21028_INVINCIBLES 5.0_Design Report

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1. Introduction

The idea of Efficycle is to encourage innovation and raise awareness among young engineers about environmentally sustainable mobility solutions.

Fossil fuels have exacted a huge toll on people and the environment, so it is time to create greener modes of transportation for a sustainable future.

The Efficycle is a dual mode tricycle (electric aided and human powered). Different topologies were evaluated based on aspects including stability, handling and maneuverability.

The vehicle's tadpole shape is ergonomic, well designed and easy to build.

We evaluated static and dynamic conditions while designing and based on various analytic results the improvements were implemented.

The mechanisms of subsystems are explored in order to support the vision of making the efficycle.

All the components are designed according to the 2021 rule book provided by the SAE-NIS for the safety of the drivers as well as vehicle.

2. Selection & Design of Sub-Systems

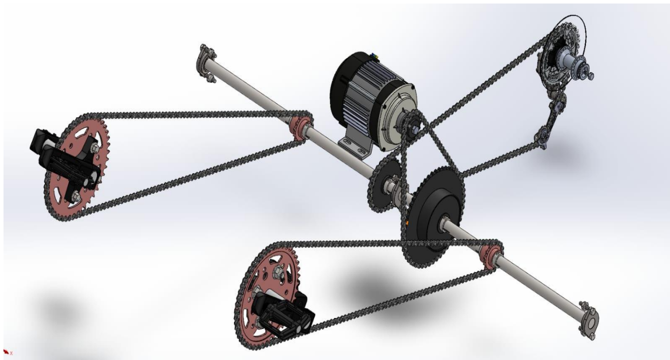
2.1. Drivetrain - *Human*

The aim of the human drive is to deliver the power produced by the driver, via pedal station to the rear wheel most efficiently. Conventionally, the mode of power transmission in efficycle is through chain drive to the rear wheel by a system of sprockets.

Here, each driver is given a discrete pedal station to give mechanical input, through a mesh of sprockets so it is transmitted to the rear. A crank arm length (7") is chosen and a crank wheel of 42T is employed in the crank set.

Sprockets of 14 teeth are used on the shaft to transmit power from the pedal station. Through a 28T sprocket the power is transmitted to the wheel hub. Here, a 9 speed (11-32T) is allocated so as to give various power inputs to the rear free hub.

This way human drive is designed to transmit maximum power to the driving wheel. In addition, shaft rotation plays a key role in the whole drive train. Mounting of shaft to the chassis plays a crucial role. Here flanged mount bearings are used to mount shaft to the chassis. These bearings grip the shaft to the chassis but also provide uniform rotational motion to the shaft.



CALCULATIONS:

Assuming input RPM given by a human = 60 rpm, the Velocity max obtained in 10s is:

Formula:

$$\frac{N_1}{N_2} = \frac{T_2}{T_1}$$

Where

N_1 = input rpm

T_1 = no. of teeth (input)

N_2 = output rpm

T_2 = number of teeth (at shaft)

$N_2 = \frac{60 \times 42}{14}$ =180 rpm at shaft

From shaft to Rear Wheel

$$\frac{N_3}{N_4} = \frac{T_4}{T_3}$$

$N_4 = \frac{180 \times 28}{11}$ = 458.18 rpm

V_{max} = R × ω

$V_{max} = \frac{2 \times \Pi \times N_4 \times R}{60}$

$= \frac{2 \times 3.14 \times 458.18 \times 0.33}{60}$

= 15.83 m/s

= 56.988 kmph

Where,

V_{max} = maximum velocity
R = radius of rear wheel = 0.33m
N_4 = speed of rear wheel
ω = angular velocity

Therefore, the maximum velocity of the vehicle in human drive = 56.988 kmph

Let,

v = final velocity

u = initial velocity

t = time taken to reach maximum velocity

a = acceleration

By the equation of motion,

v = u + at
[u = 0 , v = V_{max}]

Let us assume t = 10 sec, maximum velocity achieved in 10 sec is

a = $\frac{V_{max}}{t}$

= $\frac{15.83}{10}$

= 1.583 m/s^2

a_{max} = a − μ_rg

(μ_r = coefficient of rolling friction = 0.005)

a_{max} = 1.583 − 0.005 (9.81)

= 1.533 m/s^2

Therefore, the maximum acceleration in human drive = 1.533 m/s^2

GRADEABILITY:

On inclined planes

$$\theta = sin^{-1}(\frac{a}{g})$$

$$\theta = sin^{-1}(\frac{1.533}{9.81})$$

$$\theta = 8.99°$$

Therefore, gradeability is 8.99°.

Gear no.	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**
No. of teeth on gear	32	28	24	21	18	16	14	12	11
Gear ratio-Shaft to wheel	0.88	1	1.16	1.33	1.55	1.75	2	2.33	2.54
Gear ratio-Crank set to wheel	2.62	3	3.48	3.99	4.65	5.25	6	6.99	7.62

Gear Ratio:

The ratio of the rotational speeds of the first and final gears in a train of gears or of any two meshing gears. The gear ratio is calculated by dividing the no. of teeth in driving (input) sprocket and no. of teeth in driven (output) sprocket.

For Example:
Considering 42T sprocket in crank set, 14T sprocket on shaft then,

(i) Gear ratio = $\frac{42}{14}$ = 3

That means the rotation of one rotation crank makes, 14T sprocket and shaft to rotate 3 times.

(ii) Gear Ratio = $\frac{28}{11}$ = 2.54

So, for one rotation 28T sprocket makes, 11T sprocket rotates 2.54 times.

Considering case (i) & (ii), We get,

<u>28T</u>		<u>11T</u>
1	→	2.54
3	→	?

That gives, 3 × 2.54 = 7.62. If the crank set rotates once, 11T sprocket in the gear set rotates 7.62 times.

Gear Ratio (Crank to Rear wheel with 11T Sprocket) = 7.62

Time taken for vehicle to cover 100 m is

$$S = uT + \frac{1}{2}aT^2$$
$$S = \frac{1}{2}aT^2 \quad (u = 0)$$
$$T = \sqrt{\frac{2 \times S}{a}}$$

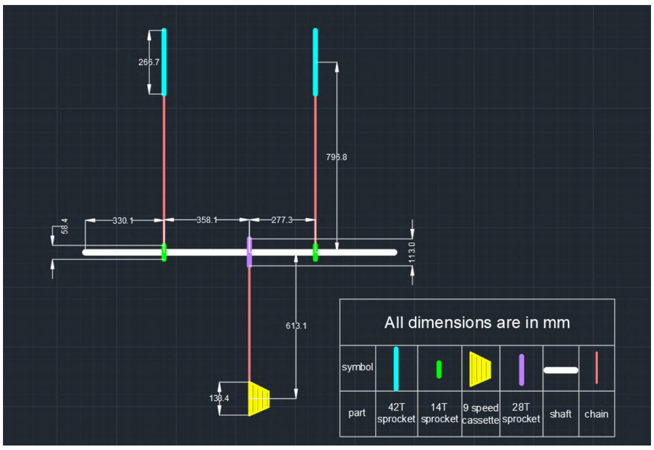
(T = time taken to travel distance S)

$$T = \sqrt{\frac{2 \times 100}{1.53}} = 11.49 \text{ s}$$

Time taken for vehicle to travel 100m is 11.49 s.

CONSIDERING ALL GEAR RATIOS:

Gear no.	9	8	7	6	5	4	3	2	1
V_{max} $(\frac{m}{s})$	6.12	6.99	8.16	9.32	10.8	12.2	13.9	14.2	15.8
a_{max} $(\frac{m}{s^2})$	1.533	1.38	1.348	1.174	1.03	0.8	0.76	0.64	0.56
Time to reach 100m (s)	11.49	12.1	12.18	11.68	14	15.9	16.3	17.7	19.1



WORKING PROCEDURE:

Initially, with discrete pedal stations, the drivers give the power to the crank set. This power is in the form of rotational energy through the chain is transmitted onto the shaft.

Here, two 14T sprockets produce torque and eventually shaft rotation starts. This finally rotates the 28T sprocket provided on the shaft.

Again, through the roller chains, this RPM of the sprockets is transferred to the cassette attached to the rear wheel hubs. As the gear set on the rear wheel hub rotates, it tends to rotate the wheel in clockwise direction. This rotation produces a translatory motion and due to friction the vehicle comes into motion.

In this way, power produced at the crank set is transferred via shaft to the rear wheel and eventually there would be a displacement observed.

2.2. DRIVE TRAIN - *ELECTRIC*

Electric drive system is a battery powered drive train (rechargeable battery). The function of electric drive system in efficycle is the conversion of electrical energy produced from the battery into mechanical energy to propel the vehicle.

COMPONENTS OF ELECTRIC DRIVE SYSTEM:

48V 600W BLDC Motor, 48V controller, 48V 35Ah Li-ion battery, Junction box, wiring harness, throttle, chain-sprocket set and kill switch.

The motor kit is provided by Vikson India. Motor part number is KTC600R.

WORKING PROCEDURE:

Motor is mounted to the chassis in between the 2 driver seats which is electrically connected to the controller. Controller is the brain of electric drive system. It is mounted beside the motor.

Electric drivetrain is powered by a 48V Li-ion battery. All the components are connected to the controller via wiring harness. Throttle is embedded on the steering handle. A 14T sprocket is placed on the motor shaft and a 52T sprocket is placed on the vehicle shaft. These 2 sprockets are connected with a chain.

The motor rotates with the power supplied by the battery which also rotates the main shaft. This shaft which is in turn connected to the rear wheel via a similar chain sprocket mechanism propels the vehicle.

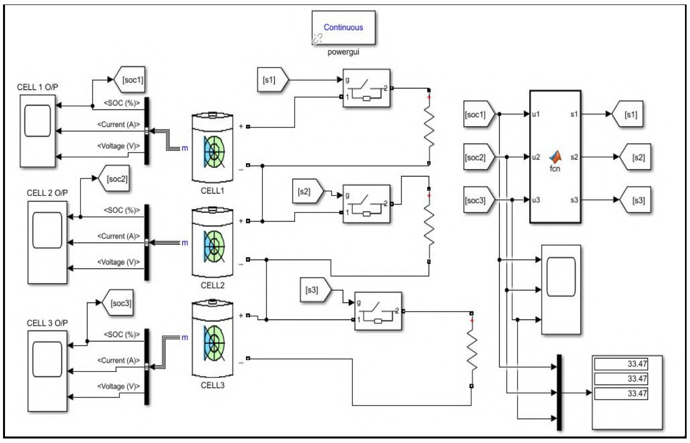
Speed of rotation of the motor is manually controlled with the throttle. The voltage input varies according to throttle input and hence the speed changes.

Kill switch is placed between the battery and the controller. It disconnects the whole electrical circuit when pressed.

CELL BALANCING:

The Li-ion battery pack in the EVs is designed using multiple cells either in parallel or series combination depending upon their Nominal Voltages and capacities. Usually, when they are overheated or overcharged, they are vulnerable to cell degradation. They are more likely to catch fire or even explode under such conditions.

Cell balancing in a battery is the process of bringing the voltages and state of charge at equilibrium when they are fully charged. Even for a pair of similar cells there are always some differences in their discharge rate, charge holding capacity, temperature characteristics even for the same model from the same manufacturer. They are also important for improving the performance and life cycle of the battery. The cell balancing circuit is usually included in the Battery Management Systems in the EV's.



Cell balancing is categorized into two types:
1) PASSIVE CELL BALANCING
2) ACTIVE CELL BALANCING

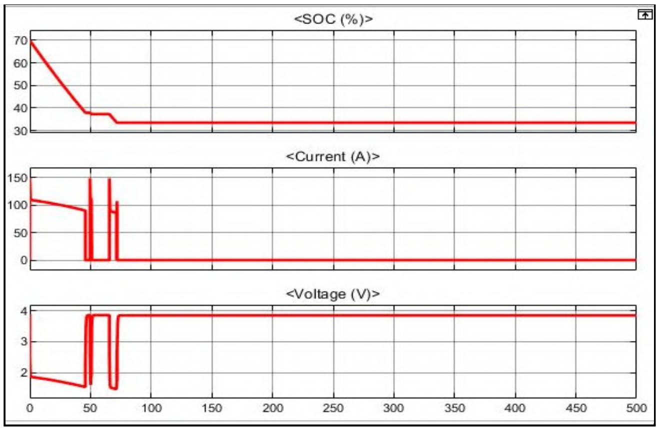
PASSIVE CELL BALANCING: This is a simple and straight forwarded method. Discharge the cells through a dissipative bypass route. This bypass can be either integrated or external to the integrated circuit (IC). Such an approach is favorable in low-cost system applications. The fact that 100% of the excess energy from a higher energy cell is dissipated as heat, makes the passive method less preferable to use during discharge because of the obvious impact on battery run time.

ACTIVE CELL BALANCING: This method uses capacitive or inductive charge shuttling to transfer the charge between the cells, This is significantly more efficient because energy is exactly transferred to the needed place. The major disadvantage of this improved efficient method is the additional cost of the high components.

