Design and Optimization of Anti-Ackerman Steering

Vijay Mistry[1], Vipul Awatade[2]

[1,2]UG Students, Department of Mechanical Engineering, Thakur College of Engineering and Technology, Maharashtra, India.

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Abstract - *The main objective of presented work is to design and optimize Anti-Ackerman steering system. The focus was to design a steering system which ensures efficient response over high-speed corners. The setup includes outer wheel steering angle greater than inner to reduce the gap between the drivers steering input and wheels direction of travel. This design allows driver to have optimal control &reduce understeer characteristics. The design & optimization was done in Solid works considering all steering parameters and tested in Lotus Shark.*

Key Words: **Steering, Anti-Ackerman, Solid works, Lotus Shark, Steering effort.**

1. INTRODUCTION

The automobile directional control system i.e. steering system gives driver the ability to control direction of travel of the vehicle by steering the front wheels as per driver input .The two major function of steering mechanism is to provide an angular turn to the front wheels as per the rotary motion of steering wheel and also to multiply the steering effort applied on the steering wheel by the driver .The consistency and preciseness of the steering response depends on the design of the steering geometry and parts i.e. steering wheel, rack and pinion and linkages .The selection of mechanism & geometry was done considering number of parameters such as response of the mechanism, lateral forces, slip, steering effort with an aim to improve the vehicle performance at higher speeds. The orientation and placements of rack and pinion has been made such that it improves the vehicle's steering response maintaining directional stability.

1.1 Problem Statement/ Selection of Steering mechanism

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. [2]

1.2 Objective

The main objective was to optimize the steering system such as to increase the steering response at high-speed corners by reducing the effect of cornering side forces at higher speeds while considering the effect of loads and slip angles on outer wheel. The orientation of rack and pinion as in Ackerman steering i.e., pinion on top of rack where front wheels turn in same direction of rotation of steering wheel cannot be used here as the movement of front wheels will be opposite due to design of Anti-Ackerman steering, so the orientation is made such that rack and pinion teeth's mesh in horizontal plane providing the directional response of wheels same as Ackerman.

2. DESIGN APPROACH

2.1 Mechanism (Rack And pinion)

The steering mechanism consists of gears and linkages and performs two functions i.e., to convert the rotary steering wheel input into linear motion which is required to turn the wheels and to reduce the steering effort of driver. While there are various types available which include the worm, recirculating ball type and the most commonly used traditional system i.e., rack and pinion mechanism. Rack and pinion mechanism is chosen for the current system because of the several advantages as it has fewer parts compared to worm type, precise and easy control, better response, easier repairs.

2.2 Anti-Ackerman Steering

The slip angle increases as the load increases thus while cornering due to increased normal force on the outer tire Ackerman system does not perform well due to less steering angle at the outer tire. In Anti-Ackerman steering the outer wheel has a higher steering angle and the inner wheel has a lower steering angle to reduce effect of slip angle on wheels. As 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. The Anti-Ackerman steering was designed for high-speed corners where the cornering forces are high to get a better steering response by reducing slip and providing more grip to the outer tire during cornering at larger radius turn.

2.3 Steering Geometry parameters

1) Steering ratio – The ratio of angle by which steering wheel is turned to angle of steer of vehicle's front wheels is called as steering ratio.

2)Steer angle - The angle at which the wheel is pointed as per driver input away from the longitudinal axis of initial wheel position is called as steering angle.

3) Rack travel –The amount by which the rack travels side by side over a certain length as the input is given through pinion rotation.

4) Slip angle – The angle between the direction in which tyre is heading and direction of actual movement of tyre with contact on the road is called slip angle. It increases with increase of lateral force.

5) Pneumatic trail –The distance from the horizontal centerline of tire contact patch when viewed from top to the distance at which the peak lateral force acts is called as pneumatic trail. It helps in self-aligning effect.

