DESIGN OF STEERING SYSTEM OF FORMULA STUDENT RACE CAR

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ABSTRACT
The 2016 iShell SAE INDIA team planned to design and fabricate a custom steering design that not only provided high accuracy and precision, but also met our budget norms. The initial design consisted of a near conventional rack and pinion design that included a large diameter pinion gear. However, that design implemented an unappreciable accuracy and cornering abilities. Moreover, that design required large steering force inputs from the driver.

A secondary design was made, that consisted of an extra idler gear between the pinion and the rack. Relatively small pinion gears were used in this new design as compared to the previous design, but still larger than the conventional systems. For this, many dual pinion steering mechanisms (including the ones of Alfa Romeo and Volkswagen) were studied. Most of them used either a hydraulic or an electrically actuated system. The advantage of our design over other dual pinion steering systems is that our design is a purely mechanical system.

On doing some further research and calculations, we determined the size and specifications of rack and pinion gears and a 3:1~4:1 steering system was ready at our disposal.

Keywords  
Rack and pinion, steering ratio, dual pinion mechanism, steering mechanism, steering system, SAE SUPRA steering mechanism, steering, working of steering system.

2. AIM AND OBJECTIVE
The aim was to design and fabricate a steering system for a race car that required less effort from driver for its working and took less revolutions from lock to lock. Accuracy, precision and cornering ability were also the major concerns.

The conventional rack and pinion system could not be used as its steering ratio is b/w 15:1-20:1, that means it takes 3 revolutions from lock to lock at an average. And its very difficult for the driver of race car to give more revolutions to steering wheel to achieve full steering, so we decided to reduce the steering ratio to 3:1-4:1.

3. DESIGN METHODOLOGY

4. CALCULATIONS
P - Power  
N - no. of revolutions  
T - Torque  
D - Diameter  
Fzl - Vertical Load, Left Wheel  
Fzr - Vertical Load, Right wheel  
d - Lateral Offset at the ground  
λ - Lateral Inclination Angle  
δ - Steer Angle  
v - Caster Angle  
Cf - Cornering Force  
Fyl - Lateral Force, Left Wheel  
Fyr - Lateral Force, Right Wheel

1. INTRODUCTION
As we are working on a national level project named SAE SUPRA in which the design and fabrication of a Formula student race car was to be done and for that we had the task to design a 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 road cars i.e we planned to reduce the steering ratio from 18:1(Conventional system) to 3:1 or 4:1.

The design consists of a conventional rack and pinion steering with an extra idler gear between them. The diameter of both the gears has been increased from the conventional size due to which, improved sensitivity has been achieved. However, the extra idler gear caused steering reversal, which was avoided by mounting the tie rods on the front part of upright rather than mounting it at the rear end which is usually done in cars now a days.
4.1 Power on steering column

\[ P = \frac{(2\pi NT)}{60} \]

\[ T = \text{Force} \times \text{Radius} \]

\[ T = 250 \times 0.018 = 4.5 \]

\[ P = \frac{(2 \times 3.14 \times 18 \times (250 \times 0.018))}{60} \]

\[ P = 8.478 \text{ watts} \]

4.2 Moment due to tractive force

\[ E_t - \text{Engine Torque} \]

\[ \eta - \text{Efficiency of Power train} \]

\[ N_g - \text{Transmission Ratio} \]

\[ N_a - \text{Driving Axle Ratio} / \text{Sprocket Ratio} \]

\[ R - \text{Tire Rolling Radius} \]

\[ \Phi - \text{Adhesion coefficient b/w tyre and road surface} \]

\[ W_r - \text{Vehicle weight component on the driving wheels} \]

\[ \text{TE} - \text{Applied tractive force based on engine torque} \]

\[ \text{TE max} - \text{Max. permissible limit of tractive force} \]

\[ F_{xl} - \text{Tractive Force, Left Wheel} \]

\[ F_{xr} - \text{Tractive Force, Right Wheel} \]

\[ d - \text{Lateral Offset at the Ground} \]

\[ \text{Depth of tire} = \frac{(\text{Aspect ratio/100}) \times \text{width of tire}}{50/100} = 175 \]

\[ \text{Outer radius of wheel} = \text{Depth of tire} + \text{Rim Radius} = 87.5 + 165 = 252.5 \]

\[ R = \text{Outer Radius of wheel} / 1000 = 0.2525 \]

\[ \text{TE} = \frac{(E_t \times \eta \times N_g \times N_a)}{R} \]

\[ \text{TE} = 2015 \text{ N} \]

\[ \text{TE max} = (\Phi \times W_r) \]

\[ = 0.7 \times 7 \]

\[ = 4.9 \text{ N} \]

\[ \text{Moment due to tractive force} = (F_{xl} - F_{xr}) \times d \]

\[ = (2015.1445 - 2015.1445) \times 25.4 \]

\[ = 0 \text{Nm} \]

