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Design and Analysis of a Shifter-Kart

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Abstract: *The paper deals with the overall designing of a go kart from scratch to CAD modelling it on SolidWorks 2016 and analyzing of the components to be fabricated Ansys18.1. It also tells the detail of the selection of the components which are going to be put in the Go-Kart. The paper initiates with the laying down of ideas of how the kart needs to be and then design methodology is discussed on how we are planning to formulate and structure it. The chassis is designed keeping in mind the constraints, components to be placed and optimal strength to weight ratio. Accordingly steering is designed for minimal turning radius and stability at corners and less steering torque. Then braking system design calculates the brake force or braking torque precisely required to stop the car in motion without skidding or turning. The brake pedal and Brake disc calculation determining the dimensions is also shown. Powertrain being the mitochondria of the kart is meticulously examined and selected and drivetrain being the muscle of the kart is designed. Formulation and determination of torque and sprockets required, also the design of axle and wheel hub for fabrication is discussed. Towards the end of the paper there is detail discussion of some additional features like Ride Height Adjustment and Sliding Seat mounts are incorporated in the CAD model to enhance dynamics and ergonomics of the kart and a complete fabrication-ready design of the kart is modelled both on SolidWorks and on paper-calculations.*

Keywords *Analysis, Ansys18.1, CAD, Calculation, Design, Dynamics, Go-kart, SolidWorks, Side-mount*

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I. Introduction

Racing is an enormously complicated activity at the higher level of the sport and significantly so at any level. At the very heart of this activity is the problem of achieving a performance from the driver-vehicle entity which, in the particular race environment, exceeds the competition. This is the challenge. It is the dynamic behaviour of the combination of high tech machines and infinitely complex human beings that makes the sport so intriguing for participants and spectators alike. As vitally important as the driver, this paper concentrates on the vehicle components which can be modified to enhance performance and facilitate driver control.

II. Literature Review

The chassis is made up of steel tubes and the main condition of a good kart chassis is that it needs to be light weight and be able to flex and twist. Therefore, before making a chassis, a lot of thought went into its design and the factors influenced in order to handle properly either on the straight or a corner. Many of us will think that the structure of a car is more complicated compared to a go-cart. In fact, it is perhaps a more difficult task to explain a go-kart than an equivalent car. The differences are the kart's lack of differential, and also its lack of suspension components. Thus, the Kart chassis is playing an important role to work as a suspension component. That is why a cart chassis needs to be flexible enough not to break or give way on a turn. The stiffness of the chassis enables different handling characteristics for different circumstances. Typically, for dry conditions a stiffer chassis is preferable, while in wet or other poor traction conditions, a more flexible chassis may work better. Best chassis allow for stiffening bars at the rear, front and side to be added or removed according to race conditions.

III. Kart design

The chassis has been outlined by taking variables like dimensional limits (width, height, length and weight), operational limitations, and administrative issues, legally binding prerequisites, financial constraints and human ergonomics as a need.

- Frame being the biggest and bulkiest, the constituent members should be weight optimized.
- The strength to weight ratio is expected to be high.

- The weight of the kart should be balanced, since we are not using differential, the COG should lie on the center line of the kart towards the rear axle.
- Front to rear weight ratio should be 40:60 and left to right to be 50:50
- The ground clearance should be more than 1.65 inch.
- Adjustable ride height and sliding seat.
- Omitting the use of differential.
- Side mounting the engine.

3.1 Design methodology

- The wheel base and track width were finalized for the vehicle.
- Extra members were introduced in the chassis for the mountings. But keeping in mind about the weight factor.
- Components were placed in accordance with the weight balance of the vehicle.
- Ground clearance was taken in account.
- Engine position was finalized for optimum weight balance.
- Ergonomics of the vehicle was kept in mind. Pencil sketches were made and design is checked and changes were made for driver's comfort.
- C bracket members were welded in the frame for the sliding seat and brackets were welded for ride height adjustment.
- Side mounting the engine to make the kart compact.
- After designing the frame, bumpers were designed
- Analysis was done to check the impact resistance of the frame and to determine the factor of safety.
- After approximate weight and acceleration of the kart was known, cross-section of chassis pipes was determined by using bending moment formula.

3.2 Material availability

AISI 1018 has excellent weld ability and produces a uniform and harder case and it is considered as the best steel for carburizing parts. The 1018 carbon steel offers a good balance of toughness, strengthened ductility. Considering the above factors, AISI 1018 was chosen for our chassis material.

Table 1 Physical properties of AISI 1018

PROPERTIES	VALUE (Metric)
Density	7.87g/cc
Yield tensile strength	370MPa
Elongation at break(in 50 mm)	15%
Poisons ratio	0.29
Modulus of elasticity	200GPa

Table 2 Frame and the pipe used

Dimension of pipes	• 1 inch diameter and 1.50mm thickness for frame. • 1 inch diameter and 1.75mm thickness for front, rear and side bumpers.
Mass of frame	14.249Kg
Welding type	Electric arc welding
Length of pipe required	20m[including wastage and material required for practical welding]

