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Design, Analysis and Fabrication of an Effective Steering System

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Abstract: ***The main objective of steering system is to manoeuvre the vehicle while maintaining pure rolling condition and giving maximum directional stability under toughest of the terrains. This article deals with the designing and manufacturing of an efficient and light weight steering system for an All-Terrain Vehicle (ATV) using rack and pinion mechanism. The main focus was given on driver comfortability, achieving maximum Ackermann percentage and steering camber control for which brainstorming and selection of proper joints along with the selection of proper material for the components was done. Various front geometries have been iterated upon and the one which met our primary targets was finalized. This study summarizes in brief the design parameters, targets and considerations followed by fabrication and assembly of the system***
Keywords: Ackermann geometry, Steering Effort, Steering Arm, Kingpin inclination,

I. INTRODUCTION

The basic function of steering system is to convert the rotational input from the driver into angular deviation of tires to give directional characteristics to the vehicle. The rack and pinion mechanism was selected for steering due its simplicity and effectiveness. In rack and pinion mechanism, the wheels are steered by the linear motion of the rack caused by the rotation of pinion. The very first step towards designing the steering system was selection of a proper compatible steering geometry. Ackermann steering geometry seemed to be the appropriate one, as it provides camber and toe adjustments along with it being compatible with suspension system as well as satisfying the law of correct steering. The primary target was to improve the design which leads to better outputs and efforts were made in that direction along with overall weight and cost reduction of the entire steering system.

A. Design Objectives

The main objective while designing the steering system was to get an optimized and stable front geometry with minimum turning radius, low steering effort and good amount of Ackermann percentage along with optimum steering camber change

B. Design Targets

To satisfy the system requirements various designing targets were set which are as follows.

1) To reduce the steering effort.
2) To reduce the turning radius of buggy.
3) To avoid any kind of play in the system.
4) To reduce weight of the components used in the system.

C. Design Considerations

To achieve above targets, following considerations were made to make the system more stable and efficient.

II. DESIGN PROCEDURE

Design procedure gives step by step description of the proceedings done while designing the steering system so as to meet the objectives and the targets.

A. By discussing with other departments, the track width and wheel base were decided.
B. By considering above parameters we started designing proper compatible steering geometry on LOTUS software.
C. In order to achieve steering targets, iterations were carried out by virtual simulation on trial and error basis method to decide the dimensions of parts and the components used.
D. The effects were studied by determining the output values from the iterations and analyzing the value of the parameters.

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E. The iterations were made on positions of components like Rack position, Steering arm length, Rack travel and Rack length to achieve optimum Ackermann percentage and turning radius with low steering camber, mechanical trail and scrub radius.

III.ITERATIONS

Iterations were made after finalizing the Steering Ratio by assuming the Values of Track Width, and Wheelbase, Rack Length, Rack Position (w.r.t Stub Axle), Rack Travel and Steering Arm Length. These above parameters were varied in order to study their effects on other parameters like Turning Radius, Inner Tire Angle, Outer Tire Angle and Ackermann Percentage and at last the best outcome was selected as the final iteration. The iterations were done on both software as well as on sketch, for cross checking the design and to implement it in practical conditions

A. Iteration 1

1) Coordinates

FRONT SUSPENSION FILENAME: final front 1.shk

TYPE 1 Double Wishbone, damper to lower wishbone

X (mm)	Y (mm)	Z (mm)	
-125.00	230.00	425.00	POINT (1) = Lower wishbone front pivot
125.00	230.00	375.00	POINT (2) = Lower wishbone rear pivot
-2.00	550.00	235.00	POINT (3) = Lower wishbone outer ball joint
-180.00	225.00	575.00	POINT (5) = Upper wishbone front pivot
180.00	225.00	480.00	POINT (6) = Upper wishbone rear pivot
0.00	520.00	380.00	POINT (7) = Upper wishbone outer ball joint
-30.00	430.00	330.00	POINT (8) = Damper wishbone end
0.00	240.00	720.00	POINT (9) = Damper body end
70.00	490.00	285.75	POINT (11) = Outer track rod ball joint
-15.00	200.00	450.00	POINT (12) = Inner track rod ball joint
0.00	240.00	720.00	POINT (16) = Upper spring pivot point
-30.00	430.00	330.00	POINT (17) = Lower spring pivot point
0.00	577.80	285.75	POINT (18) = Wheel spindle point
0.00	622.30	285.75	POINT (19) = Wheel centre point
4030.00	440.00	195.00	POINT (20) = Part 1 C of G
4170.00	520.00	450.00	POINT (21) = Part 2 C of G
4230.00	525.00	220.00	POINT (22) = Part 3 C of G
4130.00	720.00	275.00	POINT (23) = Part 4 C of G

STATIC VALUES

CAMBER ANGLE	(deg):	0.00
TOE ANGLE (SAE) (+ve TOE IN)	(deg):	0.00
TOE ANGLE (PLANE OF WHEEL)	(deg):	0.00
CASTOR ANGLE	(deg):	0.79
CASTOR TRAIL (HUB TRAIL)	(mm):	-1.30
CASTOR OFFSET	(mm):	5.24
KINGPIN ANGLE	(deg):	11.69
KINGPIN OFFSET (AT WHEEL)	(mm):	82.80
KINGPIN OFFSET (AT GROUND)	(mm):	23.68
MECHANICAL TRAIL	(mm):	5.24
ROLL CENTRE HEIGHT	(mm):	337.12