6) Mechanical trail – The horizontal distance at which the caster axis made by ball joints meets the ground contact when viewed from the side to the centerline of tire is called as mechanical trail. This also produces self-aligning effect. [1]

3. CALCULATIONS AND ASSEMBLY ANALYSIS

Table -1: Design Specifications

Sr.no	Parameters	Values in mm
1	Wheelbase (B)	1570
2	Track width (T) - front and rear	1200
3	Steering wheel radius	210
4	Tire radius	230
5	Track turning radius	2000

3.1 Steering Geometry parameters

Following analytical calculations were made considering the wheelbase and track width length mentioned above and the final steering geometry was designed on solid works software. These values will be further determine using the iteration process of the Anti-Ackerman geometry.

Inner wheel steering angle (θi) = B / (R+T/2) = 1570 / (2000 + 600) = 0.603 rad = 34.59°. [2]

Outer wheel steering angle (θ_o) = B / (R-T/2) = 1570 / (2000 - 600) = 1.12 rad = 64.24°.[2]

Inner turning radius = B/sin (θ_i) = 1570/ sin (34.59) = 2765.54 mm.

Outer turning radius = B/sin (θ_o) = 1570/ sin (64.24) = 1743.23 mm.

The final geometry values of inner, outer steering angle and radius on solid works and the above calculated analytical values were compared and were almost close. Based on the comparison the final geometry was validated. Optimum rack travel in the iterative process was found to be 35.8 mm.

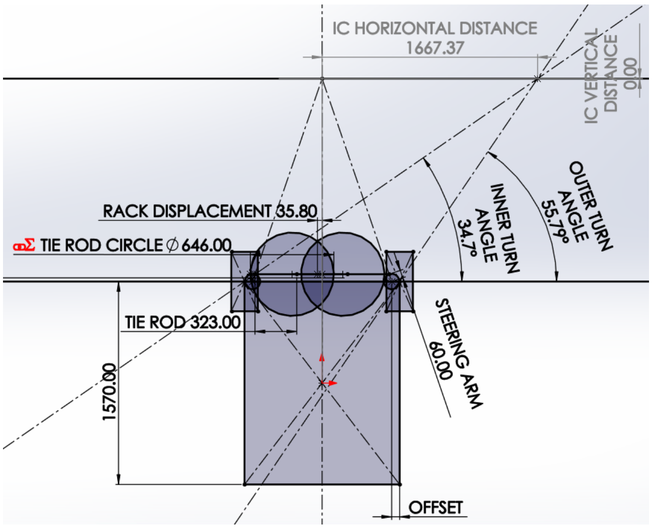

Fig -1: Final Steering Geometry

3.2 Forces on Tie Rod

Considering caster angle of 5° and a kingpin angle of 4° on the wheel the following values were obtained i.e.

Scrub patch due to kingpin axis on the ground contact = 42.41 mm.

Patch on the ground contact due to caster angle i.e., mechanical trail = 28.87 mm.

Steering arm of length 50 mm was finalized after steering geometry iterations.

1) Torque due to mechanical trail: lateral force * mechanical trail = 1000* 28.87 = 28870 N mm.

2) Torque due to scrub radius: traction force * scrub radius = 386.26*42.41 = 16381.65 N mm.

3) Total torque: 28870 + 16381.65 = 45251.65 n mm.

Now the forces on Tie rod = total torque / steering arm length = 45251.65/60=754 N.

3.3 Effort Calculations

From Suspension geometry tie rod inclination (α) was found to be 9.64°

1) Force on Rack: Forces on tie rod * cos (α)

Substituting values, 754 * cos (9.64)

Therefore, Force on rack = 743.35N.

2) Torque on Pinion: Force on rack*pinion radius

Substituting values,

Therefore, Torque on pinion = 743.35 *0.011 = 8.55 N-m

Considering torque on steering wheel equal to torque on pinion which is 8.55 N-m

3) Steering effort: Torque on steering wheel / Steering wheel radius

Substituting values,

Steering effort = 8.55 / 0.21 = 40.71 N.

3.4 Rack and Pinion Design

It is assumed to get maximum rack travel at 180 steering wheel rotation. The maximum rack travel being 35.8 mm. therefore the pitch diameter of pinion can be found by formula of length of arc.

