



DESIGN AND OPTIMIZATION OF STEERING SYSTEM IN ELECTRIC VEHICLE

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Abstract: - The Project Describes about the Design, Analysis and manufacturing of rack and pinion type steering system. The rack and pinion steering system is a pivotal element in electric vehicles (EVs), facilitating precise control and maneuverability. By translating the steering wheel's rotation into lateral motion through a pinion engaging with a toothed rack, this mechanism directs the front wheels, thereby altering the vehicle's direction. Notably responsive and efficient, EVs benefit from the system's ability to capitalize on their instant torque delivery. The compatibility of the rack and pinion setup with contemporary EV architecture supports advanced features like regenerative braking and autonomous driving. In the rapidly evolving automotive landscape driven by EVs. The steering mechanism is a cornerstone of vehicular dynamics, dictating the way a vehicle responds to the driver's inputs and navigates the road. In the context of electric vehicles (EVs), the steering system takes on added significance, aligning with the unique characteristics and potential of electric propulsion. Among the various steering mechanisms available, the rack and pinion system stand out as a versatile and efficient choice that has found widespread application in EVs. The rack and pinion steering system operates on a simple yet effective principle: converting the rotational motion of the steering wheel into the lateral movement of the vehicle's wheels. This translation is achieved through the interaction of a pinion gear and a toothed rack. When the driver turns the steering wheel, the pinion gear, meshed with the rack, causes it to move laterally. This linear motion of the rack is then translated into the angular movement of the vehicle's front wheels, enabling turns and changes in direction.

Keywords :- Design ,Analysis, Manufacturing, Autonomous ,Electric Vehicle, Regenerative braking

INTRODUCTION

The main function of steering system is to convert the rotary motion of steering handle into angular displacement of front wheels. The steering mechanism must also maintain the straight-ahead motion of vehicle while it encounters road bumps and potholes and must operate with minimum effort during operating vehicle.

The purpose of the steering system is to provide directional control of the vehicle with minimum input. The steering is designed to withstand the stress of the vehicle through any type of possible condition at the time of driving. The main concern about steering is, that it should be according to “Ackermann” condition of correct steering.

Steering system is affected by many factors like toe of steering, caster angle, chamber angle, king pin inclination. Our project is a new kind of initiative and will play around with toe of the steering.

I.

OBJECTIVES

1. The objective of this study is to address the key challenges in the design and analysis of steering, ensuring optimal performance characteristics for the vehicle.
2. The objective of this study is to address the key challenges in the design and analysis of steering, ensuring reduce the Steering Efforts.
3. To analyze the Steering design and optimize it according to its needs in terms of safety and other parameters.

II.

DESIGN & CALCULATIONS

Steering system is to provide the directional control to the vehicle by minimizing the input of driver. Thus, the steering mechanism also plays an important role in to transmit the steering movements to the wheel.

Also, it should be with stand the stresses generated while sharp cornering, jounce and rebound of the front wheel. The main thing about steering system is to follow the Ackerman Steering condition for slow speed with no slip angle. Designer should be considering the mechanism should have less play, quick response, type of suspension, space availability, manufacturing cost and weight of the system

WHAT IS RACK AND PINION?

A rack-and-pinion is a device for changing rotary motion into linear motion, in which a gearwheel (the pinion) engages with a flat toothed bar (the rack). On a rack-and-pinion steering system, the end of the steering shaft has a pinion gear that meshes with the rack.

Rack and pinions are used for lifting mechanisms (vertical movement), horizontal movement, positioning mechanisms, stoppers and to permit the synchronous rotation of several shafts in general industrial machinery. On the other hand, they are also used in steering systems to change the direction of cars.

Wheel Base	1259.84 mm
Turn Radius	1824.92 mm
Inner Turning Angle	46.629°
Outer Turning Angle	32.119°
Steering Ratio	
King Pin Inclination	5°
Steering Wheel Diameter	400 mm
Steering Effort	59.25 N
Velocity	7.90 m/s
Height of Centre of Gravity	250 mm
Steering Arm	50 mm

Tie Rods	90 mm
Total Steering Angle	78.748°

III.

GEOMETRY

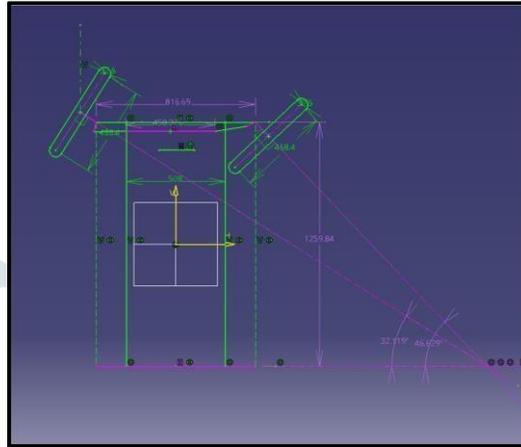


Figure 1 Ackerman geometry of Steering

1.1 Calculation of Turn Radius:

For calculating turn radius of a vehicle, we use the formula of Ackerman condition.

$$\cot \alpha - \cot \beta = \frac{c}{\text{wheel base}}$$

Where, c = KP to KP distance

α = Outer Front wheel turn angle β = Inner Front wheel turn angle

Calculate Turn radius of each wheel, R_1 = Inner

Front Wheel Turn Radius

$$R = \frac{\text{Wheelbase} - \frac{\text{Track width} - c}{2}}{\sin \beta}$$

$$1259.84 - \frac{1012.19 - 791.286}{2}$$

$$R_1 = \frac{\quad}{\sin (46.629)}$$

$$R_1 = 1622.66 \text{ mm}$$

R_2 = Outer Front Wheel Turn Radius $R =$

$$\frac{\text{Wheelbase} + \text{Track width} - c}{2}$$

$$R_2 = \frac{1259.84}{\sin \alpha} - \frac{1012.19 - 791.28}{2}$$

$$R_2 = \frac{\sin(32.119)}{2} + \frac{1259.84}{2}$$

$R_2 = 2479.99 \text{ mm}$

$R_3 = \text{Inner Rear Wheel Turning Radius } R =$

$$R_3 = \frac{\text{Wheelbase} - \frac{\text{Track width} - c}{\tan \beta}}{2}$$

$1259.84 - \frac{1012.19 - 791.28}{\tan \beta}$

$$R_3 = \frac{\tan(46.629)}{2} - \frac{1259.84}{2}$$

$R_3 = 1079.71 \text{ mm}$

$R_4 = \text{Outer Rear Wheel Turning Radius } R =$

$$R_4 = \frac{\text{Wheelbase} + \frac{\text{Track width} - c}{\tan \alpha}}{2}$$

