Design and Development of Ackerman Steering System for Formula Student Vehicle

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Abstract – *In this paper, a steering system is designed for FORMULA BHARAT student vehicle, which adopts a rack-and-pinion steering mechanism. Team planned to design and develop a custom steering design that not only provide high accuracy but also met our weight & budget norms. The theoretical modeling of the systems as well as the derivations of the optimal parameter values is presented here. By understanding the vehicle requirements first we finalized the Ackerman angle. Based on that angle the geometry of the Ackerman steering system and all the design calculations of the each component of the steering system are also presented.*

Key Words: Steering System, Formula Student, Ackerman Steering, Rack and Pinion.

1. INTRODUCTION

As we are working on a national level project named FORMULA BHARAT in which the design and development of a Formula Student race car was to be done and for that we had the task to design and develop a Ackerman Steering System that facilitated the driver to take sharp turns with less efforts or with less revolutions of steering wheel. To achieve this we decided to modify the conventional steering mechanism that is used in the normal round cars. The steering system of the car is rack and pinion based mechanism that converts the rotational motion generated at the steering wheel into a linear motion at the end of the rack. The design is based on the rule book of FORMULA BHARAT 2019, according to which drive by wire forbidden and hence we have selected a simple rack and pinion system with no additional electrical or hydraulic help. Though the tracks used for such events are flat in nature, one must account for the natural kinematic behavior of the steering system and hence it is essential to not only factor static stress but also the dynamic aspects of the steering system.

2. DESIGN PROCEDURE

The steering system is a front wheel based steering unit, as it in most formula student cars, the design involves formation of mathematical and geometrical model followed by CAD and FEA procedure. The approach in designing said system involves the following steps,

1. Identification of the vehicle requirements
2. Geometrical set up
3. Geometrical validation
4. Design of mechanism
5. Modeling and Analysis by CAD and FEA respectively

3. DESIGN SPECIFICATIONS

Parameters	Values
Turning radius	2300 mm
Inner wheel angle	42.35°
Outer wheel angle	28.12°
Ackerman angle	42.32°
Steering arm length	85.78 mm
Designed rack travel	90.64 mm
Steering ratio	3.2
Tie rod length	440.05 mm
Steering effort	83.60

Table -1: Steering Design Specifications

4. GEOMETRY DESIGN

4.1 Steering Geometry

The requirements are in accordance with the standard rulebook of FORMULA BHARAT, but are also made sure to be satisfactory to the driver comfort and also to ensure safety to the driver. That's why Ackerman steering geometry is chosen. Ackerman steering geometry is geometrical arrangement of linkages of the steering of vehicle. The kinematic condition between inner and outer wheel that allows them to turn slip-free is called Ackerman steering.

The following parameters are set up,

- Wheel Track = 1200 mm
- Wheel Base = 1550 mm
- Radius of Curvature = 2300 mm

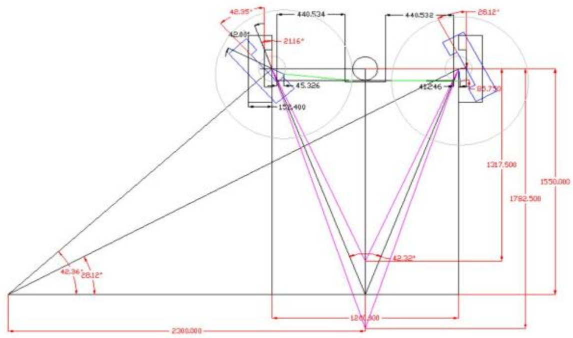

Fig -1: Ackerman Steering Geometry

4.2 Bump Steer:

To avoid the lateral motion of wheel, while vehicle is in bump, a concept of bump steer geometry is implemented Conditions to achieve bump steer are:-

- Instantaneous center of both wishbones and tie rod must be same. [2]
- Tie rod length is such that, it must be attached to the line of intersection of upper and lower ball joints. [2]

The below figure explains more about bump steer geometry.

So as to achieve bump steer in our vehicle, steering rack is place at 56 mm above floor. The following figure shows bump steer geometry drawn in AUTO CAD. Vehicle has same instantaneous of both wishbones and steering rack.