3.3 Design Analysis

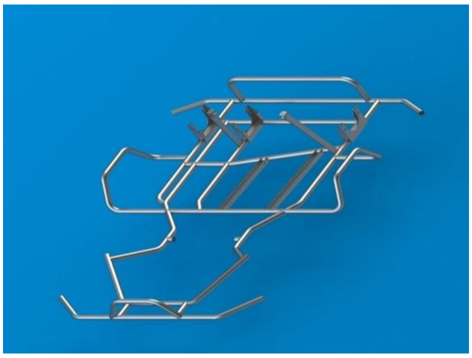


Figure 1 Isometric view of frame

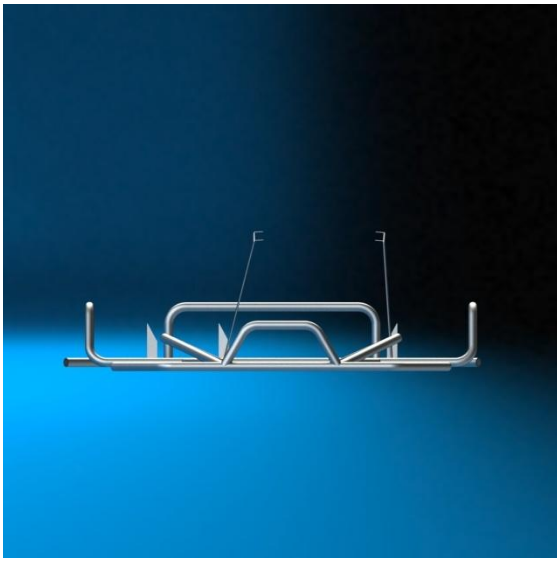


Figure 1 Front view of the frame

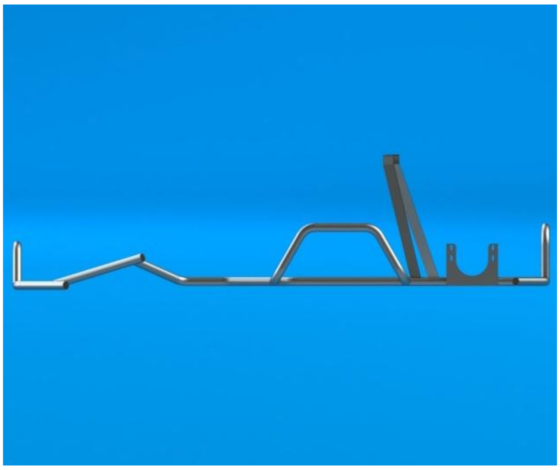


Figure 2 Side view of the frame

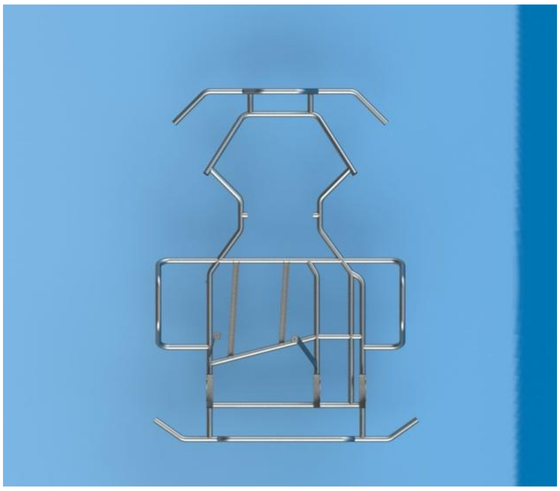


Figure 3 Top view of the frame

For the purpose of analysis, we have conducted certain test on the chassis. The following calculations were done to calculate the impact load

Table 3 Weight distribution

Parameter	Value
Weight of the kart	65kg
Weight of driver	70kg
Misc. weight (fuel, fire extinguisher etc.)	20kg
Total weight	155kg

Considering a scenario in which the vehicle hits a stationary object with a velocity of 50 km/hr (13.89 m/s), and let the impact duration be equal to 0.05 sec. Assuming the collision to be elastic in nature, the final velocity of vehicle will be 0 m/s. The impact force obtained is,

Impact force= (mass * velocity) / (2 * time)

Impact force= (155kg * 13.7m/s) / (2 * 0.05sec)

Impact force= 21,235N ≈ 14G

3.3.1 Front Impact Test

The front impact analysis has been carried out on the Ansys18.1 while constructing a perfect space frame tubular chassis on SolidWorks 2016 and then it was imported to Ansys18.1.

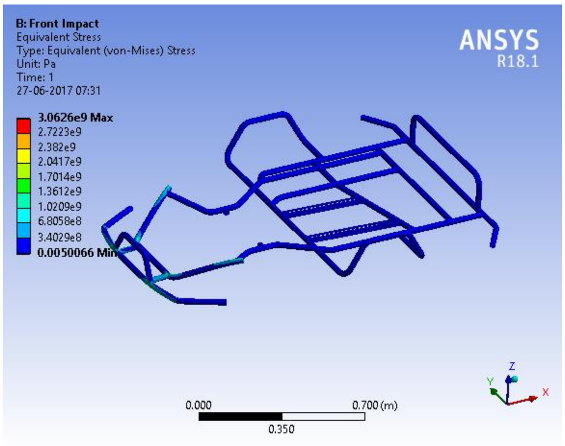


Figure 4 Stress parameters of front impact

A force of 14G was applied to the front ends constraining the body panel rods and we had seen such results in fig 5.

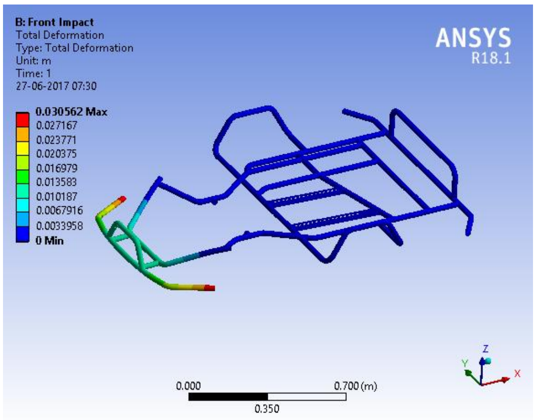


Figure 5 Deformation parameters of front impact

On applying a force of 14G the maximum deformation of 30.562mm observed in the chassis. This deformation is within the acceptable limits.

FOS = Yield strength of AISI 1018 / Mises Stress

So, FOS = 370 / 3062.6   **FOS= 0.120**

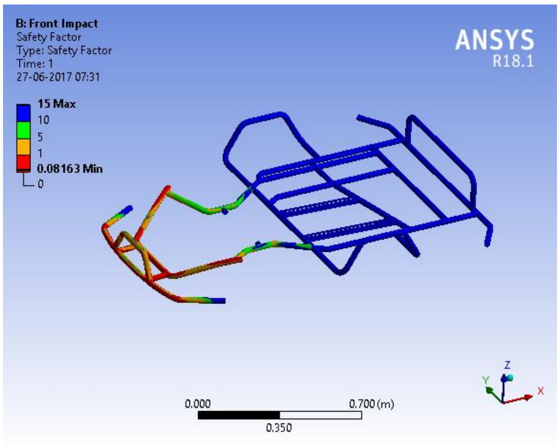


Figure 6 FOS parameters of front impact

3.3.2 Side Impact Test

The side impact analysis has been carried out on the Ansys18.1 while constructing a perfect space frame tubular chassis on SolidWorks 2016 and then it was imported to Ansys18.1.
A force of 14G has been applied and the observed deformation is 95.348mm and is within the acceptable limits.

