Steering System Report

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1 Introduction

The complete prototype of steering system and its assembly is done in SolidWorks. To get that enhancement of steering response we majorly concentrated on Ackermann principle which is a primary consideration of a vehicle, based on it the vehicle behavior depends .We are designing the steering geometry for formula cars and it's components with a specified turning radius and wheel turning angles, the system could be optimized for a low steering effort.

This design overcomes the major issue of most of the formula student cars during endurance events i.e drivers fatigue. While the lowest steering effort possible would be ideal, a compromise would have to be determined since a low steering effort also gave a slow steering response and therefore a large steering wheel angle at full lock. To allow for a fast steering response with a reasonable steering wheel angle, the full lock position was set to 360 degrees from lock to lock position. This compromise allowed for the uprights to incorporate the necessary moment arm to achieve the maximum wheel angle based upon the selected steering rack.

2 Design Considerations

2.1 Anti-Ackermann Geometry

Although Ackerman steering is efficient at low speeds it is not effective at higher speeds. This system incorporates a lower outer wheel angles and higher inner wheel angle. With increase in lateral forces there is a progressive increase in deflection i.e., the vehicle moves in a direction different to which it is actually pointing. At higher speeds lateral forces and vertical loads are significantly higher on the outer wheel causing a greater slip angle due to which Ackerman system does not perform well as the inner wheel is subjected to higher slip angles causing increase in temperature and also produces a drag The additional reasons for choosing Anti Ackermann are as follows:

- The rack and pinion arrangement is placed ahead of the front axle and likewise the steering arm points towards the front of the vehicle the outer wheel makes a small radius than the inner wheel while taking a turn which is opposite to that of Ackerman steering.

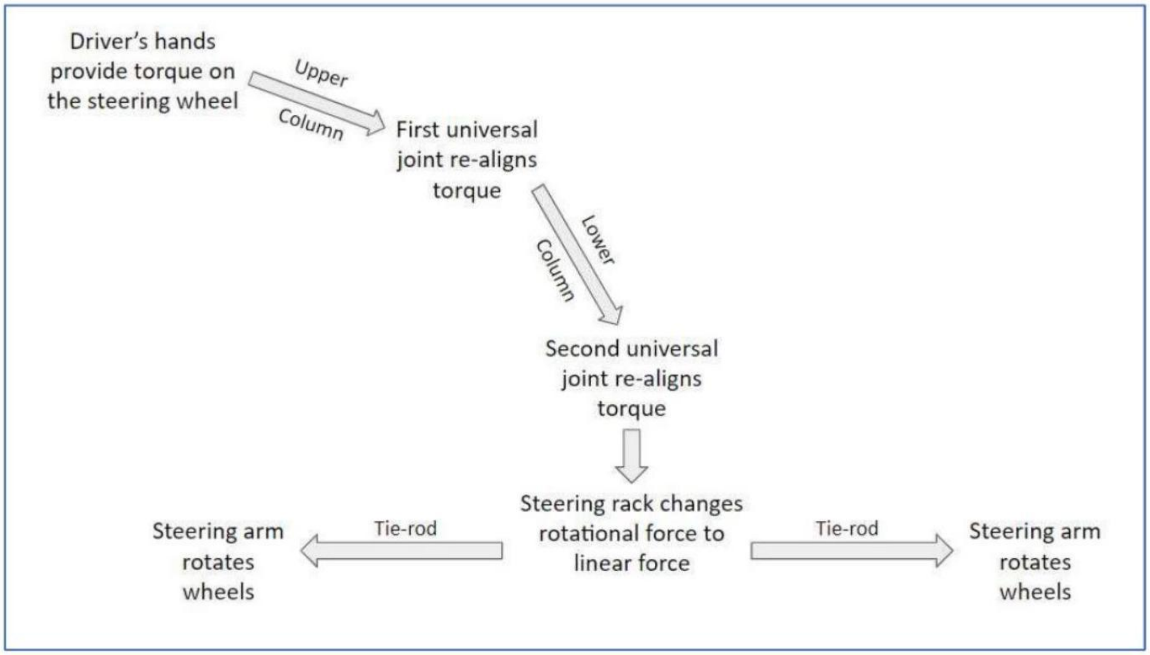


Figure 1: Steering Force Flowchart

- To get a better steering response by reducing slip and providing more grip to the outer tire during cornering at larger radius turn.

