DESIGN AND ANALYSIS OF A GO-KART

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ABSTRACT
This report documents the process and methodology to produce a low cost go-kart which is comfortable, vulnerable, durable and complete in all aspects by modeling it with CAD software. The feasibility of the go-kart design was examined through FMEA, Cost report. The team focuses on a technically sound vehicle which is backed by a profound design and good manufacturing practices. The report explains approach, reasons, selecting criteria and expected working of the vehicle parameters. The procedural way of explanation is used for different parts of the vehicle, which starts from approach with the help of known facts, then the design and calculation procedure has been explained. The best way known had been used to go on to the final result of all parameters.

1. INTRODUCTION
The go-kart will be built from the ground up to maximize the efficient use of space, and to ensure that the needs of the client are met. We approached our design by considering all possible alternatives for a system & modeling them in CAD software like CREO Parametric 2.0 and subjected to analysis using ANSYS 15.0 FEA software. Based on analysis result, the model was modified and retested and a final design was frozen.

The design process of the vehicle is iterative and is based on various engineering and reverse engineering processes depending upon the availability, cost and other such factors. So the design process focuses on following objectives:

Safety, Serviceability, Strength, ruggedness, Standardization, Cost, Driving feel and ergonomics, Aesthetics. The design objectives set out to be achieved were three simple goals applied to every component of the car: durable, light-weight, and high performance, to optimizing the design by avoiding over designing, which would also help in reducing the cost.

With this we had a view of our kart. This started our goal and we set up some parameters for our work, distributed ourselves in groups for the technical design of our vehicle.

Sub-Departments for Design:-

- Chassis Department.
- Steering Department
- Brakes and Tyres Department
- Transmissions Department

2. CHASSIS DESIGN APPROACH
The chassis has been designed by taking factors like: dimensional limits (width, height, length, and weight), operational restrictions, regulatory issues, contractual requirements, financial constraints and human ergonomics as a priority.

A basic chassis frame of circular pipes of 1.25 inch diameter and 2mm thickness was designed and selected by taking the points of strength, availability and cost into consideration.

3. MATERIAL AVAILABILITY
Tubing is available in standard fractional sizes to the 1/8th of an inch: 1, 1.12, 1.25 and 1.5. The wall thickness is limited to the common Birmingham Tubing Gauges. In this case these are: 1.5, 1.8, 2, 2.5 and 3 mm. The most commonly available materials are:

<table>
<thead>
<tr>
<th>Materials</th>
<th>Yield strength (MPa)</th>
<th>Percentage elongation at break</th>
<th>Cost per m in(₹)</th>
</tr>
</thead>
<tbody>
<tr>
<td>AISI 1026</td>
<td>260-440</td>
<td>17-27%</td>
<td>345</td>
</tr>
<tr>
<td>AISI 4130</td>
<td>435-979</td>
<td>18-26%</td>
<td>735</td>
</tr>
<tr>
<td>AISI 1020</td>
<td>230-370</td>
<td>18-28%</td>
<td>315</td>
</tr>
<tr>
<td>AISI 1018</td>
<td>270-400</td>
<td>18-29%</td>
<td>300</td>
</tr>
</tbody>
</table>

It is observed that material which has high machinability and inexpensive is AISI 1018, hence was a good choice but strength to weight ratio is greater for 4130.

AISI 1020 was rejected because of its high cost. AISI 4130 was rejected because of its high carbon content and lack of machinability, 4130 have the superior harden ability that other iron alloys like 4130 and 4140 possess. But 4130 is a popular steel in race car industry but is not easily available in India. Therefore, the material that the team chose to use is AISI 1018.

The benefit of using the AISI 1018 is that it can be easily wielded than the 4130 chromyl. The AISI 1018 has the same Modulus of Elasticity (E) and density as the 4130, so using it does not affect the weight or stiffness in member with same geometry.

