International Journal of Engineering Applied Sciences and Technology, 2018
Vol. 3, Issue 6, ISSN No. 2455-2143, Pages 41-49
Published Online October 2018 in IJEAST (http://www.ijeast.com)

OPTIMIZATION OF DESIGN PARAMETERS OF GO-KART

Vaibhav
Department of Mechanical Engineering
GLA University, Mathura, U.P., India

Nitin kukreja
Department of Mechanical Engineering
GLA University, Mathura, U.P., India

***Abstract*— the objective of the design is to optimize the working and performance of go-kart considering various factors. The modeling of our Go-kart was done using CATIA V5 software. The Finite Element Analysis was analyzed by ANSYS R18.2 software. Our central idea was to focus on designing a safe and functional vehicle that could be manufactured and be capable to compete in the market. This article summarizes the advances of vehicle dynamics which includes braking system, transmission system and steering system. Some of the attributes of the vehicle dynamics related to geometric attributes, attributes based on the mass and its distribution and attributes which are tire specific are also discussed in this paper.**

Keywords— **Go-kart, Finite Element Analysis, Vehicle dynamics, CATIA**

I. INTRODUCTION

Go-kart is an open wheel car powered by an I.C. engine/electric motor. It may be considered as a small prototype consisting of basic automobile components. The vehicle is required to be made with a combination of a frame and other integral parts such as steering, braking, transmission, etc. As weight serves critical in any vehicle operating on a small engine, there should be an optimum weight to power ratio. The frame has been designed accordingly ["Thavai et al. (2015) in their paper discussed about the material selection & modelling of the frame"]. Particular capabilities on basis of which the go-kart is judged includes static as well as dynamic.

The design of the go-kart focuses on the following objectives: Safety, Strength, Standardization, Cost, Driver ease, Ergonomics and Aesthetic consideration.

II. DESIGN CONSIDERATIONS AND ASTHETICS

Preliminary, several design considerations are to be defined before beginning the design of the frame and different parts. These decisions include transmission, intended steering, braking, assembly, fabrication methods. Besides these considerations the rules regarding the frame geometry, driver safety must also be considered ["Kyung G. et al. (2009) in their paper concentrated on the driver ease, working space & driving postures for ergonomic design"]. Our main aim was to design a Go kart which is cost effective, giving out low emissions, light in weight, high performing and safe and easy to fabricate as well["Gandhi et al. (2014) in their work addressed challenges related to safety, future fuels & reducing pollution in other words a smart Go-kart"]. CATIA V5 software has been utilized to model the design. Different views of the Go kart have been added here.

Figure 1: Front View of CAD Model of Go- Kart

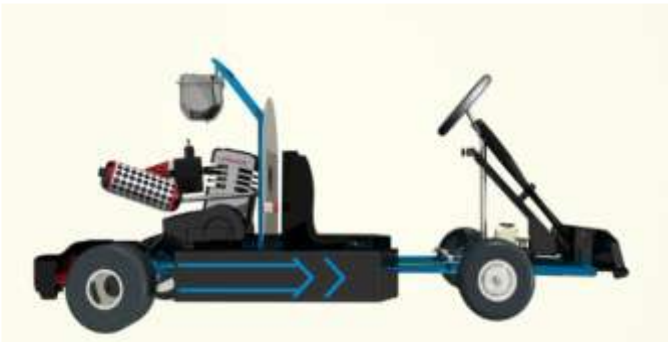

Figure 2: Side View of CAD Model of Go- Kart

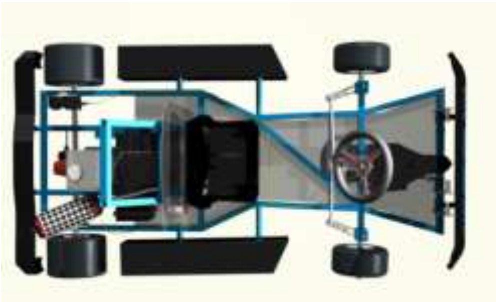
Figure 3: Top View of CAD Model of Go- Kart


Figure 4: Rear View of CAD Model of Go- Kart


Figure 5: Isometric View of CAD Model of Go- Kart

III. FRAME DESIGN & CALCULATIONS

Material selection is one of the major key points in designing the frame which increases the reliability, strength & performance of the Kart [“Yongfeng et al. (2018) in their work emphasizes on the lightweight material for the weight reduction for automobile applications.”]. For this, extensive research has been done & various materials are compared in different categories which include cost, strength & weight. So, the material that best suited our design values is AISI 1020 square pipe with 2mm wall thickness. This material has been chosen because of its weld ability, light weight, cost, strength & ductility.