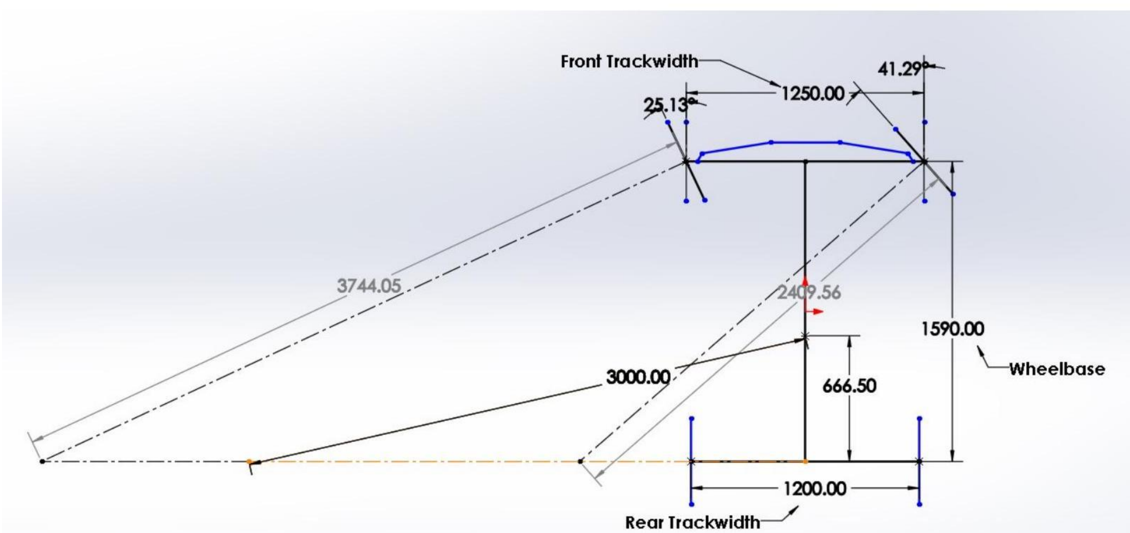


Figure 2: Anti-Ackermann Steering Geometry

2.2 Rack and Pinion Design

In rack and pinion steering mechanisms, the steering wheel turns the pinion gear; the pinion moves the rack, which is a linear gear that meshes with the pinion, converting circular motion into linear motion along the transverse axis of the car (side to side motion). This motion applies steering torque to the kingpin of the steered wheels via tie rod and a short lever arm called the steering arm. The rack and pinion design has the advantages of a large degree of feedback and direct steering "feel"; it also does not normally have any backlash or slack. That's why this system is racing designer first choice. A disadvantage is that it is not adjustable, so that

