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Review on Design & Development of Go-Kart Steering System

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Abstract: Go-kart is one of the motor sports where racing of bunch of vehicles are compete each other. This paper is written after completion of manufacturing and complete assembly. Following paper consist of diagrams, calculations and results from actual models of steering and electrical systems. Steering system is one among crucial areas in designing of go kart as even the slightest of improvement in response of this technique could reduce the lap time and help to succeed in the driving force beyond the finishing line to win an edge.

Thus, steering has got to be reliable enough such the driving force could have the entire control over the kart even within the toughest tracks. On the other hand, any failure in system could lead to serious injury or loss to the driver and the team. This paper is written with an aim to enhance a steering mechanism of go kart and overall responsiveness and control. This paper covers most of the concepts of steering mechanism of knowledgeable go kart. With the assistance of this paper one could understand and manufacture complete steering assembly individually.

The paper consists of theory, formulae, calculations, diagrams and simulation results which give top to bottom understanding of knowledgeable go kart steering mechanism.

Keywords: Ackermann, Steering, Geometry, Go-Kart, CAD, Tie-rod, Stub axle.

I. INTRODUCTION

Steering system is that the collection of components, linkages, etc. which permit a vehicle to drive through the specified course. The first purpose of this mechanism is to permit the required driving force to guide the vehicle. The steering mechanism is to realize angular motion of the front wheels to barter a turn. This is often done through linkage and gear which convert the rotation of the front wheels into angular motion of them. Secondary functions of the steering system are:

1) To supply directional stability of the vehicle when going straight ahead.
2) To supply perfect steering condition, perfect rolling motion of the road wheels in the least time
3) To facilitate and achieve straight ahead recovery after completing a turn. To minimize tire wear.

Nowadays major of the vehicles use rack and pinion steering mechanisms, where the wheel turns the pinion gear; the pinion moves the rack, which can be a linear gear that meshes with the pinion, which converts the circular motion into linear motion. Older designs often use the recirculating ball mechanism, which remains found on trucks and utility vehicles. The recirculating ball mechanism has the advantage of a way greater ratio, in order that it had been found on larger, heavier vehicles while the rack and pinion was originally limited to smaller and lighter ones. To achieve the right steering, two sorts of mechanisms are used. They are the Davis & Ackermann mechanism. Ackermann steering geometry may be a geometric arrangement of linkages within the steering of a car or other vehicle designed to unravel the matter of wheels on the within and outside of a turn wanting to trace out circles of various radius.

A simple approximation to perfect Ackermann steering geometry could also be generated by moving the steering pivot points inward so on lie on a line drawn between the steering kingpins and the centre of the rear axle. The steering pivot points are joined by a rigid bar called the rod which may even be a part of the steering system, within the sort of a rack and pinion for instance. With perfect Ackermann, at any angle of steering, the middle point of all of the circles traced by all wheels will lie at a standard point. Note that this might be difficult to rearrange in practice with simple linkages, and designers are advised to draw or analyze their steering systems over the complete range of steering angles.

II. METHODOLOGY

This project will undergo through following six phases.

1) Phase I : Literature Survey

A detailed literature survey will be carried out in the related area. Majorly the selected project is come under industrial field influence, so in this phase we will do small scale industrial visits, Feedbacks and problems faced by vendors.

2) Phase II : Concept Generation

In this phase, we are going to do schematic arrangement design and drawing of major component which we can use for completion of our project. In this phase we will generate the schematic drawing on the basis of problem statement and feedback and suggestion received from end customer and vendors.

3) Phase III : Design calculations

In this phase we are going to do the design calculations by referring the standards, catalogue and reference books. In this work we will finalize the design and components dimensions. We are also selecting the material according to parts and components function and loading conditions. In this phase we will decide the size and shape of components and its position in the assembly. Also we will decide the limit and tolerance between components and also machining methods required to select to manufacture the components.

4) Phase IV : Preparation of Drawings

In this phase we are going to prepare the design. The suitable component and assembly drawings will be prepared which will help visualize the actual project set up. In this phase we will prepare the drawing as per industrial format.