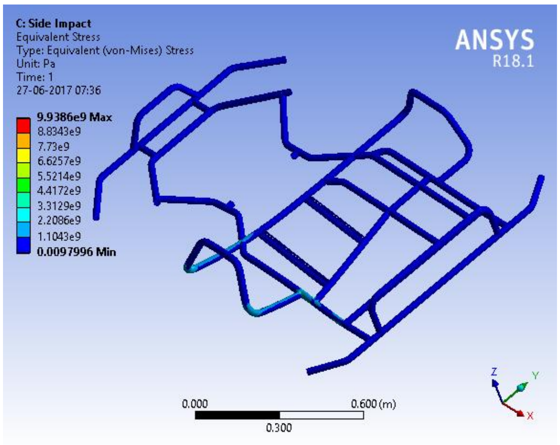


Figure 7 Stress parameters of side impact

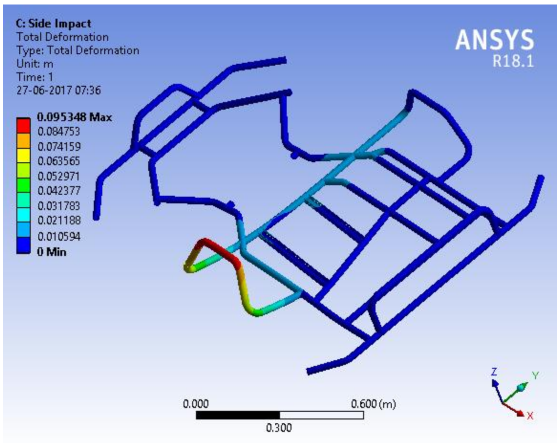


Figure 8 Deformation parameters of side impact

FOS = 370 / 9938.6 **FOS= 0.037**

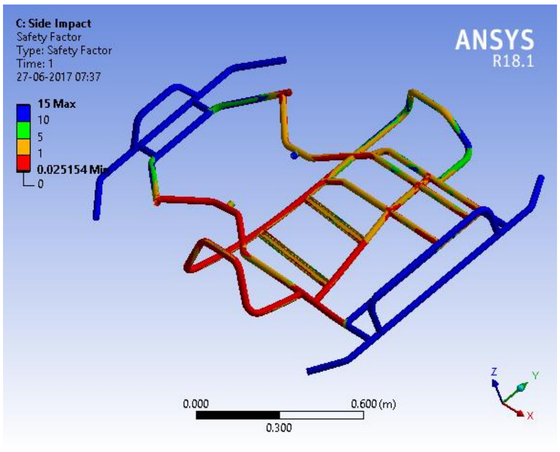


Figure 9 FOS parameters of side impact

3.3.3 Rear Impact Test

A force of 14G was applied to the rear ends by totally constraining the degree of freedom of the suspension and seen such results as shown in fig 8.

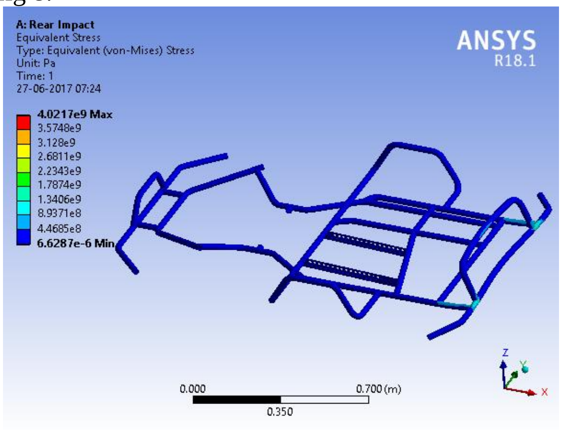


Figure 10 Stress parameters of rear impact

FOS = 370 / 4021.7 **FOS=0.092**

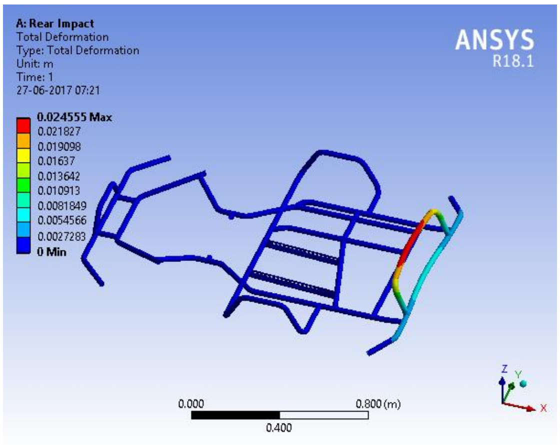


Figure 11 Deformation parameters of rear impact

A force of 14G has been applied and the observed deformation is 24.555mm and is within the acceptable limits.

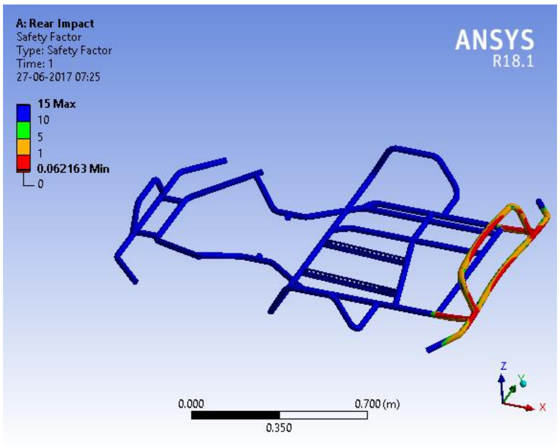


Figure 12 FOS parameters of rear impact

Table 4 Summarizing the above discussion

Elements	FOS	Maximum deformation	Maximum stress
Front impact	0.120	06mm	3062.6MPa
Side impact	0.037	95.348mm	9938.6MPa
Rear impact	0.092	24.555mm	4021.7MPa

3.4 Stability of the vehicle

In case of a four wheeled vehicle, it is essential that no wheel is lifted off the ground while the vehicle takes a turn. The condition is fulfilled as long as the vertical reaction of the ground on any of the wheels is positive in upward direction.

Mass of kart = 155 kg

Weight of the kart = 155 * 9.81

= 1520.55 N = 1G

3.4.1 Reactions due to weight

- Reaction on front wheels due to weight = (379.98 / 1084.58) * 1520.5= 532.72N = 0.35G (Upwards)
- Reaction on rear wheels due to weight = (704.6 / 1084.58) * 1520.55 = 987.83N = 0.649G (Upwards)

Weight distribution (Front : Rear) = (35:65)

Since, rear inner wheel is most vulnerable to lifting while cornering; we shall consider the reaction of the ground on that wheel only.

- Reaction on inner rear wheel due to weight = (475.2 / 954.5) * 987.83
  = 491.8N = 0.323G (Upwards)
- Reaction on outer rear wheel due to weight = (479.3 / 954.5) * 987.83
  = 496.03N = 0.326G (Upwards)

3.4.2 Reaction due to gyroscopic effect on rear wheel

- Let us assume radius of turn (R) be 5 m.
- Radius of wheel (r) =0.14m
- Track width (w) =0.9545 m
- Gear ratio (G.R) =13.75
- Moment of inertia of vehicle (I) =0.12 kgm^2
- Moment of inertia of engine (I_e) =0.024 kgm^2
- Gyroscopic couple on rear wheels (C_G) = $\{(I * v^2) / (r * R)\} + G.R * \{(I_e * v^2) / (r * R)\}$
  = $\{0.12 / (0.14 * 5) + (13.75 * 0.024) / (0.14 * 5)\} v^2$
  = $0.64\ v^2$ Nm
- Reaction due to couple on each rear wheel = $C_G / 2w$
  = $0.64\ v^2 / (2 * 0.9545)$
  = $0.34\ v^2$N (Upward)

3.4.3 Reaction due to centrifugal force on rear wheels

- Height of Centre of gravity (h) =0.306 m
- Couple due to centrifugal force (C_C) $= (m * v^2 * h) / R$
  $= \{(155 * 0.306) / 5\} * v^2$
  $= 9.49\ v^2$Nm
- Force on each rear wheels (F_C) $= C_C/2w$
  $= 4.97\ v^2$N (Upward)

3.4.4 Maximum velocity attainable at a corner of 5m radius

- $F_W = F_G+F_C$
- $491.8 = 0.34\ v^2+4.97v^2$
- $v_{max} = 9.6$ m/s

9.6m/s is the maximum speed with which the vehicle can turn without rolling.

