Steering System for "SAE SUPRA" Vehicle

Shubham Kalode[1], Kunal Patil[2], Abhijeet Bhavsar[3], Aniket Kamble[4]

[1]*Department of Mechanical Engineering, Padmabhooshan Vasantdada Patil Institute of Technology, Bavdhan, Pune, Maharashtra, 411021, India*

[2] *Department of Mechanical Engineering, Padmabhooshan Vasantdada Patil Institute of Technology, Bavdhan, Pune, Maharashtra, 411021, India*

[3] *Department of Mechanical Engineering, Padmabhooshan Vasantdada Patil Institute of Technology, Bavdhan, Pune, Maharashtra, 411021, India*

[4]*Department of Mechanical Engineering, Padmabhooshan Vasantdada Patil Institute of Technology, Bavdhan, Pune, Maharashtra, 411021, India*

***Abstract*- In this paper, a steering system is designed for SAE SUPRA car, which adopts a rack-and-pinion steering mechanism. The theoretical modeling of the System as well as the derivations of optimal Parameter values is presented here. First, the Steering angles of the front wheels are derived based on the geometry of the steering system. Second, linear equations representing the axial lines of the front wheels are derived based on the steering angles of the front wheels. Design and calculations of each component of Steering System are also presented.**

***Index Terms*- Ackerman, Casing, Geometry, Rack and Pinion, Steering System, SAE.**

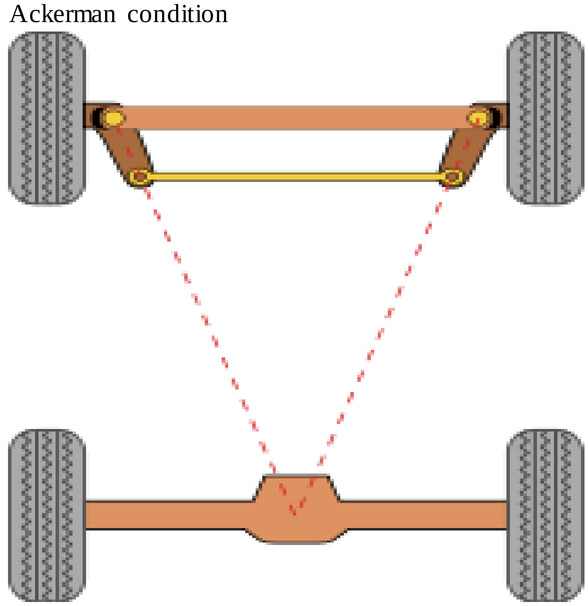
1. INTRODUCTION

All open wheeled racing oriented vehicles have to behave positively in tight turns, long straights and speedy cornering. This design paper focuses on explaining engineering and design process behind steering system in the F1 type racing vehicle. The design of the concept of this vehicle is in accordance with the specifications laid down by the rule book.

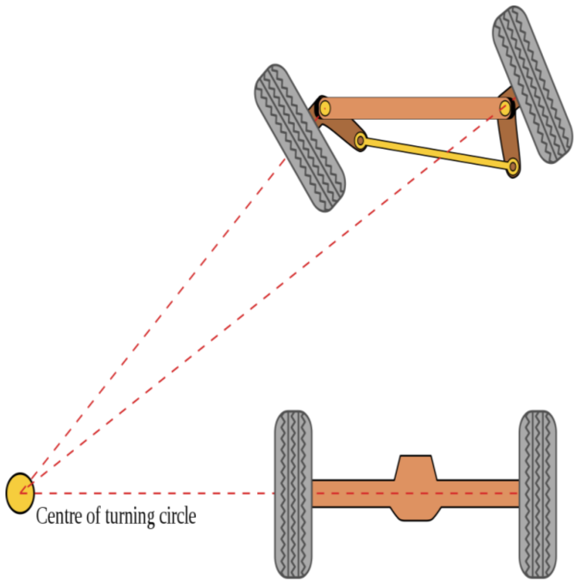
2. ACKERMAN GEOMETRY

The main purpose to use Ackerman is to avoid skidding of front tyres while turning. It is undesirable if the tyre drags while turning. Ackerman geometry helps in avoiding the skidding. It turns the front tyres optimally by allowing both tyres to turn by different angles. That is, inner tyre turns by greater angle as compared to outer angle. The geometry in which, inner angle turns by greater extent as compared to outer tyre is termed as Ackerman geometry.

