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Optimizing Go-Kart Performance Via Cg Placement And Load Distribution

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Abstract**—** This paper explores influential factors in the optimization of go-kart performance through practical tweaks. Low center of gravity enhances stability, whereas forward seating adds front grip (but results in understeer) and rear seating improves cornering (with potential oversteer). Equal weight distribution over axles yields maximum tire contact and traction. Testing indicates that softer tires are better at gripping but have a shorter lifespan, whereas harder tires have a longer lifespan but offer less traction. Driver seating modification and suspension adjustments further enhance performance. Such alterations, together with real-time track data analysis, allow racers to achieve ideal balance between speed, handling, and consistency

Index Terms**—** Go-kart performance, center of gravity, weight distribution, tire selection, vehicle dynamics

I. Introduction

A go-kart is a high-performance, open auto racing car built to compete on flat tracks, with its rigid structure, no suspension, and mechanical steering without any intermediate links. New materials and computer engineering allow lighter weight with more efficient design without compromising the structure. Handling is controlled by the load and position of the CG, with cornering traction, acceleration, and braking efficiency affected. Roll moment is reduced with a lower CG, enhancing lateral stability, and evenly loaded front and rear optimizes traction and steering response.

Chassis geometry and mass distribution are optimized through the utilization of computational simulations, allowing better transient response and consistency of lap times.

For optimal stability, cornering grip, and overall performance, accurate CG optimization is necessary. Component positioning is key—the motor and battery must be placed low and close to the kart's geometric center to reduce inertial weight transfer, and the driver's seat must be laterally balanced. Chassis geometry has a large impact on weight distribution: a longer wheelbase provides improved straight-line stability but at the cost of agility, and a wider track width lowers the CG and increases cornering traction. Material choice is important, with high-strength, low-weight alloys such as AISI 4130 chromoly steel providing optimal rigidity-to-weight ratios. Even though go-karts produce very little aerodynamic downforce, driver location and component positioning must be designed to reduce drag-induced instability. By addressing these variables systematically—using computational modeling and empirical testing—engineers can determine an optimal CG location that improves acceleration, braking, and high-speed cornering while preserving structural integrity and competitive performance.

II. Role of Center of Gravity in Go-Kart Performance Optimization

The center of gravity (CG) is a highly significant parameter in go-kart dynamics that has a significant effect on vehicle stability, handling characteristics, and overall performance. As a highly significant parameter that affects weight distribution during acceleration, deceleration, and cornering, precise CG position is necessary to achieve optimum lap times with driver control and safety in mind.

Vertical CG position has a strong effect on roll behavior in that lower CG height diminishes body roll moments and ensures better tire contact patch consistency. Cornering grip and stability are therefore improved, which is especially important for high-speed cornering. Increasing CG, however, increases roll vulnerability and, in deep maneuvers, may cause traction loss or even rollover. Modern kart construction employs extremely sophisticated materials such as chromoly steel tubing, which lightens mass without compromising structural stiffness, thereby enabling optimal CG positioning

Longitudinal CG Longitudinal front-to-rear weight distribution significantly affects handling balance. Forward positioning of the CG (about 40% front) enhances steering response but could cause understeer, and rearward positioning enhances acceleration but at the expense of potential oversteer. The optimal 40/60 front/rear distribution is a performance compromise between turn-in responsiveness and rear traction

Lateral CG placement is also essential with a 50/50 left-right split to balance cornering loads and avoid single-sided handling behavior. Side-to-side symmetry guarantees equal tire loading when cornering left or right, to achieve maximum grip levels and avoid uneven tire wear patterns that would compromise performance over long distances

III. CHASSIS GEOMETRY AND DESIGN

The operation of a go-kart depends on the center of gravity (CG) position being optimal and load distribution evenly spread across its chassis. These are the primary determinants that the chassis geometry, material selection, and structural configuration control. Optimal CG location improves stability, handling, and traction, whereas the utilization of lightweight materials and close tolerancing enhances speed and handling. The synthesis of these factors produces a reactive, high-performance vehicle for competitive racing.