GENERAL DATA VALUES

TYRE ROLLING RADIUS	(mm):	285.75
WHEELBASE	(mm):	1450.00
C OF G HEIGHT	(mm):	450.00
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:		

Fig 3.1: Iteration 1

2) Observations

Parameters	Results
Turning radius(mm)	2719
Inner tire angle	35.92°
Outer tire angle	26.21°
Percentage Ackermann	64.46%

Table 3.1

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B. Iteration 2

1) Coordinates

FRONT SUSPENSION FILENAME: final front 1.shk

TYPE 1 Double Wishbone, damper to lower wishbone

X (mm)	Y (mm)	Z (mm)		
-125.00	230.00	425.00	POINT (1)	= Lower wishbone front pivot
125.00	230.00	375.00	POINT (2)	= Lower wishbone rear pivot
-2.00	550.00	235.00	POINT (3)	= Lower wishbone outer ball joint
-180.00	225.00	575.00	POINT (5)	= Upper wishbone front pivot
180.00	225.00	480.00	POINT (6)	= Upper wishbone rear pivot
0.00	520.00	380.00	POINT (7)	= Upper wishbone outer ball joint
-30.00	430.00	330.00	POINT (8)	= Damper wishbone end
0.00	240.00	720.00	POINT (9)	= Damper body end
78.00	490.00	285.75	POINT (11)	= Outer track rod ball joint
20.00	200.00	450.00	POINT (12)	= Inner track rod ball joint
0.00	240.00	720.00	POINT (16)	= Upper spring pivot point
-30.00	430.00	330.00	POINT (17)	= Lower spring pivot point
0.00	577.80	285.75	POINT (18)	= Wheel spindle point
0.00	622.30	285.75	POINT (19)	= Wheel centre point
4030.00	440.00	195.00	POINT (20)	= Part 1 C of G
4170.00	520.00	450.00	POINT (21)	= Part 2 C of G
4230.00	525.00	220.00	POINT (22)	= Part 3 C of G
4130.00	720.00	275.00	POINT (23)	= Part 4 C of G

STATIC VALUES

CAMBER ANGLE	(deg):	0.00
TOE ANGLE (SAE) (+ve TOE IN)	(deg):	0.00
TOE ANGLE (PLANE OF WHEEL)	(deg):	0.00
CASTOR ANGLE	(deg):	0.79
CASTOR TRAIL (HUB TRAIL)	(mm):	-1.30
CASTOR OFFSET	(mm):	5.24
KINGPIN ANGLE	(deg):	11.69
KINGPIN OFFSET (AT WHEEL)	(mm):	82.80
KINGPIN OFFSET (AT GROUND)	(mm):	23.68
MECHANICAL TRAIL	(mm):	5.24
ROLL CENTRE HEIGHT	(mm):	337.79

GENERAL DATA VALUES

TYRE ROLLING RADIUS	(mm):	285.75
WHEELBASE	(mm):	1450.00
C OF G HEIGHT	(mm):	450.00
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:		

Fig 3.2: Iteration 2

2) Observations

Parameters	Results
Turning radius(mm)	2318
Inner tire angle	38.20°
Outer tire angle	27.43°
Percentage Ackermann	75.32 %

Table: 3.2

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C. Iteration 3

```
FRONT SUSPENSION       FILENAME: final front 1.shk

TYPE 1 Double Wishbone, damper to lower wishbone

      X          Y          Z
     (mm)       (mm)       (mm)
-125.00     230.00     425.00    POINT (   1 ) = Lower wishbone front pivot
 125.00     230.00     375.00    POINT (   2 ) = Lower wishbone rear pivot
  -2.00     550.00     235.00    POINT (   3 ) = Lower wishbone outer ball joint
-180.00     225.00     575.00    POINT (   5 ) = Upper wishbone front pivot
 180.00     225.00     480.00    POINT (   6 ) = Upper wishbone rear pivot
   0.00     520.00     380.00    POINT (   7 ) = Upper wishbone outer ball joint
 -30.00     430.00     330.00    POINT (   8 ) = Damper wishbone end
   0.00     240.00     720.00    POINT (   9 ) = Damper body end
  76.00     495.00     285.75    POINT (  11 ) = Outer track rod ball joint
  30.00     200.00     450.00    POINT (  12 ) = Inner track rod ball joint
   0.00     240.00     720.00    POINT (  16 ) = Upper spring pivot point
 -30.00     430.00     330.00    POINT (  17 ) = Lower spring pivot point
   0.00     577.80     285.75    POINT (  18 ) = Wheel spindle point
   0.00     622.30     285.75    POINT (  19 ) = Wheel centre point
4030.00     440.00     195.00    POINT (  20 ) = Part 1 C of G
4170.00     520.00     450.00    POINT (  21 ) = Part 2 C of G
4230.00     525.00     220.00    POINT (  22 ) = Part 3 C of G
4130.00     720.00     275.00    POINT (  23 ) = Part 4 C of G

                        STATIC VALUES
                  CAMBER ANGLE  (deg):     0.00
 TOE ANGLE (SAE) (+ve TOE IN)  (deg):     0.00
   TOE ANGLE (PLANE OF WHEEL)  (deg):     0.00
                  CASTOR ANGLE  (deg):     0.79
     CASTOR TRAIL (HUB TRAIL)    (mm):    -1.30
                CASTOR OFFSET    (mm):     5.24
                KINGPIN ANGLE  (deg):    11.69
    KINGPIN OFFSET (AT WHEEL)    (mm):    82.80
   KINGPIN OFFSET (AT GROUND)    (mm):    23.68
            MECHANICAL TRAIL    (mm):     5.24
          ROLL CENTRE HEIGHT    (mm):   337.82

                    GENERAL DATA VALUES
       TYRE ROLLING RADIUS    (mm):   285.75
                 WHEELBASE    (mm):  1450.00
             C OF G HEIGHT    (mm):   450.00
   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:
```

