# Design & Analysis of a Combined Power Go-Kart

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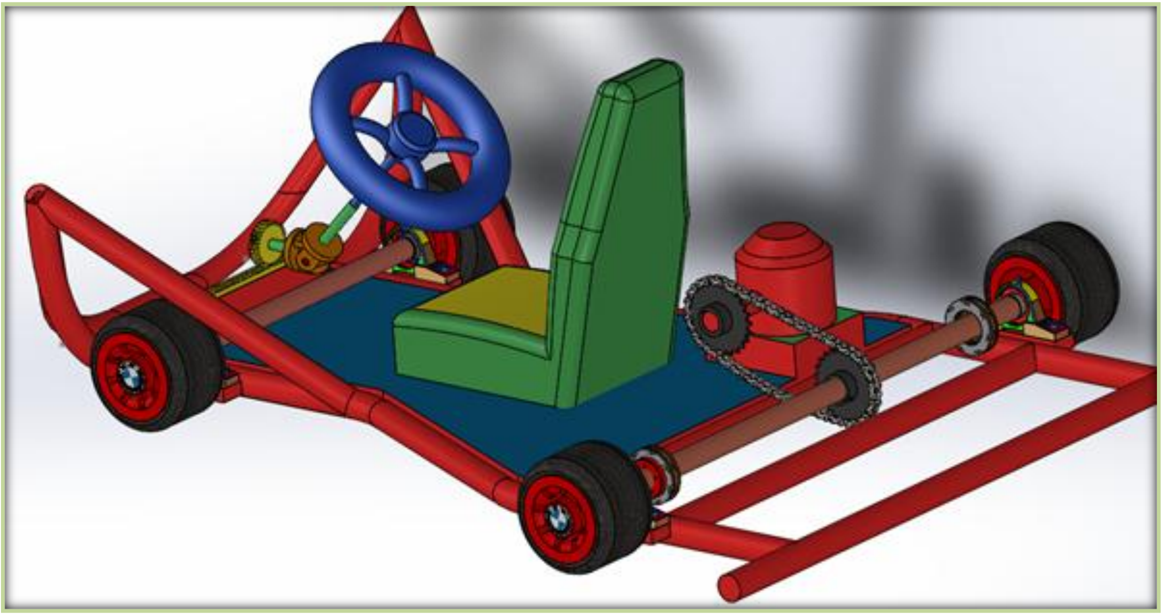

NETAJI SUBHASH ENGINEERING COLLEGE

# DESIGN & ANALYSIS OF A COMBINED-POWER GO-KART


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# COMBINED-POWER GO-KART

## I. SYNOPSIS


GO-KART is a four-wheeled motorized vehicle used for racing and amusement purposes. The main aim of this design is to design a modern day GO-KART which can run on both PETROL (conventional fuel) as well as on SOLAR ENERGY (non-conventional fuel). The principle of FEA (finite element analysis) has been used for the purpose of finding and solving potential structural or performance issues.

The main motive of our design is driver safety and inducing less harm to the environment. The vehicle has been designed such that it can carry a weight of about 80 to 90 kilograms. Chassis frame is tubular in cross-section of 6063-T6 Aluminium Alloy. All the probable analysis such as IMPACT TESTING, TORSION TESTING and DYNAMIC TESTING has been done to safeguard the vehicle against any probable damage. These vehicles usually are not equipped with suspension system and differential.

A close analysis and detailed calculation of each and every component required to manufacture this unit has been performed and shown here.



KEYWORDS – Go-Kart, Engine, Solar Energy, Components, Design, Assembly, Analysis.


## II. PREFACE

Final Year Project involves selection of projects with the latest technologies and also gives us a chance to demonstrate what we have learned. Not only does it provide insights about the future concerned, it also bridges the gap between theory and practical knowledge. The project starts with design, analysis followed by calculations of different mechanical elements which we are going to use in our proposed Go-kart. The experience and knowledge gained in the practical field during this project period was heuristic to say the least.

It was a great experience for us to know how mechanical component of Go-kart works and analysis under all the possible conditions with smooth operations to achieve desired results.


Our project guide Professor Subhrajyoti Sarkar was very kind and helpful and cleared our doubts in a very methodical and scientific manner. We hereby wish to convey our sincere regards and thank him for helping us out.

**Yours sincerely,**

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## III. ACKNOWLEDGEMENT

We would like to thank our teacher, Professor Subhrajyoti Sarkar, for this project who gave us this opportunity and entrusted upon us this project which is to be completed in our FINAL YEAR of Bachelor of technology in Mechanical Engineering and also without whose help this project would have just been an idea.

We would also like to thank our college, Netaji Subhash Engineering College, our Principal and our Head of the Department for providing us the space and necessary equipment to be used in this whole process.

## IV. INTRODUCTION

GO-KART is a simple, light-weighted vehicle used for commercial and racing purposes. Commercial karts are usually powered by a 4-STROKE engine or by an electric motor whereas racing usually employ 2-STROKE engine. Most of them are single-seated but two- seaters are equally prevalent. Being a racing vehicle it's ground clearance is extremely low which is why there is no suspension spring. Chassis or the supportive framework is considered as the most important component in this design as the whole load has to be carried by the chassis. The track could be indoor type or outdoor type.

The paper aims at design and analysis of a combined-power go-kart keeping in mind various safety issues. Modelling (part design and assembly) is performed on SOLIDWORKS software and analysis on ANSYS R3 (ver. 2019). Go-karts are usually raced on level and smooth ground with no obstructions (viz. speed breakers). Go-karts are considered as base level racing cars and are used by those who are learning racing skills as they are light-weighted and easily operated. Go-kart industry has grown to be a prolific industry in USA, JAPAN and EUROPE. Some uses of Go-kart are -

- Recreational purpose
- F-1 racing
- Concession cars
- Indoor rental
- Short distance travel etc.

# V. LITERATURE REVIEW

A.Pandiyan and his team (2016) designed go-kart chassis in identifying the strength and weakness of the design. They performed various impact testing of the chassis in ANSYS software. They used material code AISI 1018 and their chassis weight was around 20 kilograms and it could carry the weight of a 70 kilogram person. [1]

Md. Salman and his team (2017) designed and analysed the chassis model of a go-kart. Their main aim was design the go-kart keeping in mind the driver's safety and also to make the go-kart light and high strength. They performed the impact testing on go-kart (front test, rear test, side impact test). The analysis and design of their go-kart determined the stresses developed in the chassis which performed an important role in determining the safety factor and other means to safeguard the chassis by performing its analysis. [2]

Akhilesh K Dewangan and his team (2015) designed and analysed a Solar Powered Go-Kart (PV module) which was three seater as well. Their paper mainly focussed on how engineering and design process explained each system in this vehicle which can build a high-performance, cost-effective and safe electric go-kart for future use. They used trike design which has proved to be both ergonomic due to their recumbence and also performance oriented. [3]

Arjun Bopaiah (2018) published his paper on Transmission system of Go-kart where the power is transmitted from the engine to wheel drive by chain assembly. His idea was to get maximum speed with minimum load on the engine, maximum torque at starting and also to reduce various power losses. [4]

Gangesh Shukla and his team (2019), surveyed the history and future of solar and electric vehicles and provided an overview of a typical solar car. This paper discussed about the usage of solar energy to power up the vehicle. To make it cost effective, power converters, and batteries were being used.

Their calculations were satisfactory. But these disadvantages could be easily overcome by conducting further research in that area. [5]

Md. Shah Fahad and his team (2015) designed a solar-plug in vehicle. They performed calculations on rolling resistance, traction force on each wheel, air resistance. They also used PV modules and the results obtained were perfect single person bearing go-kart. [6]

Now based on the extensive research work done in each and every components of the go-kart, our design covers most of the above ideas (which were really modern and advantageous) and also some new ideas and improved design methods which will result in making of an efficient go-kart. Keeping in mind the environmental concerns we should switch to renewable energy sources and combined power (or HYBRID VEHICLES) is the best option what we feel.

We admit that long term use of petrol is harmful for the environment but at times we require much more speed which solar panel may not produce in a given time limit. Also it will take some time for vehicles to fully convert them to solar-based vehicles. For example if we drive the go-kart in busy traffic conditions at daytime then we can switch the power source to solar energy as in traffic speed required is less. Also during night the solar energy stored in battery can be used for driving limiting the use of petrol to as low as possible.

The braking system and suspension system used is also technically sound and is able to stop the vehicle in least possible time and also can take relatively high loads respectively. The bearing is designed with high accuracy and their designed life is perfect for a go-kart. The chassis design and ergonomics has been performed keeping in mind driver's safety. The engine and transmission system used is both modern and technologically advanced.

# VI. DESIGN AND METHODOLOGY

Designing is one of the most important factors considering aesthetic, functional and dimensions. The design was based on driver ergonomics. A material of less weight was chosen so that the driver finds it easy to drive the vehicle optimizing the overall performance.

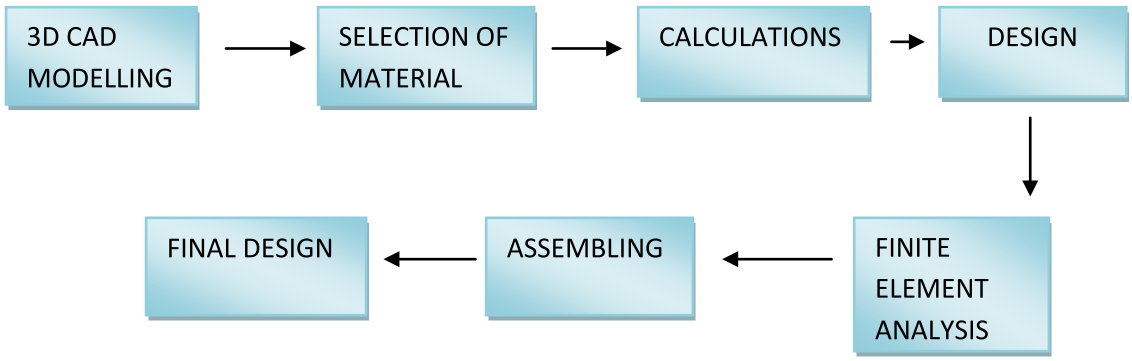



Fig. 1 Methodology

The design of the vehicle has been done keeping in mind the maximum acceleration required, speed, fuel economy, load to be carried etcetera. A finished automobile is a blend of all these components to produce the final product.

Many features of the vehicle are the result of engineer's preference which is usually based upon smooth performance of design. Extensive research has been done in order in design the vehicle that meets human needs. The next part of this paper shows the analysis and calculations of each and every component of the GO-KART starting with the chassis frame.

## VI.I CHASSIS FRAME

Chassis is a supportive framework made of tubular pipes and is of varying cross-section. It should have high strength so that it can withstand the weight of and the load of the vehicle. It also should have high rigidity to safeguard the components against action of different forces.

While selecting any chassis material and weight considerations are important factors. The material used in the chassis is 6063-T6 ALUMINIUM ALLOY (tempered). This material possesses good strength to weight ratio as well as corrosion resistance property due to availability of magnesium in the alloy and the material can also be easily anodized. Also materials like AISI 8620, AISI 1024 can be used for the chassis frame.

TABLE 1.MATERIAL PROPERTIES

| SRL. NO. | PROPERTY | VALUE |
|---|---|---|
| 1 | YOUNG'S MODULUS | 69.5 GPa |
| 2 | POISSON RATIO | 0.33 |
| 3 | TENSILE STRENGTH | 241 MPa |
| 4 | YIELD STRENGTH | 215 MPa |
| 5 | ULTIMATE STRENGTH | 220 MPa |
| 6 | THERMAL CONDUCTIVITY | 201 W/m-K |
| 7 | DENSITY | 2700 $Kg/m^3$ |
| 8 | MELTING POINT | 600 ° C |
| 9 | SHEAR MODULUS | 25.8 GPa |
| 10 | PROOF STRESS | 170 MPa |

TABLE 2.CHASSIS DIMENSIONS

| PARAMETERS | DIMENSIONS |
|---|---|
| **LENGTH** | **1510 mm** |
| **WHEEL BASE** | **1220 mm** |
| **WHEEL TRACK** | **1010 mm** |
| **PIVOT-CENTRE DISTANCE** | **800 mm** |
| **MATERIAL** | **6063 T6 Aluminium Alloy** |
| **OUTER TUBE DIAMETER** | **30 mm** |
| **INNER TUBE DIAMETER** | **24 mm** |
| **THICKNESS** | **3 mm** |
| **CHASSIS WEIGHT** | **28 Kilograms** |

The chassis frames in older designs were made very stiff in order to improve safety for the driver if involved in a collision. But technically this is not correct because the impact forces due to high deceleration are likely to cause serious damage.

The proposed design is free from such stiffness and moreover the chassis is of energy-absorbing type. This has been designed is such a manner that the front and rear end crumples in a concertina manner and thus absorbs the shock of the impact.

## CHASSIS DESIGN:

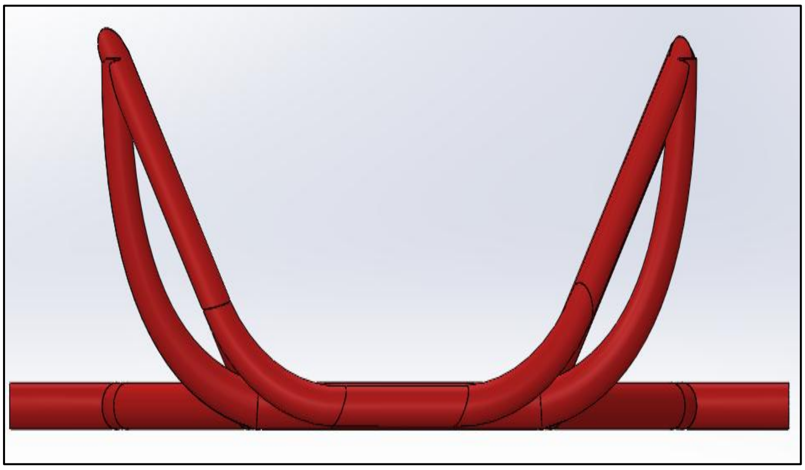


FIG. 2 FRONT VIEW

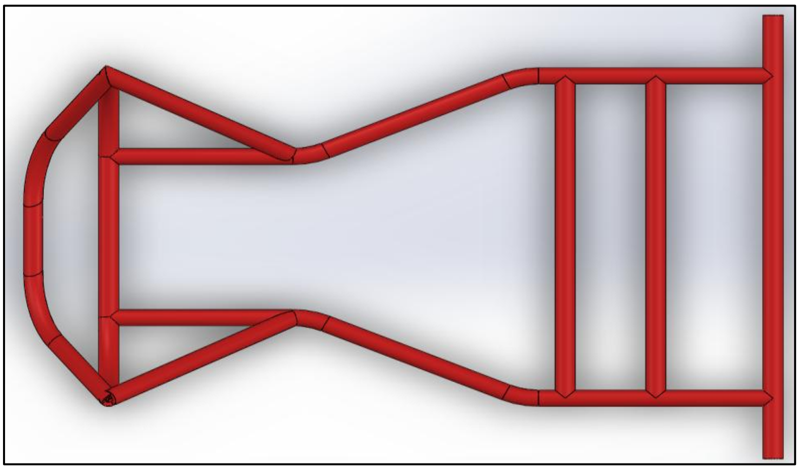


FIG. 3 TOP VIEW

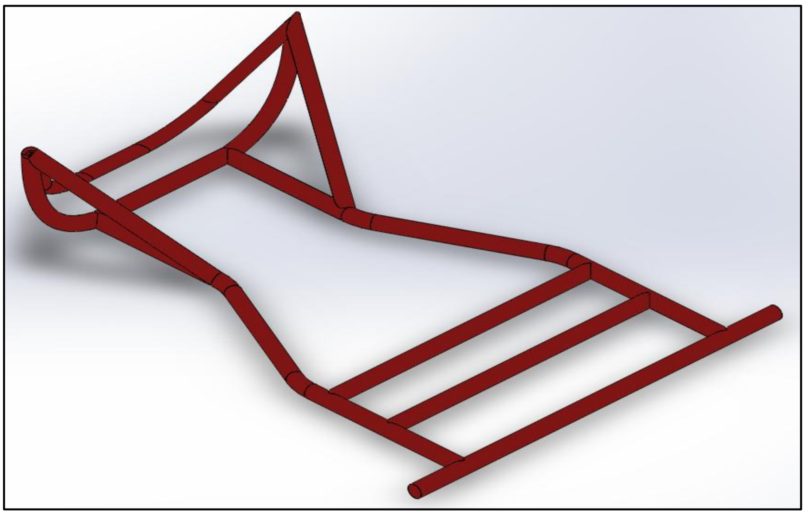


FIG. 4 ISOMETRIC VIEW

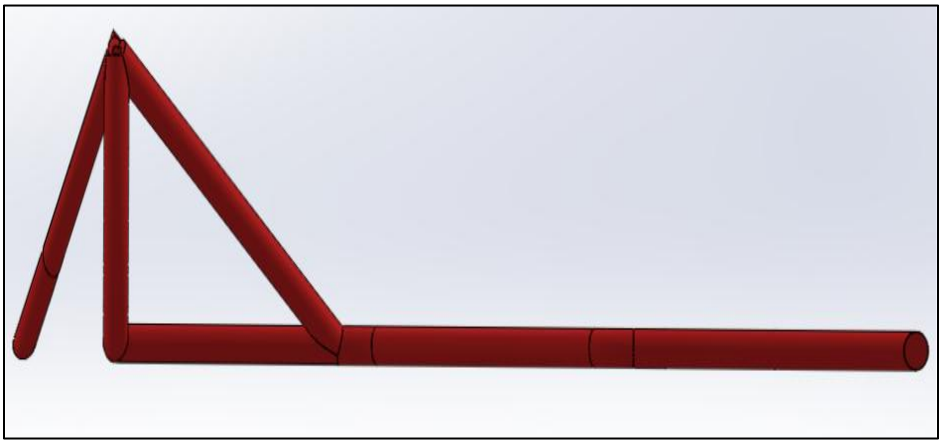


FIG. 5 SIDE VIEW

## CHASSIS CAD MODELS:

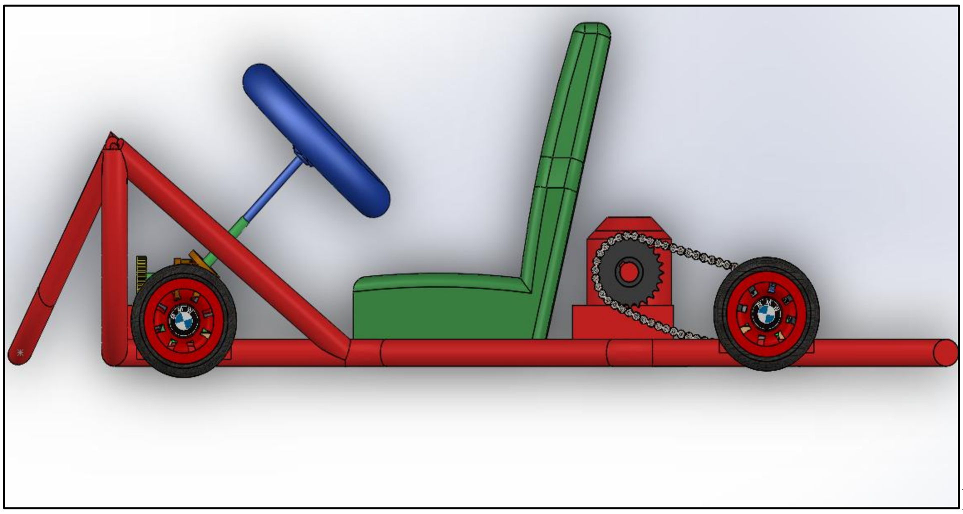


FIG. 6 SIDE VIEW

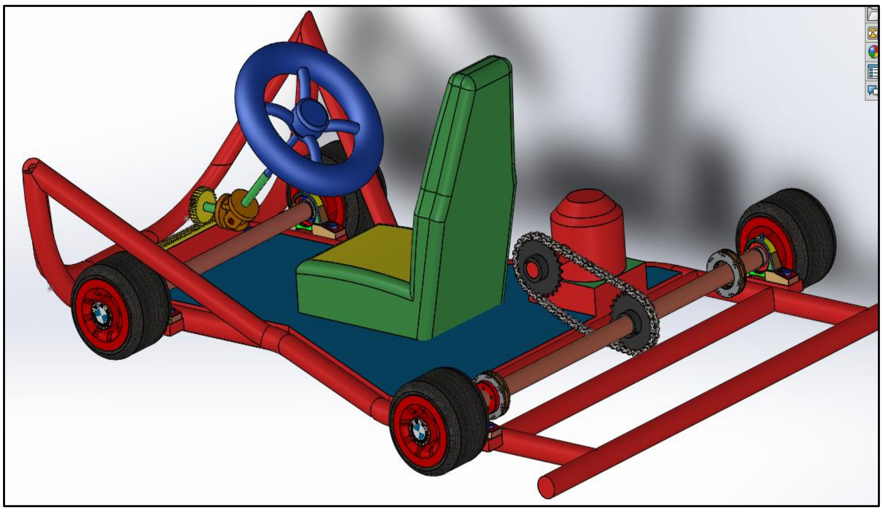


FIG. 7 ISOMETRIC VIEW

## FINITE ELEMENT ANALYSIS:-

FEA or Finite Element Analysis is a commonly used simulation method for multi-physics problem. This technique can be used to analyse forces, deformation, shear-moment diagram, vibration etc. ANSYS is one of such simulation software. The following analysis have been performed -

(a) REAR IMPACT :- For an ideal condition the acceleration is assumed as 2g (2×9.81). So the rear bump force on the vehicle body is, F= ma, where F = Impact force, m= total mass of vehicle (including the driver). Hence

$F = m \times a \Rightarrow F = 160 \times (2 \times 9.81) \Rightarrow F = 3140$ N

This amount of force is applied at the rear end keeping the frontal end fixed and the results are as:-