AISI 1018 has excellent weldability and produces a uniform and harder case and it is considered as best steel for carburizing parts. The 1018 carbon steel offers a good balance of toughness, strength and ductility. Considering the above factors we choose AISI 1018 for our chassis material.

Chemical composition of AISI 1018

<table>
<thead>
<tr>
<th>ELEMENT</th>
<th>CONTENT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon (C)</td>
<td>0.14-0.20%</td>
</tr>
<tr>
<td>Sulphur,(S)</td>
<td>&lt;=0.050%</td>
</tr>
<tr>
<td>Iron,(Fe)</td>
<td>98.81-99.26%</td>
</tr>
</tbody>
</table>
Manganese, (Mn) 0.60-0.90%
Phosphorous, (P) ≤0.040%

Physical properties of AISI 1018

<table>
<thead>
<tr>
<th>PROPERTIES</th>
<th>VALUE (Metric)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>7.87 g/cc</td>
</tr>
<tr>
<td>Yield tensile strength</td>
<td>370 MPa</td>
</tr>
<tr>
<td>Elongation at break (in 50 mm)</td>
<td>15%</td>
</tr>
<tr>
<td>Poisons ratio</td>
<td>0.29</td>
</tr>
<tr>
<td>Modulus of elasticity</td>
<td>200 GPa</td>
</tr>
</tbody>
</table>

4. VIEWS OF THE GO KART
The side view, the top and the isometric views have been displayed as under:

5. FRAME ANALYSIS
For the purpose of analysis, we have conducted Certain Tests on the Chassis, which are:-

5.1 Front Impact Test
The Front Impact Analysis has been carried out on the Ansys 15.0 while constructing a perfect space frame tubular chassis on Creo 2.0 Surface module and then it was imported to Ansys 14.5.

Gusset plates have been applied on the regions where the stress concentration was more.

A force of 7500 N was applied to the front ends constraining the body panel rods and we had seen such results as shown above and assuming the deceleration.
On applying a force of 7500N, the maximum deformation of 1.277mm for observed in the chasses. This deformation is within the acceptable limits.

\[ FOS = \frac{\text{Yield Strength of AISI 1018}}{\text{Von Mises Stress}}. \]

So, \( FOS = \frac{370}{118.75} \approx 3.11 \)

5.2 Side Impact Test

The Side Impact Analysis has been carried out on the Ansys 15.0 while constructing a perfect space frame tubular chassis on Creo 2.0 Surface module and then it was imported to Ansys 15.0 with a Force with respect to the 2G criteria.

\[ FOS = \frac{370}{112.8} \approx 3.28 \]

A force of 3650N has been applied and the observed deformation is 0.91mm and is within the acceptable limits.

5.3 Rear Impact Test

A force of 5450 N was applied to the rear ends by totally constraining the degree of freedom of the suspension points and we had seen such results as shown And assuming the deceleration of 3G

\[ FOS = \frac{370}{101.5} \approx 3.64 \]

A force of 5450N has been applied and the observed deformation is 3.9mm and is within the acceptable limit.

Summarizing the above discussions:

<table>
<thead>
<tr>
<th>Impact Type</th>
<th>FOS</th>
<th>Deformation</th>
</tr>
</thead>
<tbody>
<tr>
<td>FRONT IMPACT</td>
<td>3.11</td>
<td>1.277mm</td>
</tr>
<tr>
<td>SIDE IMPACT</td>
<td>3.28</td>
<td>0.91mm</td>
</tr>
<tr>
<td>REAR IMPACT</td>
<td>3.64</td>
<td>3.9mm</td>
</tr>
</tbody>
</table>
6. BODY AND COMPOSITES

6.1 OBJECTIVE
The purpose of the body is to prevent debris from entering the vehicle, with the intent of protecting the driver and the vehicle’s components. The seat was designed to support the driver comfortably and safely while they are operating the vehicle.