Fig Iteration 3

1) Observations

PARAMETERS	VALUES(mm)
Track width (mm)	1244.6
Wheelbase (mm)	1450
Rack Length (mm)	400
Rack Position (w.r.t stub axle)	30
Rack Travel (mm)	41
Steering Arm Length (mm)	115.4

$$\frac{1}{Steering\ Ratio} = \frac{\sin^{-1}\left[\frac{C - Factor}{Steering\ Arm\ Length}\right]}{360^{\circ}}$$

Considering steering ratio = 4.5 and C-factor as follows:-

C-FACTOR	Steering Arm Length
70	71
75	76.15
80	81.23
85	86.31
90	91.38
95	96.46
100	101.54

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For example

$$\frac{1}{4} = \frac{\sin^{-1}\left(\frac{TD}{X}\right)}{360^{\circ}}$$

= 71

IV. DESIGN SPECIFICATIONS

The selected design with all the calculations is given below

A. Ackermann Law

It states that, whenever a vehicle takes a turn, the projections of the stub axle of front wheels will meet at a point on the proj8ection of rear axle, called instantaneous centre.

For stabilizing the vehicle, the system should also follow correct steering angle. This states that for taking a proper turn with all the tires rolling on the ground having maximum contact patch, the angle turned by the inner tire should be greater than the angle turned by the outer tire. It gives a relation between inner tire angle, outer tire angle, wheelbase, and track width, which are the basic parameters to decide the stability and position of the instantaneous centre, followed by turning radius.

The mathematical expression for Ackermann is as follows;

$$cot\emptyset - cot\theta = \frac{Track\ Width}{Wheel\ Base}$$

The values of track width and wheelbase that were finalized from the iterations are given below;

Track width = 1244.6 mm

Wheelbase = 1450 mm

The value of inner tire angle turned is found by the geometry and the value of outer tire angle is found by the Ackermann's Mathematical expression. This is done to verify the outer tire angle turned by Mathematical expression with outer tire angle turned geometry.

From Geometry Inner tire angle turned = Ø = **48.20^{0}**

cot (48.2) - cot (Θ) = $\frac{1244.6}{1450}$

Therefore, Outer tire angle = Θ = **29.74^{0}**

is approximately same as that of the value which we got from the geometry simulation on Lotus).

B. Turning Radius

The Turning Radius determines the minimum value of radius in which a vehicle can take a turn. To have good manoeuvrability smaller turn radius is required with stability and without violating Ackerman law. Following formula is used to calculate the Turning Radius;

Turning Radius (inner)

$$\frac{Wheelbase}{sin\emptyset} - \frac{Track\ width - Pivot\ Distance}{2}$$

Turning Radius (Outer)

$$\frac{Wheelbase}{\sin\theta} - \frac{Track\ Width - Pivot\ Distance}{2}$$

Where

Pivot distance = 1155.6 mm

And thus, the Inner and Outer Turning Radius are Calculated which are shown below;

Inner tire Turning Radius = 1910mm

Outer tire Turning Radius = 2878.50mm

C. Ackermann Percentage

The Ackermann Percentage tells us the behaviour of a vehicle while turning i.e. Understeer, Oversteer or Neutral steer. It also shows us the limit up to which the Ackermann law is followed. And we tried to keep the Ackermann Percentage value closest to 100%. For

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calculating Ackermann Percentage, the Ackermann Value is needed to be calculated first. Formula for calculating Ackermann Value is shown below:-

Ackermann value = $\tan^{-1}\left\{\frac{Wheelbase}{\frac{Wheelbase}{\tan\theta}-Trackwidth}\right\}$

By Calculating;

Ackermann value = 48.26^0

Now we are able to calculate the Theoretical Ackermann Percentage by following formula;

Ackermann Percentage = $\frac{\theta(actual)}{Ackerman\ Value} \times 100$

Hence, Ackermann Percentage = 99.89%.