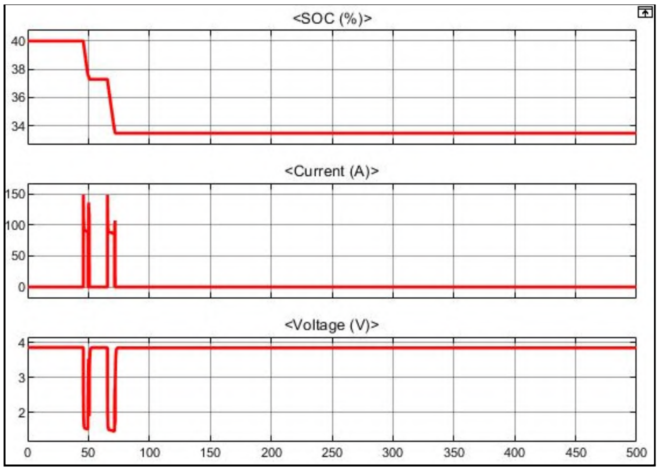
Cell Balancing also adds an element of safety to the battery. This makes it one of the emerging battery safety and extending battery life practices. Hence, increasing the usable life of battery packs including the overall safety.

Below is a simulation outcome done through Matlab for the operation of 3 similar batteries connected in series at different charge capacity stages. How simulation helps us to study the performance of these cells together.

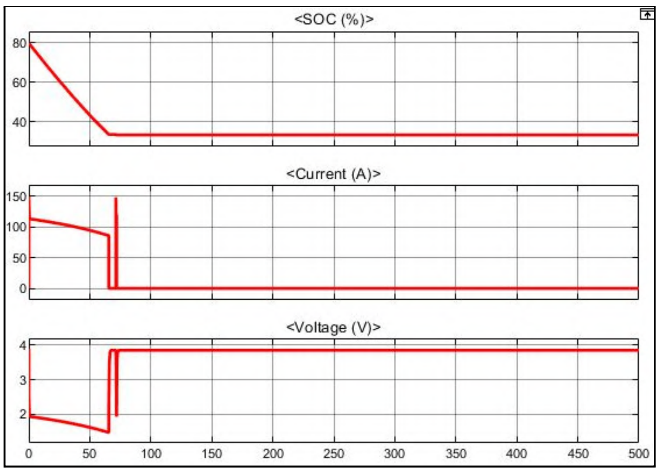
Cell 1:



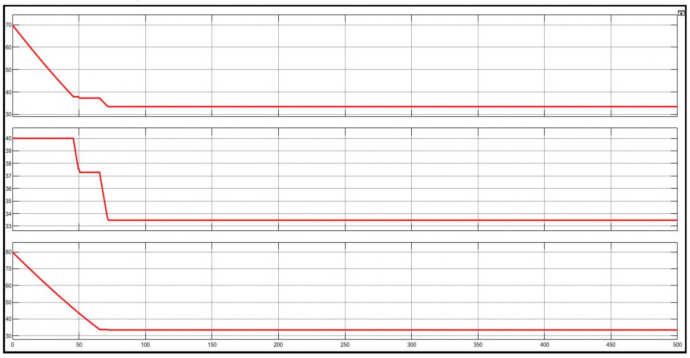
Cell 2:



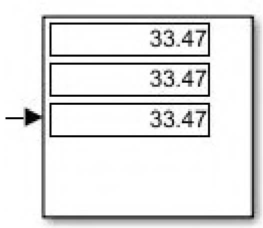
Cell 3:



Combined graphs:



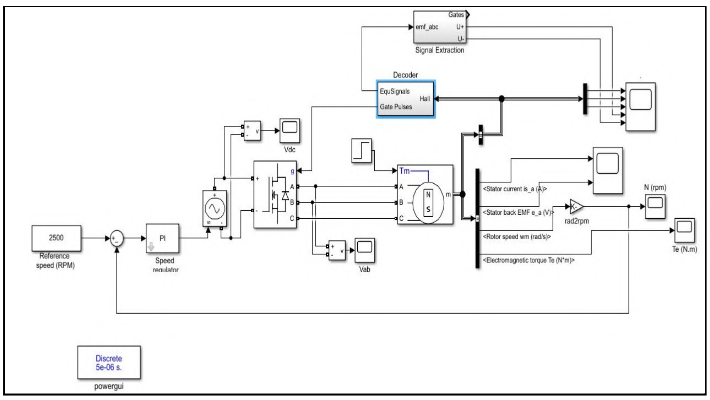
Output:



SPEED CONTROL OF BLDC MOTOR:

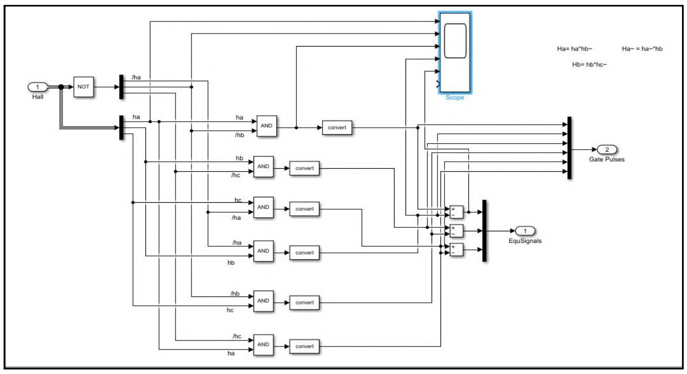
A **Brushless DC Electric** Motor (BLDC) is an electric motor powered by a direct current voltage supply and commutated electronically instead of by brushes like in conventional DC motors.

Speed control of BLDC motor is essential for making the motor work at desired rate. Speed of a brushless dc motor can be controlled by controlling the input dc voltage/current. The higher the voltage the more is the speed. Many different control algorithms have been used to provide control of BLDC motors.

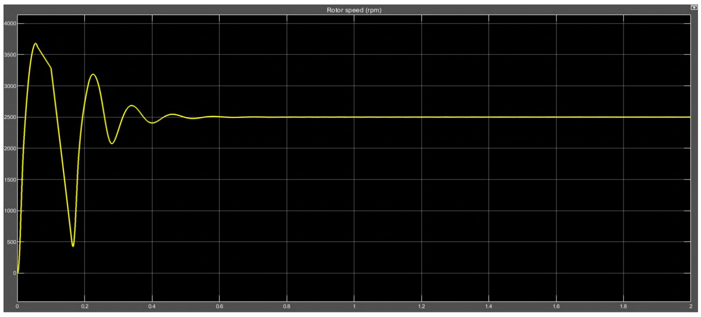


The motor voltage is controlled using a power transistor operating as a linear voltage regulator. This is not practical when driving higher power motors. High power motors must use PWM control and require a microcontroller to provide starting and control functions.

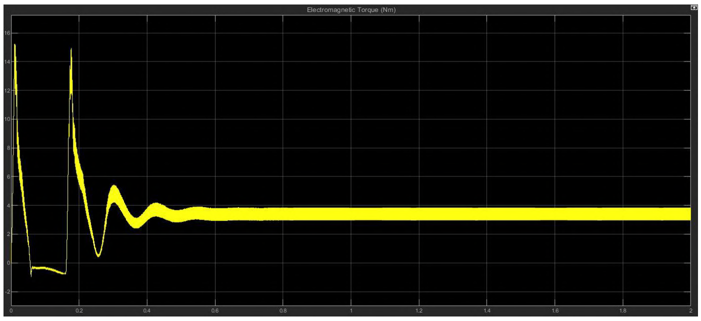
Below is a simulation outcome done through Matlab.



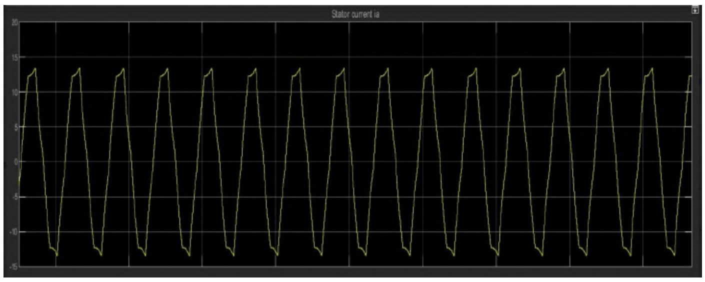
Rotor speed vs Time:



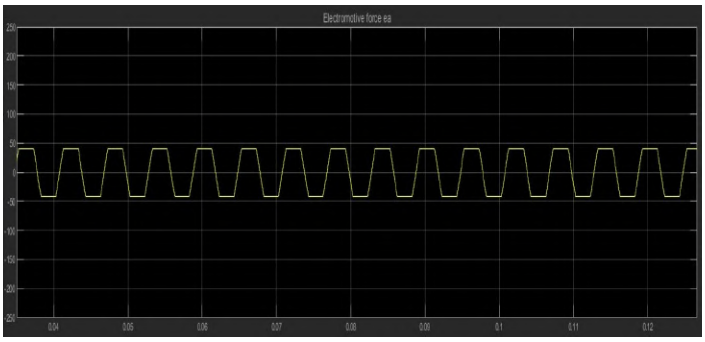
Electromotive torque vs time:



Stator current vs time:



Electromotive force ve time:



SPECIFICATIONS OF BATTERY:

SPECIFICATIONS	VALUE
Type	Li-ion battery
Dimensions	360 × 180 × 65 mm
Power Rating	48V 35Ah
Run time	2.8 hours
Charging time	8.7 hours
Weight	12 kg

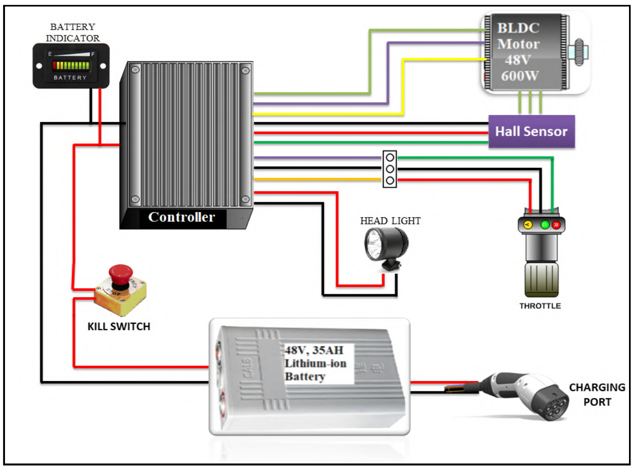
SPECIFICATIONS OF CONTROLLER:

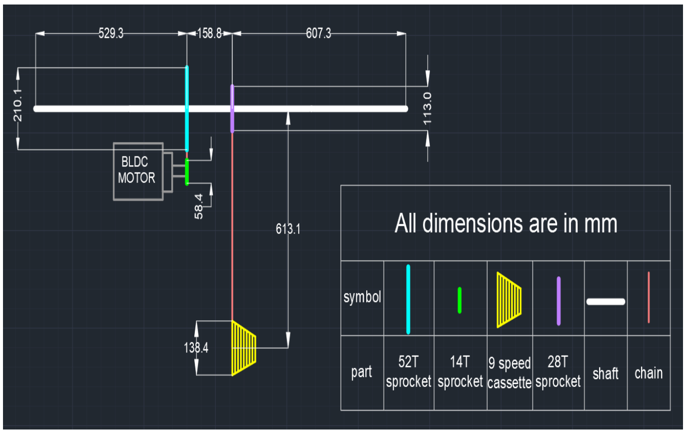
SPECIFICATIONS	VALUE
Rated voltage	48 V
Throttle voltage	0.9 – 4.3 V
Rated current	35 ± 1 A
Weight	0.55 kg

SPECIFICATIONS OF MOTOR:

SPECIFICATIONS	VALUE
Power	Rated - 600W Peak - 1400W
Rated RPM	500
Torque	Rated - 10Nm Peak - 400% of rated power for 10 seconds
Voltage	48 V
Current	Rated - 13A peak - 32A
Efficiency	80% on full load and full RPM
Protection	IP 33
Dimensions	233 × 171 × 146 mm
Shaft Diameter	Sprocket - 60mm Shaft - 12mm
Weight	4.6kg

CIRCUIT DIAGRAM:





CALCULATIONS:

1) FROM MOTOR TO SHAFT

N_1 = input rpm (rated rpm of the motor)
T_1 = number of teeth (input)
N_2 = output rpm (shaft)
T_2 = number of teeth (output)

Formula:

$$\frac{N_1}{N_2} = \frac{T_2}{T_1}$$

we know,

N_1 = 500rpm
T_1 = 14
T_2 = 52

$$N_2 = \frac{500 \times 14}{52} = 134 \text{ rpm at shaft}$$

2) FROM SHAFT TO REAR WHEEL

N_3 = input rpm (rpm of shaft)

T_3 = number of teeth (input)
N_4 = output rpm (rear wheel)
T_4 = number of teeth (output)

Formula:

$$\frac{N_3}{N_4} = \frac{T_4}{T_3}$$

we know,

N_3 = 134rpm
T_3 = 28
T_4 = 11

$$N_4 = \frac{134 \times 28}{11} = 341 \text{ rpm at rear wheel}$$

MAXIMUM VELOCITY:

V_{max} = maximum velocity
R = Radius of rear wheel

N_4 = rpm of rear wheel
ω = angular velocity

we know,

R = 0.33m
N_4 = 341rpm

$$\omega = \frac{2 \times \Pi \times N_4}{60} = 35.7\ s^{-1}$$

Formula:

$$V_{max} = \omega \times R$$

$$V_{max} = 35.7 \times 0.33 = 11.78 \text{ m/s} = 42.43 \text{ kmph}$$

Maximum velocity of vehicle in electric drive is 42.42kmph.