IV. Different views of the vehicle

Figure 13 Isometric view

Figure 14 Front view

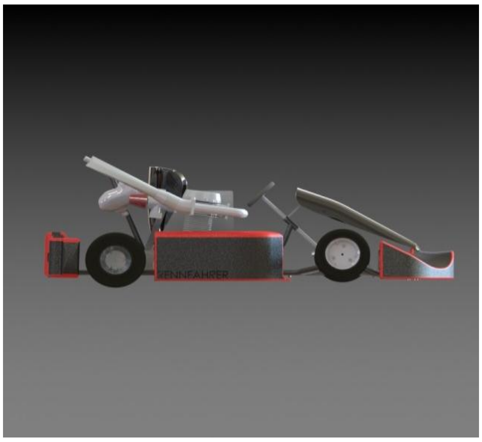


Figure 15 Side view

Figure 16 Rear view

Figure 17 Top view

V. Steering system

Figure 18 Steering system

Mechanical steering of three point linkage steering mechanism has been chosen because it is a simple mechanism with 1:1 steering ratio.
Design aim:

- Light weight mechanism.

- Proper handling with feedback.
- Appropriate steering geometry and settings for better driver control

5.1 Parameters as per frame design and space consideration:

1. Track width, front = 35.81inches =909.574mm
2. Wheel base = 42 inches =1050mm
3. Wheel Diameter = 10inches =254mm

5.2 Assumptions and Formulae used

Offset = 3.34 inches =85mm

Steering arm = 70mm.

1. Tr = offset + (Wb/sinθ), where θ= outer angle
2. Cotθ – Cotϕ = (Tw-offset)/Wb, where ϕ= inner angle
3. Sin(α+ϕ) + Sin(α-θ) = 2sinα.
4. (c-d)/2r = Sinα, where c is kingpin to kingpin, d is tie rod length and r is steering arm.

5.3 Calculations

- CotΦ – Cotθ= [TW-2(offset)]/b
  Using Φ=29 degrees, we get
  Θ=42 degrees
- Sinα= (c-d)/2r
- Φ=Outer wheel angle=29 degrees
- θ=Inner wheel angle=42 degrees
- α=Ackerman angle=26 degrees
- Turning radius=Offset+(WB/sin Φ)
  Using above values, we get,
  Turning radius=2.285metres (89.97 inches)
- Sin (α+θ) +Sin (α-Φ) = 2Sinα
  Using above values, we get, α=26 degrees

5.4 Steering and Handling settings

Settings provided on the knuckle.

- Ackerman setting as per calculated.
- Parallel steering

Tie rods will be pivoted on single point on knuckle.

As per Ackerman geometry, as the wheels will turn at different angles, the front inside being greater the steering response will be quite fast. During high speed cornering the outer tire will be shifted on the outer wheel, where in parallel steering would be effective.

Camber: negligible

Castor: **12**deg

KPI: above **10**deg

Toe: w.r.t. driver and turning of kart

5.5 Components and Analysis

1. Tie rods - Mild steel
2. Steering knuckle- 304 Stainless steel

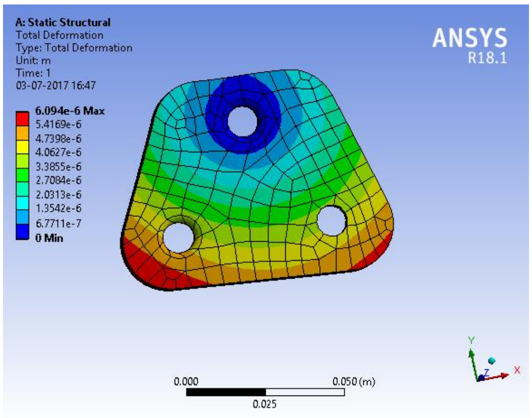
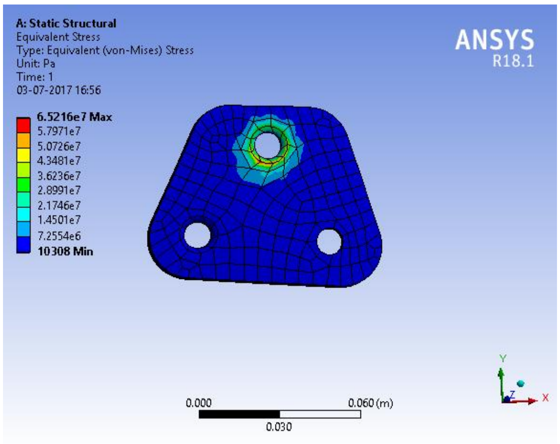


Figure 20 Steering Knuckle Maximum Deformation



Steering Knuckle Stress

3. Steering Column- Aluminum
4. Steering arms- Mild steel
5. Heim joints- Mild steel/ Chromoly
6. Kingpin Bolts- M8 Bolts
7. C bracket

