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Parametric Analysis and Optimization of the Chassis of a Go Kart Competition Vehicle by Means of Computational Software Based on Mathematical Algorithms

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Abstract

The article presents a study on the optimization of the structure and internal components of a Go Kart vehicle, using design approaches based on mathematical algorithms. The importance of vehicles, particularly chassis, providing safety to the driver at high speeds, maintaining their stability and resistance to the forces that act in motion is highlighted. The most common materials used to manufacture the structures are steel. The study included simulation tests to assess the resistance capacity of the materials used in the chassis, to select the appropriate materials to provide optimal resistance and stability to the vehicle. The results showed that AISI 4130 steel is the best option for the chassis structure due to its mechanical properties. A maximum permitted weight of 150 kg was also established to achieve optimal structural resistance performance of the vehicle, which allows a speed of 110 km/h and an acceleration of 40 m/s^2. In addition, an analysis of the results of the deformation and torsion tests on the transmission shaft is presented, as well as the relationship between the torque, power, speed, and acceleration of the vehicle under different load conditions.

Keywords

Go Kart, Chassis, Mechanical properties, AISI 4130, Transmission system, Optimization

Introduction

In the world of karting, it is essential that racing vehicles [1], especially the chassis, provide the driver with the necessary safety at high speeds [2], while maintaining their stability and resistance to the forces that act in motion [3]. That is why the use of design approaches based on mathematical algorithms for the optimization of the structure and internal components is of vital importance. The main elements of the vehicle are the chassis, wheels, and engine, all interconnected, allowing the use of a holistic design approach for the entire vehicle to generate optimal performance [4]. The most used materials for the manufacture of Go Kart structures are cylindrical section magnetized tubular steel and structural steel alloys [5], such as AISI 4130, ASTM A500 and ASTM A1011. Choosing the right engine is also crucial, as it provides the power needed to meet vehicle performance parameters, such as speed, acceleration, and maximum grade, through the drive chain [6].

A Go Kart today focuses on critical parameters such as emissions, safety, and intelligent systems. Although conventional materials such as AISI 4130, ASTM A500, ASTM A1011, among others, meet many demands; efficient integration with smart systems requires advanced materials. Nanomaterials, operating at a nanoscale, offer improved physical, chemical and electrical properties, allowing the manufacture of lighter, safer, and more economical Go Kart components

[7]. Applications stand out in tribology, rheology, electricity, and optics, optimizing the drive train, tires, vision, and surface coating [8]. These aspects may be applied in the future in small-scale competition cars; as it will result in lighter vehicles, reduced greenhouse gas emissions and carbon footprint. Nanotechnology crucially contributes to the necessary developments and production of innovative materials and processes in the automotive and aerospace sectors. For example, modern tires achieve high mileage, durability and grip through nanoscale soot and silica particles [9]. Therefore, automotive competition industries will have to invest heavily in technological research and development to meet the requirements of a Go Kart and improve its performance [10].

Experimentation

Mechanical model

The construction of the Go Kart vehicle is closely related to the mechanics since the interconnection of its components is essential for the performance and safety of the vehicle. From the engine to the tires, every part is important and must function properly for the vehicle to perform at its best. In this study, the Puma engine from IAME Motor - Grill was used, which is characterized by being an air-cooled two-stroke engine with a displacement of 85 cc [11], providing a maximum power of 14 CV at 13500 rpm and a maximum torque of 10.8 Nm at 12000 rpm.

The chain kinematics model assumes that the drive gear is on the same shaft as the motor [12], so it will rotate at the same rpm; furthermore, the power of the drive gear will be equal to the power produced by the motor, using equation 1. As a result, the mathematical model of the torque exerted on the tire axle is given by equation 2, where z_m is the number of teeth of the driving gear, z_c the number of teeth of the driven gear, and K_n the design coefficients of the kinematic chain.

$$P_{conducted} = K_1.K_2.K_3.K_4.K_5.P_{motor} \tag{1}$$

$$T_{conducted} = \frac{z_c.K_1.K_2.K_3.K_4.K_5.T_{motor}}{z_m} \tag{2}$$

For the analysis of the angle of rotation ϕ, the equations corresponding to the maximum shear strain and Hooke's law are related by means of equation 3 and replacing equation 2 you get equation 4 that allows obtaining the angle of rotation, where J is the polar moment of inertia and G is a constant of the cross section.