Maximum deformation = 0.004 m

Maximum equivalent stress = 130 MPa

Factor of Safety = 2.5

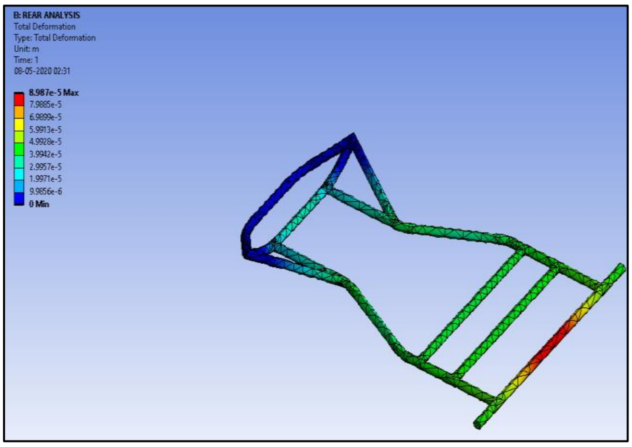


Fig. 8 Total Deformation

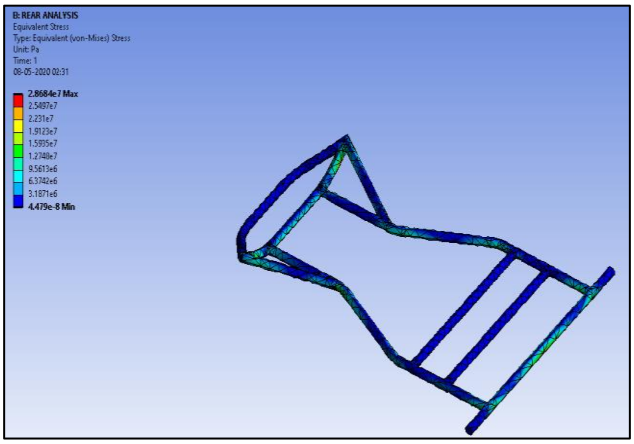


Fig. 9 Equivalent Stress

(b) FRONTAL IMPACT :- The frontal impact load is the same as the rear impact load. In this case the rear part is fully constrained. The stress and displacement is well within permissible limit. The results are as:-

Maximum deformation = $8.9\times10^{-5}$ m (0.089 mm)

Maximum elastic strain = 0.00018

Maximum strain energy = 0.0229 J

Max equivalent stress = 28.68 MPa

Factor of Safety = 2.5

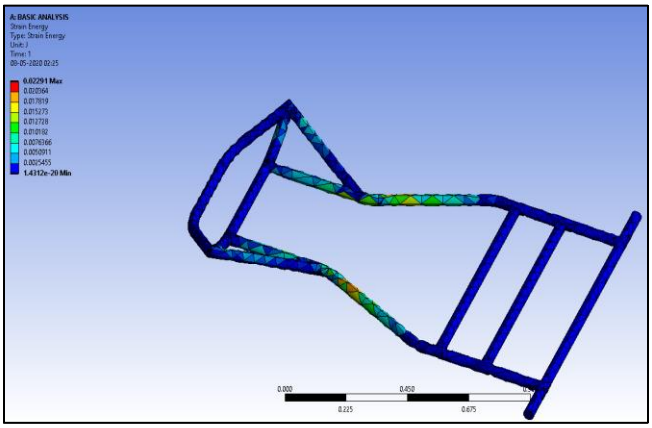


Fig. 10 Strain Energy

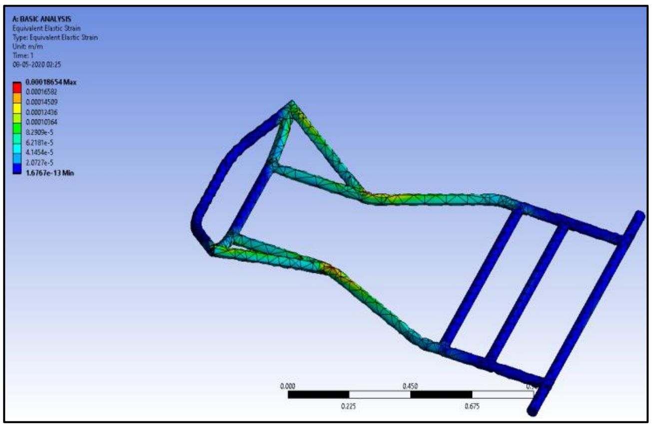


Fig. 11 Equivalent Stress

(c) TORSIONAL IMPACT :- When diagonally opposite front and rear wheels roll over bumps the two ends of the chassis are twisted in opposite directions so that they are subjected to longitudinal torsion. From the expression of torque T = r × F, we have r which is distance of C.G from either end of frame and F is the road reaction. Hence,

$T = r \times F \Rightarrow T = 0.36 \times 3020 \Rightarrow T = 1088$ Nm.

This amount of torque or twisting moment is provided on one side of chassis frame keeping the other end fixed and the results obtained are as under:-

Maximum deformation = $7.4\times10^{-6}$ m (0.0074 mm)

Equivalent strain = $1.23\times10^{-6}$

Safety factor = 2.5

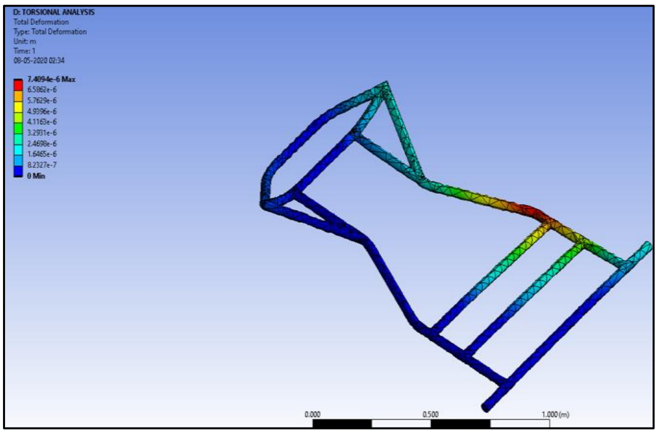


Fig. 12 Total Deformation

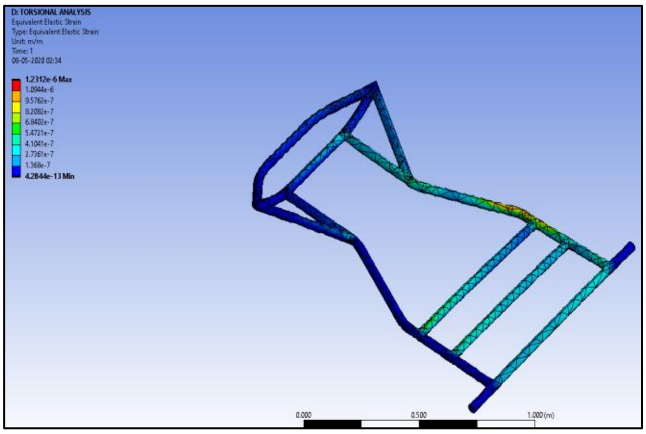


Fig. 13 Equivalent Strain

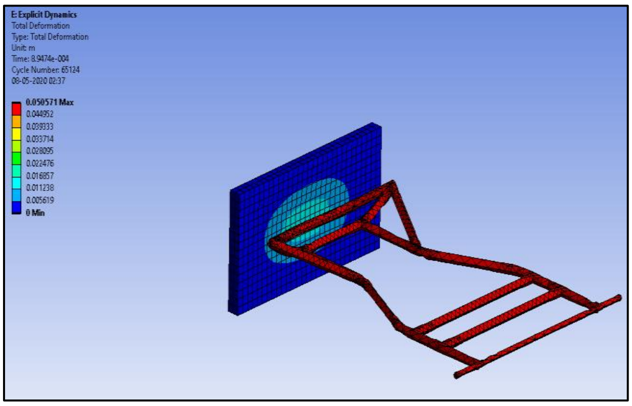
(d) DYNAMIC ANALYSIS:- This analysis has been done is response of chassis structure to head-on collisions. A concrete wall is chosen as test object on which the vehicle will dash into at some given velocity. A velocity of 16 m/sec (almost 57 Km/Hr) is given to the vehicle and the results are as under:

Maximum deformation = 0.05 m

Maximum equivalent stress = 5963 MPa

Maximum directional deformation = $2.52\times10^{-5}$ m

Safety factor = 2.6



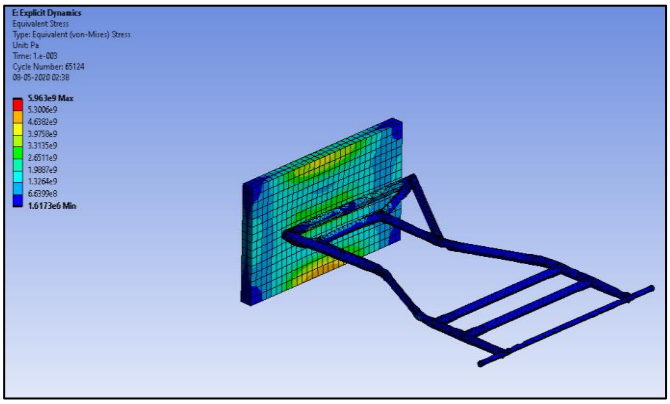


Fig. 14 (Total Deformation) Fig. 15 (Equivalent Stress)

(e) BENDING MOMENT DIAGRAM:-

Wheel base = 920 mm, UDL = 1 N/mm, Total length = 1510 mm

From the given condition, $R_1 + R_2 = 1569.6$ ------- Eqn. 1

Let $\Sigma M_{R2} = 0$;

So, Counter-clockwise moment = Clockwise moment -------- Eqn. 2

Therefore, $R_1 \times 920 = \{245.25 \times (920+150)\} + \{294.3 \times (620+150)\} + \{294.3 \times (150)\} - \{735.75 \times (150)\}$

$R_1 = 459.57$ N, $R_2 = 1110.02$ N

Bending Moment due to POINT LOAD –

BM at A = W × L = 245.25 × 150 = 36787.5 N-mm (CCW direction)

BM at B = W × L = 735.75 × 150 = 110362.5 N-mm (CW direction)

Bending Moment due to UDL –

BM at C = (w × L × L) / 2 = (1 × 150 × 150) / 2 = 11250 N-mm (CW direction)

BM at D = (w × L × L) / 2 = (1 × 150 × 150) / 2 = 11250 N-mm (CCW direction)

Hence, Maximum Bending moment = 110362.5 + 11250 = 121612.5 N-mm

From Bending Moment equation, $\sigma_b / y = M_b / I$ -------------- Eqn. 3

$M_b = 121612.5$ N-mm

I (for tubular cross-section) = $\pi/64\ (D_o^4 - D_i^4) = 23474.76\ mm^4$

$y = D_o / 2 = 15$ mm

From Eqn. 3 we have, $\sigma_{bending} = \sigma_{max} = 77.7$ MPa

Permissible bending stress for 6063-T6 alloy = 150 MPa = $\sigma_{permissible}$

So, the value of maximum bending stress is well within permissible limit.

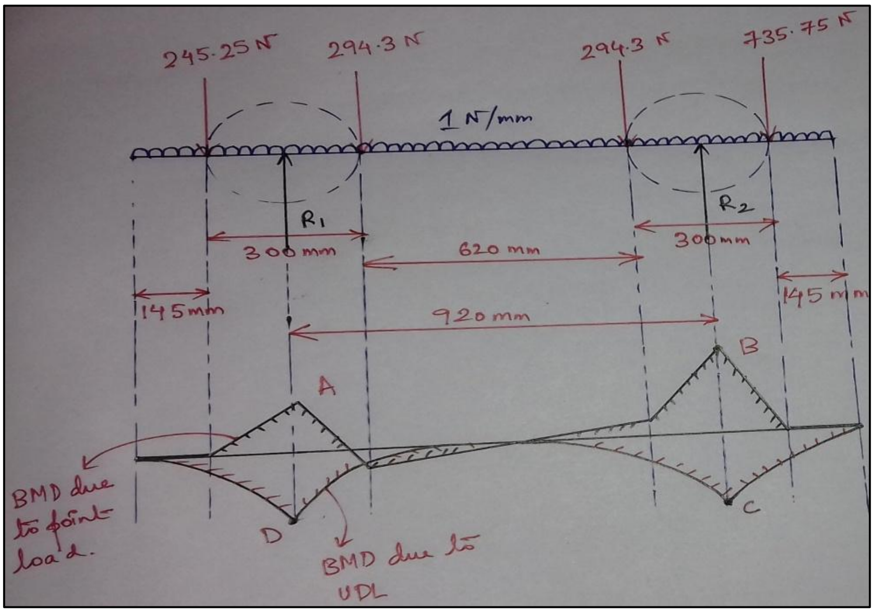



Fig.16 (Bending Moment Diagram)

(f) AIR RESISTANCE:- As understandable Air Resistance is the resistance imparted by air to the relative motion of the vehicle. The drag forces arise as a result of which the vehicle's motion is opposed and slowing down the vehicle eventually. Drag coefficient for go-kart lies between 0.25 to 0.30. So,

Drag resistance = $0.5 \times \rho \times A_{surface} \times V^2 \times C_D$, where

$\rho$ = air density = 1.225 $kg/m^3$

$A_{surface}$ = surface area of chassis = 7478.1 $mm^2$

V = relative velocity of vehicle

$C_D$ = drag coefficient = 0.25

Putting all the values we get,

Drag Resistance = 0.318 N.

# VI.II ENGINE PERFORMANCE

The Internal Combustion Engines are the most commonly used engines in automobiles and also in go-karts. The working fluid employed in these engines is either AIR and PETROL or AIR and DIESEL. But keeping in mind the environmental standards and also the harm caused by the DIESEL fuel we have used PETROL in this go-kart.

The engine used in this go-kart is HONDA GX270 9HP Commercial Petrol Engine. 4- Stroke engine is used because it shows higher thermal efficiency and also it is good from racing point of view. The power from the engine is transmitted via chain drive to the rear shaft. Chain drive is used because it provides the highest efficiency (about 98 %) and is able to withstand shocks.

TABLE 3.ENGINE SPECIFICATIONS

| PARAMETERS | DIMENSIONS/VALUES |
|---|---|
| **ENGINE TYPE** | **4-STROKE, OVER HEAD VALVE (OHV) TYPE** |
| **BORE , STROKE** | **77 mm, 58 mm** |
| **COOLING TYPE** | **Naturally aspirated; AIR-COOLED** |
| **CYLINDER (S)** | **1** |
| **DISPLACEMENT** | **270 $cm^3$** |
| **POWER OUTPUT** | **8.5 BHP (6.3 KW) @ 3600 RPM** |
| **MAXIMUM TORQUE** | **19.1 N-m @ 2500 RPM** |
| **COMPRESSION RATIO** | **8.5 : 1** |
| **IGNITION SYSTEM** | **DIGITAL CDI TYPE** |
| **STARTING SYSTEM** | **RECOIL / ELECTRIC** |
| **SHAFT ORIENTATION** | **HORIZONTAL** |
| **FUEL TANK CAPACITY** | **5.3 Litres** |
| **ENGINE OIL CAPACITY** | **1.1 Litres** |
| **FUEL** | **UNLEADED GASOLINE** |
| **OCTANE RATING** | **86 or higher** |
| **DRY WEIGHT** | **25 Kg** |
| **DIMENSIONS** | **(381 × 428 × 422) $mm^3$** |


Fig.17 HONDA GX270

ENGINE CALCULATIONS –

Power = 8.5 HP = 6.3 KW

Engine rpm at max torque = 2500

Wheel Rim diameter = 300 mm = 0.3 m = D

Wheel velocity = 16.66 m/sec

We have, $v = (\pi \times D \times N) / 60$

$N_{WHEEL}$ = 1060.60 rpm

GEAR RATIO = Engine RPM / Wheel RPM = 2500 / 1060.6 = 2.35

TRANSMISSION SYSTEM DESIGN - The mechanism by which engine power is transmitted to the driving wheels is known as transmission system. There are mainly two types of transmission used today in commercial vehicles viz. Manual and Automatic. Here we have used AUTOMATIC TRANSMISSION system. The function of transmission system is –

1. To provide the necessary torque to move the vehicle as per the road and load conditions

2. To enable the driving wheel to rotate at different speeds

3. To disconnect the engine from the drive wheels

Main components of an Automatic Transmission includes:-

a) Planetary gear set – Mechanical drive which can produce range of gear ratios for forward as well as for reverse.

b) Hydraulic system – ATF or Automatic Transmission Fluid sent under pressure by an oil pump to control the fluid clutch and band to control the planetary gear train.

c) Torque converter – Torque converter multiplies torque through the force of fluid movement. It provides a smooth automatic vehicle pick up from standstill. It has three functional parts; the driver or the impeller, turbine or the driven member and the stator.

(d) Governor and modulator – Throttle cable monitor speed and throttle position to determine when to shift.

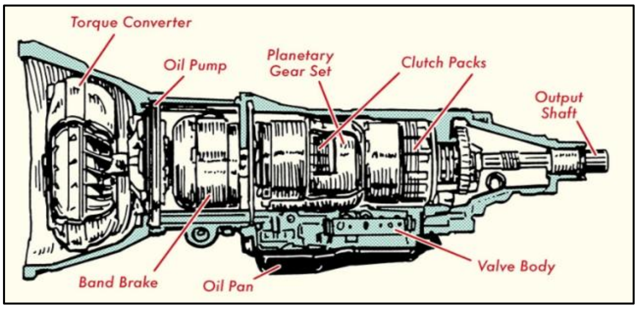



Fig.18 Automatic Transmission System

The following part shows the design and calculations of the chain drive. Chain drive chosen here is 08 B (British Standard). The material chosen for both the sprockets is CAST IRON and for the chain ALLOY STEEL.

Chain pitch = 12.7 mm (B.S)

Centre Distance = $30p < a < 50p$

$= 381 < a < 635 = 420$ mm (assuming from geometry)

Number of teeth on driven sprocket = 21 = $z_1$

Number of teeth on driving sprocket = 53 (considering a velocity ratio of 2.5) = $z_2$

Diameter of larger sprocket (installed on engine shaft) = pitch / sin (180 / 53) = 214 mm

Diameter of smaller sprocket (installed on rear shaft) = pitch / sin (180 / 21) = 85 mm

Number of links ($L_n$) = $2(a/p) + [(z_1 + z_2)/2] + [(z_2 - z_1)^2 / 4\pi^2] \times (p / a)$

= 103.92 = 104 (as number of links should be even number)

Length of chain (L) = Number of links × pitch

= 104 × 12.7

= 1320.8 mm

Actual centre distance value (a) = $p/4 \{[L_n - (z_1+z_2)/4] + ([L_n - (z_1+z_2)/2]^2 - 8[z_2-z_1]^2/4\pi^2])^{0.5}\}$

= 420.46 mm

The actual centre distance value is almost equal to the assumed value. Hence the design is satisfactory.

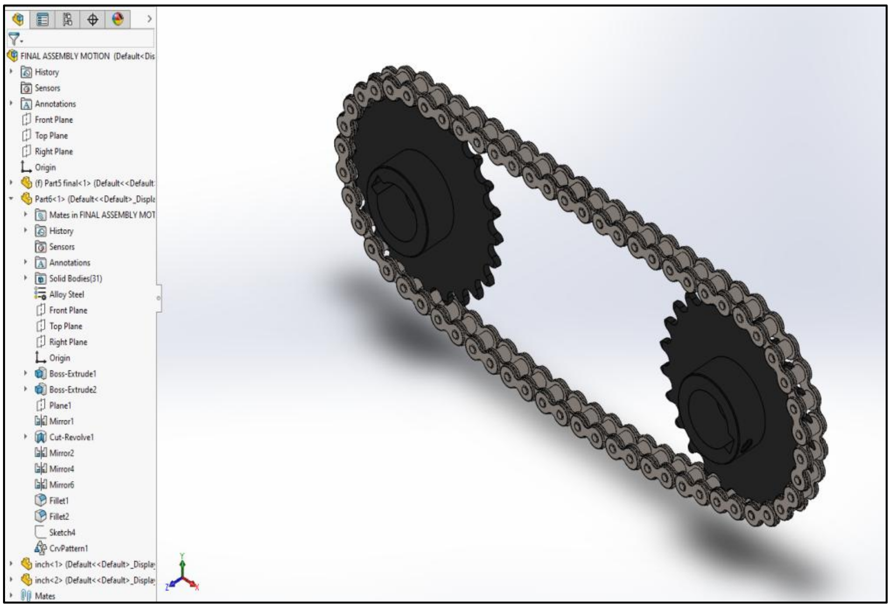


Fig. 19 Chain Drive

Engine output shaft angular speed = $(2\pi N)/60$, where N = 2500 RPM

= 261.8 radians / sec

Now, Tractive force between wheel and road surface = $\mu_t \times W$ ---- (1)

$\mu_t$ = traction coefficient = 0.5 (for wet rolled gravelled surface)

W = total weight of vehicle on surface = 160 × 9.81 = 1569.6 N

Therefore total tractive effort available = ($\mu_t \times W$) = 0.5 × 1569.6 = 784.8 N

Starting torque =Total tractive effort × wheel radius

= (784.8 × 0.3) N-m

= 235.44 N-m

Tractive effort per wheel (F) = 196.2 N

So, tractive effort from both the rear wheels = 196.2 × 2 = 392.4 N

Maximum acceleration of the vehicle (a) = F / mass of vehicle

= 2.45 m/sec$^2$

Now acceleration = dv / dt

Minimum time required to reach maximum velocity = dv / a

$T_{min}$ = 16.67 / 2.45 = 6.8 seconds

GEAR TRAIN:-

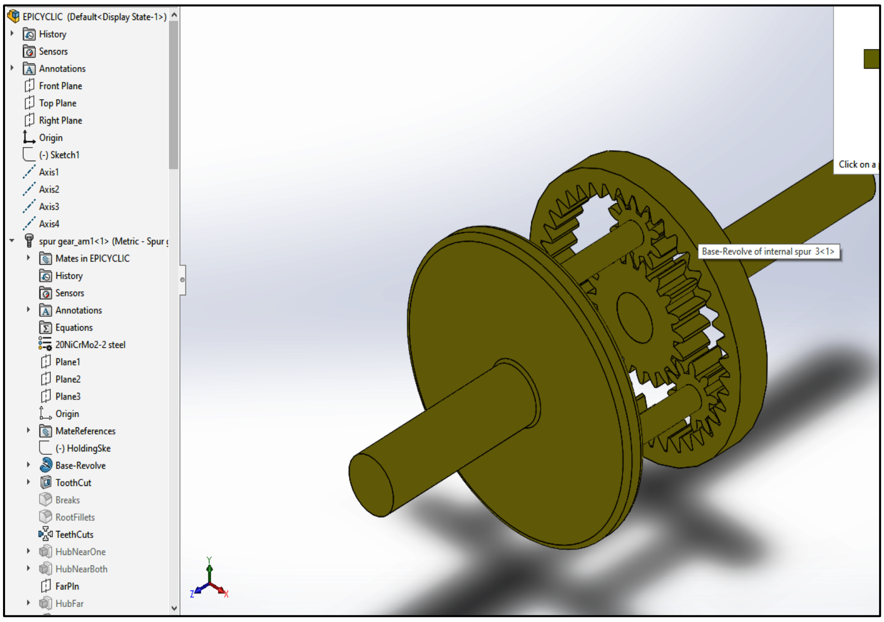



Fig. 20 Planetary Gear Train

Number of teeth in sun gear = 20

Number of teeth in ring gear = 40

Number of teeth in planet gear = 10

So, top gear ratio = 1 + ($T_{SUN}$ / $T_{RING}$) = 1 + (20/40) = 1.5

Chain drive velocity (gear) ratio = 2.5

Overall gear ratio = 2.5 × 1.5 = 3.75

Output torque at sprocket = 3.75 × Input Torque = 3.75 × 19.1 = 71.62 N-m

Minimum force on rear axle = Output torque / sprocket radius

= 71.62 / 0.23862

= 300.14 N

Minimum torque on rear axle = 300.14 × shaft diameter

= 300.14 × 0.03

= 9.0 N-m

Minimum torque at rear wheel = 300.14 × 0.3

= 90.04 N-m

Lowest gear ratio = 1 + ($T_{ANNULAR}$ / $T_{SUN}$) = 1+ (40/20) = 3

Chan drive gear ratio = 2.5

Overall gear ratio = 2.5 × 3 = 7.5

Output torque at sprocket = 7.5 × 47.75

= 358.12 N-m

Maximum force on rear axle = Output torque / sprocket radius

= 358.12 / 0.23862

= 1500.8 N

Maximum torque on rear axle = 1500.8 × 0.03

= 45 N-m

Maximum torque on rear wheel = 1500.8 × 0.3

= 450.24 N-m

So,

Maximum force acting on rear axle = 1500.8 N

Minimum force acting on rear axle = 300.14 N

STATIC ANALYSIS OF SHAFT:-

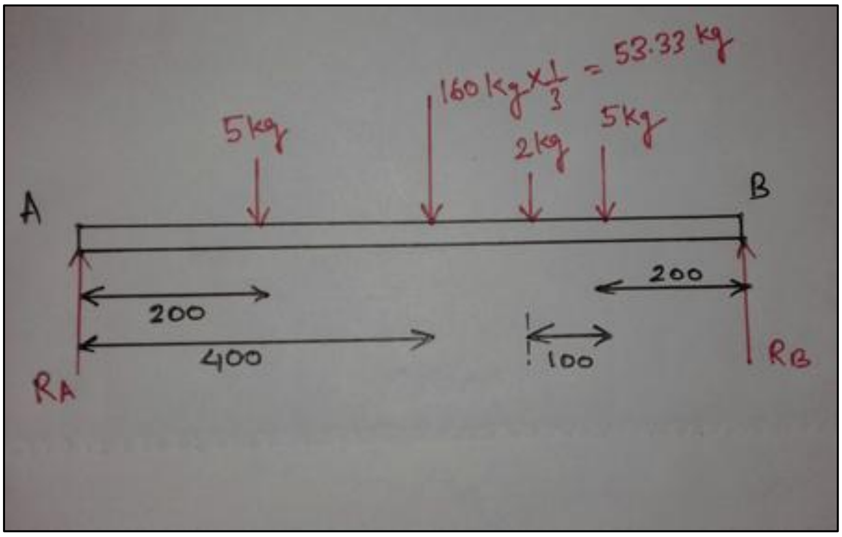



Fig. 21 Static Loading

From the given condition,

$R_A + R_B = 635.68$ N ------- eqn. 1

$\Sigma M_A = 0$;