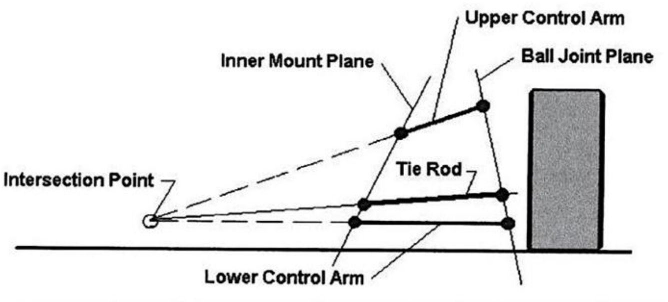

Fig -2: Bump Steer Geometry 1[5]

Value of the bump steer can be found from the below geometry.

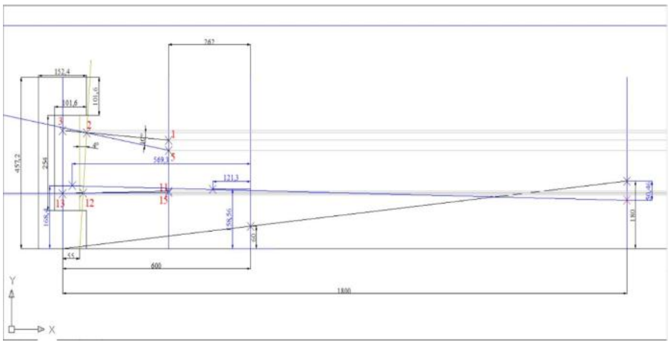

Fig -3: Bump Steer Geometry 2

5. DESIGN CALCULATIONS:

5.1 Steering System Design Calculations:

Assumptions:

Turning radius (R) = 2300 mm

Wheel base (b) = 1550 mm

Wheel Track (t) = 1200 mm

Vehicle Weight (W) = 280 Kg

$\theta = inner\ angle$

$\emptyset = outer\ angle$

TW= track width

WB= wheel base

R=radius

5.2 Ackerman geometry calculation:

$\theta = \tan^{-1}(\frac{WB}{R-\frac{TW}{2}})$ ……. [3]

$=\tan^{-1}(\frac{1550}{2300-\frac{1200}{2}})$

$=42.35º$

$\emptyset = \tan^{-1}(\frac{WB}{R+\frac{TW}{2}})$ ……… [3]

$=\tan^{-1}(\frac{1550}{2300+\frac{1200}{2}})$

$=28.12º$

Ackerman Angle $= 2 * \tan^{-1}(\frac{TW/2}{WB})$ ……….[3]

$= 2 * \tan^{-1}(\frac{\frac{1200}{2}}{1550})$

= 42.32º

Steering arm angle = $\tan^{-1}(\frac{TW}{2WB})$ .... [3]

= $\tan^{-1}(\frac{1200}{2*1550})$

=21.16º

Wheel lock to lock rotation = θ + ∅ =42.35° + 28.12 ° = 70.47°

% of Ackerman = $\frac{\theta}{\text{Ackerman Angle}} = \frac{42.35°}{42.32°}$ = 100%

5.3 Determination of Steering Ratio

Existing Steering ratio

= $\frac{\text{Existing Steering wheel rotation for total rack travel}}{\text{Wheel's lock to lock rotation}} = \frac{225.5}{70.47}$ ..... [5]

Existing Steering ratio = 3.20

5.4 Determination of Steering Effort

Weight on front axle = 0.5 × 280 = 140 kg

Weight on each front wheel = $\frac{140}{2}$ kg = 70 × 9.81 = 686.6 N

Friction force offers resistance to wheel rotations while negotiating turn.

Friction force = μ× weight on each front wheel = 0.7 × 686.6= 480.7 N

Torque required at steering wheel = Friction force× radius of pinion

Torque required at steering wheel = 480.7 × 20= 9614 N.mm

Steering Effort = $\frac{\text{Torque required at steering wheel}}{\textit{Radius of Steering wheel}} = \frac{9614}{115}$ = 83.60 N [3]

5.5 Calculation for M6 bolt:-

P= 2kN – lateral force on tie rod

Factor of safety – 2.5

Permissible tensile strength = 400 N/mm^2

$\sigma = \frac{P}{\pi * d^2/4}$ .... [1]

$160 = \frac{2000}{\pi * d^2/4}$: d=3.98mm

5.6 Bending calculations for bolt: -

P=2000N

D=8mm

$\sigma b = \frac{P/2 \; \{b/4 + a/3\} \frac{d}{2}}{\pi * \frac{d^4}{64}}$ .... [1]

$= \frac{2000/2 \; \{3/4 + 2/3\} \frac{6}{2}}{\pi * 6^4/64}$

= 66.80 N/mm^2

Double shear calculations:

P=2000 N

D=6 mm

$D = \sqrt{\frac{2*P}{\pi * \tau}}$ ...... [1]

$8 = \sqrt{\frac{2*2000}{\pi * \tau}}$

τ= 21.21 N/mm^2

During double shearing,

τ = 65.01 N/mm^2

65 > 21.21 N/mm^2 : Design is safe.

