Design & Analysis of the Steering System for All-Terrain Vehicle

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Abstract— This paper embodies the design methodology for the steering system for an All-Terrain Vehicle. The steering system has the primary function of maneuvering the vehicle in the desired direction. The research paper gives an essential procedure to design the system for effective functionality. Also, the steering system was tested under different geographical terrains. The different software such as were used to ensure precise dimensions & accurate results. The vehicle with the above-mentioned system was made to participate in SAE BAJA INDIA 2019. The actual results validated by comparing with theoretical values obtained. The design procedure started with steering angle calculations. The Rack & pinion gears were designed. Theoretical values were validated with the help of 'Lotus Shark' software. The designed system was modeled using CAD software. Structural analysis was done to check whether the designed components will sustain critical loads. All obtained results were enlisted in this paper.

Keywords: All-Terrain vehicle, maneuvering, SAE BAJA INDIA, Steering system, Ackerman/ Anti-ackerman

I. Introduction

Mechanical Engineering is one of the broadest disciplines in engineering which overlaps many other disciplines. It provides a choice of innumerable field of interest and development for a student. Automobile Design was a common interest of the members of the project team. S Our interest led us to become members of the automotive society, SAEINDIA Collegiate Club of our college, through which we were introduced to the event BAJA SAEINDIA. BAJA SAEINDIA is a competition involving teams from all over the country wherein each team has a goal to design and build a rugged single seat, an off-road recreational four-wheel vehicle intended for sale to a non-professional, weekend off-road enthusiast. The design analysis and fabrication of the various systems in the All Terrain Vehicle (ATV) was divided amongst the 4 groups consisting of 4 members each. The paper provides details about the Steering System of an All Terrain Vehicle (ATV). ATV stands for All Terrain Vehicle. It is a vehicle with rear-wheel drive intended to use for recreational, agricultural, military purposes. The project is to design the Steering system.[1]It should Maneuver through various terrains with the help of required traction at the wheels at a particular rpm. The rack-and-pinion system consists of a linearly moving rack and pinion, mounted on the firewall or a forward cross member, which steers the left and right wheels directly by a tie-rod connection. The tie-rod linkage connects to steering arms on the wheels, thereby controlling the steering angle. With the tie-rod located ahead of the wheel center, it is a forward steer configuration.

II. Steering System Problem Statement

The design of the Steering is to be done keeping in mind the various dynamic events and the level of competition both competitions. The weight of the vehicle should be less to improve the performance of the vehicle. The design of the Steering for this year should be an upgrade on the Steering system used previously. For this purpose, the drawbacks during the design and manufacturing phase as well as the failures which occurred during the main event were thoroughly brainstormed. A typical Baja vehicle uses a Rack & pinion due to its obvious advantages for off-road vehicles. The design should guarantee better performance. The Steering system for 2018 should comprise of the following components:
STEER WHEEL• RACK • PINION • TIE RODS• STEER ARM• KNUCKLE

A. Objectives:

1) To MANEUVER the vehicle through different terrains
2) Design for optimum TURNING RADIUS keeping in mind the WHEELBASE, WHEEELTRACK, CASTER and SPEED requirements.
3) To avoid UNDERSTEER due to less TRACTION during events like MANEUVERABILITY. To efficiently INTEGRATE the steering system with other subsystems of the vehicle like the roll cage and FRONT SUSPENSION keeping in mind the DRIVER COMFORT.

III. Methodology

1) To decide wheelbase & wheel track of the vehicle
2) Survey of Various Steering gear mechanisms
3) The calculation for Turning radius & Steering Ratio.
4) Design and Analysis of Rack & Pinion Assembly
5) Subsystem Integration
6) Design and Selection of Steering components
7) Subsystem Integration
8) Design Validation and Testing

A. Calculations for Steering Angles:

Turning angles:-
Assume $\phi=41.70^\circ$
$\cot\theta-\cot\phi= b/l$
Where θ=inner steer angle ϕ=outer steer angle
b=track width=1320.8m l=wheelbase=1371.6mm
$\cot(\theta)-\cot(41.7)=1320.8/1371.6$
$\theta=25.62^\circ$

