Design of Steering System for All Terrain Vehicle

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Abstract - *The steering system is a vital component of the vehicle as well as the driver interface. The basic function of steering system has to provide maximum direction control and stability to the vehicle. In this project we are designed steering system for ATV which compete in SAE BAJA competition. An ATV is a vehicle which travels on all terrains because of this the steering system of an ATV is designed for the worst possible terrain and should provide maximum directional stability, pure rolling motion to the wheel, minimum turning radius. The objective of these paper is designed efficient, durable and relatively inexpensive steering system for ATV by using rack and pinion mechanism.*

***Key Words*: All-Terrain Vehicle, Ackerman geometry, Steering design, Rack and pinion mechanism.**

1. INTRODUCTION

The steering system is group of parts that transmit the movement of steering wheel to the front wheels. The objective of steering system is to provide directional control of the vehicle, to withstand high stress in off terrain conditions, to reduce steering effort and to provide good response from road to driver. When vehicle is being driven straight ahead, the steering system must keep it from wandering without requiring the driver to make constant corrections.

Ackerman steering system is based on the four bar linkage mechanism in which different links move relative to each other and finally direct vehicle in particular direction. This system is beneficial during sharp turning and reduces steering efforts. This helps in maneuverability.

Steering rack and pinion mechanism is suitable because of obvious advantages of reduced complexity, ease of construction and less space requirement compare to other steering mechanisms. In rack and pinion mechanism, pinion is attached at the end of the steering shaft. A rack meshes with the pinion. The rotary movement of the steering moves the pinion which gives motion to the rack. The movement of the rack is responsible for turning the wheels through steering linkages.

2. DESIGN OF STEERING SYSTEM

2.1 Requirements of Steering System

1. The steering mechanism should be very accurate and easy to handle.
2. The effort required to steer should be minimum and must not be tiresome to the driver.
3. The steering mechanism should also provide the directional stability. This implies that the vehicle should have tendency to return to its straight ahead position after turning.
4. It should provide pure rolling motion to wheel.
5. It should be designed in such a manner that road shocks are not transmitted to driver.

2.2 Geometry selection

Traction is an important aspect in off-road racing as compared to speed. Since Ackerman steering geometry gives high stability at lower speed. This type of geometry is appropriate for BAJA vehicle where the speed limit is 60 kmph because of the terrain therefore we choose Ackerman steering geometry.

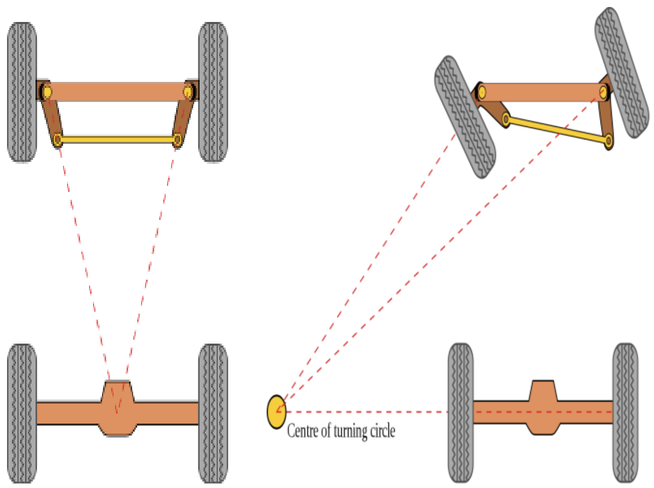

Fig -1: Schematic diagram of Ackerman Steering Mechanism

Ackerman Principle state that the extended axes of steering arm should meet at the Centre of the rear axle. While taking turns, the condition of perfect rolling is achieved if the axes of the front wheels when produced meet the rear axis at one point. This is the instantaneous center of the vehicle. When vehicle is following a curved path the inner wheel deflects by a greater angle than the outer wheel to effectively complete the cornering without skidding.

2.2 Geometry Setup

The steering and suspension systems are crucial for successful operation of any variety of cars. Due to the large responsibility that the these two major components share coupled with the fact that BAJA cars are required to be capable of taking over toughest of the terrains possible, it is obvious that consequences of failure or improper setup of the suspension and/or steering could be quite catastrophic. So combined efforts in suspension geometry and steering geometry were taken in order to design efficient system. Directional control of a road vehicle is normally achieved by steering the front wheels. This is mainly the result of steering wheel movement by the driver, but partly the result of suspension characteristics.