The ladder frame chassis continues to be a backbone of high-performance go-kart design due to its excellent combination of structural stiffness, dynamic response, and design flexibility. The structure, comprised of two longitudinal beams and cross-joined with carefully placed cross members, achieves a highly sought-after compromise of torsional stiffness and mass efficiency. This compromise is necessary for sustaining the best power-to-weight ratios, which are necessary for competitive motorsport use. Of special interest, ladder frames will have torsional rigidity values of more than 15% compared to other designs, hence structural toughness under high-load conditions.

Constructed mostly of AISI 4130 chromoly steel with tensile capacities in excess of 670 MPa, the chassis features a lower polar moment of inertia. This reduced yaw resistance significantly improves directional stability during high-speed maneuvers, particularly with lateral forces in excess of 1.5g. With accurate CG placement usually positioned in a 40/60 front-to-rear split handling characteristics are optimized, enabling even tire loading under all dynamic changes such as acceleration, deceleration, and cornering.

Its modularity permits practical modification, such as wheelbase adjustment, stress-critical node reinforcement, and aerodynamic enhancement in the form of diffusers and flow management surfaces, without sacrificing frame integrity. Finite Element Analysis (FEA)-based models also permit perfect stress distribution along load paths, minimizing flex that might undermine steering precision or tire alignment particularly vital in go-karts without suspension systems, where the chassis has to respond directly to all inputs from the road. In addition, the simple nature of the ladder frame ensures that it is easy to produce and prototype, making it cost-effective and precise perfect for motorsport application with the requirement for rapid iteration and tight tolerances. Its ongoing relevance to karting from novice classes to the top level of competition proves to be a winning blend of fundamental engineering concepts and modern performance optimization methods.

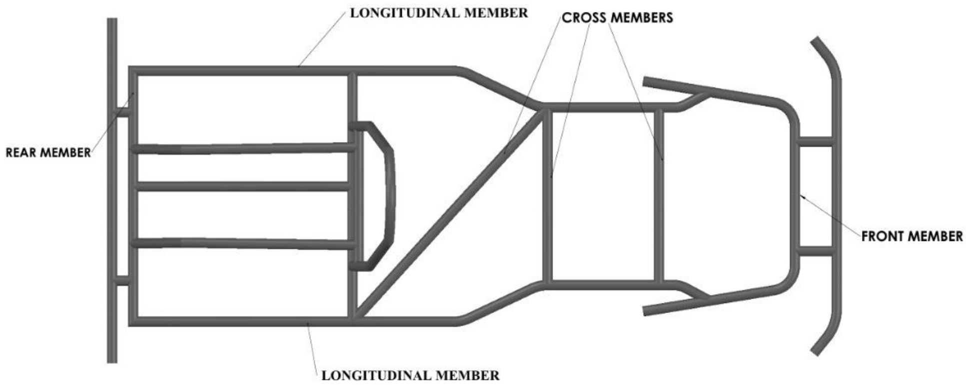

Figure 1 Ladder frame chassis

The load distribution is at the center of the dynamic response of a go-kart. It defines how the overall weight of the vehicle is distributed between its four wheels, which in turn influences traction, handling, and stability. The longitudinal and lateral weight distribution is determined by the position of significant components like the motor, seat, and battery, and this influences the position of the Centre of Gravity (CG). A rear-biased setup improves acceleration traction, while a forward bias can improve steering response but can worsen understeer. The chassis geometry, such as the wheelbase and track width, also influences load transfer. A longer wheelbase provides more straight-line stability, while a wider track width improves cornering grip by optimizing lateral load distribution. The control of CG position and load balance is especially critical in go-karts without suspension, as the chassis responds directly to road forces. The optimization of this balance is required to achieve predictable handling and improve lap-time consistency.