4.3 Moment due to lateral force

\[ C_f = \text{Cornering force} \]

\[ \text{W} = \text{Weight of vehicle} \]

\[ \text{ML} = \text{Moment due to lateral force} \]

\[ \text{L} = \text{Lateral Force} \]

\[ m = \text{Mass of vehicle} \]

\[ W = \text{m} \times 9.8 = 300 \times 9.8 = 2943 \]

\[ C_f = \frac{\text{TE}}{W} \]

\[ = \frac{2015.1445}{2943} \]

\[ = 0.6847 \text{ N} \]

\[ L = \frac{(C_f \times m)}{4} \]

\[ = \frac{(0.6847 \times 300)}{4} \]

\[ = 51.3543 \text{ N} \]

\[ M_L = - (F_{yl} + F_{yr}) \times r \times \tan(\delta) \]

\[ = - (513.543 + 513.543) \times 0.2525 \times \tan(5.7) \]

\[ = -25.88 \text{ Nm} \]

4.4 Moment due to vertical load

\[ F_{zl} - \text{Vertical Load, Left Wheel} \]

\[ F_{zr} - \text{Vertical Load, Right Wheel} \]

\[ D - \text{Lateral Offset at the ground} \]

\[ \lambda - \text{Lateral Inclination Angle} \]

\[ \delta - \text{Steer Angle} \]

\[ \text{MV} - \text{Moment due to vertical load} \]

\[ \text{MV} = - (F_{zl} + F_{zr}) \times d \times \sin(\lambda) \times \sin(\delta) + (F_{zl} - F_{zr}) \times d \times \sin(\delta) \times \cos(\delta) \]

\[ \text{MV} = - (105 + 105) \times 25.4 \times \sin(15) \times \sin(30) + 0 \]

\[ = -690.20 \text{ Nm} \]

These three factors (tractive force, lateral force, vertical force) affect the steering system. All these conditions produce a moment acting about the steer axis, and the moment is balanced through the steering rack. The total moment about the steer axis is a summation of the individual moments.

Hence, total moment about steer axis (M):

\[ \text{Moment due to tractive force + moment due to lateral force + moment due to vertical force} \]

\[ M = 0 + 17.1094 + (-788.82) \]

\[ = -771.710 \text{ Nm} \]

5. MAIN PARTS & ANALYSIS

<table>
<thead>
<tr>
<th>Parts</th>
<th>Material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Main Support Tube</td>
<td>Al-6061</td>
</tr>
<tr>
<td>Pinion Housing</td>
<td>Al-6061</td>
</tr>
<tr>
<td>Pinion Housing Cap</td>
<td>Al-6061</td>
</tr>
<tr>
<td>Rack Housing</td>
<td>Al-6061</td>
</tr>
<tr>
<td>Tie Rod End Bungs</td>
<td>Al-6061</td>
</tr>
<tr>
<td>Rack</td>
<td>Steel 8620</td>
</tr>
<tr>
<td>Gears</td>
<td>Bronze</td>
</tr>
<tr>
<td>Bronze Bushings</td>
<td>Steel 8620</td>
</tr>
</tbody>
</table>

6. DESIGN

Wheelbase(b)=1650mm

Rear Track Width(a)=1250mm

Front Track Width(c)=1200mm

Turning radius(R)=4000mm
Inner Front Wheel
Inner Real Wheel
\[ R = \frac{b}{\sin \theta} - \left[ \frac{a-c}{2} \right] \]
\[ R = \frac{b}{\tan \theta} - \left[ \frac{a-c}{2} \right] \]
\[ 4000 = 1650 / \sin \theta - \left[ \frac{1250-1200}{2} \right] \]
\[ 4000 = 1650 / \tan \theta - \left[ \frac{1250-1200}{2} \right] \]
\[ \theta = 24.2^{\circ} \text{Degree} \]
\[ \theta = 22.28^{\circ} \text{Degree} \]

Outer Front Wheel
Outer Inner Wheel
\[ R = \frac{b}{\sin \phi} + \left[ \frac{a-c}{2} \right] \]
\[ R = \frac{b}{\tan \phi} + \left[ \frac{a-c}{2} \right] \]
\[ 4000 = 1650 / \sin \phi + \left[ \frac{1250-1200}{2} \right] \]
\[ 4000 = 1650 / \tan \phi + \left[ \frac{1250-1200}{2} \right] \]
\[ \phi = 24.57^{\circ} \text{Degree} \]
\[ \phi = 22.53^{\circ} \text{Degree} \]

6.1 Steer Angle
\[ \theta = \frac{R}{L} = \text{Wheelbase} / \text{Radius of turn} \]
\[ = \frac{1645}{4000} \]
\[ = 23.57 \text{ degree} \]