2.3 Steering Design Parameters

Wheelbase	54
Track width	50
Wheel offset	80
Castor Angle	11°
Rack Position From Front Axle From Ground	 96 mm 400 mm
Rack Dimensions Length Travel	 14 in. 5 in.

Table -1: Input Parameters

Wheelbase and track width are selected considering suspension geometry, handling and stability of vehicle. Kingpin offset was decided by considering the packaging of wheel assembly and castor angle selected such that it gives straight line stability and optimum self-returning action for better handling. Position of rack were chosen so as considering pedal packaging and to avoid significant amounts of bump steer.

2.4 Steering Geometry Setup

After getting the system aims clear, the steering iterations were done in PTC Creo Parametric 2.0. Some of the predetermined parameters are listed in the above table.

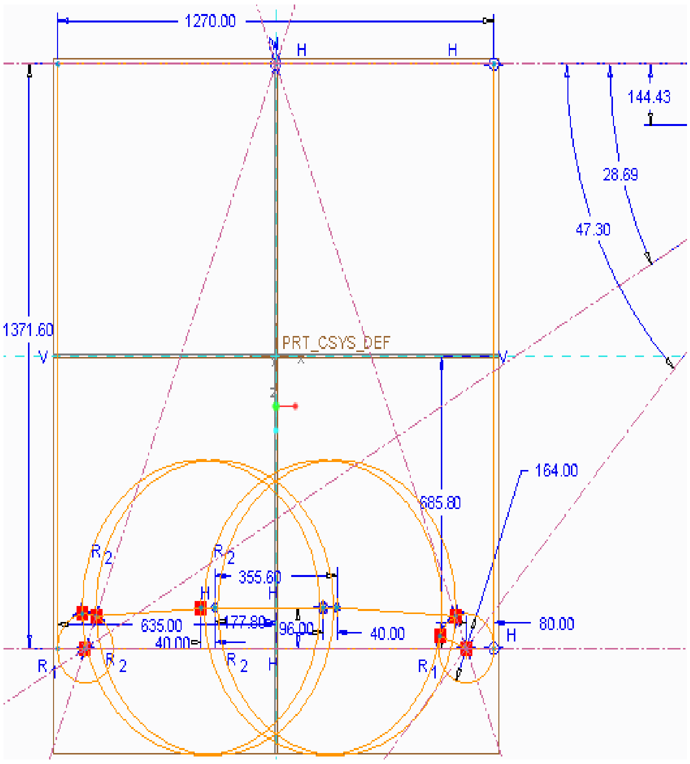

Fig-2: Instantaneous Centre Method for designing Steering System.

Final Design Parameters

Steering Mechanism	Rack and Pinion
Ackerman Angle	22.03°
Steering arm length	82 mm
Tie rod length	347.02 mm
Inner wheel lock angle	47.30°
Outer wheel lock angle	28.69°
Ackerman Percentage	89.46 %

Table -2: Design Parameters

3. Simulation:

Simulation was an important part of validating the designed steering system as it gave us the complete view of how our design was destined to perform on track. Simulations were carried out in MSC Adams Car software. The simulation environment is shown in the figure below.

Steering and suspension system combined in the simulating software were checked for the possibility and extent of bump-steer. The bump-steer graph generated is attached below. From the graph it can be inferred that steering effect with wheel travel varies minimally and hence the bump-steer condition is reduced to its minimum possibility.