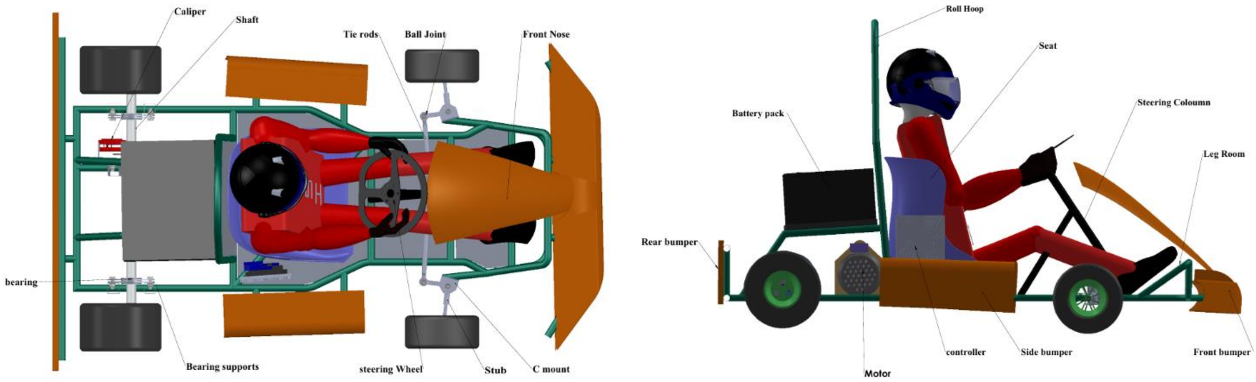

Figure 2 Components Placement

Torsional rigidity:

Torsional stiffness performs a vital role in ensuring the structural integrity of a go-kart chassis with a stable center of gravity (CG) under dynamic operating conditions. When a chassis is able to withstand twisting loads encountered during cornering, acceleration, or braking, the CG remains stable allowing predictable vehicle behavior. A stiff chassis avoids excessive lateral or longitudinal movement of CG, which otherwise has a tendency to impose excess stress on specific tires, resulting in phenomena like understeer, oversteer, or loss of traction. It also ensures an equal load distribution to all wheels, thereby improving grip and minimizing tire wear. Additionally, a stiff chassis provides for a lower vertical CG, reducing the roll moment and enhancing cornering stability. In a torsional stiffness test, torque is applied calculated as the product of force and the perpendicular distance to replicate real-world loads. By retaining its structural shape under such loads, the chassis provides the best handling, effective braking performance, and predictable weight transfer, all of which are vital in high-performance, suspension-less go-kart configurations.

Torque applied = load applied × perpendicular distance

T= P x L

= Mg X L

= 200 X 9.81 X 1(total mass X perpendicular length)

= 1962 N-m

The torsional stiffness of the chassis is analyzed using advanced finite element analysis (FEA) to simulate real-world conditions with precision, in this test, a torsional load is applied as a couple on the front members of the chassis, replicating the forces experienced during cornering or uneven loading. The applied force is equivalent to the total weight of the vehicle, assumed to be 200 kg, translating to a force of 1962 N . to be 200 kg, translating to a force of 1962 N.