$1259.84 + \frac{1012.19 - 791.28}{\tan \alpha}$

$$R_4 = \frac{\tan(32.119)}{2} + \frac{1259.84}{2}$$

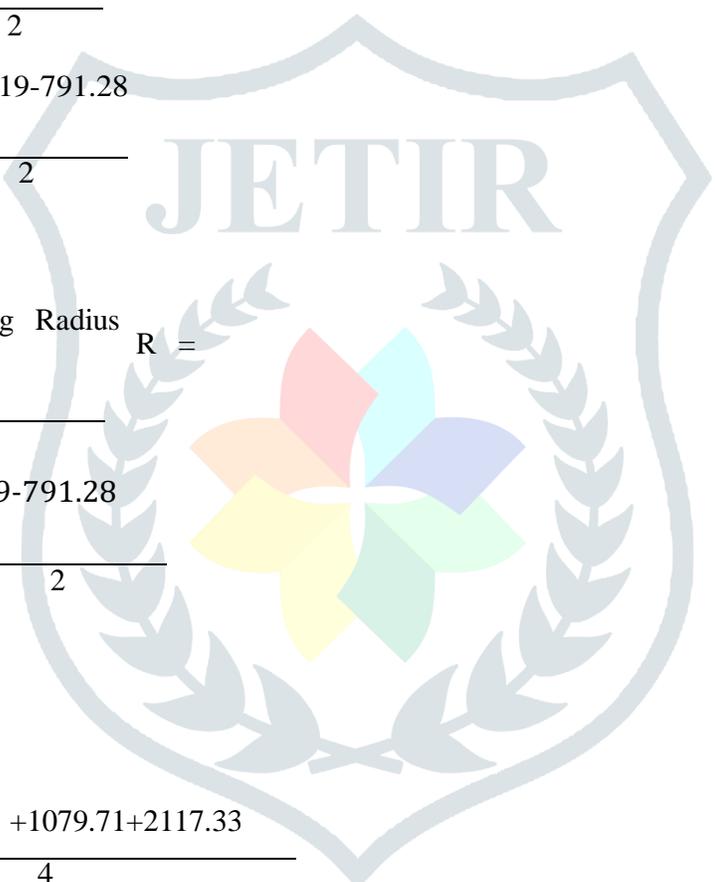
$R_4 = 2117.33 \text{ mm}$

$$R_{\text{Mean}} = \frac{R_1 + R_2 + R_3 + R_4}{4}$$

$$R_{\text{Mean}} = \frac{1622.66 + 2479.99 + 1079.71 + 2117.33}{4}$$

$R_{\text{Mean}} = 1824.92 \text{ mm}$

$\beta \text{ for } 100\% \text{ Ackerman} = \tan^{-1}$	Wheelbase
$\frac{\text{Wheelbase} - \text{track width}}{\tan \alpha}$	_____
$\beta \text{ for } 100\% \text{ Ackerman} = \tan^{-1}$	1259.848
$\frac{1259.848}{\tan(32.119)}$	_____
	-1012.19



β for 100% Ackerman=51.70°

β
3.3 Ackerman Percentage: _____ *100
 β for 100% Ackerman

Ackerman Percentage=90.19%

3.4 Steering Effort:

Mass of vehicle (M) = 80 kg

Velocity of vehicle (V) = 7.90 m/s

Turn radius (R) = 1824.92 mm

Height of C.G. = 250 mm

Track width = 1012.19 mm

Weight Distribution (F: R) = 40:60

Pitch Circle Diameter of Pinion = 44 mm

Steering Arm Length = 50 mm

Scrub Radius = 74.417 mm

Steering Wheel Radius = 200 mm

Offset Angle = 17.95°

$$1. F_{\text{static}} = M * \mu * g$$

$$= 40 * 0.8 * 9.81$$

$$F_{\text{static}} = 313.92 \text{ N}$$

$$2 \text{ Torque on Knuckle} = F_{\text{static}} * \text{Scrub Radius}$$

$$= 313.92 * 74.417$$

$$\text{Torque on Knuckle} = 23360.98 \text{ N}$$

Torque on knuckle

$$3. \quad \text{Force on Steering Arm (F}_s\text{)} = \frac{23360.98}{50}$$

Force on Steering Arm = 467.21 N

$$4. \quad \text{Force on Tie Rod (F}_T\text{)} = F_s * \cos(17.95^\circ)$$

$$= 467.21 * \cos(17.95^\circ)$$

Force on Tie Rod = 444.44 N

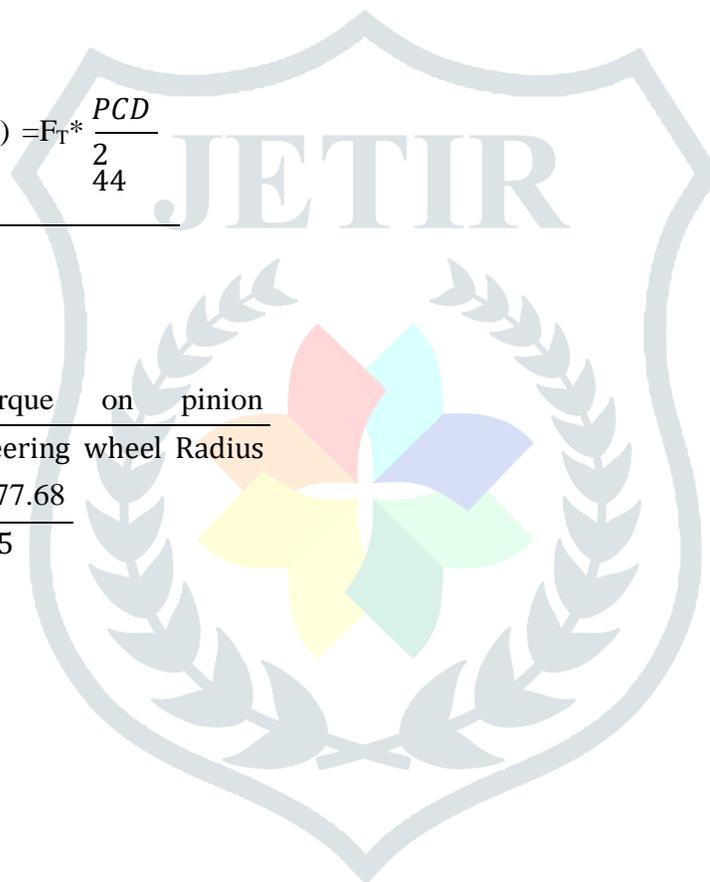
$$5. \quad \text{Torque on Pinion (T}_P\text{)} = F_T * \frac{PCD}{2}$$