5.7 Tie rod calculations:

$\theta = \frac{584 * Mt * l}{G * d^4 * (1 - c^4)}$ ..... [1]

$\theta = \frac{584 * Mt * 448}{0.8 * 10^\wedge 5 * 10^4 * (1 - .72^4)}$

= 25.6kN/mm

$\tau\,(max) = \frac{16}{\pi * d^3 * (1 - c^4)} \sqrt{Mb^2 + Mt^2}$

$92.5 = \frac{16}{\pi * 10^3 * (1 - 0.72^4)} \sqrt{Mb^2 + 25654.9^2}$

Mb= 25.8KN/mm

$\sigma b = \frac{32 * Mb}{\pi * d^3 * (1 - c^4)}$ .... [1]

$\sigma b = \frac{32 * 25.8 * 10^3}{\pi * 10^3 * (1 - 0.72^4)}$

σb= *131.3N/mm^2*

Shear stress*:* $\tau s = \frac{16 * Mt}{\pi * d^\wedge 3(1 - c^4)}$ ...... [1]

$\tau s = \frac{16 * 25.6 * 10^3}{\pi * 10^\wedge 3(1 - 0.72^4)}$

τs = 65.1 N/mm^2

5.8 Steering Column Calculations:

Required Data:

- Length of column(L)=200 mm
- C factor=di/d0=0.5
- Permissible angle of twist=3^0
- Diameter of steering wheel(D)=230 mm
- Modulus of rigidity of stainless steel= 77200Mpa
- Maximum torque on steering wheel (Mt)
  Force on steering wheel (F) = 83.6 N
  Mt=D*F=230*83.6
  Mt=19228 N-mm
- Design of Hollow shaft on basis of Torsional Rigidity:

  $\Theta= (584*Mt*L)/ 77200*d_0^4 (1-c^4)$ .....[1]

  $3 = (584*19228*200)/77200*d_0^4 (1-0.5^4)$

  d0=12.09mm i.e. d0=14mm
  Therefore di=7mm

  $\Theta= (584*Mt*L)/ 77200*(14)^4 (1-0.5^4)$
  $\Theta=1.6684^0$
  Hence, Factor of Safety =3/1.66
  =1.79

5.9 Design of steering wheel: -

Ergonomics consideration for driver and as per the rule outer dimensions of steering wheels is decided.

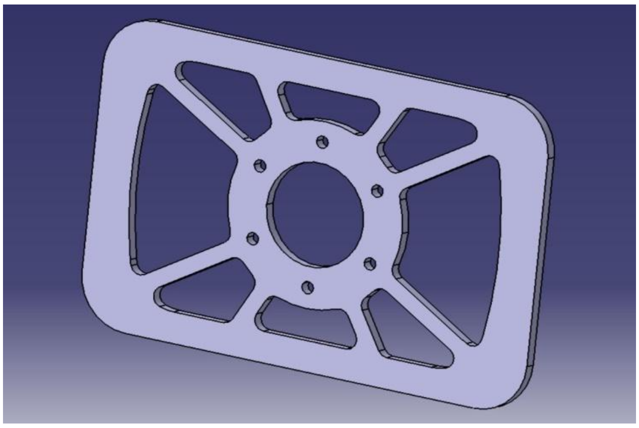

Fig -4: Steering Wheel CAD Design

5.10 Bearing calculations:

Fr = radial load

L10h = used bearing life

N = speed of rotation

Fr = 2000 N

L10h = 4000 hrs.

