DETAILED DESIGN CALCULATIONS & ANALYSIS OF GO-KART VEHICLE

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Abstract: The main objective of this paper is to give detailed design calculation and analysis of go-kart vehicles parts. Go kart is a four wheel racing vehicle & since there is no suspension and no differential, when both rear tires turn at the same speed. The intention of this paper is modelling and analysis of go-kart vehicle parts according to their design calculation and simulation of parts. The modelling and analysis are performed by using 3D CAD design software tools such as CREO PARAMETRIC 3.0, SOLIDWORKS 2016 and subjected to simulation using ANSYS WORKBENCH 16.2 etc. where impact analysis was performed in front, rear and lateral directions. Then we observed the flexural rigidity of structure and chassis deformation. The comparison of results of two different materials and then properties with final design is completed. The paper consists of design stresses and deformation result of different components of vehicle parts such as stub axle and stub arm of steering system, and temperature analysis of disc brake. Based on the result obtained from these test the design is modified accordingly.

Key words: Simulation, Design calculation, modelling.

1. Introduction

Go-kart is a compact four wheeler racing vehicle. There is a small engine single Seater racing vehicle which doesn't have any suspension nor differential. Although go-kart comes in all different shapes and as required sizes. There are different forms of it as well, from motor less model to a very high powered racing machines. Go-kart having very low ground clearance and can be work on only flat racing track. We will create a model using 3D CAD software such as CREO PARAMETRIC, SOLIDWORKS and ANSYS WORKBENCH after completing the modeling the design is tested against all types of failure, stresses and deformation by using analysis software. Based on design calculation and analysis result can be change as per further modifications in dimensions.

2. Design and Testing Methodology

The research work, which previously focus on the definition of a planned methodology of virtual design and prototyping of go-kart vehicles which can be able to apply in the designing process of an existing one. The dynamic behavior of the vehicle is strongly influenced by the structural characteristics of the tubular frame. Since the go-kart does not have a differential and suspension system, its turning behavior is strongly influenced by the torsional deformation of the axle and stress ever when the speed of the go-kart is very high and about to take a turn at high speed.

We need to take all kind of different phases of design and Turing process so that all the evaluation can be done on time. Thus, at the end, the methodology can be distinguished mainly in three systems:-

1) Purpose of designing.
2) Methodology of designing.
3) Tool of aided design.

3. Design Specifications of Go-Kart:

All Go-kart specifications are made on theoretical calculation and can be changed as per modification will be done.

Table 1

CHASSIS	
WEIGHT	8.0296 kg
MATERIAL	AISI 4130
OUTER DIAMETER	25.4mm
THICKNESS	1.6mm
ENGINE	
MODEL	HONDA CBF STUNNER
DISPLACEMENT	124.7cc
MAX. POWER	11.6BHp@8000rpm
MAX. TORQUE	11Nm@6500rpm
VEHICLE DIMENSIONS	
WHEEL BASE	45 inch
TRACK WIDTH	38 inch
OVERALL LENGHT	65 inch
OVERALL WIDTH	49 inch
GROUND CLEARANCE	3mm
STEERING	
GEOMETRY	Ackerman
TURNING RADIUS	1.9 m
BRAKING SYSTEM	
OUTER DISC DIAMETER	190 mm
INNER DISC DIAMETER	60 mm

TARGETED PERFORMANCE

MAX. SPEED	45 Kmph
OVERALL WEIGHT	130

4. Design Methodology:

By framing the goals and constraints, we quickly identify the least flexible areas of design E.g. the first step of vehicle design was the selection of drives position, track width and wheel base.

4.1. Chassis:-

4.1.1. Material Selection: The chassis undergoes various kinds of forces during locomotion, it is intact without yielding, and it should be stiff to absorb vibration also it should resist high temperatures. The material property of the criterion while designing and manufacturing the car. The two very commonly used materials for making the chassis frame are AISI4130 (chromoly) and AISI1018.

Both of these materials were analyzed for different parameters and finally decided on to use AISI4130 (chromoly) for making the frame chassis.