$$= 444.44 * \frac{44}{2}$$

Torque on Pinion = 9777.68 N

$$6. \quad \text{Steering Effort} = \frac{\text{Torque on pinion}}{\text{Steering wheel Radius}}$$

$$= \frac{9777.68}{155}$$



Steering Effort = 63.08 N

7. Force on Rack = Force on Tie Rod
= 444.44 N

I. Pinion Calculation:

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:

$$1. \quad \text{Number of Teeth (z)} = \frac{2}{\sin^2 \alpha}$$

$$= \frac{2}{\sin^2(20^\circ)}$$

(Where α is a pressure angle 20°)

$$= 17.09 = 18$$

Hence minimum number of teeth on pinion are 18 Hence we are taking it as 22. Z=

Number of teeth on pinion =

22 γ = Lewis Factor

C_s = Service factor (assume 1 for uniform load and shock) N_p = rpm

of pinion

C_v = Velocity factor

b = Face width of pinion (assume 15mm) S_{ut} =

Ultimate strength (AL6061T6 = 310Mpa) F_s =

Factor of safety taken as 1.5

$$K_w = \text{power in kw} = \frac{M_t \cdot 2\pi n}{60 \times 10^6}$$

By Beam Strength Criteria,

$$60 \times 10^6 \quad \frac{1}{k_w \cdot C_s \cdot (f_s)} \quad 3$$

$$\text{Module} = m = \left[\frac{\pi}{m} \times \left\{ \frac{b_{\text{cut}}}{Z.N.p.v.(\dots)} \right\} \right]$$

1. Module (m) = 2

2. Pitch Circle Diameter (P.C.D.) = $m \times z$
 $= 2 \times 22$
 $= 44$

3. Addendum (h_a) = $1 \times m = 2 \text{ mm}$

4. Dedendum (h_f) = $1.25 \times m = 2.5 \text{ mm}$

II. Pinion Calculation:

Number of teeth on rack

$$= \frac{2 \times \text{rack travel} \times \text{Diameter of pinion}}{\text{Circumference of pinion}}$$

$$= \frac{2 \times 27 \times 44}{\pi \times 44}$$

= 18

IV.

Graph: Rack Travel VS Ackerman %

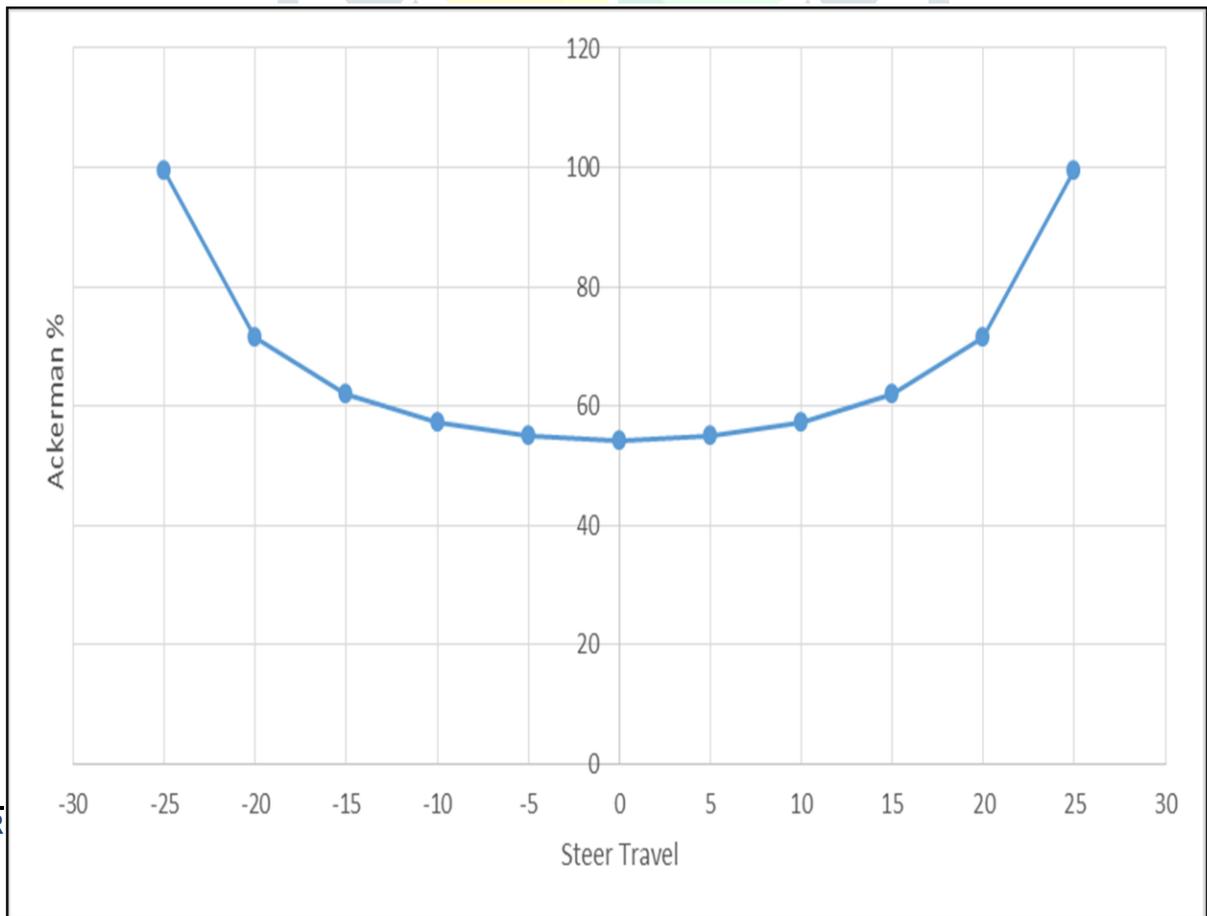


Figure 2 Rack Travel VS Ackerman %

V.