B. Ackerman Percentage:

Ackerman%=$(\theta-\phi)/\theta'$
Where θ'=Steer angle for 100% Ackerman
$\theta'=\tan^{-1}((1371.6/(1371.6/\tan(41.70)-1320.8))-41.70$
θ' =39.24

Ackerman %= -40.98%

C. *Design on Gears:*

For avoiding interference, the minimum number of teeth required on the smaller gear should be greater than 18.For 18 teeth, the diameter of pinion was >50mm, also, the size and weight of pinion increased. Thus, in order to reduce its weight, the number of teeth on pinion was chosen to be 26 after performing certain calculations.
Thus, no of teeth on pinion=26

D. *Pinion Calculations:*

In the rack & pinion mechanism, As there are not many rotations on gears, Wear is not observed. Gears are supposed to fail in bending. Hence, Pinion is always weaker.
[2]For bending failure,
$F_b=\sigma_b*b*m*Y$ (a)
$=S_{ut}/3*10m*m*(0.484-2.865/^{Z}{}_{p})$
$=635/3*0.3738*10m^2$
$=791.21m^2$
Effective load acting on gears
$F_{eff}=(K_a*K_m*F_t)/K_v$ (b)
Where K_v =Velocity factor
$K_v=3/(3+v)$
$=3/(3+v)$
$=3/(3+0.0245m)$
$F_{eff}=(3184.614+26.007m)/3m$
For safety against bending failure, From equation (a) & (b)
$F_b=N_f*F_{eff}$ (c)
Assume Factor of safety as 1.5
$791.221m^2=1.5*((3184.614+26.007m)/3m)$
m=1.25mm
Taking Standard value of module=1.25mm
$F_b=N_f*F_{eff}$
$1236.353=N_f*857.8994$
$N_f=1.4$
Therefore,
Number of teeth=26
Module=1.25mm
$PCD=m*Z_p=32.5mm$
Clearance circle Diameter $Db=0.94*D_p=30.55mm$
Addendum circle Diameter $D_a=Dp+2m=35mm$
Dedendum circle Diameter $D_d=Dp-2.5m=29.36$

E. *Rack Calculation:*

Assuming maximum turning angle steering wheel to be 150˚ for one side & Maximum turning angle for tires as 41.7˚,
Total steering wheel angle=300˚
Steering ratio=(Max turning angle for steering wheel)/(Max turning angle for tire)
Assume max turning angle for tyre=41.70˚
Steering ratio=3.6:1
Considering 18 teeth for pinion we got module 2 & FOS >2. Thus reducing module to 1.25 we achieved FOS of 1.4. We selected no of teeth on pinion=26. B=12.5mm
PCD=32.5mm
The total circumference of pinion=π*PCD
=π*32.5
=102.1017mm

For 360˚ rotation pinion travels 102.1017mm, thus for 150˚ rotates 42.54. Thus for total steering angle i.e., 300˚, total rack travel is 85.08mm.

IV. CALCULATIONS FOR INNER AND OUTER TURNING RADIUS

Ø= outer angle= 41.70˚
ϴ= inner angle= 34.26˚
Wheelbase(l) = 1371.6mm
Track width(c) = 1320.8mm (front)
(a)= 1270mm (rear)
R_{IF}= l/sinø - (a-c/2)
R_{IF}=1.969m
R_{OF} = l/sinϴ + (a-c/2)
R_{OF}=3.208m
R_{IR} = l/tanø - (a-c/2)
R_{IR}=1.464m
R_{OR} = l/tanϴ + (a-c/2)
R_{OR} =2.864m
R_{AVG}=2.012m

A. *Steering Gradient:*

Steering gradient is the parameter which is used to find the nature of steer of the vehicle i.e. understeer/oversteer/neutral steer.