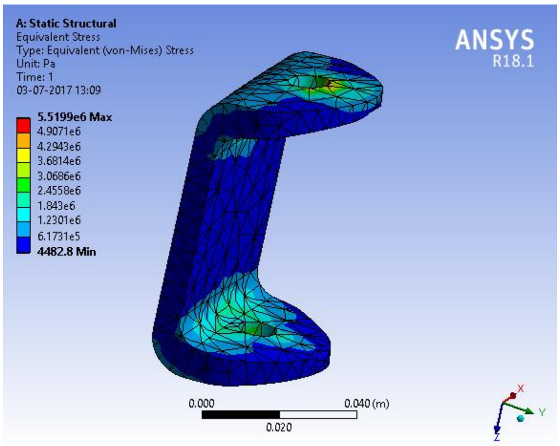


Figure 21 C Bracket Stress

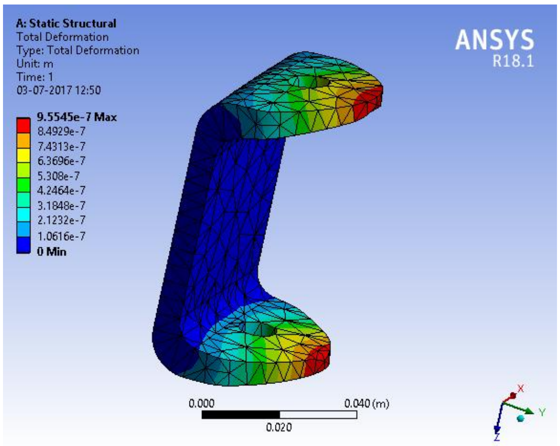


Figure 22 C bracket Deformation

8. Steering wheel hub

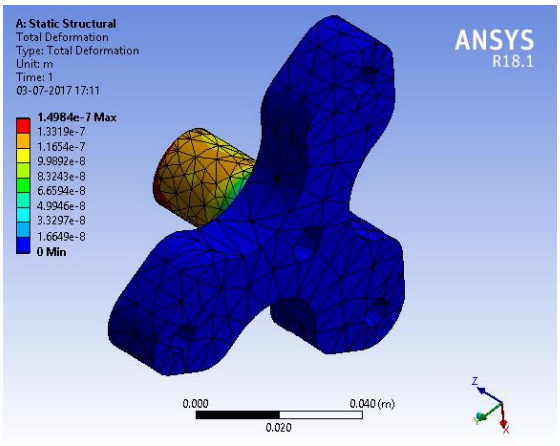


Figure 23 Steering wheel hub Maximum Deformation

9. Stub Axle

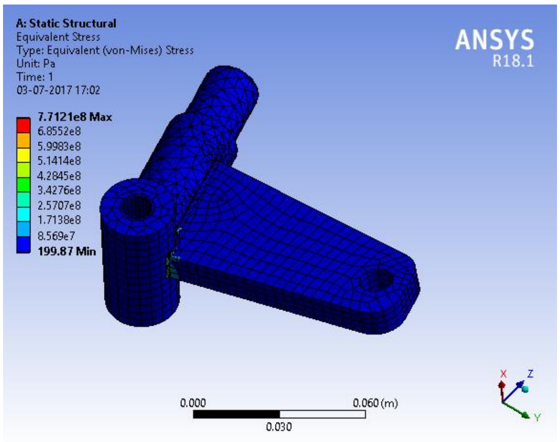


Figure 24 Stub Axle Stress

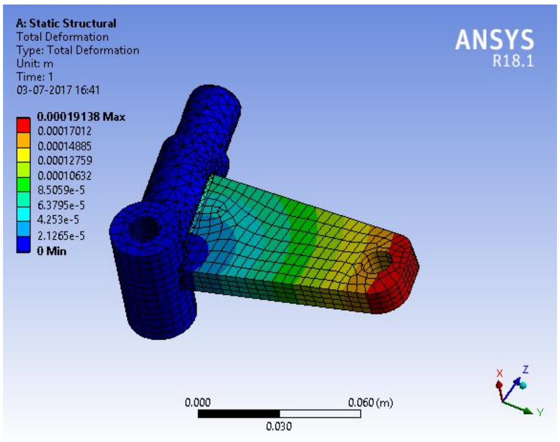


Figure 25 Stub Axle Deformation

5.5 Weight of steering assembly

- Weight of steering column(with steering wheel)-1 kg
- Weight of tie rods- 3kg(both)
- Weight of knuckle-0.10kg
- Weight of steering arms-0.40kg(both)

Total weight- 4.50 kg
Stub Axles: Steel

VI. Braking system

The main focus while designing the brakes of the kart was not only on brakes efficiency but also on the braking efficiency.

Following are the design considerations kept forward while designing & assembly of braking system:

- Effective braking in all conditions.
- Less driver fatigue.
- Simple and reliable brake system
- Adequate braking force capable of locking both rear wheels simultaneously.

6.1 Methodology

Initially, it was thought to install brakes on all the four wheels of kart but looking at the weight distribution of our kart, being biased much on the rear side of the kart carrying the engine and the batteries, installing disc brakes only on the rear wheels to save cost was decided. The proposed braking system layout for the vehicle is shown in the figure below. TVS Apache rtr160's brake disc and brake caliper have been used since they meet our requirements. The master cylinder is connected to the disc brake assembly fitted on the rear transmission shaft through brake lines**.**

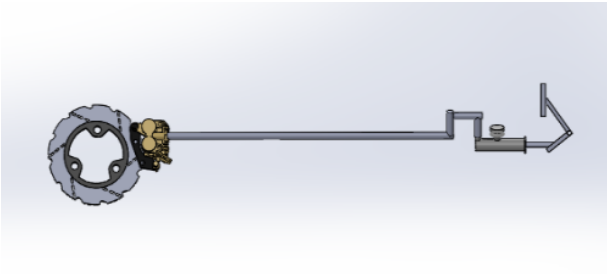


Figure 26 Brake assembly

Braking calculations were done at a velocity of 40 kmph considering the vehicle weight as 155 kg and results are shown in Table 5.

Table 5 Parameters

PARAMETER	VALUE
Brake Pedal Force	350N
Pedal Ratio	5:1
Fluid Pressure	4520000N/m^2
Braking Force	1286.78N
Stopping Distance	7.43m
Braking Torque	179.763N-m
Max. Deceleration	0.84g
Brake Fluid	DOT4

6.2 Calculations

- Vehicle Kerb Mass = 95Kg
- Centre of Gravity Height = 12.05"
- Driver Mass = 60Kg
- Static Mass on Front Axle = 91.45Kg
- Total Mass of Vehicle = 155Kg
- Static Mass on Rear Axle = 63.5Kg
- Weight Distribution = 59:41 (Front to Rear)
- Wheel Base = 42inches. (54inches)
- Mass Transfer while braking = (Total Mass * Rate of Deceleration * Centre of Gravity Height) / Wheel Base

Therefore, (155 * 0.84 * 12.05) / 42 = 37.355**kg**

Now,

- Dynamic Mass on Front Axle = Static Mass on Front Axle + Mass Transfer while braking

= 91.45 + 37.355 = 128.8kg

- Dynamic Mass on rear Axle = Static Mass on Rear Axle - Mass Transfer while braking

= 63.5 - 37.355

= 26.145kg

- Total Weight = 180 Kg

- % Front Weight (Static) = 59%
- CG Height in Inches = 12.05”
- Wheelbase (inches) = 42”

6.3 Brake pedal calculations

- Pedal Effort = 80lbs (350N)
- Pedal Ratio = 5
- Force at Master Cylinder = (Pedal Effort * Pedal Ratio)

Therefore, (350 * 5) = 1750 N

- Area of Master Cylinder Bore = 387.09mm^2
- Pressure in Brake Line = (Force at Master Cylinder/Area of Master Cylinder Bore)

Therefore, (1750 / 387.09) = 4.52 N/mm^2

- Radius of Calliper Piston = 14.5mm.
- Area of Calliper Piston = 660.51 mm^2
- Force at callipers=(pressure in brake line * area of calliper piston)

Therefore, (4.52 * 660.51) = 2986N

- Clamp force generated by callipers = f_{clamp} = F_{cal} * 2 = (2986 * 2) = 5972N
- Frictional force generated by callipers F_{fr} = F_{cl} * 0.35

Therefore, (5972 * 0.35) = 2090.27N

- Rotor Diameter = (200mm)
- Effective Radius of rotor = 86mm
- Torque at rotor = (Frictional Force * Effective radius of rotor)

Therefore, (2090.27 * 86) = 179,763.22N-mm.

The torque will be constant throughout the entire rotating assembly as follows:

- Torque at rotor = Torque at tyre
- Effective Radius of tyre = (11 / 2) * 25.4 = 139.7mm.
- Force at tyre = (Torque at rotor / Effective rolling radius of tyre)

Therefore, (179,763.22 / 139.7) = 1286.78N

- Total braking force generated = 1286.78N

6.4 Deceleration of a vehicle in motion

The deceleration of the vehicle will be given by a = F_{total} / m

a = Deceleration of the vehicle.

