Design, Analysis and Manufacturing of Efficycle-Human Powered Light Weight Hybrid Tricycle

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Abstract— Any innovation or technology cannot be possible until and unless the work is not done by a team. In such a manner members of Team STALLION (participants of SAE-NIS 2019), who are highly motivated individuals are recklessly working towards the realization of this goal since its very inception. This paper provides in detail, about the design considerations, and methodology used in designing and developing it. The term Efficycle stands for what we called all efficient cycle or hybrid cycle. It is the special kind of cycle moves with higher efficiency than the normal bicycle which has the maximum 60% efficiency & increase in magnitude is almost impossible. In regard to the recent surge of development in the automotive industry, and the growing need for alternative energy source for mobility in day to day scenario, this project carried aims at providing an energy efficient human powered three wheel electric vehicles capable of carrying two passengers. All the features like drive train, brakes, steering, and frame structure has been designed to comply with the requirements of the people.

Keywords: SAE, Efficycle, Hybrid Vehicle, Electric drive, Green Technology, Tri-Wheeler

I. Introduction

We Approached our design by considering all possible alternative for a system and modeling them in CAD software like CATIA, CREO etc. And subjected to analysis using HYPERMESH, ANSYS FEA software, ANSYS WORKBENCH. Based on analysis result model was modified and rested, the final result was frozen. The design procedure of the model iterative and based on various engineering and reverse engineering processes depending upon factors like availability, cost etc. So the design process focus on following objectives-Safety, Service Ability, Strength, Standardization, Cost, Driving feel, Ergonomics and Aesthetics. The design objectives set out to be achieved where 3 simple goals applied to every component of EFFICYCLE – Durable, Light weight and High performance, to optimize the design by avoiding over designing which will also reduce.There is increasing concern over congestion and pollution associated with the use of motor vehicle for personal transport. This project aims to design a vehicle that could be viable alternative to cars for short distance journeys. The designed vehicle is an Light Vehicle. It is powered by a hybrid human-electric drive system. Different technologies were analyzed and based on the research factors such as turning radius, stability, handling, and ease of maneuvering.

II. Literature Review:

A literature survey before the design phase, allowed us to determine the basic raw material required. The dimensional tolerance and the process for manufacturing in order to optimize the manufacturing cost and make it commercially viable, the concept of DFM [design for manufacturing] was utilized. The FMEA [failure mode and effect analysis] enabled us to fix the potential problems in the design phase itself. Hence a commercially feasible and production ready vehicle was generated.

III. Selection and Design of Subsystem

A. *Drive Train – Human*

Human power drive train consists of a parallel transmission of both drivers combined at the idler shaft transmitting it to the driven wheel. The power of the both drivers add up when at the idler shaft if they drive at the same speed. In order to get individual power of both the drivers separately they are provide with free wheel. Crank Length and number of gear teeth are designed on the basis of market availability and vehicle dynamics approach such as required speed, acceleration, grade ability, etc.

- Human power drive train consist of a parallel transmission of both drivers combined at the idler shaft before transmitting it to the main drive driven wheel.
- The power of the both drivers add up when at the idler shaft if they drive at same speed.
- For finding out the velocity in this drive train we assume ideal condition.

Force applied by one driver on pedal
$F = 300N$
Crank Length = $R = 0.1651m$
$P_{max} = T\omega$
$F * R * \omega = 300$
$= 300 * 0.1651 * \omega$
$\omega = 8.47$ rad/sec
$V = \omega * r/G$
r= radius of tyre
G = Gear ratio
$V = 8.47 * 0.97 * 0.2794 / G1G2$
$G1 = 24/44$
$G2 = x/28$.

B. *Drive Train – Electric*

- The Motor Drive Train consists of 600W BLDC motor.
- For finding out the velocity in this drive train we assume ideal condition.

