Design of Rack and Pinion Steering System for an All-Terrain Vehicle

[1]Gaurav P. Keer, [2]Sanket S. Keer, [3]Sandesh S. Jogdand, [4]Saurabh S. Dalvi, [5]V. Murali Mohan
[1,2,3,4]Students, Finolex Academy of Management & Technology, Ratnagiri 415639,
[5]Associate Professor, Finolex Academy of Management and Technology, Ratnagiri 415639.

Abstract: This paper is focused on elaboration of design processes behind steering system in All- Terrain Vehicle (ATV) to utilized in application of agriculture field. The objective of implementation of this steering system is to reduce turning radius and provide directional stability while maneuvering. The steering system has designed to meet weight reduction requirement along with keeping in mind of driver comfort. Design of the steering system includes calculations of the optimized steering parameters such as steering arm angle and turning radius and simulation of the steering components by considering actual behavior of the vehicle. This work analysis helps us to study and analyze the procedure of vehicle steering system designing and to identify performance affecting parameters of steering such as gear module and turning wheel lock angles. The steering system is thus designed in a very unique way by compelling factors like bending, wear, torque to overcome Steering Axis Inclination (SAI) and effective load.

***Index Terms* – Steering, ATV, Rack & Pinion**

I. Introduction

The All-Terrain Vehicle (ATV) was initially developed in the 1960's as a farm town vehicle in isolated, mountainous areas. During spring thaws and rainy seasons, steep mountainous roads were often impassable with conventional vehicles. Heavy tractors and tillers are unsafe in operation due to its bulky size. Therefore, it has applications in mountain terrain regions. So, to overcome these limitations, an All-Terrain Vehicle (ATV) concept is adapted for implementation work.

Work performance of vehicle design conditions satisfies the true rolling which was calculated from Ackermann steering law. Calculated values were further used to design the rack rod and modified rack tested using 3D modelled analytical simulation software of SOLIDWORKS for various modes of failure. The length of the steering column has varied according to the roll-cage dimensions and mounting points for steering system on the roll-cage designed. The steering system is in direct contact with the tires, hence it subjected to extensive different forces. Hence it has to imperative that the design is tested for failure under such conditions.

II. Literature Review

Saurabh Borse et.al. [1] conceptual design and fabricated steering device for ATV and enhance its yoke-nut assembly for better performance. This purpose has considered for kinematic behavior of the system during its operation. The concept of Yoke-nut assembly of the Rack and pinion system that keep the rack in proper mesh with pinion which highlighted main attraction of scope work. A simulation and numerical analysis was performed and it resulted depicts performance enhancement of the system due to its Yoke-nut assembly. This conceptual study and design of rack and pinion provided scope for further enhancement of subject work.

R. N. Jazar [2]studied upon steering dynamics of two wheel and four-wheel steering systems. He states formal equations for every parameter in the steering system which are derived conceptually for case work implementation. They explained steering geometry for every steering mechanism. Therefore, it helped to understand and adapt its calculation work for better implementation in work.

III. Design Procedure

As per concerns of SAE BAJA standard Rulebook [3], their restriction has come to enhance ergonomic design work. Therefore, under limited specification maximum ergonomic design and space constraints are assume and designed as shown in figure below.

1. Front track width: 57 inches
2. Rear track width: 51 inches
3. Wheelbase: 64 inches

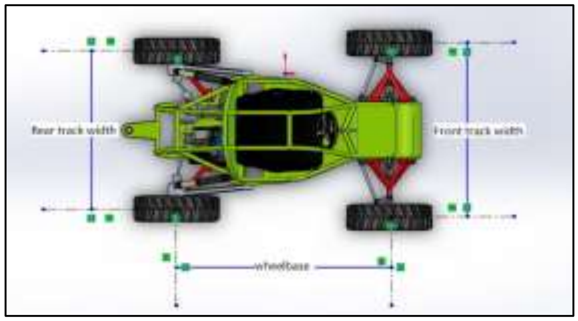


Fig 1. Basic parameters of vehicle

After studying the conditions of Ackerman steering geometry[3] of correct and slip free steering and trapezoidal mechanism[3], the inner wheel and outer wheel lock angles were calculated.

Inner wheel lock angle = 45^0
Outer wheel lock angle = 28.01^0

Followed by this, calculations were performed to calculate the turning radius of the vehicle. Turning radius is the radius of the smallest circle traced by the center of gravity of the vehicle, after locking of the wheels while turning.

