TEAM ATR

(ALL TERRAIN RACERS)

**ORIENTAL INSTITUTE OF SCIENCE AND TECHNOLOGY**

**Bhopal**

# FINAL DESIGN REPORT

*Submitted to,*

# ISNEE

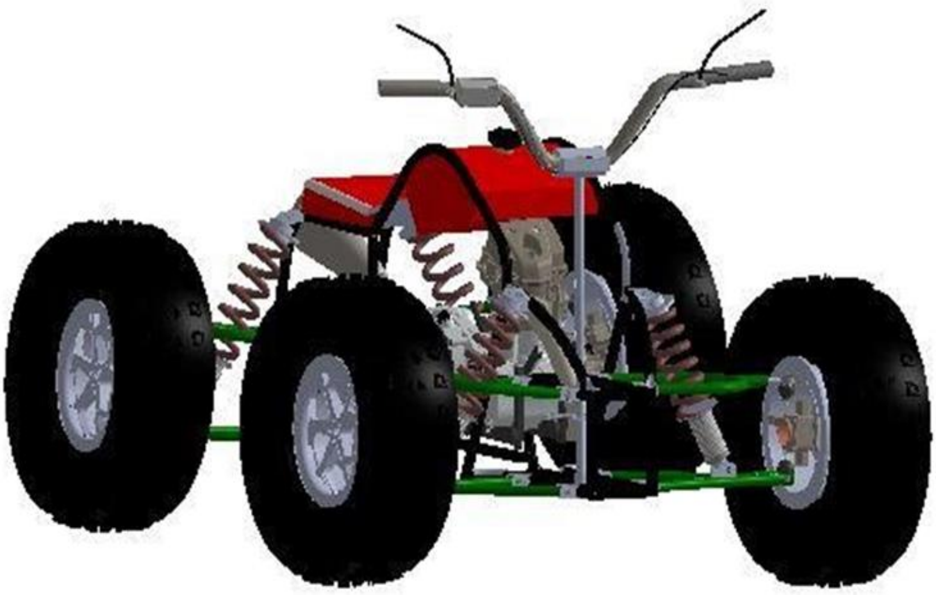
**(INDIAN SOCIETY OF NEW ERA ENGINEERS)**

## Technical Specifications:

| | |
|---|---|
| Wheel base | 1066.8 mm |
| Front Wheel track | 1016 mm |
| Rear Wheel track | 965.2 mm |
| Static Ride Height | 9” |
| C G Height | 13” |
| Curb Weight | 140 Kg |
| Weight Distribution | 55:45 |

## Performance Target:

| | |
|---|---|
| Max. Speed | 50 Kmph |
| Grade ability | 38°50” |
| Max. Accn. | 8.017 m/s$^2$ |
| Stopping Distance | 4.22 m |
| Transmission Efficiency | 90% |
| Gear Ratio | |



OBSIDIAN FURY

by team ATR

# Abstract

The “Indian Society of New Era Engineers” conduct competitions that challenge aspiring engineers to create a miniature off-road vehicle. The ISNEE QUAD TORC® competition objective is to design and fabricate a prototype vehicle that could be manufactured for competition as well as for consumer sale. The Team ATR of OIST, BHOPAL has accepted the challenge to participate in this competition. An aspect of this competition is to compose a design documentation package that creates an overview of the vehicle’s construction elements. The ATR team has created this FINAL DESIGN REPORT (FDR) while following the rule book constraints provided by ISNEE.

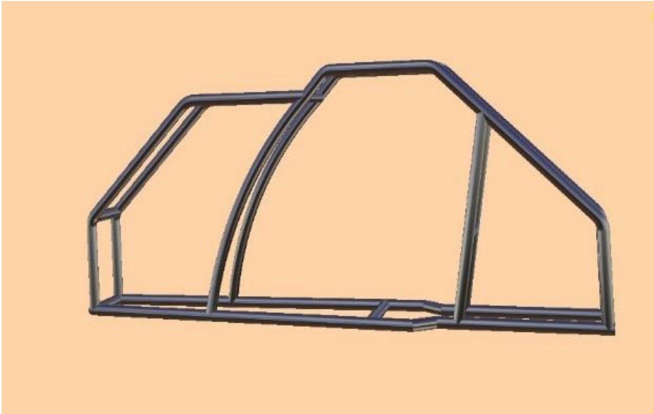
# Introduction

Designing purpose of this Quad bike is to manufacture an off road vehicle that could help in transportation in hilly areas, farming field and as a reliable experience for a weekend enthusiast. In order to accomplish this task, different design aspects of a Quad Bike vehicle were analyzed, and certain elements of the bike were chosen for specific focus. There are many facets to an off-road vehicle, such as the chassis, suspension, steering, drive-train, and braking, all of which require thorough design concentration. The points of the car that the Team ATR decided to specifically focus on were the chassis, drive-train, suspension and innovations. The most time and effort went into designing and implementing these components of the vehicle because it was felt that they most dramatically affect the off-road driving experience. During the entire design process, consumer interest through innovative, inexpensive, and effective methods was always the primary goal.

# ACKNOWLEDGEMENT

Team ‘ATR’ is thankful to Dr. A.K Tiwari, Director, OIST, for her constant support, guidance and motivation which keep our spirits high. We also thank our Faculty advisor Prof. Abhishek Choubey & HOD (Mechanical) Prof. S.L Ahirwar for his technical guidance and expertise which always keeps us going for impacts created in any certain crash or rollover. It must be strong and durable taking always in account the weight distribution for a better performance.

## Frame Design



**OBJECTIVE** – The chassis is the component in charge of supporting all other vehicle's subsystems with the plus of taking care of the driver safety at all time.

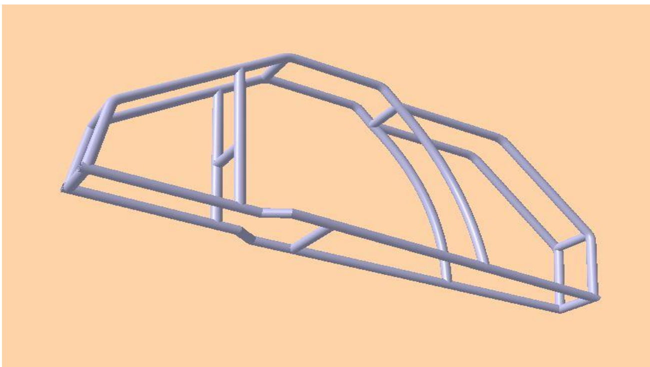
4130 Chrome-Moly Steel is the best suitable material so following it we selected it over 1018 Steel because 4130 Steel has a greater strength to weight ratio. Along with material selection, tube diameter was also taken into consideration. It was decided to create the Roll Cage using 1 inch OD and 2mm wall thickness, 4130 Steel tubing as it was thought to be more structurally sound than a larger diameter tube.

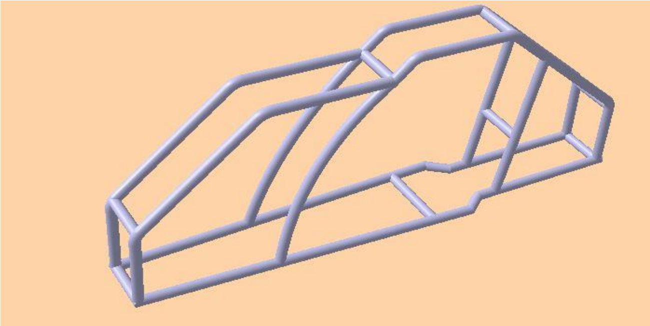
### Frame Design Considerations:

| CONSIDERATIONS | PRIORITY | REASON |
|---|---|---|
| Light-Weight | Essential | A light race ATV is a fast ATV |
| Durable | Essential | Must not deform during rugged Driving |
| Meet Requirements | Essential | Must meet requirements to Compete |
| Simple Frame | High | Majority of Frame fabrication done in House |
| Attractive Design | Desired | Easier to sell And aesthetically Pleasing vehicle |
| Cost | Low | ATV needs to be within budget |

### Material Selection:

| MATERIAL | 1080 STEEL | 4130 STEEL | 4130 STEEL |
|---|---|---|---|
| OUTTER DIAMETER | 25.40mm | 25.40mm | 31.75mm |
| WALL THICKNESS | 2.04mm | 2.04mm | 1.69mm |
| BENDING STIFFNESS | 3791.1 Nm^2 | 3791.1 Nm^2 | 3635.1 Nm^2 |
| BENDING STRENGTH | 391.3Nm | 467.4Nm | 487Nm |
| WEIGHT PER METER | 1.486 kg | 1.486 kg | 1.229 kg |





**Roll cage design made in CatiaV5**

## ROLL CAGE DESIGN SPECIFICATIONS:

| | |
|---|---|
| Type | Space Frame |
| Material | Normalized AISI 4130 Chrome- Moly. Steel |
| Mass of Roll cage | 12 kg |
| Length of Roll cage | 50 inches |
| Width of Roll cage | 8” (max.)<br>5” (min.) |
| Height of Roll cage | 26” |
| Total length of pipes | 18 m |
| Weld joints | 22 |
| No. of Bends | 16 |
| Cross section | Outer Diameter - 25.40 mm<br>Thickness – 2.04mm |

## Finite Element Analysis (FEA):

Finite element is a method for the approximate solution of partial differential equations that model physical problems such as: Solution of elasticity problems, Determine displacement, stress and strain fields. Static, transient dynamic, steady state dynamic, Roll cage analyzed at much higher forces than in real case scenario.