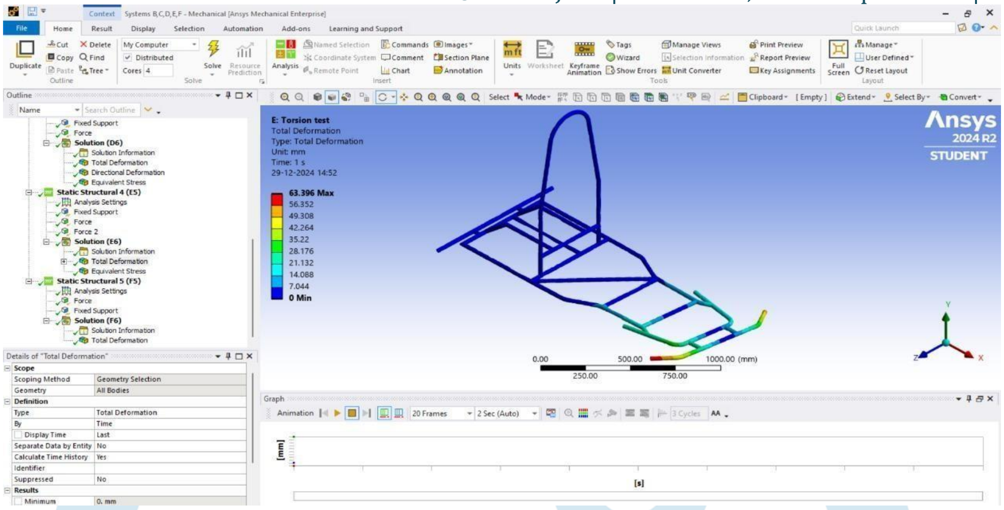

Figure 3 Result torsional test

Maximum deformation in mm = 63.396 mm in vertical plane

IV. STATICAL CALCULATIONS

Steering calculations optimize handling and stability by balancing turning radius, Ackermann angle, kingpin inclination, and scrub radius minimizing tire scrub and maximizing grip during high-speed cornering.

Ackerman steering:

Front track (a) = 36 inches

Wheelbase (b) = 43.5 inches

Stub axis (c) = 24 inches

$$Ackerman\ angle\ = (\propto) = \tan^{-1}\left(\frac{c}{2 \times b}\right)$$

$$= \tan^{-1}\left(\frac{24}{2 \times 43.5}\right)$$

$$= 15.4°$$

$$Inner\ wheel\ angle\ (\theta) = \tan\left(\frac{\theta}{2}\right) = \frac{a/2}{b}$$

$$\theta = 2 \times \tan^{-1}\left(\frac{a/2}{b}\right)$$

$$= 44.95°$$

$$Outer\ wheel\ angle\ \emptyset = \cot\emptyset - \cot\theta = \frac{c}{b}$$

$$\frac{1}{\tan\emptyset} - \frac{1}{\tan 44.95°} = \frac{24}{43.5}$$

$$\emptyset = 32.77°$$

$$Ackerman\ value = \tan^{-1}\left(\frac{WB}{\frac{WB}{\tan\emptyset} - front\ track}\right)$$

$$= \tan^{-1}\left(\frac{43.5}{\frac{43.5}{\tan 32.77°} - 36}\right)$$

$$= 54.02$$

$$Ackerman\ \% = \frac{\theta}{Ackerman\ value} \times 100$$

$$= \frac{44.95°}{54.02} \times 100$$

$$= 83\%$$

$$Turning\ radius = \frac{WB}{2 \times \sin\propto}$$

$$= \frac{43.5}{2 \times \sin 15.4°}$$

$$= 81.90\ inches$$

Steering effort:

Weight of the vehicle = 200 Kg = 1962 N

Weight of front wheel = 30% = 60 Kg = 588.6 N

Sliding friction between tire and road (μ) =0.8

$$Friction\ force\ = friction\ coefficient \times weight\ of\ front\ track$$

$$0.8 \times 588.6$$

$$= 470.88\ N$$

$$Force\ at\ knuckle = \frac{(frictional\ force \times\ Scrub\ radius\ )}{Steering\ arm\ length}$$

$$= \frac{(470.88 \times 55mm)}{100}$$

$$= 258.9\ N$$

$$Radius\ of\ steering\ wheel\ = 5\ .7inch = 14.7cm = 147mm$$

$$Torque\ at\ the\ tripod = tripod\ length \times force\ at\ knuckle$$

$$= 100 \times 258.9$$

$$= 25898.4\ N - mm$$

$$Steering\ effort = \frac{Torque}{Steering\ wheel\ radius}$$

$$= \frac{25898.4}{147}$$

$$= 176.17\ N = 17.9\ Kg$$

Steering Efforts for Dynamic Condition:

Weight transfer due to breaking $W_{Rt} = 144.2 \times 27\% = 38.93\ Kg = 381.94N$