Table 1. Mechanical Properties of Material

S. No.	Properties	Value (MPa)
1.	Density (x1000 kg/m3)	7.9
2.	Poisson’s Ratio	0.285
3.	Elastic Modulus (GPa)	200
4.	Tensile Strength (MPa)	395
5.	Yield Strength (MPa)	295
6.	Elongation (%)	36.5

Table 2. Chemical Composition of Material

S. No.	Element(S)	Weight %
1.	Manganese, Mn	0.30 - 0.60
2.	Carbon, C	0.18 – 0.23
3.	Sulphur, S	.05 (Max)
4.	Phosphorus, P	.04 (Max)
5.	Iron, Fe	Balance

A. FINITE ELEMENT ANALYSIS (FEA):

After the designing the analysis of the frame is done using ANSYS R18.2 software. Loads were calculated theoretically and placed on the frame at critical point to calculate the amount of deformation and stress generation at the time of collision, method best known as Finite Element Method. [“Pattansethi et al. (2016) in their work deals with the Design and Analysis of frame for the Go Kart using Finite Element Analysis under various stresses and forces”]. Front, side & rear impacts are the cases that were simulated. These impact values are calculated according to maximum performance of vehicle and global NCAP (New Car Assessment Program). Front, side & rear cases of impacts are calculated by the procedure given below:

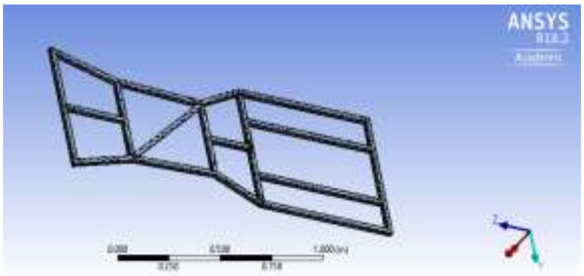

Figure 6: Meshing of Frame

A.1. Front impact

The velocity of the Kart is assumed to be 70 km/hr i.e. 19.44 m/s (V). Collision duration is small, so the duration is assumed to be 0.2 sec (t). Gross vehicle weight (assumed) = 150 kg (M), so the value of front impact force is calculated by mass moment equation:

$$F = \frac{M \times V}{t} \tag{1}$$

$$F = 14{,}580N \tag{2}$$

Now, the calculated force is applied on the frame by keeping the rear part fixed and the result is shown in the image below:

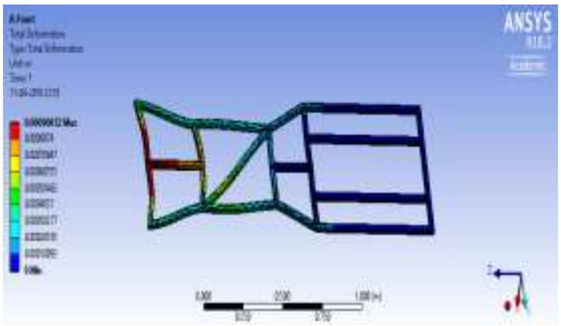

Figure 7: Result of Front Impact

A.2. Side Impact

Assumed velocity of the vehicle is 60 km/hr. i.e. 16.67 m/s. Assumed duration of time t=0.2sec.
Gross Vehicle Weight= 150kg. So the value of side impact force is calculated by mass moment equation:-

F= M x V/ t
F= 12502.5 N

Now, the calculated force is applied on the frame by keeping the side fixed and the result is.