### Loading Analysis:

To properly approximate the loading that the vehicle will see an analysis of the impact loading seen in various types of accident was required. To properly model the impact forces, the deceleration of the after impact needs to be found. It was found that human body will pass out at loads much higher than 7g. And the Crash pulse scenario standard set by industries is 0.15 to 0.3sec. We considered this to be around 2.5 sec. It is assumed that worst case collision will be seen when the ATV runs into stationary object.

### FEA of Roll cage-

**For AISI 4130 alloy steel –**

Young’s modulus-205 GPa

Poisson’s ratio- 0.27-0.29 (say0.28)

For all the analysis, weight of the vehicle is taken to be 160kg

### Objectives of FEA of Roll Cage-

1) To have adequate factor of safety even in worst case scenarios to ensure driver safety.

2) To have greater torsional stiffness to ensure less deflection under dynamic loading and enhanced performance.

3) To ensure that the natural frequency of the roll cages does not matches with the engine working range frequency to avoid resonance.

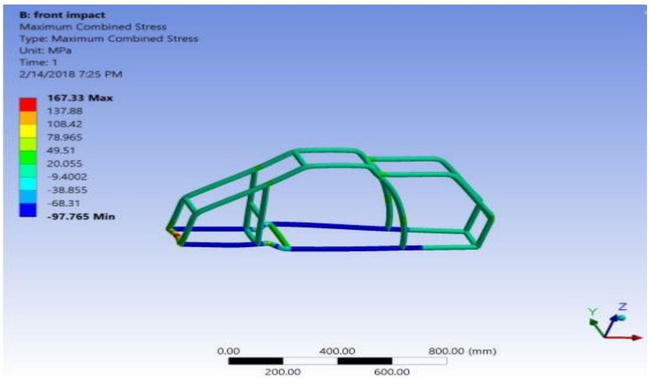
## Static Analysis:

1) Front Impact
2) Rear Impact
3) Side Impact
4) Roll Over Impact

## Front Impact Analysis

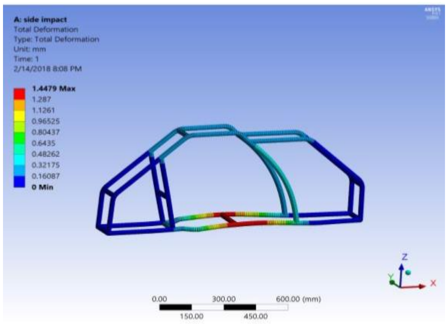
The impact test or crash test is performed assuming the vehicle hits the static rigid wall at top speed of 50 kmph. The collision is assumed to be perfectly plastic i.e, vehicle comes to rest after collision.

| Front Impact | 6G (15303.6) |
|---|---|
| Max. Deformation | 1.36 mm |
| Max. Stress | 150.331 Mpa |
| Factor of Safety | 3.05 |



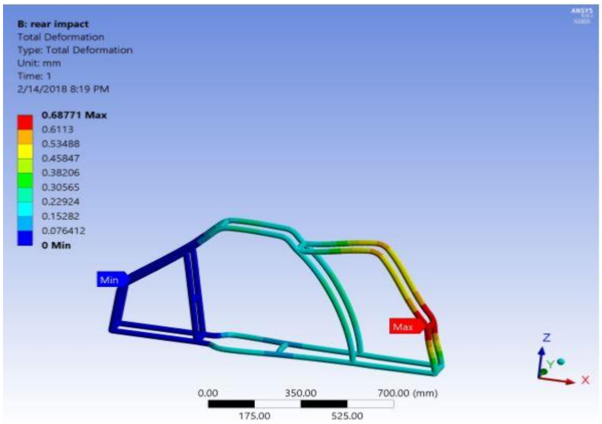

## Side Impact Analysis

| Side impact | 3G ( 7651.8 N ) |
|---|---|
| Max. Deformation | 0.95 mm |
| Max . Stress | 206.1 Mpa |
| Factor of Safety | 2.23 ( >2 Design is Safe ) |



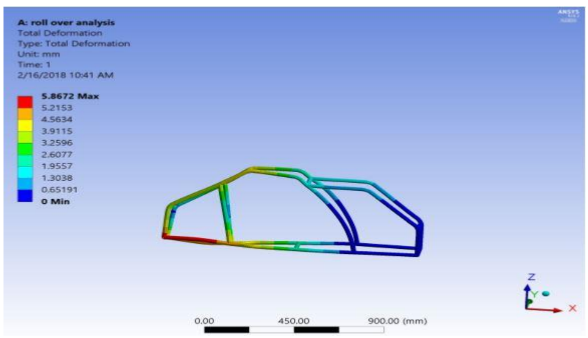

## Rear Impact Analysis

| Rear impact | 3G ( 7651.8 N ) |
|---|---|
| Max. Deformation | 0.95 mm |
| Max. Stress | 56.196 Mpa |
| Factor of Safety | 3.23 (>2 Design is Safe ) |




## Roll Over Impact Analysis

| Roll over Impact | 3.5G ( 8927.1 N) |
|---|---|
| Max. Deformation | 0.30 mm |
| Max. stress | 80.63 Mpa |
| Factor of safety | 3.05 ( > Design is Safe ) |



## Suspension Design:

**OBJECTIVE -**

1. Designing a suspension which will influence significantly on comfort, safety and maneuverability.

2. Contributing to vehicles road holding/handling and braking for good active safety and driving pleasure.

3. Protect the vehicle from damage and wear from force of

Impact with obstacles (including landing after jumping)

## DESIGN METHODOLOGY

The overall purpose of a suspension system is to absorb Impact from coarse irregularities such as bumps and distribute that force with least amount of discomfort to the driver. We completed this objective by doing extensive research on the front and rear suspension arm's geometry to help reduce as much body roll as possible. Proper camber angles are provided to the front wheels. The shocks will be set provide the proper dampening and spring coefficients to provide a smooth and well performing ride.

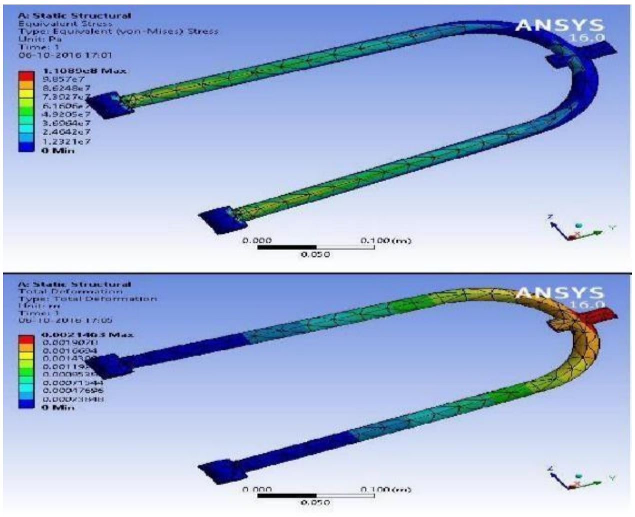
| WHEEL GEOMETRY | |
|---|---|
| Camber Angle | 30 |
| Caster Angle | 70 |
| Toe-in | 20 |
| KPI | 30 |
| Ride Height | 9" |
| Scrub radius | 50.2 mm |

| DESIGN CONSIDERATIONS | |
|---|---|
| Un sprung mass of vehicle | 70 kg |
| Static Stability Factor (Front)<br>Static Stability Factor (Rear) | 1.10<br>1.03 |

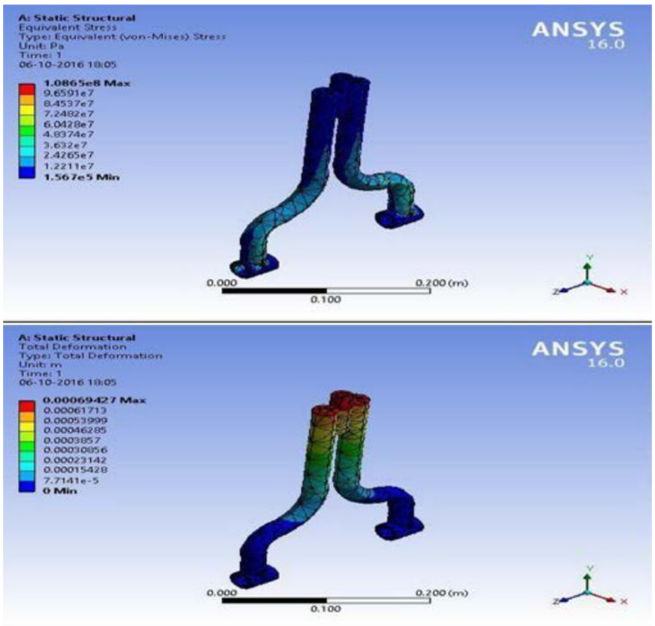
**WISHBONE ARMS :**

For both front & rear suspensions we chose one with a Double arm wishbone type suspension. It provided specious mounting position, load bearing capacity besides better camber recovery. Unequal Un-parallel double wishbone suspension. The tire need to gain negative camber in a rolling situation, keeping the tire flat on the ground.