5) Phase V : Structural Analysis of the Critical Components

In this phase we will do analyses of one component which are under critical loading condition. By doing analysis we can decide the final dimensions and material of the component.

6) Phase VI : Fabrication

a) The components will be assembled per the drawing.

b) Working trials of the project will be conducted to confirm and testing parameters (Time and speed) we will decide for to get best quality of product.

7) Phase VII : Experimental Investigations (Actual Field Trial)

The fabricated mechanism will be tested for the suitability to the intended application. This experimental testing will include the testing of vehicle at actual site.

III. FACTORS IN STEERING GEOMETRY

A. Four bar Ackermann Mechanism

A four bar Ackerman is essentially supported the four-bar linkage. In a four-bar linkage there are 4 links which are connected in loops by 4 joints. Generally, the joints are configured in order that the links move in parallel planes. The Ackerman mechanism uses four bar linkages.

The input motion from the driving force and therefore the wheel is transmitted via a steering tie rods and the steering control linkage to at least one of the steering knuckles then transmitted to the opposite one through the Ackerman mechanism .

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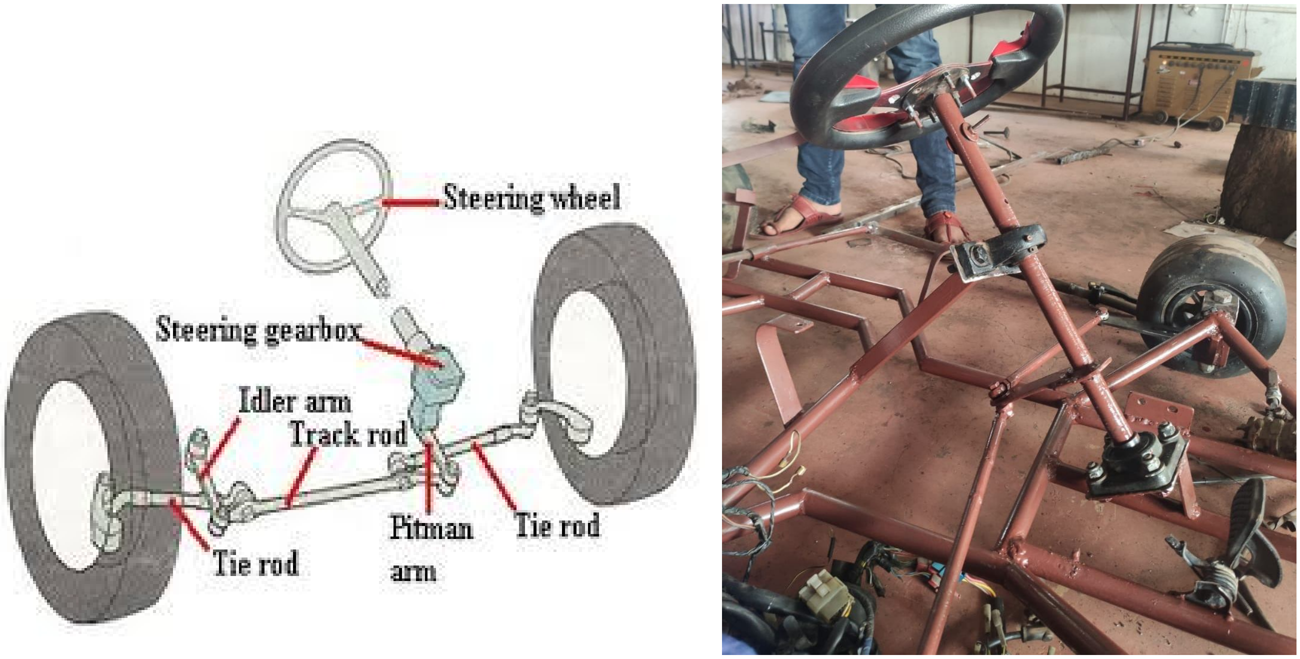

Fig..1 steering system using tie rod linkages

B. Ackermann Principle

To achieve true rolling for a four-wheel vehicle moving on a curve track the lines drawn through each of the four-wheel axes must intersect at the instantaneous centre. The actual position of the instantaneous centre constantly changes thanks to the alteration of the front wheel angular positions to correct the steered vehicles path. Since both rear wheels are fixed on an equivalent axis but the front wheel axles are independent of every other the instantaneous centre lies along an imaginary extended line drawn through the axis of the rear axle.