6.2 SEAT
The seat in this kart is also designed to be very light; it is very simple made of plastic material and is attached to the chassis by four points only.

The back rest angle of the seat is at 13 degrees which is the good position of the drivers body rest according to the ergonomics point of view and is kept almost parallel to the fire wall. The seat implemented in our go kart provides a good combination of weight reduction and ergonomics.

In an investigation of the 95th percentile man, it was found that the average male height in India is 165.3cm. Therefore all the above calculations are done according to the investigation.

6.3 VISION CONES

The vision cone shows the region accessible to the driver while sitting on the kart.

6.4 POWER TRAIN

Efficiency of CVT $\eta_T = 88\%$

Overdrive Ratio $r_o = 0.9$

Under drive Ratio $r_u = 2.7$

Sprocket teeth on CVT $t_1 = 12$

Radius of drive wheel $r = 0.1397m$

Efficiency of Chain drive $\eta_c = 98\%$

Engine Power $P = 3.5hp = 2611watts$

Maximum Engine Torque $T = 7.45 N\cdot m$

Maximum Engine rpm $N_{max} = 3300rpm$

Co-efficient of Rolling Resistance $C_{rr} = 0.012$

Co-efficient of friction for slicks $\mu = 0.9$

Gross Vehicle Weight $GVW = 185kg$

Density of Air $\rho = 1.226 \text{ kg/m}^3$
Co-efficient of drag $C_d = 0.35$

Frontal Area $A = 1 \text{ m}^2$

Let, maximum velocity of vehicle is $v$

**Maximum Speed**

Assuming a grade of $1^\circ$ on normal conditions, driving force

$$F = (GVW \times g \times C_{r}) + (0.5 \times \rho \times A \times v^2 \times C_d)$$

$$+ GVW \times g \times \sin \theta$$

$$= (185 \times 9.81 \times 0.012) + (0.5 \times 1.226 \times 1 \times v^2 \times 0.35)$$

$$+ (185 \times 9.81 \times \sin 1^\circ)$$

And, Power $P = \eta_T \times \eta_c \times F = v$  

......eq.1

Therefore, from eq.1 and eq.2:

$$32611.88 \times 0.98 \times 21.78 \times 0.214 \times 31.67 = v^2 + v$$

$$32611.88 \times 0.98 \times 21.78 \times 0.214 \times 31.67 = v^2 + v$$

Or, $v = 18.15 \text{ mph} = 65.34 \text{ kmph}$

......eq.3

**Selection of Axle Ratio and Chain Drive**

We have, $v = r \times \omega = r \times \frac{2\pi}{60} \times \frac{N_{max}}{r_o \times r_{axle}}$

Or, $18.15 = 0.1397 \times \frac{2\pi}{60} \times \frac{3300}{0.9 \times r_{axle}}$

Therefore, $r_{axle} = 2.95 \approx 3$

......eq.4

Sprocket teeth on live axle $t_2 = t_1 \times r_{axle} = 12 \times 3 = 36\text{ teeth}$

......eq.5

Sprocket available with CVT is #35 types

Therefore, Pitch $p = \frac{3}{8} \times 25.4\text{ mm} = 9.525\text{ mm}$

Pitch Diameter of sprocket on live axle $d = \frac{p}{\sin(180/36)} = 9.525 \frac{180}{\sin(180/36)} = 109.28\text{ mm}$

......eq.6

Suitable Chain for 9.525mm pitch is ISO 6-B

**Max. Tractive Torque and No-Slip Condition**

Vehicle weight distribution:

Rear: 58% and Front: 42%

Total normal reaction acing on rear drive wheels

$$R = 0.58 \times GVW \times g = 0.58 \times 185 \times 9.81 = 1052.6\text{ N}$$

Therefore, Maximum Tractive Torque

$$t_{max} = \mu \times R \times r = 0.9 \times 1052.6 \times 0.1397 = 132.34\text{ N} - m$$

......eq.7

Maximum Wheel Torque

$$t_{max} = T \times r_u \times r_{axle} = 7.45 \times 2.7 \times 3 = 60.34\text{ N} - m$$

......eq.8

From eq.7 and eq.8: $t_{max} > t_{max}$

Therefore, No-Slipping Condition is satisfied.

