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Analysis and Improvement of the Steering Characteristics of an ATV.

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ABSTRACT
The main objective is to analyze the steering characteristics of an ATV (All Terrain vehicles) in order to improve the maneuverability of the vehicle. It is found that in our last year's BAJA buggy the vehicle oversteerd more than we expected, when inputs are given in the steering wheel while cornering at sharp turns which tends to move the vehicle out of the track. We focus to design an effective steering system with a reduced steering ratio of 3:1. This helps the driver to maneuver the vehicle with ease.

I. INTRODUCTION

A Steering gear box helps to steer the front wheels of the vehicles which includes commercial vehicles, passenger vehicles, an ATV (All Terrain vehicles), etc. There are various types of steering gear boxes among which rack and pinion gear is considered to be more precise as it consists of fewer parts easier to control, respond to the inputs given and less backlash. The rack is housed in a casing.

The casing is supported on the frame near its ends. The ends of the rack are connected to the track rod with the help of a ball joint or Heim joint. The pinion shaft is carried in the bearings housed in a casing. The pinion meshes with the rack. Its main objective is to convert the rotational motion of the steering wheel to the linear motion of the rack. While the driver turns the steering wheel the column rotates the pinion gear which in turn meshes with the rack, then the linear motion is transferred to the drive wheels via the track rod.

The steering ratio is the main criteria for designing the steering system. It is nothing but the number of degrees of rotation of the steering wheel to the number of degrees of rotation of road wheels. The steering ratio varies based on the applications, which in changes the diameter of the steering wheel accordingly. Generally, the steering ratio for passenger cars is 14:1. In case of heavy vehicles it is 20:1. The number of degrees of rotation of steering wheel increases with increase in steering ratio. Rack and pinion steering is quickly becoming the most common type of steering on cars, small trucks and SUVs.

II. DESIGNED PARTS OF STEERING SYSTEM:

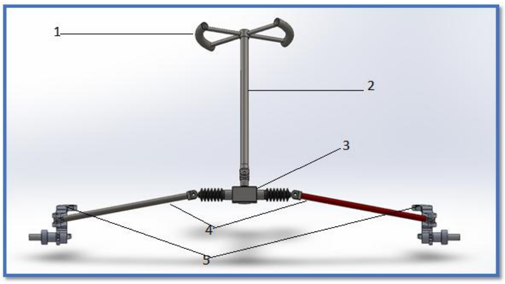

Fig 1: Parts of steering system.

1. Steering wheel 2. Steering column
3. Steering gearbox 4. Tie rod
5. Steering Upright.

Design Methodology for Steering System:

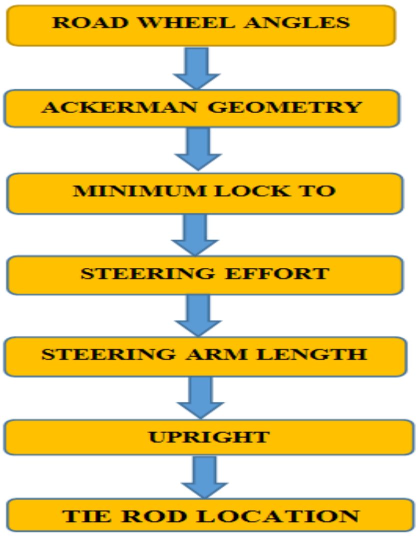

Fig 2: Design methodology flow chart.

The main objective of this project is to design a steering system based on the flow chart shown in fig 1. The design process starts with the road wheel angles, Ackerman geometry, the minimum lock to lock, steering effort, steering arm length, Upright and Tie rod location.

Ackermann steering plays an important in a steering system as it enables the vehicle to steer the vehicle without slippage of the tire. Secondly, the minimum lock to lock of the steering wheel is kept by optimizing the steering ratio of 3:1 in order to have better maneuverability for a BAJA buggy. Then followed by steering effort and wheel alignment parameters are considered. Wheel alignment parameters include camber, caster, steering axis inclination, Toe-in and Toe-out. The angles of all the wheel alignment parameters are specified based on the requirement of BAJA buggy. Hence, on the whole the dynamic behavior and stability of the vehicle is improved by studying the steering characteristics.