Basically, Ackerman geometry comprises of a trapezoid. The idea behind Ackerman is to draw lines from each of the steering arms such that they meet at centre of the rear axis. That, is the steering arms are inclined by such an angle that when we draw lines from each of the steering arm towards rear axis, they will meet at the centre of the rear axis. This condition where rack position is behind the front axis is called Ackerman condition



When the vehicle turns, the outer tyre has to turn less than the inner tyre to avoid drag and give good feeling to the driver while cornering at high speed. While turning, the geometry looks like the figure mentioned below.



3. DESIGN PROCEDURE

If the steering wheel angle for full turn of tyres on one side is set minimal then the efforts required to steer the steering wheel is quite high as it gets hard to steer. We did not want huge efforts required to steer and also not wanted to increase the steering wheel angle for full turn on one side. So, the basic idea was to have an optimal and efficient steering wheel rotation angle.

In order to minimize the bump steer effect the height of the rack with respect to chassis and the length of the tie rod are derived from the suspension geometry with upper and lower wishbone points and their ICR geometry.

Team already decided the track widths, wheelbase, front pivoted distance and maximum inner angle required for the vehicle. The perimeters are as follows:

Front Track width (a) = 1325.02mm

Wheel base (l) = 1560mm

Pivoted distance (b) = 1145mm

Max turning angle, inner wheel (θ) =40°

4. DESIGN CALCULATIONS

The geometry validation was done by performing the analytical treatment. The concept of reverse engineering was applied in order to validate the designed geometry. The following calculations were performed in order to validate the geometry: required condition is:

Angle of inside lock (θ) = 40°

According to Ackerman's geometry, the condition for perfect steering is given by:

cot (θ) - cot (φ) = (b/l)

Hence rearranging and substituting values in above equation;

cot (φ) = -(b/l) + cot (θ)

cot (φ) = -(1145/1560) + cot (40)

I. (φ) = 26.43°

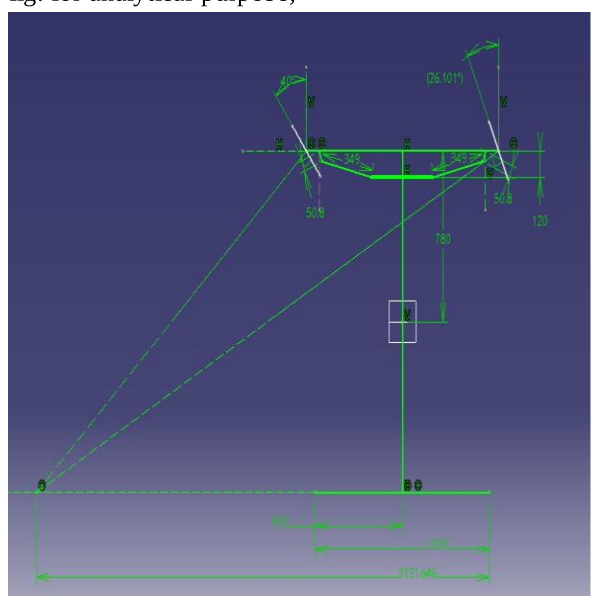
Also,

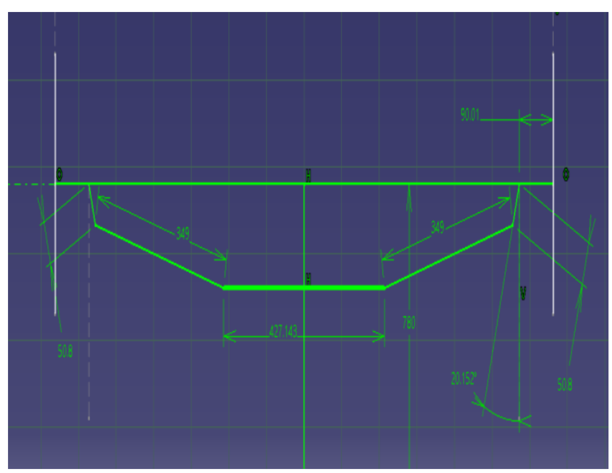
(α) = tan -1[(sin (φ) - sin (θ))/ (cos (φ) + cos (θ)-2)]