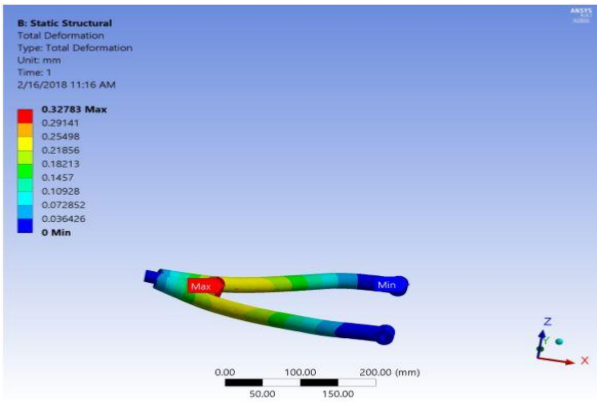
Design for optimal geometry of the control arms is done to both support the race-weight of the vehicle as well as to provide optimal performance. Design of the control arms also includes maximum adjustability in order to tune the suspension for a given task at hand.



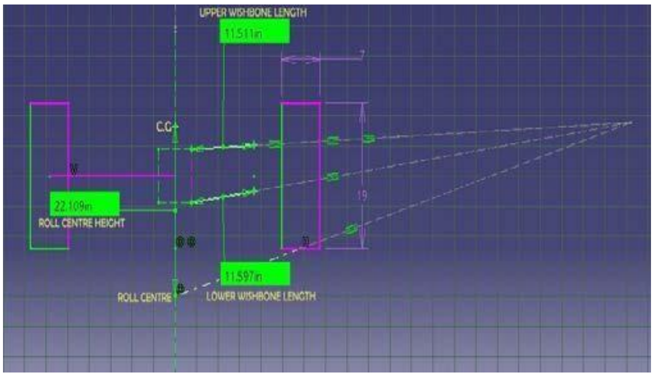

Front Upper Arm FEA




Rear Upper Arm FEA




Front & Rear Lower Arm FEA




Wishbones length Calculation in Catia V5

## SHOCK ABSORBERS:

All the suspension is equipped with custom-made shock- absorbers which allows the automatic preload adjustment in order to keep the optimal vehicle trim. The shock absorber is completely self-adopting so no HMI is needed. Less un sprung weight which helps to reduce the overall weight of the quad bike and thus provide faster acceleration.

## CALCULATIONS:

### FRONT SUSPENSIONS

Arm length (d2) = 398 mm
Length of A arm up to suspension mounting (d1) = 304.6 mm
Motion ratio = shock travel/wheel travel = d1/d2 =
**Motion Ratio (MR) = 0.701**
**Spring rate (K)** = G x $d^4$ / 8 $D^3$ n
G=81370 N/mm^2, N=10, **D**=60 mm, d=10 mm
**K= 47.08 N/mm**
**Wheel Rate (WR)** = (MR) ^2 * K * ACF
= (.701) ^2 *47.05 *cos38.9
**WR = 18 N/mm**

**Suspension Freq (SF)** = 187.8{(WR/ sprung wt)} ^1/2/ (60)
**Spring Wt** = 33.75* 9.81
SF = 187.8{(18/33.75*9.81*.2248)} ^1/2/ (60)
**SF = 1.539 Hz**

**Front Ride Rate (Krf)** = 4*(3.14) ^2*(SF) ^2*Wf
**Krf = 61458.3**

**Rear ride rate** = 4*(3.14) ^2* (SF) ^2* Wir
= 4* (3.14) ^2* (1.539) ^2 * 82.5 *9.81
**Krr** = 7714.2 * 9.81 **= 75676.302**

### REAR SUSPENSIONS

Arm length (d2) = 297.18 mm
Length up to suspension (d1) = 165.1 mm
**Motion ratio (MR) = .55**

**Spring Rate (K) = 47.08 N/mm**

**Wheel Rate (WR) = 13.06 N/mm**

**Suspension Frequency = 0.677**

**Roll Moment at front = (hf-Rcf)*Mf*g**
= (393.7-152.5)*67.5*9.81
= **159.716 Nm**
**Front roll rate factor (F¢sf)** =
3.14*(tf)^2*Wf left * Wr right /180(Wrf+Wrr)
3.14*(tf)^2*Wr/360 (tf = 40”0

**Front roll rate factor = 466170.8 N-mm-deg roll**

**Roll rate at front** = Roll Moment/ F¢sf
= **0.3426 degree/g**

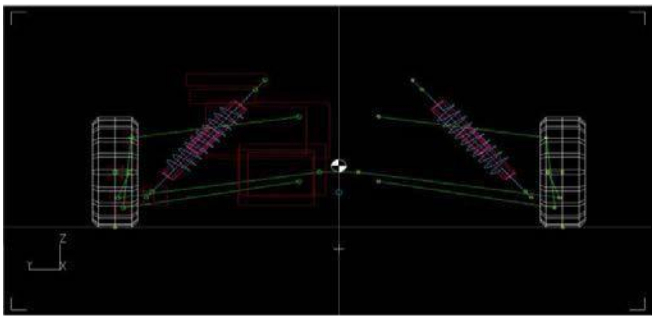
**Rear roll rate factor** = 3.14* (tr)^2* Wr/360
(tr=39”) **F¢R= 541632.292 N-mm-degroll**

**Roll moment at rear** = (hr – Rcr)*Mr*g
= (419.1-152)*82.5*9.81
**= 216170.07 Nm**

**Roll rate at rear** = 216170.7/541632.292
= **0.39 deg/g**

**Coil rate** = Wheel rate * (Suspension leverage)^2
= 18 * (1/.71)^2
= **35.70 N/mm**

Suspension and Wheel geometry:



# STEERING SYSTEM:

Objective:

The objective of steering system is to provide max directional control of the vehicle and provide easy maneuverability of the vehicle in all type of terrains with appreciable safety and minimum effort. Typical target for a quad vehicle designer is to try and achieve the least turning radius. This is achieved by selecting a proper steering system (Centre point Steering 1:1). The response from the road must be optimum such that the driver gets a suitable feel of the road but at the same time the handling is not affected due to bumps.

Design:

we have selected 4 bar linkage centralized point steering system for our Quad bike. Rear track width is kept slightly less than front track width to create a slight over steer in tight cornering situation which allows easier maneuverability at high speed.