Technical specification:
1) POWER: Rated 600W, peak 1400W
2) Rated RPM: 500
3) Torque: rated 10Nm, peak 400% of rated power for 10 second.
4) Voltage: 48V
5) Current: Rated 13A, peak 32A
6) Efficiency: approx. 80% on full load & full rpm.
7) Protection: IP 33
8) Dimension: 223mm * 171mm* 146mm

9) Shaft diameter: sprocket 60mm, shaft 12mm
10) Weight: controller 0.55kg, motor 5kg

$V = \omega * R/G = (2\pi N/60) * (R/G)$

$G = G_1 * G_2$

$\tau_1 = 10Nm$

$G_1 = 28/14$

$G_2 = x/28$

$\tau = \tau_1 * G_1 * G_2$

Gear	Velocity	Torque
1	25.5	20
2	29.8	17.14
3	34.05	15
4	39.7	12.85
5	44.70	11.42
6	51.08	10

C. Drive Train- Hybrid

For obtaining better performance in endurance we use hybrid drive train.

- The power of the both drivers add up when at the idler shaft if they drive at same speed.
- For finding out the velocity in this drive train we assume ideal condition.

τ max = (2 * τ human power) + τ motor
= (2 * 19.216) + 23.57
= 62.002 N m

Vmax = maximum value between motor and human power
= 44.126 kmph

Rolling resistance = µmg
= 0.003 * 235 * 9.81 =96.91 N

Frictional resistance = µmg
= 0.4 * 235 * 9.81
= 45922 N

Drag resistance = ½ CD ρ * A (V)2
= ½ * 0.88 * 1.1 * 1.225 *(11.11)2
= 73.18 N

For finding acceleration (A) of hybrid drive train system

Amax = (tractive force – (rolling resistance + frictional resistance + drag resistance)) / mass of vehicle
= (1118.533 – (6.91 + 922 + 73.18)) /235
= 0.8084 m/s2

Gear	Velocity	Torque
1	25.5	47.01
2	29.8	40.29
3	34.05	35.26
4	39.7	30.21
5	44.70	26.85
6	51.08	23.5

D. HUB:

Internal diameter and splines are designed as per shaft and external diameter according to free wheel available

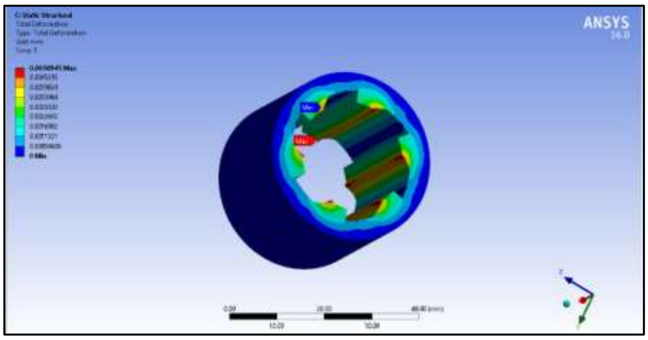

Fig. 1: Hub

1) Sprockets calculations

46 teeth

Module (m) = D/T
= 4 mm
Addendum = 1m
Dedendum = 1.57m

PCD = p/sin(180/Z)

Where,
p = pitch of sprocket
Z = no. of teeth

NO. OF TEETH	46	28	24	20
PCD (mm)	186.101	113.42	97.29	81.18
MODULE (mm)	4	4	4	4
ADDENDUM (mm) (1*m)	4	4	4	4
DEDDENDUM (mm) (1.157*m)	4.628	4.628	4.628	4.628
TOOTH THICKNESS (mm) (1.157*m)	4.628	4.628	4.628	4.628

E. Chain calculations:

Length of chain = L_n * P

Where,
L_n = no. of links

$L_n = 2 (a/p) + ((Z_1 + Z_2)/2) + ((Z_2 – Z_1)/2\pi)^2 * (p/a)$

NO. OF TEETH	NO. OF LNKS
46-29	174
14-33	644
28-14	110

F. Shaft calculations:

For shaft material we are using EN 19.