The Ackermann Principle is predicated on the 2 front steered wheels being pivoted at the ends of an axle-beam. The main advantage of Ackermann Principle is that in cornering the wheel slippage is minimum and hence gives a far better traction which is required in racing.

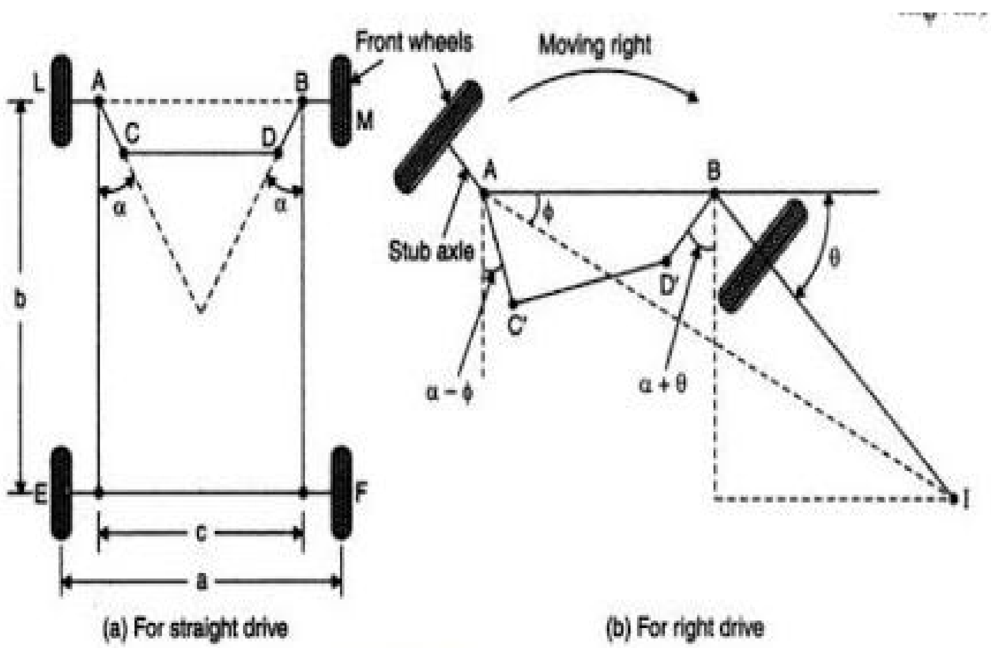

Fig.2 Ackerman Geometry

C. Ackermann Calculations

For the Ackermann analysis the Ackermann condition is used to determine the relationship between inner and outer wheel in a turn and the radius of turn.

General equation

Where:

θ= turn angle of the wheel on the outside of the turn

ϕ= turn angle of the wheel on the inside of the turn

C= track width

B= wheelbase

b= distance from rear axle to centre of mass

From the general equation we can calculate the turn angle of the wheel on the outside of the turn for a given inside wheel angle as follows:

B= 1070 mm

C= 560 mm

$\theta = 40°$

$\tan\theta = B/Y + C$

$\cot \Phi = Y/B + C/B$

Therefore,

$\mathrm{Cot}\ \Phi = \cot \theta + C/B$

Now

$(1/\tan\Phi) = (1/\tan 40) + (560/1070)$

$= 1.19+0.5233$

$= 1.7133$

$(1/\Phi) = \tan^{-1}(1.7133)$

$\Phi=33.43^{o}$

- Ackerman Formula

$\mathrm{Tan}\ \alpha = (\sin \Phi - \sin \theta) / (\cos \Phi+ \cos \theta -2)$

$= (\sin 33.43 - \sin 40) / (\cos 33.43 +\cos 40 - 2)$

$= 0.23$

$\alpha = 12.95^{o}$

$\alpha - \Phi = 12.95- 33.43$

$= -20.48$

$\alpha + \theta = 12.95 + 40$

$= 52.95^{o}$

- Turning ratio for inner front wheel

$RIF = (b / \sin\alpha) - ((c-a)/2)$

$= (1070 / \sin 40) - ((715 - 560) / 2)$

$= 1587.1244$ mm

$= 1.58$ m

- Turning ratio for outer front wheel

$ROF = (b/\sin \theta) + ((c - a) / 2)$

$= (1070 / \sin 30.24) + ((715 - 560) / 2)$

$= 2202.1$ mm

$= 2.2$ m

D. Steering Movement Ratio

The overall steering ratio is defined as the ratio between steering wheel angle and front wheel angles is known as overall steering angle. This ratio does varies between 20:1(slower) to 10:1 (faster). Usually, the steering ratio used for go-kart is 1:1. Common values in Go-Kart racing are 16:1 to 18:1. And the steering ratio used for our go-kart is kept 1:1.

The rack and pinion mechanism is meant to transfer the circular input motion of the pinion into linear output movement of the rack. It was measured that for a full travel of the rack of 295 mm the pinion has to be rotated 3.5 turns, therefore for one turn, the rack travel will be:

Xo= 295*3.5=84.28

Considering the pinion to form one revolution then the input steering movement is

Xi=2πR

Where, R = 190 mm is the radius of the steering wheel.

And the output rack movement is:

Xo=$2\pi r$

r = 84.28*2π=13.42

Then, the movement ratio is often calculated as input movement over output:

MR= Xi/Xo= $2\pi R/2\pi r$ = 190/14=13.57

Therefore, the movement ratio is 14:1

We needed to know the movement ratio in order to determine the output load transmitted to the tie rods for a given input load. For an effort of 20 N applied by each hand on the steering wheel and considering no friction, the output load will be:

Fo= F1 x MR=560

Therefore, the load transmitted to the tie rods is 560.

IV. BASIC STEERING COMPONENTS

About 99% of the world's car steering systems are made from an equivalent three or four components. The wheel , which connects to the steering mechanism , which connects to the track rod, which connects to the tie rods connects to the steering arms. The steering mechanism are often one among several designs, which we'll enter further down the page, but all the designs essentially move the track rod left-to-right across the car. The tie rods hook up with the ends of the track rod with ball and socket joints, then to the ends of the steering arms, also with ball and socket joints. The purpose of the tie rods is to permit suspension movement also as a component of adjustability within the steering geometry. The rod lengths can normally be changed to realize these different geometries. In the simplest sort of steering, both the front wheels always point within the same direction. You turn the wheel; they both point an equivalent way and round the corner you go. Except that by doing this, you finish up with tires scrubbing, loss of grip and a vehicle that 'crabs' round the corner. So why is this? Well, it is the same thing you would like to require into consideration when watching transmissions. When a car goes around a corner, the surface wheels travels further than the within wheels. In the case of a transmission, it's why you would like a differential (see the Transmission Bible), but within the case of steering, it's why you would like the front wheels to actually point in different directions. This is the diagram from the Transmission Bible. You can see the within wheels travel around a circle with a smaller radius than the surface wheels.

In order for that to happen without causing undue stress to the front wheels and tires, they need to point at slightly different angles to the centerline of the car. The following diagram shows an equivalent thing only zoomed in to point out the relative angles of the tires to the car. It is all to try to with the geometry of circles. This difference of angle is achieved with a comparatively simple arrangement of steering components to make a trapezoid geometry (a parallelogram with one among the parallel sides shorter than the other). Once this is often achieved, the wheels point at different angles because the steering geometry is moved. Most vehicles now don't use 'pure' Ackermann steering geometry because it doesn't take a number of the dynamic and compliant effects of steering and suspension under consideration, but some derivative of this is often utilized in most steering systems.