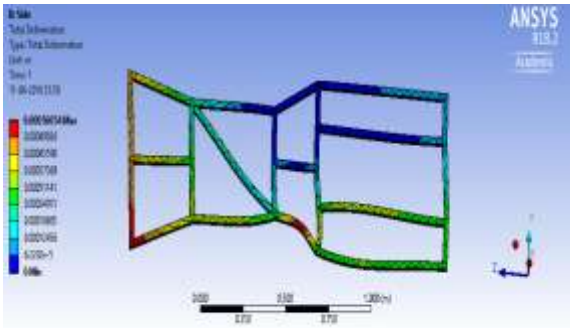

Figure 8: Result of side impact

A.3. Rear Impact

The velocity of the Kart is assumed to be 75km/hr i.e. 22.22 m/s (V). Collision duration is small, so the duration is assumed to be 0.2 sec (t). Gross vehicle weight (assumed) = 150 kg (M), So the value of rear impact force is calculated by mass moment equation.

$$F = \frac{M \times V}{t}$$

$$F = 16{,}665N$$

Now, the calculated force is applied on the frame by keeping the front fixed and the result is shown in the image below:-

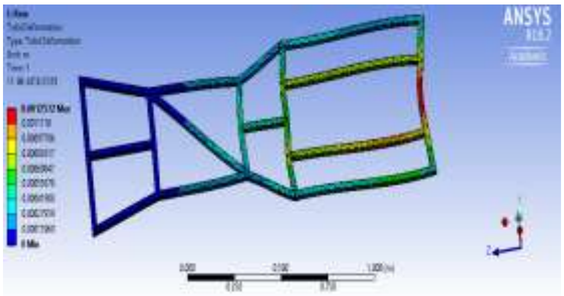

Figure 9. Result of rear impact

A.4. Torsional Rigidity

Overall structure rigidity is also important to enhance the capabilities and performance of the vehicle so the force is calculated by taking the average of our driver's weight and kerb weight i.e. 150 kg So,$F = 1500N$. This calculated force is placed on one of the corners of the frame while the rest of the corners are kept fixed. The result is displayed below in the image.

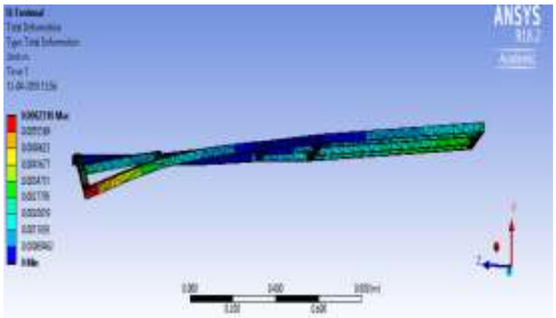

Figure 10. Result of torsional analysis

After the impact testing is done, the factor of safety is calculated by the formula.
F.O.S = (material strength/generated stress).

Hence the result Provided from these analysis modes are accurate for the amount of loading that was applied to the frame. With the collected data from FEA the safety factor has been found to be **2.01**. However, this result does not depict the actual scenario of the collision as the test was conducted by fixing the parts in spite of moving parts (dynamic form) during the real collision ["Renuke (2012) in their work discussed about modifying the frame according to the analysis of the frame related to the disturbances and natural frequencies on ANSYS"].

International Journal of Engineering Applied Sciences and Technology, 2018
Vol. 3, Issue 6, ISSN No. 2455-2143, Pages 41-49
Published Online October 2018 in IJEAST (http://www.ijeast.com)

IV.CENTER OF GRAVITY:

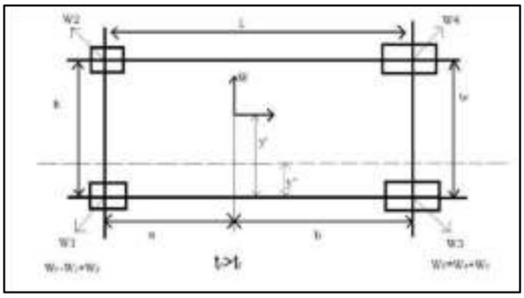

Figure11. Weight distribution of Go-kart

A. DESIGN PARAMETERS

Wheel base (l) = 1062mm = 41.811 inches
Track width (t) = 910 mm = 35.827 inches
tf = 910 mm = 35.827 inches
tr = 960 mm = 37.795 inches

The weight distribution of vehicle is assumed to be 2:3. The total weight of the vehicle without driver is 90 kg.