Putting values in above equation;

II. (α) = 20.730°

Referring Ackerman geometry shown in following fig. for analytical purpose;





5. TURNING RADIUS

I/sin (θ) = 1560/ sin (40)
III. (Rif) = 2426.92 mm

(Rof) = I/ sin (φ) = 1560/sin (26.43)

IV. (Rof) = 3504.79 mm

6. DESIGN OF MECHANISM

We are using rack and pinion mechanism because of obvious advantages of reduced complexity, ease of construction and less space requirement compare to other steering mechanisms. The analytical steps involved in designing rack and pinion system are as follow:

1) Selection of gear tooth profile
A gear tooth profile was selected based on BIS (Bureau of Indian Standard) recommendation and the manufacturability. A 20° full depth involute profile system was selected because of the following advantages:

- It reduces the risk of undercutting
- It reduces the interference Turning radius

Due to increase in pressure angle, the tooth becomes slightly broader at the root this makes the tooth stronger and increases its load carrying capacity it provides better length of contact
The properties of teeth of 20° full depth involute profile system are as follow:
TABLE
m= Module

Pressure angle (φ)	20°
Addendum (ha)	1m
Dedendum (hf)	1.25m
Clearance (c)	0.25m
Working depth	2m
Whole depth	2.25m
Tooth thickness	1.5708m

2) Minimum number of teeth on pinion
The minimum number of teeth required on pinion in order to avoid the interference was computed using following relation:

$$Zp = 2*ha/ (m*\sin^2 (\varphi))$$

Substituting values in above equation;

$$Zp = 2*m/ (m*\sin^2 (\varphi)) = 2/\sin^2 (\varphi)$$

Zp = 17.09

Hence number of teeth on pinion are 18.
Rack Length=427.143mm
Rack Travel= 31.4mm
The rack travel is 314mm and we wanted the steering wheel to turn 90 to achieve the maximum turning angle of tyre of 40. So, in short, 90° pinion rotation for 31.4mm rack travel on one side.
For 90° pinion, circumference for pinion is 31.4mm
So for 360°,
Circumference is 125.6mm
Addendum circle diameter=40mm
Module=2
Pitch circle diameter= Outside diameter-(2m)
PCD= 40-(2*2)
=36mm
Pitch is the distance between corresponding points on adjacent teeth.
p = Pi x Module = πm
p=2π
=6.28mm
Tooth thickness (s) is basically half the value of pitch (p).
Pitch (p) =πm

s=πm/2
S=3.14mm

The following are calculations of Tooth depth (h) / Addendum (ha) / Dedendum (hf) for gear with module2.
h = 2.25 m = 2.25 × 2 = 4.50mm
ha = 1.00 m = 1.00 × 2 = 2.00mm
hf = 1.25 m = 1.25 × 2 = 2.50mm
The size of gears is determined in accordance with the reference diameter (d) and determined by these

other factors; the base circle, Pitch, Tooth Thickness, Tooth Depth, Addendum and Dedendum.