D. C-Factor

C-Factor is the linear distance travelled by rack in one rotation of the pinion. To make steering system more responsive. The value of C-Factor should be as large as possible But, it should be kept in optimum range to keep the Pinion diameter in limit.

C-factor = $\frac{Rack\ Travel}{Lock\ to\ Lock\ angle}$

By Calculating, we get C-Factor = 101 mm

The Steering Ratio is the ratio of Angle of Steering Wheel Rotation to Angle Deviated by the Tire. It's the measure of responsiveness of steering system. For a steering system with rack and pinion as the steering gears, steering ratio calculated by the formula as follows;

$$\frac{1}{Steering\ ratio} = \frac{\sin^{-1}\left[\frac{C-factor}{Steering\ Arm\ Length}\right]}{360^\circ}$$

Therefore, values of C-factor and Steering Arm Length are as follow;

C-factor= 101 mm

Steering Arm Length = 111.71 mm

We get Steering Ratio as;

Steering ratio = 5.6:1

This value of steering ratio shows that the system designed is responsive and will ease the driver to control the vehicle.

E. Steering Effort

It is the amount of force applied at the steering wheel to steer a vehicle. If the force is less, it is more comfortable for driver to control a vehicle. Various parameters influencing the value of steering effort are mechanical trail, scrub radius, lateral force and traction force.

Following parameters, we took from Steering Geometry

Mechanical trail = 5.33 mm

Scrub radius = 22.37 mm

For calculating the steering effort, we determined the steering characteristic of the vehicle by calculating the understeer gradient.

$\delta = \left(\frac{57.3 \times L}{R}\right) + \left(\frac{k \times V \times V}{g \times R}\right)$

Therefore,

k = 3.55.

The negative under steer gradient gives the oversteer characteristic of the vehicle.

Now Calculating the Moment on Tire;

Lateral force on front tires = 4000 N

Moments on Tire:

Lateral Force × Trail = 4000 × 5.33 = 21320 N-mm

Traction Force ×Scrub =555.23 × 22.37= 12420 N-mm`

Total Torque = 21320 + 12420 = 33740 N-mm

Now Calculating the Forces on Tie Rod;

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$$Force = \frac{Total\ Torque}{Steering\ Arm\ Length}$$

Force = 449.87N
Tie rod inclination (α) = 29.64
Force on rack = Force × cos 29.64
Force on rack = 390.96 N
Pinion radius = 16.95mm = 0.0169m
Torque on Pinion = Force on Rack × Pinion Radius
Torque on pinion = 6.626N-m
As we know that,
Torque on steering wheel = Torque on pinion
Therefore,
Torque on steering wheel = 6.62 N-m
Steering wheel radius = 150 mm = 0.15 m

$$Steering\ Effort = \frac{Torque\ on\ Steering\ Wheel}{Steering\ Wheel\ Radius}$$

Steering Effort = 44.17 N-m.

F. Rack and Pinion Gear Design

The main consideration while designing the gears of rack and pinion is to avoid the interference between Rack and Pinion and to minimize the backlash. So as to avoid any play in the system.
As we know that for no interference the module should be $m \leq R \times \sin^2 Ø$.
Where;
Ø = pressure angle

C-Factor = 101 mm = $2\pi \times R$

R = 16 mm and PCD = 32 mm

$m \leq 16 \times \sin^2 20^0$

$m \leq 1.85$

The module shouldbe less than or equal to 1.85.

Some standard module available are as follows:

1st choice	1.00	1.5	2
2nd choice	1.25	1.75	2.25

Therefore,
Module (m) = 1.5 mm
Standard Addendum = m = 1.5 mm
Standard Dedendum = 1.25 x m = 1.875 mm
Now we know that,
PCD = m x No. of Teeth on Pinion
No. of Teeth on Pinion = 21
To Calculate No. of Teeth on Rack using following formula;

$$\text{No. of Teeth on Rack} = \frac{C-Factor}{(m \times \pi)}$$

Teeth on rack = 21

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Centre distance = PCR of pinion + Rack Shaft Radii - Module
CD= 16 +16 -1.5 = 31.5 mm
After designing of rack and pinion they were analysed using ANSYS software which assured us that the teeth of Rack and Pinion will not fail with the factor of safety coming out to be 2.5.
Specifications of the Selected Geometry.

Parameters	Results
Turning radius	1910 mm
Inner tire angle (Ø)	48.2^0
Outer tire angle (Θ)	29.74^0
Percentage Ackerman	99.89%
Rack travel	82 mm
Steering arm length	111.72 mm
Tie rod length	332 mm

V. THEORETICAL DATA ANALYSIS

A. Spline Calculations

Formula for torque transmission is as follows;

$M_t = P_m \times A \times R_m$

Where;

$P_m = 6.5$ N/mm^2

$A = \frac{1}{2(D-d)} \times l \times n = 630$ mm^2

$Rm = \frac{(D+d)}{2} = 8.75$ mm

So, Maximum Torque we can transmit is

$M_t = 35.831$ N-m

And Torque Transmitted by Driver is 7000 N-mm which is safe.