$R_B \times 800 = 49.05 \times 200 + 19.62 \times 500 + 523.16 \times 400 + 49.05 \times 600$

$R_B = 322.89$ N

From 1, $R_A = 312.78$ N

So, Maximum bending moment = (523.16 × 500) – (312.78 × 200) = 199024 N-mm

Maximum torsion = 45 N-m = 45000 N-mm

Shaft diameter calculation (d)

From Maximum shear stress theory we have,

$\tau max = 16 / \pi d^3 (M_B + M_T)^{0.5}$ ----------------- eqn. 1

$\tau max = 0.3 \times S_{yt} = 0.3 \times 460 = 138$ MPa

$\tau max = 0.18 \times S_{ut} = 0.18 \times 560 = 100.8$ MPa

We are to consider the minimum of these two values i.e. 100.8 MPa

If there are keyways on the shaft, then

$\tau max = 0.75 \times 100.8 = 75.6$ MPa

$M_B + M_T = 244024$ N-mm

Putting all the values in equation 1, we get

Shaft diameter (d) = 23.95 mm = 24 mm

Considering entire load to be acting on the rear axle the shaft diameter is taken as 30 mm (FOS = 3)

## SOLAR PANEL

Solar energy is the energy available from the sun in the form of solar irradiation. Solar vehicles harness solar energy from the sun using solar panels. A solar panel is a packaged assembly of photovoltaic cells that can convert solar energy to electrical energy which can be stored in batteries. They are noiseless and pollution free and also requires very little maintenance.

India receives solar energy of more than 5000 trillion KWh per year, which is much more than total annual consumption. Though the energy density is low (about 1 $kW/m^2$) it has now become possible to harness this abundantly available energy very reliably. Energy is produced in the sun by fusion reaction i.e. 4 hydrogen atoms combines to form 1 helium atom and releasing 26.7 Mega electron Volt of energy.

NEED FOR SOLAR ENERGY – Energy is universally recognized as one of the most important input for economy and human development. It is the main driving force in all sectors. One of the major challenges the world is facing is that almost 15 % people live without any access to modern day energy.  Also in the recent year’s excessive use of fossil fuels (viz. petrol, diesel, coal etc) have led to global warming, ozone layer depletion and environmental imbalance. These resources are limited in quantity which is being depleted at a faster rate which will leave us mineral-scarcity in the near future. So it’s high time that we should switch to non-conventional energy sources which are pollution free as well as easily available in ample quantity.

Some advantages of solar energy (non-conventional) over conventional power sources are:

a) It converts solar power directly into mechanical power hence no mechanical linkage is required.

b) Available in ample quantity.

c) They are reliable and durable and generally maintenance free

d) They have much longer life span (about 20 years)

The solar panel used here is of type 100W MONOCRYSTALLINE (manufacturer RENOGY 100D). This variety produces the highest efficiency per unit area. It is available both in 12 Volts as well as 24 Volts variants. The advantages of this variant used are:-

a) Highest PTC rating

b) No hotspots

c) Quick and inexpensive mounting

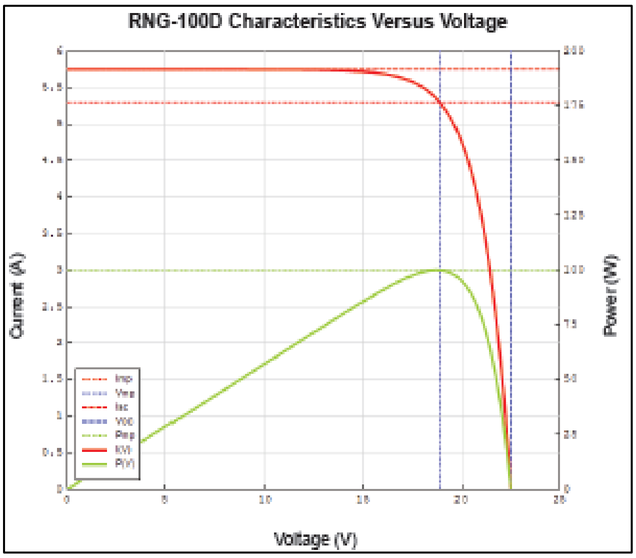



Fig.22 I-V characteristic

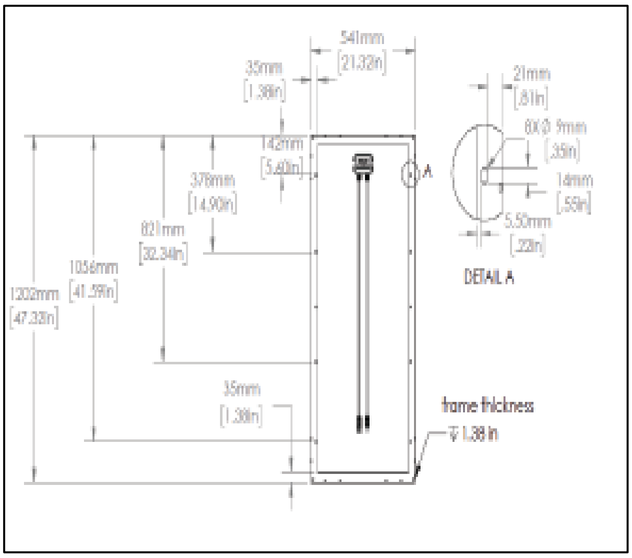



Fig.23 Module Circuit

TABLE 4A.PANEL SPECIFICATIONS (ELECTICAL DATA)

| SERIAL NUMBER | PARAMETERS | VALUES |
|---|---|---|
| 1 | **MAXIMUM POWER (at STC)** | **100 WATTS** |
| 2 | **OPERATING VOLTAGE** | **18.9 VOLTS** |
| 3 | **OPERATING CURRENT** | **5.29 AMPERES** |
| 4 | **OPEN CIRCUIT VOLTAGE** | **22.5 VOLTS** |
| 5 | **SHORT CIRCUIT CURRENT** | **5.75 AMPERES** |
| 6 | **MODULE EFFICIENCY** | **16 %** |
| 7 | **MAXIMUM SYSTEM VOLTAGE** | **600 VOLTS** |
| 8 | **SERIES FUSE RATING** | **15 AMPERES** |

TABLE 4B.PANEL SPECIFICATIONS (MECHANICAL DATA)

| SERIAL NUMBER | PARAMETER | VALUES |
|---|---|---|
| 1 | **SOLAR CELL TYPE** | **MONOCRYSTALLINE SQUARE (12.5 cm × 12.5 cm)** |
| 2 | **NUMBER OF CELLS** | **36** |
| 3 | **DIMENSIONS** | **1202 mm × 541 mm × 35 mm** |
| 4 | **WEIGHT** | **7.5 Kilograms** |
| 5 | **FRONT GLASS** | **TEMPERED (3.2 mm)** |
| 6 | **FRAME** | **ALUMINIUM ALLOY (anodized)** |
| 7 | **CONNECTOR** | **MC4 type** |
| 8 | **FIRE RATING** | **Type C** |

TABLE 4C.PANEL SPECIFICATIONS (THERMAL DATA)

| SERIAL NUMBER | PARAMETERS | VALUES |
|---|---|---|
| 1 | **OPERATING MODULE TEMPERATURE** | **-40°C to 80°C** |
| 2 | **NORMAL OPERATING CELL TEMPERATURE** | **45°C to 49°C** |
| 3 | **TEMPERATURE COEFFICIENT OF $P_{MAX}$** | **-0.44% / °C** |
| 4 | **TEMPERATURE COEFFICIENT OF $V_{OC}$** | **-0.30% / °C** |
| 5 | **TEMPERATURE COEFFICIENT OF $I_{SC}$** | **0.04% / °C** |

TABLE 4D. JUNCTION BOX SPECIFICATIONS

| SERIAL NUMBER | PARAMETERS | VALUES |
|---|---|---|
| 1 | **IP RATING** | **IP 65** |
| 2 | **DIODE TYPE** | **HY 10SQ050** |
| 3 | **NUMBER OF DIODES** | **2** |

TABLE 4E.DC MOTOR AND BATTERY SPECIFICATIONS

| SERIAL NUMBER | PARAMETERS | VALUES |
|---|---|---|
| 1 | **POWER RATING** | **600 WATTS** |
| 2 | **MOTOR RPM** | **2750** |
| 3 | **OPERATING VOLTAGE** | **48 VOLTS** |
| 4 | **BATTERY** | **4×12 Volts; 16 AH capacity** |
| 5 | **BATTERY TYPE** | **LEAD-ACID** |
| 6 | **TOP SPEED** | **27 KMPH** |

CALCULATIONS:-

Energy consumed by motor in 4 hours = 600 × 4 = 2400 Watt hours = 2.4 KWh

Energy produced by solar panel per day = Panel wattage × sunshine hours × 0.75(loss factor)

= 100 × 8 × 0.75 = 600 Watt hours

Energy stored by battery = Capacity × Voltage = 16 × 48 = 768 Watt hours

Discharging time (from 100 % to 0 %) = (768 / 600) = 1.28 hours (without power loss)

Discharging time (from 100 % to 0 %) = (768 / 600) × 0.75 = 1 hour (with power loss)

Power of panel = Load / Sunshine hours = 2400 / 8 = 300 Watts

Charging time (from 0 % to 100 %) = (Energy stored by Battery / Power of panel) = 2.56 hours

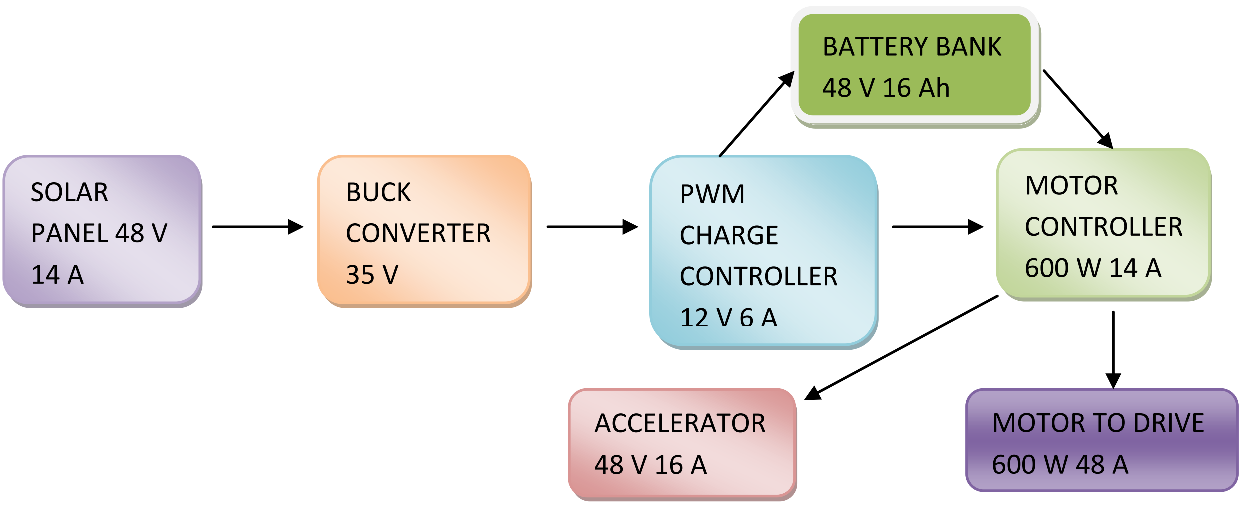
Charging time (from 0 % to 100 %) = (Energy stored by Battery / Power of panel) × 1.8 = 4.6 hours

(Considering a power loss factor of 1.8)

TABLE 4F.PETROL CAR versus SOLAR CAR

| POINT OF DISTINCTION | PETROL CAR | SOLAR CAR |
|---|---|---|
| **FUEL** | **GASOLINE** | **SOLAR ENERGY** |
| **FUEL UNIT RATE** | **INR 73/litre** | **INR 2.5 /KWh** |
| **MILEAGE** | **15 KMPL** | **30 KM per charge** |
| **AVERAGE SPENDING ON FUEL/YEAR** | **INR 13,450 (approx)** | **INR 540 (approx)** |
| **TOTAL COST PER YEAR** | **INR 20000 (approx)** | **INR 10000 (approx)** |

CIRCUIT DIAGRAM:



# VI.III DISC BRAKE

Brakes are mechanical devices for increasing the frictional resistance that retards the turning motion of vehicle wheels. It absorbs either potential or kinetic energy and dissipates this in the form of heat. In this go kart DISC BRAKES are used because:-

a) Disc brake produces constant braking

b) Design is much simpler compared to drum brakes

c) There is no self-locking in disc brakes

d) Uniform pad wear due to flat friction contact between disc and pads.

This brake assembly consists of two annular plates ribbed together by radial vanes which act as heat sink. Ventilation in the plates increases convection heat transfer by 70 %. The material chosen for this brake is CAST IRON (CI).

TABLE 5.MATERIAL PROPERTIES

| SERIAL NUMBER | PROPERTIES | VALUES |
|---|---|---|
| 1 | YOUNG'S MODULUS | 147 GPa |
| 2 | POISSON RATIO | 0.287 |
| 3 | YIELD TENSILE STRENGTH | 449 GPa |
| 4 | ULTIMATE TENSILE STRENGTH | 479 GPa |
| 5 | SHEAR MODULUS | 58.4 GPa |
| 6 | MELTING POINT | 1150 ° C |
| 7 | THERMAL CONDUCTIVITY | 26.6 W/m-K |
| 8 | DENSITY | 7220 Kg/$m^3$ |
| 9 | CTE | $12.7 \times 10^{-6}$ °$C^{-1}$ |
| 10 | SPECIFIC HEAT | 506 J / Kg-K |

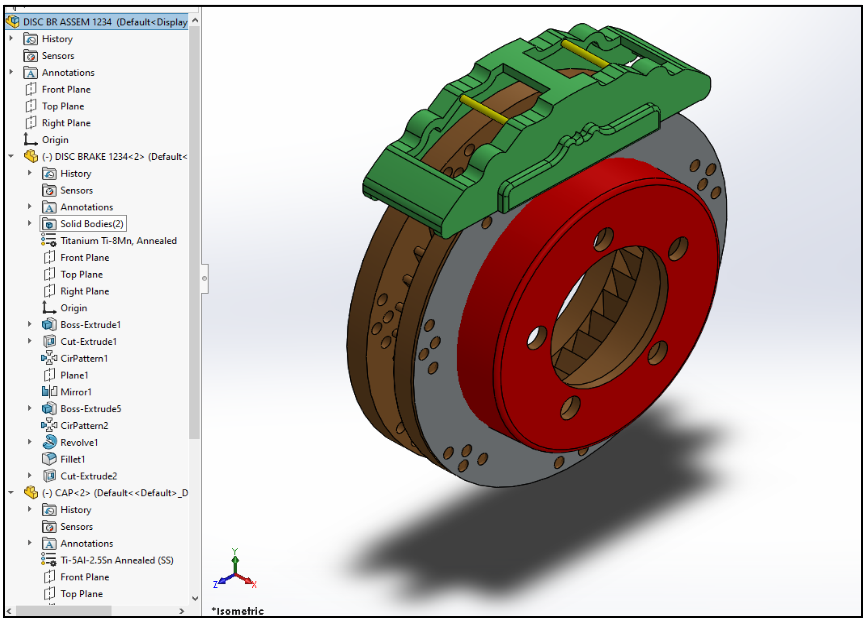


Fig. 24 ISOMETRIC VIEW

## FINITE ELEMENT ANALYSIS:-

The following part shows the Steady-state thermal analysis of the disc brake in ANSYS and the results are shown here.

a) Temperature Analysis: - A temperature of 32ºC was given on 8 faces of the disc and results are –

Maximum temperature = 1002.5ºC

b) Maximum heat flux: - Heat flux of 30 $KW/m^2$ and a convection film coefficient of 230 $W/m^2$ ºC and the results obtained are -

Maximum heat flux = 4640 $KW/m^2$

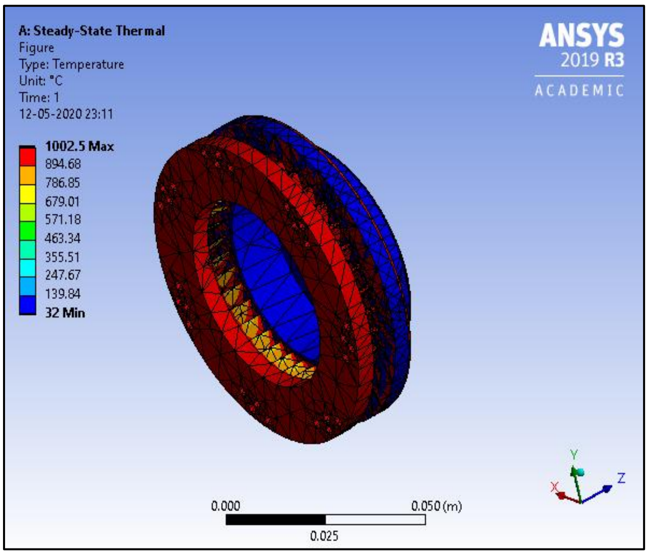



Fig. 25 Temperature Distribution

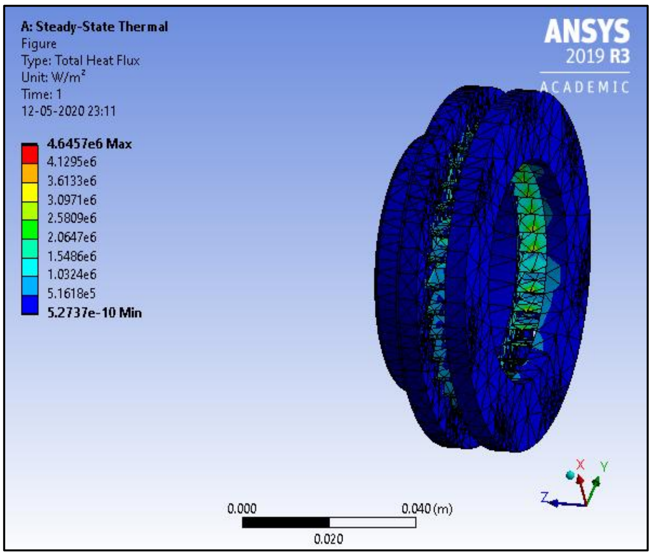



Fig.26 Total Heat Flux

The following part shows the calculations of various braking fundamentals.

Kinetic energy of a vehicle during braking = 0.5 × M × $(U^2 – V^2)$, where

M = mass of vehicle = 160 Kg (with the driver)

V = final velocity after braking = 0

U = initial velocity = 16.67 m/sec

So, K.E = 22231.11 J = 22.23 KJ

The braking of vehicle occurs at ground level so effective braking force acts on the ground which causes the vehicle to pitch forward so the dynamic weight transfer from rear wheel to front wheel is

$W_T = [\{(\mu h) / b\} \times [f / g] \times W]$, where

μ = adhesion coefficient = 0.6

b = wheel base = 1220 mm = 1.22 m

h = height of centre of gravity from ground = b/3

W = weight of car = M × g = 160 × 9.81 = 1569.6 N

f = deceleration of vehicle = μg (for all wheel braking) = 0.6 × 9.81 = 5.88 $m/s^2$

g = 9.81 $m/s^2$

So, Dynamic weight transfer = 188.35 N

Let the static load on front axle = 0.4 × vehicle weight = 627.84 N

Let the static load on rear axle = 0.6 × vehicle weight = 941.76 N

So, Dynamic weight on front axle = Static load + Dynamic load

= 627.84 + 188.35 = 816.19 N

Dynamic weight on each front wheel ($R_F$) = 408 N

Dynamic weight on rear axle = Static load + Dynamic load

= 941.76 + 188.35 = 1130.11 N

Dynamic weight on each rear wheel ($R_R$) = 565 N

Frictional force on each front wheel = μ × $R_F$ = 0.6 × 408 = 244.8 N

Frictional force on each rear wheel = μ × $R_R$ = 0.6 × 565 = 339 N

Braking torque at front wheel = $F_F$ × Rolling radius

= 244.8 × 0.324

= 79.31 N-m

Braking torque at rear wheel = $F_R$ × Rolling radius

= 339 × 0.324

= 110 N-m

HYDRAULIC BRAKING SYSTEM: - A hydraulic braking system transmits pedal force to the wheel brakes through pressurized fluid. This fluid pressure is equally transmitted throughout the fluid to front and rear disc callipers (not shown). Some advantages of this system are –

a) Provides equal braking effort on all wheels

b) Efficiency is greater the mechanical system

c) Each brake receives its full share of pedal effort

Hydraulic braking system works on the principle of Pascal's Law which states that pressure applied to a fluid at rest is transmitted equally in all directions within the container i.e.