**Maximum Acceleration**

Engagement of centrifugal clutch of CVT takes place at 2200 rpm.

From power curve of engine, the power (P) at 2200 rpm is 2.25 hp

$$P = \frac{2\pi}{60} \times \frac{1678.5}{2200} = 7.28\text{ N} - m$$

Therefore, wheel torque

$$t_{wheel} = T \times r_u \times r_{axle} = 7.28 \times 2.7 \times 3 = 58.96\text{ N} - m$$

And, driving force available

$$F = \frac{t_{wheel}}{r} = \frac{58.96}{0.1397} = 422\text{ N}$$

......eq.9
Driving force
\[ F = \text{Rolling Resistance} + \text{Air Resistance} + \text{Acc. force} \]
Or,
\[ F = (GVW \times g \times C_n) + \left( \frac{D}{2} \times A \times V^2 \times C_d \right) + (m \times a) \]
Where, \( a \) = acceleration
Therefore, substituting eq.9
\[ 422 = (185 \times 9.81 \times 0.012) + \left( \frac{1.226}{2} \times 1 \times 18.15^2 \times 0.35 \right) \]
\[ + (185 \times a) \]
Or, acceleration \( a = 1.78 \text{m/s}^2 \)

**Grade Ability**

Let, \( \Theta \) be the grade angle. Then Grade resistance
\[ R_{\text{grade}} = GVW \times g \times \sin \Theta \] .......eq.10

Driving force \( F \)

= Rolling resistance + Air resistance + Grade resistance

.........eq.11

Using eq.9, eq.10 and eq.11
\[ 422 = (185 \times 9.81 \times 0.012) + \left( \frac{1.226}{2} \times 1 \times 18.15^2 \times 0.35 \right) \]
\[ + (185 \times 9.81 \times \sin \Theta) \]

Therefore, \( \Theta = 10.46^\circ \)

6.5 STEERING

The control of an automobile is done by means of a steering system which provides directional changes to the moving automobile.

**Ackermann principle of steering**

To solve the problem of wheels on the inside and outside of a turn needing to trace out circles of different radius, Ackermann principle of steering is used.

**Assumptions**

- 100% Ackermann steering geometry.
- Maximum road bank angle is 20°.
- Optimum kingpin inclination angle range is 4° to 8°.
- Front to rear weight ratio is 42:58.
- Taking acceleration due to gravity as 10m/s^2

7. **CALCULATION**

Assuming the total weight of vehicle to be 185 KgF, therefore weight on front tyres is 77.7kgf.

- Vertical force (on one tyre) \( V = 388.5 \text{ N} \).
- Lateral force (on one tyre) \( L = V \times \sin(\Theta) = 132.87 \text{ N} \)

Where, \( \bar{\Theta} \) – maximum road bank angle

- Total Aligning Torque (\( M_z \)) is
\[ M_z = M_a + M_b \cos \sqrt{\( \lambda^2 + \upsilon^2 \)} \]

Where,
\( M_a \) – aligning torque on left tyre,
\( M_b \) – aligning torque on right tyre,
\( \lambda \) - Kingpin angle,
\( \upsilon \) - Caster angle.
- Mechanical trail (\( m \)) = 48.75 mm
(Mechanical trail is calculated using geometry with the help of castor angle)

- $M_a = L \times m = 6477.4 \text{ Nmm}$

Aligning torque on each tyre will be same so “$M_a=M_b$” and assuming the total aligning torque($M_z$) to be 12000Nmm the castor angle ($\psi$) and kingpin angle ($\lambda$) is thus calculated by hit and trial method in the equation above.