$$\phi = \frac{T.L}{J.G} \tag{3}$$

$$\phi = \frac{z_c.K_1.K_2.K_3.K_4.K_5.T_{motor}}{z_m.J.G} \tag{4}$$

For the analysis of the bending deflection of the Go Kart axle, a mathematical model is considered that considers the longitudinal and frontal transfer of the vehicle, which allows determining the distribution of forces acting perpendicularly, as shown in figure 1. The shaft structure is modeled as a single supported beam and the flexural deflection is expressed by equation 5, using the singularity method. The variables used in this model are detailed in table 1. This approach allows the evaluation of the resistance and performance of the Go Kart axle, which is essential to guarantee safety and success in competition.

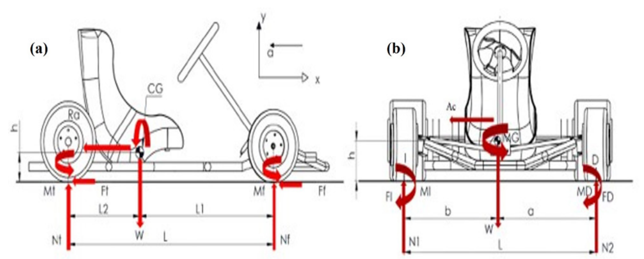

Figure 1: Diagram of **(a)** Lateral and **(b)** Frontal transfer of forces in the competition vehicle.

$$EIY = \frac{Wb}{6L}x^3 - \frac{W}{6}(x-a)^3 + \frac{Wb^3 - WbL^2}{6L}x \tag{5}$$

Once the produced torque has been calculated, it is necessary to determine the driving force that the tire transmits to the ground; therefore, the parameter of maximum speed in resistance conditions is reached when the acceleration capacity of the vehicle is zero. This is how we have equation 6 where r_t is the total forward resistance, μ_r is the rolling resistance, C_x is the drag coefficient, S_{eff} is the effective airfoil, P_o is the maximum power and V is the speed.

$$\frac{75.r_t.P_o}{V} = \mu_r.W + \frac{C_x.S_{eff}}{16}.V^2 \tag{6}$$

Consequently, the calculation of the maximum plane acceleration γ is expressed by equation 7, where $(1 + \varepsilon_i)$ is a correction factor for the vehicle mass m due to rotational inertia. And from the equation 6, the maximum slope n with respect to the horizontal is obtained, as observed in equation 8. Thus, from equation 8, the maximum slope can be obtained in a static state, so both the initial speed of the Go Kart and the aerodynamic resistance (proportional to speed) will be zero.

$$\gamma = \frac{75.r_t.P_o - \left(\mu_r.W + \frac{C_x.S_{eff}}{16}.V^2\right)V}{V.m.(1+\varepsilon_i)}.9.81 \tag{7}$$

$$n = \frac{T_{motor}.r_t.\mu_r}{W.R} - \mu_r - \frac{C_x.S_{eff}}{16W}.V^2 \tag{8}$$

Simulation testing

To evaluate the resistance capacity of the materials used in the Go Kart chassis, simulation tests were carried out to determine the maximum turning angle with respect to the applied torque and the characteristic rigidity modulus of each material. The results are shown in table 2, where the ASTM A500 steel has the highest maximum angle of twist and the highest modulus of rigidity, followed by the ASTM A1011 structural steel and finally the AISI 4130 steel. These data are important for the design and optimization of the chassis, since they allow selecting the appropriate materials that provide maximum re-

Table 1: Variables used in the model EIY.

Lateral	Front
a distance between the center of gravity and the posterior shaft (m)	*a* distance from the center of gravity to the left tire (m)
b distance between the center of gravity and the front shaft (m)	*b* distance from the center of gravity to the right tire (m)
W force applied at the center of gravity (N)	*W* force applied at the center of gravity (N)
L distance between axles (m)	*L* girder width (m)

Table 2: Maximum turning angle (°) for each material under study.