A. Minimum Turning Radius

After obtaining the Ackermann angle by geometry method, minimum turning radius has to be decided. Usually in professional go karts the minimum turning radius doesn't exceed 2.5m. Hence after deciding the minimum turning radius of go kart the maximum inner and outer angle values can be found by the calculations provided in this paper. These values wouldn't permanently fix the MTR after manufacturing the system therefore the designer needn't worry.

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These values are only for the designer to understand what proportion the inner and outer wheel would rotate for that specific MTR. The MTR might be also adjusted after manufacturing the steering mechanism . The steering arm have to be restricted from rotating beyond a point preventing the locking of four bar mechanism.

This helps to adjust the minimum turning radius as well as it makes safe steering system.

B. Camber

It is recommended to not focus much on this think about go kart and keep internet camber value zero represented in Fig. 4. Net camber is that the final camber angle value of the wheel of loaded kart regardless of the stub axle angle with king pin to which is typically kept to compensate the king pin inclination.

While performing a activate unbanked road the CG of go kart shifts outwards from direction of steered turn (i.e., towards left when steered clockwise and vice versa) thanks to force. This might cause understeer which the camber would prevent by playing its part.

Camber angles play an important role when suspensions are available picture. Usually, negative camber is meant for tires of economic racing cars for better grip while cornering. It results proper traction of the outer wheel when performing a turn because the lateral force pulls the rubber of tire to make a far better tire-road interface which is assisted by weight shift of car and depends on velocity of car at that instant. This traction formed, helps the car to remain in its intended track throughout the corner.

For go karts, either zero or few degrees of negative camber is generally required. Since there are not any suspensions in go kart, there's no chance of chassis' relative deflection with reference to wheels then only deflection that might occur is due to deformation and flex in go kart frame. This deflection could lose the traction in tire with the road surface which could be restricted by negative camber, but these deflections are minute and difficult to analyse in go kart as it doesn't make much sense for negative camber.

Manufacturing such minute angles is not possible without automized equipment or proper designed jigs and fixtures. Thus, setting camber on go kart may be a complex design challenge for the designer.

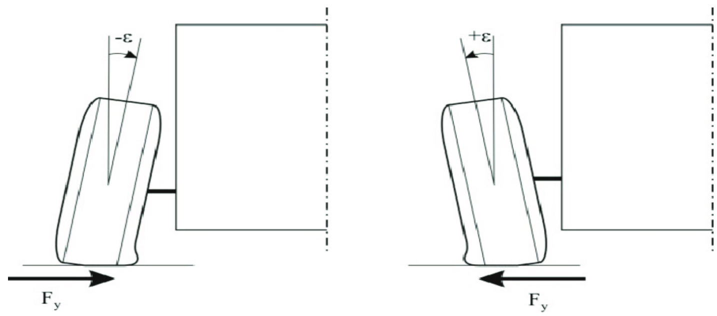

Fig.3. Camber angle

C. Caster Angle

Rise in this angle will increase the jacking effect but also will rise the steering effort. Caster angle leads to elevation of left front side of kart and lowering of the right front side of kart when steering wheel is spun anticlockwise and vice versa. It is called jacking effect. This happens thanks to rotation of stub axle during a plane which is oblique to the bottom. This assists the go kart to perform turns in sharper manner by inclining the front a part of go kart, there by resulting greater manoeuvrability.

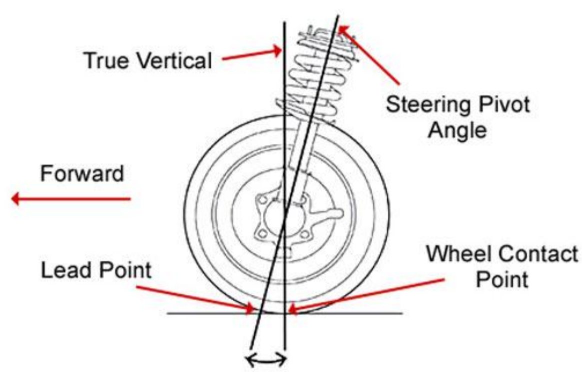

Fig.4. caster angle

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D. King pin Inclination (KPI)

Imagine a knuckle assembled on the king pin at certain KPI. Now, when an upward force (weight of the kart acting against the ground) is acted at the end of stub axle the stub axle will tend to rotate about the pivot (KP) and settle at topmost position in height as its equilibrium. Imagine the similar scenario when the is wheel assembled to the stub axle. The wheel tends to roll on the ground surface and place itself to certain position which is the desired neutral position. Fig. 6 shows 10° KPI which was used in go kart.