4.1.2. Front Impact: For the front impact, engine and driver load was given at respective points. The kingpin mounting points and rear wheels position kept fixed. Front impact was calculated for an optimum speed of 45kmph. From impulse momentum equation, 5g force has been calculated. The loads were applied only at front end of the chassis because application of forces at one end, while constraining the other, results in a more conservative approach of analysis. Time of impact considered is 0.2 seconds as per industrial standards.

F x t = m x (Vi - Vf)
F x 0.2 = 130 x (12.5 - 0)
F=8.125 KN

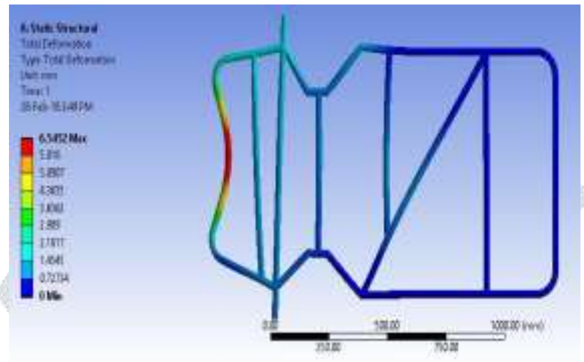

Fig. 4.1.2 Total deformation

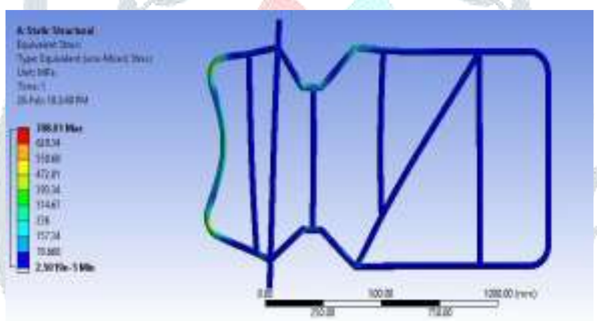

Fig. 4.1.2 Equivalent stresses

4.1.3. Rear Impact: Considering the worst case collision for rear impact, force is calculated as similar to front impact for speed of 45kmph. The value of 5g force has been calculated. Load was applied at rear end of the chassis while constraining front end and king pin mounting points. Time of impact considered is 0.2 seconds as per industrial standards. F x t = m x (Vi-Vf)
F x 0.2 = 130 x (12.5 -0)
F=8.125 KN

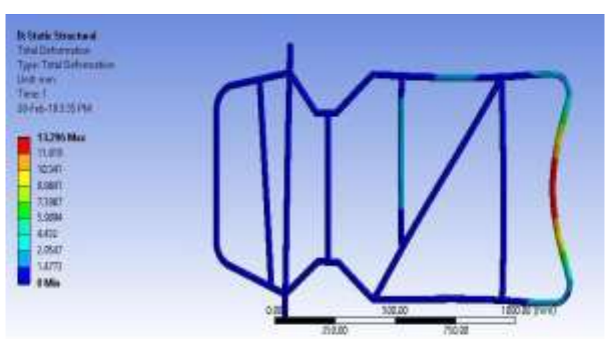

Fig. 4.1.3 Total deformation

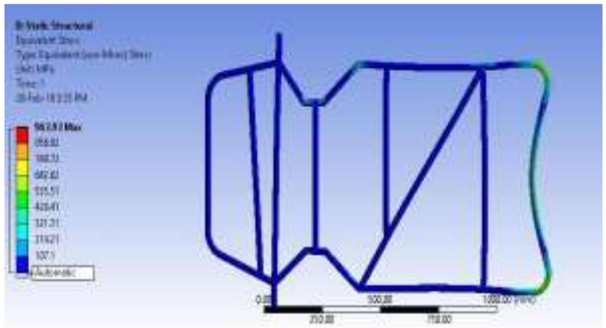

Fig. 4.1.3 Equivalent stresses

4.1.4. Side Impact: The most probable condition of an impact from the side would be with the vehicle already in motion. So it was assumed that neither the vehicle would be a fixed object. For the side impact the velocity of vehicle is taken 45kmph and time of impact considered is 0.2 seconds as per industrial standards. Impact force was applied by constraining left side of chassis and applying load equivalent to 2.5g force on the right side.