Calculations:

$$w_1 + w_2 + w_3 + w_4 = w \quad (3)$$

Let, $w_1 = 18kg, w_2 = 18kg, w_3 = 27kg, w_4 = 27\ kg$

$w = 90kg$

$$d = tr - tf/2 \quad (4)$$

=50/2
= 25 mm

$wf = 36kg,$

$wr = 54kg$ Now (Taking moment about rear axis)

$$b = (wf \times l/w) \quad (5)$$

= (36*1062)/90
= 424.8mm

$a = l - b$

= 1062 – 424.8
= 937.20mm

$$y' = w_2 \times (tf - d)/w - w_1 \times (d)/w + w_4 \times tr/w \quad (6)$$

= 18*(910-25)/90 – 18*(25)/90 + (27*960)/90
= 177– 5 + 288
= 460mm

Lateral Shift (With driver on board, W= 150kg)

$$y'' = w_2 \times tf\text{-}d/w - w_1 \times (d/w) + w_4 \times tr/w - tr/2 \quad (7)$$

= 18*(910-25)/150 – 18*(25)/150 + (27*960)/150-960/2
= 106.2-3+172.88-480 = -203.92 mm

Negative sign denotes shifting of C.G to the front.

The co-ordinates of C.0.G are (424.8, 460) from rear axle (without driver).

The co-ordinates of C.0.G are (628.7, 460) from rear axle (with driver).

Thus the center of gravity of the kart is calculated [“Mango N. (2004) in their work emphasizes the calculation of vehicle`s Centre of Gravity (CG) affecting the performance parameters of the car & the method used by racing teams”].

V. STEERING SYSTEM DESIGN AND CALCULATIONS:

The steering system converts the rotation of the steering wheel into a swiveling movement of the road wheels in such a way that the steering-wheel rim turns a long way to move the road wheels a short way. It is used for changing the direction of the vehicle. Vehicle steering is not only required on a curved road but also on maneuvering on the busy traffic roads. A steering system must offer sufficient precision for the driver to actually sense what is happening at the front tyres contact patch as well as enough “feel” to sense the approach to cornering limit of the front tyres. It must be structurally stiff to avoid components deflections and for this purpose Ackermann steering mechanism has been used [“Zhao J.S.et al. (2013) studied about different types of mechanism related to the steering geometry & investigating the turning geometry for steering wheels in their work.”].

A. OBJECTIVE

The basic aim of steering is to ensure that the wheels are pointing in the desired directions. This is typically achieved by a series of linkages, rods, pivots and gears. The main goal for steering is to have turning radius 3m or less. It is designed to withstand the stress of safely maneuvering the vehicle.

B. DESIGN

The steering system must be able to turn the front wheels sharply yet easily and smoothly. So while designing the steering system, the constraints that we possessed were Centre alignment of steering system, track width human effort at the steering wheel and the desired response of the steering system.

Figure 12: Ackermann steering arrangement

While designing the important angles kept in mind were:

B.1. Caster angle

It is the angle between the pivot line (in a car - an imaginary line that runs through the Centre of the upper ball joint to the Centre of the lower ball joint) and vertical. Caster angle can be adjusted to optimize a car's handling characteristics for particular driving situations.

B.2. Camber Angle

Camber (or camber angle) is the angle of the tire from zero degrees (meaning straight up and down) when viewed from the front of the vehicle. If the bottom of the tire is further out than the top of the tire, this is known as Negative Camber Angle. And if the top of the tire is further out than the bottom, this is known as Positive Camber Angle.

B.3. King Pin Inclination

The king pin is tilted at the top towards the vehicle in order that the weight of the load will be thrown towards the tire Centre. It is used to give self-centering effect of the vehicle.
For correct steering:

$$\cot Ø - \cot \theta = w/L \tag{8}$$

For inner angle: $Tan\ \theta = L/(R - w/2)$ (9)
For outer angle: $Tan\ Ø = L/(R + w/2)$ (10)