(1286.78 / 155) = 8.30m/s^2

Kinematic relationships of vehicles experiencing deceleration

S = v^2 / (2a)

S = Stopping distance of the vehicle

Therefore,

- Case 1: For (40kmph)

  {11.11^2 / (2 * 8.3)} = 7.43mt.

- Case 2: For (30kmph)

  {8.33^2 / (2 * 8.3)} = 4.18mt.

6.5 Hub design

For the disc to be mounted on axle, hub had to be designed that fits in the axle and which can bear the braking torque of 179.63N-m easily. The designing was done on Solidworks 2016. Analysis was done in Ansys 18.1. On applying a force of 1286.78N the maximum deformation of 7.515*10^{-3}mm is observed in the chassis. The deformation is within the acceptable limits.

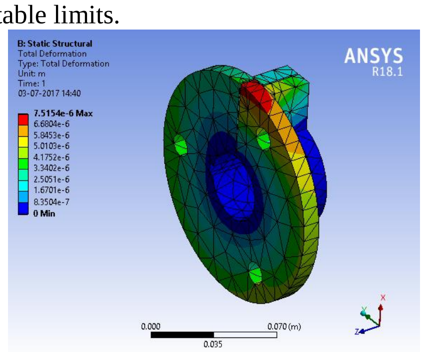


Figure 27 Deformation parameter of brake disk’s hub

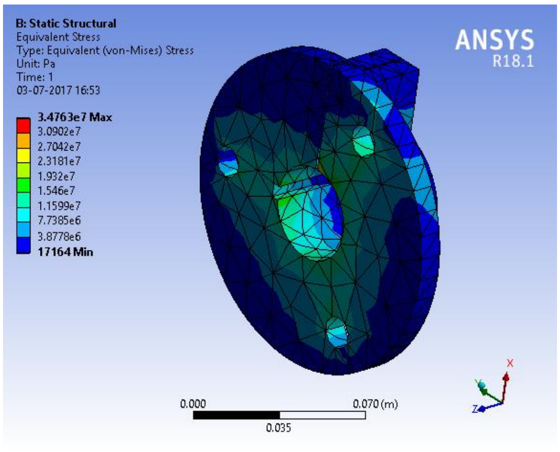


Figure 28 Stress parameters of brake disk's hub

FOS = yield strength of alloy6061 / Mises stress
So, FOS =276/34.763 **FOS=7.93**

VII. Powertrain

- Find the best power producing unit,
- Select an appropriate final drive
- Design appropriate rear axle

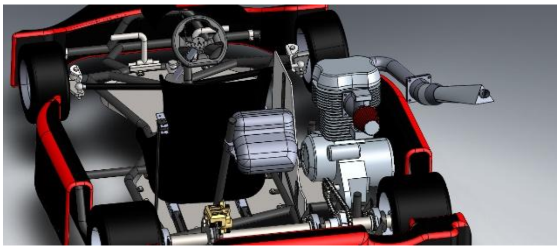


Figure 29 Powertrain View

7.1 Engine Selection

Ignitor produced a higher torque at lower rpm than the others. Also, it comes with Advanced Tumble Flow Induction technology. So, it overshadows Stunner and TVS Phoenix. Bajaj however produces highest power but there is no chance we can rev up to 9000rpm on the max straight path possible so we won't achieve that peak power.

7.2 Engine Specifications

Gearbox type	Manual
Number of gears	5 (1-N-2-3-4-5)
Type of clutch	Multi plate Wet Type
Primary ratio	3.350
1st gear	3.076
2nd gear	1.944
3rd gear	1.473
4th gear	1.190
5th gear	1.038
Type of drive	Chain drive

8. Drivetrain

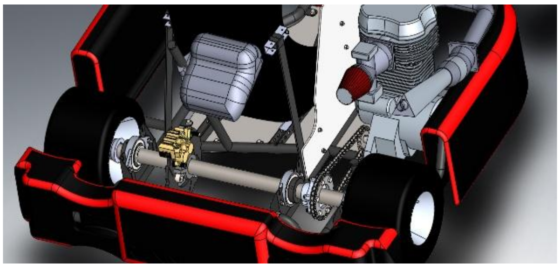


Figure 30 Drivetrain View

- Design of final drive sprocket
- Design of live axle

Better torsional rigidity, Better acceleration, Better braking, Reduced weight, provides more flexibility (softness) which helps in lift of inner rear wheel during turning so that it eliminates the need of differential.

7.1 Axle

Keeping in mind the above factors it was decided to use hollow axle instead of conventional solid one. Keeping the OD 40mm wall thickness was calculated required using Theories of bending, torsion and Tresca criterion (Maximum Shear Stress theory).

7.1.1 Calculation

Material chosen is AISI 4140 with
Yield strength = 415MPa.
d_o = outer dia = 40mm
d_i = inner dia
M = $\sqrt{(M_{weight})^2 + (M_{friction})^2}$ = 114345.93321N-mm
$M_{friction}$ = µLW = 76493.39652N-mm
µ = Coefficient of friction = 0.9 for slicks
L = Centre plane distance between wheels and outboard bearings = 169.88mm
W = Effective weight carried by each rear tire = 500.31N
= (Weight of vehicle×9.81×0.6)/2
Taking weight distribution = 40:60.
M_{weight} = µW = 84992.6628N-mm
T = 210000 N-mm
$\sigma_{allowable}$ = σ/F.O.S. = 166MPa
$T_{allowable}$ = T/F.O.S. = 95.782MPa
$T_{max} = \frac{\sqrt{(\frac{\sigma}{2})^2 + T^2}}{F.O.S.}$ = 126.3458MPa
F.O.S. = 2.5
K = ${}^{d_i}/_{d_0}$

Theory of Bending $\sigma_{allowable} = \frac{32 \times M \times d_o}{\pi((d_0)^4 - (d_i)^4)}$; thickness ≈ 2mm

Theory of Torsion
$T_{allowable} = \frac{16 \times T \times d_o}{\pi((d_0)^4 - (d_i)^4)}$; thickness ≈ 2mm

Tresca Criterion
$(d_0)^3 = \frac{16 \times \sqrt{M^2 - T^2}}{\pi \times T_{max} \times (1 - K^4)}$; thickness ≈ 3.6mm

7.1.2 Conclusion

Putting in the values of all the above data as given we get the value of wall thickness to be 3.5mm for safe design.