Torque [Nm]	AISI 4130	ASTM A500	ASTM A1011
7	0.1731	0.1817	0.1773
8	0.1978	0.2077	0.2026
9	0.2225	0.2336	0.2279
10	0.2472	0.2596	0.2533

sistance and stability to the vehicle. In addition, the equation 5 determines the maximum deflection produced at different applied weights for each material, as detailed in table 3.

Table 4 shows the values assigned to the variables that were considered during the design of the Go Kart. These variables include total forward resistance, rolling resistance, drag coefficient, among others. Assigning precise values to these variables is essential to achieve optimal design and ensure maximum vehicle performance.

Results and Discussion

The stiffness modulus, inversely proportional to the angle of torsional rotation of the transmission shaft, causes a greater torsional deformation at a greater length of the shaft. The ASTM A500 material, which has a modulus of rigidity of 200 MPa, exhibits higher torsional strain. Increasing the applied torque increases the strain produced as seen in figure 2a. In the deformation resulting from the application of weight, the ASTM A500 material also presents greater deformation according to figure 2b, considering the minimum weight of the driver without equipment (75 kg) and the maximum weight when incorporating the weight of the equipment and vehicle components (150 kg) at a safety factor equal to 1.5.

As seen in figure 3 the AISI 4130 drive shaft has its maximum bending deformation in the middle length with a weight of 135 kg, while the maximum torsional deformation occurs near the transmission system. The stiffness study at a load of 150 kgf produces a torque of 1155.24 Nm and a maximum bending strain of 0.00447 m as seen in figure 4 and a torsional strain of 0.33 degrees, comparable to the values obtained by the mathematical model indicated in table 3.

Figure 4a shows three critical points: the minimum torque (point A), the equilibrium torque (point B), and the maximum torque (point C). Peak torque is at an rpm below full power, and both balance torque and horsepower are at 9600 rpm. As you go up to 12000 rpm, torque and power gradually increase as well. However, the maximum applied load is related to the increase in torque and can affect the speed of the vehicle. Op-

Table 3: Maximum displacement for each material under study.

Weight [Kgf]	AISI 4130	ASTM A500	ASTM A1011
70	-0.0023	-0.0024	-0.0023
80	-0.0026	-0.0027	-0.0026
90	-0.0029	-0.0030	-0.0030
100	-0.0032	-0.0034	-0.0033
110	-0.0035	-0.0037	-0.0036
120	-0.0039	-0.0041	-0.0040
130	-0.0042	-0.0044	-0.0043
140	-0.0045	-0.0047	-0.0046

Table 4: Data sheet of variables considered in the design of the competition vehicle.

Parameter	Value
r_t	0.9
P_o	14 (CV)
S_{eff}	0.945 (m^2)
V_{max}	13500 (rpm)
μ_r	0.015
C_x	0.85
R	0.13 (m)
$(1 + \varepsilon_i)$	1.04

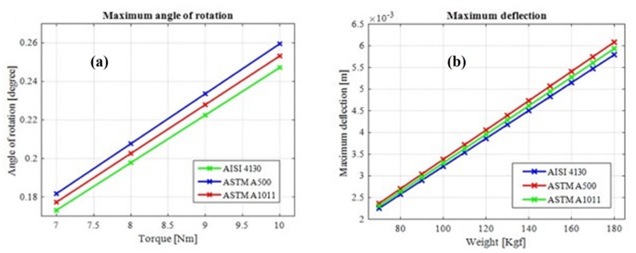

Figure 2: The maximum **(a)** Angle of rotation (torsional strain) and **(b)** Deflection (bending strain).

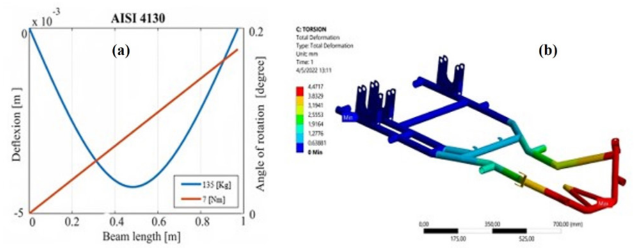

Figure 3: The **(a)** Deformation (bending and torsion) - AISI 4130 and **(b)** Total deformation.