- Kingpin angle ($\lambda$) = 6.9°
- Castor angle ($\psi$) = 21°

Analysis on the stub axle is done by applying a force of 410N in upward direction and FOS calculated is $4.15$.

**Ackermann angle ($\alpha$)**

$$\alpha = \tan^{-1}\left(\frac{0.5 \times \text{Track Width}}{\text{wheel base}}\right) = 22.17^\circ.$$ 

**Inner angle ($a$)**

$$\tan a = \left[\frac{H}{R - \left(\frac{W}{2}\right)}\right]$$

$a = 35.37^\circ$

**Outer angle ($b$)**

$$\tan b = \left[\frac{H}{R + \left(\frac{W}{2}\right)}\right]$$

$b = 23.74^\circ$

**Turning radius ($R$)**

$$R = \frac{W}{2}H / \sin \gamma$$

Where, $\gamma$ (average steer angle) = $(a + b/2)$

- $W$ (track width) = 1018mm
- $H$ (wheel base) = 1250mm

$$R = 3043.16 \text{ mm or } 3.04 \text{ m}$$

The centre to centre distances between the sleeve and steering arms are 89mm & 127mm respectively.

The angular distance covered by the steering arm is 78.23mm, corresponding to the inner steer angle of 35.37°.

The same angular distance must be travelled by the sleeve and corresponding to the angular distance the sleeve should rotate by 50.52°, which is same the steering wheel has to be turned to rotate the inner wheel by 35.37°.

**Steering ratio ($r$)**

$$r = \frac{\text{angle turned by steering wheel}}{\text{angle turned by wheel}}$$

$$r = 50.52^\circ / 35.37^\circ = 1.42$$

**Max Steering effort ($E$)**

$$E = \frac{\text{vertical load of tyres}}{\text{steering ratio}}$$

$$E = 98.35 \text{ N}$$

**RESULTS**

<table>
<thead>
<tr>
<th>Track width</th>
<th>1018mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wheel base</td>
<td>1250mm</td>
</tr>
<tr>
<td>Ackermann angle</td>
<td>22.17°</td>
</tr>
<tr>
<td>Inner steer angle</td>
<td>35.37</td>
</tr>
<tr>
<td>Outer steer angle</td>
<td>23.74</td>
</tr>
<tr>
<td>Kingpin angle</td>
<td>6.9°</td>
</tr>
<tr>
<td>Caster angle</td>
<td>21°</td>
</tr>
<tr>
<td>Camber angle</td>
<td>1°</td>
</tr>
<tr>
<td>Steering ratio</td>
<td>1.42:1</td>
</tr>
<tr>
<td>Steering effort</td>
<td>98.35N</td>
</tr>
</tbody>
</table>
### 7.1 BRAKING CALCULATIONS

The calculations for the selected components of the brake system were done by considering a driver input force of 70lbs (i.e., and mechanical leverage as 4:1. For better stability of the vehicle during braking, the main aim was to have minimum weight transfer along with an optimum stopping distance. Iterations were performed accordingly.

<table>
<thead>
<tr>
<th>Inputs</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gross weight (m)</td>
<td>185 kg</td>
</tr>
<tr>
<td>Deceleration in g’s (a)</td>
<td>0.9 g</td>
</tr>
<tr>
<td>Height of C.G (h)</td>
<td>250 mm</td>
</tr>
<tr>
<td>Wheel base (b)</td>
<td>1250 mm</td>
</tr>
<tr>
<td>Initial velocity (u)</td>
<td>12.5 m/s</td>
</tr>
<tr>
<td>Final velocity (v)</td>
<td>0 m/s</td>
</tr>
</tbody>
</table>

- **Stopping distance (s)** = \(\frac{(v^2-u^2)}{2 \cdot a \cdot g} = \frac{(12.52-0^2)}{2 \cdot 0.9 \cdot 9.81} = 8.8 \text{ m} \)