ACCELERATION:

Power(P) = $\frac{W}{t} = \frac{1 \times m \times V^2}{2 \times t}$
P = peak power of the motor
W = work done (change in kinetic energy)
t = time taken to reach maximum velocity
m = total mass of vehicle along with 2 drivers
V = maximum velocity

we know,

P = 1400W (peak power)
m = 118 + 75 + 75 (vehicle + $driver_1$ + $driver_2$) = 268kg
V = 11.78 m/s

Formula:

$$t = \frac{1 \times m \times V^2}{2 \times P}$$

$$t = \frac{1 \times 268 \times 11.78^2}{2 \times 1400} = 13.28 \text{ s}$$

Time taken to reach maximum velocity is 13.28 s

FROM EQUATIONS OF MOTION:

$$v = u + at$$

v = final velocity
u = initial velocity
a = acceleration
t = time taken to reach maximum velocity

we know,

u = 0
v = 11.78 m/s
t = 13.28 s

$$a = \frac{v - u}{t} = \frac{11.78}{13.28} = 0.89\ m/s^2$$

Now considering frictional losses

$$a_{max} = a - \mu_r g$$

μ_r = coefficient of rolling friction = 0.005

a_{max} = 0.89 - 0.005 × 9.81 = 0.84 m/s^2

Maximum acceleration when 11T gear is used at the rear wheel is 0.84 m/s^2.

GRADEABILITY:

On inclined planes

$$\theta = sin^{-1}(\frac{a}{g})$$
$$\theta = sin^{-1}(\frac{0.84}{9.81})$$
$$\theta = 4.91°$$

Therefore, gradeability is 4.91°.

TIME TAKEN FOR VEHICLE TO TRAVEL 100M:

From equations of motion

$$S = uT + \frac{1}{2}aT^2$$
$$S = \frac{1}{2}aT^2 \quad (u = 0)$$
$$T = \sqrt{\frac{2 \times S}{a}}$$
$$T = \sqrt{\frac{2 \times 100}{0.84}} = 15.43 \text{ s}$$

Time taken for vehicle to travel 100m is 15.43 s.

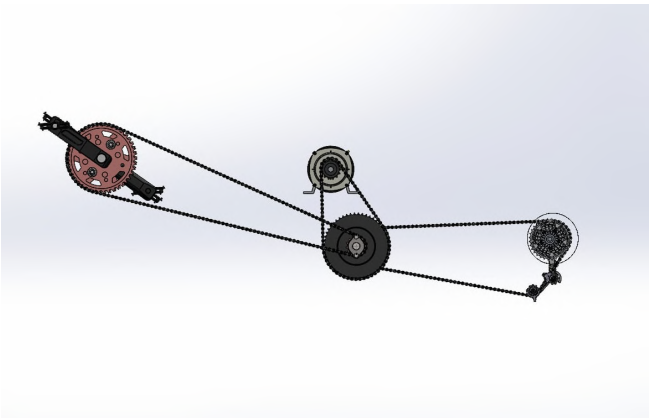
CONSIDERING ALL GEAR RATIOS:

gear no.	1	2	3	4	5	6	7	8	9
teeth no.	32	28	24	21	18	16	14	12	11
a_{max} $(\frac{m}{s^2})$	2.53	2.21	1.88	1.64	1.35	1.24	1.08	0.92	0.84
V_{max} $(\frac{m}{s})$	4.05	4.63	5.4	6.17	7.42	8.1	9.26	10.8	11.78
max Gradeability (°)	14.94	13	11.08	9.64	7.96	7.26	6.31	5.37	4.91

2.3 Drive train - *hybrid*

Efficycle is a human-electric hybrid vehicle designed to carry two passengers. It can be driven in electric-only mode or pedal-only mode or electric + pedal mode.

Hybrid drivetrain is a system that makes use of human power and electric power in a vehicle whose drivetrain consists of a person, an electric motor and an electricity storage device (battery).



CALCULATIONS:

Human drivetrain:

V_{max} is 57.006 kmph or 15.835 m/s

a_{max} is 1.533 m/s^2

Gradeability is 8.99°

Electric drivetrain:

V_{max} is 42.43 kmph or 11.78 m/s

a_{max} is 2.53 m/s^2

Gradeability is 14.94°

Hybrid drivetrain:

1. **Maximum average velocity**

$$v = \frac{57.006 + 42.43}{2} = 49.718 \text{ kmph}$$

2. **Maximum average acceleration**

$$a = \frac{1.533 + 2.53}{2} = 2.03\ m/s^2$$

3. **Maximum average gradeability**

$$\frac{8.99 + 14.94}{2} = 11.965°$$

2.4. Steering

The basic aim of steering is to ensure that the wheels are pointing in the desired directions. The purpose of steering is to allow the driver to guide/control the direction of the vehicle by turning the front wheels.

CONSIDERATIONS WHILE DESIGNING THE STEERING ARE AS FOLLOWS:

- Less steering effort
- Cost efficient and less weight
- Minimum turning radius
- Easy and comfortable to handle
- More efficient

We have chosen Linkage Mechanism (Trapezoidal) for our steering subsystem.

WORKING PROCEDURE:

The vehicle is steered in the desired path by turning the steering handle in that direction.

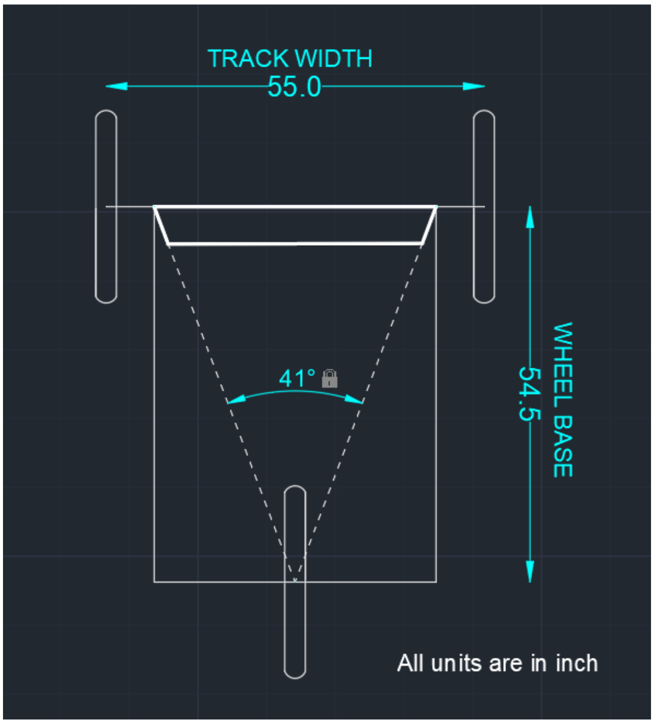
As the driver turns the handle bar, the movement of the steering column causes movement in the tie rod connected to the right wheel. This movement indirectly imparts motion to the left wheel through the linkage mechanism.

This way the wheels turn in the desired direction. Direct and simple steering is employed because there is no urge requirement of rack and pinion steering system.

CONDITION FOR PERFECT STEERING :

- All wheels must turn about the same instantaneous centre.
- The above condition is satisfied when the inner wheels make a larger turning angle than the angle subtended by the axis of the outer wheel.

STEERING GEOMETRY:



Ackerman Geometry is chosen to ensure that the inner wheel turns a higher angle than the outer wheel in order to allow the vehicle to rotate around the midpoint of the rear wheel axis, so as to prevent the slipping of wheels at low speeds.

$$\cot\alpha - \cot\beta = \frac{b}{L}$$

[α = outer wheel turning angle]

[β = inner wheel turning angle]

[b = track width]

[L = wheel base]

For a correct steering geometry, the above conditions must be met and applied. So, through calculations the inner and outer turning radius practically are

α = 28.6883°, β = 50.8465°.

ADVANTAGES OF ACKERMANN GEOMETRY:

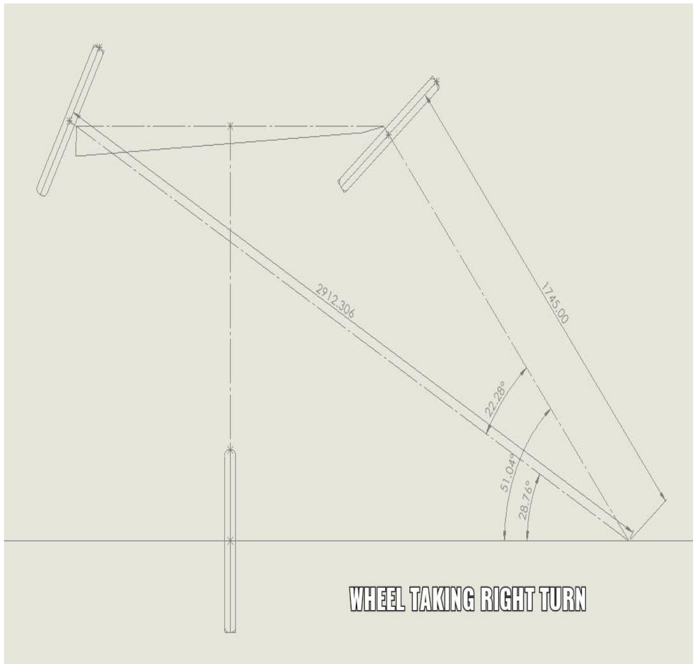
1) Experience of smooth steering feels as sliding contact is changed to rolling contact.

2) It gives easier and more compact control over vehicle.

3) Avoids front tire slippage and helps in achieving pure rolling.

4) Turning pairs employed reduces the friction and wear in the mechanism.

5) Easier maintenance.



CALCULATION OF TURNING RADIUS:

Outer wheel turning radius =

$$\frac{wheelbase}{\sin\alpha} - \frac{(track\ width - pivot\ distance)}{2}$$

$$= \frac{54.5}{0.48} - (3.0825)$$

= 2.9 m

Outer wheel turning radius = 2.9 m

Inner radius turning radius =

$$\frac{wheelbase}{\sin\beta} - \frac{(track\ width - pivot\ distance)}{2}$$

$$= \frac{54.5}{0.775} - (3.0825)$$

= 1.7 m

Inner radius turning radius = 1.7 m

CALCULATION OF CRITICAL VELOCITY:

According to Newton's 1st law of motion, a body moving in a straight line will continue in a straight line unless acted on by an external force.

A body moving on a circular path with constant speed will have a changing velocity due to body's changing direction.

This velocity change with time, called Centripetal Acceleration, has a radial direction towards the centre of the circular movement and is given by

a = $\frac{V^2}{R}$

[v = velocity ($\frac{m}{s}$)]

[R = Radius of turn (m)]

Since Newton's 2nd law tells us that a force has to act on the body to produce acceleration.

Therefore,

F = ma [m = mass of the vehicle]

[a = centripetal acceleration ($\frac{m}{s^2}$)]

[F = centripetal force]

F = $\frac{mV^2}{R}$

Now the lateral force on a vehicle while moving in a circular motion on a pavement surface is powered by the frictional force between the tyres and the roadway as follows:

F = μN [N = normal reaction]

Therefore, F = μmg

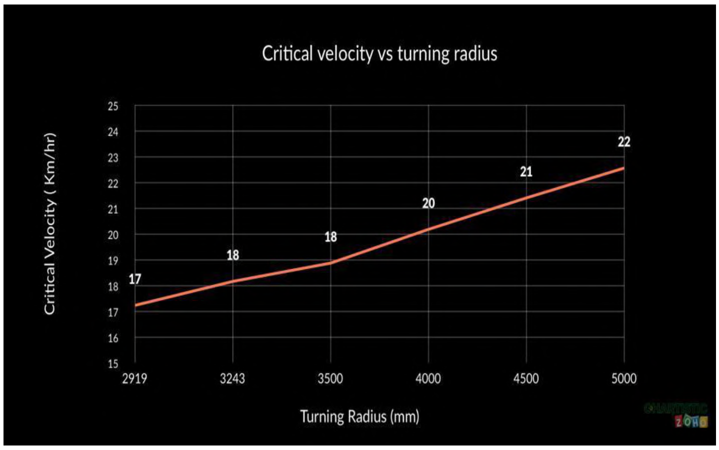
Condition for a vehicle not to slip (centripetal force > frictional force). Therefore, equating both forces will give critical velocity, which should not be exceeded.