F x t = m x (Vi-Vf)*0.5
F x 0.2 = 130x (12.5 -0)*0.5

F=4.062 KN

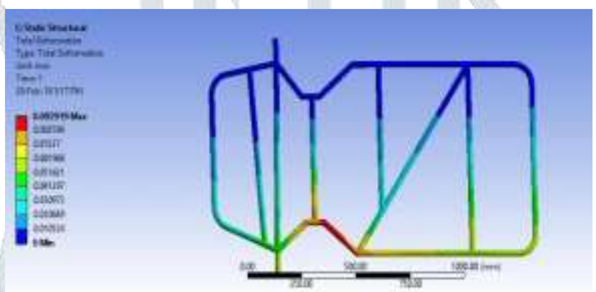

Fig. 4.1.4 Total deformation

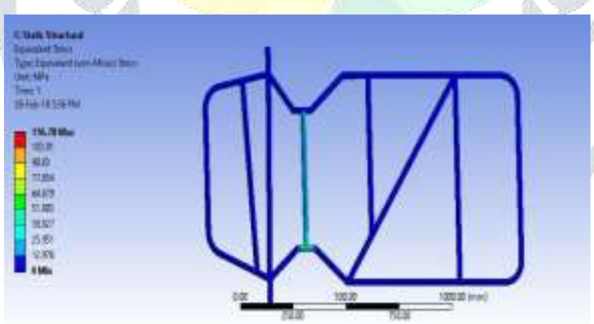

Fig. 4.1.4 Equivalent stresses

4.1.5. Modal Analysis: During Modal analysis, the chassis was tested at the Maximum engine RPM that is 8000. This frequency was applied to the entire chassis, and the total deformation was obtained.

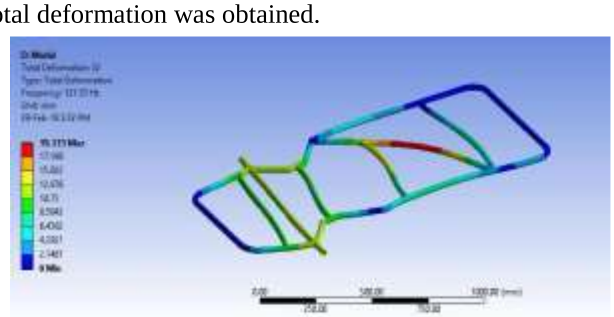

Fig 4.1.5 Modal analysis total deformation

4.2. Engine & Transmission

4.2.1. Introduction: A single cylinder four stroke 124.7 cc engines is selected. So there had number of options for the selection of engine such as Honda shine, Bajaj discover, TVS Phoenix etc. After long research work and survey, it is decided to use Honda stunner 125 engine to power a kart. It have inbuilt gear box of manual 5 speed constant mesh gear box, with the multi plate wet clutch. So the design is according to the engine specification.

4.2.2. Design Objectives

- Each component must be properly designed, analyzed and optimized before manufacturing and prototyping.
- All forces and loads must be considered while analyzing the CAD geometry.
- Most precise meshing should be used in order to get

most accurate results.

Table 2

Engine	**Honda CBF Stunner**
Displacement	124.7cc
Max. Power	11.6BHp@8000rpm
Max. Torque	11 Nm@6500rpm
Compression Ratio	9:2:1
Transmission	5 Speed
Bore	52.4 mm
Stroke	57.8 mm
Dry Weight	28 Kg
Starting Type	Self start
Transmission	
Gear Box	5 Speed
Clutch	Wet,Multi-plate
Transmission Type	Chain Drive
Gear Reduction Ratio	
Primary Reduction	3.35
First Gear	3.076
Second Gear	1.944
Third Gear	1.473
Fourth Gear	1.19
Fifth Gear	1.038
Final Reduction	3.071

4.2.3. Engine Calculations

No of teeth on gear=34

No of teeth on pinion=14

Pinion RPM=1200 (Tachometer)

Rated Power=14.32KW

Ref1-Selection Of chain No. (10A)

From Chain No. 12A

Ref 1- pitch of roller chain 19.05mm Now, K.W. Rating Of chain=(KW to be Transmitted)*Ks/(K1*K2) Where,

Ks=1.2

K1=1.0

K2= 0.78

Therefore,

KW rating is 12.38KW=16.59HP Now, by using KW=16.59, RPM=1200

Pitch=15.875mm

Therefore, PCD of sprocket will be =pitch/sin (180/teeth) =172.05mm

4.3. Steering

4.3.1. Introduction

Four bar linkage mechanism which consist of following steering component which were designed in CAD software PTC CREO 3.0 and analyzed by means of CAE software ANSYS 16.2.