Turning radius (R_{cg}) = 2.42 m

Design of rack and pinion steering system

3.1. Selecting 20^0 full depth involute gear profile

The reason behind selecting this profile are as follows: -

a. Avoids undercutting
b. Reduces interference
c. This value of pressure angle makes tooth slightly broader at the root, making tooth stronger and increase its load carrying capacity.

Table 1

Parameter	**Value**
Pressure angle (α)	20^0
Addendum (h_a)	m
Dedendum (h_a)	1.25m
Clearance (c)	0.25m
Working depth	2m
Whole depth	2.25m
Tooth thickness	1.5708m

Table 1 comprises of gear tooth proportions of standard 20^0 full depth involute gear profile

3.2. Calculation for minimum no. of teeth on pinion

The minimum number of teeth required on pinion in order to avoid the interference were computed using following relation:

$$Z=\frac{2\frac{a}{m}}{\text{SIN}\,\alpha^2} \qquad \text{Eq 1.}$$

Substituting values in above equation
Minimum number of teeth on pinion are :18.
Standard gear ratio for rack and pinion = 1.5
Therefore, No. of teeth on rack = 18*1.5 = 27

3.3. Selection of material

Material selected – Al T6 7075

Table 2

Physical properties	Metric
Density	2.81 g/cc
Mechanical properties	

Brinell Hardness	150
Ultimate tensile strength	572 MPa
Tensile yield strength	503 MPa
Modulus of elasticity	71.7 GPa
Poisson's ratio	0.33

Table 2 consists of physical and mechanical properties of Aluminum alloy T6 7075

$\sigma_t = S_{ut}$/ Factor of safety = 572/3 = 190.67 N/mm^2

3.4 Calculation of beam strength

Beam strength of gear tooth is the maximum tangential load that gear tooth can take without tooth damage.

Assumptions in analysis of beam strength

1. The full load is acting at the tip of a single tooth
2. The effect of radial force is neglected
3. The load is uniformly distributed across the full-face width
4. Effect of stress concentration is neglected
5. Frictional force due to teeth sliding are neglected

Lewis form factor (Y) = π(0.154 – (0.912/Z))

$Y_{pinion} = 0.324$

$Y_{rack} = 0.377$

Strength factor(Fb) = σ_y *Y

Fb_{pinion} = 61.77 N/mm^2

Fb_{rack} = 71.88 N/mm^2

Here strength factor of pinion is found to be lesser than that of rack. Hence, Pinion is the weaker element.

Therefore, Design of Pinion has to be implemented.

The beam strength is given by,

$P_b = \sigma_t{*}b{*}m{*}y$

Where σt = 190.67 N/mm^2

b= Face width = 10*m

$P_b = 617.77m^2$

3.5. Calculation of Wear strength

The failure of the gear tooth due to pitting occurs when the contact stresses between two meshing teeth exceed the surface endurance strength of the material. Pitting is a surface fatigue failure, characterized by small pits on the surface of the gear tooth. In order to avoid this type of failure, the proportions of the gear tooth and surface properties such as surface hardness, should be selected in such a way that the wear strength of the gear tooth is more than the effective load between the meshing teeth. The analysis of the wear strength was done by Earle Buckingham, in his paper.

Buckingham's theory is based on Hertz theory of contact stresses. From his theory we can conclude that Wear strength of a gear tooth is the maximum tangential load the gear tooth can take without causing a pitting failure.

$P_w = b.Q.dp.k$

Q = Ratio factor for external gear pair

= [(2.Zg) / (Zg + Zp)]

Here Zrack = 27, Zp = 18 So that Q = 1.14

dp= m*Zp

$P_w = 207.765m^2$

3.6. Calculation of Effective load

Effective load was calculated based on functional requirement of the system.

While cornering the drive applies an effort to steer the vehicle, it is a necessary effort require to provide the alignment torque to overcome the friction couple generated by the reaction forces acting at point of contact of wheel with the surface of road.

The friction couple is generated because of offset between wheel contact point and traction point and also because of castor trail.

While setting geometry for suspension system following parameters were finalized to improve system performance

1. Steering axis inclination: 4^0
2. Castor angle: 6^0

3. Scrub radius: 67.31 mm
4. Castor trail: 24.31 mm

Assuming human tendency to apply brakes while turning, we considered 50% distribution of weight.

Thus, the reaction forces are computed based on above assumptions are calculated as follow:

So, F_T = 50% weight of vehicle = 1471.5 N on both suspension joints.