Design Process, Analysis And Simulation:

The complete CAD modelling is done in solid works software, including Rack and pinion gear box, steering upright, Tie rod, steering column and steering wheel. It is important to carry out analysis for all the custom made parts. Ansys software is used for analysis to check the component strength, fatigue life which includes components factor of safety, equivalent stress and total deformation to ensure safety design standards.

After carrying out analysis, simulation is carried out with the help of Lotus shark software by generating the coordinates from the CAD assembly. The dynamic behavior of wheel alignment parameters, bump steer and Ackerman percentage is checked with respect to steer travel. The graphs were generated after simulation.

Design Caculations:

1.1.1 Rack and pinion gear Box:

Parameter	Rack	Pinion
No. of Teeth	22	23
Length	356mm	52mm
Diameter	20mm	34mm
Pressure Angle	20º	20º
Travel	4.25”	270 deg

Table 1: Rack and Pinion Specifications

2. Steering Components Cad Modelling:

2.1 Rack:

Cad modelling is done for rack with 4.25” rack travel. It consists of 22 teeth. Heim joints are fixed at both the ends of the rack to connect the tie rod. The total length of rack travel is 14”.

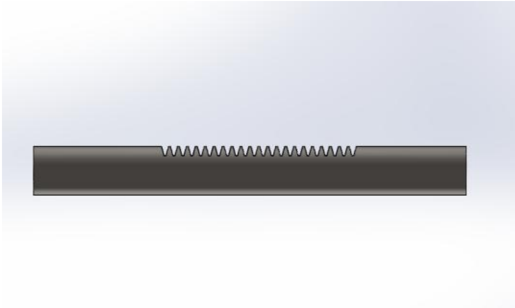

Fig 3: CAD modelling of rack.

2.2 Pinion:

Pinion gear is designed with 23 teeth with a travel of 270 degree from lock to lock.

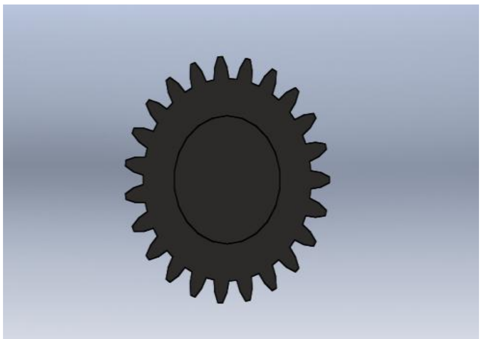

Fig 4: CAD Modelling of pinion.

2.3 Rack and pinion gearbox casing:

Casing is designed to cover the rack and pinion assembly. Nylon bushings are provided at both the ends of casing for rack sliding. Ball Bearing is placed at the center to enhance the rotatory motion of the pinion. Backlash adjustment is provided at the bottom side of the casing to avoid the free play.

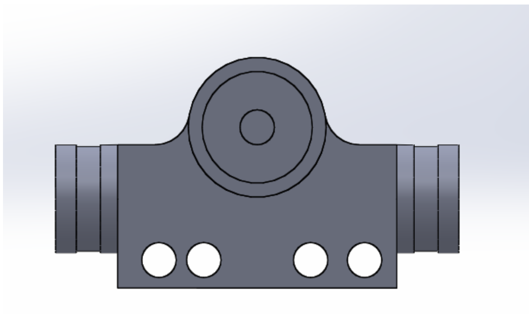

Fig 5: CAD modelling of rack and pinion Gear box casing.

The casing also incorporates the mounting holes of 8mm diameter of 4holes for rigid mounting. The casing is supposed to made-up of the Aluminium materials for weight reduction.

2.4 Rack and pinion assembly:

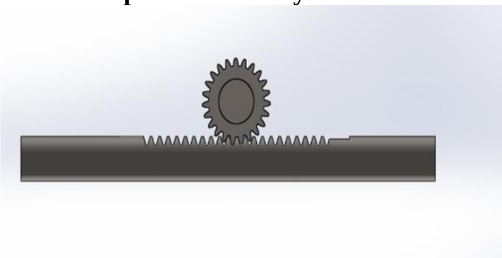

Fig 6: Rack and pinion assembly.