Iterations:

Tyre	Track Width	Wheel Base	KP Distance (C)	α	β	Radian	α in Rad	β in Rad	$\sin \beta$ or α	Wheel Base / $\tan \beta$ or α	Track Width - C	(Track Width - C)/2	R about IC	Total Radius	R/4 (inch)	R mean (mm)
INNER FRONT	39.858	49.000	31.976	31.137	44.9220	0.02		0.78	0.706	69.39	7.88	3.94	65.45 R1	294.396	73.599	1869.415
OUTER FRON	39.858	49.000	31.976	31.137	44.9220	0.02	0.54	0.517	94.76	7.88	3.94	98.70 R2				
									$\tan \beta$ or α	Wheel Base / $\tan \beta$ or α	Track Width - C	(Track Width - C)/2				
INNER REAR	39.858	49.000	31.976	31.137	44.9220	0.02		0.78	0.997	49.13	7.88	3.94	45.19 R3			
OUTER REAR	39.858	49.000	31.976	31.137	44.9220	0.02	0.54	0.604	81.11	7.88	3.94	85.05 R4				

Tyre	Track Width	Wheel Base	KP Distance (C)	α	β	Radian	α in Rad	β in Rad	$\sin \beta$ or α	Wheel Base / $\tan \beta$ or α	Track Width - C	(Track Width - C)/2	R about IC	Total Radius	R/4 (inch)	R mean (mm)
INNER FRONT	39.858	49.600	31.960	33.964	49.9630	0.02		0.87	0.766	64.78	7.90	3.95	60.83 R1	268.874	67.219	1707.350
OUTER FRON	39.858	49.600	31.960	33.964	49.9630	0.02	0.59	0.559	88.78	7.90	3.95	92.73 R2				
									$\tan \beta$ or α	Wheel Base / $\tan \beta$ or α	Track Width - C	(Track Width - C)/2				
INNER REAR	39.858	49.600	31.960	33.964	49.9630	0.02		0.87	1.190	41.67	7.90	3.95	37.72 R3			
OUTER REAR	39.858	49.600	31.960	33.964	49.9630	0.02	0.59	0.674	73.63	7.90	3.95	77.58 R4				

Tyre	Track Width	Wheel Base	KP Distance (C)	α	β	Radian	α in Rad	β in Rad	$\sin \beta$ or α	Wheel Base / $\tan \beta$ or α	Track Width - C	(Track Width - C)/2	R about IC	Total Radius	R/4 (inch)	R mean (mm)
INNER FRONT	39.858	49.600	31.960	32.097	46.4710	0.02		0.81	0.725	68.41	7.90	3.95	64.46 R1	287.952	71.988	1828.498
OUTER FRON	39.858	49.600	31.960	32.097	46.4710	0.02	0.56	0.531	93.35	7.90	3.95	97.30 R2				
									$\tan \beta$ or α	Wheel Base / $\tan \beta$ or α	Track Width - C	(Track Width - C)/2				
INNER REAR	39.858	49.600	31.960	32.097	46.4710	0.02		0.81	1.053	47.12	7.90	3.95	43.17 R3			
OUTER REAR	39.858	49.600	31.960	32.097	46.4710	0.02	0.56	0.627	79.08	7.90	3.95	83.03 R4				

Tyre	Track Width	Wheel Base	KP Distance (C)	α	β	Radian	α in Rad	β in Rad	$\sin \beta$ or α	Wheel Base / $\tan \beta$ or α	Track Width - C	(Track Width - C)/2	R about IC	Total Radius	R/4 (inch)	R mean (mm)
INNER FRONT	39.858	49.600	31.960	26.680	40.1400	0.02		0.70	0.645	76.94	7.90	3.95	72.99 R1	344.929	86.232	2190.299
OUTER FRON	39.858	49.600	31.960	26.680	40.1400	0.02	0.47	0.449	110.47	7.90	3.95	114.41 R2				
									$\tan \beta$ or α	Wheel Base / $\tan \beta$ or α	Track Width - C	(Track Width - C)/2				
INNER REAR	39.858	49.600	31.960	26.680	40.1400	0.02		0.70	0.843	58.82	7.90	3.95	54.87 R3			
OUTER REAR	39.858	49.600	31.960	26.680	40.1400	0.02	0.47	0.503	98.70	7.90	3.95	102.65 R4				

Tyre	Track Width	Wheel Base	KP Distance (C)	α	β	Radian	α in Rad	β in Rad	$\sin \beta$ or α	Wheel Base / $\tan \beta$ or α	Track Width - C	(Track Width - C)/2	R about IC	Total Radius	R/4 (inch)	R mean (mm)
INNER FRONT	39.858	49.600	31.153	32.097	46.4710	0.02		0.81	0.725	68.41	8.71	4.35	64.06 R1	287.952	71.988	1828.498
OUTER FRON	39.858	49.600	31.153	32.097	46.4710	0.02	0.56	0.531	93.35	8.71	4.35	97.70 R2				
									$\tan \beta$ or α	Wheel Base / $\tan \beta$ or α	Track Width - C	(Track Width - C)/2				
INNER REAR	39.858	49.600	31.153	32.097	46.4710	0.02		0.81	1.053	47.12	8.71	4.35	42.76 R3			
OUTER REAR	39.858	49.600	31.153	32.097	46.4710	0.02	0.56	0.627	79.08	8.71	4.35	83.43 R4				

Tyre	Track Width	Wheel Base	KP Distance (C)	α	β	Radian	α in Rad	β in Rad	$\sin \beta$ or α	Wheel Base / $\tan \beta$ or α	Track Width - C	(Track Width - C)/2	R about IC	Total Radius	R/4 (inch)	R mean (mm)
INNER FRONT	39.858	49.600	31.153	32.119	46.6290	0.02		0.81	0.727	68.23	8.71	4.35	63.88 R1	287.390	71.847	1824.926
OUTER FRON	39.858	49.600	31.153	32.119	46.6290	0.02	0.56	0.532	93.29	8.71	4.35	97.64 R2				
									$\tan \beta$ or α	Wheel Base / $\tan \beta$ or α	Track Width - C	(Track Width - C)/2				
INNER REAR	39.858	49.600	31.153	32.119	46.6290	0.02		0.81	1.059	46.86	8.71	4.35	42.50 R3			
OUTER REAR	39.858	49.600	31.153	32.119	46.6290	0.02	0.56	0.628	79.01	8.71	4.35	83.36 R4				

VI.**Lotus Software:**

The steering geometry had to be analyzed using a particular software to determine the steering parameters for best values of better steer. The software chosen was LOTUS Shark analyzer due to its ease of use and accurate results. The process used was to determine the 2D suspension points in Catia V5 and then input them into LOTUS Shark analyzer. After the first set of points were entered into the software, a number of iterations were carried out to determine the best possible values for the steering geometry.