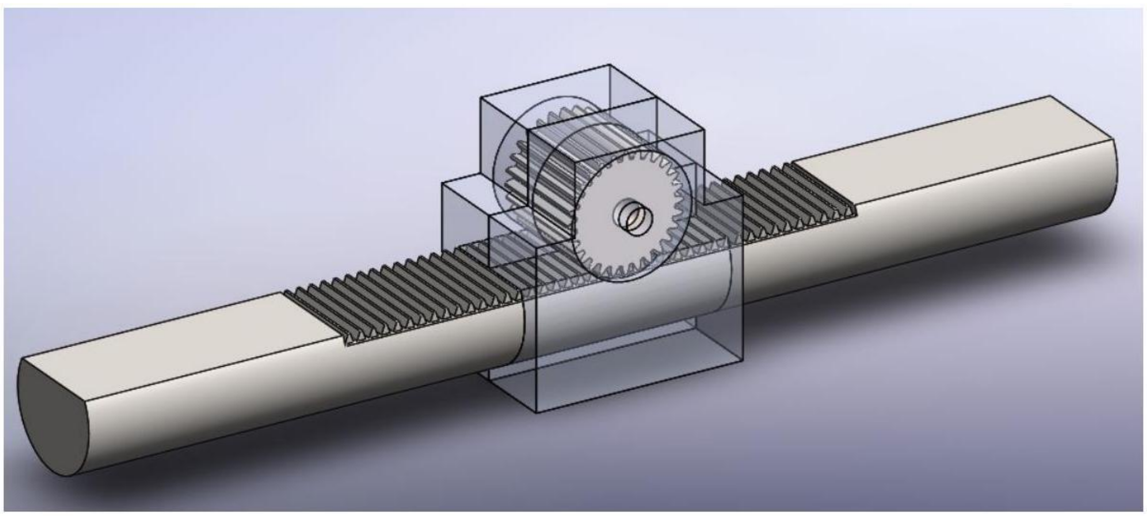


Figure 3: Rack and Pinion Assembly

when it does wear and develop lash, the only cure is replacement. Some more advantages of this system are as follows:

- Economical and Simple to manufacture
- Easy to operate due to good degree of efficiency
- Contact between steering rack and pinion is free of play and even internal damping is maintained
- The idler arm (including bearing) and the intermediate rod are no longer needed
- Tie rods can be joined directly to the steering rack
- Easy to limit steering rack travel and therefore the steering angle.

2.3 Steering Wheel

An ergonomically designed steering wheel of 9 cm diameter is made of laser-cut aluminum to reduce the steering effort of the driver. As a requirement of the design for the competition, the steering wheel must be equipped with a quick-release mechanism. Following discussion with the team it was decided to make use of the last year quick release mechanism.

Figure 4: Steering Wheel Assembly

3 Calculations

3.1 Anti-Ackermann Calculations

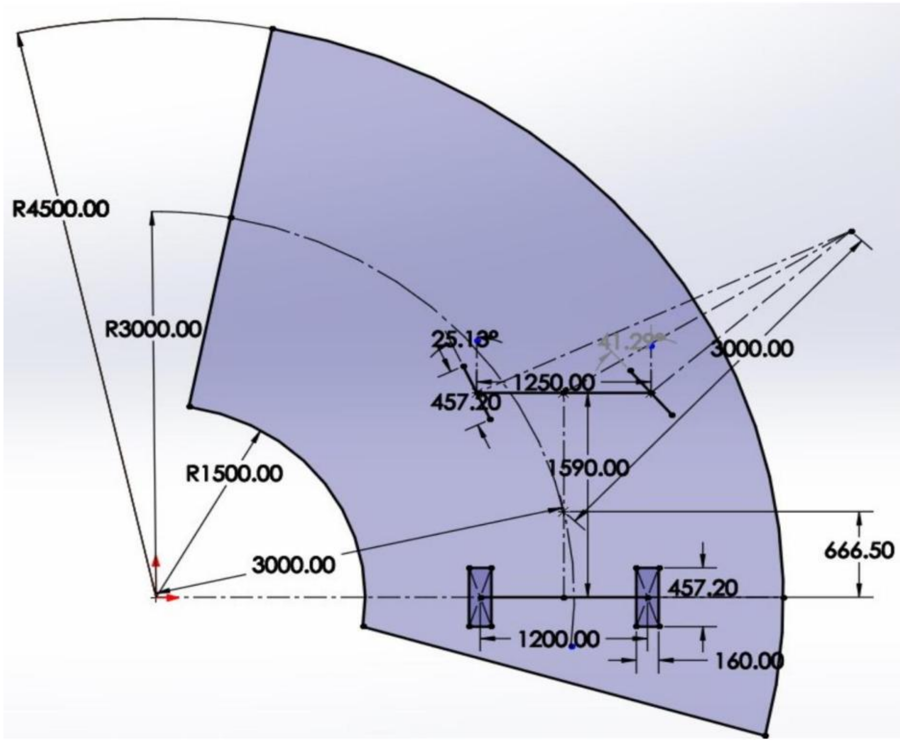


Figure 5: Turning Radius Geometry for Anti-Ackermann Condition

The sketch for Anti-Ackermann Geometry was drawn for the car in SOLIDWORKS keeping in mind the rules of FSEV 2024. Keeping the distance of CG from IC as 3000 mm (turning radius), we got the inner and outer wheel angle as

$$\theta_i = 25.13^\circ \ \theta_o = 41.29^\circ$$

The design specifications are listed below:

Sl. No	Parameters	Values
1	Wheelbase	1590 mm
2	Front Trackwidth	1250 mm
3	Rear Trackwidth	1200 mm
4	Tire Width	160 mm
5	Tire Radius	9 inches
6	Track Turning Radius	3000 mm

Table 1: Design Specifications

3.2 Ackermann Angle

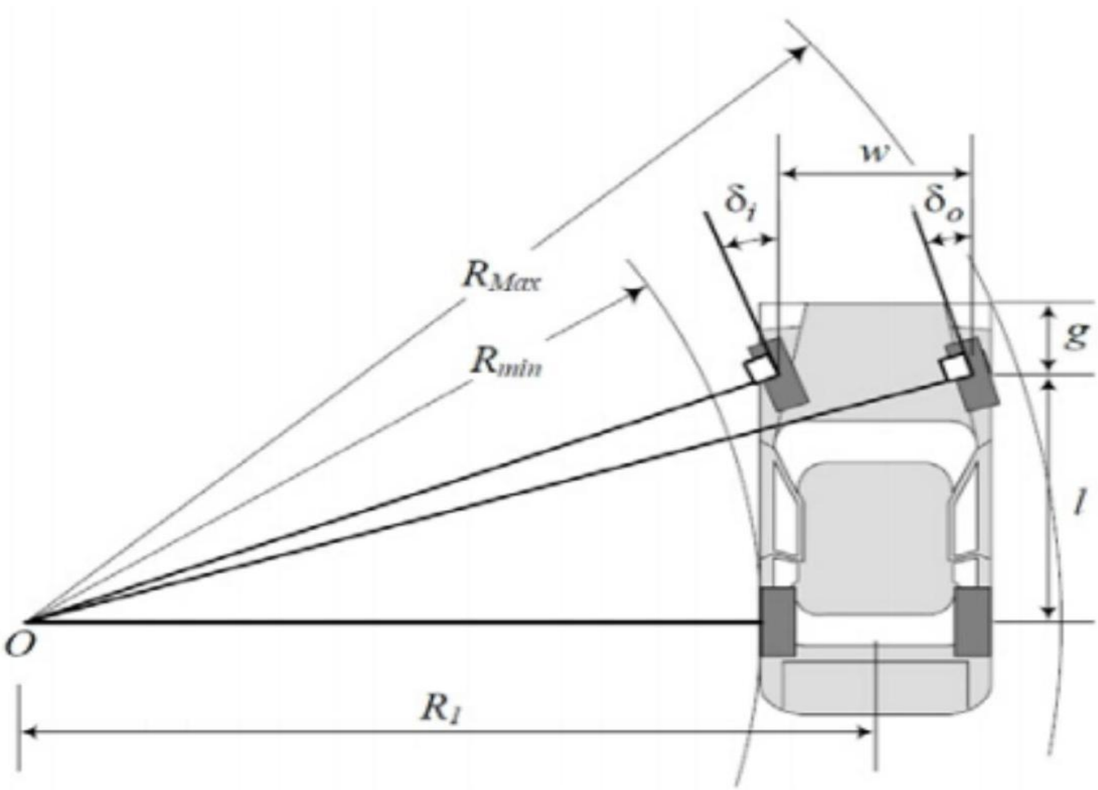
Let ϕ be the Ackerman angle, then using the formula

$$\tan\ \phi = \frac{\text{kingpin pivot } - \text{ point to pivot } - \text{ point distance}}{2 \cdot \text{ wheelbase}}$$

$$\tan\ \phi = \frac{1147.70}{2 \cdot 1590} = 0.3609$$

$$\phi = \mathbf{19.84}\text{deg}$$

3.3 Space Required for Turning



The space required for turning is the space between the two circles in which the whole vehicle fits without going out of the circle. Space required for turning is given by the formula:

$$\Delta R = R_{\max} - R_{\min}$$

$R_{\max}$ and $R_{\min}$ are obtained from the formulas below:

$$R_{\max} = \sqrt{(R_{\min} + w)^2 + (l + g)^2}$$

where g is the distance from front axle to nose

$$R_{\min} = \frac{l}{\tan\ \theta_i} = \frac{1590}{\tan\ 25.13°} = 3389.66 \text{ mm}$$