$\frac{mV^2}{R}$ = μmg

[μ = 0.8]

[g = gravitational constant = 9.81$\frac{m}{s^2}$]

[R = outer turning radius = 2.919 m]



Therefore, we get critical velocity

$V_{critical} = \sqrt{\mu g R}$

$= \sqrt{0.8 \times 9.81 \times 2.919} = 4.78\frac{m}{s}$

This is 17.23 Kmph, a desirable critical velocity of a vehicle.

CALCULATIONS OF STEERING RATIO:

Steering ratio in linkage mechanism is determined by the ratio of the relative lengths of the pitman arm and steering arm.

Pitman arm - 4.495 in
Steering arm - 4.854 in

Therefore, we get steering ratio = $\frac{4.495}{4.854}$ = 0.926:1

CALCULATION OF ACKERMANN PERCENTAGE:

Ackermann value = $tan^{-1}(\frac{L}{\frac{L}{tan\,\alpha} - b})$

Substituting, [wheel base = 54.5 in]

[Outer wheel turning angle (α) = 28.6883°]

[Track width (b) = 55 in]

Therefore, we get Ackermann value = 50.7

Ackermann Percentage = $\frac{\alpha}{Ackermann\ value}$

After calculation, Ackermann Percentage = 56.58%

CALCULATIONS OF STEERING FORCES (NO PAYLOAD):

Total mass = 120 kg

Mass of the vehicle with drivers =195 kg

Mass on the front =0.534 × 195 = 104.13 kg

Mass on the Rear = 0.436 × 195 = 85.02 kg

Corner mass (front) = $\frac{M_{front}}{2}$ = 52.065 kg

Corner mass (rear) = M_{rear} = 85.02 kg

The basic concept is, torque required to turn the wheel should be more than resisting torque by friction.

Force of friction (on one wheel) = μ × g × corner mass (front)

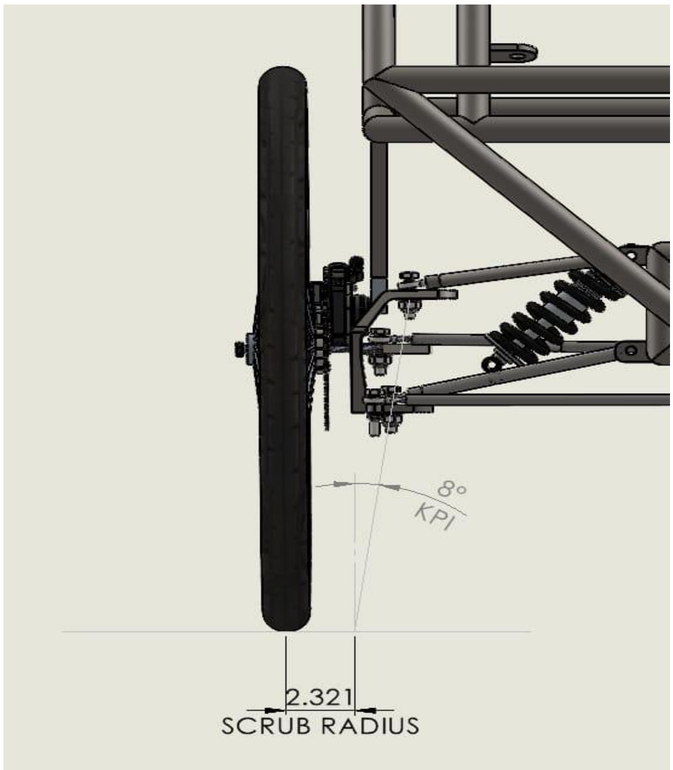
[μ = 0.8] , [g = 9.81m/s^2]
=0.8 × 9.81 × 52.065 kg
=408.60612 N

We get, Force of Friction = 408.60612 N. As we know that steer happens about the kingpin axis of the corner.

Input Torque from the ground (on one wheel) =
Force of Friction × perpendicular distance from contact patch to line joining kingpin centres

=408.60612 × 2.32 × 0.0254 Nm
=24.078 Nm

The scrub radius is the distance in front views between the kingpin axis and the centre of the contact path of the wheel.



Torque due to Friction Force

= 408.60612 × 2.32 × 0.0254

= 24.078 Nm

The torque will be equal to the lateral push from the tie rods.

Torque due to lateral push from (back tie rod) =
F_t × perpendicular distance between the outer tie rod point end to line joining the kingpin centres.

$F_t = \frac{24.078}{2.9358 \times 0.0254}$ N

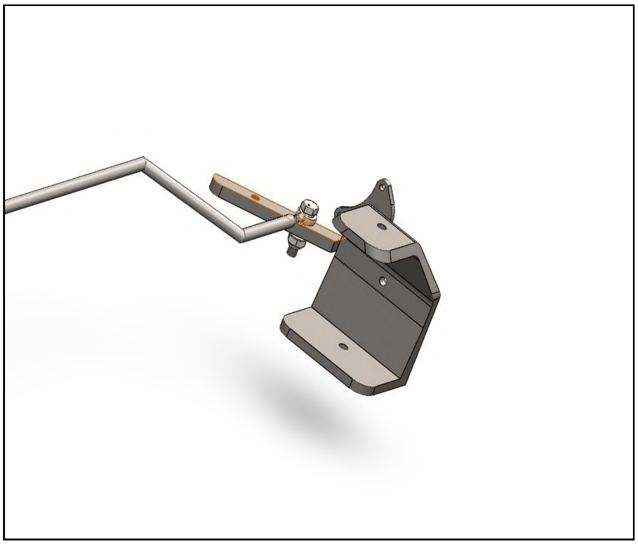
= 322.8941 N

Since, both Torques have to be equal

Torque due to lateral push (front tie rod) =
F_t × perpendicular distance between the outer tie rod point end to line joining kingpin centres

$F_t = \frac{24.078}{4.5065 \times 0.0254}$ N

= 210.352 N



As the tie rods pushing the tire from one side and pulling it from another side

Total force = 322.8941 + 210.352

= 533.2461 N

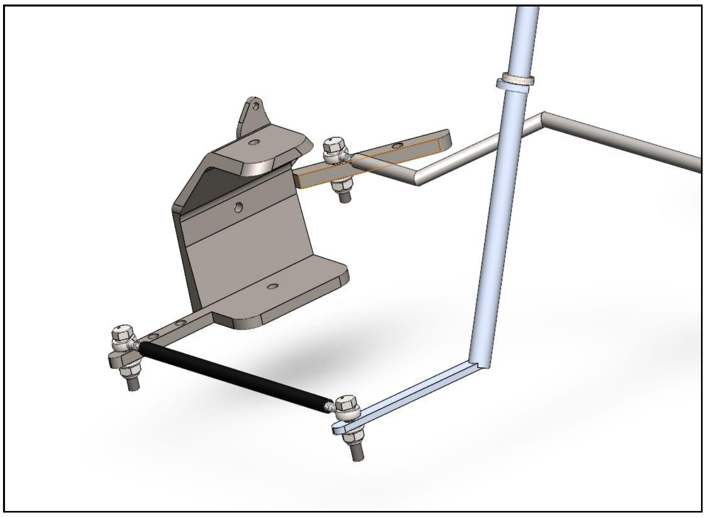
Length of the steering column plate = 1.9645 in

= 0.04989 m

Torque on steering column =
Total force $\times$ perpendicular distance of the link between steering column and tie rod from the line joining kingpin centres

= 533.2461 $\times$ 1.9645 $\times$ 0.0254 Nm

=26.608 Nm



Torque on steering handle = torque on steering column = 26.608 Nm

Length of steering handle (half length) = 9 in
= 0.2205 m

Force applied by the driver to steer the wheels = Torque on steering handle/length of steering handle

$= \frac{26.608}{0.2205}$ = 120 N

After we get, Force applied by the driver to steer the wheels = 120 N

STEERING PARAMETER:

Parameter	Value
Track Width	55 in
Wheel Base	54.5 in
Turning Radius	2.919 m
Critical Velocity	17.208 kmph
Steering Ratio	0.926:1
Pitman arm length	4.495 in
Steering arm length	4.854 in
Ackermann Percentage	56.58%

2.5. Suspension

Suspension is a system of tires, springs, shock absorbers and linkages that connects a vehicle to its wheel and allows relative motion between them.

Suspension provides comfort to the driver, making sure the car stays in contact with the ground and that the driver has control over the tires. It helps while braking, cornering the vehicle.

WORKING PROCEDURE:

A suspension works on the principle of force dissipation which involves converting force into heat thus removing the impact that force would have made. It uses springs and dampers to achieve this. Springs are used to store the energy while a damper will convert into heat.

Keeping in view the above mentioned aspects and to account for the dynamic weight distribution of the vehicle, we have opted to install both rear & front suspension.

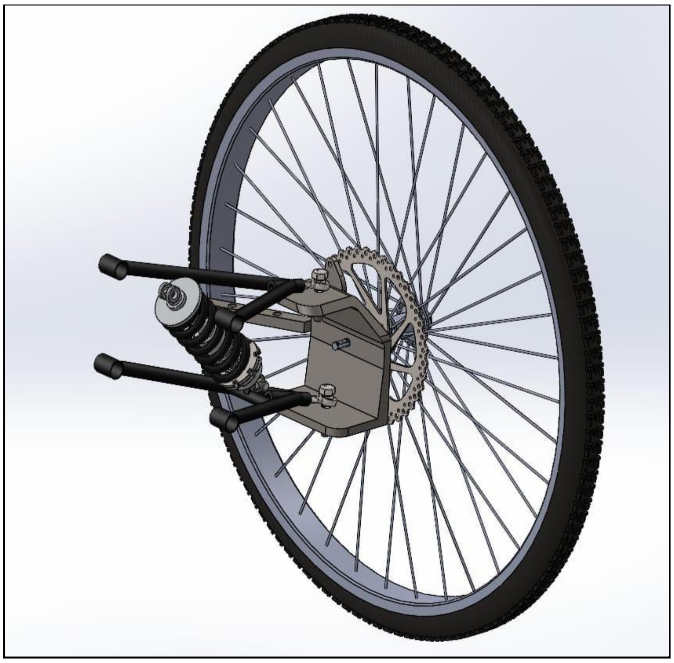
FRONT SUSPENSION

We have the following option for front suspension.

1. Suspension forks (double forks)

2. Lefty (single forks)

3. Double wishbone



a)Suspension forks: Suspension forks will reduce acceleration. If the forks bob up and down as you pedal this steals pedalling efforts, so reduces maximum speed. Therefore this option is rejected.

b)Lefty: It has a single fork. The advantages are it consumes less space and is higher. But this option was eliminated too due to unavailability of product in our local market and huge price if offered from outside of state.

c)Double wishbone: It plays a key role at rider comfort. This allows the control over vehicle with stability and consistency. It also gives much more freedom with placement of the damper.

Therefore keeping in the account of availability, maintenance and performance we have decided to use double wishbone as our front suspension system.

SPECIFICATIONS:

Spring rate of a spring	750 lbs/in = 131.345 N/mm
Wire diameter	10 mm
spring diameter	50.5 mm
Number of coils	6
Maximum spring travel	36 mm
Number of springs used	2

ANALYSIS OF FRONT SUSPENSION:

Assumption made:

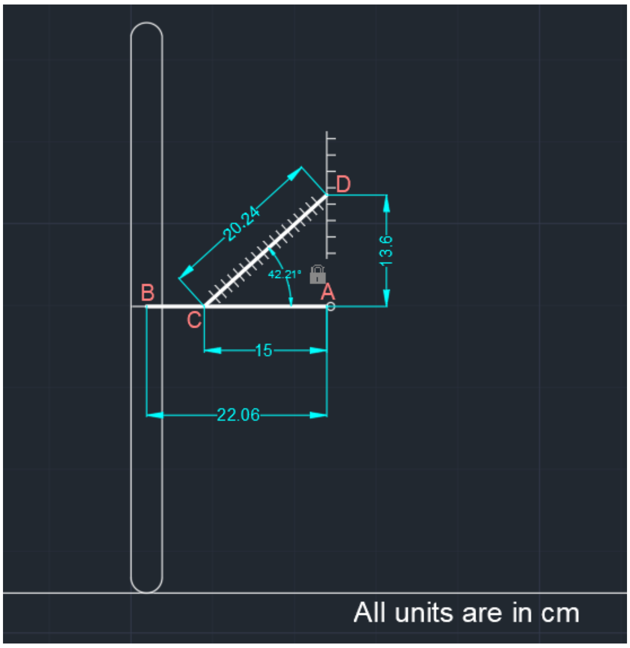
The rods used are assumed to be straight.

1) 46.6% of weight (280kgs (vehicle weight 120 kgs)+riders weight(80+80 kgs)) is acting on the front axle.