4.3.2. Design objectives

- Each component must be properly designed, analyzed and optimized before manufacturing and prototyping.
- All forces and loads must be considered while analyzing the CAD geometry.
- Most precise meshing should be used in order to get most accurate results.

4.3.3. Steering parameters

Table 3

WHEEL BASE	1143 mm
TRACK WIDTH	965.2 mm
KING PIN DISTANCE	964.964 mm
TIE-ROD LENGTH	856.234 mm
MINIMUMTURNING RADIUS	1.95 m
CASTER ANGLE	0
ACKERMAN ANGLE	22.89

4.3.4. Weight distribution

The total weight of vehicle is 130 kg (including driver)

The weight distribution is very important to understand the vehicle dynamics and analyzing the vehicle in various aspects.
The weight distribution was assumed to be 43% of total weight (that is 130kg) in the front and 57% in rear portion of vehicle.
By the theoretical calculation the location of center of gravity was calculated. The height of center of gravity is 3 inches.

4.3.5. Steering calculation

Outer steering angle
Inner steering angle
£o= L/(R+t/2)
£I= L/(R-t/2)
£o=25.10
£I=37.78
Where,
L=wheelbase
R=turning radius
t=track width

- **Ackerman arm angle**

$\alpha = \tan^{-1}[0.5*t/l]$
$\alpha = 22.89$

- **length of tie rod**

LT=DKC-2RAB*sin (α)
LT=33.71inches
Here,
LT=length of tie rod
Dkc=kingpin Centre to center distance=37.99inches
RAB=Ackerman arm radius=5.5inches

- **Turning radius**

R=L/2*sin (α) R=1.95m

Assumptions:-
Mass in front tires (m) =55.9kg
Average velocity (v) = 30 km/hr =8.33 m/s μ =0.6

Normal Force on Stub Axle:
N=m*g
=55.9*9.81
= 548.37 N
Lateral Force on stub axle:
Lateral Force = mv^2/r
= 2021.68 N
Tractive force:
Force due to traction = μ*Normal force
= 329.02 N

4.3.6. Structural analysis of Stub axle of wheel: mass in front was considered 55.9 Kg, thus weight on one wheel stub axle would be half of front weight.

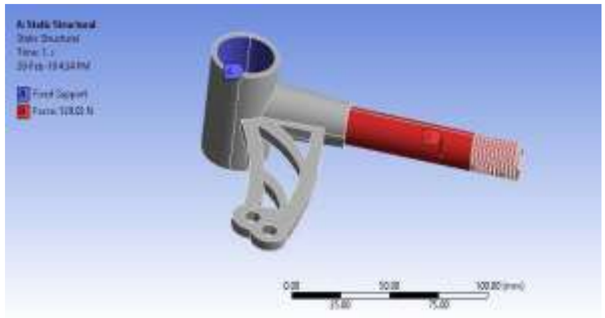

Fig.4.3.6. loading condition of stub axle of wheel

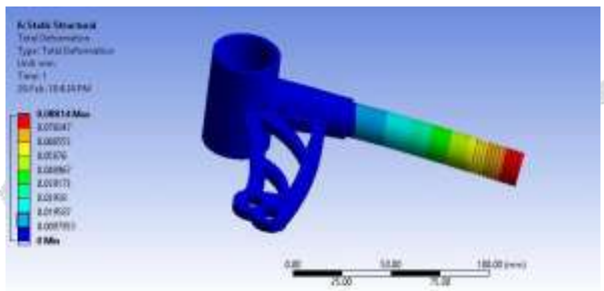

Fig.4.3.6. Total deformation

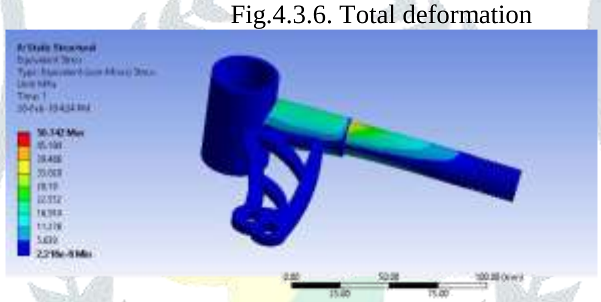

Fig 4.3.6. Equivalent stresses

0.08814 mm of deformation is occurred when the load of 329.02 N was applied. Internal stress generated due load applied was 50.742 Mpa.