$S = r * \theta$
Substituting the values in the equation:
$35.8 = r \times 180 \times \pi/180$
Therefore, r = 11.40 ~ 11.5mm
Pitch diameter = 2×r =23 mm

Now to determine the module we use Lewis's form factor,
$P_t = \sigma_b \times D.P \times b \times y$
Where Pt is transverse load 1000N,
σ_b is permissible stress considered as 100MPa,
D.P is diametral pitch given by $\pi \times m$,
b is face width considered as 8m,
y is Lewis form factor which is 0.484 for pinion taken as from design data book

Therefore, substituting the values in equation,
$1000 = 100 \times \pi m \times 8m \times 0.484$
We get, m = 0.91 ~ 1.25
Therefore, module is taken as 1.25mm
No. of teeth on Pinion (T) = D/m = 23/1.25
Therefore, T = 18.4 ~ 19
For rack we need maximum 35.8 mm travel on both sides,

Considering 10-20 mm clearance on both sides we get,
Minimum length of teeth on Rack = 90 mm.

Since addendum = module
Addendum circle i.e., outer diameter = (Pitch diameter + 2*m) = D +2m = 23 + 4 = 27mm
[3]

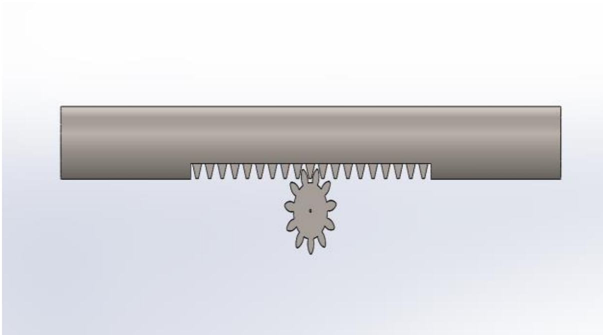

Fig -2: Rack and pinion assembly- Top View

3.5 Lotus shark Analysis

After fixing the coordinates from suspension geometry the values were substituted in lotus to finalize the steering geometry.

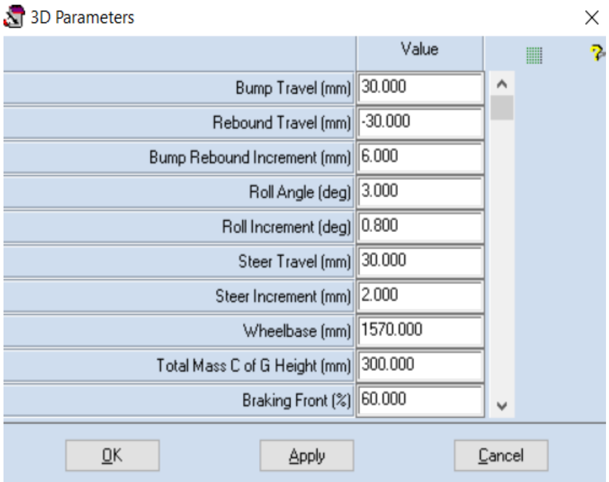

Fig -3: Lotus shark software input parameters

FRONT SUSPENSION - STEER TRAVEL
RHS WHEEL (+ve Y)
TYPE - Double Wishbone, Rocker Arm Damper [corner]

1) INCREMENTAL GEOMETRY VALUES

Steer Travel (mm)	Toe Angle {SAE} (deg)	Toe Angle {SAE} (deg)
30.00	-34.96	61.35
25.00	-29.14	41.76
20.00	-23.40	29.67
15.00	-17.69	20.76
10.00	-11.93	13.19
5.00	-6.06	6.36
0.00	0.00	0.00
-5.00	6.36	-6.06
-10.00	13.19	-11.93
-15.00	20.76	-17.69
-20.00	29.67	-23.40
-25.00	41.76	-29.14
-30.00	61.35	-34.96