$F_1 / A_1 = F_2 / A_2$

Master cylinder bore diameter = 20 mm

Master cylinder area = $0.7854 \times 20^2 = 314\ mm^2$

Foot pedal force = 120 N (assumption)

Force on master cylinder = Force on foot pedal × Leverage ratio

= 120 × 4

= 480 N

Pressure acting on master cylinder = $F_1 / A_1$

= 480 / 0.000314

= 1.53 MPa

Pressure in brake pipe lines (p) = Pedal force / master cylinder area

= 120 / 0.000314

= 0.382 MPa

Braking Torque (for non servo systems) = $T_B = 2P \times a \times \mu \times R \times n$, where

P = brake fluid pressure = 0.382 MPa

a = area of piston per calliper = $0.7854 \times 0.045^2 = 1.59 \times 10^{-3}\ m^2$

$\mu$ = friction coefficient of pad material = 0.4

R = mean radius = 90 mm = 0.09 m

n = 4

So, $T_B$ = 174.92 N-m

Clamping Force = $T_B / 2\mu R$ = 2429.44 N

Clamping force for 4 wheels = 4 × 2429.44 = 9717.76

Deceleration = - (force / mass of vehicle)

= - (9717.76 / 160)

= - 60.73 radians/sec$^2$

Stopping distance (S) = $v^2 – u^2 / 2f$, where

V = final velocity after braking = 0

U = initial velocity = 16.67 m/s

So, S = 2.28 meters

Force on each front cylinder piston = p × area of front pistons

$$= 0.382 \times 10^6 \times (0.7854 \times 0.045^2)$$

$$= 607.81 \text{ N}$$

Force on each rear cylinder piston = p × area of rear piston

$$= 0.382 \times 10^6 \times (0.7854 \times 0.0315^2)$$

$$= 297.7 \text{ N}$$

Total force on front brakes = 607.81 × 4 = 2431.34 N

Total force on rear brakes = 297.7 × 4 = 1190.8 N

The brake fluid used in most braking system is glycerine-alcohol fluid with certain additives. These fluids are hygroscopic in nature which is why vapour locking may arise due to presence of water and high temperature generated due to braking. Some special silicone based fluids are used to overcome this problem. Some major characteristics of brake fluid are low viscosity, resistance to chemical ageing, high boiling point, non-reactive to rubber components etc.

TABLE 6.CALCULATED VALUES

| PARAMETER | VALUES |
|---|---|
| **MAXIMUM SPEED** | **60 KMPH (16.67 m/sec)** |
| **PEDAL EFFORT** | **120 N** |
| **ADHESION FACTOR** | **0.6** |
| **FRICTION FACTOR (BRAKE PAD)** | **0.4** |
| **STOPPING DISTANCE** | **2.28 m** |
| **STOPPING TIME** | **0.274 s** |
| **PEDAL LEVERAGE RATIO** | **4** |

Friction pads are one of the components used in disc brake which is sandwiched between each piston and the disc face and they are held by retaining pins. Materials used in these pads are usually asbestos-free. Sintered materials have long life but low friction coefficient. Ceramics cause wear and tear on discs. Nowadays DuPont’s Kevlar, a high strength fibre is used.

# VI.IV STEERING ASSEMBLY

The function of steering system is to convert the rotary motion of the steering wheel into angular movement of the front wheel. The assembly is designed to enable the driver to control and continuously adjust the steered path of the vehicle. The steering type used in this vehicle is electronic operated rack and pinion steering gear mechanism. The components required are:-

a) Steering column and wheel

b) Universal joint

c) Steering gears

d) Electronic Control Unit (ECU) (not shown)

e) Torque sensor and motor (not shown)

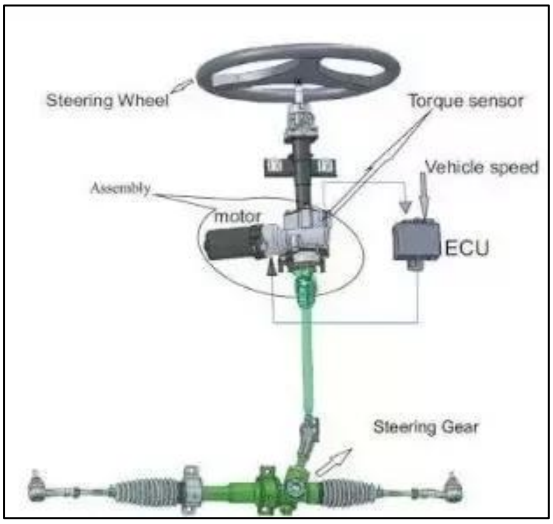



Fig.28 Steering Assembly

Universal Joint – These are capable of transmitting torque and rotational motion when shafts are inclined at an angle at each other and which may constantly vary under working conditions. The steering shaft connects the steering wheel to the gear unit using a universal joint coupling.

ECU – Electronic Control Unit is the most intelligent part of electric steering system it calculates optimal assistance and sends it to the electric motor based on the signal received from torque sensor.

Torque Sensor – This is basically a transducer which converts torque to electric signals. The sensor contains magneto-resistive elements which can sense field direction changes.

Steering gears – This enables the driver to exert a large force at the road wheel with minimum effort and to control the direction of the wheel.

## FINITE ELEMENT ANALYSIS:-

The following part shows the static structural analysis of universal joint and steering gears. The material used for universal joint is Stainless steel (ferritic) and AISI 304 for gear components.

a) Static structural – Both the ends plates are fixed. A twisting moment of 200 N-m and a force of 150 N are applied on the component and the results are as under:-

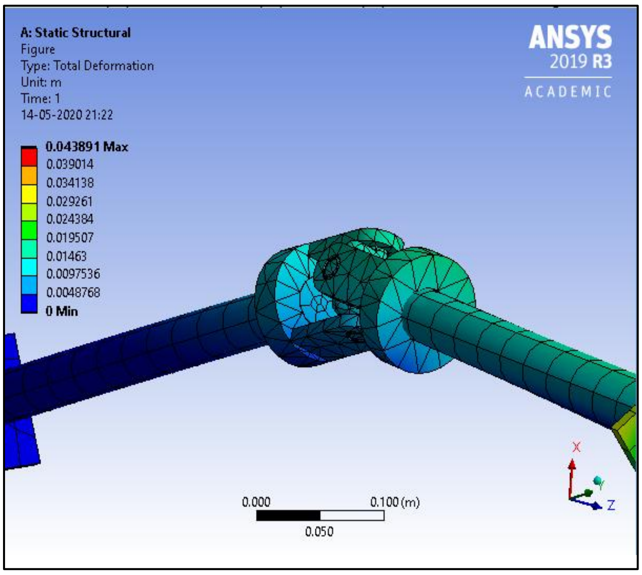



Fig. 29 Total Deformation

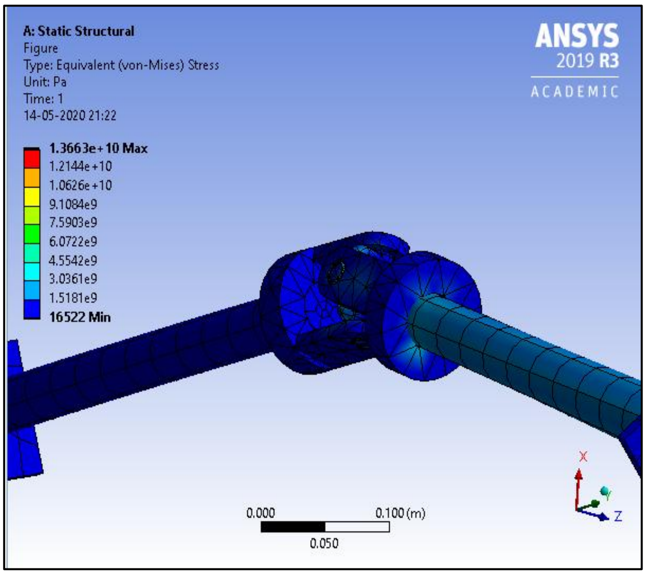



Fig. 30 Equivalent stress

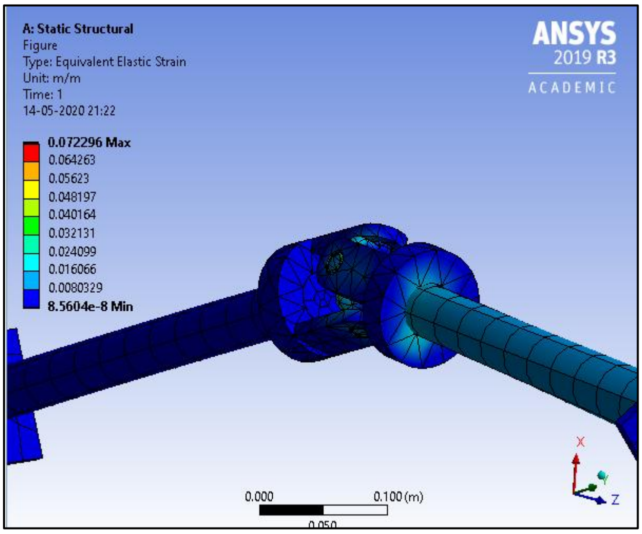



Fig.31 Equivalent strain

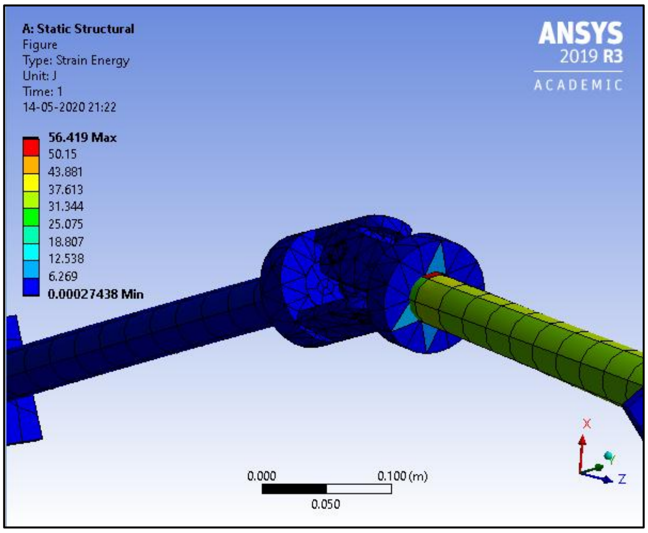



Fig.32 Strain Energy

Maximum total deformation = 0.0439 m

Maximum stress = 13663 MPa

Maximum strain = 0.0723 m/m

Strain Energy = 56.42 J

b) Static structural – The bottom face of the rack is fixed and a force of 30 N is given on the spur rack. The displacement is fixed in x and z direction. Also a rotational velocity of 5 radians/sec is given on the body and the results are as under:-

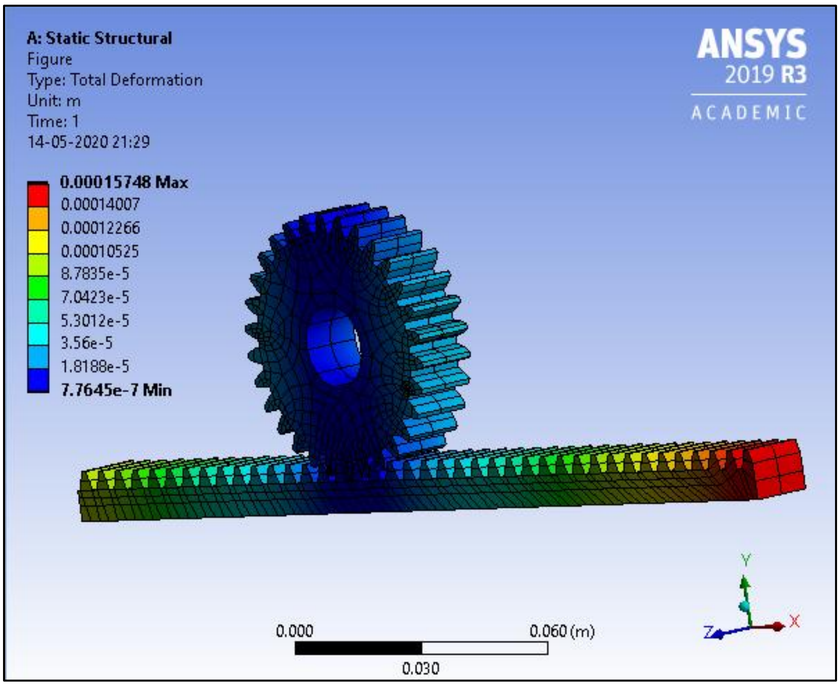
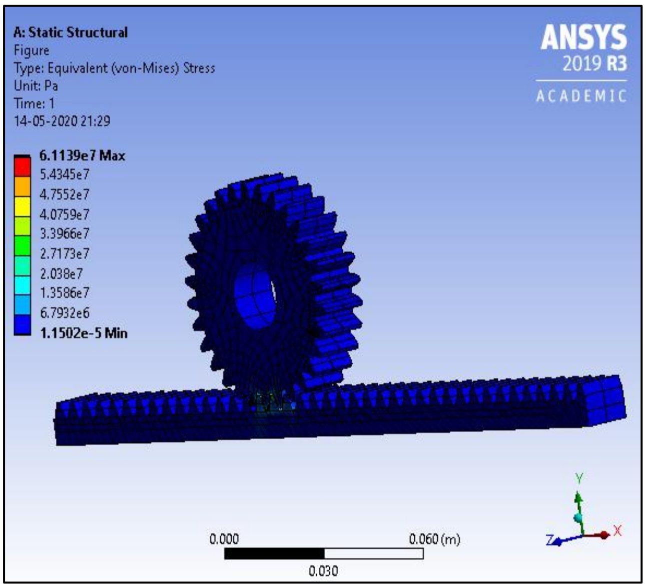



Fig. 33 Total Deformation



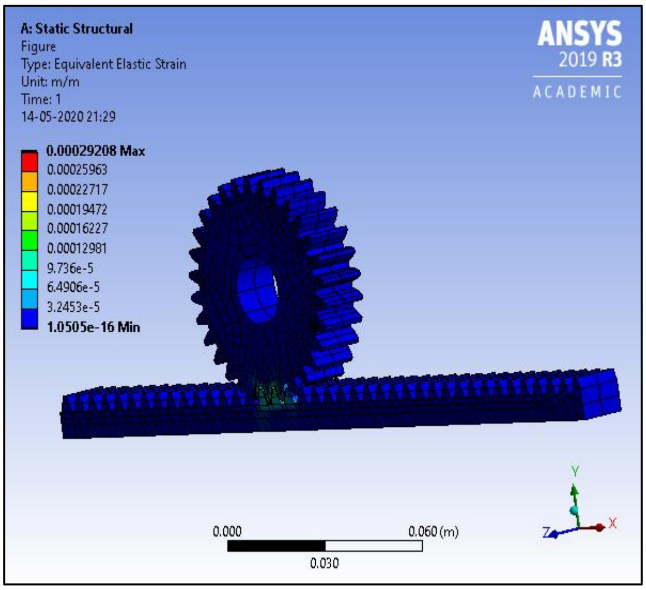




Fig.34 Equivalent stress

Fig.35 Equivalent strain

Maximum deformation = 0.157 mm

Maximum stress = 61.1 MPa

Maximum strain = $2.9 \times 10^{-4}$ m/m

CALCULATIONS:-

Steering wheel radius (R) = 170 mm

Spur pinion pitch radius (r) = 13 mm

Pinion teeth (t) = 6

Circular pitch (p) = 12 mm

Movement ratio = R / r = 13:1

Force on tie rods = Effort on steering wheel × Movement ratio

$= 2 \times 20 \times 13$

$= 520$ N

Weight on each wheel = 40 Kg = 392.4 N

Torque on pinion = 392.4 × 13 = 5101.2 N-mm (considering $\mu=1$)

Force on steering wheel = 5101.2 / 170

= 30 N

The steering mechanism chosen for this vehicle is the Ackermann Linkage Geometry as this allows the vehicle to turn about a centre i.e. instantaneous centre while not allowing the tyres to skid. Also this geometry is chosen for small and light vehicles which is why the reason. In Ackermann type the inner wheel turns more than the outer wheel.

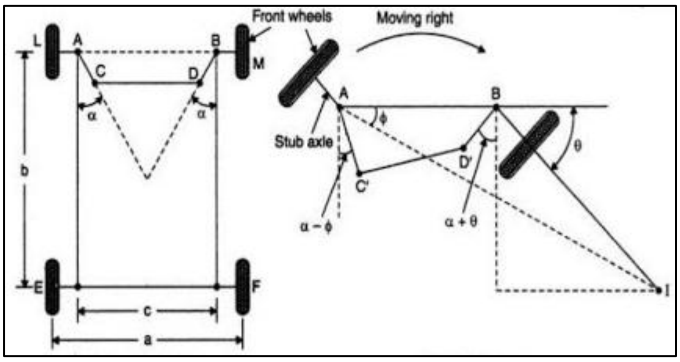



Fig. 36 Ackermann Geometry

Let us assume that the inner wheel turns through an angle = 40 º = θ

From the law of correct steering and the from the above figure we have,

$\cot \phi - \cot \theta = c / b$ ------------ Equation 1, where

$\phi$ = angle turned by outer wheel

$\theta$ = angle turned by inner wheel

c = distance between pivot centres = 800 mm

b = wheel base = 1220 mm

a = wheel track = 1010 mm

So from 1, $\phi = 29.17^{\circ}$

Now, Turning circle radius for outer front wheel = $R_{OF} = (b/\sin \phi) + (a-c)/2 = 2.86$ m

Turning circle radius for inner front wheel = $R_{IF} = (b/\sin \theta) - (a-c)/2 = 1.97$ m

Turning circle radius for outer rear wheel = $R_{OR} = b\cot\phi + (a-c)/2 = 2.57$ m

Turning circle radius for inner rear wheel = $R_{IR} = b\cot\theta - (a-c)/2 = 1.57$ m

Gear ratio value for steering should be within 8:1 to 24:1, although the actual value will depend on the weight and type of the vehicle. Steering ratio value will be within 12:1 to 20:1.

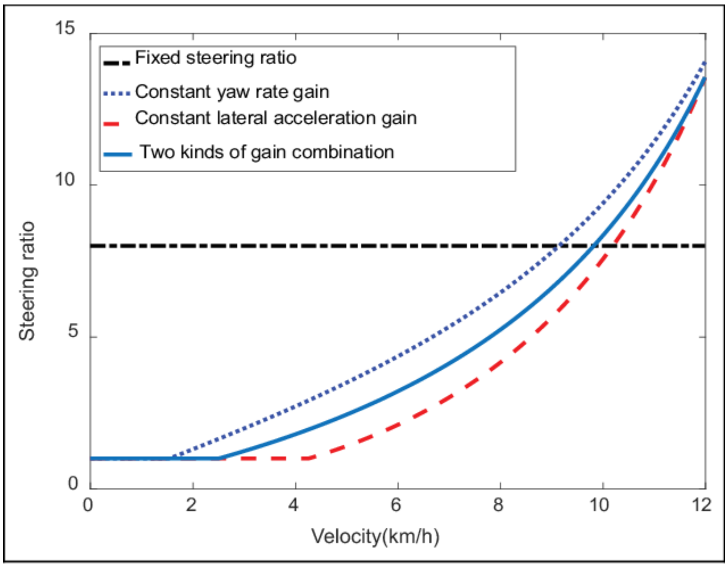



Fig.37 Steering ratio versus velocity curve

Ackermann Percentage tells us by how much do the inner tyre turns with respect to the outer tyre. Also this tells us the behaviour of vehicle during over steering, under steering or neutral steering. The value of this is kept as close as possible to the ideal value. So,

Ackermann Value = $\tan^{-1} \{(b) / (b/\tan \phi) - a\}$

$$\theta_{ACKERMANN} = 44.23^{\circ}$$

Ackermann Percentage = $\theta / \theta_{ACKERMANN} = 0.9043 = 90.43$ %. Hence the assumptions and the design specifications are within satisfactory limit.

# VI.V WHEEL ASSEMBLY

The wheel assembly support the whole weight of the vehicle, resist the strains and transmits the driving torque. They must be rigid enough to retain their shape under all working conditions. Tyres should be rigidly mounted on the rim and they must absorb shocks from road undulations. The material chosen for wheel rim is 6061 ALUMINIUM ALLOY and for the tyres IMPREGNATED RUBBER. We propose the use of tubeless tyres in this go-kart which are beneficial in air retention, safety, rider comfort, cooling of tyre and balancing.

