Go Kart Steering Design and analysis

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Abstract— This paper deals with the design of steering system for go-kart. The purpose of the steering system is to turn the go-kart but turning of go-kart without surging and tire squeaking is important. The geometrical relationship of turning radius and go-kart line of action is important for even tire wear and smooth turning. In this paper, the design and analysis of full Ackermann steering mechanism for go-kart is done.

Keywords—CAD, PVC, Ackerman etc.

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I. INTRODUCTION (*HEADING 1*)

Go-Kart racing is gaining tremendous popularity in professional as well as in beginner routine. Go-kart is relatively small in size but have enough power to create excitement while driving hence it is loved by both adults and kids. For kids it is used as fun and entertainment and for adults there is altogether different racing segment created because of go-kart. Go-kart consist of four tires which are smaller then normal tire but they are slick tires. It has small engine either two stroke or four stroke. Transmission can be automatic or cvt or manual depending on the choice of the individual. Steering system is normally of Ackermann type

II. EXISTING SYSTEM

Steering systems can be classified into two different designs:

- The Ackermann steering system
- The Bogie Steering system

Bogie system is the most efficient and simplest in design. The bogie system is where pivot in the center section and the two front wheels are mounted on a beam or the same axel. It is also called wagon style steering system, because it is similar to the system on simple pull behind wagons. This system is efficient because the wheels uniform ally scrub the same when the vehicle is turned and the follow the turn center, or turn circle in the correct geometrical relationship. The downside to typical wagon style steering is that the amount of movement required to make the gokart turn can be quite large.

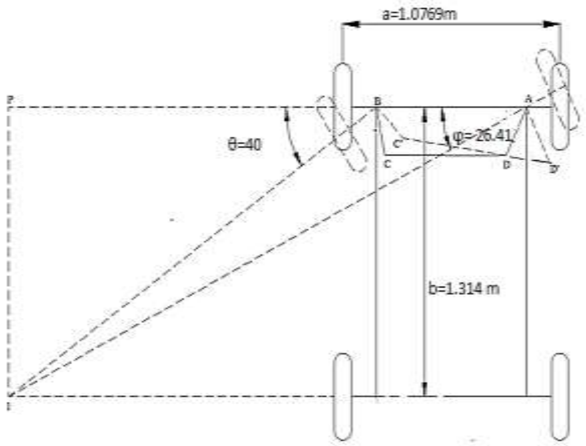
The Ackermann steering system is where the axles are mounted on knuckles out and away from the go kart. The wheels rotate vertically around these pivots, and cause the wheels to turn. There are number of parameters which needs to be found out for Ackermann steering system.

III. METHODOLOGY

- Mathematical calculations.
- CAD designing and stress analysis
- PVC model
- Final design

A. *Mathematical calculations.*

Steering of go kart is very hard to turn due to lack of differential it tends to move straight. Depending on wheel base and wheel track we calculated the turning radius and the turning angle of inner and outer wheel. In this we have used inversion of 4 bar link mechanism "Ackermann steering mechanism".



B. *CAD Designing and Stress Analysis*

The steering system consists of many components. Stress taking components are stub axle, spindle and knuckle bracket. We designed all the components safe for bending and shear load. Assuming the load distribution on kart as 60% on rear and 40% on front as this in important factor in deciding the effort required for turning the vehicle and assuming the gross weight of vehicle including the driver as 190 kg. Software used for designing and analysis is "Solid Works".

C. *PVC Model*

After designing on software and making all the components safe in various aspects we made the steering out of PVC pipe which helps us in understanding the placement of all components and driver positioning which should not compromise with the driver safety and ergonomics.

D. *Final design*

Finalizing on all above steps we finally made all the parts out of MS and for stub axle material used is AISI 4140 as this component has lot of stress concentration while turning.