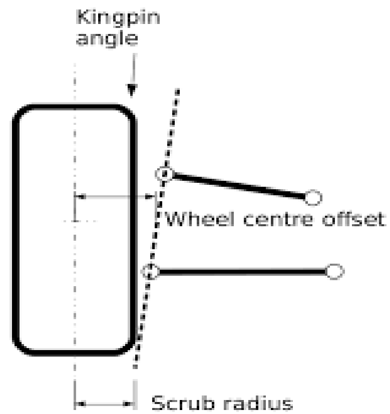

Fig.5. King Pin Inclination

V. DESIGN CALCULATIONS

Various calculations are tabulated as follow according to the vehicle specifications.

Table 1. Steering Specifications

Turning	
Inner Front Turning Radius	1602. 47m m
Outer Front Turning Radius	2622. 07m m
Turning Radius of C.G	1779. 79m m
Inner Locking Angle	40°
Outer Locking Angle	24.7°
King pin Inclination	12^0
Castor Angle (Positive)	2^0
Steering Wheel Diameter	13 Inch
Torque at Steering Wheel	11.59 8N-m

Table 2. Steering Actual Dimensions

Actual Dimensions Taken						
Parts	Length (mm	Width (mm	Height (mm	Outer dia (mm	Inner dia (m	Thickness
Steering arm				-	-	-
Tie rod	380/319	-	-	14.00	10	2
Knuckle fork	170.00	32	5	-	-	-
King pin post	-	-	100	20	10	5
Stub axle	165	-	-	17	-	-
Pitman arm	122	-	40	-	-	-
Steering column	490	-	-	25.46	20.46	2.5
Steering wheel	-	-	-	320	260	30
Bolts	-	-	-	6	-	-
Rose joint	-	-	-	6	6	-
King pin	70	-	-	10	-	-

A. Steering Effort Calculations

1) Mass of a kart including driver (M) = 150 kg
2) Mass on one wheel (m) = 30 kg
3) Scrub radius (r) = 72.65mm
4) Length of steering arm (s) = 170 mm
5) Length of pitman arm (p) = 122 mm
6) Radius of steering wheel (R) = 160 mm
7) Static coefficient of friction (μ) = 1
8) Angle of steering arm with longitudinal axis = 0^o
9) Restoring torque (τ) = mg*r* = 21.38 Nm
10) Perp. force on steering arm (F) = s = 21.38 /0.122 = 175.46 N
11) Force on pitman arm (F_1) is given by = F/Cos (17.45) = 183.92 N
12) (T_1) = F_1 p = 22.44 Nm
13) Steering effort = T_1/R = 22.44 /0.16 = 140.239 N = 14.29 kg
14) Force on one hand = 14.29/2 = 7.15 kg

B. Weight Transfer Calculations:

Lateral weight transfer:

g' = Cornering power

R'= Turning radius in feet

T' = Time required to complete a complete circle of radius R'

V' = Maximum velocity without slipping for turning radius of R'

$$g' = \frac{1.225 \times R'}{T'^2} = \frac{1.225 \times 1.853}{0.3 \times 2.82^2} = 0.9514 \text{ N}$$

$$\text{Lateral weight transfer} = = \frac{M \times height\ of\ C.G \times g'}{Trackwidth} \frac{150 \times 9.81 \times 0.2 \times 0.9514}{0.910} = 307.69 \text{ N}$$

Weight increase on front wheel = 307.69 x 0.4 = 123.07 N
Weight transfer due to braking:
g" = deceleration due to braking

Weight transfer = $\frac{M \times g \times g'' \times H}{Wheelbase} = \frac{150 \times 9.81 \times 1 \times 0.2}{1.07}$ = 275.04 N