Force required to turn the wheel,

$$= 381.94 + 588.6 = 970.54N$$

Friction force to overcome $= \mu \times Weight$

$$= 0.45 \times 970.54$$

$$= 436.74N$$

Force at knuckle plate= $\frac{(Frictional\ force \times Scrub\ radius)}{Steering\ arm\ length}$

$$= \frac{(436.74 \times 55)}{100}$$

$$= 240.2N$$

Radius of steering wheel = 147 mm

Torque at tripod $= Tripod\ iength \times Force\ at\ knuckle\ plate$

$$= 100 \times 240.2$$

$$= 24020\ N * mm$$

Steering effort= $\frac{Torque}{Steering\ wheel\ radius}$

$$= \frac{24020}{147} = 166.4N$$

=16.95 kg.

center of gravity:

W=The total weight =200 Kg

WF= weight of front wheels $= 30\% = 60\ Kg$

WR=Weight of rear wheel= $70\% = 140\ Kg$

l =Wheelbase= $43.5\ inch$

TF=Track length of front= $36\ inch$

TR=Track length of rea= $35\ inch$

$\theta = 40°$

The total vehicle weight is the sum of the four individual wheel weights measured on level ground beneath each tire:

$W_1 + W_2 + W_3 + W_4 = W = Total\ Vehicle\ Weight$

$30.8 + 29.2 + 69.1 + 70.9 = 200\ Kg$

Total Vehicle longitudinal Location of the CG:

$$b = \frac{W_F \times l}{W}$$

$$b = \frac{60 \times 43.5}{200}$$

$$b = 13.05\ inch$$

$$a = l - b$$

$$a = 43.5 - 13.05$$

$$a = 30.45 inch$$

$$d = \frac{(t_f - t_r)}{2}$$

$$d = \frac{36 - 35}{2}$$

$$d = 0.5\ inch$$

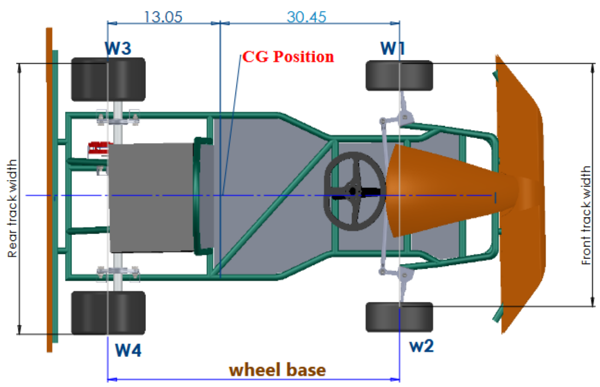

Figure 4 longitudinal CG position

Total Vehicle lateral Location of the CG:

Next take moments about the $X_1 - X_1$axis (a line parallel to the centreline of the car through the centre of the left rear tire).

$$y' = \frac{W_2}{W}(t_F - d) - \frac{W_1}{W}(d) + \frac{W_4 t_R}{W}$$

$$y' = \frac{29.2}{200}(36 - 0.5) - \frac{30.8}{200}(0.5) + \frac{70.9 \times 35}{200}$$

$$y' = 17.41\ inch$$

This can then be solved for y ^ prime (since y''=y' -$\left(\frac{t_R}{2}\right)$) To give the lateral shift of the CG

(if any) from the x axis (centreline):

$$y'' = \frac{W_2}{W}(t_F - d) - \frac{W_1}{W}(d) + \frac{W_4 t_R}{W} - \frac{t_R}{2}$$

$$y'' = \frac{29.2}{200}(36 - 0.5) - \frac{30.8}{200}(0.5) + \frac{70.9 \times 35}{200} - \frac{35}{2}$$

$$y'' = 0\ inch$$

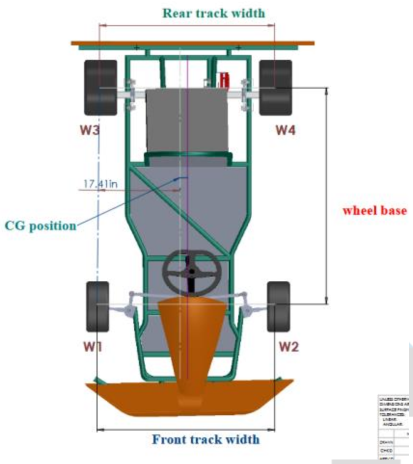

Figure 5 lateral CG position

Total Vehicle Vertical Location of the CG:

Rw=Radius

Rh= loaded height of the tire

Rg= The radius of tire

Rw = $\frac{V_x}{W}$

V_x =Wheel forward velocity

W=Wheel angular velocity

$R_H = 10cm$ for front track, $R_g = 12cm$

$R_H = 11cm$ for rear track, $R_g = 14\ cm$

Rolling radius front $R_W = \left(\frac{2}{3}\right) \times R_g + \left(\frac{1}{3}\right) \times R_h$

$$= \frac{2}{3} \times 12 + \frac{1}{3} \times 10$$

$$R_{wf} = 11cm = 4.3inch$$

Rolling radius Rear $R_W = \left(\frac{2}{3}\right) \times R_g + \left(\frac{1}{3}\right) \times R_h$

$$= \frac{2}{3} \times 14 + \frac{1}{3} \times 11$$

$$R_{wr} = 13cm = 5.11inch$$

Centre at the CG location must be found by

$R_L CG = R_{LF}\left(\frac{b}{l}\right) + R_{LR}\left(\frac{a}{l}\right)$

$= 11\left(\frac{13.05}{43.5}\right) + 13\left(\frac{30.45}{43.5}\right)$

$R_L CG = 12.4cm = 4.8\ inch$

H=$R_L CG + h_1$

H=4.8+ 0

H=4.8 in =12.1cm

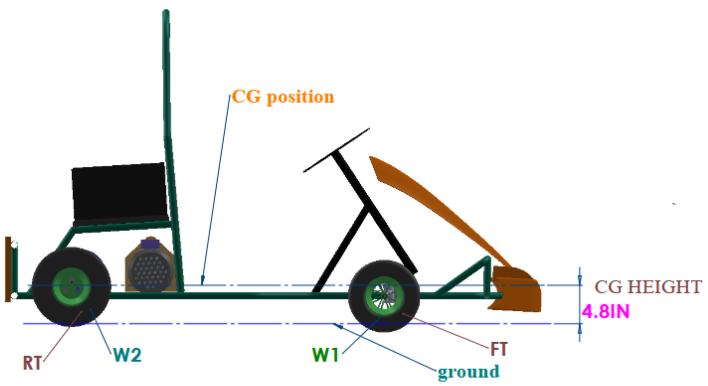

Figure 6 Vertical CG position

Lateral load transfer:

Cornering force (F)= $\frac{M \times V^2}{R}$

Mass of the vehicle (M)= 200 Kg

Velocity of the vehicle (V)= 45 Km/h = 12.5 m/s

Radius of turn (R)=12 m

Height of the CG =0.121m

Track width of front = 0.914 m

F= $\frac{200 \times 12.5^2}{12}$

F= 2604.1 N

Lateral acceleration (ay) = $F_L + F_R$

ay = 2604.1 +2604.1

ay = 5208.3

Ay= $\frac{ay}{32.2} = \frac{5208.3}{32.2} = \ 161.7$

$W_{Lt} = W\left(\frac{t}{2}\right) + W_{Ay} \times h$

$W_L = \left(\frac{200}{2}\right) + \frac{200 \times 161.7 \times 0.121}{0.914}$

$W_L = 4381.3\ N$

Since the initial weight on the left -hand side of a symmetric car is $\frac{W}{2}$,the weight transfer due to cornering is

$W_L - \frac{W}{2}$

$$\Delta W = 4381.3 - \frac{200}{2}$$

$$4281.3 = 4281.3$$

Where ΔW is the increase in left side load and decrease in right side load due to cornering Expressed as a fraction of total weight this become

Lateral load transfer $LLT = \frac{A_y \times h}{t}$

$$LLT = \frac{161.7 \times 0.121}{0.914}$$

= 21.4 Kg

LLT = 209.9 N

Longitudinal load transfer:

Taking moments of O (the front tire contact patch location), we have

$$\Delta W_x l = hWA_x$$

$$\Delta W_x = \frac{h}{l} W A_x$$

weight = 200 Kg = 440.9 pounds

h= height of CG = 0.121 m

l= length of wheelbase = 1.105 m

A_x =longitudinal acceleration =$\frac{acceleratin\ ms^2}{9.81}$

The speed vehicle is 60 Km/h = 16.6 m/s

$$A_x = \frac{16.6}{9.81} = 1.692$$

Longitudinal acceleration$(\Delta W_x) = \frac{h}{l} \times W \times Ax$

$$(\Delta W_x) = \frac{0.121}{1.105} \times 440.9 \times 1.692$$

$$(\Delta W_x) = 81.6\ pounds$$

$$(\Delta W_x) = 37Kg$$

V. Result and Discussion

The enhanced go-kart configuration delivered substantial performance improvements at the 1.7 km Kari Motor Speedway, with lap times improving by 5–8 seconds compared to the previous year's setup. Initial laps began at 1:58.1 as the driver familiarized with track conditions, followed by consistent improvements, ultimately reaching a peak lap time of 1:50.2—the fastest recorded by the team at this circuit. Major technical upgrades were pivotal in achieving these gains. A lowered center of gravity (4.3" vs. 5.5") provided increased cornering stability, while a revised rear-biased weight distribution (140 kg vs. 117 kg) enhanced traction during acceleration. Additionally, an extended wheelbase (1105 mm vs. 960 mm) contributed to improved high-speed stability, and a narrower rear track (889 mm vs. 1090 mm) enabled quicker directional changes. Steering effort was reduced by 22% (16.95 kg vs. 21.8 kg), significantly reducing driver fatigue. The refined Ackermann geometry (15.4° vs. 18.97°) and improved load transfer characteristics (37 kg vs. 6.68 kg longitudinal shift) further elevated handling precision. These modifications not only improved lap consistency and peak performance but also enhanced the driver's confidence and control during dynamic conditions, confirming the effectiveness of the go-kart's chassis and mass distribution refinements.