TABLE 7.MATERIAL PROPERTIES OF 6061 ALLOY

| SERIAL NUMBER | PROPERTIES | VALUES |
|---|---|---|
| **1** | **YOUNG'S MODULUS** | **70 GPa** |
| **2** | **POISSON RATIO** | **0.33** |
| **3** | **TENSILE STRENGTH** | **260 MPa** |
| **4** | **PROOF STRESS** | **240 MPa** |
| **5** | **THERMAL CONDUCTIVITY** | **166 W/m-K** |
| **6** | **THERMAL EXPANSION** | **23.4 × $10^{-6}$ °$C^{-1}$** |
| **7** | **DENSITY** | **2700 Kg/$m^3$** |

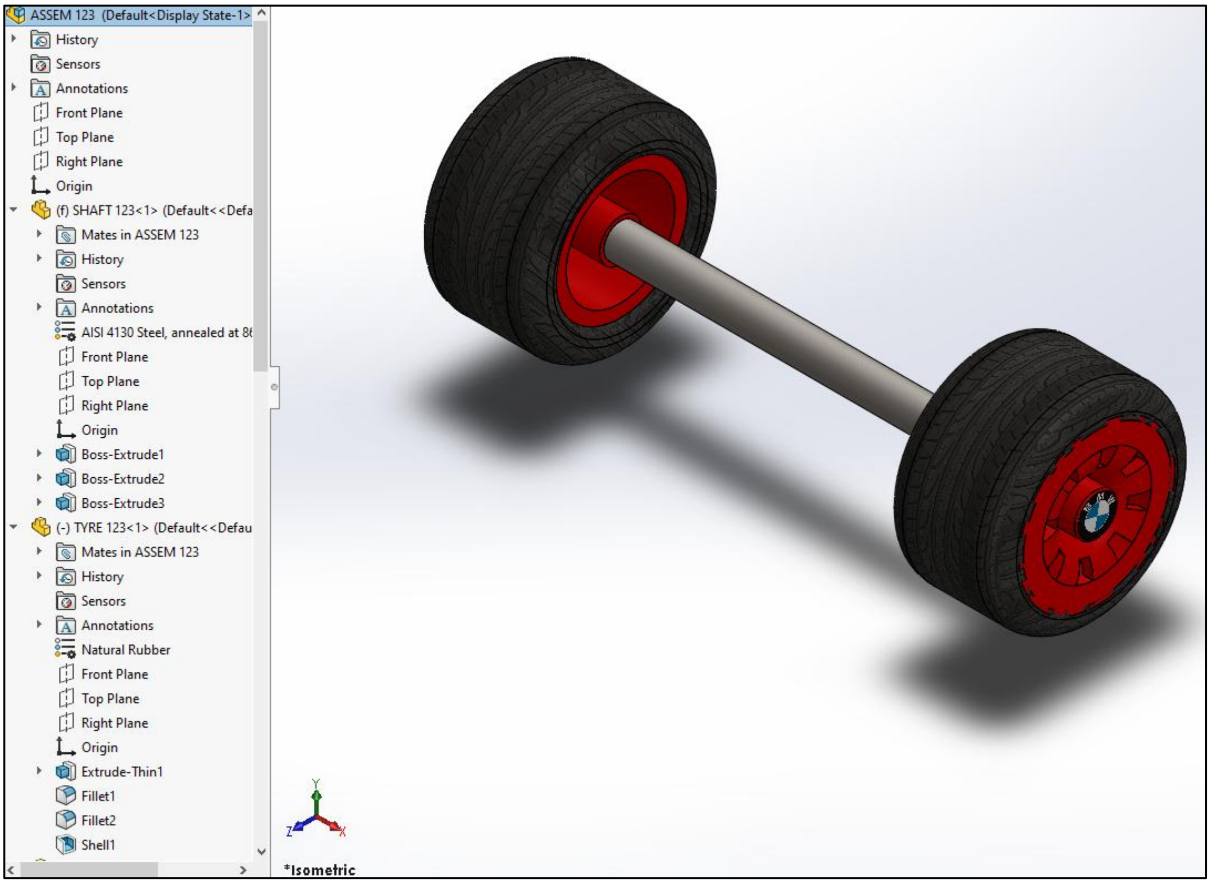



Fig.38 Wheel Assembly

## FINITE ELEMENT ANALYSIS:-

The following part shows the static structural analysis and modal analysis of the wheel assembly. The material used for wheel is ALUMINIUM ALLOY and AISI 4130 for the shaft. A concrete structure (treated as road surface) has been shown to study the effect (balance force) on the wheels by the road reaction.

a) Static Structural – Each of the wheel were loaded with a force of 2000 N the concrete support was fixed and a torque of 60 N-m was given on the shaft. The displacement was fixed on x and y directions and the z direction were kept free. The results are as under:

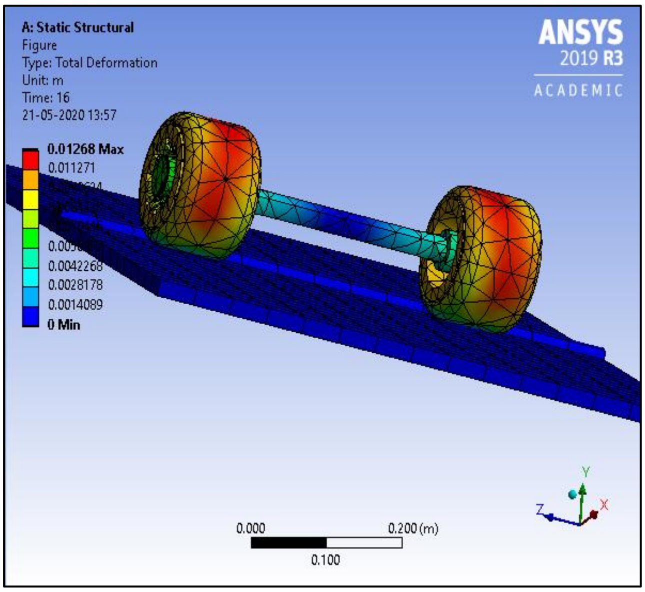



Fig.39 Total Deformation

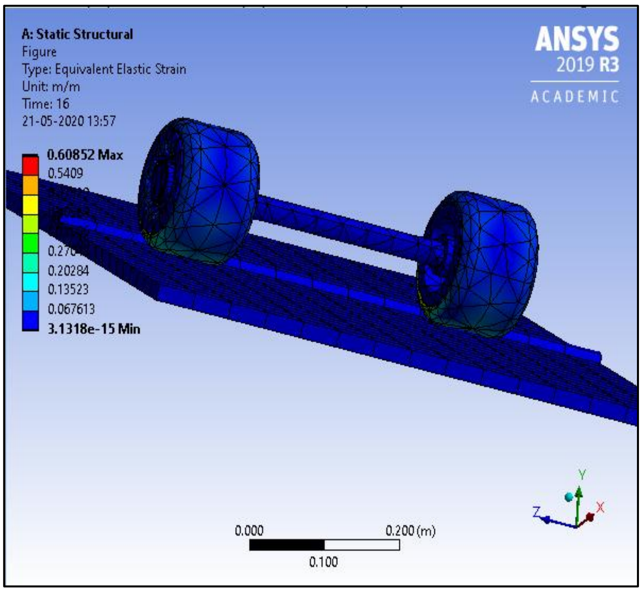



Fig.40 Equivalent strain

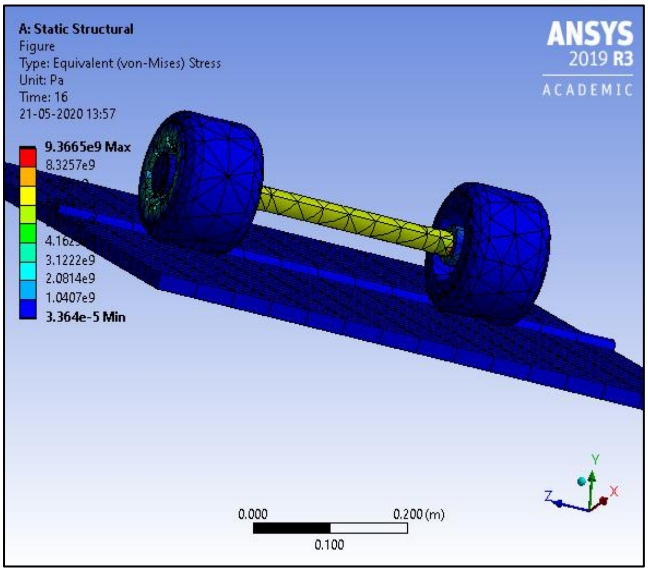



Fig.41 Equivalent stress

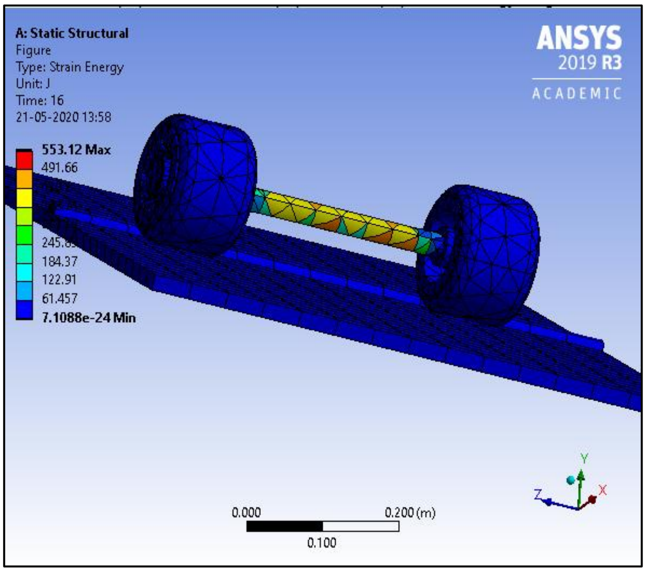



Fig.42 Strain Energy

Maximum deformation = 12.68 mm

Maximum equivalent stress = 9366 MPa

Maximum equivalent strain = 0.6

Maximum Strain energy = 553.12 J

| Graphics Properties | | | | | | |
|---|---|---|---|---|---|---|
| Visible | Yes | | | | | |
| Transparency | 1 | | | | | |
| **Definition** | | | | | | |
| Suppressed | No | | | | | |
| Stiffness Behavior | Flexible | | | | | |
| Coordinate System | Default Coordinate System | | | | | |
| Reference Temperature | By Environment | | | | | |
| Treatment | None | | | | | |
| **Material** | | | | | | |
| Assignment | Concrete | 6061 ALLOY WHEEL RIM | | RUBBER TYRE | | AISI 4130 SHAFT |
| Nonlinear Effects | Yes | | | | | |
| Thermal Strain Effects | Yes | | | | | |
| **Bounding Box** | | | | | | |
| Length X | 0.70505 m | 0.127 m | | 0.167 m | | 3.2e-002 m |
| Length Y | 0.41331 m | 0.127 m | | 0.167 m | | 3.2e-002 m |
| Length Z | 0.8 m | 0.116 m | | 0.11228 m | | 0.5 m |
| **Properties** | | | | | | |
| Volume | 1.2995e-002 m³ | 6.7014e-004 m³ | | 5.7502e-004 m³ | | 3.3941e-004 m³ |
| Mass | 29.888 kg | 1.8094 kg | | 0.57502 kg | | 2.6644 kg |
| Centroid X | -3.955e-002 m | 4.5774e-002 m | 4.5779e-002 m | 4.5796e-002 m | 4.5789e-002 m | 4.5772e-002 m |
| Centroid Y | 0.25316 m | 0.31189 m | 0.3119 m | 0.31189 m | 0.31192 m | 0.3119 m |
| Centroid Z | 0.44005 m | 0.67637 m | 0.25504 m | 0.67186 m | 0.25954 m | 0.4657 m |
| Moment of Inertia Ip1 | 1.595 kg·m² | 4.2074e-003 kg·m² | 4.2064e-003 kg·m² | 2.3099e-003 kg·m² | 2.31e-003 kg·m² | 4.5612e-002 kg·m² |
| Moment of Inertia Ip2 | 3.1641 kg·m² | 4.2063e-003 kg·m² | 4.2075e-003 kg·m² | 2.3083e-003 kg·m² | 2.3082e-003 kg·m² | 4.5612e-002 kg·m² |
| Moment of Inertia Ip3 | 1.5712 kg·m² | 3.6869e-003 kg·m² | 3.687e-003 kg·m² | 3.3819e-003 kg·m² | | 2.9987e-004 kg·m² |
| **Statistics** | | | | | | |
| Nodes | 3882 | 6677 | 6693 | 2648 | 2652 | 772 |
| Elements | 570 | 3388 | 3398 | 1267 | 1266 | 339 |
| Mesh Metric | None | | | | | |

Fig.43 ANSYS REPORT

CALCULATIONS:-

Wheel velocity (v) = 16.66 m/sec

Rolling radius (r) = 0.162 m

Mass of unbalanced rubber = 0.2 kg

Radius of wheel rim ($r_b$) = 0.15 m

Angular velocity = v / r

= 51.45 radians / sec

Out of balance force = $m \times r \times \omega^2$

= 85.76 N

Now, $m_b \times r_b \times \omega^2$ = out of balance force = 85.76

So, balance weights required ($m_b$) = 0.216 kg = 216 g

Thus 216 gm is equally distributed and a balance mass of 108 g is placed on each rim side opposite to the unbalanced mass.

b) MODAL ANALYSIS – The values of modal analysis were kept the same as in static structural. This analysis calculates the frequency of the system and determines the dynamic characteristics of the system. The total deformation values at different frequencies are as under:

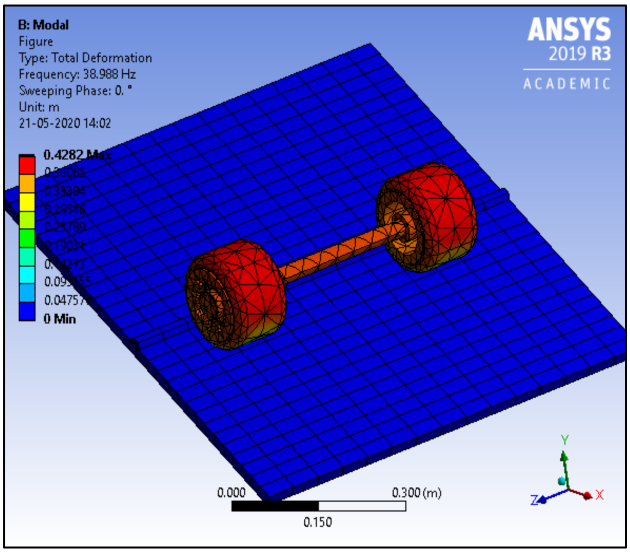



Fig.44

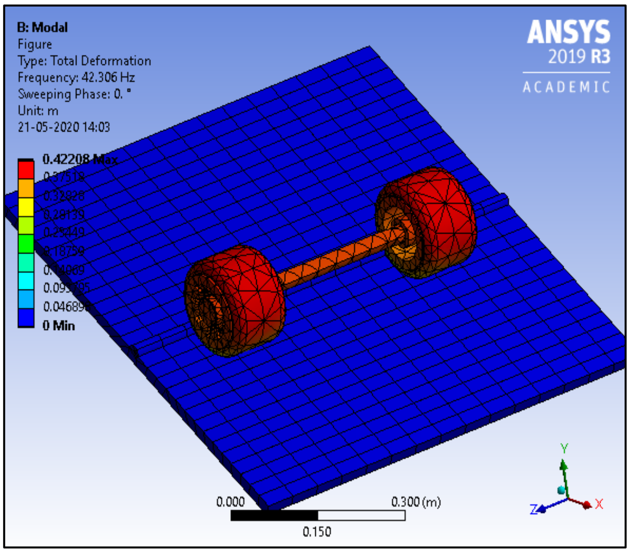



Fig.45

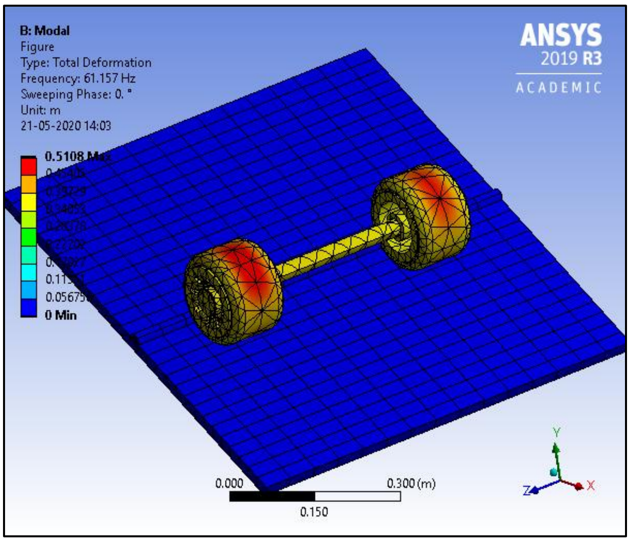



Fig.46

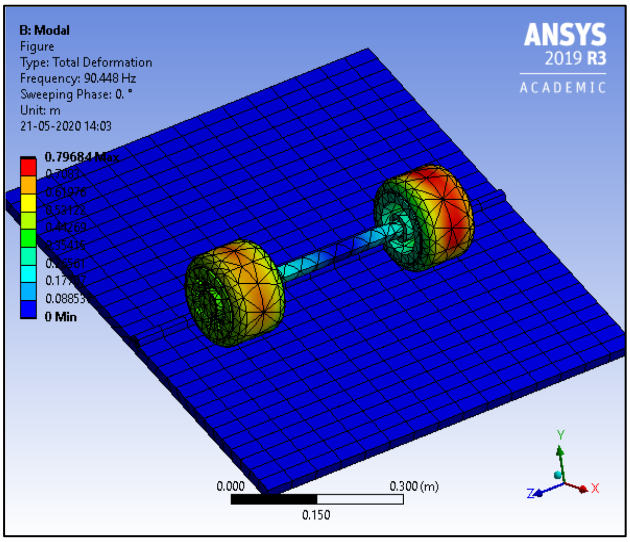



Fig.47

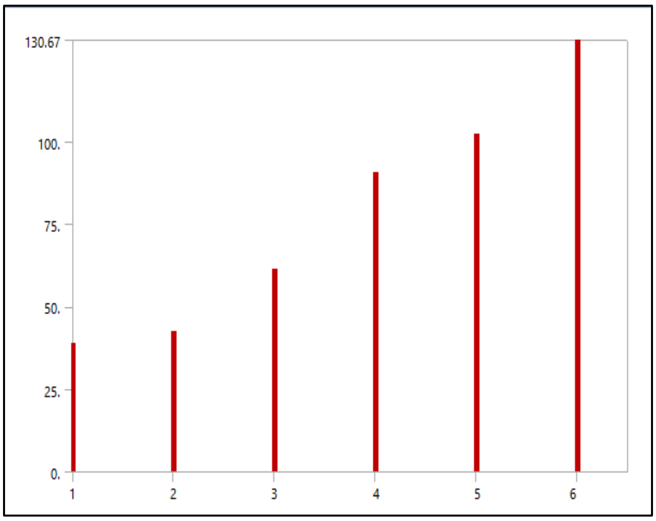



Fig.48 Frequency versus Number of modes

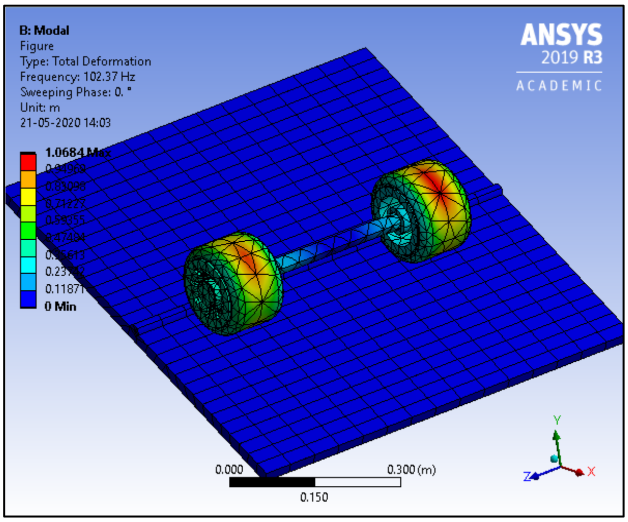



Fig.49

## TABLE 8.CALCULATED VALUES

| NUMBER OF MODES | FREQUENCY (Hz.) | MAX. DEFORMATION(m) |
|---|---|---|
| 1 | 38.988 | 0.4282 |
| 2 | 42.306 | 0.4220 |
| 3 | 61.157 | 0.5108 |
| 4 | 90.448 | 0.7968 |
| 5 | 102.37 | 1.0684 |
| 6 | 130.67 | 1.1343 |

Now, Critical Speed of Shaft ($N_C$) = $(60 / 2\pi) \times (\pi^2 / l^2) \times (EI / \rho A)^{1/2}$ , where

I = moment of inertia = $\pi/64\ (D^4 - d^4) = 8.59 \times 10^{-08}\ m^4$

E = Young's modulus = $2.05 \times 10^{11}\ N/m^2$

$\rho$ = density of material = 7850 $Kg/m^3$

A = Area of cross-section = $\pi/4\ (D^2 - d^2) = 5.49 \times 10^{-4}\ m^2$

L = length of shaft = 1 m

So, Critical speed ($N_C$) = 6024.53 rpm (Design speed)

From design consideration critical speed should be 60% more than the engine speed at maximum power condition i.e.

Engine speed at max power = 3600 rpm

So, Minimum critical speed required = 3600 × 1.6 = 5760 rpm (Calculated speed)

As, Design Speed > Calculated Speed, hence the design is SAFE.

# VI.VI BEARING

BEARING is a machine element that is used to allow relative motion between two members which are in contact (such as shaft and bearing housing) with very less friction. There are two major types of bearing viz. SLIDING CONTACT and ROLLER CONATCT. In this go-kart ROLLER CONTACT BEARING has been used. Deep-groove ball bearing the most frequently used bearing is employed. Some advantages of this bearing are:-

a) High load carrying capacity

b) Due to point contact temperature rise is less in this bearing and

c) It can bear load both in axial and radial direction.

The bearing consists of four parts i.e. the outer race and inner race, rolling element (balls), roller and the housing. The material used for the races are AISI 440 C (Martensitic), for the rolling elements ZIRCONIUM DIOXIDE ($ZrO_2$) and for the roller and housing CAST IRON.