7.1.3 Axle Final Element Analysis

It was checked for the Bending Load and Torsion also

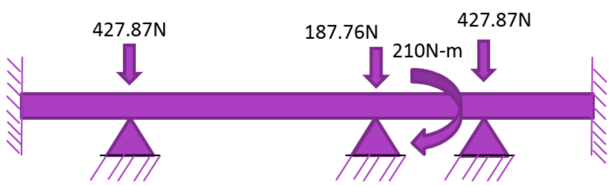


Figure 31 FBD of Force and Torque

Analysis was done in Ansys 18.1

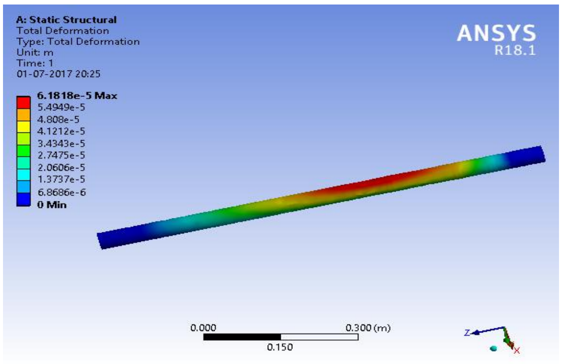


Figure 32 Total Deformation Axle

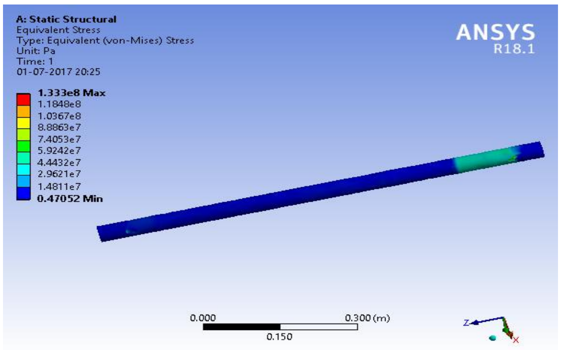


Figure 33 Equivalent Stress Axle

7.2 Ride Height Adjustment

Adjustment of axle height. It can be uplifted or lowered by using set of screws and bolts. Based on driver size if it is a taller driver (centre of gravity –higher) so he/she would be struggling with centre corner speed.

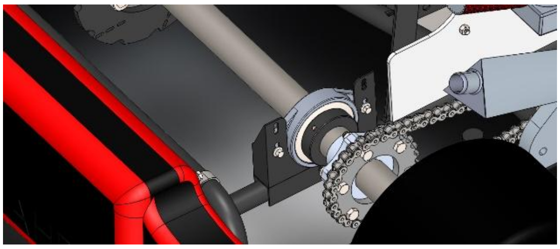


Figure 34 Ride Height Adjusting Brackets(left)

So raising the axle will result in lowering the Centre of Gravity and that will allow to carry a little bit more corner speed & be a little bit more free at centre exit of corner.

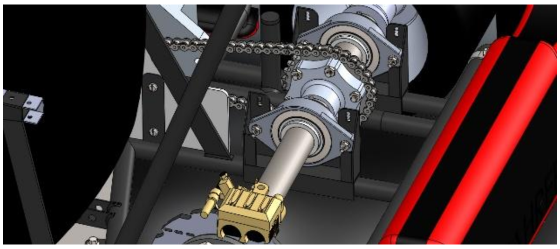


Figure 35 Ride Height Adjusting Brackets(right)

7.3 Sprocket Design

To calculate the acceleration which is maximum possible for our kart and set our top speed according to track length. Calculating the torque requirements of the kart and combining the results producing final drive ratio. Various considerations regarding track were made and final results were bought down.

7.3.1 Calculation of Total Torque

The technical specifications are given in the above tables. The gearbox efficiency is 95% and redline rpm is 9500. We have 14 teeth on our front sprocket. Using the formulas:

- Final drive ratio = $\frac{\text{Redline rpm}}{\text{primary reduction} \times \text{5th gear ratio} \times \text{Wheel rpm}}$
- Rear Sprocket teeth = Front sprocket teeth × Final drive ratio.

If the track and max possible span is 150-160 mtrs. Also, the maximum distance for acceleration test is 23 mtrs. Top speed was selected accordingly and final drive ratio designed.

We have two rear sprockets whose calculations are shown below. This is done to get best possible from the kart for different conditions. We have a sliding type lockable two-way hub on which we've mounted both of them and slide to change along with chain.

Maximum frictional torque: μNr

μ = coefficient of friction = 0.9

N = Dynamic weight at rear axle

r = radius of the wheel = 5.5 inches = 0.1397m

N= $\frac{(W\times r_s)+[W\times h\times (^{a}/_{g})]+(R_x\times H_x)}{L}$

Where,

$W\times r_s$ = static weight at rear axle = W× (distance of COG from front axle)

W= total weight of kart= 155 kg

h = distance of COG from ground = 0.306m

= 1071.373Nm

$\frac{W\times a\times h}{g}$ = Longitudinal mass transfer due to acceleration = 364.729Nm

$\frac{a}{g} = \frac{\mu\times\text{Dist.of COG from front axle}}{L-(\mu\times h)}$

Dist. Of COG from front axle = .7046m

L = Wheelbase = 1.084m

=0.78373g

Drawbar Pull = $R_x\times H_x$

R_x is chosen keeping in mind the traction test in which the kart will be pulling a payload, assuming M = 1400kg (keeping in mind the effect on karts performance). Max acceleration is considered to be same as of the kart.

R_x = Weight of vehicle × Acceleration of kart

H_x = Height of hitch point = 0.05334m

= 556.4927 Nm

N = $\frac{1071.373+367.429+556.4927}{1.084}$

=1838.0661N

Considering static condition,

For rolling resistance:

R = (a + b×V) × W; a = 0.015, b = 0.00016, V = 0(initial velocity) = 22.0725N

Total N = 1860.8125N

= 1.223g

Frictional torque = μNr = 230.85280Nm

For maximum torque transmitted to wheels:

T = $\frac{\eta\times\text{Power@5000rpm}\times\text{Gear ratio}\times 60}{2\pi\times N}$

Efficiency of gearbox = 95%

Power = $\frac{2\pi NT}{60}$; N = 5000rpm (Max torque at 5000rpm), T = 11Nm

Gear ratio = $\frac{Primary\ reduction\times Current\ G.R.\times No.of\ teeth\ on\ rear\ Sprocket}{No.of\ teeth\ on\ front\ Sprocket}$

7.3.2 Sprocket Teeth

A sheet containing the no of teeth required on rear sprocket for different speeds is mentioned below along with torques.

Similar calculations were run for choosing our second sprocket eliminating the factor of drawbar pull and including aerodynamic resistances.

Vehicle Speed(km/hr)	Vehicle Speed(m/s)	Wheel Speed(rad/s)	Wheel RPM	Final Drive Ratio	Rear Sprocket Teeth	Rear Sprocket teeth(Integer)
90	24.9999993	178.5714236	1706.0964	1.601319077	22.41846708	22
85	23.61111045	168.6507889	1611.31327	1.695514317	23.73720044	24
83	23.05555491	164.6825351	1573.40002	1.736370084	24.30918117	24
82	22.77777714	162.6984081	1554.44339	1.757545329	24.6056346	25
81	22.49999937	160.7142812	1535.48676	1.779243419	24.90940787	25
80	22.2222216	158.7301543	1516.53014	1.801483962	25.22077547	25
79	21.94444383	156.7460273	1497.57351	1.824287556	25.54002579	26
78	21.66666606	154.7619004	1478.61688	1.847675858	25.86746202	26
77	21.38888829	152.7777735	1459.66026	1.871671649	26.20340308	26
76	21.11111052	150.7936466	1440.70363	1.896298907	26.5481847	27
75	20.83333275	148.8095196	1421.747	1.921582893	26.9021605	27
74	20.55555498	146.8253927	1402.79037	1.947550229	27.26570321	27
73	20.27777721	144.8412658	1383.83375	1.974228999	27.63920599	28
72	19.99999944	142.8571388	1364.87712	2.001648847	28.02308385	28
71	19.72222167	140.8730119	1345.92049	2.029841084	28.41777517	28
70	19.4444439	138.888885	1326.96387	2.058838814	28.82374339	29
69	19.16666613	136.9047581	1308.00724	2.088677057	29.2414788	29
68	18.88888836	134.9206311	1289.05061	2.119392896	29.67150055	30