yield strength = 480 MPa

$\tau = 16/\pi d^3 * ((M_b * K_b)^2 + (M_t * K_t)^2)^{1/2}$

$= 16/\pi d^3 * (247976.266 * 1.4)^2 + (62 * 1.4)^2)^{1/2}$

$\tau = 05. S_{yt}/FOS$
= 754.62 MPa

d= 23 mm

let A and F = bearing
B, C, D and E = sprocket

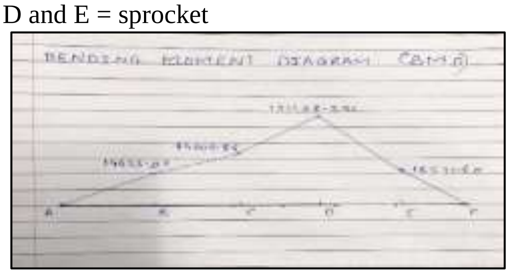

Fig. 2: BMD figure

Fig. 3: CAD model of Shaft

IV. STEERING

A. Objective:

The steering is designed to withstand the stress of safety maneuvering the vehicle through any type of possible condition at the tie of driving. The purpose of the steering system is to provide directional control of the vehicle with minimum input. The main thing about steering is that it should be according to Ackermann condition of correct steering.

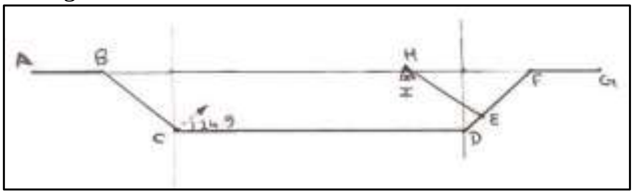

Fig. 4: Working mechanism

While designing, the main thing is that to keep in mind is simplicity of the system. Therefore system should be light in weight, less play, should give a quick response to the driver, and better-running feedback. Thus we are using Ackermann geometry.

Formulae that we used for calculation of various steering parameters are:

$\alpha = 29$

$\beta = 49.14$

C = 39”

where,

c = distance between KP to KP

α = front outer turning angle

β = front inner turning angle

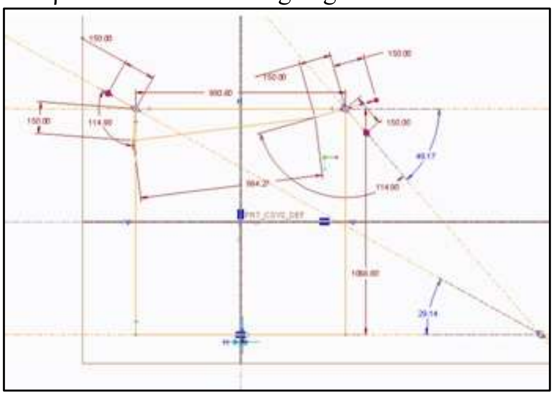

Fig. 4: Steering simulation

Now calculating the inner front wheel turning radius (R1) which is given by,

R_1 = (wheel base / sin β) – (track width – c/2)

= (1066.8/sin49.14) – (1117.6 – 990.6/2)

R_1 = 788.23 mm

Similarly, for front outer turning radius (R_2)

R_2 = (wheel base / sin α) + (track width – c/2)

= (1066.8/sin29) + (1117.6 – 990.6/2)

R_2 = 2822.75 mm

Calculating the rear wheel turning radius,

R_3 = (wheel base/tan β) – (track width – c/2) + (c/2)

Where,

R_3 = rear wheel turning radius

= (1066.8/tan49.14) – (1117.6 – 990.6/2) + (990.6/2)

R_3 = 795.78 mm

So mean Turning radius is given by,

$R_{mean} = (R_1 + R_2 + R_3)/3$

R_{mean} = 1468.9 mm = 1.47 m

β for % Ackermann

= $\tan^{-1}$ (wheel base / ((wheel base / tanα) – c))

= $\tan^{-1}$ (1066.8/ ((1066.8 / tan 29) – 990.6))

= 49.14˚

Ackermann = β/ (β for 100% Ackermann)

= (49.14 / 49.14) * 100

= 100% (theoretically)

PARAMETER	VALUE
Mass	44 Kg
Turning radius	1.468 m
α	29
β	49.14
KPI	3˚
Steering wheel diameter	400 mm
Tie rod length(1)	863.6 mm
Tie rod length(2)	228.6 mm
Castor	3˚
Pitman length (1)	101.6 mm (4”)
Track width	1117.6 mm (44”)
Wheel base	1066.8 mm (42”)
C.G height	533.4 mm (21”)
Velocity	20 Kmph
μ	0.7
Distance between KP	990.6 mm (39”)
Steering ratio	1.22:1
Scrub radius	138.04 mm