Weight increase on one wheel = (275.04/2) = 137.52 N
Total weight on front wheel = (30 x 9.81) + 123.07 +137.52 = 554.9 N = **56.67 N**
(*Note*: *The above value of mass or weight will be used for design calculations.)*

C. Design Calculations

1) Stub Axle:

Stub axle is considered as a cantilever beam of circular cross section.
Factor of safety is considered as 2.
Material selected = EN8
Yield Strength = 323 N/mm2
Stub axle length= 165 mm
Bending Moment calculations
Bending moment = 35236.15 Nmm
Bending moment diagrams for vertical loading and horizontal are equal because coefficient of friction is considered as 1.
Horizontal Bending Moment = Vertical Bending Moment= 35236.15 Nmm

Resultant Bending Moment = $\sqrt{(M_v^2 + M_h^2)}$ = 49831.44 Nmm

Design of Stub Axle Length
Bending Moment of Stub Axle = 49831.44 Nmm
For Solid Shaft
d = diameter of stub axle
Bending Stress of Stub Axle = 323 N/mm^2

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

$$\frac{49831.44}{\frac{\pi d^4}{64}} = \frac{323}{2 \times \frac{d}{2}}$$

d = 13.04 mm

2) Steering Arm

Steering arm is taken as a cantilever beam of rectangular cross-section
Force is assumed to act through bolt in hole where tie rod is attached to steering arm.
Factor of safety is considered as 1.5.
Material Selected = AISI 1018 (plate)
Yield strength = 255 N/mm^2
Here,
b = width of steering arm
d = thickness of steering arm
Moment on steering arm = M =
Bending stress on steering arm = σ = 255 N/mm^2

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

$$\frac{40244.8}{\frac{bd^3}{12}} = \frac{255}{1.5 \times \frac{d}{2}}$$

$Bd^2 = 1420.38$

Assuming,

b : d = 4 : 1

$1420.38 = 16 \times b^3$

$b^3 = 88.774$

b = 4.46 mm

& d = 17.84 mm

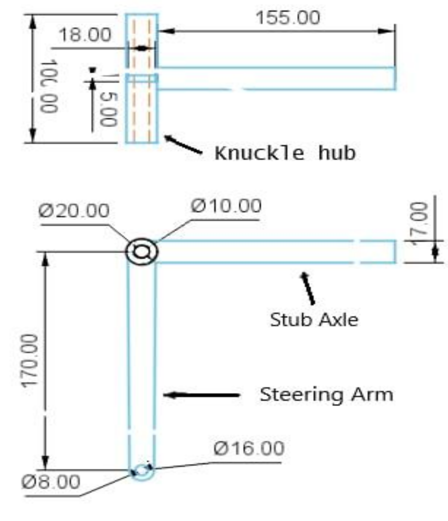

Fig.6. Knuckle joint

3) Pitman Arm

Pitman arm is also assumed as a cantilever beam of T section.

Force is considered to act on the beam through bolt.

Factor of safety is considered as 1.5

Material Selected = AL 6063

Yield Strength = 170 N/mm2

Here,

b = width of pitman arm

d = thickness of pitman arm

Moment on Pitman Arm (M)

$$= \frac{Moment\ on\ steering\ arm \times pitman\ arm\ length}{Steering\ Arm\ Length \times \cos(angle\ at\ which\ perpendicular\ force\ is\ acting)}$$

$$= \frac{40244.8 \times 122}{170 \times \cos(17.45)} = 30274.84\ \text{Nmm}$$

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

$$\frac{30274.84}{\frac{bd^3}{12}} = \frac{170}{1.5 \times \frac{d}{2}}$$

$Bd^2 = 1602.78$

Assuming,
(b/d) = 0.25
16 x b^3 = 1602.78
b^3 = 100.17
b = 4.64 mm
d = 18.58 mm

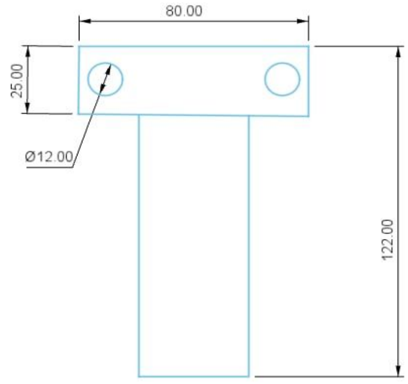

Fig.7. pitman arm

4) Steering Column

Steering column is assumed to be hollow and undergo torsion due to the torque acting through the steering wheel. It is assumed to be fixed at the position where pitman arm is attached.