Figure 36 Sprocket and Speed Data

Sprocket Teeth(Rear)	Torque G1(Nm)	Torque G2(Nm)	Torque G3(Nm)	Torque G4(Nm)	Torque G5(Nm)
27	207.6744921	131.2481186	99.44880589	80.3422125	70.08001393
28	215.36614	136.10916	103.132095	83.31785	72.67557
29	223.0577879	140.9702014	106.8153841	86.2934875	75.27112607
30	230.7494357	145.8312429	110.4986732	89.269125	77.86668214

Figure 37 Desired Teeth and Corresponding Torque

7.3.4 Conclusion

For the calculation of frictional torque, a number of factors were taken into consideration such as dynamic weight at rear axle plus the static rolling resistance. Rear sprocket decided to be of 30 teeth so that it does not exceed frictional torque and the maximum possible speed is attained. Now the big question why to limit our top speed at 68kmph only. We have run several configurations of track. So according to us the game is about accelerating faster. Hence, we settled on a compromise and decided to achieve 68kmph quickly with the max able acceleration of our kart. Also, our second sprocket with 27 teeth can take our kart up to 76kmph which stands as an option during endurance and changes will be made in final drive ratio if needed.

7.3.4 CAD modelling Chain and Sprocket in Assembly

Sprocket – 29 teeth , 14 teeth

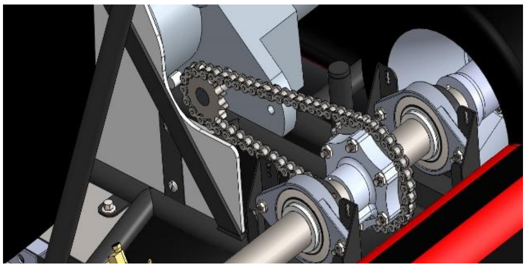


Figure 38 Chain and Sprocket Assembly

7.4 Wheel Hub Design

Wheel hub was designed for the given design of the axle. The design was done keeping in mind the Shear stress theory and Strain energy per unit volume theory.
Wheel hub was modelled in SolidWorks and its analysis was done in Ansys18.1 for safety factor and maximum deformation.

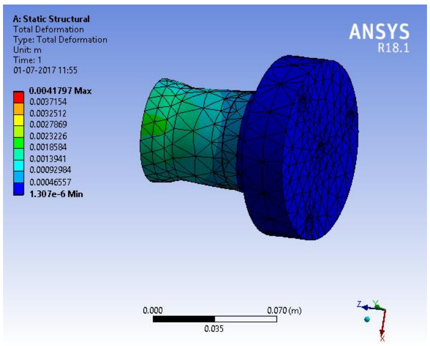


Figure 39 Maximum Deformation Wheel hub

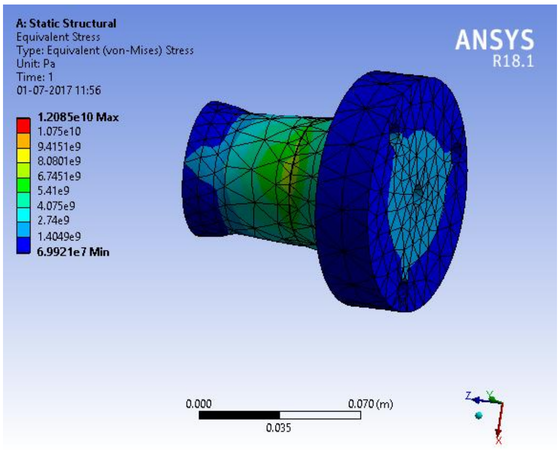


Figure 40 Equivalent Stress Wheel Hub

10 Safety and ergonomics

- Kart Dimension according to Ergonomics, Comfortability and Reachability.
- Floor Sketches – Hand sketches
- Ergonomics checked at every design stage

Figure 41 Final Ergonomics model of the Kart in SolidWorks

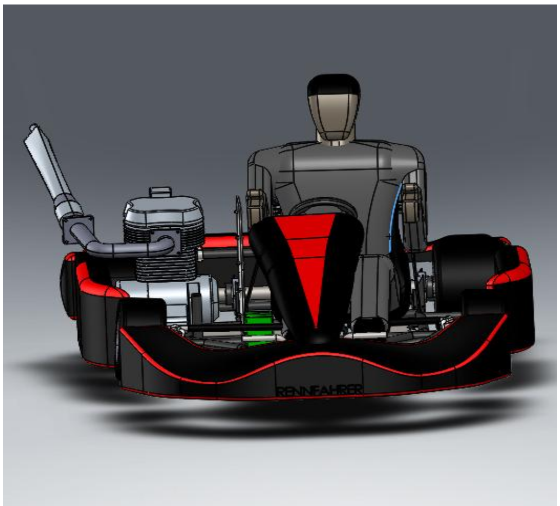


Figure 42 Distance between Driver and Engine

Engine's piston is at least 3 Inches away from the driver and distance between firewall and driver is at least 2 inches.
Driver comfortably reaches the pedals. To ensure it adjustable seat mounts have been incorporated in the frame of the Kart. Seat can slide up to 5 inches span (both forwards and backwards) according to driver's comfort. This was done using C section rods.

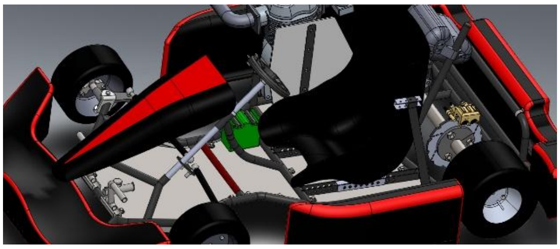


Figure 43 Seat Sliding Mounts

VIII. Conclusion

The design of go kart has become a vast and challenging task as the number of people getting attracted towards this activity is continuously increasing. Keeping this in mind, the design of the kart required to be technically sound, aesthetically pleasing and at the same time a value for money affair. The three before mentioned requirements shaped the methodology used to design the vehicle. Chassis material was selected which can be affordable and at the same time does not fail in the occasion of any unforeseen circumstance. Designing was done to keep the vehicle at par with other members of the segment. Analysis was done keeping in mind the safety of the vehicle components, the driver and the by standers watching the vehicle. A series of rigorous calculations and assumptions were used to finalize the steering geometry, chassis dimensions accompanied by afore taken engine specifications. The team is confident with the work and takes pride in it.

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