B. Steering effort calculation

F_{static} = mass * μg

= 302.146 N

Load transfer = (mv^2 / r * ht. of C.G)/ track width

= (44*$(5.55)^2$ /1.47 * 0.5334) / 1.0668

= 460.98 N

Torque at left kingpin (Ʈ) = F_{static} * scrub radius

= 302.146 * 107.83

= 32580.4 N mm

Castor (3˚) = cos 49.14*sin3 * $F_{load\ transfer}$

= 15.78

Scrub radius = 302.146*cos 93* sin49.14

= - 11.96

Total torque at kingpin (B) = Ʈ + castor+ scrub radius

= 32584.22 N

Perpendicular force at c (F_C) = Ʈ$_t$ / steering arm

= 217.22 N

Force on tie rod (F_{cd}) = F_c cos 24.9

F_d = 197.03 N

Torque at kingpin (Ʈ$_e$) = F_{cd} * cos24.9 * steering arm

= 26807.97 N mm

Force on steering arm (F_{eg}) = Ʈ$_e$ / steering arm

= 178.71 N

Force on pitman = F_{eg}* sin 24.9

= 75.24 N

Cornering force = 460.98 N

Total force at point H (F_h) = Force on pitman + cornering force

= 75.24 + 460.98

= 536.23 N

Force on steering = (F_h * pitman dist.) / diameter of steering wheel

= (536.23 * 101.6) / 400

= 136.20 N

Force per hand = 68.1 N

= 6.94 Kg

Component	Material	Yield Stress	Max. Stress	FOS (Factor Of Safety)
KNUCKLE	MS	370	103.84	>1
PITMAN	MS	370	137.63	>1
SHAFT	MS	290	49.973	>1
TIE ROD	Al 7075 T6	260	109.93	> 1.5

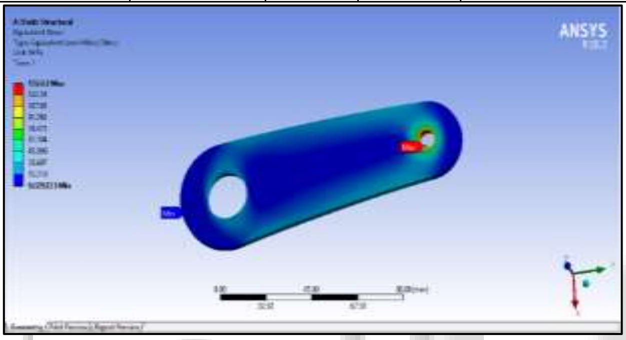

Fig. 4: Analysis of pitman (stress)

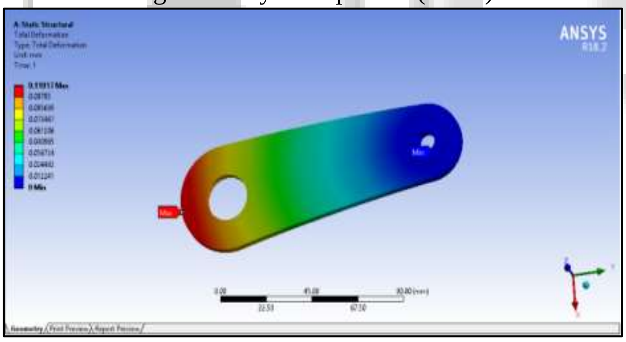

Fig. 5: Analysis of pitman (deformation)