Reference diameter (d)
$d = zm$

Tip diameter (da)
$da = d + 2\ m$

Root diameter (df)
$df = d - 2.5\ m$

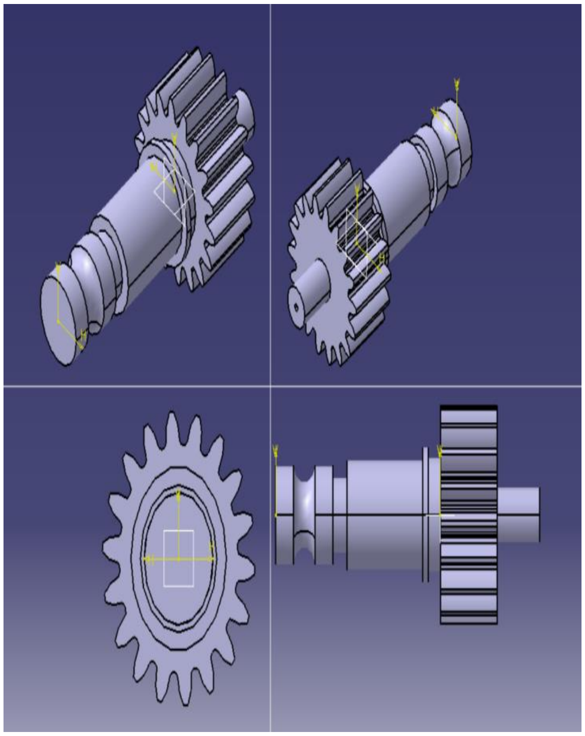
The following are calculations of Reference diameter / Tip diameter / Root diameter for a spur gear with module (m) 2, and 18 teeth (z).

$d = z\ m = 18x\ 2 = 36mm$
$da = d + 2\ m = 36 + 4 = 40mm$
$df = d – 2.5\ m = 36 – 5 = 31mm$

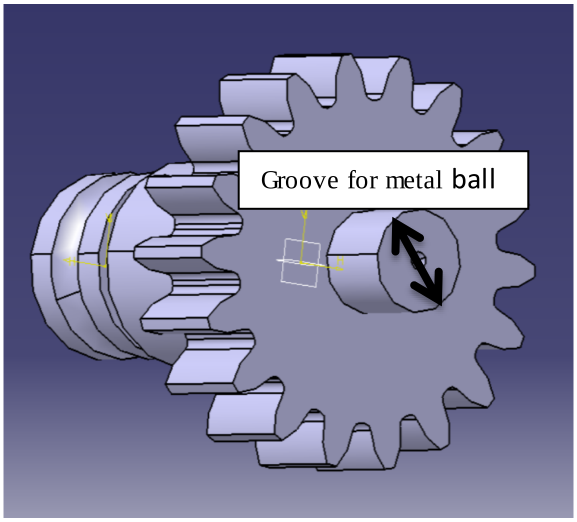
All parameters such as pressure angle, pitch, depth of tooth, etc. are same as that of pinion.

Rack needed at least 10 teeth, 5 teeth for each lock to lock turn. We designed our rack for 15 teeth to be on the safer side.

7. DESIGN OF PINION



Pinion with 18 teeth is shown in the above figure. Pinion and shaft is made one piece. Shaft on the pinion is splined to ensure good grip in between pinion and universal joint.

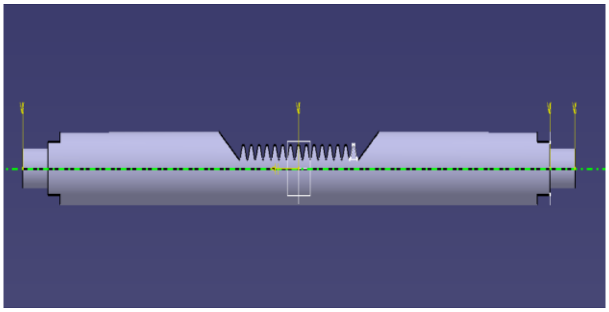


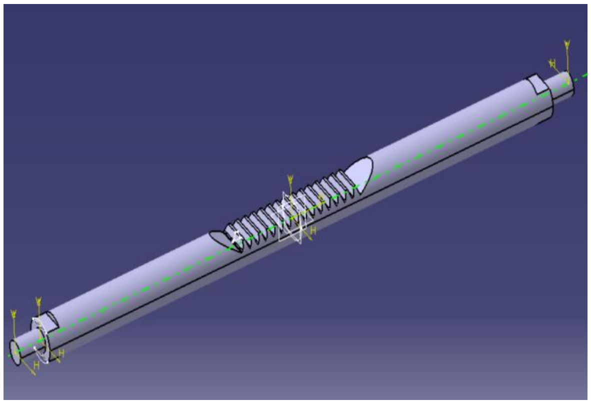
The extended portion which is shaft like is grooved at the centre for a steel ball to allow roll as shown in the fig which is to ensure minimum friction and to avoid wear and tear of the pinion.