The Wheel Base of vehicle, L =1220mm Width to wheel base ratio is 0.852
Fundamental steering equation is $\cot Ø - \cot \theta = w/L$
To get a turning radius of 1800mm
$Tan\ Ø = L/(R+w/2)$
$Tan\ Ø = 1220/(1800+1040/2)$
$Ø = 27.73$
$Tan\ \theta = L/(R-w/2)$
$Tan\ \theta = 1220/(1800-1040/2)$
$\theta = 43.62°$
Angle made by outer sub axle should be 27.73° and inner stub axle should be 43.62°. Putting these values in fundamental equation of steering and checking if they are satisfying or not.
$\cot Ø - \cot \theta = w/L$
$\cot 27.73° - \cot 43.62° = 0.645$ W/L =0.645
For Ackermann angle;

$$Tan\ \alpha = (\sin Ø - \sin \theta)/(\cos \theta + \cos Ø - 2) \tag{11}$$

$Tan\ \alpha = (\sin 27.73° - \sin 43.62°)\ /\ (\cos 43.62° + \cos 27.73° - 2$
$\alpha = 29.4°$

Value of Ackermann angle is 29.4°.

Where $Ø$ = outer angle of the wheel (During turning), θ= inner angle of the wheel (During turning), R= turning radius of the wheels, w= width of the car, L= Wheel base of the car.

C. STEERING SPECIFICATIONS

Table 3. Specification of Steering System

S. No.	Specification	Value
1.	Wheel Base	1220mm
2.	Track Width	1040mm
3.	Pin to Pin (spindle)	603.2mm
4.	Turning Radius	1800mm
5.	Outer Turn Angle	27.73°
6.	Inner Turn Angle	43.62°
7.	Ackermann Angle	29.4°
8.	Ackermann angle %	148.3%
9.	Steering Wheel dia.	304.8mm
10.	Steering Ratio	6:1
11.	Steering Wheel Torque	13.24Nm
12.	Turns (lock to lock)	1220mm
13.	King Pin Inclination	0°
14.	Caster Angle	8°
15.	Camber Angle	2°

VI. BRAKING SYSTEM

The brakes of a vehicle have to absorb the energy given to the vehicle by the engine plus that due to the momentum of the vehicle [“Zulkifly M.B.,(2007) in their work addressed issues related to suitable braking system keeping in consideration the speed of the car, weight of the car etc. & they also discussed the challenges and the troubleshooting related to the braking system”].
The brake must also pull up the vehicle smoothly and is a straight line.

Fig. 13: Disk Brake Arrangement

International Journal of Engineering Applied Sciences and Technology, 2018
Vol. 3, Issue 6, ISSN No. 2455-2143, Pages 41-49
Published Online October 2018 in IJEAST (http://www.ijeast.com)

A. DESIGN METHODOLOGY:

Following are the optimum brake parts which are combination help to achieve target. Optimize brake design. Major components of braking system are:

- Brake pedal.
- Master cylinder.
- Brake line.
- Brake disc.
- Brake caliper.

B. DESIGN OF BRAKING SYSTEM:

Brake pedal: we assumed pedal ratio of 4:1 i.e. when driver applies 1N force it gets multiplied by 4 times to produce a 4N force. Master cylinder: Maruti Suzuki 800. Caliper & rotor: We compared the rotor assemblies of Apache RTR 160 and Honda Aviator finally we zeroed on Apache RTR which had following dimension- Diameter of rotor = 200mm. Caliper piston diameter = 32mm.

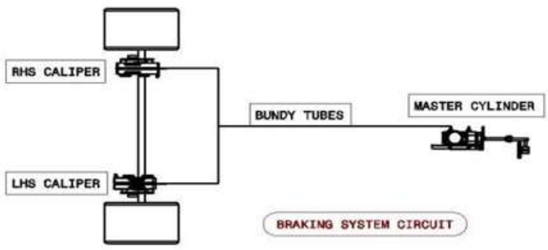

Fig 14: Braking System Circuit

C. BRAKING CALCULATION

Maximum force a driver applies on brake pedal = 22g.
Pedal ratio = 4:1.
Force on master cylinder piston (FMC) = 22 x 9.81 x 4
= 863.28 N.
Area of master cylinder (AMC) = [π x $(0.018)^2$]/4
= 2.545 x 10^{-4} m^2.
Pressure develops in the system= FMC/AMC (12)
= 3.392 MPa.
According to **Pascal's law** states-"that a pressure change occurring anywhere in a confined incompressible fluid is transmitted throughout the fluid such that the same change occurs everywhere".