TABLE 9.MECHANICAL PROPERTIES

| SRL. NO. | PROPERTY | AISI 440 C | $ZrO_2$ (Z 201N) | GREY CAST IRON (ASTM GRADE 40) |
|---|---|---|---|---|
| **1** | **YOUNG'S MODULUS** | **200 MPa** | **200 GPa** | **180 GPa** |
| **2** | **DENSITY** | **7800 $Kg/m^3$** | **6000 $Kg/m^3$** | **7500 $Kg/m^3$** |
| **3** | **POISSON RATIO** | **0.28** | **0.31** | **0.29** |
| **4** | **SHEAR MODULUS** | **84 GPa** | **86.4 MPa** | **69 GPa** |
| **5** | **YIELD TENSILE STRENGTH** | **1900 MPa** | **330 MPa** | **200 MPa** |
| **6** | **ULTIMATE TENSILE STRENGTH** | **1970 MPa** | **410 MPa** | **310 MPa** |
| **7** | **COEFFICIENT OF THERMAL EXPANSION** | **$10.2 \times 10^{-06}$ / K** | **$11 \times 10^{-06}$ / K** | **$11 \times 10^{-06}$ / K** |
| **8** | **MELTING TEMPERATURE** | **1756 K** | **2973 K** | **1453 K** |
| **9** | **SPECIFIC HEAT CAPACITY** | **460 J/Kg-K** | **510 J/Kg-K** | **490 J/Kg-K** |
| **10** | **THERMAL CONDUCTIVITY** | **24.2 W/m-K** | **4 W/m-K** | **46 W/m-K** |

## BEARING DESIGN:-

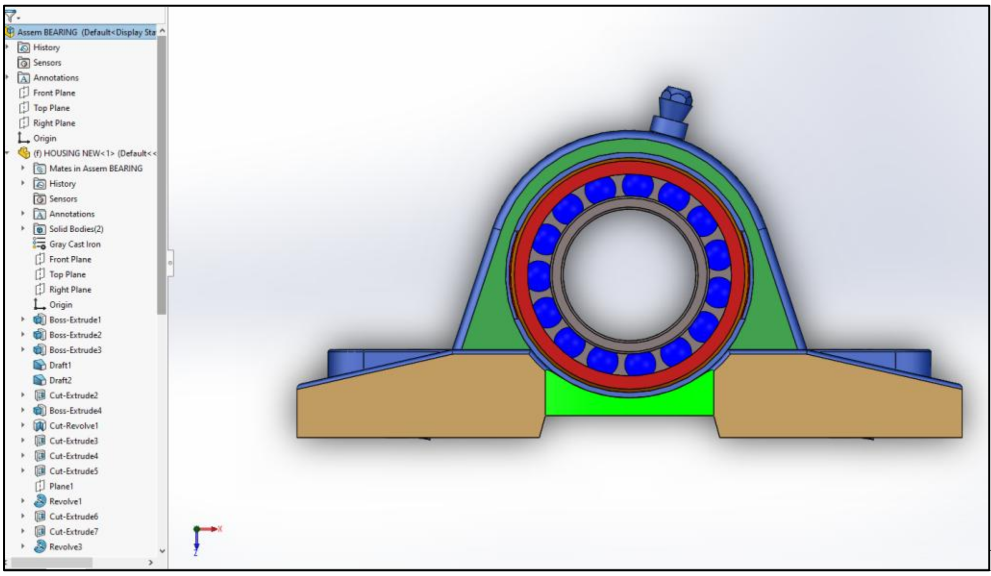


Fig.50 FRONT VIEW

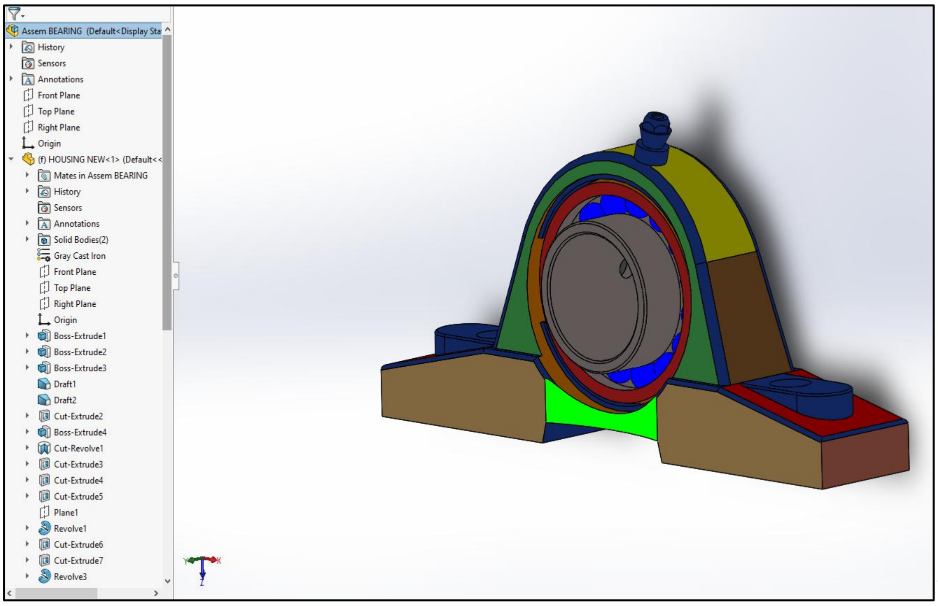


Fig.51 ISOMETRIC VIEW

## FINITE ELEMENT ANALYSIS:-

The following part shows the static structural, transient structural and steady-state temperature analysis of the race. The materials used have already been discussed.

a) STATIC STRUCTURAL – The outer cross-section of the bearing was fixed and a pressure of 20635 Pa (ramped) was given on the hollow surface along with a radial force of 2220 N on the fixed geometry. Both the values were calculated based on the dimensions of the hollow geometry. The following results were obtained:

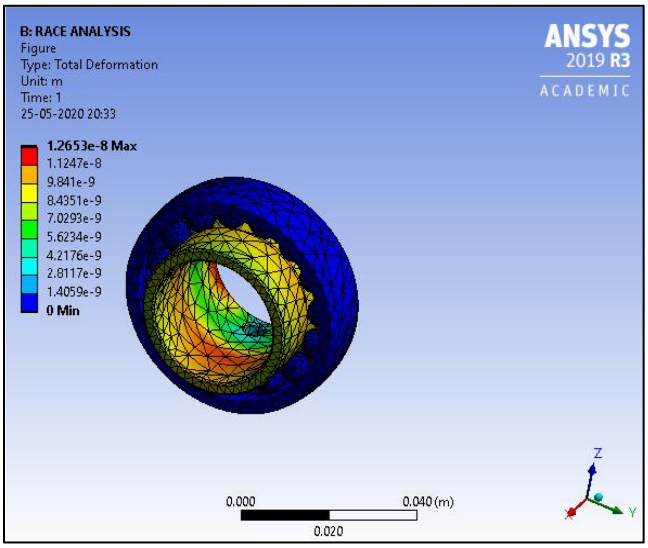



Fig.52 Total Deformation

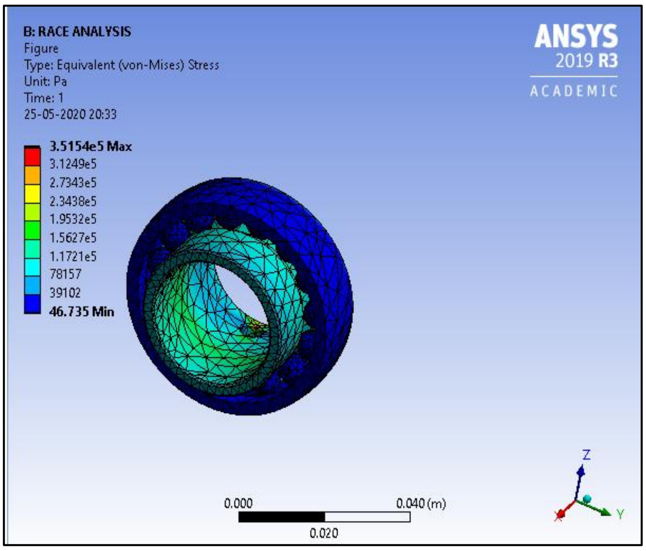



Fig.53 Equivalent Stress

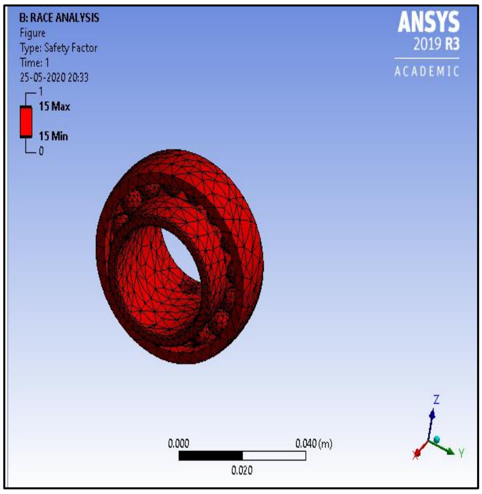



Fig.54 SAFETY FACTOR

| | | | | | |
|---|---|---|---|---|---|
| State | Meshed | | | | |
| **Graphics Properties** | | | | | |
| Visible | Yes | | | | |
| **Definition** | | | | | |
| Suppressed | No | | | | |
| Stiffness Behavior | Flexible | | | | |
| Coordinate System | Default Coordinate System | | | | |
| Reference Temperature | By Environment | | | | |
| Treatment | None | | | | |
| **Material** | | | | | |
| Assignment | ZIRCONIUM DIOXIDE | | AISI 440C | ZIRCONIUM DIOXIDE | |
| Nonlinear Effects | Yes | | | | |
| Thermal Strain Effects | Yes | | | | |
| **Bounding Box** | | | | | |
| Length X | 7.e-003 m | | 3.2e-002 m | 7.e-003 m | |
| Length Y | 7.e-003 m | | 5.3656e-002 m | 7.e-003 m | |
| Length Z | 7.e-003 m | | 5.3656e-002 m | 7.e-003 m | |
| **Properties** | | | | | |
| Volume | 1.7959e-007 m³ | | 1.6492e-005 m³ | 1.7959e-007 m³ | |
| Mass | 1.0057e-003 kg | | 0.12699 kg | 1.0057e-003 kg | |
| Centroid X | -1.5276e-009 m | -1.5486e-009 m | -5.9922e-004 m | -1.5316e-009 m | -1.5098e-009 m |
| Centroid Y | 1.6585e-002 m | -1.8728e-002 m | 7.3323e-006 m | -1.8728e-002 m | -2.05e-002 m |
| Centroid Z | -1.205e-002 m | 8.3381e-003 m | 4.9445e-007 m | -8.3381e-003 m | 2.7456e-012 m |
| Moment of Inertia Ip1 | 4.8579e-009 kg·m² | | 5.3577e-005 kg·m² | 4.8579e-009 kg·m² | |
| Moment of Inertia Ip2 | 4.8577e-009 kg·m² | | 3.402e-005 kg·m² | 4.8577e-009 kg·m² | |
| Moment of Inertia Ip3 | 4.8484e-009 kg·m² | | 3.4025e-005 kg·m² | 4.8484e-009 kg·m² | |
| **Statistics** | | | | | |
| Nodes | 403 | 377 | 8313 | 391 | 406 |
| Elements | 194 | 179 | 4158 | 192 | 201 |
| Mesh Metric | None | | | | |

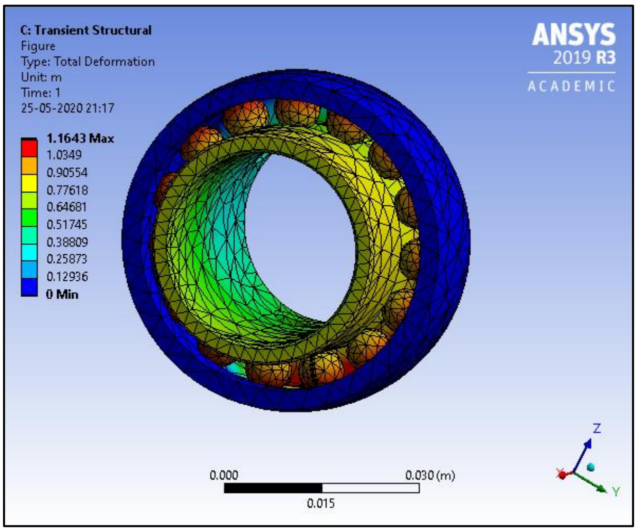
Fig.55 ANSYS REPORT

Maximum Deformation = $1.26 \times 10^{-05}$ mm

Maximum stress = 351.54 KPa

Safety factor = 15

b) TRANSIENT STRUCTURAL – Keeping the same values as that of static structural the initial time was set as 0.1 sec, the maximum time as 0.1 sec, the step end time as 0.01 sec and Joint load (rotational) ground to race and ground to multiple was set and the following results were obtained:



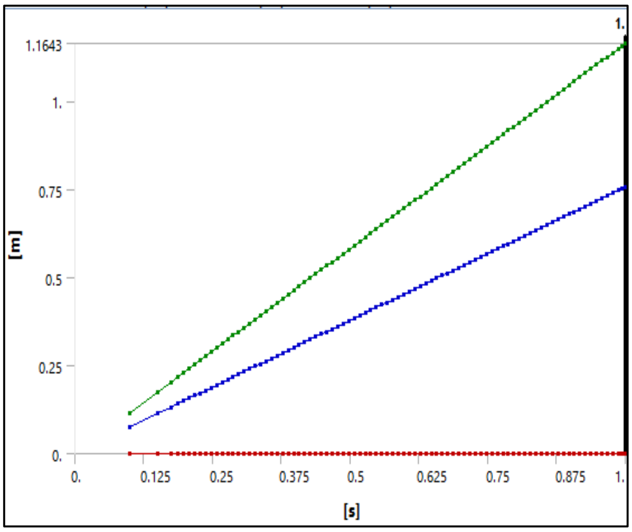
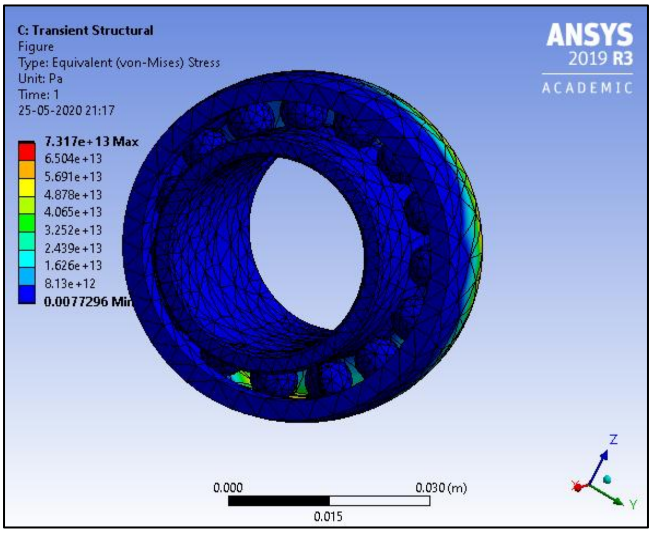




Fig.56 Total Deformation

Fig.57 Deformation versus Time Curve



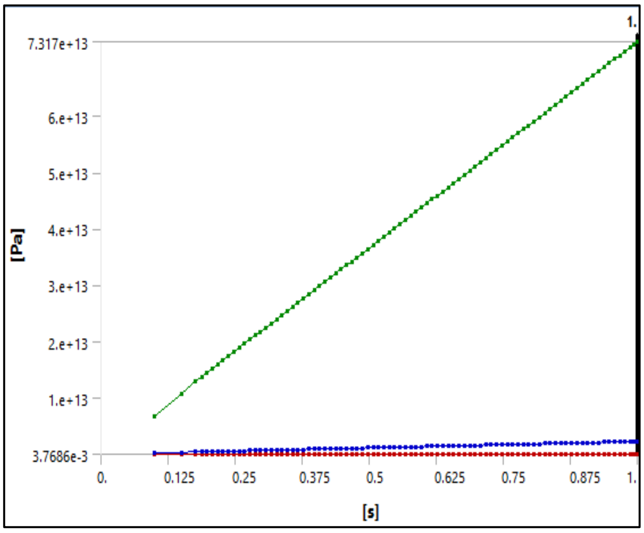
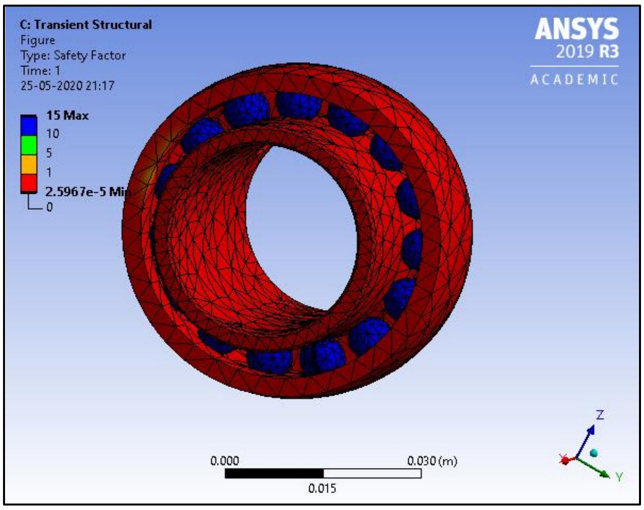




Fig.58 Equivalent Stress

Fig.59 Stress versus Time Curve



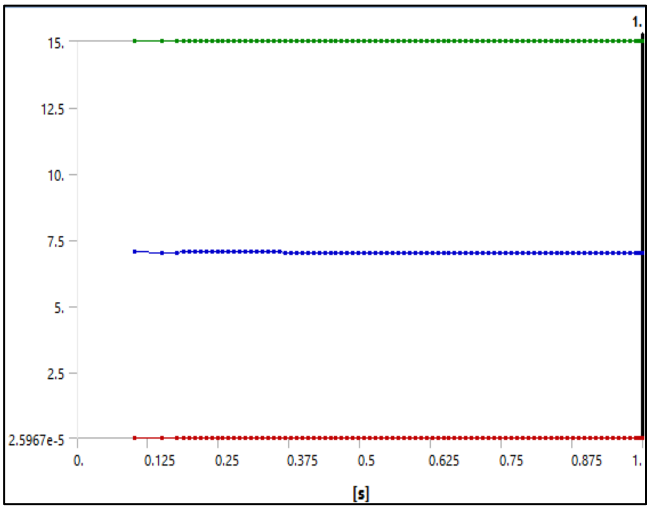




Fig.60 Safety Factor

Fig.61 Safety factor versus Time Curve

Maximum deformation = 1.1643 m

Maximum Equivalent stress = 73170 GPa

Maximum Safety Factor = 15

c) STEADY-STATE THERMAL – An estimated temperature value of 110°C was given on the inside hollow surface. As the bearing will be naturally cooled, convection coefficient of 220 $W/m^2K$ and a heat flux of 11.15 $KW/m^2$ were given on the outer race and the results obtained are:

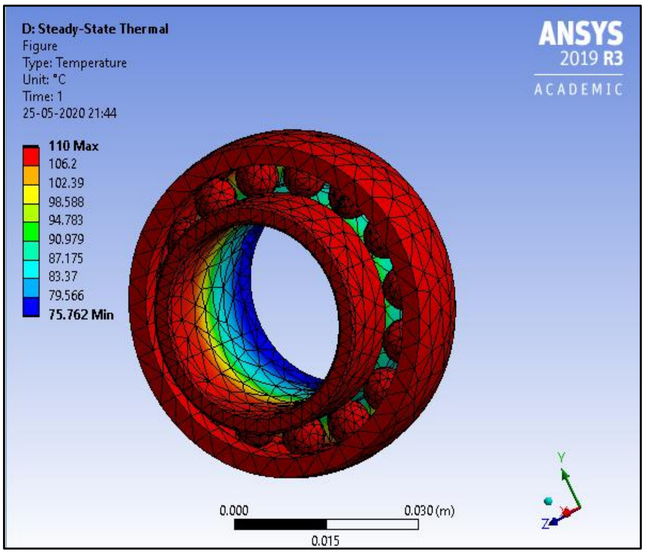



Fig.62 Temperature

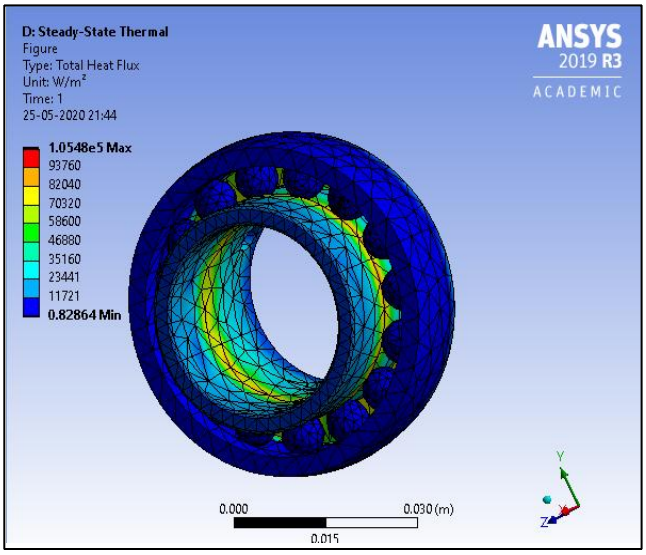



Fig.63 Total Heat Flux

Maximum temperature = 110°C

Maximum heat flux = 105.48 $KW/m^2$

CALCULATIONS:-

STRIBECK's EQUATION gives the static load capacity of the bearing i.e.

$$C_O = (kd^2z) / 5 \quad \text{-------- equation 1, where}$$

$C_O$ = Static load

$P_1$ = load on ball = 3260 N (from analysis)

k = constant factor = $P_1 / d^2$ = 8.15 $N/mm^2$

d = diameter of balls = 20 mm

z = number of balls = 15

So, Static load capacity = 9780 N

Now, Equivalent dynamic load (P) = $XF_R + YF_A$ ----------equation 2, where

$F_R$ = Radial load = 3260 N

$F_A$ = Axial load = 978 N

X and Y are radial and thrust factors respectively and the values chosen are:-

$F_A / C_O = 0.1$ and $F_A / F_R = 0.3$ so from the table X = 1, Y = 0.

| $\left(\frac{F_a}{C_0}\right)$ | $\left(\frac{F_a}{F_r}\right) \le e$ | | $\left(\frac{F_a}{F_r}\right) > e$ | | $e$ |
|---|---|---|---|---|---|
| | $X$ | $Y$ | $X$ | $Y$ | |
| 0.025 | 1 | 0 | 0.56 | 2.0 | 0.22 |
| 0.040 | 1 | 0 | 0.56 | 1.8 | 0.24 |
| 0.070 | 1 | 0 | 0.56 | 1.6 | 0.27 |
| 0.130 | 1 | 0 | 0.56 | 1.4 | 0.31 |
| 0.250 | 1 | 0 | 0.56 | 1.2 | 0.37 |
| 0.500 | 1 | 0 | 0.56 | 1.0 | 0.44 |

Fig.64 X and Y factors

So, P = 3260 N.

| | | | | | |
|---|---|---|---|---|---|
| 20 | 32 | 7 | 2700 | 1500 | 61804 |
| | 42 | 8 | 7020 | 3400 | 16404 |
| | 42 | 12 | 9360 | 4500 | 6004 |
| | 47 | 14 | 12700 | 6200 | 6204 |
| | 52 | 15 | 15900 | 7800 | 6304 |
| | 72 | 19 | 30700 | 16600 | 6404 |

Fig.65 Static and Dynamic load capacities of Ball Bearings

From the table corresponding to static load capacity the dynamic load capacity (C) value is taken as 16800 N for bearing designation 6304.

From Load-Life relationship,

$$L_{10} = (C / P)^{n} \quad \text{-----------equation 3, where}$$

$L_{10}$ = rated bearing life (in million revolutions)

C = Dynamic load capacity = 16800 N

P = Equivalent dynamic load = 3260 N

n= 3 (for ball bearing)

So, $L_{10}$ = 137 million revolutions.

Also Rated bearing life (in hours) = $L_{10h} = (L_{10} \times 10^6) / 60 \times N$, where

N = rotational speed = 382 rpm (40 radians/sec)

Therefore, $L_{10h}$ = 5977 hours (can be approximated to 6000 hours)

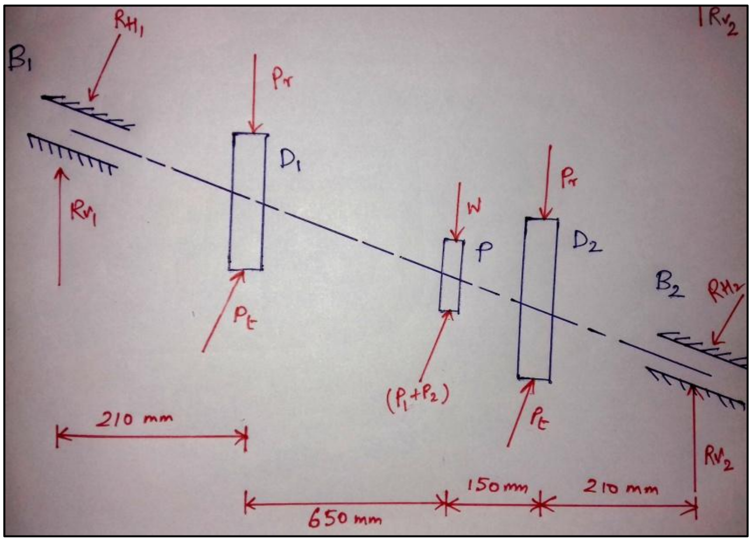



Fig.66 Load Distribution

$P_R$ = 49.05 N (for each brake)

$P_T$ = 595.4 N (for each brake)

W = 19.62 N (weight of smaller sprocket)

$P_1+P_2$ = 140 N (tangential force on the chain assembly)

Considering the vertical forces and taking moment about $B_1$ i.e. $\Sigma M_{B1} = 0$;

$P_R \times 210 + W \times 860 + P_R \times 1010 = R_{V2} \times 1220$

So, $R_{V2}$ = 62.88 N

Considering all the vertical forces,

$R_{V1} + R_{V2} = P_R + W + P_R$

Therefore, $R_{V1} = 54.84$ N

Considering the horizontal forces and taking moment about $B_1$ i.e. $\Sigma M_{B1} = 0$;

$P_T \times 210 + (P_1 + P_2) \times 860 + P_T \times 1010 = R_{H2} \times 1220$

So, $R_{H2} = 694.08$ N

Considering all the horizontal forces,

$R_{H1} + R_{H2} = P_T + P_T + (P_1 + P_2)$

Therefore, $R_{H1} = 636.72$ N

Now, the BEARING REACTIONS are given by

$R_1 = [(R_{V1})^2 + (R_{H1})^2]^{1/2}$

$R_1 = 639.07$ N

$R_2 = [(R_{V2})^2 + (R_{H2})^2]^{1/2}$

$R_2 = 696.92$ N

Therefore the radial reactions are 639.07 N and 696.92 N respectively at bearing 1 and 2. As there is no axial force on the bearings so it's value is zero.