**Calculations**

| | |
|---|---|
| Wheelbase(L) | 42”=1.066m |
| Front Wheel Track | 38”=0.965m |
| Rear Track Width | 36”=0.914m |
| Weight Distribution | 55:45 |
| Total Weight | 140kg+60Kg (Driver) |
| Turning Radius | 2.41m |
| Static Weight on Front Wheel | 110.5 Kg |
| Static Weight on Rear Wheel | 90 Kg |
| Axle Length | 914.4mm |
| Mechanical Trail | 25.5mm |
| Scrub radius | 50.2mm |
| Turning Velocity | 5 m/s |
| Steering Arm Length | 140mm |

| | |
|---|---|
| **Front**<br>Tyre width<br>Radius of Tyre<br>Radius of Steering | <br>190.5mm<br>292.1mm<br>342.9 mm |
| **Rear**<br>Tyre Width<br>Radius of Wheel<br>Radius of Steering | 254<br>mm<br>279.4mm<br>342.9m |
| | |
| Load on Front Wheel<br>Load on each Front Wheel | 110.5Kg<br>55.25 Kg |
| | |
| Tripod Radius<br>Tripod Height | 80 mm<br>75.4 mm |
| Inner Turning angle Outer turning angle | $32.62^0$<br>$22.63^0$ |

# ACKERMANN CALCULATION:

Ackermann Angle (φ3)

= $\tan^{-1}$ (Axle Length / (2 x Wheel base))

= $\tan^{-1}$ (914.4 / (2*1066.8))

= 23.19

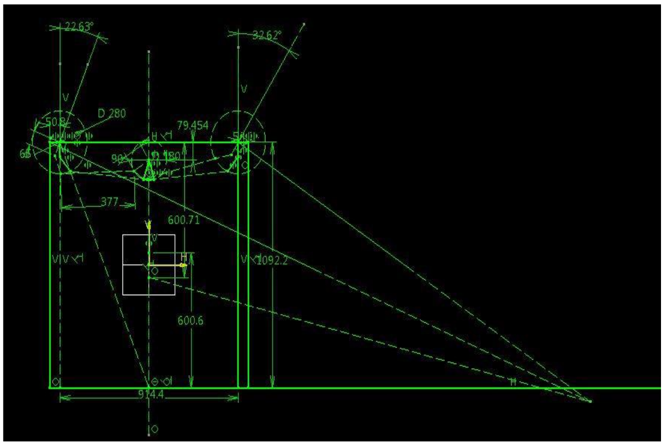
1.3. ACKERMANN PERCENTAGE:

Ackermann Percentage is calculated from referring to the CREO model of the Steering system; for error. [Error = 10.03 mm] Ackermann Percentage

= {(1092.2 + 10.03)/1092.2} x 100

= 1. x 100

= 100.93 % (*Over steer)*



Ackermann Angle Calculation Drawing in Catia V5

**DYNAMIC CONDITION**:

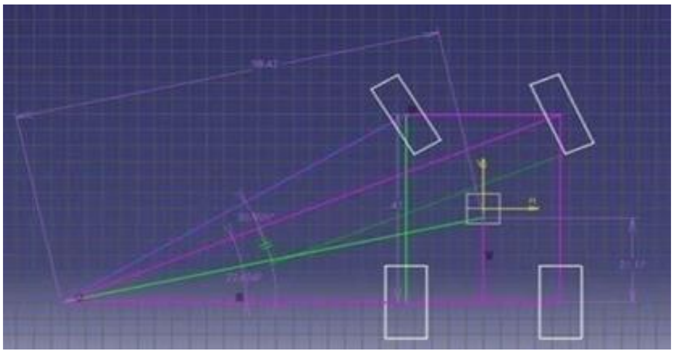
Turning Radius Centre of Gravity
=2.414m
Turning Radius Inner Wheel = 2.026 m
Turning Radius Outer Wheel =2.839mm
Height Centre of Gravity = 330.2mm
Tie-Rod Angle top view (ϕ1) = 50
Tie-Rod Angle front view (ϕ2) = 1
Ackermann Angle (ϕ3) = 23.1
I. Cornering Force -
= (Turning Velocity) ^2 /
(Turning Radius Centre of Gravity x
g)
= 5^2 / (2.414x 9.81)
= 1.05N
II. Weight transfer at cornering -
= (Cornering Force x Height C.G. x Front Axle Load) / Track width
= (1.05 x330.2 x 103.5) / (1066.8)
= 46.40 kg
III. Weight on Inner Wheel -
= (Load on Inner Wheel - Weight transfer at cornering)
= (51.75- 46.40)
= 5.35kg
IV. Weight on Outer Wheel -
= (Load on Outer Wheel + Weight transfer at cornering)
= (51.75+ 46.40)
= 98.15 kg
V. Lateral Force on Inner Wheel -
= (Weight on Inner Wheel x Turning Velocity2) / Turning Radius Inner Wheel
= (5.35 x 52) / 2.026
= 66.016N
VI. Lateral Force on Outer Wheel -
= (Weight on Outer Wheel x Turning Velocity2) / Turning Radius Outer Wheel
= (98.15 x 52) / 2.839
= 864.3N

VII. Moment due to Lateral Force - A. Moment Inner Wheel
= (Lateral Force on Inner Wheel x Radius of Wheel x tan (caster angle))
= (66.016 x 292.1 x tan (7))
= 2367.68 N-mm
B. Moment Outer Wheel
= (Lateral Force on Outer Wheel x Radius of Wheel x tan (caster angle))
= (864.3x 292.1 x tan (7))
= 30998.43 N-mm
Moment at Kingpin due to Lateral Force
= Moment Inner Wheel + Moment Outer Wheel
= 2367.68+ 30998.43
= 33366.1N-mm
VIII. Self-Aligning Torque –
= lateral force*cos(caster)*mechanical trail
=930.3*cos7*(1.41*25.4)
=33069.98Nmm
IX. Net Moment at Kingpin -
= Moment due to Self-Aligning Torque + Moment due to Lateral Force
= 33069.98+ 33366.1
=66436.08 N-mm
X. Perpendicular force at Steering Arm -
= Net Moment at Kingpin / Steering Arm Length
= 66436.08 / (25.4*5.5)
= 288.34N
XI. Force along Tie-rod at arm end -
= Perpendicular force at Steering Arm / [cos (ϕ1+ϕ3) x cos (ϕ2)]
= 288.34/ [cos (5.00+22.714) x cos (1)]
= 325.76N
XII. Force along Tie-rod at tripod end -
= Force along Tie-rod at arm end / tripod efficiency
= 325.76/0.9
= 361.96N
XIII. Horizontal force at tripod end -
= Force along Tie-rod at tripod end / [cos (ϕ1) x cos (ϕ2)]
=361.76//[cos(5)*cos(1)]= 363.4N
XIV. Actual Horizontal force at tripod end –
= Horizontal force at tripod end / tripod efficiency
= 363.47 / 0.9
= 403.86 N
XV. Moment at Steering Column -
= Actual Horizontal force at tripod end x
Radius
= 403.47 x 90
= 36347.4 N-mm
XVI. Steering Effort -
= Moment at Steering Column / Radius of Steering
= 36347/342.9
= 106N

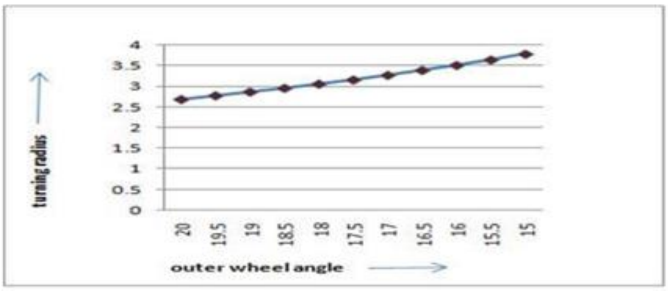
## Cornering Stiffness:

**For front**, C.S=42.18 N/degree
**For Rear**, C.S=27.071N/degree

Turning Radius

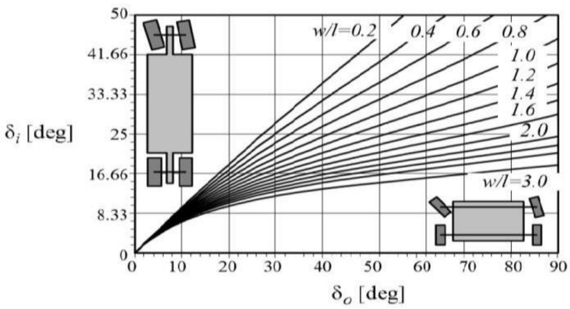


Turning Radius Drawing in Catia V5




Graph

**Effect of the w/l on the Ackermann condition for the front wheel steering vehicles:**

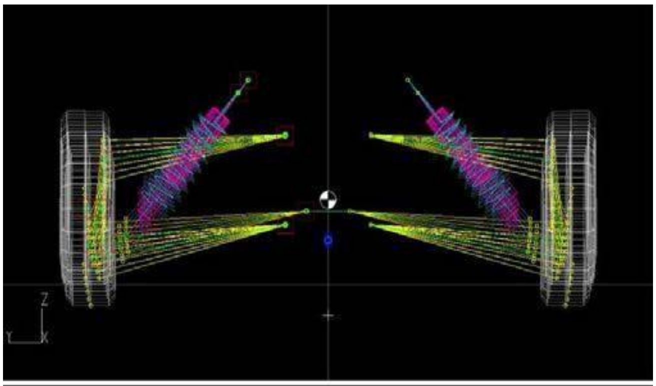



BUMP AND ROLL STEER

Steer with ride travel is very common in all terrain scenarios steer with ride travel is undesirable because if the wheel steers when it runs over a bumps or when the car rolls in a turn the car will travel on a path that a driver did not select intentionally.

Ride/Bump and roll steer are a function of the steering geometry. If the tied rod is not aimed at the instant axis of the motion of the suspension system, then the steer will occur with ride because the steering and suspension are moving about different centers.

