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Simplified Numerical Approach for Steering System Design Parameter Selection for Off-Road Racing

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Abstract – *Vehicle dynamics is considered to be one of the most compact and complex systems to analyse. The steering system is part of the Vehicle Dynamics department and plays an important role in providing directional control to the vehicle. Physically, the steering system appears simple, but the mathematical approach to parameter selection and design is extremely complicated. The purpose of this study is to present a simplified and optimal mathematical approach for selecting steering system design parameters for Off-Road racing vehicles using a rack and pinion mechanism. The findings of this study can be applied to vehicles developed for off-road racing competitions such as mBaja, eBaja, ESI, ATVC, and many others.*

Key Words: Vehicle Dynamics, Steering System, Off-Road racing, Rack & Pinion mechanism, Steering system design.

1. INTRODUCTION:

Mechanical engineers are fascinated by the subject of automobiles, and the automotive industry is a popular field of study and employment. Vehicles are a compact system that combines all mechanical subjects such as materials, production, design, dynamics, transmission, power plant, fuels, and ergonomics into one unit. Vehicle Dynamics is the primary subject that must be studied and analyzed before the vehicle could be designed.

Suspension, steering, braking performance, driver comfort level with ride handling characteristics, and tyres are all covered in Vehicle Dynamics. Because the steering system is solely responsible for directional control, it must be carefully developed and examined.

I In order to have an extraordinarily high-performing automobile, we need a more robust steering system with a high level of comfort for the driver, which is especially important for Off-Road racing cars due to the extremely difficult tracks used for testing and racing. The role of the steering system is to convert rotational motion provided by the steering wheel to translatory motion via a rack and pinion mechanism. When compared to other conventional steering systems, this rack and pinion mechanism is employed in front wheel steering systems, and the rack and pinion gear box can be readily attached.

Provided study aims to give an optimized and simplified numerical approach to finalize the parameters of steering system for designing and manufacturing.

2. METHODOLOGY:

Certain stages, as shown in figure 1, must be followed while designing a steering system. The steering geometry is always chosen initially because it assists in determining the primary dynamics characteristics such as understeer and oversteer. The required turning radius is next determined, after which the track width and wheel base are calculated, followed by the rack and pinion steering gear mechanical calculation, which briefly involves rack and pinion study and calculation. At the penultimate step, steering effort is assessed to determine if the system is operable by the driver or not. The system is validated in the last step by testing the designed prototype.

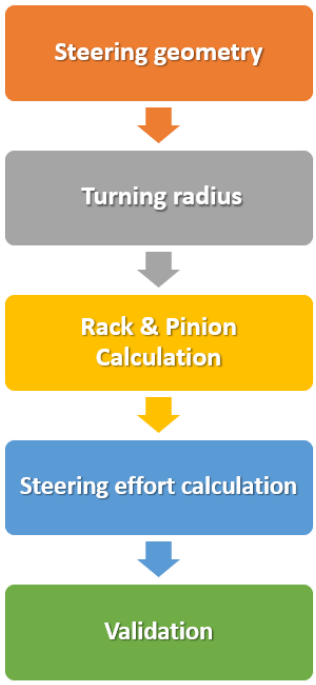

Figure 1 - Methodology flow chart.

3. STEERING GEOMETRY:

Ackermann, Anti-Ackermann, and Parallel steering geometry are the three basic forms of steering geometry. Ackermann appears to be the greatest fit for low speed turning among these three, for Off-Road and slow-moving vehicles like BAJA ATV. It assists in spinning the front inner and outer wheels in such a way that both have a different turning radius, preventing tyre sliding when following a curved path as shown in figure 2. Ackermann percentage can be provided by the software itself where front geometry is designed, software like Lotus Suspension Analysis SHARK and Adams.