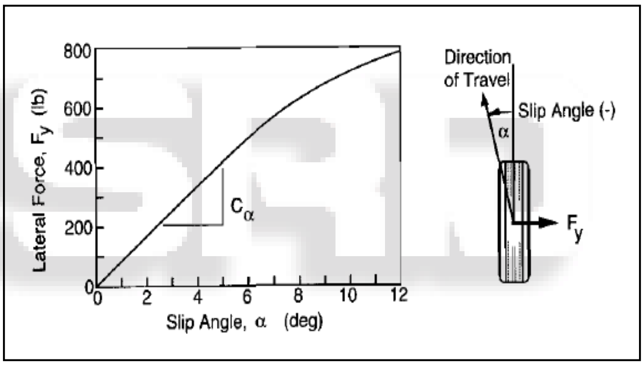

Fig. 1: Steering gradient

The above graph gives us the relation between lateral force for the front axle and the slip angle for the two wheels.[1]
From the calculations, we got the weights for the front and rear axle as:
F_{YF} = 82.66kg = 182.223 lb
F_{YR} = 117.34kg = 258.69 lb
For the above corresponding values of weights of front and rear axle, we found the slip angles for the respective axles from the above graph.
$\alpha_F = 2^{\circ}$
$\alpha_R = 2.83^{\circ}$
$\alpha_F < \alpha_R$
This Is the Condition for Oversteer Nature of Vehicle.

V. CALCULATION OF STEERING WHEEL FORCE

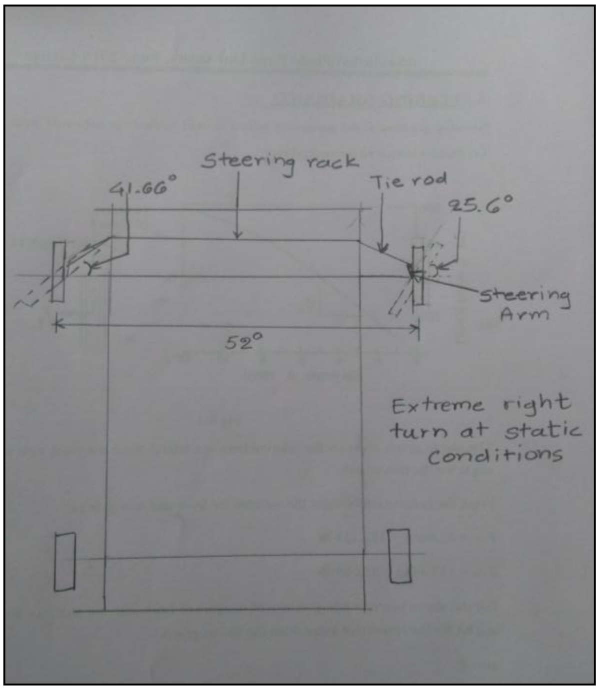

Fig. 2: Calculation of steering wheel force(top view)

A. *For top view:*

The angle between Steering Arm& Axle=43.3^0
The angle between tie rod & rack=41.633^0
For front view:-
The angle between Steering arm & axle=0^0(Horizontal)
The angle between tie rod & rack=16.5^0
The force required to push/pull tire at static conditions is μmg.

$F_{tyre}=\mu mg$ (d)
$=0.85\times41.33\times9.81=344.63N$

Resolving the obtained force into steering arm & tie rod inclinations
From equation (d)
Steering arm force $=344.63/\cos(43.3^0)=473.54N$ (e)
From equation (e)
Tie rod force $=[473.54/\cos(43.3^0-41.633^0)]/\cos(16.5^0)$ (f)
$=494.082N$
From equation (f)
Force on rack & pinion$=494.082\times\cos(16.5^0)\times\cos(41.633^0)$
$=354.081N$
Torque on pinion$=354.081\times32.5=11507.63N\text{-mm}$
Torque on steering wheel=Force applied × Radius of the steering wheel
$11507.63=F_{wheel}\times115$
$F_{wheel}=100N$

B. *Lotus Analysis:*

The theoretical values needed to compare with the practical condition. Here comes the role of validation. We preferred virtual validation over the manual because of a lack of data acquisition instruments. Further improvement can be done by using electronic sensors for measuring different parameters like turning radius, Toe change, Camber change, Steer travel.