Mechanical properties

Properties	Values
Tensile strength, ultimate	655 MPa
Tensile strength, yield	415 MPa
Bulk modulus, (typical for steel)	140 GPa
Shear modulus (typical for steel)	80 GPa
Hardness, Brinell	197
Poisson's ration	0.27-0.30

Chemical properties

Properties	Values %
Iron, Fe	96.785 - 97.77
Chromium, Cr	0.80 - 1.10
Manganese, Mn	0.75 - 1.0
Carbon, C	0.380 - 0.430
Silicon, Si	0.15 - 0.30
Molybdenum, Mo	0.15 - 0.25
Sulfur, S	0.040
Phosphorous, P	0.035

IV. RESULTS AND ANALYSIS/CALCULATIONS.

Analysis is done on 3 major component of steering system. The steering system is a mechanism of rigid linkages that helps in directing, handling in a desirable manner and effort. Below is the Solid-Work Model of steering system used.

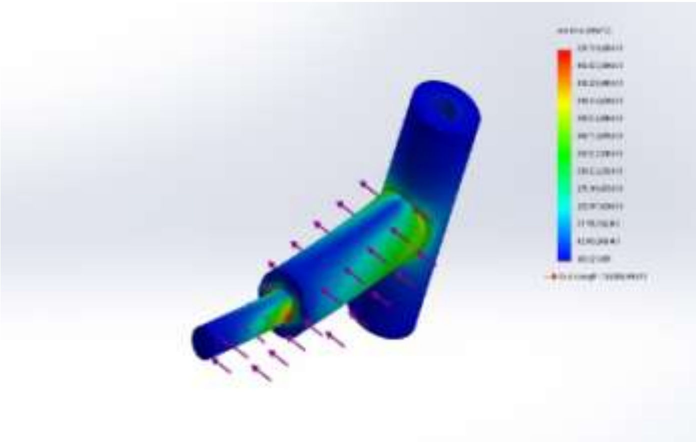
A. *Stub axle anlysis.*

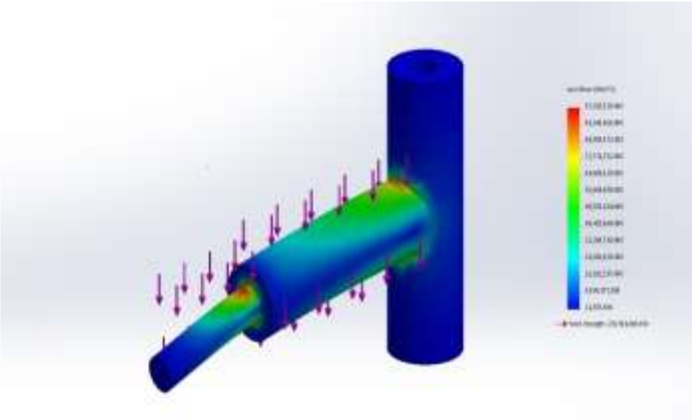
Stub axle analysis is depicted below
Vehicle Weight is 190kg with the driver. Vehicle Weight is very important to understand the vehicle dynamics and analyzing the vehicle dynamics in various aspects. Weight Distribution is 30-70%.
But for analysis we are using 40-60% distribution.
Assumptions -
1) Mass on front tires - 76kg
2) Average Velocity = 11.11m/s or 40kmph.

Normal Force on Stub Axle
= m x g= 76 x 9.81 = 745.5N

Lateral Force on Stub axle = mv2/r = 75 x 11.112/2.47 = 3797.9N

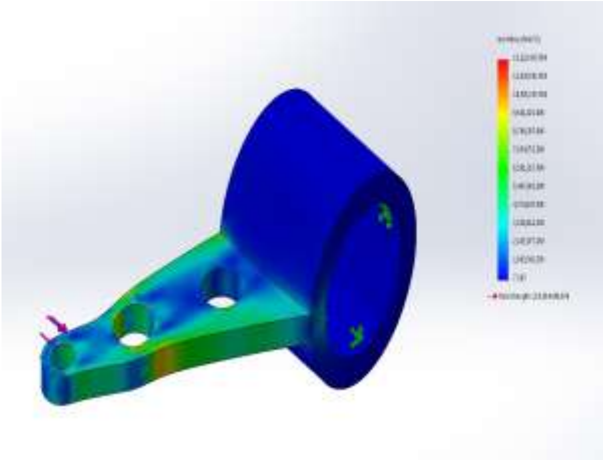
Tractive force that is Force due to traction
= μ x normal force = 0.7 x 745.5 = 521.85N