Reliability of bearing is defined as the number of successful revolutions completed by the bearing before fatigue failure. Mathematically,

$(L / L_{10})^b = [\ln (1/R)] / [\ln (1/R_{90})]$ ------------equation 4, where

R = reliability for a given life

L = given life = 5000 hours (assumed)

$R_{90} = 0.90$

$L_{10}$ = life corresponding to 90 % reliability = 6000 hrs

b= constant = 1.17

So, R = 0.9184 = 91.84 % is the reliability of the designed roller bearing.

Lubrication in rolling contact bearings is to reduce the friction between the balls and races. Also dissipation of heat generated, protection from dirt and corrosion are other reasons for lubrication. Oil is preferred to grease for lubrication material as it is more effective in carrying away the extra heat generated due to friction and also the flow ability characteristic of oil into various contact areas make it a suitable lubricating material.

# VI.VII SUSPENSION SPRING

Springs are machine elements which deflect under the action of the load and returns to its original shape as soon as the load is removed. Go-karts are usually not employed with suspension due to very little ground clearance available. But recreational purpose go-karts can use suspension for safety and to reduce vibrations. Suspensions are mainly used for:-

a) Absorb shocks and vibrations

b) Store energy

c) Measurement of forces etc.

We propose the use of helical spring in this go-kart as it is the most popular and commonly used spring nowadays because:-

a) Easy to design and manufacture

b) High reliability and cheaper

c) Deflection is proportional to the force applied

The material used for the suspension is AISI 6150 VANADIUM CHROME STEEL (tempered). Chromium present in small percentage (0.8 – 1.1) % prevents corrosion, vanadium and manganese induces hardness in the metal.

TABLE 10.MECHANICAL PROPERTIES OF AISI 6150

| SERIAL NUMBER | PROPERTIES | VALUES |
|---|---|---|
| 1 | YOUNG'S MODULUS | 190 GPa |
| 2 | POISSON RATIO | 0.29 |
| 3 | SHEAR MODULUS | 73 GPa |
| 4 | DENSITY | 7800 $Kg/m^3$ |
| 5 | YIELD TENSILE STRENGTH | 1160 MPa |
| 6 | ULTIMATE TENSILE STRENGTH | 1200 MPa |
| 7 | FATIGUE STRENGTH | 750 MPa |
| 8 | SPECIFIC HEAT CAPACITY | 470 J/Kg-K |
| 9 | THERMAL CONDUCTIVITY | 46 W/m-K |
| 10 | COEFFICIENT OF THERMAL EXPANSION | $12 \times 10^{-06}$ $K^{-1}$ |
| 11 | MELTING TEMPERATURE | 1683 K |

The helical spring is made from wire of a circular cross-section which is bent in the shape of a helix. In helical compression springs the external force tends to compress the spring. Although the spring is under compression the wire is not subjected to compressive stress.

SUSPENSION DESIGN:-

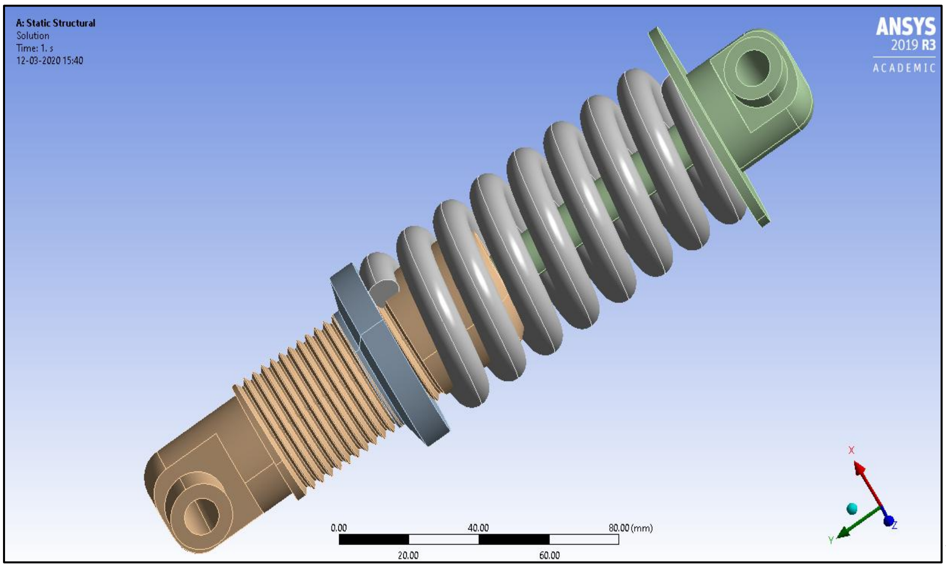



Fig.67 ISOMETRIC VIEW

FINITE ELEMENT ANALYSIS:-

The following part shows the static structural and transient structural analysis of the suspension member.

a) STATIC STRUCTURAL – The lower part and the top part is fixed. The top rod is subjected to a force of 3630 N (downwards) and the bottom part is subjected to a force of 1210 N (upwards) and the results obtained are as:

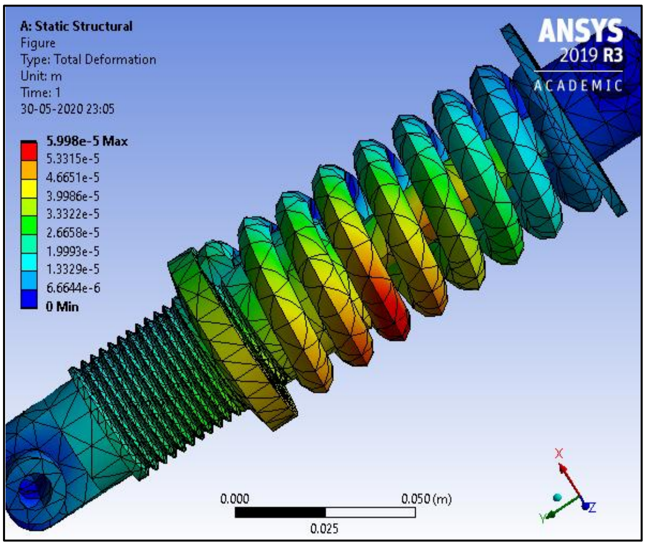



Fig.68 Total Deformation

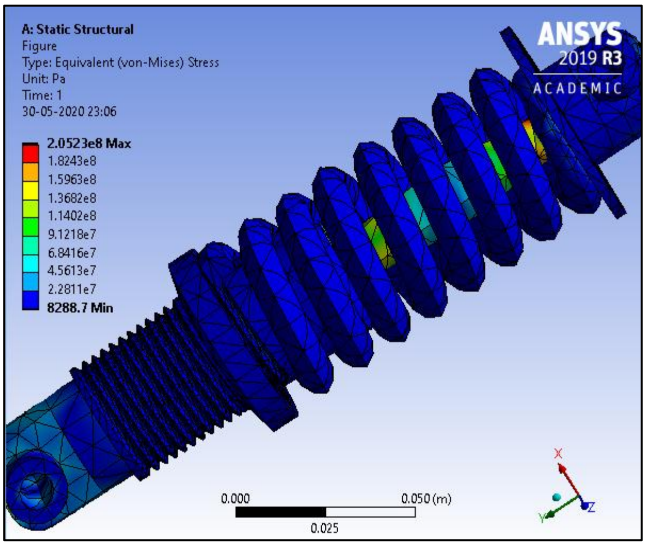



Fig.69 Equivalent Stress

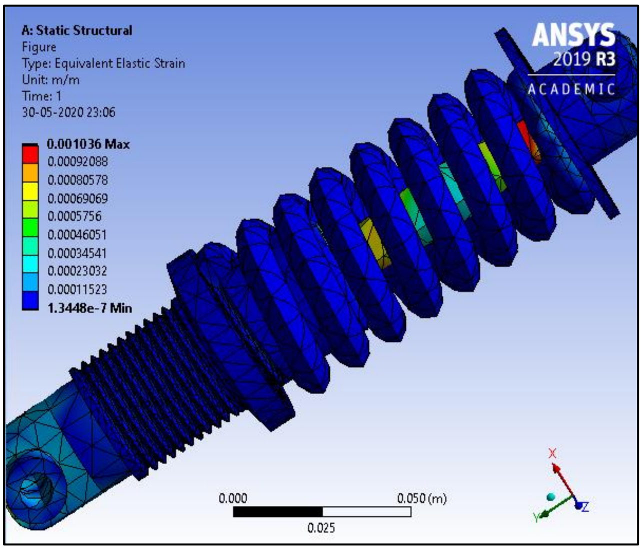



Fig.70 Equivalent Strain

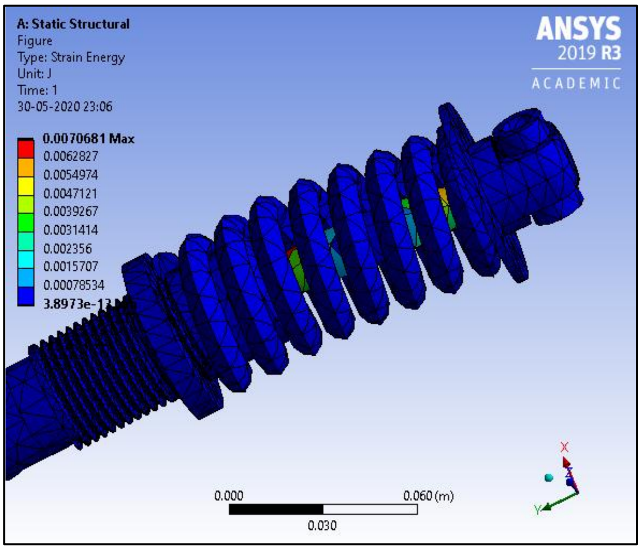



Fig.71 Strain Energy

Maximum deformation = 0.06 mm

Maximum equivalent stress = 205.2 MPa

Maximum equivalent strain = 0.001

Maximum Strain energy = 0.007 Joule

| Object Name | *Solid* | *Solid* | *Solid* | *Solid* |
|---|---|---|---|---|
| State | Meshed | | | |
| **Graphics Properties** | | | | |
| Visible | Yes | | | |
| Transparency | 1 | | | |
| **Definition** | | | | |
| Suppressed | No | | | |
| Stiffness Behavior | Flexible | | | |
| Coordinate System | Default Coordinate System | | | |
| Reference Temperature | By Environment | | | |
| Treatment | None | | | |
| **Material** | | | | |
| Assignment | AISI 6150 VANADIUM CHROME STEEL | | | |
| Nonlinear Effects | Yes | | | |
| Thermal Strain Effects | Yes | | | |
| **Bounding Box** | | | | |
| Length X | 5.e-002 m | 5.2e-002 m | 6.e-002 m | 3.5e-002 m |
| Length Y | 0.10315 m | 1.3125e-002 m | 0.148 m | 0.115 m |
| Length Z | 5.e-002 m | 5.2e-002 m | 6.e-002 m | 3.5e-002 m |
| **Properties** | | | | |
| Volume | 7.7983e-005 m³ | 1.4134e-005 m³ | 3.7594e-005 m³ | 8.2595e-005 m³ |
| Mass | 0.60827 kg | 0.11025 kg | 0.29323 kg | 0.64424 kg |
| Centroid X | 1.7861e-002 m | 1.7899e-002 m | 1.7961e-002 m | 1.796e-002 m |
| Centroid Y | -1.3358e-002 m | 4.4118e-002 m | -5.0252e-002 m | 5.8674e-002 m |
| Centroid Z | 0.1004 m | 0.10062 m | 0.10057 m | |
| Moment of Inertia Ip1 | 6.2187e-004 kg·m² | 2.3926e-005 kg·m² | 4.7562e-004 kg·m² | 6.8741e-004 kg·m² |
| Moment of Inertia Ip2 | 2.5159e-004 kg·m² | 4.5998e-005 kg·m² | 4.5309e-005 kg·m² | 9.4722e-005 kg·m² |
| Moment of Inertia Ip3 | 6.2291e-004 kg·m² | 2.3926e-005 kg·m² | 4.7635e-004 kg·m² | 6.8817e-004 kg·m² |
| **Statistics** | | | | |
| Nodes | 2540 | 4637 | 1365 | 13928 |
| Elements | 941 | 2575 | 644 | 7171 |
| Mesh Metric | None | | | |

Fig.72 Ansys Report

b) TRANSIENT STRUCTURAL – The top and bottom portions are fixed. All the connections are removed. Joint load of ground-solid type is provided and the analysis is as under:

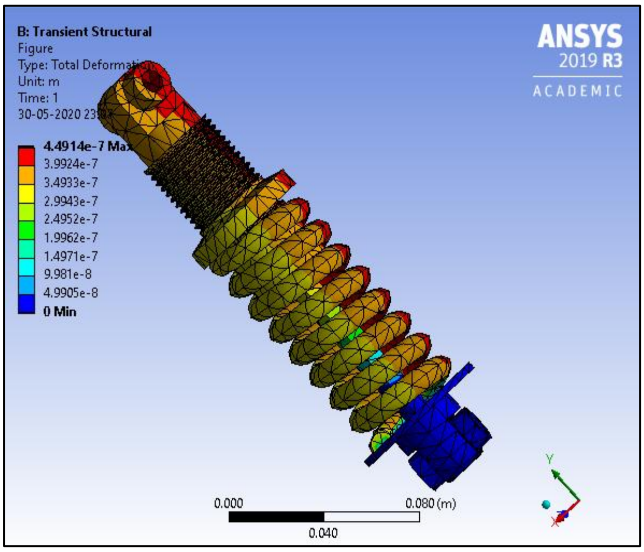



Fig.73 Total Deformation

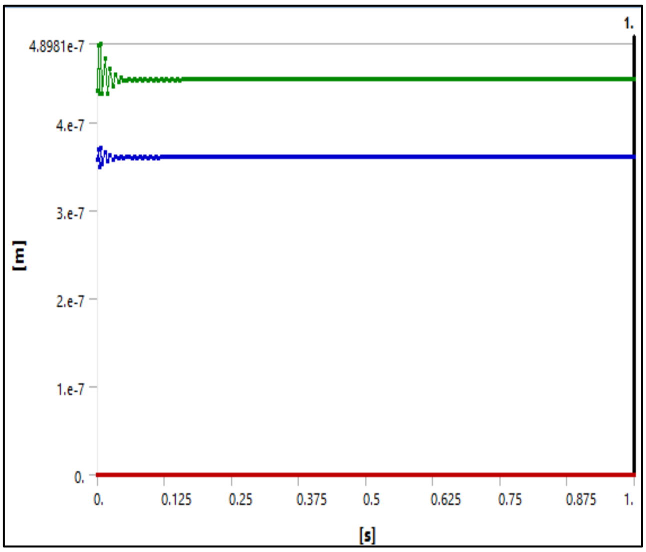



Fig.74 Deformation versus Time curve

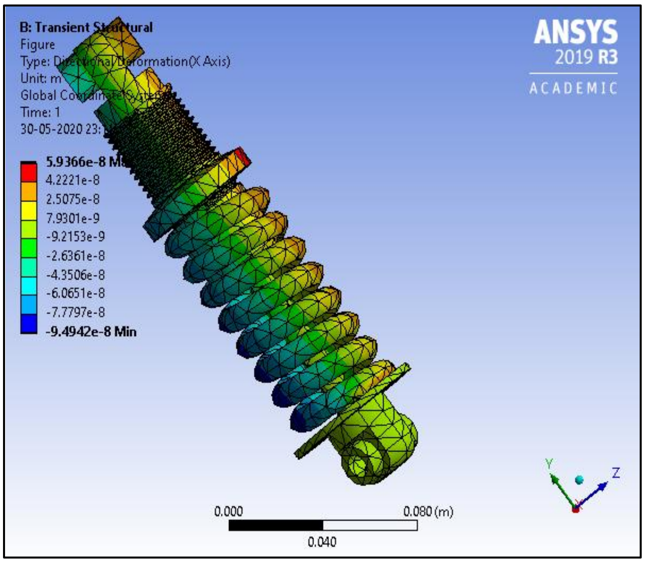



Fig.75 Directional Deformation

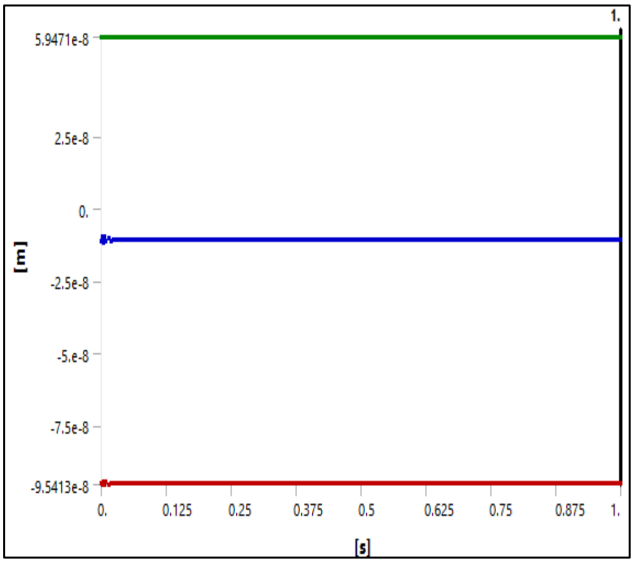



Fig.76 D. Deformation versus Time curve

Maximum deformation = $4.49 \times 10^{-4}$ mm

Maximum directional deformation = $5.93 \times 10^{-5}$ mm

CALCULATION:-

$\tau = (8PCK) / \pi d^2$ -------------equation 1, where

$\tau$ = Permissible shear stress = 0.5 × Yield tensile strength = 580 MPa

C = spring index = 6

d = wire diameter

K = Wahl factor = (4C-1 / 4C-4) + (0.615/C) = 1.2525

P = Maximum force = 3630 N

So, the wire diameter (d) = 10.94 mm = 12 mm (for design consideration)

So, Actual shear stress = 482.40 MPa (from equation 1)

As actual shear stress is less than permissible shear stress so the design is satisfactory with wire diameter of 12 mm.

Now, C = D / d

D = Mean coil diameter = 6 × 12 = 72 mm

Again, Max deflection = $\delta = (8PC^3N / Gd)$ ------------equation 2, where

N = number of active coils

G = shear modulus = 73 GPa

$\delta$= Maximum deflection = 40 mm

So, N = 5.58 = 6 coils

Total number of coils ($N_T$) = N + 2 = 8 (number of active and inactive coils = 2)

Actual deflection = 57.28 mm = $\delta_{ACTUAL}$

Solid length = $N_T$ × d = 8 × 12 = 96 mm

Axial gap = 0.15 × actual deflection = 8.59 mm

Free length of spring = Solid length + Axial gap + Actual deflection

= 161.87 mm = 162 mm

Pitch of coil = Free length / ($N_T$ – 1)

= 23.14 mm

Required spring rate (k) = P / maximum deflection

= 90.75 N/mm

Actual spring rate = $Gd^4 / 8D^3N$ --------------equation 4

= 84.49 N/mm

DETERMINATION OF SAFETY FACTOR OF SUSPENSION MEMBER:-

For oil-hardened and tempered steel wire –

Endurance limit ($S_E$) = 0.22 × $S_{UT}$ ($S_{UT}$ = Ultimate tensile strength)

= 0.22 × 1200

= 264 MPa

Torsional yield limit ($S_{SY}$) = 0.45 × $S_{UT}$

= 540 MPa

Now, $\sigma_{MAX}$ = 205.32 MPa , $\sigma_{MIN}$ = 22.81 MPa

Mean stress ($\sigma_M$) = ($\sigma_{MAX}$ + $\sigma_{MIN}$) / 2 = 114.06 MPa

Amplitude stress ($\sigma_A$) = ($\sigma_{MAX}$ - $\sigma_{MIN}$) / 2 = 91.25 MPa

$S_A$ = Combined equivalent stress = N × $\sigma_A$ = N × 91.25

$S_M$ = Mean stress = N × $\sigma_M$ = N × $\sigma_M$ = N × 114.06

By, GERBER THEORY

$(S_A / S_E) + (S_M / S_{UT})^2 = 1$

N = 2.7

By, SODERBERG THEORY

$(S_A / S_E) + (S_M / S_{YT}) = 1$

N = 2.25

By, GOODMAN THEORY

$(S_A / S_E) + (S_M / S_{UT}) = 1$

N = 2.27

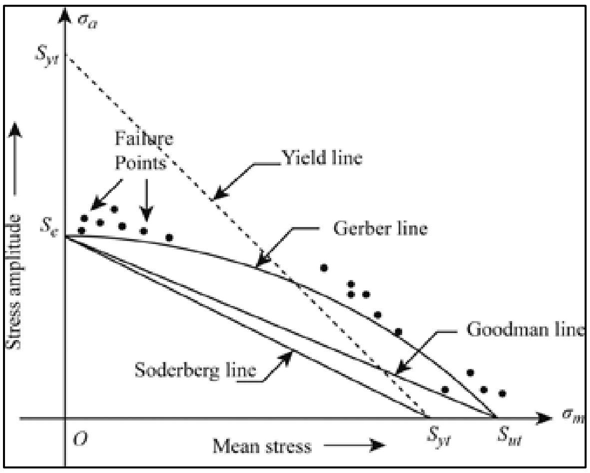



Fig.77 GERBER PARABOLA

So GERBER > GOODMAN > SODERBERG (from calculation)

As evident from the figure GERBER parabola gives more accurate safety factor whereas SODERBERG and GOODMAN are conservative theories. Hence the design is satisfactory.

# VI.VIII WISHBONE SUSPENSION

The wishbone suspension consists of two links usually parallel in normal position to provide stiffness to front and rear part and to resist the braking torque. The vehicle weight is transferred to the coil spring. If there is any side thrust present then it will be resisted by the stiffness of the wishbone members and pivot joints.