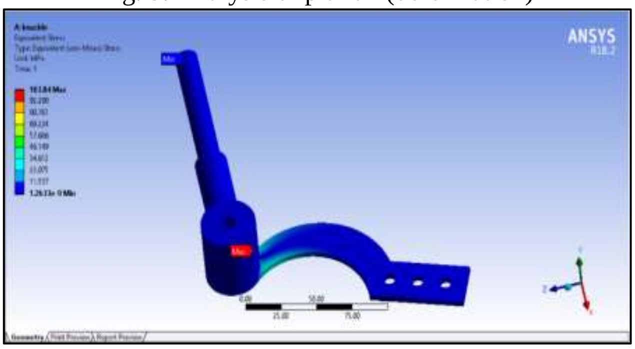

Fig. 6: Stress Analysis of knuckle

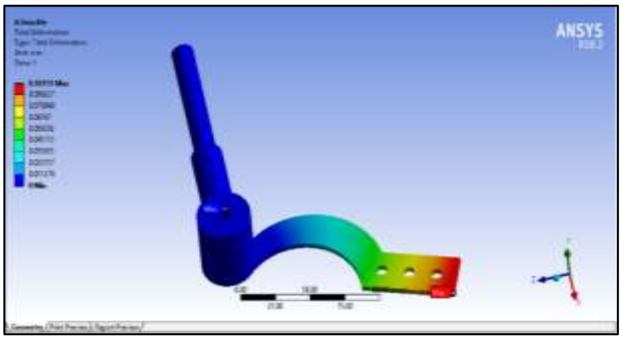

Fig. 7: Deformation of Knuckle

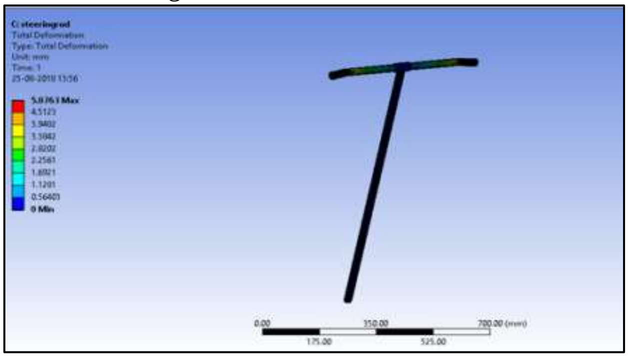

Fig. 8: Steering handle

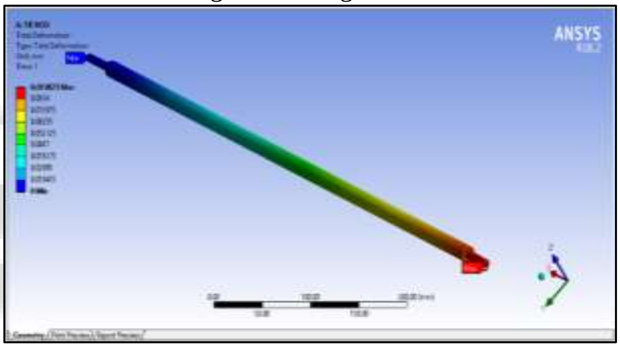

Fig. 9: Tierod

V. WHEELS & TYRES

- The wheels and tyre hat we are using are having following specifications:
- Front wheel size:
  Radius = 10”
- Rear wheel size:
  Radius = 10”
- Front tyre width = 1.5”
- Rear tyre width = 1.5”
- Larger wheels have less rolling resistance.
- They have greater leverage and less deformation.
- Smaller wheels involve higher chain and hub friction losses.

VI. BRAKING SYSTEM

The purpose of brake is to slow down the vehicle or stop the vehicle when required. For better performance the brakes are designed to lock all the wheels. We use Mechanical Disc brakes which are mounted on all three wheels. Disc Brakes are Used because of its high heat dissipation capability, torque transmitting capacity, good efficient in wet weather conditions (because centrifugal force tends to flings water off the brakes disc and keeps it dry).

Parameter	Value
Muscular force	50N
Pedal ratio	3:1
OD of disc	140mm
ID of disc	104mm
Mass of vehicle	230kg