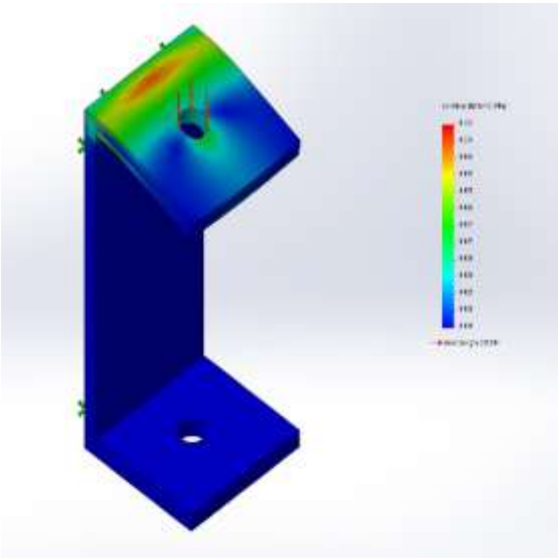
B. *Spindle analysis.*

A force of 60N was applied to it for calculation of shearing effect.



C. Force calculation on knuckle bracket.

The knuckle, is subjected to two nature of failures viz. Shear and Bending.
Considering,
SHEAR Failure at weakest section



Stress $\tau = \frac{W}{A}$

$= \frac{294}{b*t}$ … w= (30% of 200)/2

$= \frac{294}{5*80.8}$

∴ τ = 1.1529 N/mm along length

BENDING Failure at weakest section

The critical section for bend is rectangular
Thus we use governing equation for Bending stress with due respect to moment of inertia

$\frac{M}{I} = \frac{\sigma}{Y}$

Where, M= 294*40 N.mm

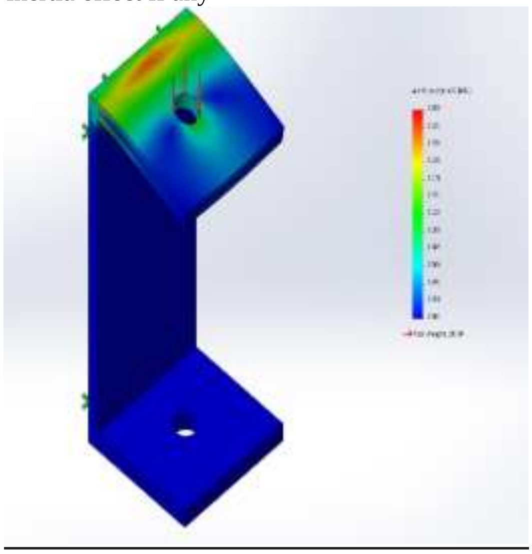
$I = \frac{b*d^3}{12} = \frac{40*30^3}{12}$ = 213333.33 N

$Y = \frac{b}{2} = \frac{40}{2}$

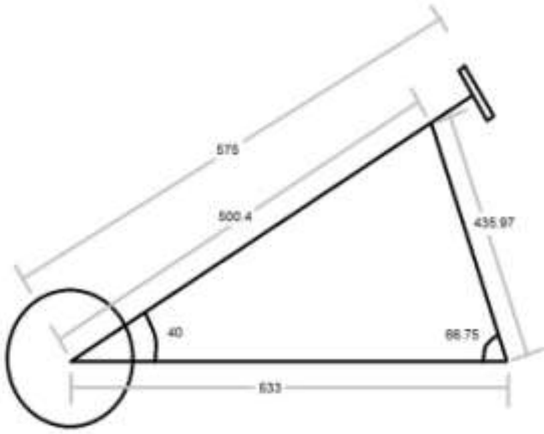
∴**Y= 20 mm** (the fiber of neutral axis)

$\frac{40*294}{0.1306} = \frac{\sigma}{20}$ ∴ $\sigma = 2.6133$ **Nmm** (bending stress)

Considering factor of safety 3 to compensate dynamics and nullify inertia effect if any



D. Steering column dimension diagram



All dimension are in mm in above figure.