4.3.7. Analysis at Lateral Force (while turning): Lateral force calculated was 2021.68 N

Only half load are applied in one side 1010.8 N

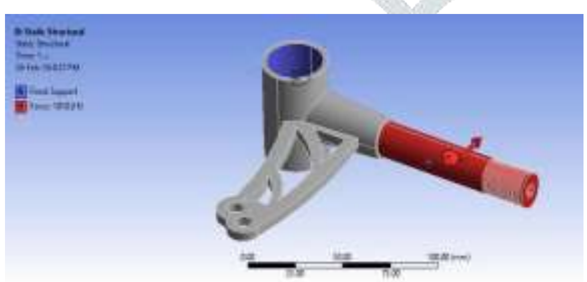

Fig.4.3.7. loading condition of stub axle (while turning)

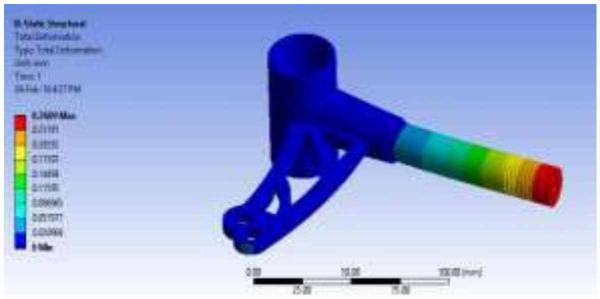

Fig.4.3.7. Total deformation

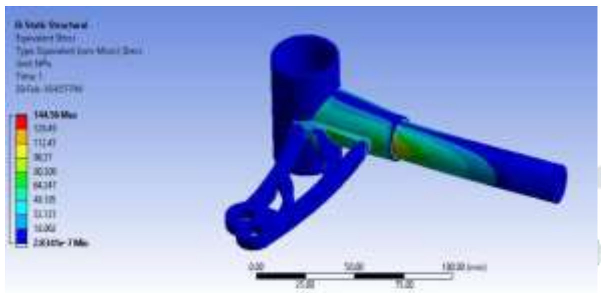

Fig.4.3.6. Total deformation

0.2609mm of deformation is occurred when the load of 1010.8 N was applied. Internal stress generated due load applied was 144.56 Mpa.

4.3.8. Structural Analysis of stub arm: load on the both the front wheels, so on one stub arm 50% load with factor of safety was taken.