Now Calculating Shear Stress on Splines as follows;

Considering FOS as 2.5 and Yield Strength as 276 MPa

$T_{permissible} = (0.5 \times S_{yt}) \div 2.5 = 55.2$ N/mm^2

Force is 777.78 N and Area is 17.5 mm^2

$T_{max.} = F \div A = 44.46$ N/mm^2

So,
$T_{max.} \leq T_{permissible}$ Therefore, design is safe.

B. Bolt Design Calculations

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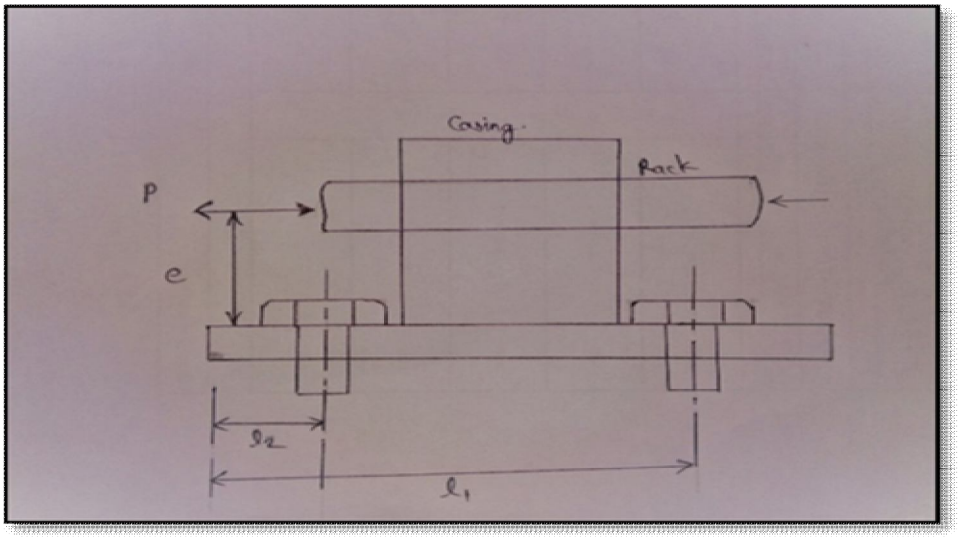


FIG 5.1: ROUGH SKETCH OF BOLT DESIGN

No. of Bolt(s) = 4
Considering same forces of rack on casing,
So,
Total force on casing = 443.25 + 777.78 = 1221 N
Therefore,
Force on each bolt= P = 305.25 N
Now calculating the Resisting force on bolts are as follows;
Considering, $T_{max.}$ = 70 N/mm^2.

$$P1' = \frac{P \times e \times l1}{2(l1 \times l1 + l2 \times l2)}$$

= 357.77 N

$$P_2'' = \frac{P \times e \times l2}{2(l1 \times l1 + l2 \times l2)} = 61.68 \text{ N}$$

Therefore,

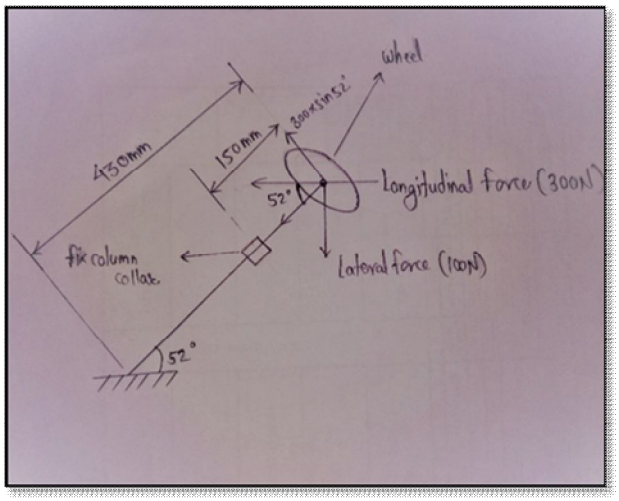


Fig 5.2 : Forces Acting on the Wheel

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$$T = \frac{P}{A} = \frac{305.25}{A}$$

$$\sigma_t = \left(\frac{P1}{A}\right) = \left(\frac{367.77}{A}\right)$$

$$T_{max.} = [(\sigma_t \div 2)^2 + T^2]^{0.5}$$

Therefore,

$$\text{Area} = \left(\frac{\pi}{4}\right) \times d^2 = 6.71 \text{ mm}^2$$

d = 3 mm

But, for considering higher FOS we choose M8 Bolt.

C. *Steering Column Analysis for Lateral and Longitudinal Forces*

$$\sigma t = \frac{Syt}{FOS} = 124 \text{ N/mm}^2$$

$$\sigma_{b(longituinal)} = \frac{(Mb \times y)}{I} = 105.92 \text{ N/mm}^2$$

$$\sigma_{b(lateral)} = 44.80 \text{ N/mm}^2$$

$\sigma_{b(longitudinal)}$ and $\sigma_{b(lateral)} \leq \sigma_t$

Therefore, the design is safe.