Aligning torque required to overcome the steering axis inclination (SAI) is given by;

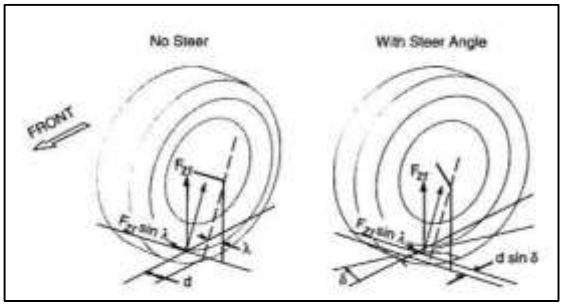


Fig.2 Lateral forces on wheel while cornering

$F_{ZR} = 0.5\ (F_T)$.

Moment to overcome offset of SAI axis

$= (F_{ZR} + F_{ZL}).\ d.\mathrm{Sin}\lambda.\mathrm{Sin}\delta$

$= (1471.5)*67.31*\mathrm{Sin}4*\mathrm{Sin}45.3$

$= 4911.01$ N-mm

The friction couple generated while cornering is given by;

Friction couple per wheel $= m*v^2/\ 4*R_{cg}$

$= [300*((5/18)*v)^2]/\ 4*2.42$

$= 2152.20$ N

So, Aligning torque for friction couple (M_f) is given by;

Aligning torque (M_f) $= 2480.45*(R/2)*\ \mathrm{Sin}\mu.*\ \sin\delta$

Where R is the tire radius

$= 2152.20*(279)*\sin(6)*\sin(45.03)$

$= 44405.26$ N-mm

$M_{eff} = (M_f + M_T)$

$= (4911.01 + 44405.26)$

$= 49316.2737$ N-mm

$M_{eff} = F_t$ * steering arm length

Hence; $F_t = 493.16$ N

From design data book;

For accurate mounting and moderate shock

Ka = Application factor = 1

Km=Load concentrationfacto0= 1.3

Kv = velocity factor = 1

$P_{eff} = (Ka * Km)* F_t\ / Kv$

$P_{eff} = 641.11$ N

3.7. Estimation of module

As gear pair is weaker in wear than in bending so, system must be design against wear failure

Assuming factor of safety (N_f) = 1.8

For given system;

$P_w = N_f * P_{eff}$

$207.765m^2 = 1.8*641.11$

$m = 2.35$

Table 3

Parameter	Pinion
d (mm)	42.42

Addendum(ha)	2.35
Dedendum (hf)	2.9375
Face width(b)	23.5
Module(m)	2.35
Z_p	18

Table 4

Parameter	**Rack**
d (mm)	63.45
Addendum(ha)	2.35
Dedendum (hf)	2.9375
Face width(b)	23.5
Module(m)	2.35
Z_g	27

Table 3 and Table 4 consists of final gear proportions of designed Rack and Pinion respectively.

IV. CAD MODELS

Fig 3 Cad model of Rack and Pinion

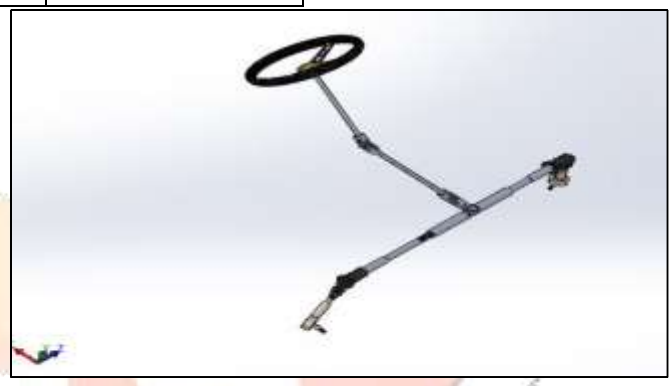
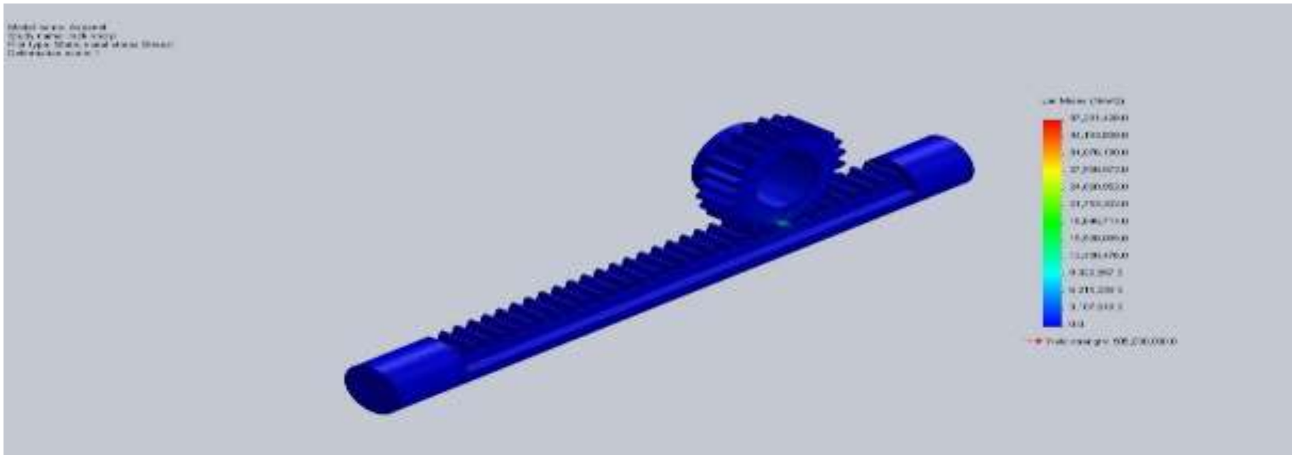