A needle bearing was provided for the extended portion of pinion to rotate smoothly inside the casing without causing friction and wear and tear in the casing.

Pinion material: Stainless steel 303

8. DESIGN OF RACK

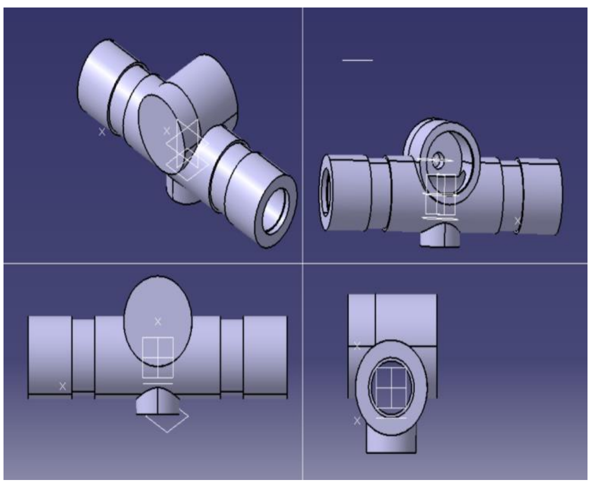




Rack with diameter 25 mm is shown in the figure above. At the ends of the rack threads were made for assembling the rack with tie rods. The surface finish needed is very fine for the rack to easily slide in the casing, so accordingly rack surface finish was achieved.

Material: Stainless steel 303

9. DESIGN CASING

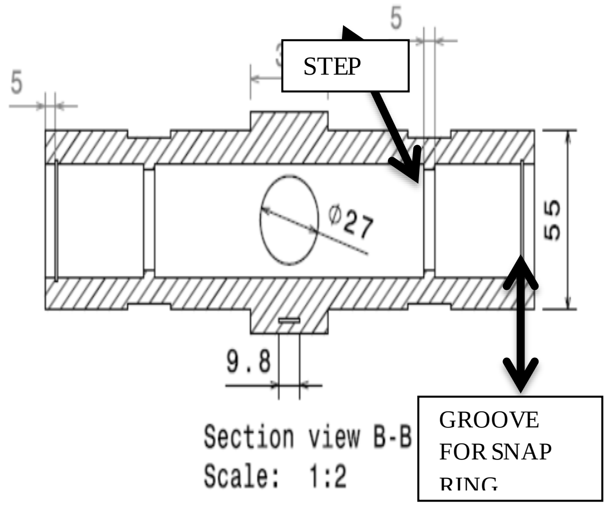


On both side of the casing grooves were made for mounting the casing into the chassis with the help of clamps.

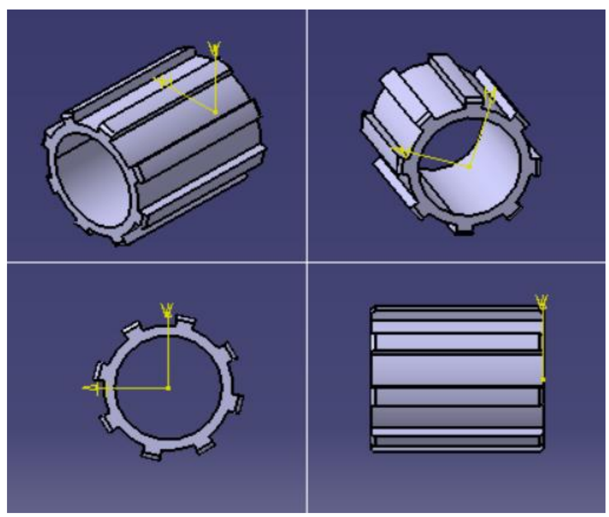
Once the pinion in inserted into the casing, to avoid pinion from coming back and to avoid lubricant from coming out of the casing, a cap was given on top of the pinion as shown below.