1. Ackerman angle (β)

Ackerman angle can be calculated by using following equation

$\beta = \tan^{-1}\left(\frac{wheel\ track}{2 \times wheel\ base}\right)$

$\beta = \tan^{-1}\left(\frac{1.0769}{2 \times 1.3104}\right)$

β= 23degree

2. Distance between two king pin.

H=wheel track – (2 × distance between center of front tire and king pin)
H =1.0769 – (2 ×0.13258)
H =0.812m

3. Length of Tie- rod.

$$\sin\beta = \left(\frac{y}{length\ of\ ackerman\ arm}\right)$$

The length of steering arm =1.056m

$$\sin(23) = \left(\frac{y}{1.056}\right)$$

y = 0.4299m

4. Turning radius

Turning angle of inner wheel positively at 40 (θ= 40˚)

$$\cot\phi - \cot\theta = \frac{\text{wheel track}}{\text{wheel base}}$$

Thus turning angle of outer wheel (ϕ) =26.41˚

$$R1 = \frac{Wheel\ base}{\tan 40} + \frac{Wheel\ track}{2}$$

$$= \frac{1.3104}{\tan 40} + \frac{1.0769}{2}$$

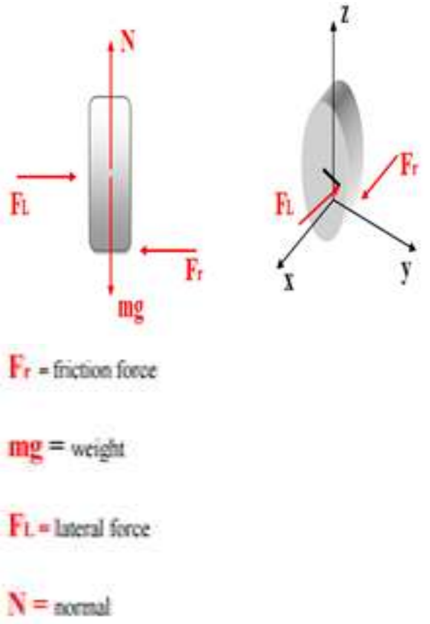
=2.1m

$$R = \sqrt{R1^2 + b^2}$$

$$R = \sqrt{(2.1)^2 + (1.3104)^2}$$

R = 2.47 m.

E. Tangential Force required by driver to turn wheel.



Mg =weight of wheel
Fr=frictional force
N= Normal force on wheel.
Fl=lateral force /force on tie rod.

∑Fx =0
-Fr + Fl =0 ……… (1)
Fl =Fr
∑Fy=0
N – Mg =0 ……….. (2)
N = Mg

Weight of cart = 120kg (without considering driver)
Weight of cart = 190kg (including driver)
30% of cart weight is on the front assembly
Thus equation is
=30/ (100) ×190
= 57 kg (on both wheels)
Thus load on each wheel will be = 57/2 = 28.5kg
=28.5×9.81
=279.58N
To calculate friction force (Fr)
Fr = μ×N
= 0.7× 279.58........ (μ is coefficient of friction. Considering safety μ=1)
= 195.706 N
Fr =Fl =195.706 N.
Thus we can calculate the torque on steering column which is transmitted by the tie rod to mediating link.
T = Fl × r ……… (Length of mediating link, r =0.14m)
= 195.706 ×0.14
= 27.398 Nm.
Tangential force required by the driver to turn the wheel (F)

F = (Torque)/ (Radius of steering wheel)
= 39.14/0.139
F = 280.57N (for each front wheel)

F. Results

Wheel Base(b)	1.3104m
Wheel Track	1.0769m
Turning radius	2.47m
Length of tie-rod	0.429m
King pin inclination	7˚ positive
Camber angle	2˚ positive.
Front wheel toe in	0
Rear wheel toe in	0
Steering wheel diameter	0.2794m
Scrub angle	To be intersected just below surface

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