D. Material and Manufacturing Processes of Different Components.

The material selection and the manufacturing processes involved in the process are as follows;

1) *Rack:* Rack design is of composite material to reduce the weight. It is composition of steel (EN8) and Nylon 6.

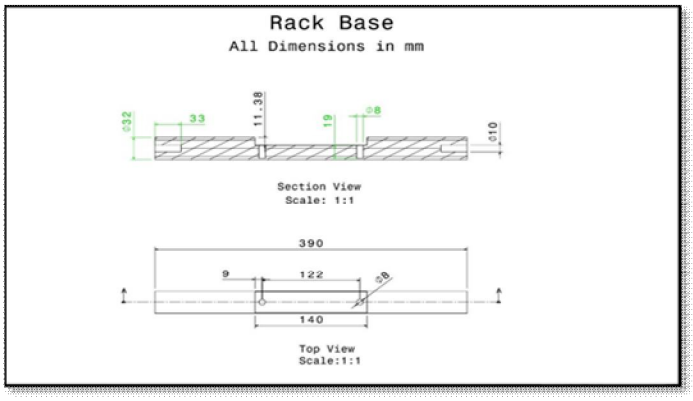


Fig 5.3: Dimensions for Rack Base

a) Material selected for Rack plate is bright EN8 steel. We choose EN8 because of its higher wear resistance, strength and low cost.
b) Material selected for Rack Base is Nylon 6. We choose nylon 6 because it is lighted in weight
c) On the shaft, turning and facing operations were done by lathe machine
d) Holes of M10 were made on both the ends of the rack shaft
e) Milling is done on rack base to make a pocket to fit rack plate in it
f) Milling is also done for making teeth on rack plate
g) Through hardening was done on rack plate.

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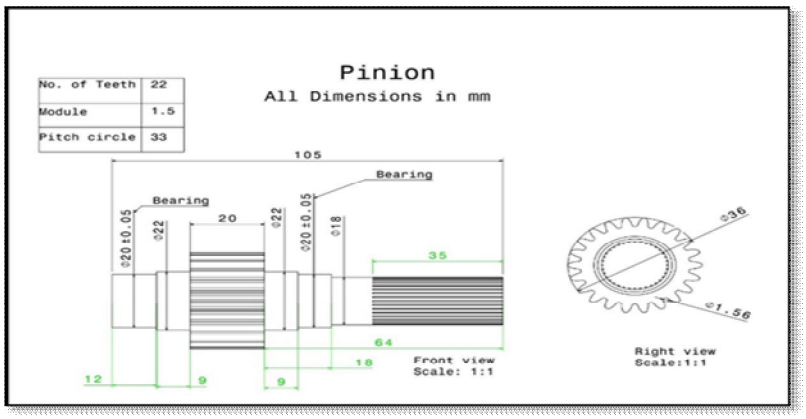


Fig 5.4: Draft for Pinion Gear



Fig 5.5 : Pinion Manufactured with the Splines

2) Casing

a) Material : The material selected for the manufacturing of rack and pinion casing was Nylon 6. This material was selected by analysing the theoretical and experimental data of material to reduce the weight of component.

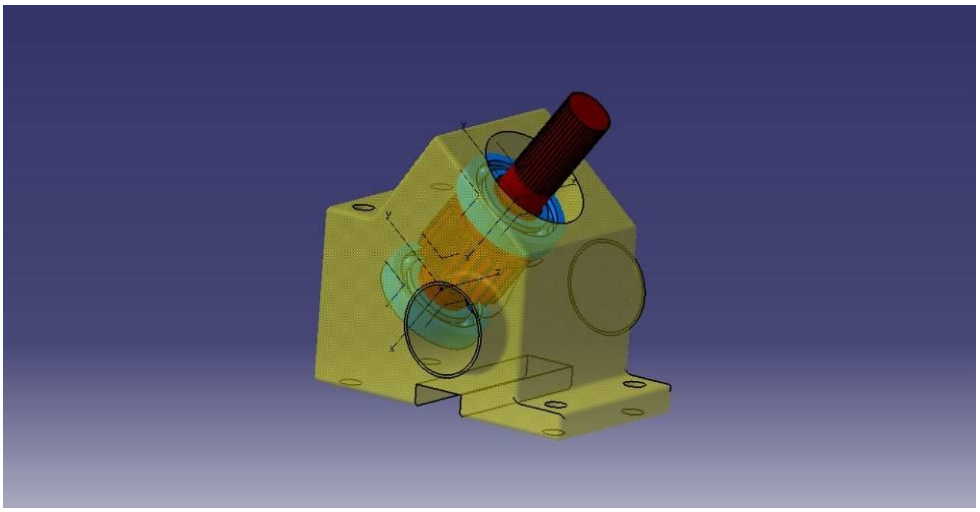


Fig 5.6 : Casing (CAD Model)

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b) Manufacturing

i) The operation is done on Vertical Milling Centre (VMC) Machine. This machine is selected to reduce the cost and time required.