- **Stopping time (t)** = \(\frac{(v-u)}{a} = \frac{12.5}{0.9 \cdot 9.81} = 1.4 \text{ sec} \)

#### Static weight distribution

- Weight of car at front axle=77.7kg
- Weight of car at rear axle=107.3kg

Weight transfer to front (Wt.) = coefficient of friction * mass of vehicle * height of center of gravity /wheelbase

\[ = 0.9 \times 185 \times \frac{250}{1250} = 33.3 \text{ kg} \]

#### Dynamic weight distribution

- Weight of car at front axle=111kg
- Weight of car at rear axle=74kg

Hence, the above calculated values satisfy the condition of having mass transfer less than half of the weight of a vehicle.

#### Master cylinder

- Diameter=19.05mm
- Area of master cylinder=285.87 mm²

#### Caliper

- Diameter=27mm
- Area of calipers=572.55 mm²

#### Tire

- Diameter=11”=279.4mm

**Rotor:** Outer diameter=198mm

**Inner diameter=51mm**

**Required Braking Force**

\[ = \text{mass } \times \text{deceleration} = 185 \times 0.9 \times 9.81 = 1633.365 \text{ N} \]

**Locking force**

\[ = \text{wt. on rear axle } \times \text{coefficient of friction} \times g = 74 \times 0.9 \times 9.81 = 653.346 \text{ N} \]

**Required Torque (T)**

\[ = \text{locking force } \times \text{tire radius} = 653.346 \times 5.5 \times 0.0254 = 91.27 \text{ N} \cdot \text{m} \]

**Required Clamping Force**

\[ = \frac{T}{(\text{no. of friction surface } \times \text{coefficient of friction } \times \text{disc radius})} = \frac{91.27}{(2 \times 0.9 \times 0.1)} = 507 \text{ N} \]

**Generated clamping force and torque**

- Brake pedal force= 311.5*4:1=1246N
- Pressure generated at master cylinder, \(P_{mc}\)
  \[=\text{brake pedal force } \div \text{area of master cylinder} = 4.35 \text{ N/mm}^2 \]

Since \(P_{mc}=P_{cal}\)

Force generated at the caliper = \(P_{cal} \times \text{area of caliper} = 2495.59 \text{ N} \)

**Clamping force**

\[2 \times 2495.5 = 4991.06 \text{ N} \]

**Force of friction**

- Clamping force * coefficient of friction between the pads=4991.06*0.45
  \[=2245.977 \text{ N} \]

**Torque produced**

\[= \text{force of friction } \times \text{effective radius of rotor} = 2245.977 \times 47 \times 10^{-3} = 105.48 \text{ Nm} \]

#### 8. INNOVATION

The 3 way adjustable steering wheel designed keeping in mind the Ergonomics of the vehicle.

Adjustable steering is not being used in the go karts in general, so keeping that into the notice it would be an innovation using the ADJUSTABLE STEERING in the go karts.
9. MECHANISM

The steering wheel is connected to the steering column and further connected to the column tube which is free to rotate about a pivot point and fixed at a certain angle by using a spring loaded lever mechanism. The lower steering column is fixed to sleeve and supported on a bearing which is free to rotate. Only the upper steering column can rotate about the pivot point and hence the steering wheel.

10. MOTIVE

The adjustable steering system is designed to provide a comfort zone to the driver in handling the vehicle by varying the angle of the steering column keeping in mind the ergonomics of the vehicle.

CALCULATIONS

Diameter of holes for spring = 10mm
Length of steering column = 320mm.
Weight of adjustable assembly = 2kg
Distance between the holes = 20mm
Angle variation = 40° to 54°

11. CONCLUSION

We used the finite element analysis system to evaluate, create, and modify the best vehicle design to achieve its set goals. The main goal was to simplify the overall design to make it more light-weight without sacrificing performance and durability. The result is a lighter, faster, and more agile vehicle that improves go kart design.

12. REFERENCES