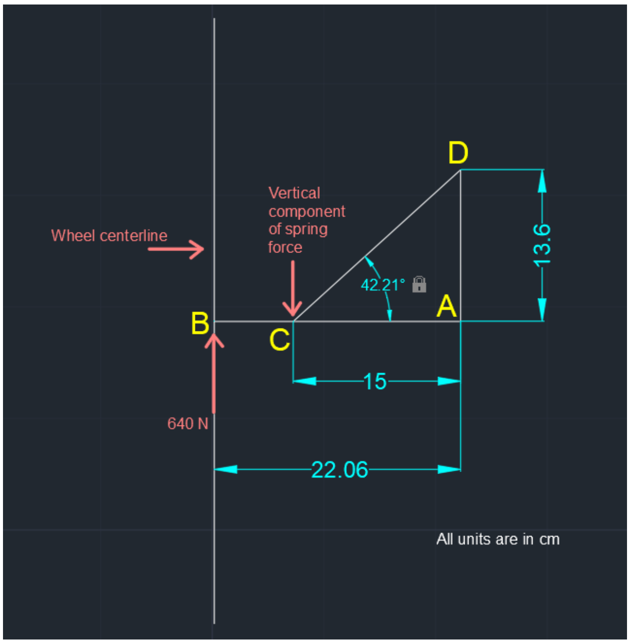
2) The springs are replaced with rigid rods, for case though it doesn't make any significant difference.

3) One end of the spring is attached to the chassis member which is assumed to be a rigid motionless wall.

SCHEMATIC DIAGRAM OF THE SUSPENSION SYSTEM:



FREE BODY DIAGRAM:



CALCULATIONS:

As we calculated 46.6% of the weight is on the front wheels.

So, on each wheel = 23.3%

Damping calculations:

Let I_r = 0.66 (Instantaneous ratio)

k_t = 75 (Tyre rate)

f_r = 0.1Hz (Ride rate frequency)

$k_r = 4\pi^2 f_r^2 m$ (Ride rate)

$= 4 \times 10 \times (0.1)^2 \times 65.24$

$= 2609.6 \times 0.01$

$k_r = 26.096$

$$K_r = \frac{k_w \times k_t}{k_w + k_t}$$

$$26.096 = \frac{k_w \times k_t}{k_w + k_t}$$

$$\left(\frac{1}{k_w} = \frac{1}{k_r} - \frac{1}{k_t}\right)$$

$$k_w = \frac{k_r \times k_t}{k_t - k_r}$$

(k_w = wheel rate)

$k_w = 40.21$

where, $k_w = k_s\, I_r^2 \cos\theta$

(k_s = spring rate)

(θ = Angle made by the spring with chassis)

where θ = 47 degrees

$40.21 = k_s \times (0.66)^2 \times \cos(47)$

k_s = 135.351 N/mm

The spring we found is near to this value of k is 131.345 N/mm.

SPRING CALCULATIONS:

As we calculated 46.6% of the weight is on the front wheels.

So, on the each wheel 23.3%

23.3% of 280 kg = 65.24 kg

Therefore, the reaction force from the ground on the wheel = 640 N = R_t

The force on the spring is F_s

Now taking moments about point A

$R_t \times AB - F_{sv} \times AC = 0$

(F_{sv} = vertical component of spring)

$640 \times 220.61 = F_{SV} \times 150$

($F_{sv} = Fs \times \sin\theta$)

$640 \times 220.61 = F_s \sin\theta \times 150$

($\theta = 42.21°$)

(θ is angle of spring made with arms)

(movement due to horizontal component = 0)

$640 \times 220.61 = F_s \sin(42.21) \times 150$

$F_s = 1401.01$ N

We know that any spring satisfies the equation

$F_s = KX$

[K = spring constant]

[X = compression in the spring]

[F_s = spring constant]

As we are using spring constant = 750 lbs/inch

= 131.345 N/mm

From the calculation above spring force = 1401.01 N

The compression in the spring = $\frac{F}{K}$

$X = \frac{1401.01}{131.345} = 10.667$ mm

Here X is compression of the spring when an axial force of 1401.01 N or 65.24 kg of a reaction is applied.

We know that maximum possible compression in the spring,

$X_{max} = 36$ mm

$F_{max} = K \times X_{max}$

$F_{max} = 131.345 \times 36 = 4.728$ KN

The relation between axial force (F) on the spring and reaction force on the wheel is linear

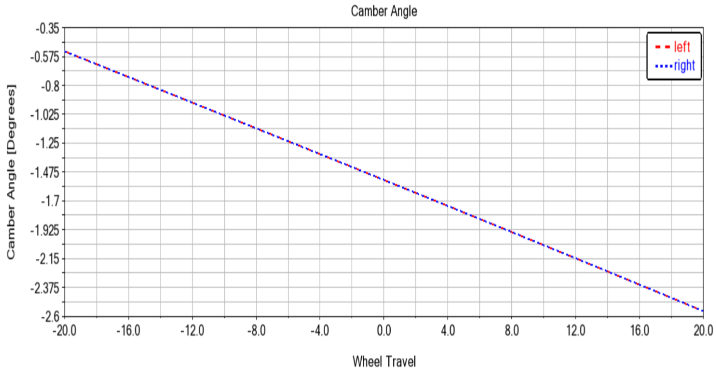
Therefore,

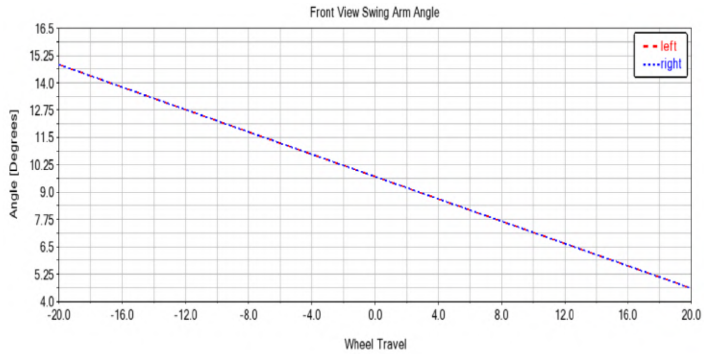
$$\frac{F}{F_{max}} = \frac{R_t}{R_{t\,max}}$$

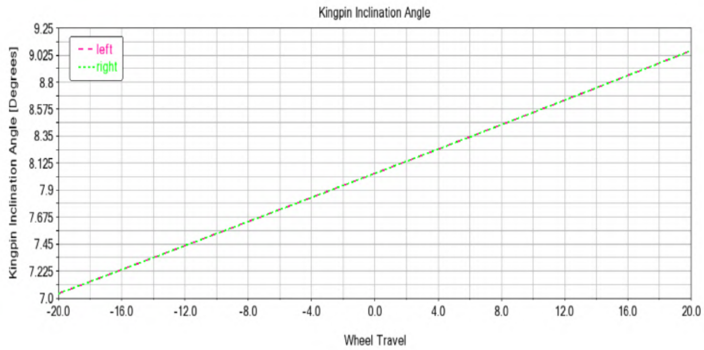
$$\frac{1401.01}{4728} = \frac{640}{R_{t\,max}}$$

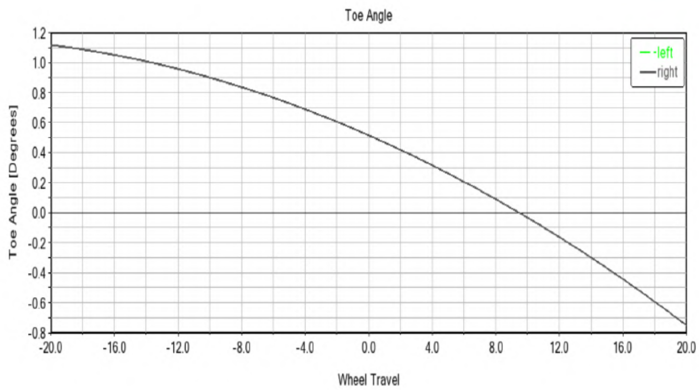
$R_{tmax} = 2159.81$ N

Therefore, a maximum of 2159.81 N of a reaction force is the safety limit.









ANALYSIS OF A SUSPENSION SYSTEM IF THE VEHICLE IS THROWN FROM CERTAIN HEIGHT

Assumption made:

When the vehicle is thrown from the maximum height of 40m 46.6% of the reaction force acts on the front wheels. So on each wheel 23.3% of reaction force acts.

CALCULATION:

F = m $\left(\frac{\Delta v}{\Delta t}\right)$

$\Delta V = V_2 - V_1 = \sqrt[2]{2gHm} - 0 = \sqrt[2]{2gHm}$

Δt = reaction time = 0.5 sec (assumption)

M = 23.3% of 120 = 27.96 Kg

H_m = 40 m

$F = 27.96 \frac{\sqrt[2]{2 \times 9.81 \times 40}}{0.5}$

= 1566.55 N

As we know max reaction force = 1566.55 N

Therefore the front suspension springs are safe if the vehicle is thrown from a height of 40 m.

WEIGHT DISTRIBUTION AND FORCES ACTING ON EACH WHEEL DURING BRAKING

The vehicle is under negative acceleration during braking. Inertial reaction force denoted by (Gxa/g) acting at centre of gravity opposite to the direction of the acceleration (G = mg).

During the vehicle's deceleration load is transformed from rear to the front wheel. By considering the equilibrium of the moments about the front and rear tyre ground contact points the normal loads on the front and rear wheels are

$G_{Fadyn} = G_{lr}$ + m.a.H/L

$G_{Radyn} = G_{lr}$ - m.a.H/L

Where,

L_r = Distance from rear axle to C.G

L_F = Distance from front axle to C.G

H = Height of C.G

a = acceleration of vehicle

As we know that,

L_r = 751.4 mm = 29.58 inch

L_f = 632.34 mm = 24.89 inch

Wheel base L = 54.5 inch

G = mg

a_{max} = 1.53 m/s$^{2.}$

$G_{Fadyn} = \frac{(2746.8 \times 29.58) + (280 \times 1.53 \times 32.91)}{54.5}$

=1749.52 N

$G_{Radyn} = \frac{(2746.8 \times 24.89) - (280 \times 1.53 \times 32.91)}{54.5}$

=1000.71 N

Weight % on the front wheels during braking

$W_{Fadyn} = \frac{G_{Fadyn}}{G_{Fadyn} + G_{Radyn}} \times 100$

$= \frac{1749.52}{1749.52 + 1000.71} \times 100$

W_{Fadyn} = 63.6%

On each front wheel 31.8% of weight is applied.

So, reaction force on the front wheel during braking is 0.318 × 280 × 9.81 = 873.48 N

Compression on the spring during braking is,

$R_t \times AB = F_{SX} \times AC$

873.48 × 220.61 = F_S Sin(42.21) × 150

F_S = 1912.11 N

Since, F_S = KX

1912.11 = 131.345 × X

X = 14.55mm

Compression in the spring during maximum braking is 14.55 mm.

REAR SUSPENSION

We chose coil over suspension system since it is easy to construct. It is made up of triangular swing arm components coupled to the chassis by the spring positioned on the joint.

The springs utilized have a consistent weight and are readily available in the market, as well as lighter and less expensive.

SPECIFICATIONS:

Spring rate of a spring	450 lbs/in = 78.8 N/mm
Wire diameter	12.4 mm
spring diameter	80 mm
Number of coils	6
Maximum spring travel	36 mm
Number of springs used	2

ANALYSIS OF THE REAR SUSPENSION

Assumption made:

a) The rods used are assumed to be straight lines.

b) 53.4% of the weight (vehicle weight (120 kgs) + Riders weight (80+80kg) = 280 kgs) is acting on the rear wheel.

c) The spring is replaced with rigid rods, for ease, though it doesn't make any significant difference.

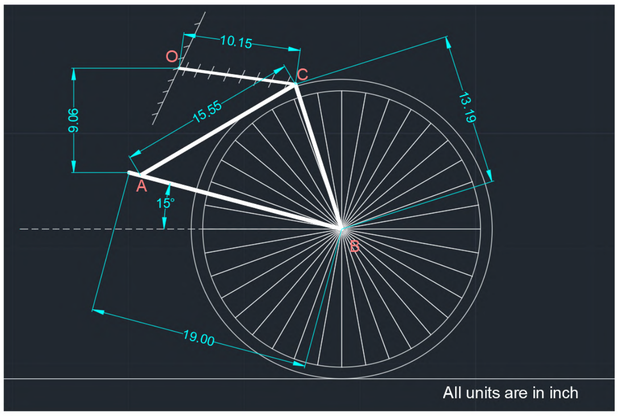
d) One of the springs is attached to the chassis member which is assumed to be a rigid, motionless wall.