If the tie rod is not of the correct length for its location, then it will not continue to point at the instant axis when the suspension travelled in ride.



Bump rolls & Steer Simulation in Lotus

**DIFFERENTIAL:**

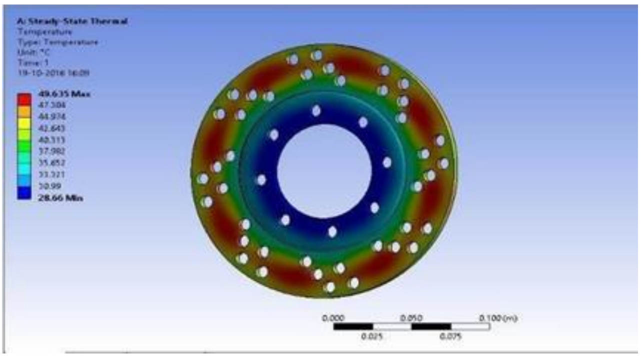
An open differential is a type of differential that allows its two output shafts to rotate at different speeds while cornering. In our ATV we used a differential instead of single shaft drive because of certain dynamic advantages it provided while cornering and while riding through non-uniform terrains. The torque transmitted by an open differential will always be equal at both wheels.

**BRAKING SYSTEM:**

Design

The braking system is composed of disc brakes on both front and rear wheels. A rotor disc & caliper assembly is mounted on

each front & rear axle to satisfy the braking requirements of our quad bike such as terrain of the track, speed limits, driver ergonomics and other rulebook constraints.




Disc Analysis

Calculations:

**The total weight of the vehicle along with an average weight of driver (60 kg) was estimated to be 200 kg. The weight distribution for the quad bike was estimated to be approximately 55:45 from the front to the rear. The static weight distribution of the vehicle is 110kg at the front and 90 kg at the rear.**

## Rear Braking:

### a) Known and assumed Parameters:

1. Total weight of vehicle = 200kg

2. Radius of Tire = 0.2794m

3. Weight distribution factor:

Front wheels = 0.55

Rear wheels = 0.45

4. Master cylinder diameter = 0.01905m=19.05mm

5. Caliper cylinder diameter = 0.0255m=25.5mm

6. Rotor Disc Diameter = 0.175m=17.5cm

Effective diameter of rear disk = 0.0155m=15.5cm

7. No of disc used = 1

Assumed Parameters:

1. Paddle Force = 200N

2. Paddle ratio = 5

### b) Calculating the actual braking force (Fr) and braking torque (Tr):

- Force transmitted to the piston of Tandem Master Cylinder= Paddle force* Paddle ratio = 200*5 = 1000 N
- Area of piston in master cylinder = π*0.01905*0.01905/4=2.85*10^(-4)m2
- Area of caliper piston= π *0.0255*0.0255/4 = 5.107*10^-4 m2
- Pressure generated=force transmitted/area of master cylinder piston =(200*5)/(2.85*10^-4)=35.087 *10^5 N/m^2

Force at caliper= pressure generated* area of caliper piston= (35.087*10^5*5.107*10^-4) = 1791.89N

- Braking force provided by caliper= force at caliper*coefficient of friction between caliper and disc = 1791.89*0.5 = 895.9 N
- Total braking force by both calipers on the disc =895.9*2= 1791.89 N
- Total braking force acting on both discs (Fr) = 1791.89* 2 =3583.6N
- Braking torque at disc= total braking force* distance from center of disc where the effective pressure acts= 3583.6*0.0775
- **Tr (generated) = 277.74 Nm**

### c) Decelerations of vehicle:

- Decelerations of vehicle = Fr / Weight of vehicle = 3583.6/200 = 17.918 m/sec$^2$

### d) Static load on wheels:

- Load on front wheel = weight distribution factor on front wheel * mass of vehicle *g= 0.55*200*9.81 = 1079.1 N
- Load on rear wheel = weight distribution factor on rear wheel * mass of vehicle*g = 0.45*200*9.81 = 882.9 N

### e) Relative centre of gravity (X):

- X = Vertical distance of C.G from ground / wheel base = 13 / 42 = 0.309

### f) Dynamic axle load:

- Dynamic load transfer: (CG height/wheel base)*(mass of vehicle)*Decelerations= 0.309*200*17.918 =1107.332 N
- Dynamic load on rear axle= static load+ load transfer = 1079.1 + 1107.332 = 2186.432 N
- Dynamic load on rear wheel= load on axle /2= 2186.432/2 = 1093.216 N
- Frictional force on rear wheel= dynamic load*coefficient of friction between road and tires = 1093.216*0.65 = 710.5904 N
- Braking torque required at rear wheel= Frictional force on rear wheel* rolling radius wheel = 710.59* 0.2794 = 198.538 Nm
- **As, the Tr (generated) = 277.7Nm >Tr (required) = 198.538Nm**

  **Our design is safe.**

## g) Stopping distance:

- For Speed of 40km/hr Stopping Distance is,

$V^2=u^2+2as$

$S=(v^2-u^2)/2a$=speed^2/(2*deceleration)= (11.11*11.11)/(2*17.918) = **3.44 m**

But, the actual stopping distance depends upon the driver's reaction time, his applying force and hence for reaction time of 1sec, another 1 m of distance is covered by tire,

Hence the actual stopping distance will be= 3.44 +0.7=4.14m

The stopping distance may vary from driver to driver.

## h) Stopping time:

Theoretical:

We have t = v/a = 11.11 / 17.918 = 0.62sec

Where, v - Initial velocity

a - deceleration

Hence, t = 0.62sec

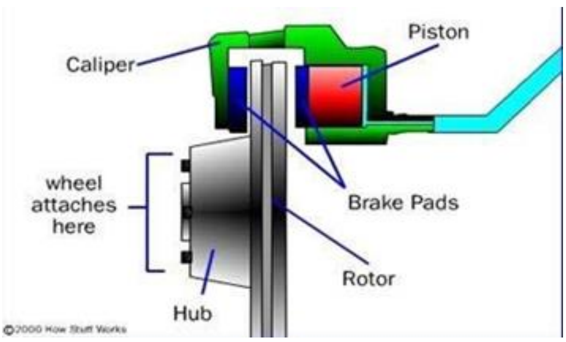
The above time is when there is no slip of tires, i.e. the coefficient of friction is 0.65 (dry earth).

The above stopping time is theoretical; the actual time depends upon the response time of the driver, the surface of road, tire grip etc.

As per, thinking time used in the Highway Code, an additional time of 0.6sec,

**t = 0.62+ 0.6= 1.22sec**

## Front Braking:




A brake is a mechanical device which inhibits motion, slowing or stopping a moving object or preventing its motion.

## Tandem master cylinder :

The tandem master cylinder transforms applied brake force into hydraulic pressure which is transferred to the wheel units through two separate circuits. This provides residual braking in the event of fluid loss, i.e. two separate braking mechanism installed in a single unit.

### a) Known and assumed Parameters:

Known Parameters:

1. Total weight of vehicle = 200kg

2. Radius of Tire = 0.292m

3. Weight distribution factor:

Front wheels = 0.55

Rear wheels = 0.45

4. Master cylinder diameter = 19.05mm

5. Caliper cylinder diameter = 25.5mm = 0.0255m

6. Rotor Disc Diameter = 200mm=0.2m=20Cm

Effective diameter of front disk = 0.18m

7. No of disc used = 2

Assumed Parameters:

1. Paddle Force = 150N

2. Paddle ratio = 5

## b) Calculating the actual braking force (Fr) and braking torque (Tr):

- Force transmitted to the piston of Tandem Master Cylinder= Paddle force* Paddle ratio = 150*5= 750 N
- Area of piston in master cylinder = π*0.0195*0.0195/4=2.986*10^-4m2
- Area of caliper piston= π *0.0255*0.0255/4= 5.107*10^-4 m2
- Pressure generated=force transmitted/area of master cylinder piston=750/(2.986*10^-4)= 25.11*10^5
- Force at caliper= pressure generated* area of caliper piston= (25.11*10^5*5.107*10^-4) = 1282.36N

- Braking force provided by caliper= force at caliper*coefficient of friction between caliper and disc = 1282.36*0.5 = 641.18 N
- Total braking force by both calipers on the disc =641.18*2= 1282.36N
- Total braking force acting on both discs (Fr) = 1282.36 * 2 =2564.73N
- Braking torque at disc= total braking force* distance from center of disc where the effective pressure acts= 2564.73*0.09=230.82
- **Tr (generated) = 230.82 Nm**

### c) Decelerations of vehicle:

- Decelerations of vehicle = Fr / Weight of vehicle = 2564.73/200=12.823m/sec$^2$