Area of brake caliper
$A = [\pi \times 0.032^2]/4.$
= 8.042 x 10^{-4} m^2.
Therefore, the force on brake caliper
$F = P \times ACAL.$ (13)
= 2728.089 N.

Total force on rotor = 2728.089 x 2.
= 5456.178 N.
Total friction force=clamp force × coefficient of friction.
= 5456.178 N x 0.65
= 3546.51 N.
Torque on rotor = Frictional force × effective radius of rotor
= 2728.165 x 0.085.
= 283.7213N-m.
Force acting on one tire = Torque on rotor /radius of tyre
= 283.7213/0.1397
= 2026.581 N
Total brake force using selected brake force
= 1659.94 x 2
= 4053 N.
Deceleration = $force$/mass of vehicle
= 4053/140
= 28.95 m/s^2
= 2.95g.
Stopping distance:
At 40 Km/h
$Stopping\ distance = (maximum\ velocity)^2/(2 \times deceleration)$

= $(11.11)^2$/ (2 x 28.95)
= 2.13 m.
At 60 Km/h
$Stopping\ distance = (maximum\ velocity)^2/(2 \times \text{deceleration})$
= $(16.66)^2$/ (2 x 28.95)
= 4.8 m.
At 80 Km/h
$Stopping\ distance = (maximum\ velocity)^2/(2 \times \text{deceleration})$
= $(22.22)^2$/ (2 x 28.95)
= 8.53 m.

Table 4: Braking System Specification

S. No.	Types of Brakes	Disk brake arrangement system
1.	Disc diameter	200mm
2.	Thickness of disc	4.5mm
3.	Effective radius of disc	85 mm
4.	Brake pedal force	22g
5.	Pedal ratio	4:1
6.	Coefficient of friction	0.5
8.	Torque on rotor	283.7213N-m
9.	Stopping distance (at 60kmph)	4.8 m
10	Disc diameter	200mm

VII. DRIVETRAIN DESIGN:

The drivetrain includes the transmission, the driveshaft, the axles, and the wheels. Simply put, it works in conjunction with the engine to move the wheels. The drivetrain system is an essential component of a vehicle and the transmission is an integral part of the drivetrain.

Figure15: Engine coupled at rear axle.

A. CALCULATIONS:

Vehicle design criteria:

Gross vehicle weight (GVW): 140Kg =1372 N
Weight on each drive wheel (WW):35Kg=343 N
Radius of wheel/tire (R_w): 0.14 m
Desired top speed (V_{max}): 20.83 m/s
Desired acceleration time (t_a): 10 sec
Maximum incline angle (β): 3 degree
Working surface: Asphalt (good)

It is necessary to determine the total tractive effort (TTE) requirement for the vehicle to select motors capable of producing enough torque to propel the kart.

$$TTE\ [N] = RR\ [N] + GR\ [N] + FA\ [N] \quad (14)$$

Where:

TTE = total tractive effort [N]
RR = force necessary to overcome rolling resistance [N]
GR = Grade resistance to climb a slope [N]
FA = force required to accelerate to final velocity [N]
The components of this equation will be determined in the following steps.

<u>Step One: Determine Rolling Resistance</u>

Rolling Resistance (RR) is the force necessary to propel a vehicle over a particular surface. The worst possible surface type to be encountered by the vehicle should be factored into the equation.

$$RR\ [N] = GVW\ [N]\ x\ Crr \quad (15)$$

RR = 1372 x 0.017 = 23.324N
Where:
RR = rolling resistance [N]
GVW = gross vehicle weight [N]
Crr = surface friction (0.017 for fair asphalt)

<u>Step Two: Determine Grade Resistance</u>

Grade Resistance (GR) is the amount of force necessary to move a vehicle up a slope or "grade". This calculation must be made using the maximum angle or grade the vehicle will be expected to climb in normal operation.
To convert incline angle, α, to grade resistance:

$$GR\ [N] = GVW\ [N]\ x\ sin\ (\alpha) \quad (16)$$

GR = 1372 x sin (3°)
= 71.80N Where:
GR = grade resistance [N]
GVW = gross vehicle weight [N]
α1=maximum incline angle [degrees]