N = 750 rpm

Bearing life –

$$L10 = \frac{60*n*L10h}{10^6} \quad ....[1]$$

$$= \frac{60*750*4000}{10^6}$$

= 180 million revolution

Dynamic load capacity –

P = Fr = 2000N

$$C = P*(L10)^{1/3} \quad .... [1]$$

$$= 2000*(180)^{1/3}$$

= 11292.43 N

6. CAD Assembly of Steering System: -

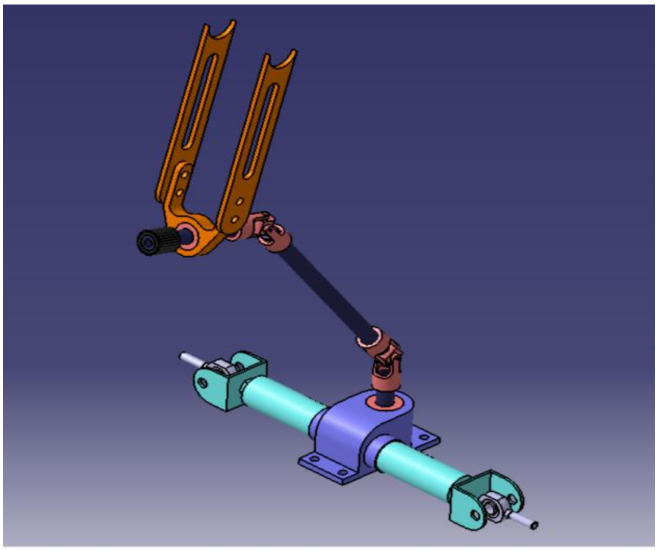

Fig -5: Steering System Assembly CAD Design

7. Analysis of components: -

7.1 Tie rod analysis:-

- Tie rod loading [6]

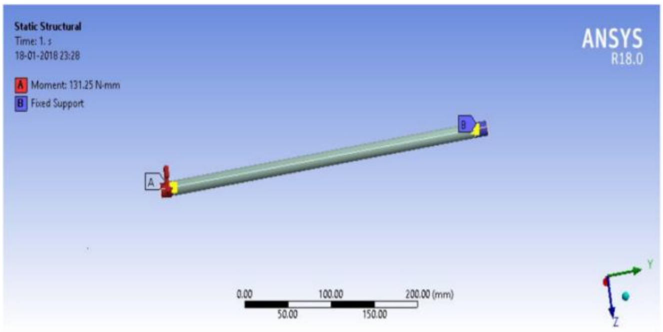

Fig -6: Structural analysis of tie rod

7.2 Tie rod analysis:-

- Tie rod total deformation [6]

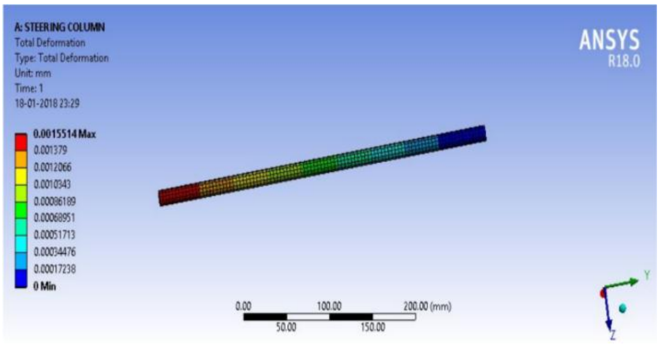

Fig -7: Tie rod total deformation

8. Testing and validation of steering system: -

For testing of steering system 2 methods are followed and they are: -

- Constant speed method.
- Constant radius methods.

Time required in both the methods are calculated by stopwatch and the results are plotted on graph.

8.1 Constant speed method[4]

Here speed of the vehicle is kept constant at 30 km/hr. and time required to complete different turning radii are calculated.

Chart -1: Constant Speed Method

8.2 Constant radius method [4]

Here turning radius is kept constant and time for different speed to complete the circle is calculated

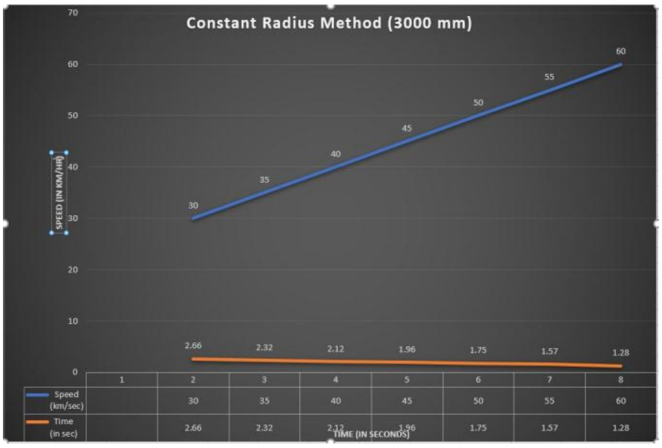

Chart -2: Constant Radius Method

9. CONCLUSIONS

While designing the vehicle the primary objective was to make the vehicle light, compact, ergonomic and safe for the driver. While designing the system all the conditions of a racing environment such as Speedy cornering, proper turn and steering response were considered. Steering system functions desirably and the resulting vehicle is safe, attractive, reliable, economical and fun to drive.

10. ACKNOWLEDGEMENT

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