CALCULATIONS:

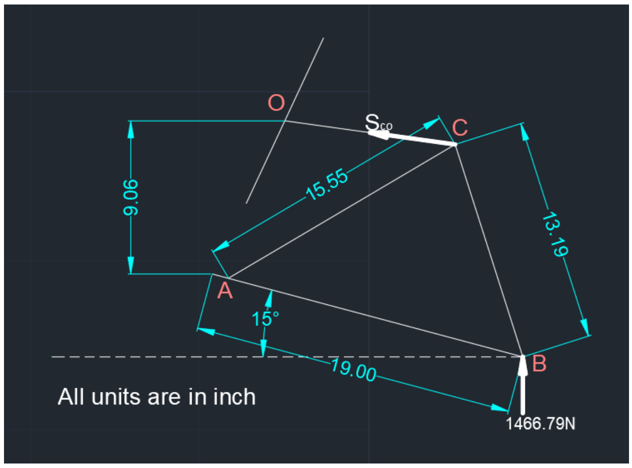
1) A is the pivot (hinge), and can rotate the X-Y plane defined.

2) The vehicle is in a static condition with the drivers.

SCHEMATIC DIAGRAM OF THE SUSPENSION SYSTEM:



FREE BODY DIAGRAM:



AB = 48.26 mm
BC = 30.96 mm
AC = 39.49 mm
Angle CAB = 45.58 degrees
Angle OAC = β = 34.66°
OA = AC × cos(β)
= 39.49 × cos(34.66)
= 32.48 cm

As we calculated that 53.4% of the weight on the rear wheel, 53.4% of the 280 kgs = 149.52 kgs.

Therefore the reaction force by the ground on the wheel is,

$R_t = 149.52 \times 98.1 = 1466.79$

The Axial Force on the members OC be S_{co}

Now taking moments about point A

$R_t \times AB + S_{co} \times 32.48 = 0$

$S_{CO} \times 32.48 = -1466.79 \times 48.26$

$S_{CO} = -\ 2179.41$ N

As we are using 2 springs of spring constant 78.8N/mm, the effective spring constant is 157.6 N/mm.

K_{eff} = 157.6 N/mm

We know that spring satisfies the below equation.
F=KX

where,

[F is axial force]

[K is effective spring constant]

[X is compression in the spring]

From the above calculations,
F = -Sco = 2179.41N
(As spring restoring force is opposite to the axial force)

$$X = \frac{F}{K_{eff}} = \frac{2179.41}{157.6} = 13.84 \text{ mm}$$

Here X is the compression in the spring when an axial force = 2179.41N of reaction force applied.

We know that Maximum possible compression in the spring = 36 mm

$$F_{max} = K_{eff} \times X_{max}$$

$$F_{max} = 157.6 \times 36 = 5673.6 \text{ N}$$

The relation between axial force F on the spring and the reaction force on the wheel is linear.

Therefore,

$$\frac{F}{F_{max}} = \frac{R_t}{R_{t\,max}}$$

$$\frac{2179.41}{5673.6} = \frac{1466.79}{R_{t\,max}}$$

$R_{t\,max}$ = 3818.455 N
Therefore, a maximum of 3818.455 N of the reaction force is the safety limit.

ANALYSIS OF THE SUSPENSION SYSTEM IF THE VEHICLE IS THROWN FROM A CERTAIN HEIGHT

Assumption made:

When a vehicle is thrown from a maximum height of 40m. 53.45% of the reaction force acts on the rear wheel.

CALCULATIONS:

$$F = m\left(\frac{\Delta v}{\Delta t}\right)$$

$$\Delta V = V_2 - V_1 = \sqrt[2]{2gHm} - 0 = \sqrt[2]{2gHm}$$

Δt = reaction time = 0.5 sec (assumption)
h_{max} = 40 m
m = 53.4% of 120 = 64.08 kg

$$F = 64.08\ \frac{\sqrt[2]{2 \times 9.81 \times 40}}{0.5}$$

F= 3590.31N

As we know that maximum reaction force is 3818.455N. Therefore the rear spring is under safety limit.

2.6. Wheels & Tyres

Wheels provide traction in all kinds of surfaces without slipping and even road shocks are first absorbed by the tyres. And then it is transmitted to suspension. They must be as light as possible to help keep unsprung weight to be minimum. Hence we choose general available tyres from the market.

Front:
Wheel rim diameter= 20 inch
Wheel diameter= 22 inch

Rear:
Wheel rim diameter= 24 inch
Wheel diameter= 26 inch

A large rear wheel is chosen as it is the powering wheel, hence more distance is covered in a single rotation, and the front wheel has to be small, as steering action on the front wheel must be easy to steer.

2.7. Braking System

Brakes are an essential and important part of the vehicle. Mechanical disc brakes are being used on all three wheels of the vehicle. Mechanical disc brakes are preferred over hydraulic disc brakes because mechanical disc brakes provide efficient braking at a fraction of cost of hydraulic disc brakes.

Dual cable levers are used to activate the front two brakes. Discs of diameter 160mm are used.

Brake specification: independent disc with lever actuation.

Disc brakes were preferred over rim brakes because

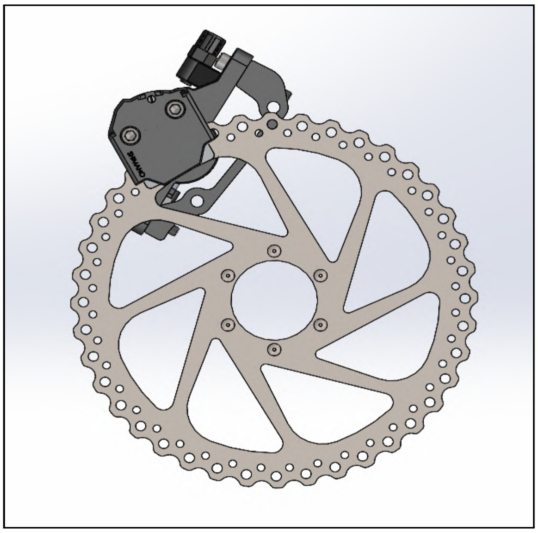
(i) Disc brakes are more efficient.

(ii) Rim brakes failure frequency is much higher than disc brakes.

(iii) Disc brakes can be trusted in situations where sudden application of brakes are required.

(iv) Disc brakes do not alter the size and shape of the rim in any given situation.

(v) Rim brakes require frequent changing of brake pads.



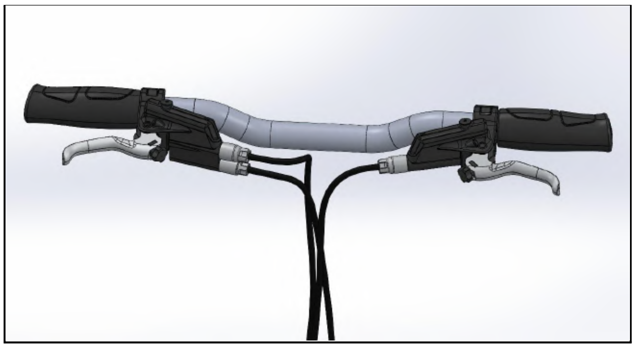
WORKING PROCEDURE:

In a Disc brake mechanism, the disc is attached to the wheel and the disc moves along with the wheel.

A calliper containing 2 brake pads is mounted over the disc in such a way that there is some gap between the pads and the disc.

The calliper is connected to the brake levers via cables. When the brake levers are pulled, the work is transferred to the calliper via brake cables.

The brake pads inside the calliper get activated and squeeze against the moving disc. This causes friction between the disc and the brake pads thus stopping the vehicle.



CALCULATIONS:

Speed = 25kmph = 6.94 m/s

Vehicle mass (m) = 270 kg

Rotor radius (r) = 80 mm

Front wheel radius (R_f) = 9" = 0.225 m

Rear wheel radius (R_r) = 12" = 0.301 m

Area of brake pad (A) = 500 mm^2

Coefficient of friction between disc and brake pad (μ) = 0.5

Pressure (p) = 5.0 MPa

No. Of surfaces of contact (n) = 2

Force exerted by brake pads on rotor will be (clamping force)
$F_c = P \times A = (5 \times 10^6) \times (500 \times 10^{-6}) = 2500$ N

The braking torque at disc will be
$\tau_d = n\mu F_c R = 2 \times 0.5 \times 2500 \times 80 \times 10^{-3} = 200$ Nm

Since disc is fixed to wheel hub,
the torque on disc = torque on wheel

Front wheel brake force(F_f) :-
$T_w = T_d$
$F_f \times R_f = 200$
F_f (0.225) = 200 $F_f = 888.889$ N

Rear wheel brake force (F_r):-
$T_w = T_d$
$F_r \times R_r = 200$
F_r (0.301) = 200
$F_r = 664.451$ N

Average braking force = 776.67 N

Wheel base (L) = 54.5'' = 1.3842 m

Centre of gravity (h) = 25.57" = 0.6446 m

Distance of cog to front axle (x) = 27.028" = 0.6865 m

Coefficient of friction between wheel and ground (μ') = 0.7

Deceleration of vehicle under braking

$$d = \frac{L \times \mu' \times F_f}{m\,(Lx - \mu h)}$$

= 5.101 m/s^2

Stopping distance of vehicle is (s) = $\frac{V^2}{2d}$

= 4.721 m

Avg Braking Force	776.67 N
Braking torque	200 N
Deceleration	5.101m/s^2
Stopping distance	4.721 m

2.8. Seats

The seat system is designed in a way that it provides immense safety and comfort to the drivers. Both the drivers are provided with individual seats. They are placed side-to-side such that both the drivers are completely inside the chassis.

The seats are 14.5" apart and have 0.25" clearance from the chassis. Seat base is inclined at an angle of 22 degrees from horizontal. Seat back is inclined at an angle of 25 degrees from vertical.

The dimensions are:-

seat base (horizontal) = 16" × 15"

seat base (vertical) = 22.3" × 15"

The seats are placed at a height of 20" from the ground.

The seats are provided with 2" cushions to bear the weight of the drivers.

Sitting space height for both drivers is 43.4". Non Movable seats are used in the vehicle, no adjustment mechanism is provided to the seats.

3. Safety features of vehicle

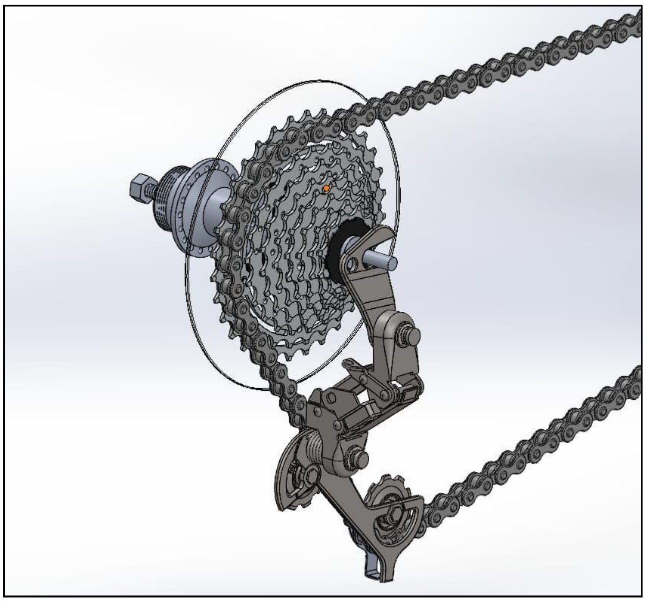
To ensure the safety of the vehicle and the strength of the chassis in unfortunate situations, impact analysis has been performed from different sides i.e. front impact, side impact, front roll, side roll.

The results of these analysis were promising and assured that neither the vehicle nor the driver will be affected during impacts.

The vehicle has been equipped with a fairing for protection of the driver from any frontal effects and to improve aerodynamics. Body panels are also used to protect the drivers from any types of weather.

3 point belt mechanism which covers the entire torso is being used. Doors made up of 2mm thick acrylic sheets are provided on both sides of the vehicle to ensure the safety of the driver in case of an unfortunate seat belt malfunction. The vehicle is fully enclosed to ensure the safety of the drivers.

Mud guards are provided in the vehicle to prevent small debris being flown up by the tyres of the vehicle. They protect the drivers and spectators from flying debris.



Dork disc is mounted to protect the spokes of the wheel from the chain coming off.

Cassette guard is mounted to ensure that the derailer or chain does not get caught in the spokes of the wheel.

Chain guards are being used to prevent the harm caused to drivers or vehicle due to any unfortunate malfunctioning chain or sprocket.

Sufficient mounting tabs were used for mounting various parts ensuring safety of the vehicle.

Helmets, elbow and knee guards are used by the drivers. Proper insulation is provided to the wires to prevent any short circuit and to avoid live wires coming in contact with the drivers.