### d) Static load on wheels:

- Load on front wheel = weight distribution factor on front wheel * mass of vehicle *g= 0.55*200*9.81 = 1079.1 N
- Load on rear wheel = weight distribution factor on rear wheel * mass of vehicle*g = 0.45*200*9.81 = 882.9N

### e) Relative centre of gravity (X):

- X = Vertical distance of C.G from ground / wheel base = 13 / 42 = 0.309

### f) Dynamic axle load:

- Dynamic load transfer: (cg height/wheel base)*(mass of vehicle)*Decelerations

Dynamic load transfer: (13 / 42)*200*12.823= 793.8N

- Dynamic load on front axle= static load+ load transfer = 1079.1 + 793.8 = 1872.9N
- Dynamic load on front wheel= load on axle = 1872.9/2 = 936.45N
- Frictional force on front wheel= dynamic load*coefficient of friction between road and tires = 936.45*0.65 =608.69 N
- Braking torque required at front wheel= Frictional force on front wheel* rolling radius wheel = 608.69* 0.292 = 177.73Nm
- **As, the Tr (generated) = 230.82Nm >Tr (required) = 177.73Nm**

**Our design is safe.**

### g) Stopping distance:

- For Speed of 40km/hr Stopping Distance is,$V^2=u^2+2as$
  $S=(v^2-u^2)/2a$=speed^2/(2*deceleration)

  = (11.11*11.11) / (2*12.823) = 4.812m

But, the actual stopping distance depends upon the driver's reaction time, his applying force and hence for reaction time of .6sec, another 1m of distance is covered by tire,

Hence **the actual stopping distance will be=4.812+1=5.812m**

The stopping distance may vary from driver to driver.

### h) Stopping time:

Theoretical:We have t = v/a = 11.11 / 12.823 = 0.866sec

Where, v - Initial velocity a - deceleration

Hence, t = 0.866sec

The above time is when there is no slip of tires, i.e. the coefficient of friction is 0.65 . The above stopping time is theoretical; the actual time depends upon the response time of the driver, the surface of road, tire grip etc.

As per, thinking time used in the Highway Code, an additional time of 0.4sec, **t = 0.866+ 0.4= 1.26sec**

## DRIVE TRAIN-DESIGN:

**Objective-**

The drive train includes the engine, transmission, differential & drive shafts for transmitting the power to the wheels. We will be having a Rear wheel drive and the engine and the transmission both will be placed such that center of gravity of both of them lie more or less in the center.

## SHIFTER TRANSMISSION (MANUAL GEARBOX)

The shifter transmission has high output and power transfer. The power is more easily controlled on with the gear shifter. Therefore, we have decided to use the shifter transmission. We are using the stock shifter transmission that comes with engine LIFAN 167FMM.

## Calculations:

Transmission calculations:

Rolling resistance
F-rolling = c*weight of vehicle
= 0.050*230*9.81
=112.81 N
Where,
F-rolling- Frictional / Rolling resistance

c- Rolling resistance coefficient c = 0.050(car tire dry earth)
c= 0.03 for car tire on tar / asphalt
Trolling – Rolling Torque
R – Wheel Radius

Trolling = Fr * R
= 112.81 * 0.2794
= 31.5 Nm

Maximum peak Torque of Engine – 16 Nm

Maximum Torque at rear axle = Max peak torque of engine * efficiency of engine * sprocket ratio * Transmission efficiency
= 16 * 0.92 * 3.75 * 0.97
=53.54 Nm

## ENGINE SPECIFICATIONS:

The engine specifications below are specified as per the provider.

| | |
|---|---|
| Engine | LIfan 167 MN |
| Displacement | 229.00 cc |
| Bore X Stroke | 67 mm X 65 mm |
| Compression Ratio | 9:1 |
| Power | 15.5 bhp @ 7000 rpm |
| Torque | 16 Nm @ 6000 rpm |
| Idle Speed | 1420 rpm +/- 100 rpm |
| Starter | Electric & Kick |
| Cooling | Petrol |
| Fuel | Air Cooled |
| Lubrication | Forced & Wet Sump |
| Lube Oil | 10W30 MB oil |

## Maximum Speed of Vehicle:

Rated rpm provided by the throttle controller = rpm
Final Gear Ratio of vehicle = 3.75
Losses in transmission system = 0.03
Efficiency of engine = 0.92
Circumfrence of tire = 2* π * 0.275
= 1.72 m

Velocity of vehicle (considering zero drag) = rpm of engine *
circumfrence of tire * Final gear ratio
= 6000 * 0.92 * 1.5 / (3.75*3.33*0.96)
= 813 m/min
= 48.78 km/hr

### Calculations at different rpm:

1) Engine RPM : 1000 rpm
Wheel diameter: 0.558 m (22”)
Primary reduction = 3.33
Engine shaft rpm = 1000 / 3.33
= 300.30
Power shaft rpm = 300.30 / 3.75
=80.08 rpm
Speed (kmph) = power shaft rpm * π * wheel dia 8 *
60 / 1000
= 80.08 * 3.14 * 0.558 * 60 * 10^-3
= 8.4 kmph

2) Engine rpm : 1800 rpm
Wheel diameter: 0.558m (22”)
Primary reduction = 3.33

Engine shaft rpm = 1800 / 3.33
= 540.54
Power shaft rpm = 54.54 / 3.75
= 144.14 rpm
Speed = 80.08 * 3.14 * 0.558 * 60 * 10^-3
= 15.15 kmph

3) Engine rpm :
3200 rpm Wheel diameter: 0.558 m (22”)
Primary reduction = 3.33

Engine shaft rpm = 3200 / 3.33
= 960.96
Power shaft rpm = 960.96 / 3.75
= 256.25 rpm
Speed (kmph) = 256.25 * 3.14 * 0.558 * 60 * 10^-3

= 26.9 kmp

4) Engine rpm : 4500 rpm
Wheel diameter: 0.558 m (22")
Primary reduction = 3.33

Engine shaft rpm = 4500 / 3.33
= 1351.13
Power shaft rpm = 1351.13 / 3.75
= 360.36 rpm
Speed (kmph) = 360.36 * 3.14 * 0.558 * 60 * 10^-3
= 37.88 kmp

5) Engine rpm : 6000 rpm
Wheel diameter: 0.558 m (22")
Primary reduction = 3.33

Engine shaft rpm = 6000 / 3.33
= 1801.80
Power shaft rpm = 1801.80 / 3.75
= 480.4 rpm
Speed (kmph) = 480.4 * 3.14 * 0.558 * 60 * 10^-3
= 50.51 kmp

Torque excerted on the wheels:

The maximum engine torque = 16 Nm
Transmission efficiency = 90% = 0.9

Drive torque (Nm) = engine torque * combined gear ratio * transmission efficiency

Combined gear ratio = primary reduction * gear ratii * reduction at each engine gear ratio

Calculations:

1 Drive torque =16.33*3.75*2.769*0.9 =497.92 Nm
2 Drive torque =16.33*3.75*1.882*0.9 =338.42 Nm
3 Drive torque =16.33*3.75*1.450*0.9 =260.73 Nm
4 Drive torque =16.33*3.75*1.130*0.9 =203.1 Nm
5 Drive torque =16.33*3.75*0.960*0.9 =172.62 Nm

**Vehicle tractive effort / drive torque formula:**

Drive force = drive torque / Wheel radius

Calculations:

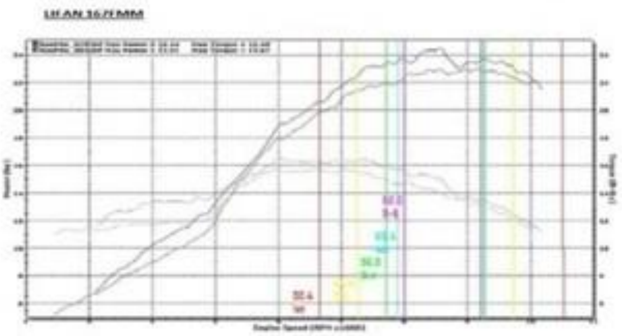
1) 497 / 0.279 = 1781.36 N
2) 338.42 / 0.279 = 1211.4 N
3) 260.73 / 0.279 = 934.5 N
4) 203.1 / 0.279 = 727.9 N
5) 172.62 / 0.279 = 618.70 N

Grade ability

Drive force = mg sin Θ  m = 230 kg  g = 9.81 $m/s^{-2}$  Θ = $sin^{-1}$ (Drive force / mg)

Calculation

1) Θ = $sin^{-1}$( 1781.36 / (230 * 9.81)) = 42□
2) Θ = $sin^{-1}$ ( 1211 / (230 * 9.81)) = 32□
3) Θ = $sin^{-1}$ ( 934.5 / (230 * 9.81)) = 24□
4) Θ = $sin^{-1}$ ( 727.9 / (230 * 9.81)) = 18.82□
5) Θ = $sin^{-1}$ ( 618 / (230 * 9.81)) = 15.89□