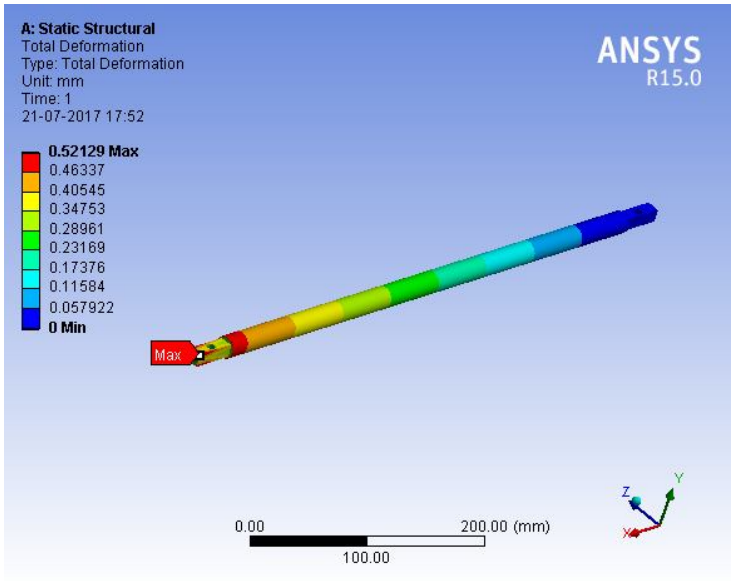


Fig 5.7: Structural Analysis of Tie Rod

3) Manufacturing

a) Selected pipe is cut into required number of pieces (two pieces) of same size.

b) Turning and facing operation is done on shaft for bushing on lathe machine.

c) Press fitting of bushes on tie rod ends is done by Hydraulic Press Fitting Machine.

d) Tungsten Inner Gas (TIG) welding is done after press fitting on bushes and pipes of tie rod.



Fig 5.8 : Tie Rods

4) Steering Arm

a) Material

a) The material selected for Steering arm is Al7075 T6 due to its high strength by reducing its weight.

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The operation is done on Vertical Milling Centre (VMC) Machine. This machine is selected to reduce the cost and time required.

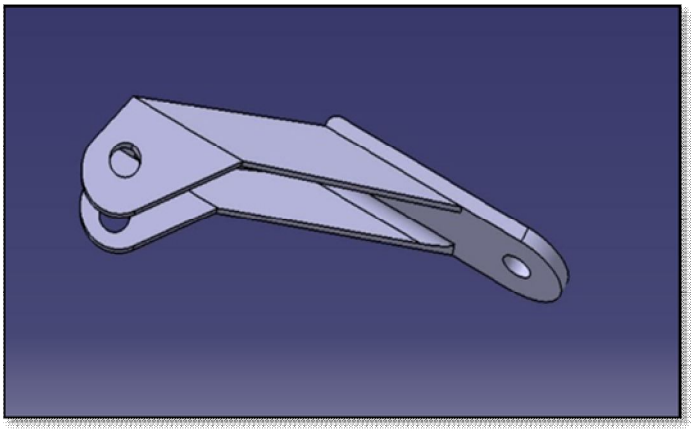


Fig 5.9 : Steering Arm (CAD Model)

5) Nuts And Bolts

a) Nuts and bolts of standard grade and size are purchased with dimensions according to different functions.

6) Bearing

a) Bearing of standard size were selected and also the design of parts is made by considering the size of the bearing. And the size of bearing is as follows

I.D. = 20 mm
O.D. = 42
Width = 8 mm

7) Joints: Standard Joint Selected is Heim-joint and it is used on both the ends of Tie Rod. The size of Heim-joint is M8, This Joint is chosen because it has 12 Degree of Freedom and light in weight.

Manufactured U-joint is selected for connecting tie rod and rack. Material selected for it, is EN8. And Manufacturing of this

VI. VALIDATION

A. Rack Travel

1) Steering wheel and pinion were set to the centre position and a marking is made on the rack with a marker.

2) The steering wheel is turned to either of the side to full lock position and then another marking is made at position where last marking was taken

3) After that the distance between both the markings is measured and the distance between them tell us the rack travel.

B. Turning Radius

1) It is implemented practically by driving the vehicle and measuring the smallest radius the vehicle could turn with the help of measuring tape.

2) The patches of the wheels helped us in taking the measurements.

3) In static condition it is validated by turning the wheel to full lock condition and then projecting the stub axles of the front wheels to the rear axle projection.

4) Then the distance is measured between the intersection point and the certain point on the vehicle.

C. Angles turned by the tires

1) The projections of the tire centres were mark on the ground in static straight position of the vehicle.

2) Then the tires were turn to full steer condition and another projection is made on the ground at tire centre.

3) The angle between both the projection is measured, which is the angle turned by the tire.

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D. Steering Wheel Rotation

1) Steering wheel rotation is validated by turning the steering wheel from one side to another side full lock positions.
2) The angle turn by the steering wheel is measure and is converted into number of rotations.

E. Ackermann Percentage

1) Angles of tire deviation are measured and it is used to calculate 'Ackermann Percentage'.
2) Then the theoretical value of Ackermann percentage is compare with practical value.

F. Steering Ratio

1) It is validated by turning the steering wheel to full lock condition to either side.
2) Then the angle by which steering wheel rotates and the angle deviates by the tire is measured.
3) The ratio of degrees of steering wheel rotation to the degrees of tire deviation gives us the steering ratio on which the vehicle is going to respond.