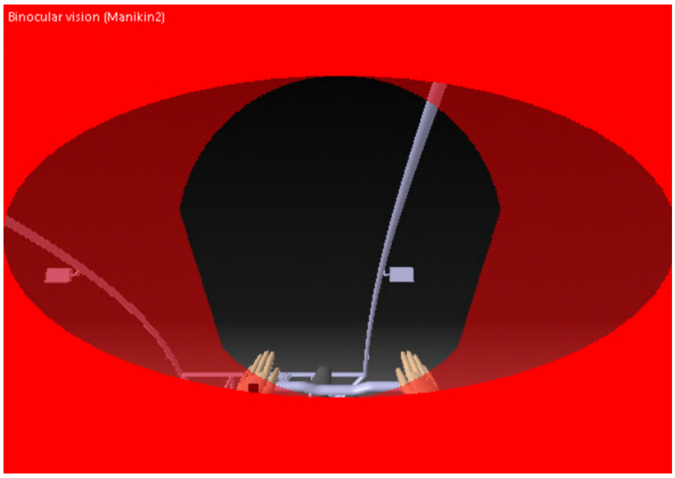
4. Ergonomics & Comfort Features

The vehicle is ergonomically engineered for an adult driver of above average height and weight. It has been designed such that it can also fit a 95th percentile adult human.

The vehicle is engineered in such a way that it remains stable at maximum speed and at full load capacity. The centre of gravity is positioned in such a way that there is no chance of toppling of the vehicle.

The fairing augments comfort the driver and the chassis design bestows a good line of sight.

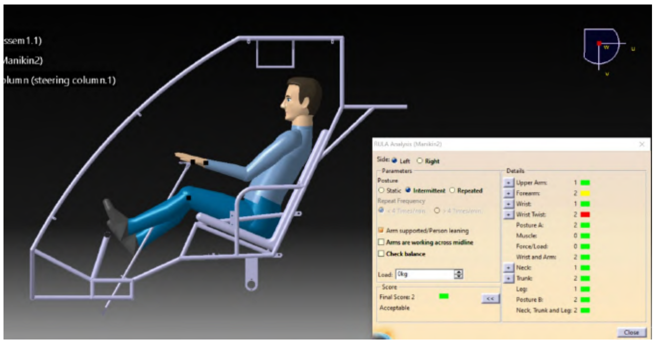
The figure below shows the line of sight of the driver. Special care has been taken while designing that one mirror is in direct sight and the other is in the peripheral vision.



The steering is at optimum distance from the driver. The driver can use it without applying any unnecessary effort.

The seats are fixed in a way that they are at optimum distance from the handle bars.

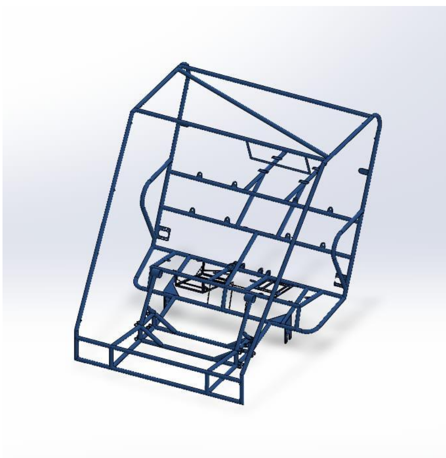
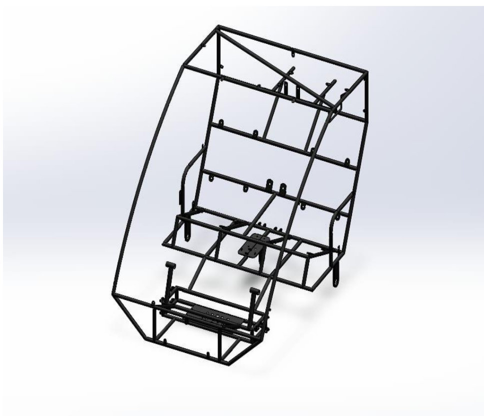
The image below shows RULA (Rapid Upper Limb Assessment) analysis which has been done to decide the optimal seat back angle for the driver. A RULA score of 2 was obtained, which is accepted as a comfortable seating position for automobiles.



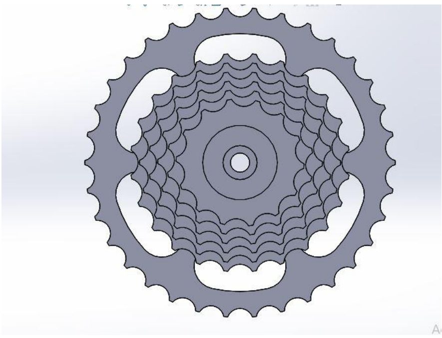
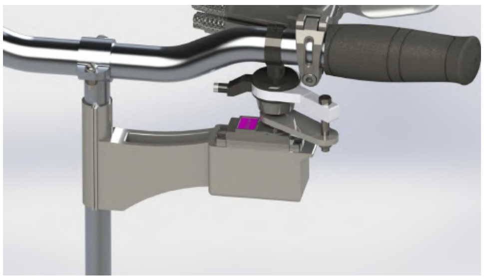
Pedals are placed in a position such that maximum torque can be drawn without tiring the drivers easily.

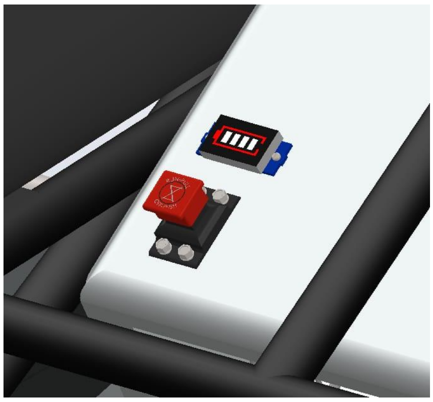
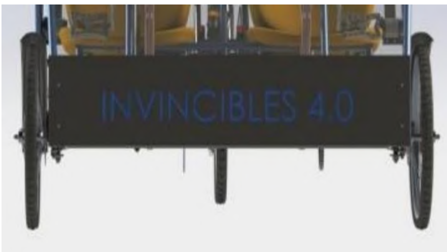
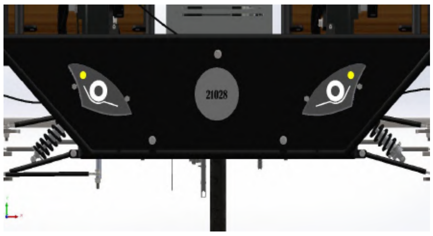
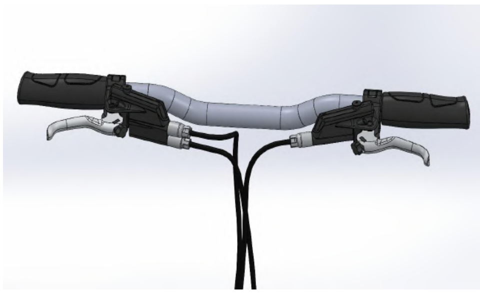
The suspension system absorbs almost all the shocks such that both the drivers do not feel any sudden jerks.

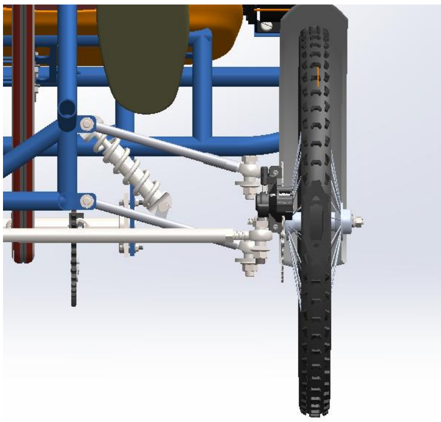
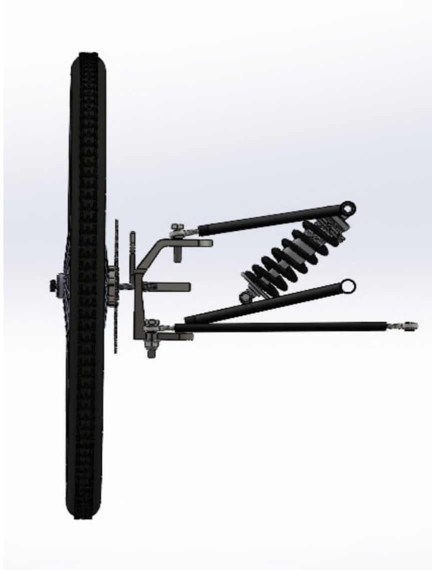
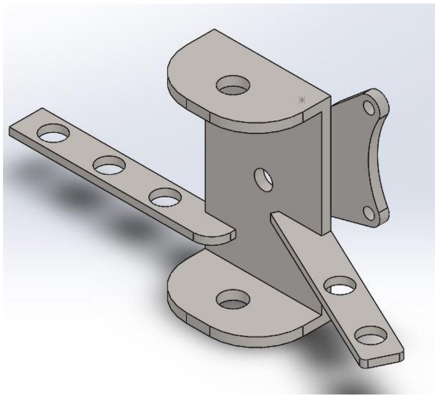
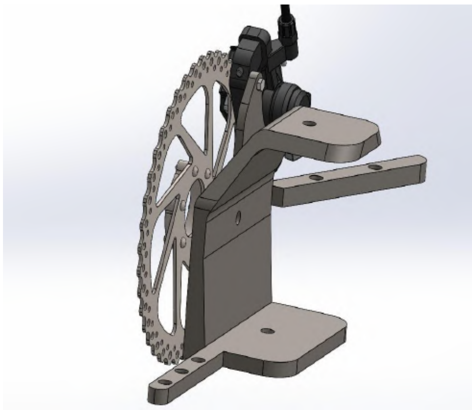
5. PREVIOUS VEHICLE COMPARISON

SYSTEM	Y'20	Y'21	Purpose of change/ Modify
FRAME	The seat back angle was too less	The seat back angle is according to drivers comfort	**To increase the comfort for the driver**
			
FRAME	The chassis material was AISI 1018	The chassis material is AISI 4130	**To reduce weight of the chassis and increase strength**
			

FRAME	Driver sitting position is congested and uncomfortable	Driver has larger sitting space and comfort	**To increase the comfort for the driver**
			
FRAME	The vehicle is exposed to atmosphere	Full enclosure of the vehicle was given	**To protect from various environmental conditions**
			

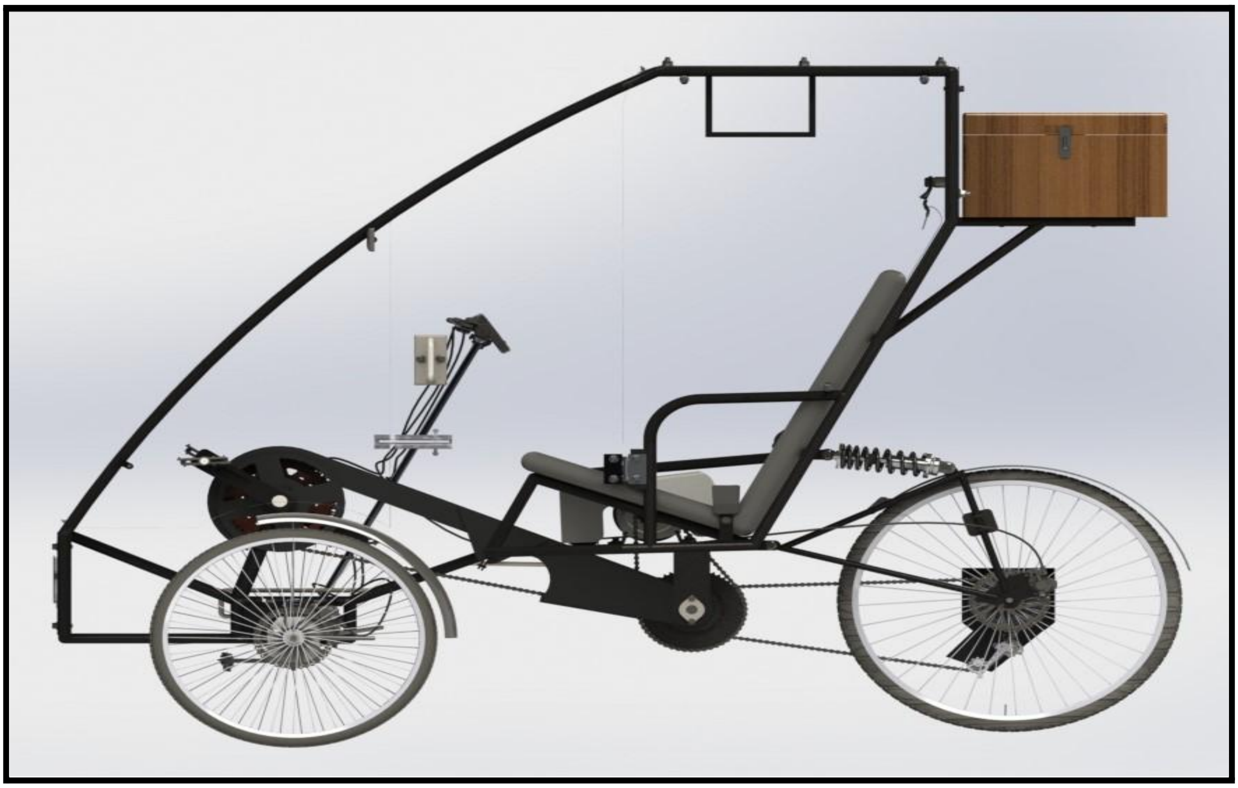
FRAME	Side view mirrors weren't present	Side view mirrors have been used	**Helps the driver to see outside the peripheral vision**
			