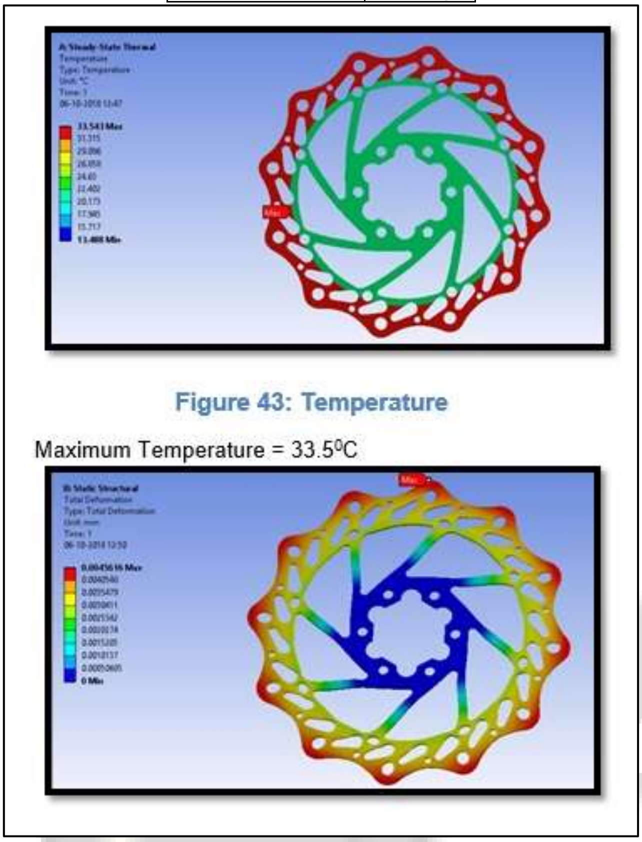

Fig. 10: Analysis of Disc

A. DISC

Teams allow to cover distance of 50m in 50 sec
Initial velocity of efficycle (U)
U = dist (s)/time (t)
= 50/15
=3.33m/s
Braking distance(S)= 5 m
Final velocity of efficycle after braking (V) = 0m/s
By using kinematic equation of motion,
$V^2 – U^2 = 2aS$
$0 – 3.33^2= 2*aS$
$a = -1.10\ m/s^2$ (deceleration)

1) Case 1:

Required braking torque to stop efficycle
Braking force (BF) =
BF = mass (m) * deceleration (a)
= 220*1.10
= 242 N
Required breaking torque(BT)=
BT = BF* (R/r_p)
Where,
R is static laden radius of tire
r_p is the ratio of radius of brake disc to the static laden radius of the wheel
r_p = Radius of brake disc / R
= 70mm / 254 mm
= 0.2755
Now,
BT = 240N * (0.07m/0.2755)
= 44.67 Nm (Required Torque)
Actual braking force applied due to brake pads on tire
BF = μ * load on pad
Where,
μ = coff. Of friction
= 267 N (applied force)
Actual breaking torque applied on tire

BT = BF * (R/r_p)
=49.81 Nm (Applied torque)

VII. SEATS

This includes seatback angle of 26 degree. The seat includes seat rest and supports entire torso and thighs. The seating space height for both the driver is 37inch. We are using in house fabricated seat which is reclined seat for overall weight reduction and the manufacturing process include tapering a free end of a cylindrical tube so as to form a bent end; forming the bent end of cylindrical tube so as to form a seat support; and drilling a hole in the seat support. The specifications of our design is based on an average human's posture with the sole aim of providing a cozy ride.

VIII. SAFETY FEATURES OF VEHICLE

In order to ensure the complete safety of the drivers, many safety features were installed in the vehicle. Some of them are,

- Helmets and knee pads
- 3point seat belt
- SPM is provided for the protection from the side impact.
- Kill Switches

IX. ERGONOMICS & COMFORT FEATURES

The vehicle is designed by considering ergonomics

- Seat reclined at 26 degree for an optimum ease in ride and proper visibility.
- Easy steering handle provided.
- Easy entrance and exit from vehicle.