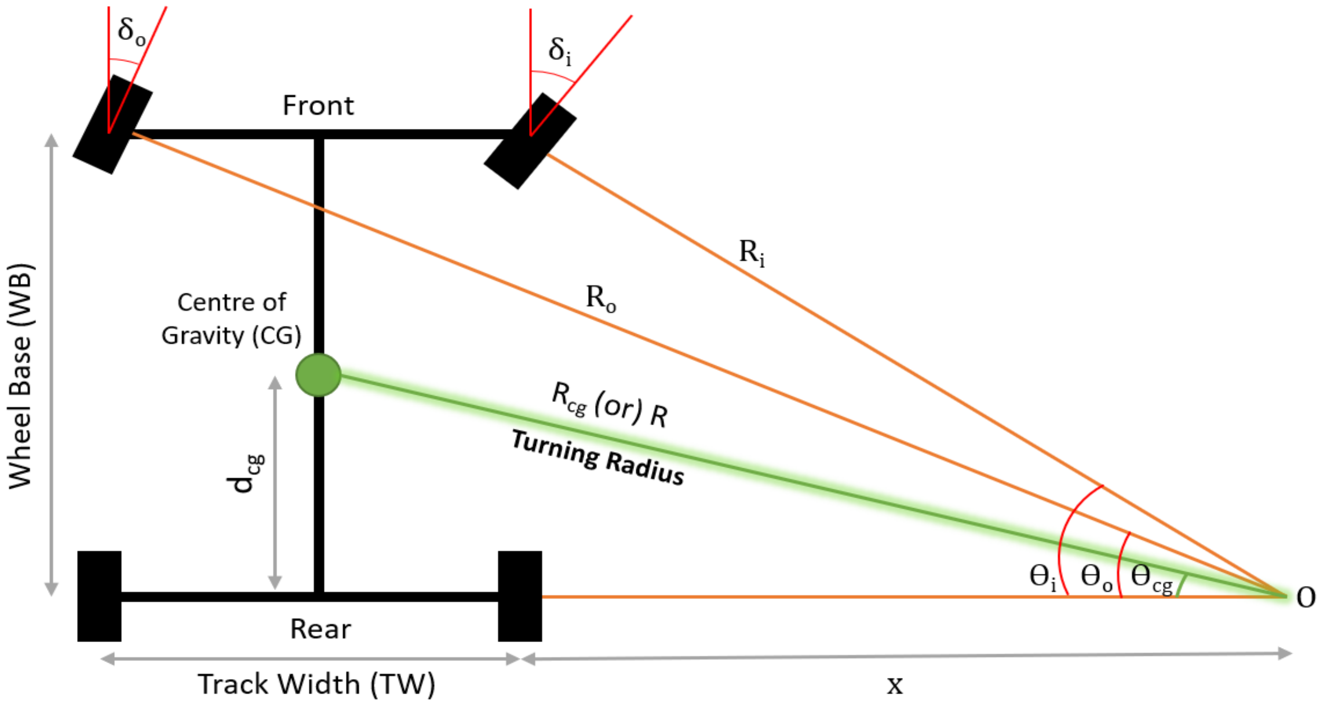

Figure 2 - Ackermann Steering geometry.

4. TURNING RADIUS:

In order to manage the sharp and compact turns, the ideal turning radius must be determined. For Off-Road racing, the turning radius is usually between 2 and 2.5 meters. Now the track width and wheel base must be fixed in order to achieve the required turning radius.

The following equations can be derived from figure 2 to relate track width, wheel base, and turning radius with respect to center of gravity:

$$\cot\theta o - \cot\theta i = \frac{Track\ Width}{Wheel\ Base} \quad \text{------①}$$

$$R^2 = (\frac{TW}{2} + x)^2 + d_{cg}{}^2 \quad \text{------②}$$

$$x^2 = (\frac{WB}{\sin\theta i})^2 - WB^2 \quad \text{------③}$$

Let's consider that vehicle has a 50:50 weight distribution, which means the vehicle's center of gravity (CG) is in the middle i.e., equidistant from the front and rear axles.

Performing several iterations with a track width and wheel base between 1200 mm and 1800 mm, and taking maximum inner wheel turning angle (δi) as 45^0. Under no slip conditions, a track width of 1300 mm and a wheel base of 1500 mm produce a turning radius of 2.2 meters. Let's take these values in preparation for future calculations.

From equation 1, determining outer wheel lock angle (δo) and is found as 28.18^0. Now the maximum average wheel angle deviation (δ) becomes average of inner and outer wheel angle.

$$\delta = \frac{\delta_i + \delta_o}{2} \quad \text{------④}$$

Hence $\delta = 36.6^0$ approximately.

Table 1 - Steering geometry selected parameters.