TRANSMISSION	14-34 Teeth 7 speed freewheel was used	11-32 Teeth 9 speed cassette was used	**To increase the range of speeds**
			
TRANSMISSION	Shifting of gears was only through manual mode	Automatic gear shifting was introduced	**Easy to use in heavy traffic. Accurate and effortless shifting of gears**
			

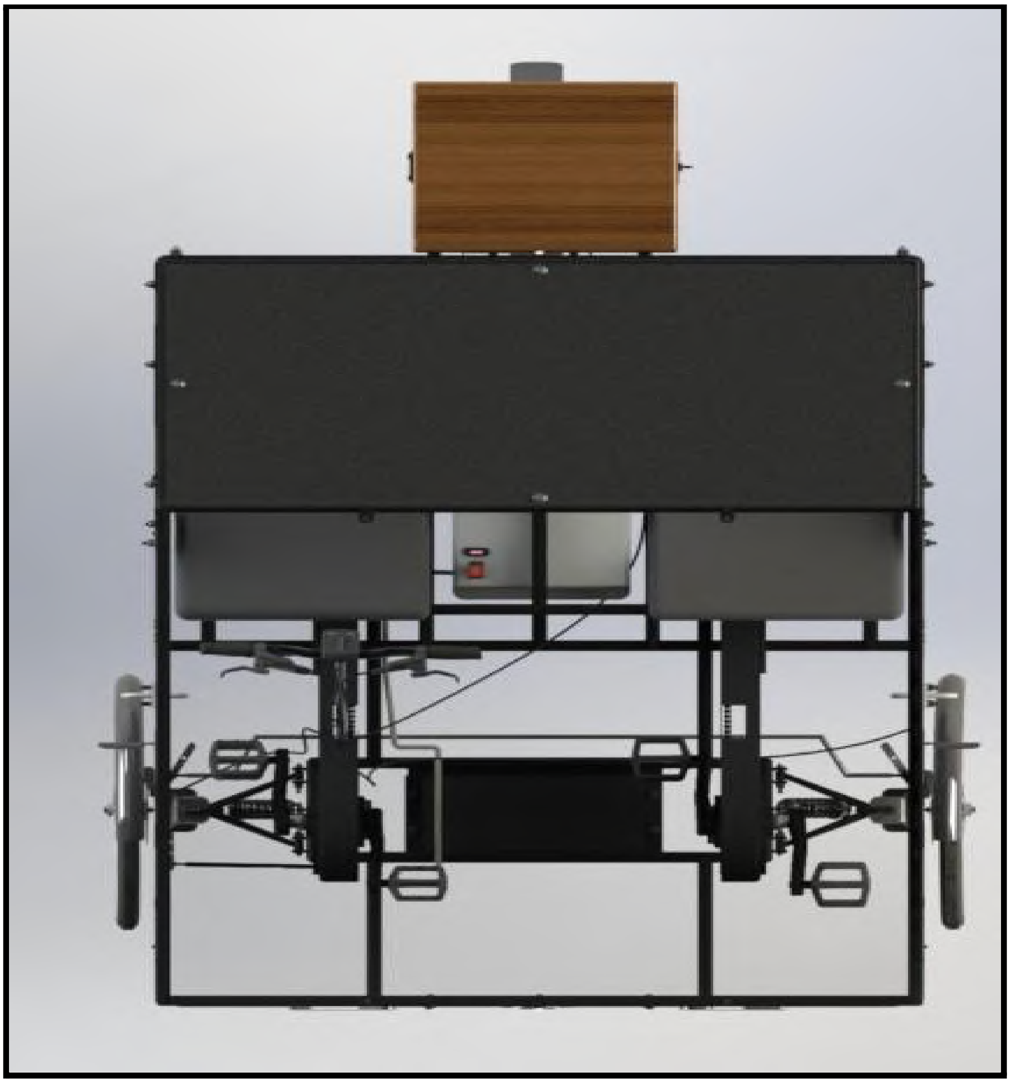
ELECTRIC DRIVE	Battery indicator and was not there	Battery indicator was introduced	**To indicate the battery capacity**
			
ACCESSORIES	Headlights are not used	Headlights are used	**To improve visibility at night. Use indicators while changing lanes, overtaking, taking turns**
			
BRAKES	Single brake levers are used	Dual brake levers are used	**To apply brakes at both the front wheels at the same time**
			

SUSPENSION	A-Arms were parallel	A-Arms are not parallel	**To avoid camber change during bumps**
			
SUSPENSION	Knuckle was designed without considering stress concentrations	Knuckle was designed by conducting topology optimisation	**Weight of knuckle is decreased**
			
STEERING	Tie rod was in front of steering column	Tie rod is behind steering column	**To increase comfort, clearance and to follow proper Ackermann geometry**
			

APPENDIX-1 : PICTORIAL PRESENTATIONS

VEHICLE VIEW

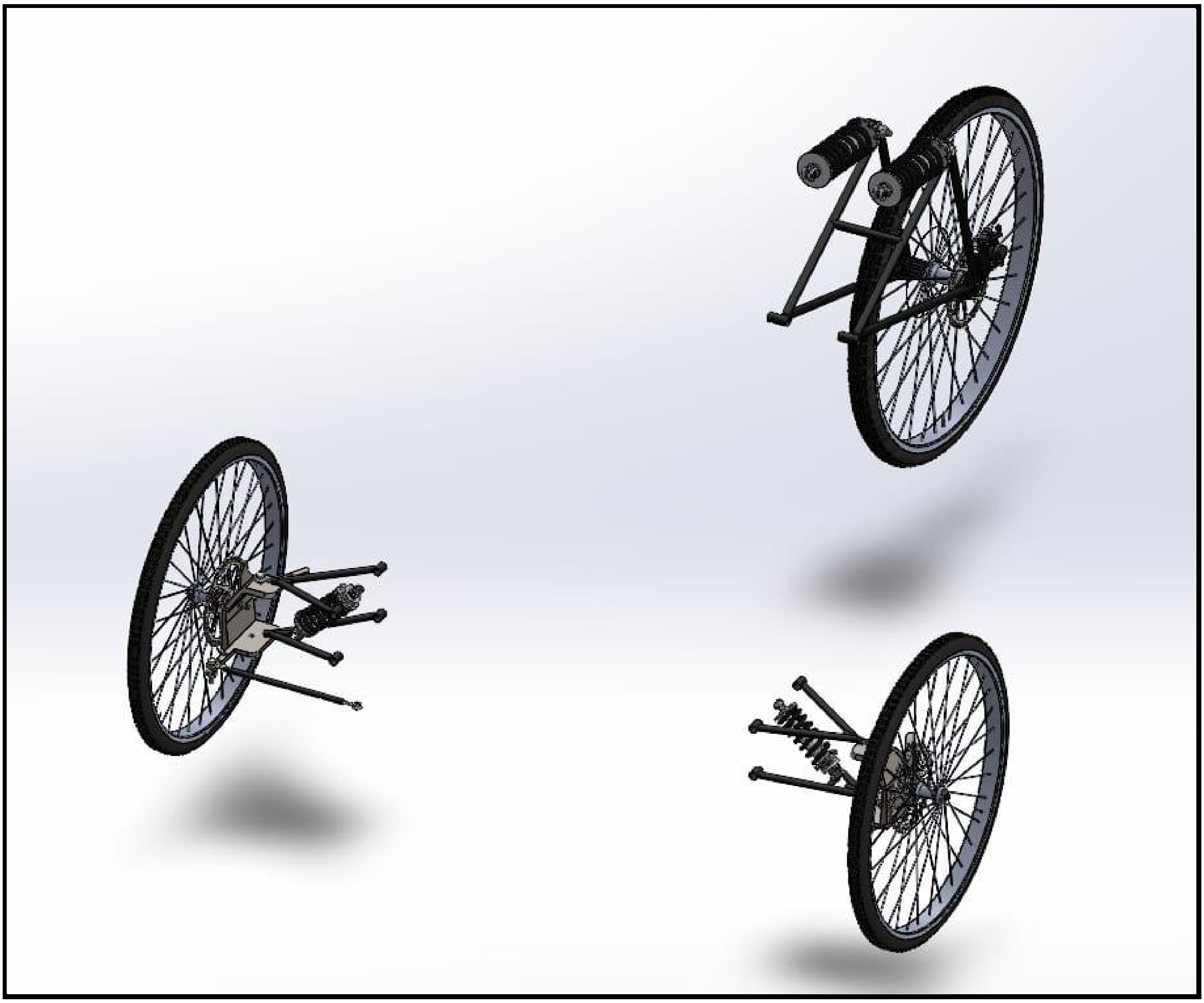




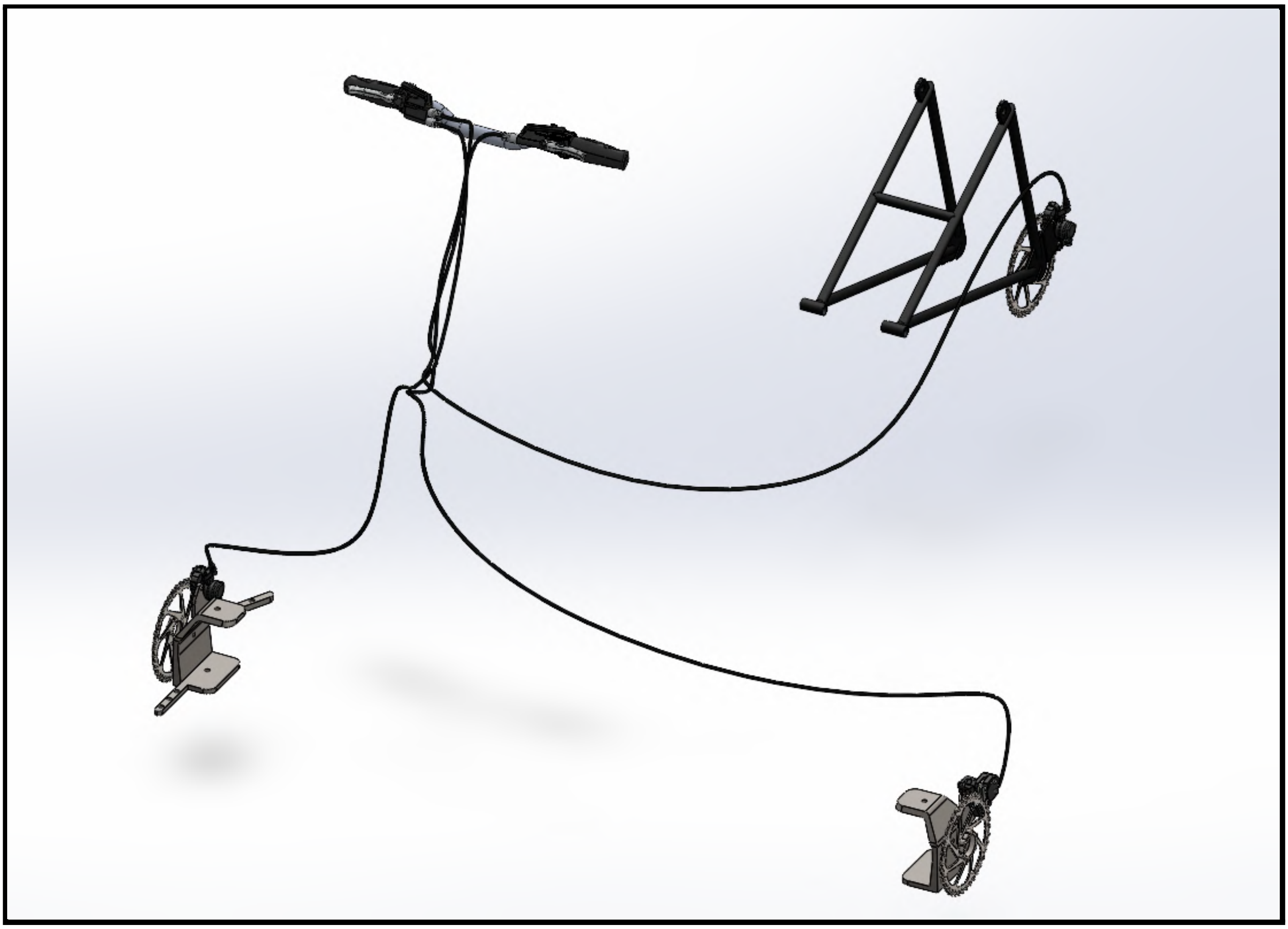
VEHICLE VIEW WITH DOORS

STEERING SUBSYSTEM

SUSPENSION SUBSYSTEM



BRAKING SYSTEM



TRANSMISSION SYSTEM