Fig -4: Lotus shark incremental geometry -1

Camber Angle (deg)	Camber Angle (deg)	Ackermann (%)	Overall Turning Circle Dia (mm)
3.75	-1.04	-84.69	5439.85
3.03	-1.84	-73.15	6783.08
2.35	-1.67	-64.99	8454.62
1.71	-1.35	-61.23	10980.34
1.11	-0.95	-59.32	15878.33
0.53	-0.50	-58.44	30434.94
0.00	0.00	-58.21	0.00
-0.50	0.53	-58.44	30434.94
-0.95	1.11	-59.32	15878.33
-1.35	1.71	-61.23	10980.34
-1.67	2.35	-64.99	8454.62
-1.84	3.03	-73.15	6783.08
-1.04	3.75	-84.69	5439.85

Fig -5: Lotus shark incremental geometry -2

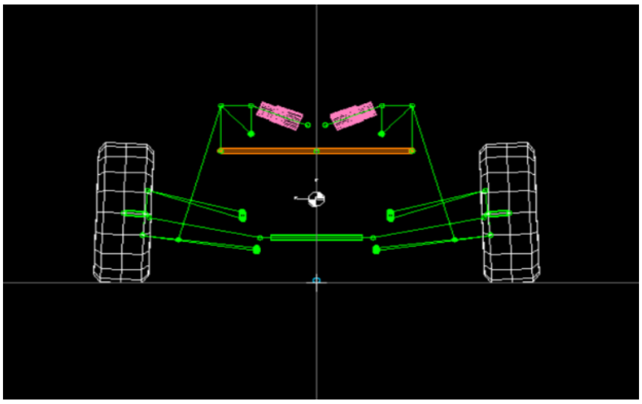

Fig -6: Lotus shark Simulation Front view

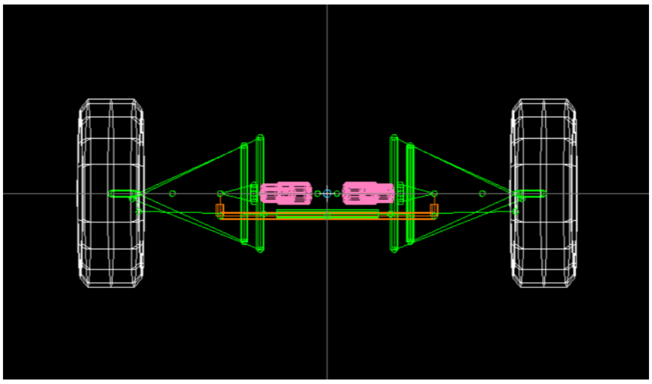

Fig -7: Lotus shark Simulation Top view

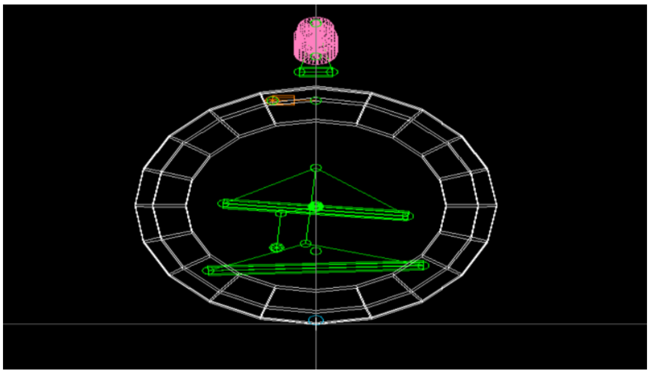

Fig -6: Lotus shark Simulation side view

3.6 Final Assembly

Using the coordinates that were finalized from the lotus simulation the part models for steering system were designed and assembled in solid works. A dual UV joint system is used for attaching the rack to the steering wheel. Quick release is added for easy removal and attachment of steering wheel.

Fig -8: Steering Assembly

Fig -9: Steering Assembly

4. CONCLUSIONS

At higher speeds and loads with the help of Anti-Ackerman steering system the effect of high lateral forces and slip angles is reduced as the outer wheel turns more providing a greater steering angle thereby improving the steering response and directional stability. The radius of turn of outer wheel is less than the inner wheel opposite to the Ackerman steering. The final optimized results from the lotus simulations were used to design steering system as seen in the fig.9 using solid works. The assembly suggested is applied in the vehicle for the usage.

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BIOGRAPHIES

Student: Pursuing B.E in Mechanical Engineering, Thakur College of Engineering and Technology, Mumbai.

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