Therefore,

$$R_{\max} = \sqrt{(3389.66 + 1250)^2 + (1590 + 600)^2} = 5130.55 \text{ mm}$$

Therefore Approx width or spacing = Rmax −Rmin = 1740.89 mm ≈ 1.75 m

3.4 Turning Radius

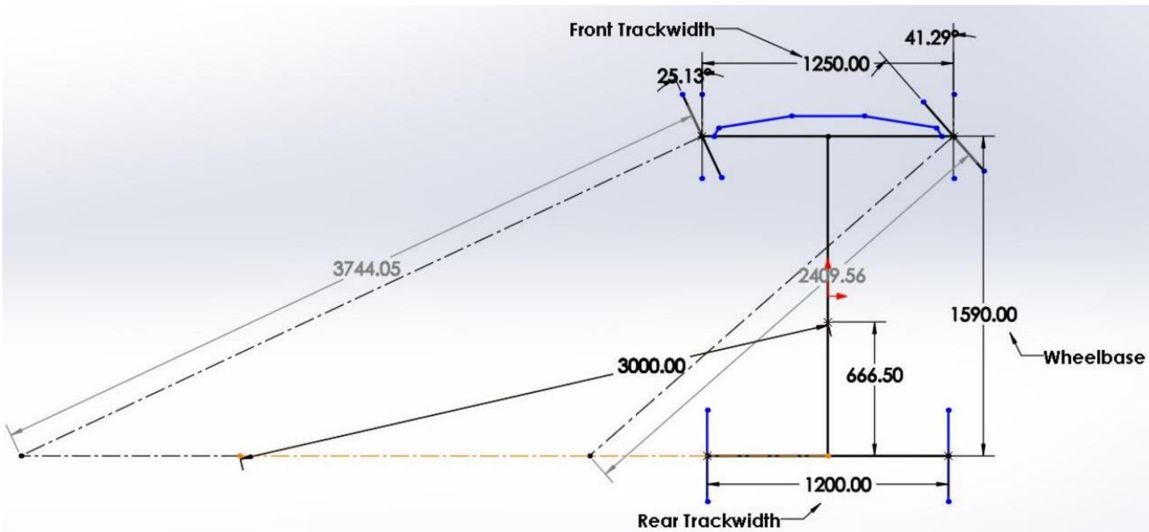


Figure 6: Top View of steering geometry

Inner Front Turning Radius, $R_{if} = \left(\frac{\text{Wheelbase}}{\sin\ \theta_i}\right)$

Outer Front Turning Radius, $R_{of} = \left(\frac{\text{Wheelbase}}{\sin\ \theta_o}\right)$

Inner Rear Turning Radius, $R_{ir} = \left(\frac{\text{Wheelbase}}{\tan\ \theta_i}\right)$

Outer Rear Turning Radius, $R_{or} = \left(\frac{\text{Wheelbase}}{\tan\ \theta_o}\right)$

$$R_{if} = \left(\frac{1590}{\sin\ 25.13}\right) = 3744.05 \text{ mm}$$
$$R_{of} = \left(\frac{1590}{\sin\ 41.29}\right) = 2409.56 \text{ mm}$$
$$R_{ir} = \left(\frac{1590}{\tan\ 25.13}\right) = 3389.67 \text{ mm}$$
$$R_{or} = \left(\frac{1590}{\tan\ 41.29}\right) = 1810.50 \text{ mm}$$

3.5 Tie Rod Calculation

- x = steering arm length
- y = tie-rod length
- z = rack ball joint center to ball joint length
- d = distance between front axis and rack center axis
- B = distance between left and right kingpin centerline
- β = Ackerman angle