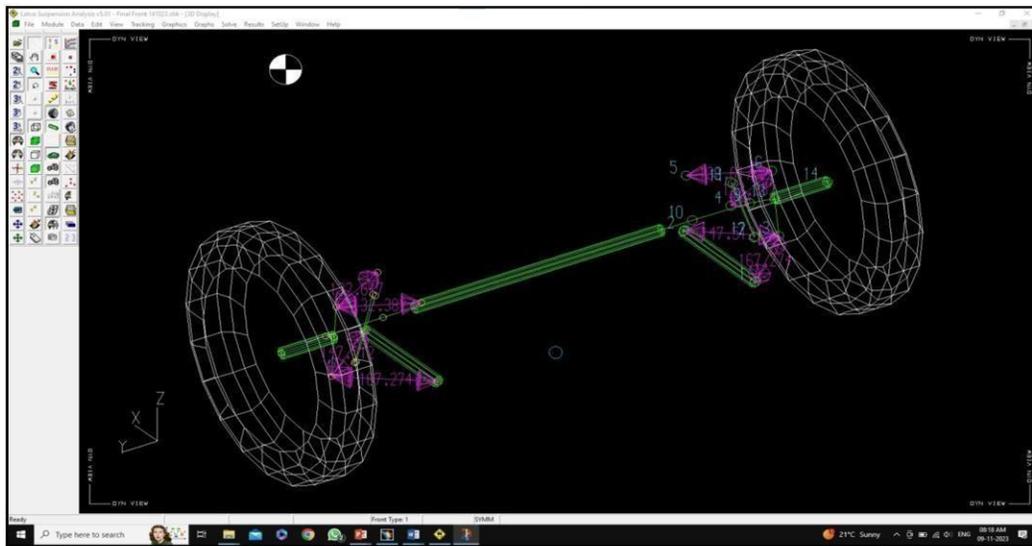


Figure 3 After input values

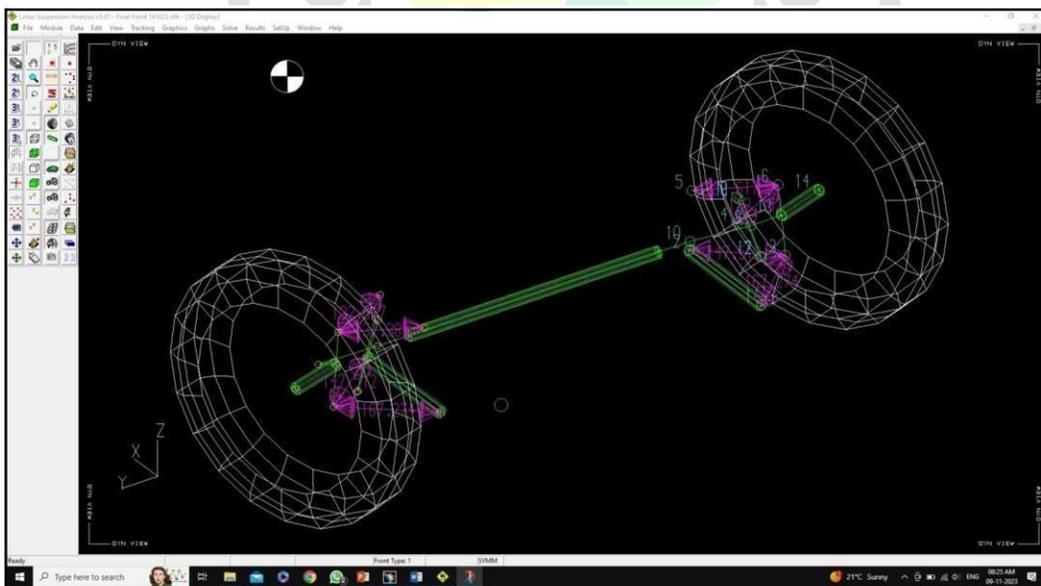


Figure 4 Running the software

LOTUS SUSPENSION ANALYSIS v5.01

FRONT SUSPENSION

FILENAME: Final Front 141023.shk

TYPE 1 Double Wishbone, damper to lower wishbone

STATIC VALUES

X (mm)	Y (mm)	Z (mm)		
3976.03	-294.32	198.21	POINT:1	Lower wishbone front pivot
4179.00	-295.86	200.35	POINT:2	Lower wishbone rear pivot
4092.00	-414.73	192.56	POINT:3	Lower wishbone outer ball joint
4029.84	-286.83	322.48	POINT:4	Upper wishbone front pivot
4154.25	-286.08	322.15	POINT:5	Upper wishbone rear pivot
4092.50	-403.45	322.50	POINT:6	Upper wishbone outer ball joint
4093.80	-370.27	203.60	POINT:7	Damper wishbone end
4098.59	-331.42	321.21	POINT:8	Damper body end
4140.06	-392.93	240.51	POINT:9	Outer track rod ball joint
4140.06	-286.63	241.94	POINT:10	Inner track rod ball joint
4092.11	-331.42	321.19	POINT:11	Upper spring pivot point
4091.10	-369.68	206.24	POINT:12	Lower spring pivot point
4092.50	-408.35	266.50	POINT:13	Wheel spindle point
4092.50	-506.10	266.50	POINT:14	Wheel centre point
4030.00	-440.00	195.00	POINT:15	Part 1 C of G
4170.00	-520.00	450.00	POINT:16	Part 2 C of G
4230.00	-525.00	220.00	POINT:17	Part 3 C of G
4130.00	-720.00	275.00	POINT:18	Part 4 C of G