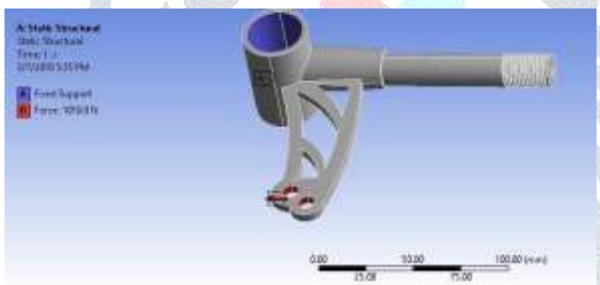

Fig.4.3.8. loading condition of stub arm

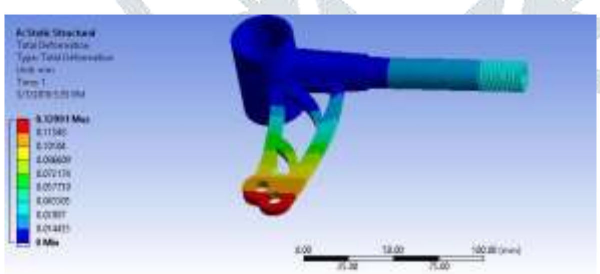

Fig.4.3.8. Total deformation

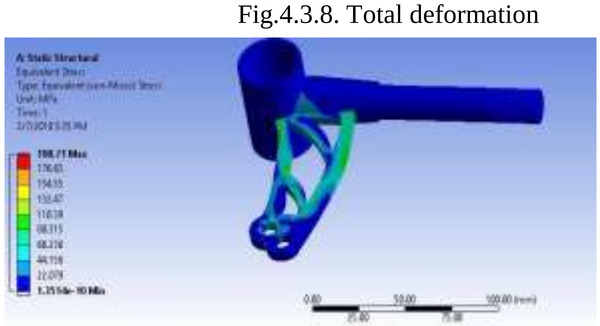

Fig.4.3.8. Equivalent stresses

0.12991 mm of deformation is occurred when the load of 1010.8 N was applied. Internal stress generated due load applied was 198.71 Mpa.

4.4. braking

4.4.1. Introduction

A brake is a mechanical device that inhibits motion by absorbing energy from a moving system. It is used for slowing or stopping a moving vehicle, wheel, axel, or to prevent its motion, most often accomplished by means of friction.

4.4.2. Design objectives

- The objective of Braking System is to provide reliable and prompt deceleration of vehicle.
- In order to achieve maximum performance from the braking system, the brakes have been designed to lock up rear wheel, while minimizing the cost and weight.
- Moreover, the driver must have complete control of the vehicle while brakes are actuated.

4.4.3. Braking parameter

Brake Type	Hydraulic disc brake
Disc diameter	190mm
TMC diameter	5.6mm
Caliper Piston Diameter	20mm
Pedal Force Applied	98.1N
Pedal Ratio	4:1
Velocity	16.67m/s

4.4.4. Calculations

- Overall mass=130kg
- Overall weight = 1275.3N
- Outer diameter of disc = 190mm
- Inner diameter of disc = 60mm
- Pedal ratio = 4:1
- Velocity after braking = 0
- Velocity before braking (Vv) = 12.5m/s
- Pedal effort by driver = 98.1N
- Weight distribution ratio = 57:43
- Inner diameter of rim = 5 inches
- Outer diameter of rim = 11 inches
- Deceleration = v^2-u^2/2s

  =0-$(12.5)^2$/2×4.25

  =18.38m/s
- Stopping distance (SDv)

  =12.5^2/2 × (0.44+0.73) × 1.6 × 9.81

  =4.25m
- Where, SDv is stopping distance of the vehicle. Vv is the velocity of the vehicle in motion. The kart is decelerating at 18.38 m/sec², Therefore the kart moving at 45km/h.
- Stopping Distance Time

= v-u/a

=12.5/18.38

=0.68s

- Static weight on front axel

=Weight distribution on front wheel × Overall weight

= 0.43×1275.3

=548.379N

- Static weight on rear axel

=Weight distribution rear wheel× Overall weight = 0.57×1275.3

= 726.921N

- The absolute weight transferred from the rear axle to the front axle(WT):

$WT = (av / g^2) \times (hcg / WB) \times Vt$

Where, g is the acceleration due to gravity. hcg is the vertical distance from the CG to the ground. Vt is the total vehicle vertical force.

WT= 287.099 N

- Dynamic weight on Front Axel

= (static front weight +dynamic weight transfer)

= 548.379+287.099

=835.478N

- Dynamic weight on Rear Axel

= (static rear weight - dynamic weight transfer)

= 726.921-287.99

=439.822N

- Friction Force at Each Front Wheel

=0.35×417.739

=146.208N

- Friction Force at each rear wheel

=0.35×219.911

=76.968N

4.4.5. BRAKE PEDAL CALCULATION

- Pedal effort=98.1N
- Pedal ratio=4:1
- Force at master cylinder=pedal effort× pedal ratio

=98.1×5

=490.5N

- Area of master cylinder=7.068mm^2
  - Pressure in brake line=490.5/7.068 =69.397N/mm^2
- Radius of caliper piston=10mm
- Area of caliper piston=314.159mm^2

- Force at caliper=69.397×314.159 =21.802×1000N
- Clamp force generated by caliper=43604N
- Frictional force generated by caliper=43604×0.35
  =15261.4N
- Rotor diameter(disc)=190mm
- Effective radius of rotor=60mm
- Torque at rotor=15261.4×60
  =915684N
- Torque at rotor=torque at tire
- Effective radius at tyre=139.7mm
- Diameter of tyre=11 inches