6.2 Steering Ratio
The steering ratio is the ratio of how much the steering wheel turns in degrees to how much the wheel turns in degrees. Approximating maximum turn to be of 30 degrees and steering wheel movement to be 180 degrees the steering ratio can be calculated as
\[ \text{S.R} = \frac{180}{30} \]
\[ = 6 \]

6.3 Rack Travel
The steering ratio has been calculated the rack travel needs to be decided.

The steering wheel travel for one complete rotation
\[ = 2\pi \times r \quad (r = \text{radius of wheel}) \]
\[ = 0.816 \text{m} \]

Considering maximum steer angle and max rack travel is reached at complete rotation of the steering wheel.

The Steering ratio can be equated to Steering wheel travel to the Rack Travel

Steering Ratio=Steering Wheel Travel/ Rack Travel
\[ 6 = 0.816 / \text{Rack Travel} \]
Rack Travel=0.136m or 5.35inch

\[ G = \text{Gear ratio} = \frac{T}{t} = \text{Number of teeth on the pinion, } T = \text{Number of teeth on the wheel} \]

Gear Ratio=1:3

No of teeth on Pinion(T₁) can be calculated

\[ t = \frac{2 A_w}{G \left[ 1 + \left( \frac{1}{G} \left( \frac{1}{2} \right) \sin^2 \phi - 1 \right) \right]} \]

No of teeth on wheel or Rack(T₂) can be calculated

\[ t = \frac{2 \times A_w}{\sqrt{1 + 3 \sin^2 \phi - 1}} \]

\[ P_c = \pi D/T \]
\[ m(\text{module}) = D/T \]

The Values of module(m) generally considered as 1,1.25,1.5,2,2.5,3,4,5,6,8,10,12,16 and 20 in Indian Standards.

We decided G=1:3, so the value of Adendum(A₁), Dedendum(D₁) and Pressure Angle(φ) Consider According to Indian Standards

\[ A_1 = 1 \text{ module} \]
\[ D_1 = 1.25 \text{ module} \]
\[ \phi = 20 \text{ Degree} \]

Sometimes, the spur gears (i.e. driver and driven) are to be designed for the given velocity ratio and distance between the centres of their shafts.

Let \[ x = \text{Distance between the centres of two shafts} \]
\[ N_1 = \text{Speed of the driver,} \]
\[ T_1 = \text{Number of teeth on the driver,} \]
\[ d_1 = \text{Pitch circle diameter of the driver,} \]
\[ N_2, T_2 \text{ and } d_2 = \text{Corresponding values for the driven or follower, and} \]
\[ p_c = \text{Circular pitch.} \]

We know that the distance between the centres of two shafts
\[ x = d_1 + d_2 \]

Speed ratio or velocity ratio can be calculated as
\[ N_1 / N_2 = T_2 / T_1 = d_2 / d_1 \]
\[ T_1 = 12 \text{teeths} \]
\[ T_2 = 36 \text{teeths} \]
\[ T_2 / T_1 = d_2 / d_1 = 36 / 12 \]
\[ = 3 / 1 \]
\[ d_2 = 3 d_1 \]
\[ x = d_1 + d_2 / 2 \quad (x = 86 \text{inch or } 21.77 \text{mm}) \]

By Solving we get,
\[ d_1 = 10.92 \text{mm} \]
\[ d_2 = 31.76 \text{mm} \]

Now we calculate Circular pitch(Pᵢ)
\[ P_i = \pi d_2 / T_2 \]
\[ = 3.14 \times 31.76 / 36 \]
\[ = 2.77 \text{mm} \]
Diametral Pitch \(P_d = \frac{\pi}{P_c}\)

\[= \frac{3.14}{2.77}\]

\[= 1.13\text{mm}\]

\(m\text{(module)} = \frac{D}{T}\)

\[= \frac{.90 ~ 1}{7.}\]

7. ASSEMBLY OF STEERING ASSEMBLY

- **7.1 Rack and Pinion**

7.2 **Akerman%**

8. CONCLUSION

After all the calculations were completed and analysis in LOTUS Shark suspension analyzer was conducted the final steering assembly was designed in Solidworks. The above picture shows the final design incorporated into the chassis of the SAE car. This steering system designed for the turns generally encountered in the SAE events was optimal to counter negative impacts of bump and roll steer and also possessed self-returning capability. Universal joints have been added in the steering column to line it up nicely with the pinion shaft. It also provides for the columns to fold up in the event of a hard frontal collision preventing it from being forced into the cockpit and injuring the driver.

9. ACKNOWLEDGMENTS

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10. REFERENCES


[4] https://www.youtube.com/watch?v=vzRNlg9rrt0