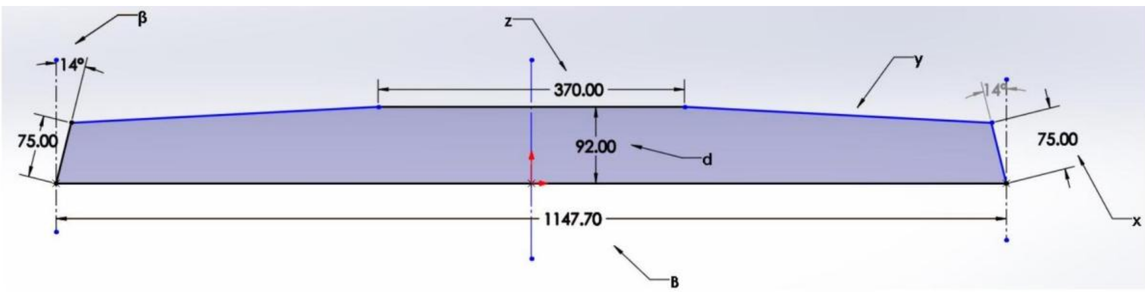


Figure 7: 2D sketch of steering rack assembly in top view

From the above figure, we get

$$y = \sqrt{\left[\frac{B-z}{2} - x\sin\,\beta\right]^2 + [d - x\cos\,\beta]^2}$$

The tie-rod length came out to be approximately 371.20 mm .

3.6 Rack Travel

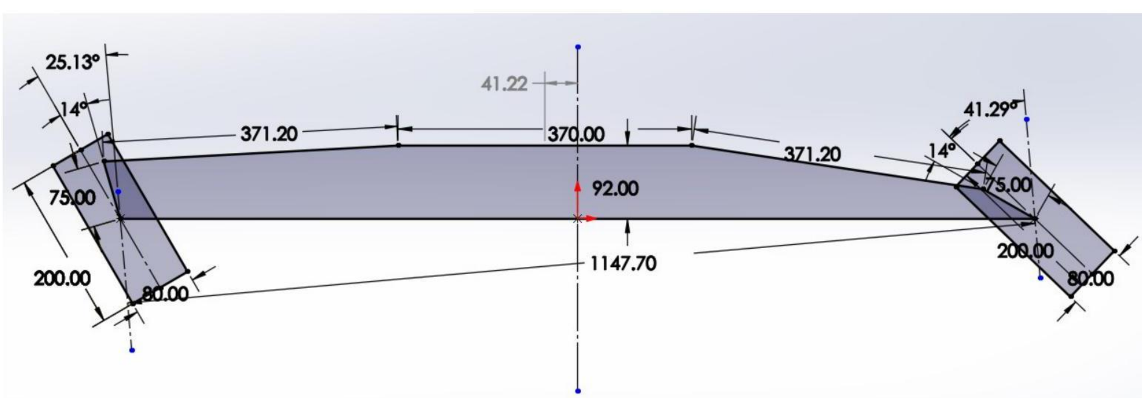


Figure 8: Rack Travel for maximum turn angles

Rack travel δ for maximum wheel turn angle was found out in SOLIDWORKS:

$$\boldsymbol{\delta = 41.22mm}$$

3.7 Steering Ratio

The steering ratio is calculated as

$$\text{Steering Ratio } = \frac{\text{Angle of steering wheel}}{\text{Angle of turning on the left } + \text{ right wheels}} = \frac{180}{25.13 + 41.29} = \mathbf{3.71}$$

3.8 C-factor

C-factor is the linear distance travelled by the rack in one rotation of the pinion.

$$C - \text{ factor } = \frac{\text{Total Rack Travel (in mm)}}{\text{Lock } - \text{ to } - \text{ lock angle (in rad)}}$$

where

$$\text{Lock } - \text{ to } - \text{ lock angle } = \theta_i + \theta_o = 25.13 + 41.29 = 66.42° = 1.16\text{rad}$$