STATIC VALUES

Camber Angle (deg):	0.00
Toe Angle {Plane} (deg):	0.00
Toe Angle {SAE} (deg):	0.00
Castor Angle (deg):	0.22
Castor Trail (hub) (mm):	-0.22
Castor Offset (grnd) (mm):	1.08
Kingpin Angle (deg):	4.97
Kingpin Offset (w/c) (mm):	97.78
Kingpin Offset (grnd) (mm):	78.23
Mechanical Trail (grnd) (mm):	1.08
ROLL CENTRE HEIGHT (mm):	61.16

GENERAL DATA VALUES

TYRE ROLLING RADIUS (mm):	225.00
WHEELBASE (mm):	1259.84
C OF G HEIGHT (mm):	250.00
BREAKING ON FRONT AXLE (%)	60.00
DRIVE ON FRONT AXLE (%)	0.00
WEIGHT ON FRONT AXLE (%)	40.00
OUTBOARD FRONT BRAKES:	
INDEPENDENT FRONT SUSPENSION:	
RACK TYPE STEERING ARTICULATION:	

RUN DETAILS FRONTSUSPENSION ONLY:

BUMP TRAVEL (mm):	60.00	INCREMENT (mm):	20.00
REBOUND TRAVEL (mm):	60.00	INCREMENT (mm):	20.00
ROLL ANGLE (deg):	3.00	ROLL INCREMENT (deg):	0.50
STEERING TRAVEL (mm):	27.00	STEERING INCREMENT (mm):	5.00

LOTUS SUSPENSION ANALYSIS v5.01

***** FRONT

SUSPENSION - STEERING TRAVEL

LHS WHEEL (-ve Y)

TYPE 1 Double Wishbone, damper to lower wishbone

INCREMENTAL GEOMETRY VALUES

Steer Travel (mm)	Toe Angle {SAE} (deg)	Toe Angle {SAE} (deg)	Camber Angle(deg)	Camber Angle(deg)	Ackerman (%)	Turning circle radius(mm)
25.00	43.73	-28.46	1.50	0.48	99.44	1820.70
20.00	29.12	-22.82	0.71	0.29	71.44	2627.98
15.00	20.16	-17.21	0.37	0.15	62.09	3749.37
10.00	12.76	-11.59	0.16	0.05	57.33	5853.32
5.00	6.15	-5.88	0.05	0.00	54.95	11963.83
0.00	0.00	0.00	0.00	0.00	54.22	0.00
-5.00	-5.88	6.15	0.00	0.05	54.95	11963.83
-10.00	-11.59	12.76	0.05	0.16	57.33	5853.32
-15.00	-17.21	20.16	0.15	0.37	62.09	3749.37
-20.00	-22.82	29.12	0.29	0.71	71.44	2627.98
-25.00	-28.46	43.73	0.48	1.50	99.44	1820.70

LOTUS SUSPENSION ANALYSIS v5.01

FRONT SUSPENSION - STEERING TRAVEL

RHS WHEEL (+ve Y)

TYPE 1 Double Wishbone, damper to lower wishbone

SteerTravel (mm)	Toe Angle {SAE} (deg)	Toe Angle {SAE} (deg)	Camber Angle(deg)	Camber Angle(deg)	Ackerman (%)	Turning circle radius(mm)
25.00	43.73	-28.46	1.50	0.48	99.44	1820.70
20.00	29.12	-22.82	0.71	0.29	71.44	2627.98
15.00	20.16	-17.21	0.37	0.15	62.09	3749.37
10.00	12.76	-11.59	0.16	0.05	57.33	5853.32
5.00	6.15	-5.88	0.05	0.00	54.95	11963.83
0.00	0.00	0.00	0.00	0.00	54.22	0.00
-5.00	-5.88	6.15	0.00	0.05	54.95	11963.83
-10.00	-11.59	12.76	0.05	0.16	57.33	5853.32
-15.00	-17.21	20.16	0.15	0.37	62.09	3749.37
-20.00	-22.82	29.12	0.29	0.71	71.44	2627.98
-25.00	-28.46	43.73	0.48	1.50	99.44	1820.70

VII.

CAD Models:

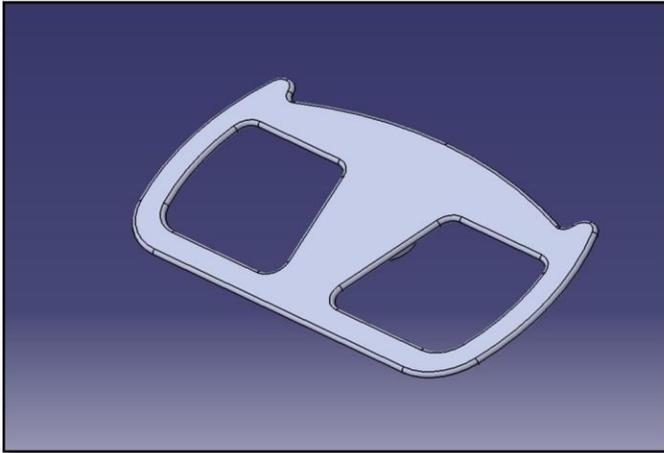


Figure 5 Steering Wheel

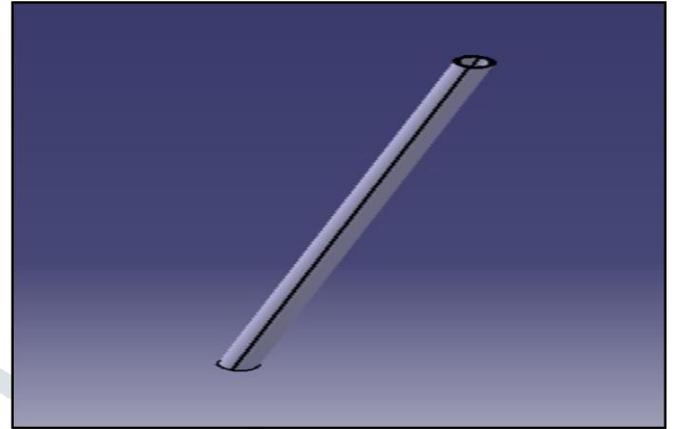


Figure 6 Steering column

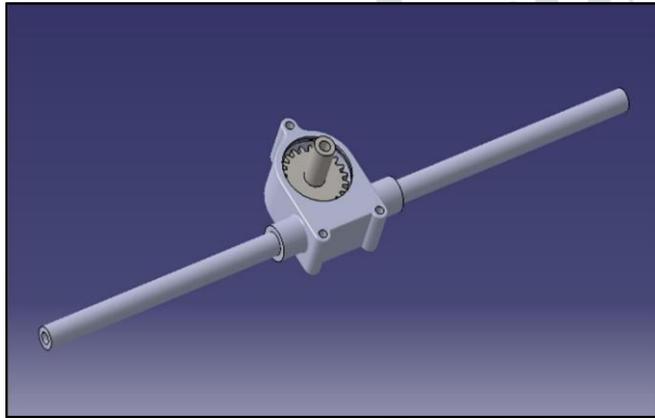


Figure 7 Rack & Pinion

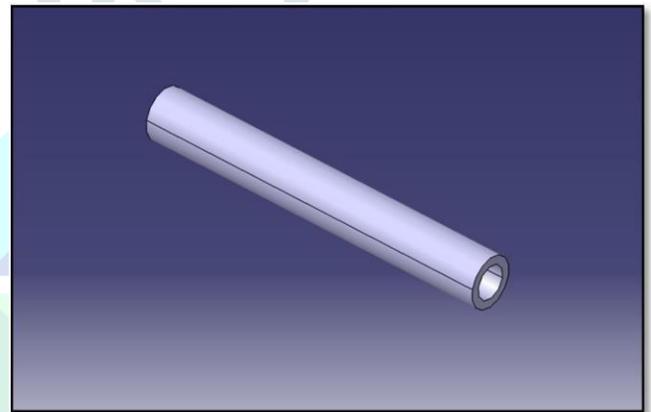


Figure 8 Tie Rods

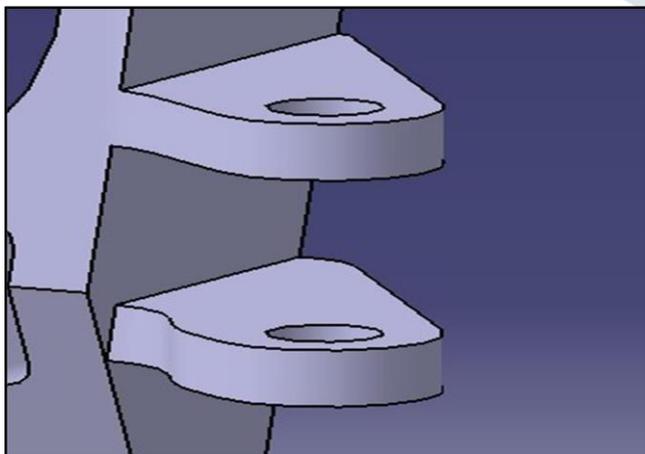


Figure 9 Steering Arm

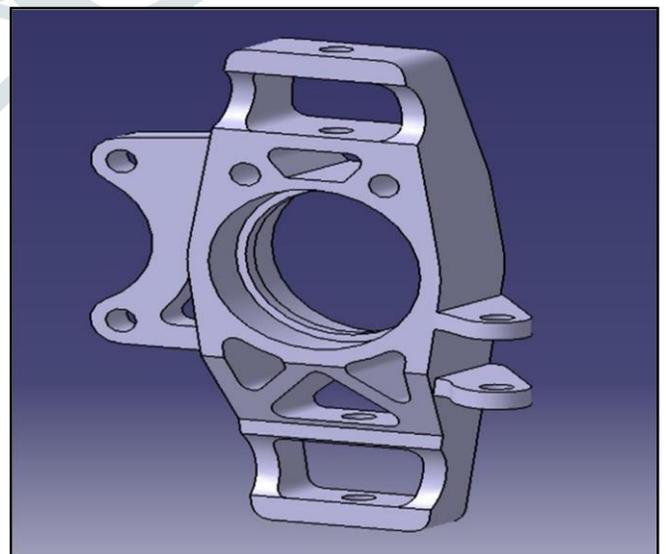


Figure 10 Knuckle

VIII.

Actual Models:



Figure 11 Knuckle

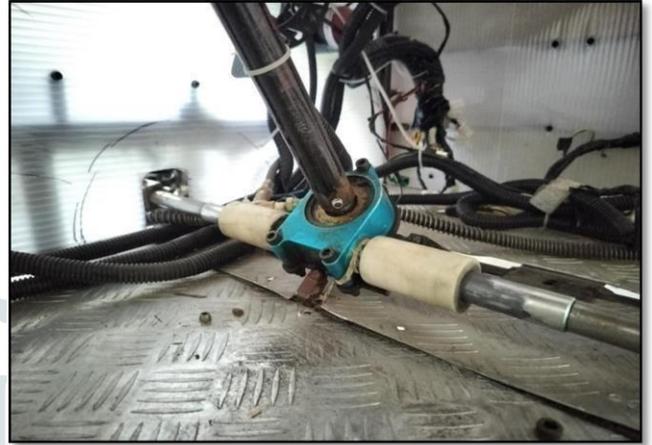


Figure 12 Rack & Pinion

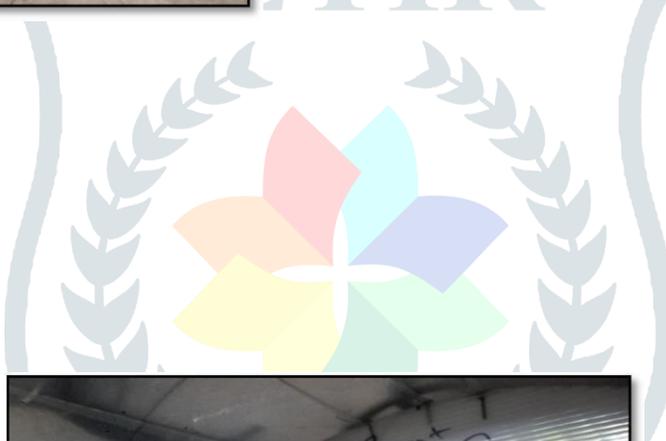


Figure 13 Assembly

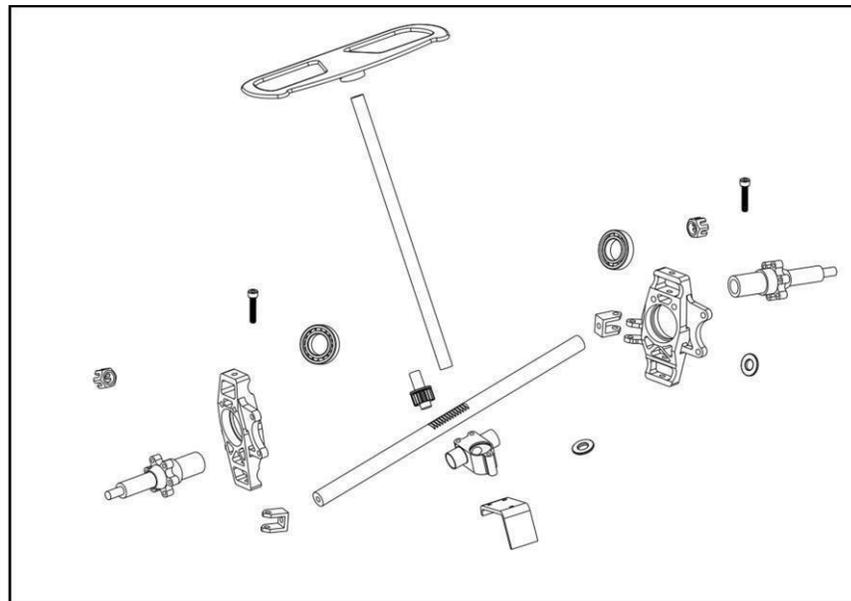


Figure 14 Exploded View

Table 3 Steering Parts

Sr. No	Part Name	Sr. No	Part Name
1.	Steering Wheel	8.	Tie Rods
2.	Steering Column	9.	Rose Joint
3.	Rack	10.	Knuckle
4.	Pinion	11.	Axle
5.	Brass Washer	12.	Bearing
6.	Casing	13.	Castle Nut
7.	Rack Bracket	14.	Casing Mountings

IX. STEERING KNUCKLE CALCULATION:

Steering Knuckle is important part of the vehicle which connects steering, suspension, brakes, and wheel on axle to chassis. Knuckle undergoes different types loading in various conditions.

Knuckle is then further classified into Hub type knuckle and spindle type knuckle in which we are design the spindle type knuckle. Knuckle has steering arm for steering purpose, brake mounting for brake calipers, axle or spindle for wheel mount.

Design of knuckle depends on main suspension type like independent type or dependent type and suspension like connection to wheel, steering arm, brake calipers mounting and then optimization of weight.

Forces Acting on Knuckle:

- 1) Cornering Force: 676.205 N
- 2) Braking Moment: 121.023 Nm
- 3) Lateral load transfer: 588.6 N
- 4) Longitudinal Load Transfer: 1172.4 N

Material Selection:

Material for knuckle is Mild Steel for Steering column, steering arm and Aluminum 6061 T6 for Knuckle with brake mounting.

Properties of Material:

Table 4 Material Details

Material	Al6061 T6
Density	$2.7e^{-6}$ kg/mm ³
Young's modulus	68900 MPa
Poisson's Ratio	0.33
Bulk Modulus	67549 MPa
Shear Modulus	25902 MPa
Tensile Ultimate Strength	310 MPa
Tensile Yield Strength	276 MPa

Table 5 Material Details

Material	Mild Steel
Density	$7.8e^{-6}$ kg/mm ³
Young's modulus	$2.1e^5$ MPa
Poisson's Ratio	0.29
Bulk Modulus	$1.6667e^{-5}$ MPa
Shear Modulus	81395 MPa
Tensile Ultimate Strength	440 MPa
Tensile Yield Strength	370 MPa

Parameters:

Table 6 Steering Parameters

Track width	1012.19 mm
Wheelbase	1259.84 mm
King Pin Inclination	5°
Scrub Radius	74.417 mm
Weight of Vehicle	200 kg
Weight distribution	40:60

X. FORCES ACTING ON KNUCKLE:

1. **Vehicle Weight:** 200Kg

Weight distribution 40:60

Hence the weight on front knuckles = 80Kg But on

Each Knuckle the Weight = 40×9.81

$$= 392.4 \text{ N}$$

2. **Bump Force:**

= Mass of vehicle \times acceleration of vehicle

$$\text{Acceleration} = \frac{\text{Velocity}}{\text{Time}}$$

$$= \frac{7.9}{3.97}$$

$$= 1.98 \text{ m/s}^2$$

$$\text{Bump Force} = 200 \times 1.98$$

$$= 396 \text{ N}$$

3. **Cornering force:**

$$\frac{m \times v^2 \times \text{Height of C.G.}}{\text{Track Width} \times R}$$

$$\frac{80 \times 7.9^2 \times 0.25}{1.012 \times 1.824}$$

$$= 676.205 \text{ N}$$

4. **Force on Steering Arm** = 467.21 N

5. **Braking Torque** = 121.023 N.m



XI.**CONCLUSION**

Electric vehicles are becoming a popular alternative for the foreseeable future. Therefore, this study covers the design, modeling, and simulation procedures for an electric car's steering system in detail. Electric vehicles have become a worldwide obsession. It became a popular activity throughout time. As a result, the demand for affordable and safe electric vehicles is rising. Hence designing a low weight and safe steering system is a first step towards manufacturing a low budget electric vehicle. Furthermore, our optimization efforts have only led to improvements in vehicle handling and maneuverability but also contributed to overall vehicle performance, fuel efficiency, and safety standards have not only considered performance metrics but also taken into account factors such as manufacturing feasibility, cost-effectiveness, and sustainability. Through the adoption of lightweight materials, streamlined manufacturing processes.

The design and optimization of the steering system represent a dynamic and evolving field within automotive engineering, where continual innovation and collaboration drive progress innovation and collaboration drive progress towards safer, more efficient, and more enjoyable driving experiences for all.

XII.**FUTURE SCOPE**

Electric vehicles are becoming a popular alternative for the foreseeable future. Therefore, this study covers the Steering design, modelling, and simulation procedures for an electric car in detail. Electric vehicles have become a worldwide obsession. It became a popular activity throughout time.

As a result, the demand for affordable and safe electric vehicles is rising. Hence designing a low weight and low cost, safe steering is a step towards manufacturing a low budget electric vehicle.

XIII.**References**

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