Parameter	Value
Track Width	1300 mm
Wheel Base	1500 mm
Turning Radius	2.2 m
Inner wheel lock angle	45^0
Outer wheel lock angle	28.18^0
Average wheel deviation	36.6^0

5. RACK & PINION CALCULATION:

The "C-factor" is the quantity around which other parameters and calculations revolve in rack and pinion mechanisms. The linear distance traversed by the rack for one complete revolution is defined as the C-factor. In mathematical terms C-factor is the circumference of the Pitch circle of pinion gear. Assume a pitch circle diameter (PCD) of 33 mm and a gear module of 1.5 for the development of the mechanism.

$$Module\ (M) = \frac{PCD}{N_t} \quad \text{-------⑤}$$

In equation 5 Nt is the number of teeth and as per considered values Nt calculated as 22.

$$C - factor\ (Cf) = \pi * PCD \quad \text{-------⑥}$$

From equation 6 C-factor is calculated as 103.67 mm/rev. As per standards C-factor is expressed in inches/rev hence C-factor becomes 4.08 inches/rev approximately.

$$Rack\ travel = Cf * \left(\frac{Steering\ wheel\ angle}{360^0}\right) \quad \text{-------⑦}$$

For finding rack travel we need to finalize the steering wheel lock angle for present study let's assume driver is interested in rotating steering wheel no more than 120^0 (SWA). Then from equation 7, rack travel becomes 34.56 mm on both sides.

$$Rack\ trave = SAL * \delta \quad \text{-------⑧}$$

In equation 8 the SAL represents the Steering Arm Length which is based upon Rack travel and average wheel deviation. It is calculated as 54 mm approximately.

Now the final part is the steering ratio calculation and for rack and pinion, equation 9 is used for steering ratio calculation:

$$\frac{1}{SR} = \sin^{-1}\left(\frac{C - factor}{SAL * 360}\right) \quad \text{-------⑨}$$

From above equation 9 steering ratio (SR) is calculated as 3.3.

Table 2 - Rack & Pinion parameters

Parameter	Value
PCD	33 mm
C-factor	4.08 in/rev
Gear Module	1.5
Rack travel	34.56 mm
Steering Arm Length	54 mm
Steering ratio	3.3

6. STEERING EFFORT CALCULATION

The steering effort must be optimal and manageable by the driver. When compared to a passenger vehicle on the road, Off-road racing vehicles or ATVs should have more hard steering because there are more bumps and droops from which the vehicle should not self-turn.

To proceed with further calculation following assumptions have been made and are mentioned in table 3.

Table 3 - Assumed parameters for steering effort calculation.

Parameter	Value
Net Vehicle weight (with driver) (m)	200 Kg
Weight distribution	50:50
Friction Coefficient (μ)	0.5
Wheel radius (r)	250 mm
Steering wheel radius (Sr)	15 cm

According to the weight distribution the front axle load is reduced to 50% of the net weight (250 kg), or 125 kg, and the weight on each wheel is lowered to 62.5 kg. The friction coefficient times the load on a tyre equals the total frictional force acting on a wheel. This frictional force causes a couple at the contact patch level on both front tyres while rotating. This coupling is passed to the steering arm, which the driver must apply via the rack and pinion mechanism to negotiate a turn.

$$Net\ Movement\ at\ CP\ (MCP) = 2 * \mu * Tire\ load * wheel\ radius \quad \text{------(10)}$$

MCP = 2*0.5*62.5*9.8*250 Nmm = 153 Nm (approximately)

We know this MCP is equal to movement at steering arm, hence:

$$F * SAL = MCP \quad \text{-------(11)}$$

F = MCP/SAL = 153 Nm / 54 mm = 2833 N (approximately)

The F in above equation 11 is the maximum force that can be transferred to rack via tie-rod and this force (F) will generate a torque with pinion which needs to be provided by driver.

$$Torque\ at\ Pinion\ (TP) = F * PCD/2 \quad \text{------(12)}$$

TP = 2833 N * 33/2 mm = 46.74 Nm

46.74 Nm is the amount of torque which needs to be provided by driver. Now the force needs to be excreted by the driver on the steering wheel is:

$$FSW * Sr = TP \quad \text{------(13)}$$

FSW = TP / Sr = 46.74 Nm / 15 cm = 311.6 N; take 310 N approximately

In above equation 13 FSW represents Force on steering wheel and Sr represents steering wheel radius.

Force excreted by each hand will be half of the net force i.e., 155 N. Hence this can be conclusive steering effort for negotiating a turn and is to be provided by driver in the absence of power assistance.

3. CONCLUSION

An optimized numerical approach to finalize steering system design for Off-Road racing ATVs is presented and calculations have been done with some assumed and estimated values.

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