Therefore

$$\text{C-factor } = \frac{2 \cdot (41.22)}{(1.16)} = \mathbf{71mm}$$

3.9 Pinion Calculation

It is assumed to get maximum rack travel at 180 steering wheel rotation. The maximum rack travel being 41.22 mm , the pitch diameter of the pinion can be found by the formula of the length of an arc.
The rotation angle of the pinion will not exactly match the steering wheel angle due to play in the universal joints and steering column. So, assuming an efficiency of 90%, we calculate

Arc length is,

$$s = r\omega$$
$$41.22 = r \cdot (0.9) \cdot \left(\frac{180\pi}{180}\right)$$

Therefore,

$$r = \frac{41.22}{0.9 \cdot \pi} = 14.58 \text{ mm}$$

So, PCD of the pinion is $2r = 29.16 \approx 30$ mm
Module (M) $= 2$
PCD $= 30$ mm

No.of teeth on pinion $(T_p) = \frac{PCD}{M} = \mathbf{15}$
Circular Pitch $= \pi \cdot M = 6.28$ mm

3.10 Rack Calculation

Rack travel for both side $= 2 \cdot \delta = 82.44mm$
For perfect meshing of rack and pinion, module should be same for both.

Therefore, Module(M)=2
Axial pitch of rack $= \pi \cdot M = 6.28$ mm
No.of teeth on rack $= \frac{2 \cdot \delta}{\pi M} \approx 15$ (considering some amount of clearance)
Therefore, the actual length of rack $(L_r) = T_r \cdot \pi \cdot M = \mathbf{94.25mm}$
[Note: This is actually the minimum length of rack required]

3.11 Steering Effort Calculation

Mass of the car, M = 310 kg
Weight distribution is 45% (front) and 55% (rear)
The force acting on any of the front wheels is

$$W = \frac{1}{2} \cdot \frac{45}{100} \cdot 310 \cdot 9.81 = 684.25N$$

Considering the coefficient of friction between tyre and road as $\mu = 1.4$, Frictional Force is

$$F_r = \mu \cdot W = 957.95N$$

Considering the lateral coefficient of friction between tyre and road as $\mu_l = 2.1$, Lateral Force is

$$F_y = \mu_l \cdot W = 1436.92N$$

Resisting/Traction Torque on the wheel

$$T_R = F_r \cdot \text{ScrubRadius}(S_r) = 957.95 \cdot 30 = 28738.50\text{Nmm}$$

Lateral Torque on the wheel

$$T_L = F_y \cdot \text{ Mechanical Trail } (M_r) = 1436.92 \cdot 9.22 = 13254.50\text{Nmm}$$

Total Torque on the wheel

$$T = T_L + T_R = 41993\text{Nmm}$$

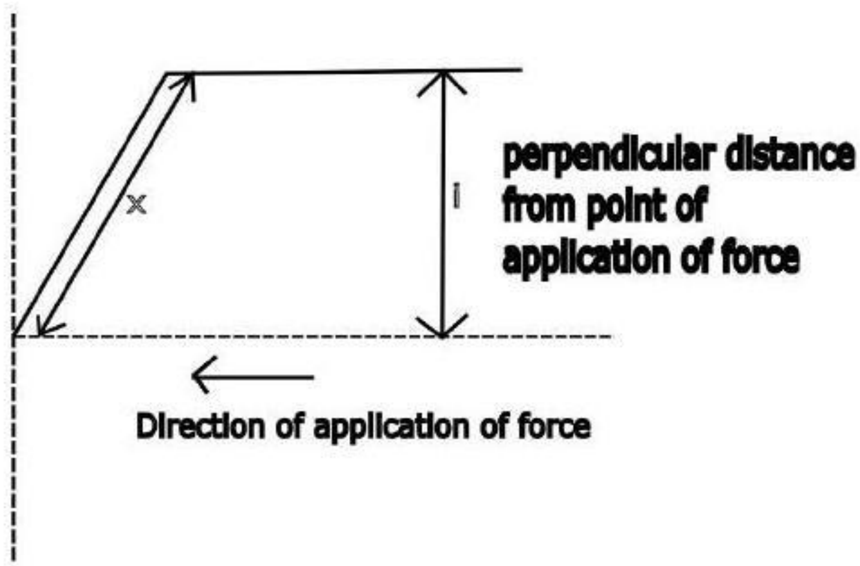
Let the force applied on the steering arm by the tie-rod be P. Considering i as the perpendicular distance measured from the point of application of force to the front axis, we get