VII. DESIGN COMPARISON TABLE

The difference between the designed value and the actual is due to the errors made in the manufacturing phase or errors due to play or clearances.

Parameters	Designed Value	Actual Value
Trackwidth	1244.6 mm	1244.6 mm
Wheelbase	1450 mm	1450 mm
Turning radius	1970 mm	1910 mm
Ackermann percentage	94 %	98 %
Lock to lock steering wheel rotation	0.82	0.82
Inner Tire deviation angle	44.16^0	48.26^0
C-factor	101 mm	100 mm
Steering Ratio	5.5:1	5.5:1
Steering Effort	40 N	44 N

Table 7.1: Design Comparison

VIII. CONCLUSION

From the testing phase of the vehicle, we got to know that the designed value of the parameters and the actual value of the parameters are close enough to function without failures and satisfy our required needs. Hence, the designed system works as per the driver input and make the vehicle more responsive to act over the driver's input and to control it more easily.

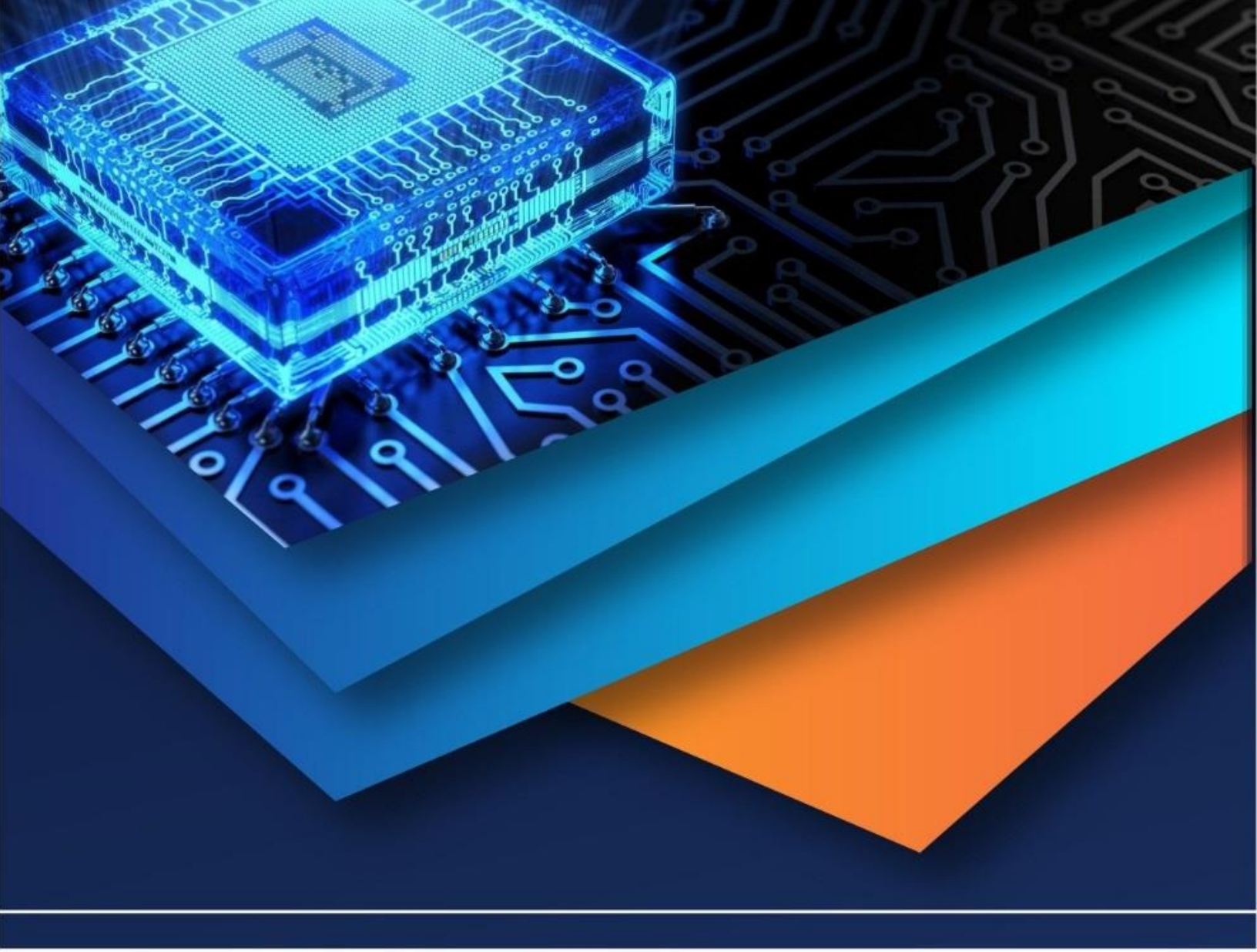
IX. ACKNOWLEDGMENT

We are grateful to all those who helped us out to make this article possible. All the efforts taken by the members are remarkable and we will remain obliged to it.

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