We have used the 'Lotus shark' software for validating theoretical values. Suspension hard points were plotted for designing the geometry of wheels. Then steering parameters like steer travel, steer increment, wheelbase, wheel-track were put in the software. The obtained results are shown below. [3]

Toe change= -36º to 46º
Camber change= -1º to 4º
Ackerman %= -35% (Anti-ackerman)
Turning Radius=1.5m

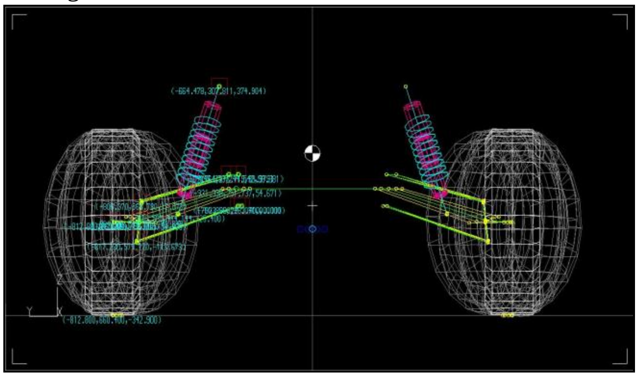

Fig. 3: Lotus Analysis interface

FRONT SUSPENSION - STEERING TRAVEL

TYPE 1 Double Wishbone, damper to lower wishbone

INCREMENTAL GEOMETRY VALUES

RACK TRAVEL (mm)	TOE ANGLE RHS (deg)	TOE ANGLE LHS (deg)	CAMBER ANGLE RHS (deg)	CAMBER ANGLE LHS (deg)	ACKERMANN (%)	TURNING CIRCLE RADIUS (mm)
-45.00	46.45	-36.84	-1.05	4.65	-35.06	1567.18
-40.00	38.88	-32.56	-1.23	3.78	-30.73	1924.35
-35.00	32.63	-28.37	-1.23	3.05	-27.96	2340.97
-30.00	27.09	-24.26	-1.17	2.42	-26.04	2862.58
-25.00	22.02	-20.19	-1.05	1.88	-24.67	3560.81
-20.00	17.26	-16.16	-0.89	1.41	-23.69	4574.17
-15.00	12.74	-12.14	-0.71	1.00	-23.01	6222.71
-10.00	8.38	-8.12	-0.50	0.63	-22.57	9463.89
-5.00	4.14	-4.08	-0.26	0.30	-22.33	19081.44
0.00	0.00	0.00	0.00	0.00	-22.25	0.00
5.00	-4.08	4.14	0.30	-0.26	-22.33	19081.44
10.00	-8.12	8.38	0.63	-0.50	-22.57	9463.89
15.00	-12.14	12.74	1.00	-0.71	-23.01	6222.71
20.00	-16.16	17.26	1.41	-0.89	-23.69	4574.17
25.00	-20.19	22.02	1.88	-1.05	-24.67	3560.81
30.00	-24.26	27.09	2.42	-1.17	-26.04	2862.58
35.00	-28.37	32.63	3.05	-1.23	-27.96	2340.97
40.00	-32.56	38.88	3.78	-1.23	-30.73	1924.35
45.00	-36.84	46.45	4.65	-1.05	-35.06	1567.18

Fig. 4: Lotus Output results

C. *CAE Analysis:*

Firstly components were designed in CATIA V5R21 Software. The components were then subjected to Virtual loading using ANSYS 18.0 Software. While doing static analysis, the rack was fixed. The pinion shaft was given a frictionless support as it is given bearing support. The pinion is subjected to a momentum of 11507 N-mm. For Steering column, bearing support is given at section C. At section A force of 100 N is subjected through steering wheel &at section B force of 100 N is applied to pinion. For Steering Arm, Considering same case as tie rod. A pull force of 600 N is subjected at section B & the section A is fixed. Tetrahedron Elements and Hex dominant Elements were used to get the maximum deflections and maximum stresses. Analyses were carried out with the number of nodes and the size of elements by auto generated. From this we can get the max deformation and max stress induced. The strength of the designed components was identified by this analysis. The obtained results were as follows:-