Fig 4 Cad Model of Steering assembly

Table 5

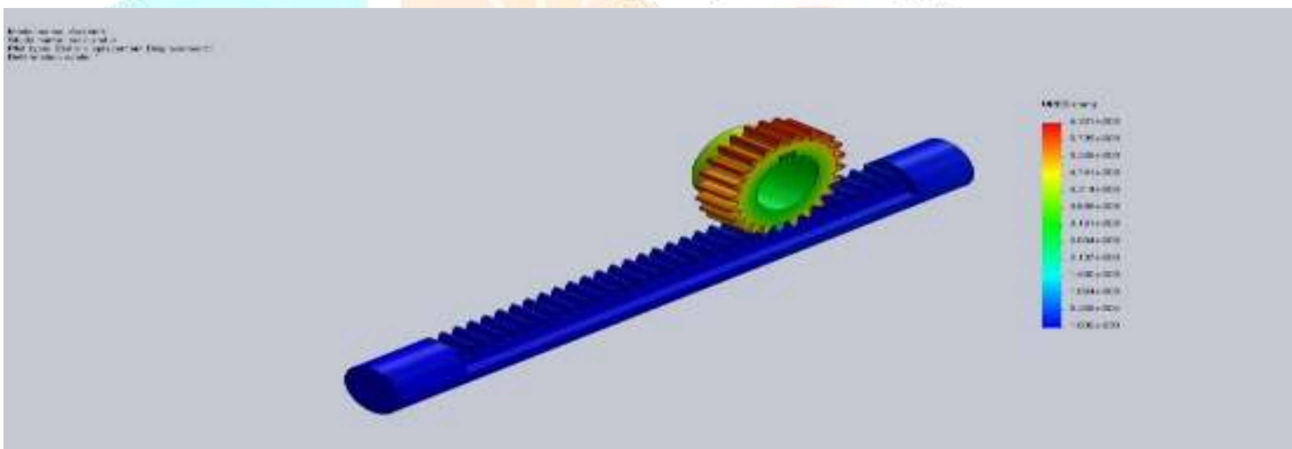
Name	**Type**	**Min**	**Max**
Stress1	VON: von Mises Stress	0.0353465 N/m^2 Node: 41852	9.42914e+007 N/m^2 Node: 40778

Name	**Type**	**Min**	**Max**



Assem4-rack and p-Stress-Stress1

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0 mm Node: 1	0.00632145 mm Node: 291265



Assem4-rack and p-Displacement-Displacement1

Table 5 is the analysis report of designed Rack and Pinion performed in SOLIDWORKS 2016
Maximum induced stress = 9.42914e+007 N/m^2
Tensile strength of material = 503 MPa

Factor of safety = Tensile strength / Max. Induced stress = 5.33

V. Conclusion

For the fulfilment of the application of an all-terrain vehicle in agricultural sector especially in the mountain terrain regions can be achieved by implementing firm design procedures. Consideration of every parameter is important and hence the research paper and the theory material are obtained. Following the standardized and optimum design procedure for the optimum outcome is the aim of the project. Design of rack and pinion has been implemented by consideration of bending and wear failures of the pinion. Safe value of gear module was estimated and with standard gear proportions, the parameters of rack and pinion were finalized. After analysing in simulation software, the factor of safety obtained was 5.33, which is positive value and safe. Hence, the design procedure followed is safe.

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