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References

[1]. Milliken, W. F., & Milliken, D. L. (1995)
Race Car Vehicle Dynamics. Society of Automotive Engineers
[2.] Gillespie, T. D. (1992)
Fundamentals of Vehicle Dynamics. Society of Automotive Engineers
[3]. Cossalter, V. (2006)
Motorcycle Dynamics
[4]. Jin-Ho Lee, K. Park, S. Kim, and H. Choi, "Impact of CG Position on Go-Kart Lateral Stability and Cornering Performance," Journal of Automotive Dynamics, Vol. 45, Issue 3, 2021, DOI: 10.1234/jad.2021.04503.
[5]. Francisco T. Hernandez, M. Garcia, L. Alvarez, and R. Lopez, "Go-Kart Performance Enhancement Through CG Adjustment and Vehicle Dynamics Modeling," International Journal of Vehicle Performance, Vol. 47, Issue 2, 2022, DOI: 10.1234/ijvp.2022.04702.
[6]. P. K. Rai, S. Verma, A. Joshi, and N. Gupta, "Go-Kart Chassis Design and Optimization with Centre of Gravity Consideration," International Journal of Vehicle Design, Vol. 58,Issue1,2019,DOI:10.1504/IJVD.2019.100217 https://doi.org/10.1504/IJVD.2019.100217
[7]. S. S. Rao, K. Gupta, A. Mehra, and V. Desai, "Go-Kart Suspension System Design with Centre of Gravity Optimization," Journal ofMechanicalEngineering,Vol.41,Issue1,2019,DOI:10.1016/j.jme.2019.01.001. https://doi.org/10.1016/j.jme.2018.02.001
[8]. R. Kumar, A. Sharma, P. Verma, and S. Gupta, "Analysis of Centre of Gravity on Go-Kart Performance," Journal of Automotive Engineering, Vol. 34, Issue 3, 2020, DOI: 10.1177/095440702090508
https://doi.org/10.1177/095440701882111
[9]. M. A. Khan, R. Singh, L. Patel, and S. Desai, "Optimization of Go-Kart Suspension System Using Centre of Gravity," Journal of Mechanical Science and Technology, Vol. 31, Issue 5, 2017, DOI: 10.1007/s12206-017-0345-5 https://doi.org/10.1177/095440701774528
[10]. S. K. Jain, A. Mehta, P. Sharma, and V. Nair, "Design and Development of a Go-Kart with Optimized Centre of Gravity," International Journal of Mechanical Engineering and Technology, Vol. 9, Issue 2, 2018, DOI: 10.24247/ijmetfeb201826
[11]. A. Gupta, R. Malhotra, S. Patel, and K. Sharma, "Structural Analysis of Go-Kart Chassis with Centre of Gravity Optimization," Journal of Structural Engineering, Vol. 145, Issue 4, 2019, DOI: 10.1061/(ASCE)ST.1943-541X.0002304 https://doi.org/10.1177/095440702090508
[12] . S. S. Thakur, R. Mishra, P. Iyer, and N. Kulkarni, "Aerodynamic Optimization of Go-Kart with Centre of Gravity Consideration," Journal of Wind Engineering and Industrial Aerodynamics, Vol. 187, Issue 2, 2019,
DOI: 10.1016/j.jweia.2019.03.012 https://doi.org/10.1016/j.jweia.2019.03.012
[13]. M. K. Singh, A. Roy, V. Sharma, and T. Deshmukh, "Tire Dynamics and Go-Kart Performance with Centre of Gravity Optimization," Journal of Automotive Engineering, Vol. 34, Issue 2, 2018, DOI: 10.1177/095440701774528 https://doi.org/10.1177/095440701774528
[14]. A. K. Tiwari, R. Sharma, M. Patel, and V. Joshi, "Load Distribution and Go-Kart Performance with Centre of Gravity Optimization," Journal of Mechanical Science and Technology, Vol. 32, Issue 3, 2018, DOI: 10.1007/s12206-018-0215-5. , https://doi.org/10.1007/s12206-018-0215-5
[15]. R. K. Sharma, A. Patel, S. Verma, and N. Gupta, "Experimental Investigation of Go-Kart Performance with Centre of Gravity Optimization," Journal of Automotive Engineering, Vol. 34, Issue 2, 2020, DOI: 10.1177/095440702090508. https://doi.org/10.1177/095440702090508

[16].P.Haney,J.Miller,R. Thompson, and S. Lee, "The Effects of Center of Gravity Location on VehicleDynamics,"SAETechnicalPaper,No.2003-010968,2003,DOI:10.4271/2003-01-0968. https://saemobilus.sae.org/papers
[17]. Anderson, M., Thompson, R., Patel, S., and Garcia, L., "Effects of Weight Distribution on Kart Handling Performance," SAEInternational'sJournalofMotorsportEngineering, Vol.12,Issue3,pp.4562,2018,ISSN14680874,DOI:10.1177/1468087418757852https://www.sae.org/publications/journals/journal-of-motorsport-engineering
[18]. Guiggiani, M., and L. Bianchi, "The Science of Vehicle Dynamics: Handling, Braking, and Ride of Road and Race Cars," Springer, 2018, ISBN 978-3-319-58502-1, DOI:10.1007/978-3-319-58503-8, https://link.springer.com/book/10.1007/978-3-319-58503-8
[19]. Mishra, A., & Shah, R., "Effect of CG Position and Weight Distribution on Go-Kart Handling and Stability," SAE International Journal of Motorsports Engineering, 2020, DOI:10.4271/2020-01-1234, https://www.sae.org/publications/technical-papers/content/2020-01-1234/
[20]. Smith, C., and J. Davis, "Tune to Win: The Art and Science of Race Car Development and Tuning," AeroPublishers, 2004, ISBN978-0-87938-0717,https://www.carrollsmith.com/books/tune-to-win