Therefore,

$$P = \frac{T}{i} = \frac{41993\text{Nmm}}{92\text{ mm}} = 456.44\text{ N}$$

From Suspension geometry tie rod inclination α was found to be 9.64°. Therefore,

$$\begin{aligned}\text{Force on Rack } &= (\text{ Forces on tie rod }) \cdot (\cos\ \alpha) \\ &= 456.44 \cdot \cos(9.64) = 450\text{ N} \\ \text{Torque on Pinion } &= (\text{ Force on rack }) \cdot (\text{ pinion radius }) \\ &= 450 * 15 = 6750\text{ N} - \text{mm}\end{aligned}$$



The torque in the pinion will not be the same as the torque in the steering wheel or the torque required to turn the steering wheel due to play in the universal joints. Hence, assuming an efficiency of $\eta = 90\%$, we get the minimum required torque at the steering wheel as

$$T_{S_{\min}} = \frac{\text{Torque on pinion}}{\eta} = \frac{6750\text{Nmm}}{0.9} = 7500\text{Nmm}$$

$$\text{Steering effort } = \frac{\text{Torque on steering wheel}}{\text{Steering wheel radius}}$$

Substituting values,

$$\text{Steering effort } = \frac{7.5}{0.5} = 15\text{ N}$$

3.12 CAD Model of Steering Assembly

The whole steering wheel assembly was made in SOLIDWORKS. The rendered version of the same is shown below.

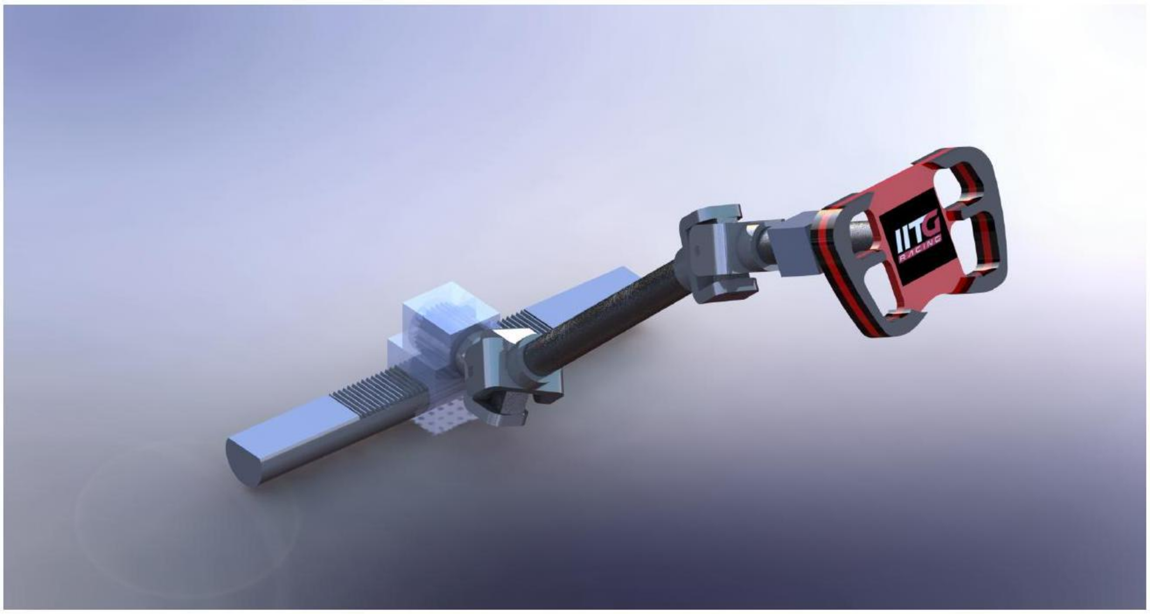


Figure 9: Steering Wheel Assembly