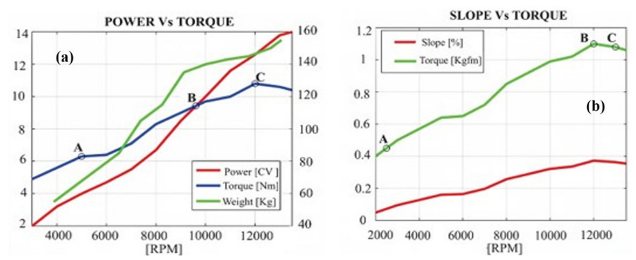

Figure 4: Curve of **(a)** Power - torque and **(b)** Slope - torque.

timum performance is obtained at 12000 rpm, 10.4 Nm and 12.2 CV. At a minimum load value, power and torque increase, producing a ride with gradual gain in speed. The optimal performance in this case is obtained at 5000 rpm with a minimum load of 75 kg, 4 CV and 6.2 Nm. Figure 4b shows that at steady state at 2500 rpm, airspeed and drag are zero with a minimum slope of 9.8% at a torque of 0.5 Kgf m; the full torque running state is obtained at 12000 rpm with a maximum slope of 39% at 1.1 Kgf m, and the full speed running state is obtained at 13500 rpm with a 36% slope at 1.06 Kgf m.

As seen in figure 5a, the maximum speed of the vehicle depends on the weight and the gear ratio, being 26.1 m/s for a balance weight and a gear ratio of 8.7. In figure 5b two inflection points are observed; point (A) at 4000 rpm marks the beginning of an inversely proportional behavior between speed and acceleration, gradually decreasing to point (B). The higher the rpm, the higher the speed (110 km/h) and acceleration (40 m/s^2), resulting in a maximum rpm value and a run time of 0.24 sec.

Conclusion

Based on the results of this study, it can be stated that AISI 4130 steel is the best option for the chassis structure due to its mechanical properties, providing greater resistance to bending and torsion loads. Compared to ASTM A500 and ASTM A1011 steels, the use of AISI 4130 significantly reduces the deformations obtained by simulation in the chassis structure, being 0.0002 m and 0.0001 m lower, respectively.

To achieve optimal structural strength performance of the vehicle, a maximum allowable weight of 150 kg has been established, which allows for a speed of 110 km/h, an acceleration of 40 m/s^2, and an execution time of 0.24 sec. It is important to consider that as the applied weight increases, permanent deformations will occur in the structure, resulting in a decrease in vehicle speed and acceleration. Likewise, modifying the speed through the transmission ratio will result in a decrease in the acceleration parameter, with the optimal value established between 1 and 10.

According to the developed mathematical model, it is observed that the torque, power, and speed increase up to 13500 rpm, with significant increases in the speed and acceleration parameters at values of 130 m/s and 68.4224 m/s^2, respectively, and a decrease in the time to 0.24 sec. However, beyond this point, the torque progressively decreases until reaching a value of 10.4 Nm. Furthermore, the achieved slope depends on the torque increase, with the minimum slope being equal to 9.8% at a torque of 0.5 Kgf m in static conditions at 2500 rpm, and the maximum slope being 36% at 1.06 Kgf m during full speed operation at 13500 rpm.

Optimizing the studied model requires relating the effective aerodynamic surface with the aerodynamic coefficient to improve the displacement force while maintaining the established design characteristics for the type of tire to be used. Additionally, it is important to consider the rolling resistance according to the type of terrain on which the competition will take place, and the implementation of an electronic control system to ensure optimal performance.

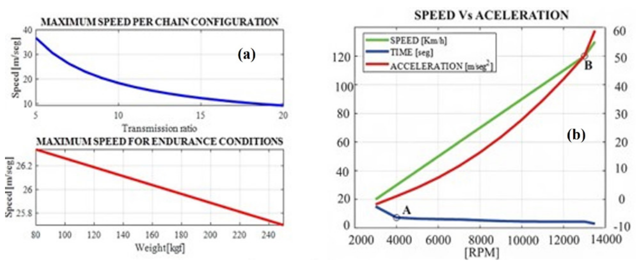

Figure 5: Curve of **(a)** Maximum speed and **(b)** Speed - acceleration.

Acknowledgments

None.

Conflict of Interest

None.

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