- Force at tyre=915684/139..7 =6554.645N
- Total breaking force generated=6554.645N

4.4.6. BRAKE DISC

- Coefficient of friction (μ) = 0.35
- Area of brake pad = 2411.43 mm^2
- Overall mass = 130 kg
- Stopping time (t) = 0.680 sec
- Kinetic energy = ½ mv^2
  $=1/2 \times 130 \times 12.5^2$
  =10156.25 J
- Braking power (Bp) = K.E / t
  =10156.25/0.680 =14935.66 W
- Heat Flux (Q) = Bp/A
  $=14935.66/2411.43\times 10^{-6} = 6.193\times 10^{6}\ W/m^2$
- Maximum Temperature of disc

$=0.527\times Q\times\sqrt{t}/\sqrt{PCK} + T_{atm}$

$=0.527\times 6.193\times 10^{6}\times\sqrt{0.68}/\sqrt{7250\times 500\times 58}+27$

$=185.26^{o}c$

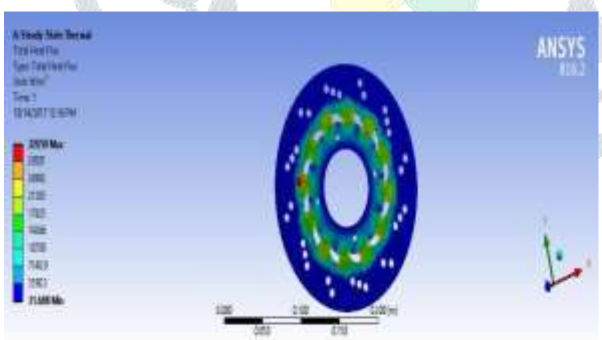

Fig.4.4.6. heat flux analysis

5. Result:

Chassis impact test: The maximum forces are applied on the chassis and the test is performed and finally we got the following result.

Sr. no.	Impact	Force	Total Deformation	Equivalent-stress

1.	Front	8.125 KN	6.54 mm	708.01Mpa
2.	Rear	8.125 KN	13.29 mm	963.93Mpa
3.	Side	4.062 KN	0.092 mm	116.78Mpa

Modal Analysis: The following result is found by applying the possible maximum frequency.

Sr. no.	Frequency	Total deformation
1.	121.53 Hz	19.315 mm

Wheel stub axle: The maximum force is applied on the wheel stub axle and calculated the result as shown below.

Sr. no.	Force	Total deformation	Equivalent-stress
1.	329.02 N	0.08814 mm	50.742 Mpa

Turning stub axle: The maximum force is applied during the turning of the wheel and tested which gave result as follows.

Sr. no.	Force	Total deformation	Equivalent-stress
1.	1010.8 N	0.2609 mm	144.56 Mpa

Stub arm: The Tie-rod force is applied on stub arm and tested to get the following result.

Sr. no.	Force	Total deformation	Equivalent-stress
1.	1010.8 N	0.12991 mm	198.71 Mpa

Brake disc: Temperature of disc brake is increased to 185.26 °C which gave the total heat flux of 32059 W/m^2.

Sr. no.	Temperature	Total Heat flux
1.	185.26 °C	32059 W/m^2

6. Conclusion:

AISI 4130 is better material in terms of strength, reliability and performance when compared with AISI 1018. It is also suitable for large-scale production. Successful analysis was carried out on the chassis of CAD modal using ANSYS WORKBENCH to determine, equivalent stresses, and total deformation results. The engine selected and drive train are designed such as to give maximum performance in terms of speed as well as fuel economy. The steering system designed for the kart less effort while encountering a turn. It also performs effectively during high speed turns with negligible chances of skidding. The stub axles are also designed so as to work in all working conditions and under heavy loads. The braking system is so designed and mounted so as to lock the rear wheels and stop the kart within safe distance. Under condition of brake failure, there is an over travel switch located behind the brake pedal which seizes the engine and ensures driver's safety. This paper provides adequate knowledge and proper guidelines for designing a go-kart model. Thus, after all the calculations and analysis, it is finally concluded that this go-kart is safe for fabrication under healthy engineering practices and meets the performance targets.

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