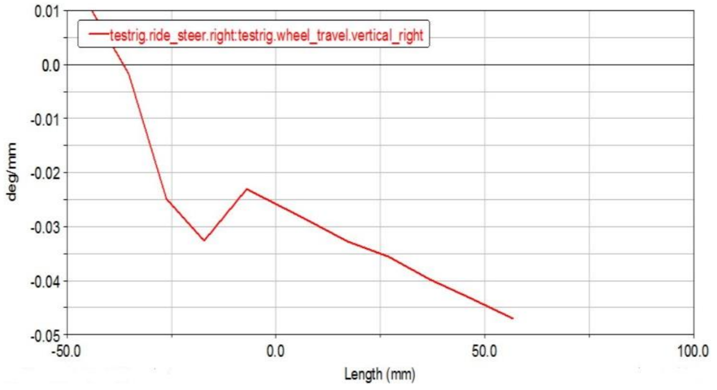

Chart -1: Bump steer Graph

4. Calculation

Velocity at turning	5 m/s
Turning Radius of C.G. (Rcg)	2.086 m
Turning Radius of inner wheel (Ri)	1.616 m
Turning Radius of outer wheel (Ro)	2.626 m
C.G. height	18"
Radius of wheel	292.1 mm
Contact Patch	65.81 mm
Caster Angle (θ)	11°
Tire width	177.8 mm
Front axle load	60 kg
Load on each tire	30 kg
Scrub radius	48.95 mm
Coefficient of friction	0.7
Diameter of steering wheel	150 mm
Rack efficiency	0.9
Tie rod efficiency	0.95
Radius of steering wheel	150mm
Pinion radius	20 mm
Tie rod angle in Top View (ϕ1)	3.30°
Tie rod angle in front view (ϕ2)	12.02°
Ackerman angle (ϕ3)	22.03°

Table -3: Technical Specification of steering system

1) Cornering g force (N) = V^2 /R*g
= 5^2/ (2.086*9.81)
=1.22168 N

2) Wt. transfer at cornering = (g force *C.G. height* front axle load)/ (Track width)
= 1.22168*457.2*60/1270
=26.3882 kg

3) Wt. on inner wheel (Wi) = W- Wt. transfer at cornering
= 30 - 26.3882
= 3.61176 kg

4) Wt. on outer wheel (Wo) = W+ Wt. transfer at cornering
= 30 + 26.38824
= 56.38824 kg

5) Lateral force on inner wheel (Fi) = Wi*V^2 /Ri
= 3.61176 * 5^2/ 1.616
= 55.87498 N

6) Lateral force outer wheel (Fo) = Wo*V^2 /Ro
= 56.38824*5^2/ 2.626
= 536.8264 N

7) Total lateral force (Ft) = Fi + Fo
= 55.87498+ 536.8264
= 592.7013 N

8) Moment of inner wheel due to lateral force (Mi) = Fi*R* tan θ
= 55.87498* 292.1* tan 11
= 3172.48 N-mm

9) Moment of outer wheel due to lateral force (Mo) = Fo*R* tan θ
= 536.8264* 292.1*tan 11
= 30480.19 N-mm

10) Moment at kingpin due to lateral force (M_f) = Mi + Mo
= 3172.48+30480.19
= 33652.5 N-mm

11) Moment at kingpin due to self-aligning torque (M_t) = Total lateral force * contact patch/6
= 592.7013* 65.81/6
= 6500.946 N-mm

12) Net moment at kingpin = M_f +M_t
= 33652.5+ 6500.946
= 40153.45 N-mm

13) Perpendicular force at steering arm = Net moment at kingpin/Steering arm length
= 40153.45/82
= 489.6762 N

14) Force along tie rod at arm end = Perpendicular force at steering arm/ [cos (ϕ1+ϕ3)*cos ϕ2]
= 489.6762/ [cos (3.30+22.03)*cos (12.02)]
= 553.9056 N

15) Force along tie rod at rack end = Force along tie rod at arm end/Tie rod efficiency

= 553.9056/ 0.95

= 583.0585 N

16) Horizontal force at rack = Force along tie rod at rack end/ (cos ϕ2*cos ϕ1)

= 583.0585/ [cos (12.02)*cos (3.30)]

= 597.1186 N

17) Actual force required at rack = Horizontal force at rack/Rack efficiency

= 597.1186/ 0.9

= 663.4652 N

18) Moment at steering shaft/pinion = Actual force required at rack*Pinion radius

= 663.4652*20

= 13269.3 N-mm

19) Force at steering wheel/steering effort = Moment at steering shaft/pinion/(Radius of steering wheel)

= 13269.3/ 150

= 88.462 N-mm

4. Assembly of Steering System

Fig -3: Assembly of Steering System

Fig -4: BAJA Vehicle

5 CONCLUSIONS

We have designed steering system then simulated in Adams software and tested on various rough terrains. The stipulated objectives namely providing better directional stability, minimum turning radius, less steering effort and minimal bump steer.

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