<u>Step Three: Determine Acceleration Force</u>

Acceleration Force (FA) is the force necessary to accelerate from a stop to maximum speed in a desired time.

$$FA\ [N] = GVW\ [N]\ x\ Vmax\ [m/s]\ /\ (9.8\ [m/s2]\ x\ ta\ [s])$$

FA= 1372 x20.83 / (9.8*10)
= 291.62N Where:
FA = acceleration force [N]
GVW = gross vehicle weight [N]
Vmax = maximum speed [m/s] ta = time required to achieve maximum speed [s]

<u>Step Four: Determine Total Tractive Effort</u>

The Total Tractive Effort (TTE) is the sum of the forces calculated in steps 1, 2, and 3. (On higher speed vehicles friction in drive components may warrant the addition of 10%-15% to the total tractive effort to ensure acceptable vehicle performance.)

$$TTE\ [N] = RR\ [N] + GR\ [N] + FA\ [N]$$

TTE = 23.324+71.80+291.62= 386.74N

<u>Step Five: Determine Wheel Motor Torque</u>

To verify the vehicle will perform as designed in regards to tractive effort and acceleration, it is necessary to calculate the required wheel torque (Tw) based on the tractive effort.

$$Tw\ [Nm] = TTE\ [N]\ x\ Rw\ [m]\ x\ RF\ [-] \quad (17)$$

Tw = 386.74 x 0.14 x 1.1= 59.55Nm

Where: Tw = wheel torque [Nm] TTE = total tractive effort [N]
Rw = radius of the wheel/tire [m]
RF = "resistance" factor [-]
The "resistance factor" accounts for the frictional losses between the caster wheels and their axles and the drag on the motor bearings. Typical values range between 1.1 and 1.15 (or 10 to 15%).

<u>Step Six: Reality Check</u>

The final step is to verify the vehicle can transmit the required torque from the drive wheel(s) to the ground. The maximum tractive torque (MTT) a wheel can transmit is equal to the normal load times the friction coefficient between the wheel and the ground times the radius of the drive wheel.

$$MTT = Ww\,[N]\; x\; \mu\,[-]\; x\; Rw \tag{18}$$

MTT= 412 x 0.6 x 0.14= 34.608 Nm

Where:
Ww = weight (normal load) on drive wheel [N]
μ = friction coefficient between the wheel and the ground (~0.7 for slick tyre on asphalt) [-]
Rw = radius of drive wheel/tire [m]
An appropriate acceleration time must be chosen such that the required Tw < MTT × number of drive wheels.
53.50<34.608x2 i.e. true

VIII. OVERALL SPECIFICATIONS

So the best suited and available components for the above values are:

Engine: Honda Stunner 125cc or similar engine of rated capacity.
Capacity: 124.7cc
Power (PS): 11@8000 rpm
Torque (Nm): 11@6500 rpm
Final Reduction (Bigger Sprocket) =3.071
Final Reduction (Smaller Sprocket) = 2.71

IX.CONCLUSION

After various calculations and research we came to the conclusion that the optimization of the various design parameters including vehicle dynamics parameters achieved by us is best to our consideration which involves the factors such as steering calculations, braking calculations and various design calculations.

In our research we achieved Ackermann angle of 29.4 degrees, stopping distance (60km/hr.) of 4.8 m, whereas we achieved the overall Factor of Safety (F.O.S) of 2.01.

Table 5: Conclusion of the safety analysis

S. No.	Front	Side	Rear	Torsion
Test Parameters	70 Km/hr	60 Km/hr	75 Km/hr	150 kg
Load (N)	14580	12502.5	16665	1500
Total Deformation (m)	0.908	0.56	1.25	6.25
Equivalent Stress	126	125	172	177
Factor of Safety	2.33	2.35	1.71	1.66

X. REFERENCES

1. Journals

[1] Patil N.R., Kulkarni Ra.R., Mane B.R., Malve S.H., (2014) "Static Analysis Of Go-Kart Chassis Frame By Analytical And Solid works Simulation", International Journal Of Scientific Engineering And Technology (ISSN: 2277-1581), Volume No.3 Issue No.5, Pp : 661-663