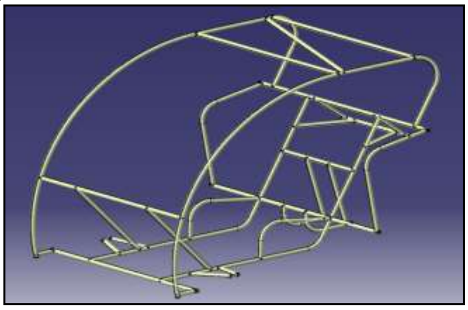

Fig. 11: Front View

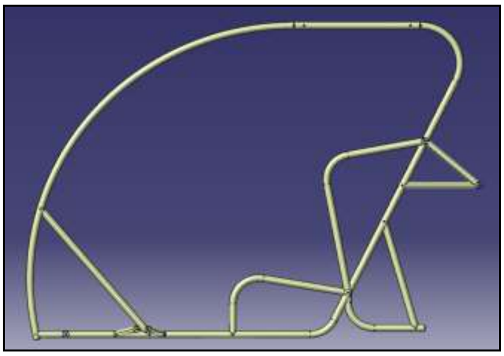

Fig. 12: Side View

X. CAE ANALYSIS OF FRAME

A frame of the vehicle plays the most important role in safety of passenger. The frame contains the operator, motor, brake system, steering mechanism and must be of adequate strength to protect the operator in the event of impact or rollover. Finite Element Analysis is a mathematical modeling technique used to determine the response of real structures to external and internal loads. CAE Analysis on the frame is performed to evaluate the safety offered by the frame to drivers in case of any accident.

A. Frontal Impact Analysis

It is the impact wherein there is a possibility of vehicle crashing into another vehicle head on during the race.

Material-1 (EN15, Circular, Outer diameter =25.4, wall thickness= 2 mm)

1) Assumption & Considerations:

Ansys Workbench 16.2 is used to create 2-D meshing of the frame. For boundary conditions for front impact test, the frame is fixed from all tyre position and on rear side and the front members will come across the applied load. To approximate the worst-case scenario that the vehicle will undergo on the track, the impact force was considered as 2G.

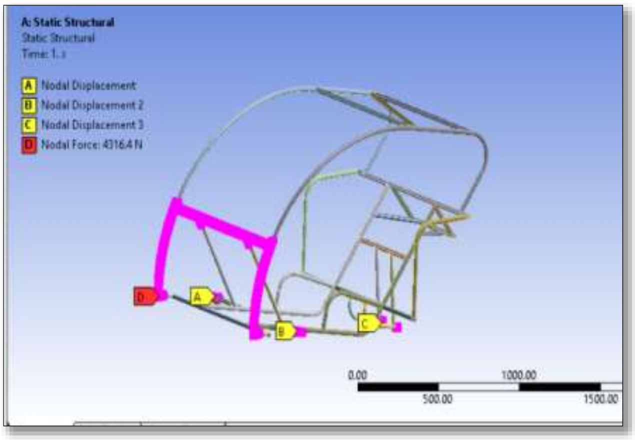

Fig. 13: Boundary condition

2) Calculation of Impact Forces:

Weight of the vehicle, M= 220kg (Including two drivers.)

Impact force F = 2*G N

Therefore, G = M*g = 220 * 9.81

G =2158.2 N

F = 3*G = 2* 2158.2

F = 4316.4 N

3) Analysis Results:

Maximum stress = 593.75MPa

Minimum stress = 0 MPa

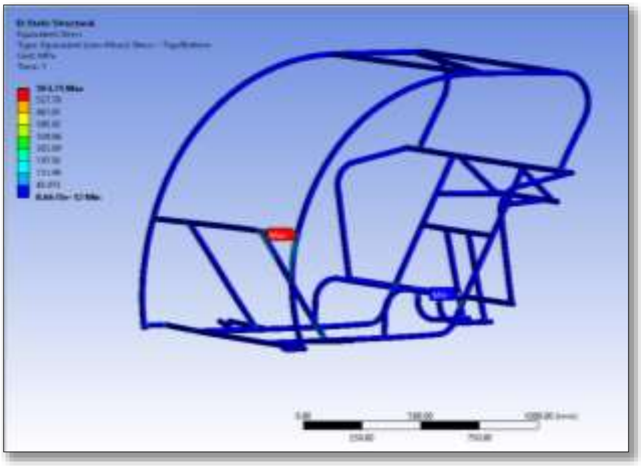

Fig. 14: Stress Analysis

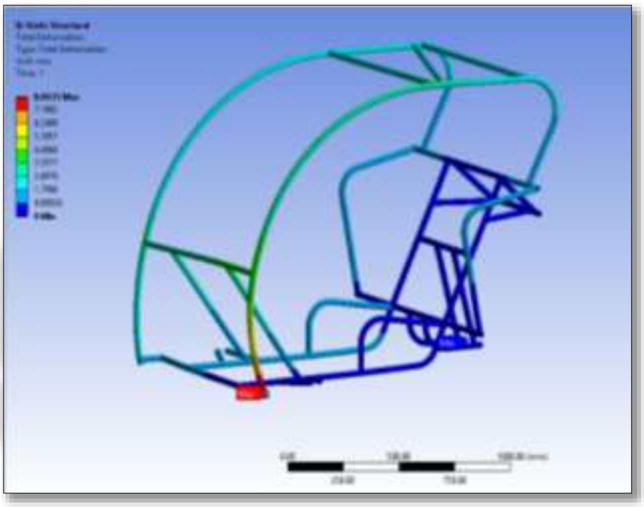

Fig. 15: Deformation

B. REAR IMPACT ANALYSIS

It is the impact wherein there is a possibility of vehicle crashing into other vehicles rear(back) on during the race.

1) Material-1 (EN15, Circular, Outer diameter =25.4-inch, wall thickness= 2 mm)

a) Assumption & Consideration: Ansys workbench 16 is used to create 2-D meshing of the frame. For boundary conditions for Rear impact test, the frame is fixed from front side and front tyre position and the load is acting on the nearest part of the chassis (rear tyre mountings). To approximate the worst-case scenario that the vehicle will undergo on the track, the impact force was considered as 2G.

Fig. 16: Boundary condition

b) Calculation of Impact Forces:

Weight of the vehicle, M= 220kg (Including two drivers.)

Impact force F = 2*G N

Therefore, G = M*g = 220 * 9.81

G =2158.2 N

F = 2*G = 2* 2158.2

F =4316.4 N

c) Analysis Results:

Maximum stress = 425.83 MPa

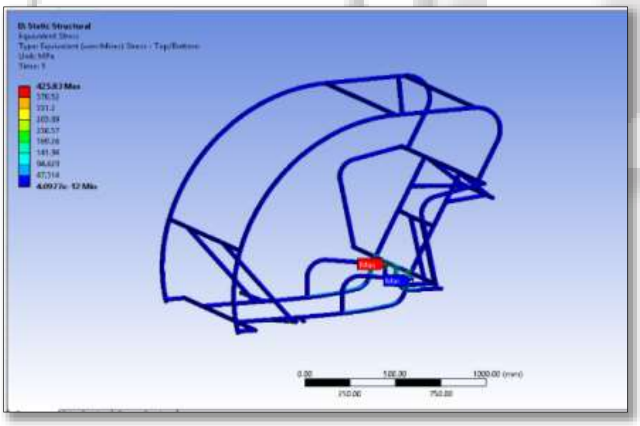

Fig. 17: Analysis (stress)

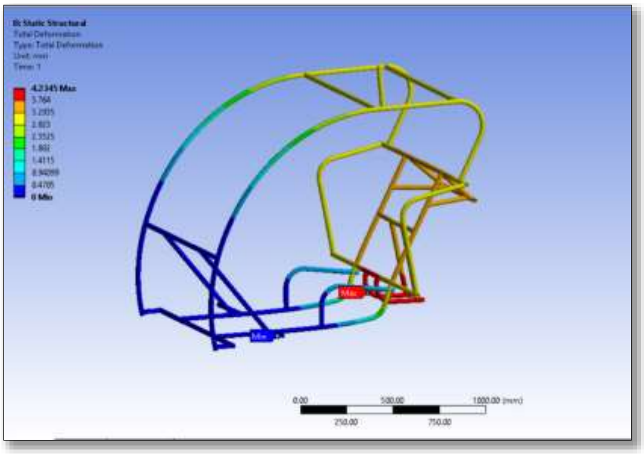

Fig. 17: Analysis (deformation)

FOS = Yield strength / Max working stress

= 805 / 425.83

= 1.89

XI. CONCLUSION:

Pollution being the emerging issue all over the world, there should be an alternative for transport. Efficycle could be the appropriate answer for the issues regarding the same. Design & Analysis becomes easier with the use of software like CREO PARAMETRIC, CATIA, and ANSYS. It is an eco-friendly human powered vehicle with an electric drive system. Such a lightweight tricycles can be modified and can be used for transporting good and can even be used for disaster management. India being a load of multi lingual and multicultural festivals, such tricks becomes a useful element for carrying out cleaning and sanitations activities in public places during such festival.

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