Engine Torque Curve

### d) Sprocket Calculations:

Pitch of chain = p = 14.8 mm
Centre distance = a = 584.2 mm

z1 = no. of teeth on driving sprocket = 12
z2 = no. of teeth on driven sprocket = 45

Sprocket ratio = 45 / 12 = 3.75

Velocity = v 13.52 m/s

(n1) = driving sprocket speed = 6000 / (3.33*0.96) rpm
= 1876 rpm
Chain speed = v= z1*p*n1 / 60000
= 12*14.8*6000/60000

=17.7m/s
Chain tension = p1 = 533.33 N
Chain width = b1 = 11mm roller
Diameter = d1 = 10mm

NO of links = Ln2 (a/p) + (z1+z2)/2 + (z2-z1/2
π)*(p/a) = 2(584.2/14.8) + (12+45)/2 + (45-12+2
π)*(14.8/584.2) = 109

Length of chain (L) = L*P = 109*14.8 = 1613.2mm = 1.6 m
Power to be transmitted (KW) = (2 πnT)/60000 = 10KW

From table,
Service factor = Ks = 1.4
Single chain K1 = 1

For 12 teeth (driving) = K2 = 0.76

KW rating of chain = (KW to be transmitted) * Ks / (K1*K2) KW rating of chain

= 10*1.4/1*0.76 = 18.42 KW

PCD of driving sprocket = D1 = P/sin(180/Z1)

=14.8/sin(180/12) = 57.18mm

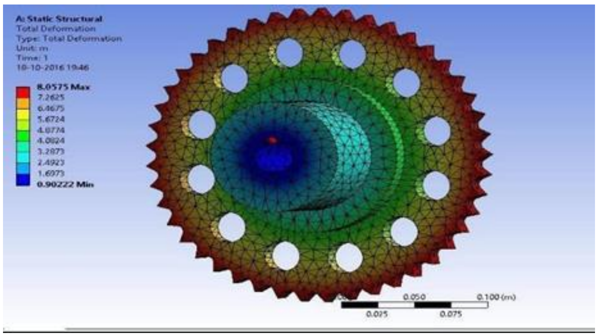
PCD of driven sprocket = D2 = P/sin(180/Z2)

=14.8/sin(180/45) = 212 mm

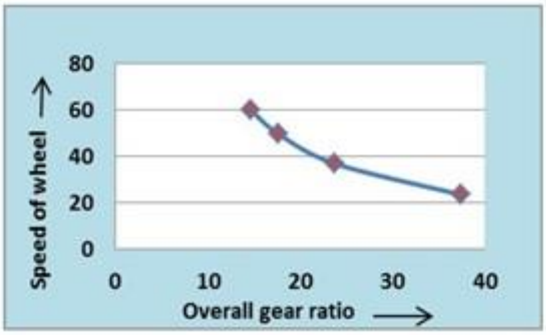
Do mean = D1+P(1-1.6/17)-D1

= 57.18+14.8(1-1.6/17)-10 = 70.58 mm

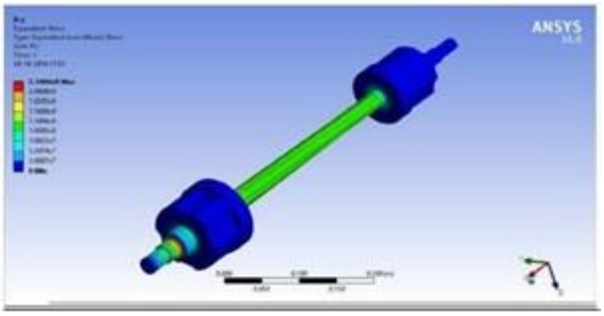
Tooth width = 0.90*b1 = 0.9.*11 = 8.2 mm



Sprocket FEA




Speed of Wheel to Gear Ratio




Drive Shaft FEA

TYRES AND RIMS

## Objective

Traction is one of the most important aspects of both steering and getting the power to the ground. The ideal tire has low weight and low internal forces. In addition, it must have strong traction on various surfaces and be capable of providing power while in puddles.

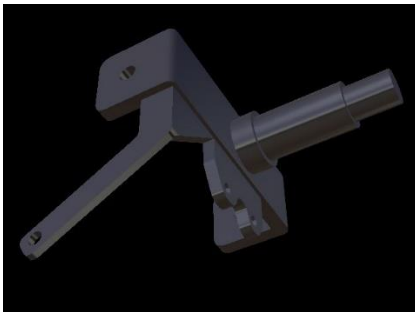
## TYRES & RIMS

We decided to use 4-Ply rating Carlisle & Far East tyres, tubeless tires and that have got specific thread pattern so as to provide a very strong and firm grip on all kinds of terrains After going through the calculations we have finalized the diameter of front tires to 23 inches and the diameter of rear tires to 22 inches which would help us to transmit maximum power. The dimension of Front tires is finalized as 23x08 inches and Rear tires 22x10. The Rims shall be made up of Aluminium to minimize un sprung weight. By reducing the width of the rim the inertia will be directly decreased and subsequently this will also reduce the overall weight. The diameter of Front rim is 12 inch whereas 10 inch at Rear rims.

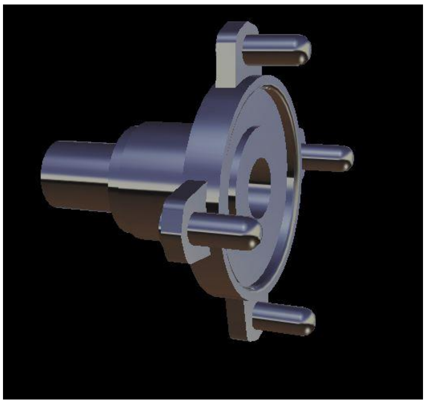
## WHEEL END

The wheel end is made up of the following parts - Rim, Hub, Disc, Milled bearing, and knuckle in sequence. Their compatibility with each other is a major design issue. The machined Hub and Knuckle designs that are made in Catia V5R20 of different ends are shown below.

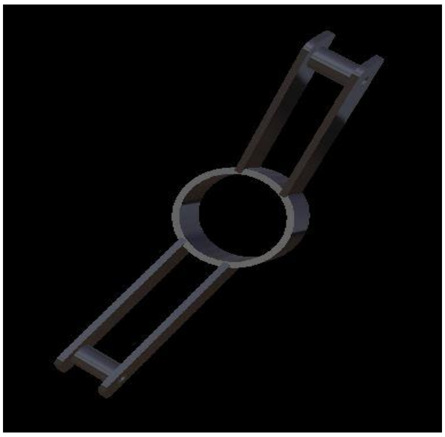
Front Wheel Hub



Front Wheel Knuckle



Rear Wheel Hub



Rear Wheel Knuckle

**<u>Tire Specifications</u>**:

| | |
|---|---|
| Front | 23x8-12 |
| Rear | 22x10-10 |

## <u>ELECTRICAL SYSTEM:</u>

### <u>Objective</u>

The electronic system for the car Quad Bike was designed to fulfil two key purposes. First, the electronics system supports. The mandatory safety equipment, specifically the brake light and the kill switch circuit

### <u>COMPONENTS OF ELECTRICAL SYSTEM</u>

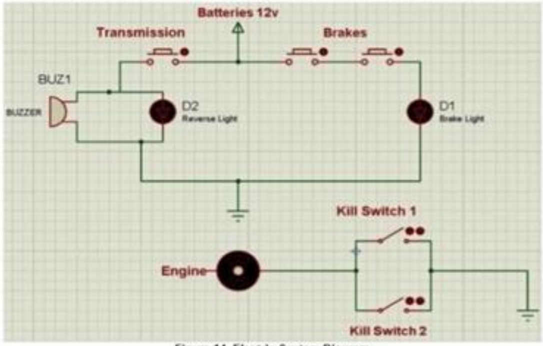



Figure 14. Electric System Diagram

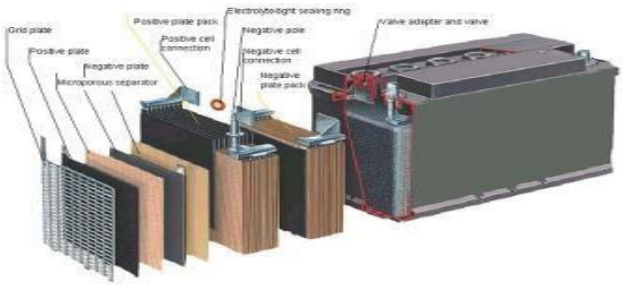
### <u>1. Kill Switch</u>

These kill switches are able to cut off all the Electrical connections including ignition system and are rigidly mounted near the steering handle where the driver can easily control it. Second kill switch is placed on left side of vehicle near driver seat.