The material used for wishbone member is AISI 304 (annealed). In this go-kart we have not used wishbone suspension instead we have used rigid-axle beam suspension which is having following advantages:-

a) Beam-axle is capable of rough handling

b) Good road grip and prolonged tyre wear etc.

TABLE 11.MECHANICAL PROPERTIES OF AISI 304

| SERIAL NUMBER | PROPERTIES | VALUES |
|---|---|---|
| **1** | **YOUNG’S MODULUS** | **200 GPa** |
| **2** | **POISSON RATIO** | **0.28** |
| **3** | **DENSITY** | **7800 Kg/m$^3$** |
| **4** | **SHEAR MODULUS** | **77 GPa** |
| **5** | **YIELD TENSILE STRENGTH** | **230 MPa** |
| **6** | **ULTIMATE TENSILE STRENGTH** | **580 MPa** |
| **7** | **FATIGUE STRENGTH** | **210 MPa** |
| **8** | **MODULUS OF RESILIENCE** | **140 KJ/m$^3$** |
| **9** | **SHEAR STRENGTH** | **400 MPa** |

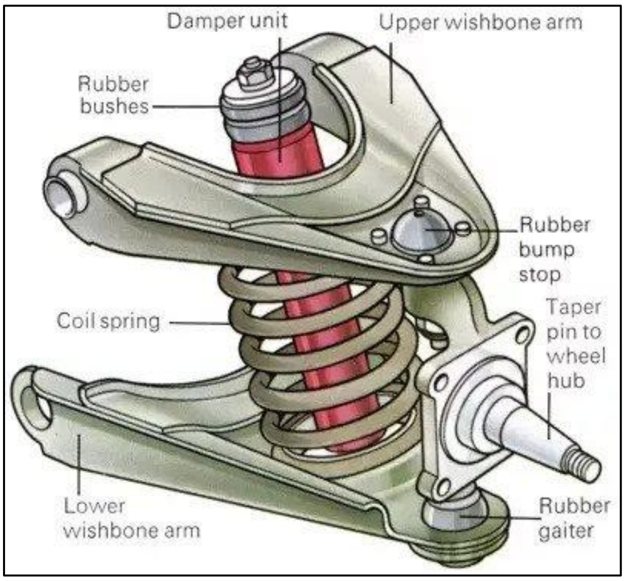



Fig.78 Double Wishbone Member (transverse)

WISHBONE DESIGN:-

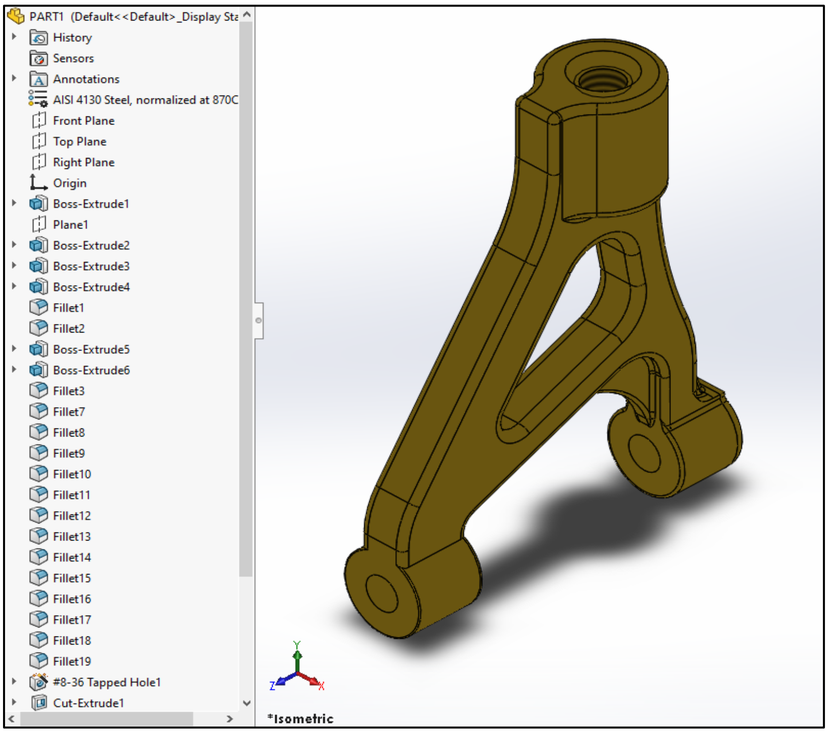



Fig.79 ISOMETRIC VIEW

FINITE ELEMENT ANALYSIS:-

The following part shows the static structural analysis of the member.

a) STATIC STRUCTURAL - The end are fixed and a load of 3360 N is subjected on the top surface and the results are as under:

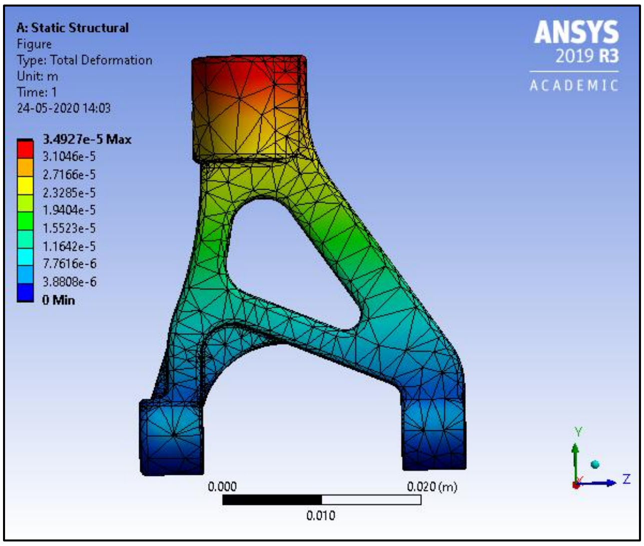



Fig.80 Total Deformation

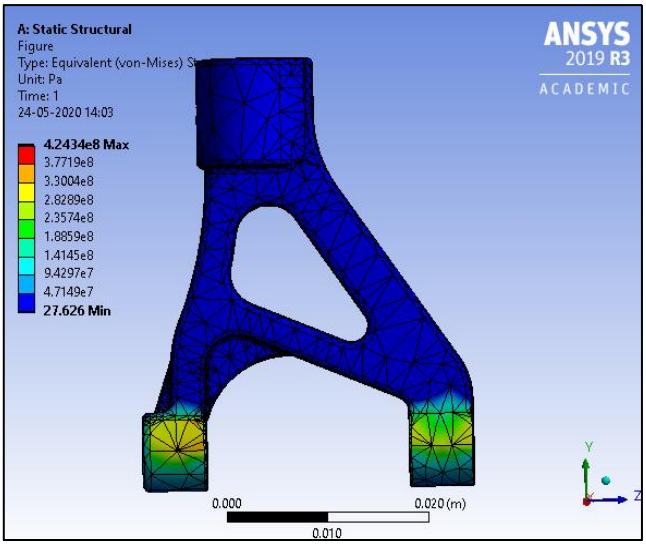



Fig.81 Equivalent Stress

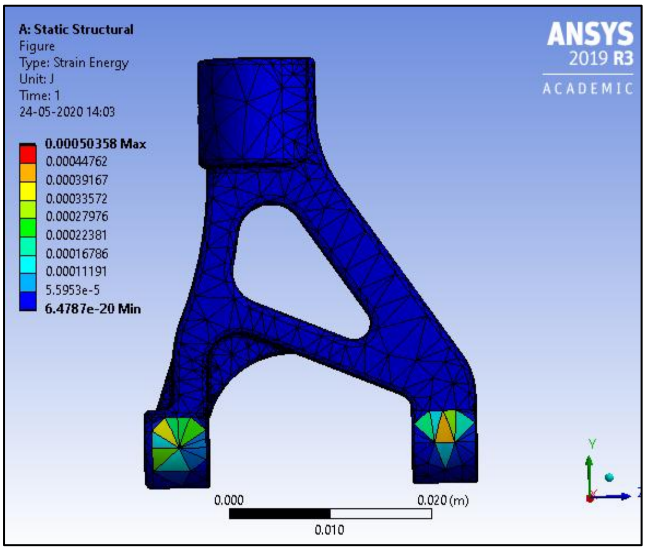



Fig.82 Strain Energy

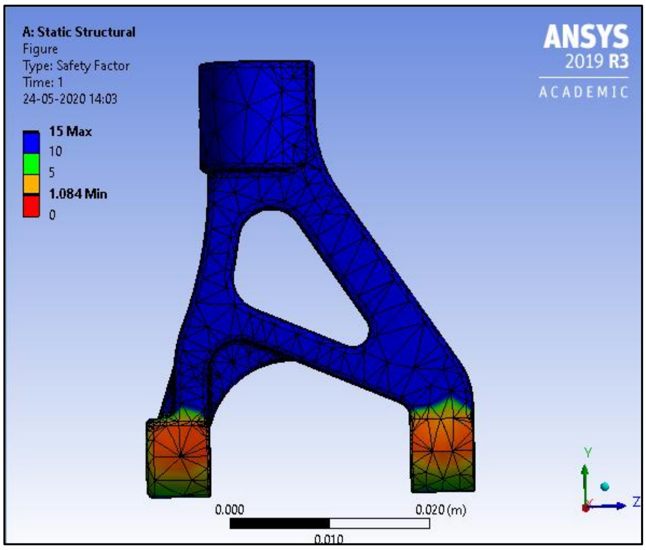



Fig.83 Safety Factor

Maximum deformation = 0.0349 mm

Maximum equivalent stress = 424.3 MPa

Maximum strain energy = 0.0005 J

Maximum safety factor = 15

# VII. RESULTS

With this our combined-power go-kart design seems to have been accomplished. Various engineering principles and design methods have been used to create the vehicle with optimum performance and safety. The use of modelling software SOLIDWORKS and simulation software ANSYS assisted a lot in the successful completion of our project. Now here is a display of all the components and their analysis results:-

- CHASSIS FRAME – The material used for the chassis frame is ALUMINIUM ALLOY 6063-T6. The tests which were performed are Static Structural (Front impact, Rear impact and Twisting impact) and Explicit Dynamics and the calculated values are:-

TABLE 12.STATIC STRUCTURAL RESULTS

| REAR IMPACT | FRONT IMPACT | TWISTING IMPACT |
|---|---|---|
| **Force = 3140 N** | **Force = 3140 N** | **Torsion = 1088 Nm** |
| **Maximum deformation = 0.004 m** | **Maximum deformation = 0.089 mm** | **Maximum deformation = 0.0074 mm** |
| **Maximum Equivalent stress = 130 MPa** | **Maximum Equivalent stress = 28.68 MPa** | **Maximum Equivalent strain = $1.23 \times 10^{-6}$** |
| **Safety factor = 2.5** | **Safety factor = 2.5** | **Safety factor = 2.5** |

For the dynamic analysis the values obtained are:-

i) Maximum deformation = 0.05 m

ii) Maximum equivalent stress = 5963 MPa

iii) Maximum directional deformation = $2.52 \times 10^{-5}$ m

iv) Safety factor = 2.6

Maximum bending stress obtained = 77.7 MPa

Air resistance = 0.318 N

- ENGINE - The engine used in this go-kart is HONDA GX270 9 HP commercial petrol engine. The calculations are tabulated as under –

TABLE 13.ENGINE POWER RESULTS

| PARAMETERS | VALUES |
|---|---|
| **Power** | **6.3 Kilowatts** |
| **Engine RPM (at maximum torque)** | **2500** |
| **Gear Ratio** | **2.5** |
| **Chain drive pitch** | **12.7 mm** |
| **Larger sprocket diameter** | **214 mm** |
| **Smaller sprocket diameter** | **85 mm** |
| **Number of links** | **104** |
| **Chain Length** | **1320.8 mm** |
| **Actual centre distance** | **420.46 mm** |
| **Velocity Ratio** | **2.5** |
| **Overall Gear Ratio** | **3.75** |
| **Maximum force on rear axle** | **1500.8 N** |
| **Minimum force on rear axle** | **300.14 N** |
| **Moment on shaft** | **244024 N-mm** |
| **Shaft Diameter** | **24mm (= 30mm)** |
| **Safety Factor** | **3** |

TABLE 14.SOLAR PANEL RESULTS

| PARAMETERS | VALUES |
|---|---|
| **Energy consumed by Motor** | **2400 Watt hours (in 4 hours)** |
| **Energy produced by Solar panel** | **600 Watt hours (in 8 hours)** |
| **Energy stored by battery** | **768 Watt hours** |
| **Discharging time** | **1 hour** |
| **Charging time** | **4.6 hours** |
| **Power of panel** | **300 Watts** |

➢ DISC BRAKE – The material used for the disc brake is CAST IRON (CI). Steady-state thermal analysis is done and the results are shown –

TABLE 15.DISC BRAKE RESULTS

| PARAMETERS | VALUES |
|---|---|
| **Maximum Temperature** | **1002.5°C** |
| **Maximum Heat Flux** | **4640 KW/m$^2$** |
| **Braking Energy** | **22.23 KJ** |
| **Deceleration of vehicle** | **(-) 60.73 m/s$^2$** |
| **Static load (front axle)** | **627.84 N** |
| **Static load (rear axle)** | **941.76 N** |
| **Dynamic load (front axle)** | **816.19 N** |
| **Dynamic load (rear axle)** | **1130.11 N** |
| **Frictional force (each front wheel)** | **244.8 N** |
| **Frictional force (each rear wheel)** | **339 N** |
| **Braking Torque (each front wheel)** | **79.31 N-m** |
| **Braking Torque (each rear wheel)** | **110 N-m** |
| **Master cylinder bore** | **20 mm** |
| **Pressure on Master cylinder** | **1.53 MPa** |
| **Brake Fluid Pressure** | **0.382 MPa** |
| **Force on Master cylinder** | **480 N** |
| **Braking torque (assuming friction)** | **174.92 N-m** |
| **Clamping Force (all 4 wheels)** | **9717.76 N** |
| **Stopping Distance** | **2.28 m** |
| **Stopping time** | **0.274 s** |
| **Total force on Front brakes** | **2431.34 N** |
| **Total force on Rear brakes** | **1190.8 N** |

➢ STEERING ASSEMBLY - The following part shows the static structural results of universal joint and steering gears. The material used for universal joint is Stainless steel (ferritic) and AISI 304 for gear components.

TABLE 16.STATIC STRUCTURAL

| COMPONENTS | DEFORMATION | EQUIVALENT STRESS | EQUIVALENT STRAIN |
|---|---|---|---|
| **UNIVERSAL JOINT** | **43.9 mm** | **13663 MPa** | **0.0723** |
| **STEERING GEAR** | **0.157 mm** | **61.1 MPa** | **$2.9 \times 10^{-4}$** |

TABLE 17.STEERING GEOMETRY RESULTS

| PARAMETERS | VALUES |
|---|---|
| **Circular Pitch** | **12 mm** |
| **Movement Ratio** | **13:1** |
| **Force on Tie rods** | **520 N** |
| **Force on steering wheel** | **30 N** |
| **Angle turned by Outer wheel** | **29.17°** |
| **Angle turned by Inner wheel** | **40°** |
| **Turning circle radius (Outer front wheel)** | **2.86 m** |
| **Turning circle radius (Inner front wheel)** | **1.97 m** |
| **Turning circle radius (Outer rear wheel)** | **2.57 m** |
| **Turning circle radius (Inner rear wheel)** | **1.57 m** |
| **Ackermann Value ($\theta_A$)** | **44.23°** |
| **Ackermann Percentage** | **90.43 %** |

- WHEEL ASSEMBLY – The material chosen for wheel rim is 6061 Aluminium Alloy and Impregnated rubber for tyre. The results for Static structural and Modal Analysis are listed below –

a) Total Deformation = 12.68 mm

b) Maximum stress = 9366 MPa

c) Maximum strain = 0.6

d) Maximum strain energy = 553.12 J

TABLE 18.MODAL ANALYSIS RESULTS

| FREQUENCY (Hertz) | MAXIMUM DEFORMATION(m) |
|---|---|
| **38.988** | **0.4282** |
| **42.306** | **0.4220** |
| **61.157** | **0.5108** |
| **90.448** | **0.7968** |
| **102.37** | **1.0684** |
| **130.67** | **1.1343** |

TABLE 19.SHAFT RESULTS

| PARAMETERS | VALUES |
|---|---|
| **Moment of Inertia** | **$8.59 \times 10^{-08}$ $m^4$** |
| **Cross-sectional Area** | **$5.49 \times 10^{-04}$ $m^2$** |
| **Shaft length** | **1 m** |
| **Design Speed** | **6024.53 RPM** |
| **Critical Speed** | **5760 RPM** |

- BEARING HOUSING – The material used for the races are AISI 440 C (Martensitic), for the rolling elements ZIRCONIUM DIOXIDE ($ZrO_2$) and for the roller and housing CAST IRON. The results of Static structural, transient structural and steady-state thermal are listed below –

TABLE 20.RACE RESULTS

| ANALYSIS | TOTAL DEFORMATION | EQUIVALENT STRESS | SAFETY FACTOR |
|---|---|---|---|
| **STATIC STRUCTURAL** | **1.26 × $10^{-05}$ mm** | **351.54 KPa** | **15** |
| **TRANSIENT STRUCTURAL** | **1.1643 m** | **73170 GPa** | **15** |

TABLE 21.BEARING DESIGN RESULTS

| PARAMETERS | VALUES |
|---|---|
| **Maximum Temperature** | **110°C** |
| **Maximum Heat Flux** | **105.48 KW/$m^2$** |
| **Radial Load** | **3260 N** |
| **Axial Load** | **978 N** |
| **Number of balls** | **15** |
| **Ball diameter** | **20 mm** |
| **Static Load Capacity** | **9780 N** |
| **Dynamic Load Capacity** | **16800 N** |
| **Bearing Revolutions** | **137 ( in million revolutions)** |
| **Rated Bearing Life** | **5977 hours** |
| **Radial Resultant Reaction at bearing 1** | **639.07 N** |
| **Radial Resultant Reaction at bearing 2** | **696.92 N** |
| **Reliability** | **0.9184 (91.84 %)** |

- SUSPENSION SPRING – The material used for suspension spring is AISI 6150 VANADIUM CHROME STEEL. The Static structural and Transient structural results are listed as below -

TABLE 22.STATIC STRUCTURAL RESULTS

| TOTAL DEFORMATION | MAXIMUM EQUIVALENT STRESS | MAXIMUM EQUIVALENT STRAIN | MAXIMUM STRAIN ENERGY |
|---|---|---|---|
| **0.06 mm** | **205.2 MPa** | **0.001** | **0.007 J** |

TABLE 23.TRANSIENT STRUCTURAL RESULTS

| MAXIMUM DEFORMATION | MAXIMUM DIRECTIONAL DEFORMATION |
|---|---|
| **$4.49 \times 10^{-4}$ mm** | **$5.93 \times 10^{-5}$ mm** |

TABLE 24.DESIGN RESULTS

| PARAMETERS | VALUES |
|---|---|
| **Permissible Shear stress** | **580 MPa** |
| **Maximum Force** | **3630 N** |
| **Wire Diameter** | **12 mm** |
| **Wahl factor** | **1.2525** |
| **Actual Shear stress** | **482.40 MPa** |
| **Mean coil diameter** | **72 mm** |
| **Total number of coils** | **8** |
| **Actual Deflection** | **57.28 mm** |
| **Free Length of spring** | **162 mm** |
| **Coil pitch** | **23.14 mm** |
| **Spring rate** | **84.49 N/mm** |
| **Safety factor (GERBER THEORY)** | **2.7** |
| **Safety factor (SODERBERG THEORY)** | **2.25** |
| **Safety factor (GOODMAN THEORY)** | **2.27** |

- WISHBONE SUSPENSION – The material used for this component is AISI 304. Static structural analysis and their results are listed below –

Maximum deformation = 0.0349 mm

Maximum equivalent stress = 424.3 MPa

Maximum strain energy = 0.0005 J

Maximum safety factor = 15

# VIII. CONCLUSION

In our proposed Go-Kart, we have presented the use of both Solar Energy and Fossil fuel energy (Petroleum) and the advantages of using such hybrid systems.

- The source of power can be switched according to required conditions, say if it is up for a heavy performance testing, long duration operation, testing of the body's endurance limits at top speeds- the petrol engine can come at handy.
- For conservation of fuel energy, eco-friendly and short duration operations, light performance testing the solar energy powering the Go-Kart should be the selected source.
- Petroleum and other such forms of fossil fuels have been the top sources of energy for automobiles till date. It can produce a great range of power, it is very much abundant, easy to harness, and very reliable. But because of the extreme amounts of it being used daily and the fact that it is non-renewable, there will be a time which will go through the scarcity of fossil fuels.
- Also, fossil fuels, like petrol, produce and give out a lot of heat and in addition to it a lot of pollutants which has been one of the most major causes of global warming and many other environmental and health damages as we all know.
- Solar energy has always been present for us in wide abundance. Its uses are eco-friendly and the source will never run out. Our idea in this project has been to promote the use of an alternative, and more importantly, a renewable and clean form of energy.
- Promoting such uses of renewable sources can and will bring changes which are very much needed in the near future for the sake of our environment and our future generation.

With this, we conclude the major project for our course. Thank you for going through.

# IX. FUTURE SCOPE

For our future prospects, we look forward to increase the performance and efficiencies of our Go-Kart, introducing the use of more Novel materials, giving it some more accessories and hopefully bringing up or putting forward a newer, better and efficient ideologies for Go-Karts. Some areas of improvement could be:-

- The go-kart can be transformed to an Amphibious Vehicle which can run on water as well using propellers, battery and a separate motor arrangement. The motor can be powered using the solar energy.
- IC ENGINES can be completely replaced by electric fuel cells which do not emit unwanted hydrocarbon particles and can be considered as a "CLEAN FUEL".
- Using of bio-fuel in place of petrol or diesel as bio-fuels is considered as "GREEN FUEL".
- Using of Dual-fuel powered engine; Hydrogen fuel can be considered as an excellent substitute for carbon-based fuels.

➢ Use of lean-burn engine which can lead to emission of very less hydrocarbons thus enabling far more efficient operation.

# X. REFERENCES


- GIRI, N.K. Automobile Mechanics, Khanna Publishers, 2019
- JOSEPH, HEITNER. Automobile Mechanics, East West Press, 2019
- TIMOSHENKO & YOUNG, Engineering Mechanics, Tata McGraw Hill Education, 2017
- SHAMES & PITARRESI, Solid Mechanics, Prentice Hall India, 3e
- BHANDARI, V.B. Design of Machine Elements, Tata McGraw Hill Education, 2019
- SHIGLEY & MISCHKE, Mechanical Engineering Design, Tata McGraw Hill Education, 2018
- RATAN, S.S. Theory of Machines, Tata McGraw Hill Education, 2018
- CALLISTER, W.D. Material Science and Engineering, Willey India, 2016
- KHAN, B.H. Non-Conventional Energy Resources, Tata McGraw Hill Education, 3e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