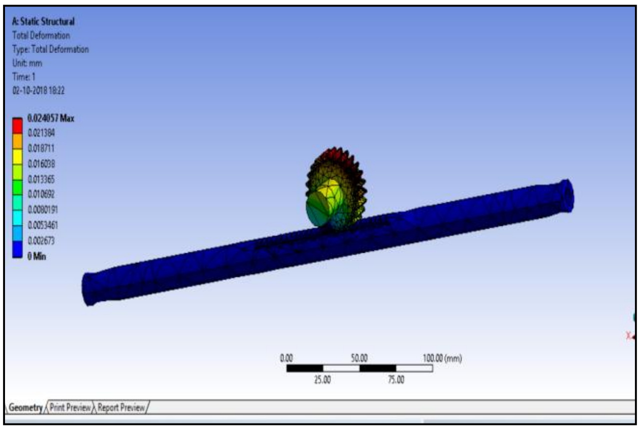

Fig. 5: Rack Pinion deformation

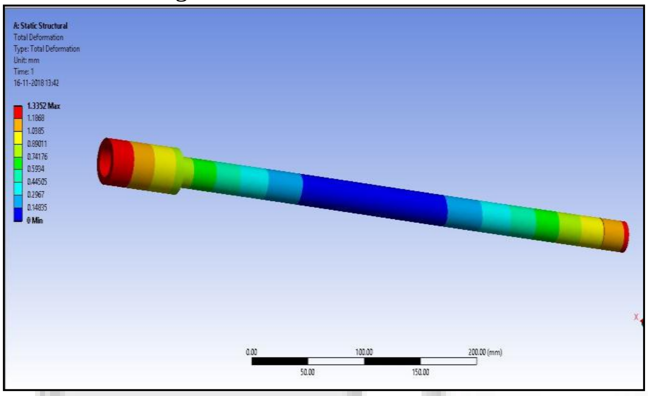

Fig. 6: Column deformation

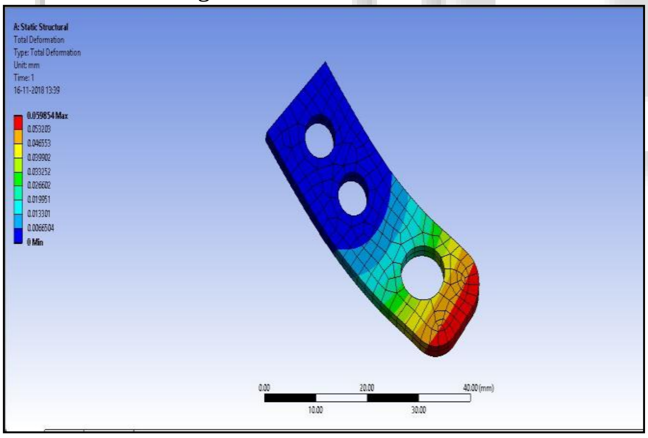

Fig. 7: Steering arm deformation

D. CAE Results:

All The Designs were validated in ANSYS 19.0.

1) Rack Pinion

MAXIMUM DEFORMATION =0.00135 mm
MAXIMUM STRESS = 37.512 Mpa
FOS = 8.1

2) Column

MAXIMUM DEFORMATION = 1.3352mm
MAXIMUM STRESS = 90.985Mpa
FOS = 2.3

3) Steering arm

MAXIMUM DEFORMATION = 0.05985mm
MAXIMUM STRESS = 120.52Mpa
FOS = 1.75

From above Obtained results we can conclude that the safety factor was observed to be within limits. Hence the design can be considered as safe design.

VI. CONCLUSION

An All-Terrain Vehicle needs a light-weight, agile & a durable design for its performance to be best. Various calculations were done for steering angles, Rack Pinion design, turning radius. The values calculated in the paper may differ practically due to steering linkages error or due to improper steering geometry, so these values are useful to understand the interdependency of the quantities on each other and to design an ideal manual rack and pinion system for the vehicle. The theoretical values were validated using lotus shark software. The design of the components was done in CAD software CATIA V5 R21. All the designed components are checked to the real world conditions using Finite Element Analysis. The software used to perform the FEA was ANSYS 19.1. The components are checked for the Von- Mises theory of failure & the safety factor. The type of analysis done is Static Structural. All the analysis results are properly studied. More optimization is carried out on it to make it more light-weight.

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