### <u>2. Battery</u>

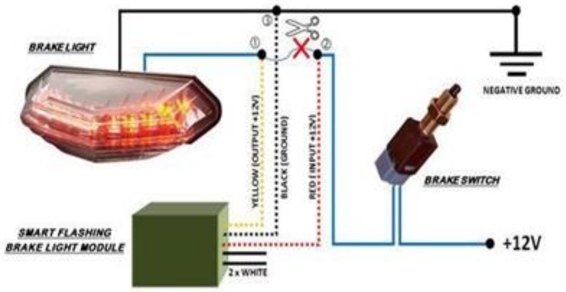
A maintenance free lead-acid storage battery is an electrochemical device that produces voltage and delivers electric current.

SPECIFACTONS: *9V 6Ah.*




### <u>3. Brake Light</u>

The brake light is installed at the back of our vehicle red in color which is clearly visible from the rear, and is visible in very bright sunlight, and is mounted between the wheel centerline to serve the purpose of indicating that the driver has applied brake and the vehicle tends to stop.




4. <u>Electric Start</u>

The modern starter motor is either a permanent-magnet or a series- or series-parallel wound direct current electric motor with a solenoid switch (similar to a relay) mounted on it. When current from the starting battery is applied to the solenoid, usually through a key-operated switch, it pushes out the drive pinion on the starter driveshaft and meshes the pinion with the ring gear on the flywheel of the engine.

## INNOVATION:

## DRIVER'S SAFETY & ERGONOMICS:

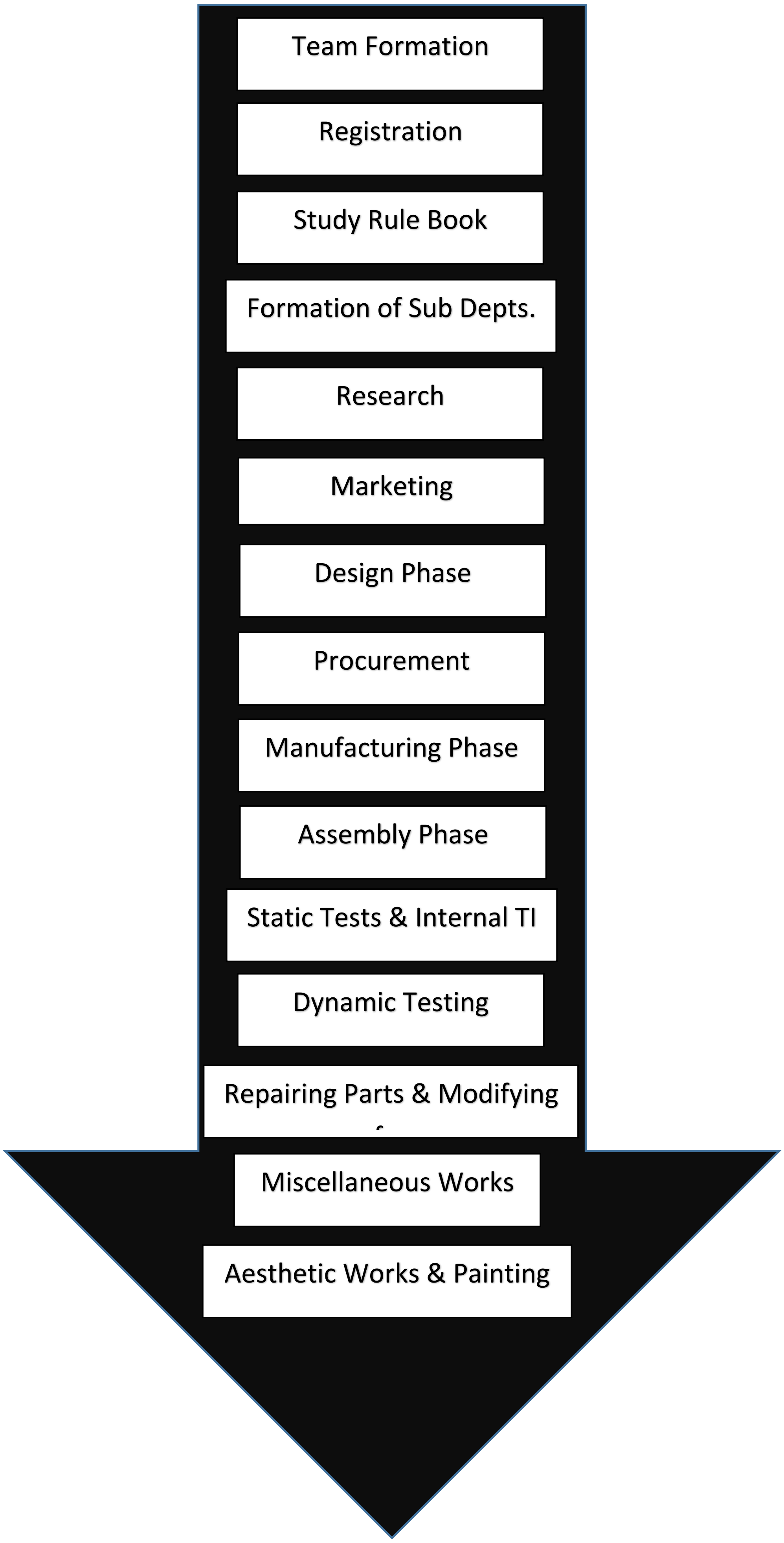
Driver's safety is the most important concern for our ATV. For better perspective we have made a 1:1 PVC model of the roll cage and further improvised our design in CATIA V5R20 according to driver's ergonomics. For the comfort and well-being of the driver, the use of standard helmet, goggle, driver suit, gloves, neck brace, shoes & fire safety equipment will be used to ensure driver safety. For the rugged, up and down track the vehicle will be provided with a hitch point bumper with spring support installed in the front of the vehicle to absorb energy from collision. Two Fire extinguisher and two kill switches specified in the rulebook will also be used for the case of emergency. Ergonomics include the foam padding of the front, rear and side body panels, gear shifting indicator, turn light indicators, standard rear view mirrors and such other things.

# Anthropometry Data

Anthropometry is a branch of anthropology concerned with comparative measurements of the human body and it's parts as well as the variables which impact these measurements, often presented in tabular format or annotated diagrams of human figures. The primary dimensions of bones, muscles, and adipose tissue. This data is used in human factors/ergonomics in order to ensure that designs and standards and realistic.

| Dimension Number | Dimension | Description |
|---|---|---|
| 1 | Age (year) | |
| 2 | Weight (kg) | Total mass (weight) of the body |
| 3 | Stature | Vertical distance from the floor to the highest point of the head (vertex). |
| 4 | Shoulder (biacromial) breadth | Distance along a straight line from acromion to acromion |
| 5 | Hip Breadth, sitting | Breadth of the body measured across the widest portion of the hips |
| 6 | Shoulder height, sitting | Vertical distance from a horizontal sitting surface to the acromion |
| 7 | Elbow height, sitting | Vertical distance from a horizontal sitting surface to the lowest bony point of the elbow when it is bent at a right angle with the forearm horizontal |
| 8 | Buttock-popliteal length (seat depth) | Horizontal distance from the hollow of the knee to the rearmost point of the buttock |
| 9 | Lower leg length (popliteal height) | Vertical distance from the footrest surface to the lower surface of the thigh immediately behind the knee, bent at right angles |
| 10 | Upper hip bone height, sitting | Distance from floor to the uppermost point of the left hipbone. The hipbone is traced by palpating [11, 16]. |
| 11 | Lowest rib bone height, sitting | Distance from floor to the bottom of the lowest left rib. The lowest left rib is traced by palpating [11, 16]. |

**Project Planning:**



## CONCLUSION

The objective of designing a single-passenger off-road race vehicle with high safety and low production costs seems to be accomplished. The design is first conceptualized based on personal experiences and intuition. Engineering principle and design processes are then used to verify and create a vehicle with optimal performance, safety, manufacturability, and ergonomics. The design process included using Solid Works, CATIA and ANSYS software packages to model, simulate, and assist in the analysis of the completed vehicle. After initial testing it will be seen that our design should improve the design and durability of all the systems on the car and make any necessary changes up until the leaves for the competition.

## REFERENCES

1) John C. Dixon; Suspension analysis and computation geometry; ISBN: 978-0-470-51021-6; October 2009

2) Thomas D. Gillespie; Fundamental of Vehicle Dynamics; ISBN: 978-1-56091-199-9; February 1992

3) Race car vehicle dynamics – William F Miliken

4) Fundamental of vehicle dynamics – Gillespie

5) Front axle and steering in automobile engineering –

6) Dr. Kripal Singh