Material: 64430 T6 ALUMINIUM

To make the surface contact of the rack with the casing minimum, two sleeves were provided on both the sides of the rack. The purpose of the sleeves was to minimise the surface contact and allow smooth sliding of the rack inside the casing, minimising driver's effort to turn the steering wheel. A very fine surface finish was achieved on the inner surface of the sleeves and these sleeves were then press fitted inside the casing, rack was through these sleeves inside the casing. Casing material step was provided on closer to centre ends of the sleeve and internal grooves for snap rings were provided at the extreme ends of the inside casing to make the sleeves fixed inside the casing. This resulted in smooth sliding of the rack inside the casing.



10. DESIGN OF SLEEVES

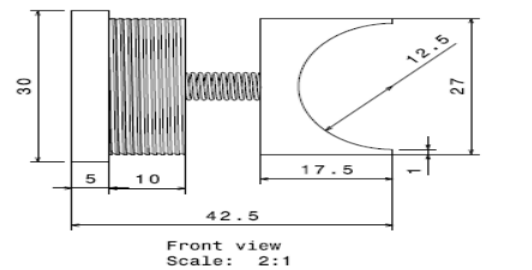


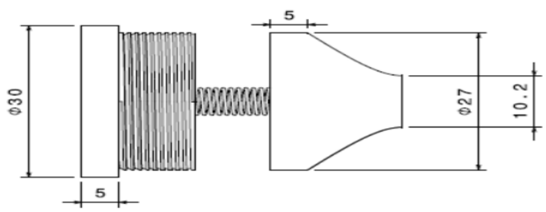
Sleeve with fine surface finish is showed in the figure above

Material: Cast iron

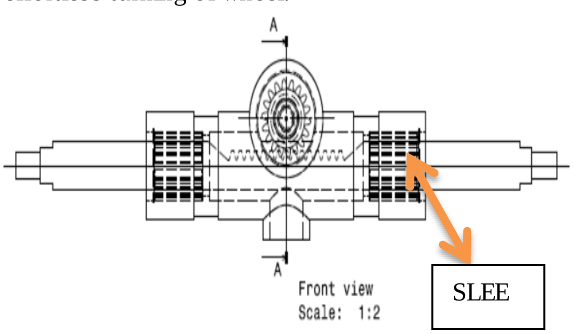
11. RACK SUPPORT

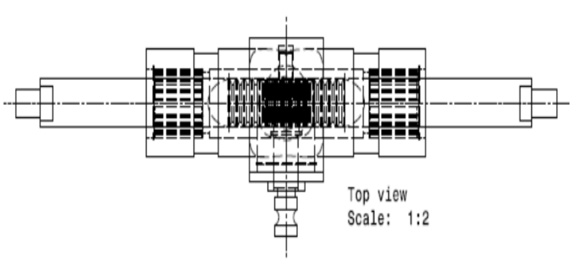
As due to continuous sliding of the rack, the rack can be extremely smooth to slide which is undesirable. So to slide the rack according to our vehicle's need, we provided rack support, a nylon of the shape of the outer surface of the rack was attached with the spring and a screw was provided as shown in the figure.





As we tighten the screw, the nylon holds the rack tightly and in case if the screw is opened, nylon is pulled back by the screw and the rack slides smoothly. The purpose of providing this feature was to make sliding of the rack desirable to the driver, making the driver comfortable by providing tight slide of the rack in case the rack slide is extremely smooth and making the driver effort minimum to turn the wheel by Opening the screw and enabling the smooth slide of the rack in case the driver desires effortless turning of wheel.





12. ASSEMBLY WITH TIE RODS

13. CONCLUSIONS

This paper gives detailed information about the steering system available today. As the design Component of the paper, various mathematical formulas were derived from the fundamental to calculate the exact steer angle under the Ackermann principle. 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.

REFERENCE

[1] Thomas D. Gillespie , "Fundamentals of Vehicle Dynamics", SAE Inc.

[2] Milliken and Milliken , "Race Car Vehicle Dynamics" , SAE Inc.

[3] John C. Dixon , "Suspension Geometry and Computation" , Wiley and Sons Ltd.

[4] Reza N. Jazar , "Vehicle Dynamics Theory and Application" , Springer Publication.

[5] Hans B. Pacejka, "Tyre and Vehicle Dynamics", Second Edition