The length of steering column used for design calculation is approximately calculated according to driver's comfort (600mm).

Factor of safety is considered as 1.5
Material Selected = AL 6061 T6
Shear Strength = 207 MPa
Here,
d0 = outer diameter of shaft
di = inner diameter of shaft
Torque on Steering Column (T) =30274.84 Nmm

Polar Moment of Inertia (J) =$\pi \times \frac{{d_o}^4 - {d_i}^4}{32}$

Shear strength of column (τ) = 207N/mm^2
Outer Diameter of steering column (d_o) = 25 mm (1inch)
Radius of steering column (R) = 12.5 mm
Shear Modulus (G) = 25800 MPa
Length of Steering column= 490 mm
Angle of Twisting (θ) = 3 = $3 \times \frac{\pi}{180}$
Thickness (t) = 2.5 mm

$$\frac{\tau}{R} = \frac{T}{J}$$

$$\frac{207/1.5}{d_o/2} = \frac{30274.84}{\pi \times \frac{{d_o}^4 - {d_i}^4}{32}}$$

d_i = 25.07 mm
t = 0.162 mm

The dimensions as calculated on rigidity basis yield greater dimensions therefore they are selected.

5) Tie Rod

Tie rod is designed for buckling load as the force acts along the longitudinal axis. The two ends are assumed to be hinged.

Factor of safety is considered as 1.5

Material Selected = AISI 1018

Yield Strength = 170 N/mm2

Here,

n = factor accounting for end conditions
E = modulus of elasticity
I = moment of inertia about neutral axis
L = length of longer tie rod
Factor of safety = 1.5

Force on tie rod (F) = Moment on steering arm) / (Length of steering arm)
= 40244.28 / 170
= 236.73 N

$$F = \frac{n\pi^2 EI}{L^2}$$

$$236.73 = \frac{1 \times \pi^2 \times 6.89 \times 10^4 \times \pi \times (d_o{}^4 - d_i{}^4)}{1.5 \times 64 \times 380^2}$$

$$(d_o{}^4 - d_i{}^4) = 1537.21$$

Here we assume (d_o / d_i) = 0.8

So,

d_o = 7.14 mm

d_i = 5.71 mm

Thickness = 1.24 mm

(Note: Tie Rod dimensions according to calculations are very small. Therefore, the dimensions are taken through manufacturing and geometrical considerations and ease of servicing.)

VI. CONCLUSION

In this paper, the design and assembly of the go-kart steering system is presented with the help of the researchers and their documentations, the purpose of the steering system their requirements, the design methodology, the overview of the steering system used in the go-karts are all mentioned in this paper. The steering mechanism used which is Ackermann steering geometry is discussed with the steering conditions and is explained with terms considered in mechanism and the assumptions made during the solution of the Ackermann steering geometry problems.

The analysis of the steering system components through design load calculations can be performed which determines the stresses, loads and deformation of the steering system from which the design engineers can predict the safety of the system and can also be modified and minimization of the errors in the systems can be done. The manual mechanical linkages steering mechanism isn't utilized in heavy weight vehicles due to high axle loads, although it's simple in design and straightforward to manufacture, therefore it is commonly used in light weight vehicles. The values calculated within the paper may differ practically due to steering linkages error or due to improper steering geometry, so these values are useful to know the interdependency of the quantities on one another and to style an ideal manual mechanical linkages system for the vehicle. Considering all the data mentioned, this can be used as better system for Go-Kart.

VII. ACKNOWLEDGEMENT

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