[2] Abdul Aziz F. & Sapuan S. & Mohd ariffin M. K. A. & Baharudin, B T & Supeni E.,(2009). "Design and fabrication of a student competition based racing car". Scientific Research and Essay. Vol. 4 (5) pp. 361-366 ISSN 1992-2248

[3] Parvez Hussain S. D, C. N. Veeramani, B. Amala Priya Shalini, R. Karthika.,(2014) "An Innovative Energy Efficient Automobile Design, "International Journal of Innovative Science and Modern Engineering.Vol. 2, ISSN: 2319-6386.

[4] Kyung G. & Nussbaum M.A. (2009) "Specifying comfortable driving postures for ergonomic design and evaluation of the driver workspace using digital human models, Ergonomics", DOI: 10.1080/00140130902763552

[5] Gandhi A. & Kansal H.,(2014), "Troubleshooting and Its Solutions in Designing of a Smart Go Kart", IJRMET Vol. 4, Issue: 2, spl- 2 May -October 2014 ISSN: 2249-5762 (Online).

[6] Yongfeng P. & Ma F. & Zhang J. & Yang M. (2018). "Optimal lightweight material selection for automobile applications considering multi-perspective indices". IEEE Access. PP. 1-1. 10.1109/ACCESS.2018.2804904.

[7] Dr. Ravikanth D., Nagaraja C., Dr. Rajagopal K., Dr. Murthy V.S.S., (2015) "Fabrication of a Model Go-Kart (With Low Cost)" Journal of Mechanical and Civil Engineering (IOSR-JMCE) e-ISSN: 2278-1684,pISSN: 2320-334X, Volume 12, Issue 6 Ver., PP 24-30

[8] Pattanshetti V.S., "Design & Analysis of Go-kart Chassis",(2016), International Journal of Mechanical and Industrial Technology, Vol. 4, Issue 1, pp: (150-164) ISSN: 2348-7593 (Online)

[9] Nath A., Vikram C.J., Lalchhanchhuah, Lalrinsanga, Nongrum L., Marboh P.,(2015) "Design and Fabrication of a GoKart" International Journal of Innovative Research in Science, Engineering and Technology, Vol. 4, Issue 9, ISSN No.: 2319-8753.

[10] Renuke P.A., (2012), “Dynamic Analysis Of A Car Chassis”, International Journal Of Engineering Research And Applications, ISSN: 2248-9622, Vol. 2, Issue 6, November- December 2012, PP.955-959 955.

[11] Mango N. (2004) “Measurement and calculation of vehicle center of gravity using portable wheel scales”. DOI: 10.4271/2004-01-1076.

[12] Thavai R.,Shahezad Q.,Shahrukh M.,(2015), “Static Analysis of GoKart Chassis by Analytical and Solid Works Simulation”, International Open Access Journal of Modern Engineering Research (IJMER) Vol. 5, Iss.4, Apr. 2015.

[13] James Valder, Vishwas Naik, Zarnab Ahmed, Vishwajit Ghorpade, Daniyel Ahmed, Design and Fabrication of Magnetic Braking System, Journal of Mechanical Engineering and Automation, Vol. 7 No. 4, 2017, pp. 116-118. doi: 10.5923/j.jmea.20170704.06.

[14] Zhao, Jing-Shan & Feng Xiang Liu, Zhi-Jing & Dai, Jian. (2013). Design of an Ackermann Type Steering Mechanism. Journal of Mechanical Engineering Science. 227.
DOI: 10.1177/0954406213475980.

[15] Hac, A., Brown, T., and Martens, J., (2004), "Detection of Vehicle Rollover,”. DOI: 10.4271/2004-01-1757

2. Books

[1] Essay UK, Criteria for Selection of Material for Automobile Design.

[2] Gillespie Thomas D., “Fundamentals of Vehicle Dynamics” Society of Automotive Engineers Inc. Publications.

[3] Dr. Kirpal Singh, “Automobile Engineering vol-1&2”, Standard Publishers Distributors.

[4] William Crouse and Donald Anglin, ‘’Automotive Mechanics Tenth Edition’’