2.5 Final assembly of Steering gear box:

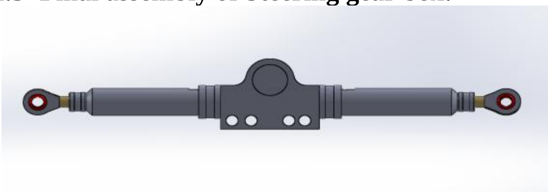

Fig 7: Steering gear box assembly.

III. CALCULATIONS

3.1. 1. Specifications:

Wheel base	1561mm
Front track	1350 mm
Rear track	1244mm
Turning circle radius(g)	1900mm
Lock to lock angle(Steering wheel)	360 degrees
Diameter of the steering wheel	300 mm
Rack travel	107.95mm
Tie rod length	320 mm
Steering effort	65N
Steering ratio	3:1
Caster angle	10 deg
Kingpin inclination	10 deg

Table 2: Specifications

3.1.2. Steering Effort:

VERTICAL MOMENT:
Mv = - (Fzl+Fzr)dsin γsinϬ + (Fzl-Fzr)dsin Ʋcos Ϭ
Dynamic load = FAW+L. T
= (210*0.45*9.81) + (w*a*h/g*L)
= (927)+(283)
= 1410 N
Fzr = 710N , Fzl = 700N

LEFT TURN:
Mv = - Fzl d sinγ sinϬi - Fzr d sinγ sinϬo + Fzl d sinγ cosϬi - Fzr d sinγ cosϬo
= -700*45*sin(10)*sin(45)-710*45*sin(10)*sin(36) +700*45*sin(10)*cos(4)-710*45*sin(10)*cos(36)
= -7749.5Nmm

RIGHT TURN:
Mv = - Fzl d sinγ sinϬo - Fzr d sinγ sinϬi + Fzl d sinγ cosϬo - Fzr d sinγ cosϬi
= - 700*45*sin (10) *sin (36) -710*45*sin (10) *sin (45) +700*45*sin (10) *cos (36) -710*45*sin (10) *cos (45)
= -6646Nmm

LATERL MOMENT:
ML = (Fyl + Fyr)*r*tan Ʋ
Fyl = μFzl = 0.45*700 = 315N
Fyr = μFzr = 0.45*710 = 319N
= (315+319) *285*tan (10)
= 31860Nmm

TRACTIVE MOMENT:
Mt = (Fxl – Fxr) *d
= (710-700)*45 = 450Nmm

EQUIVALENT TORQUE:
Equivalent torque= 28963Nmm

STEERING EFFORT:
Steeringeffort=EquivalentTorque/(steeringwheel radius*S. R)
= 28963/(150*3)
= 65N.

3.1.3. PINION DESIGN AND CALCULATION:

Pinion Torque = Equivalent Torque/Steering ratio
= 28963/3
= 9654Nmm

Ft = Pinion Torque/(PCD/2)
= 9654/23
= 419N.

Bending Stress σb = Ft*q.k*q.e / b*m
= 419*1*2.7/ 3.5*2
= 162 N/mm^2.

Design bending stress (Tension):

$$[\boldsymbol{\sigma}_b] = \frac{k_{bl}}{n.k_{\sigma}}\sigma_{-1}$$

x = (t/m – pi/2)/(2*tanα)
Where, t = thickness of teeth.
m = module.
α = pressure angle.

x = (3.5/2 – 3.14/2)/(2*tan(20))
= 0.25

k_{σ} **= 1.3**

$\boldsymbol{\sigma}_{-1} = 0.35\boldsymbol{\sigma}_u + 1200$
$= 3211.005 + 1200$
$= 4411.005$ kgf/cm^2

$[\sigma_b] = \frac{1*4411.005}{2*1.3} = 1696$ kgf/cm^2
$= 170$ N/mm^2

$\sigma_b < [\sigma_b]$

So, the material is safe.
Pressure angle α= 20 deg
Module m = 2
Number of teeth Z1 = 23

Pitch circle dia(d1) = Z1*m
= 23*2
= 46mm.
Tip circle dia = d1+2m
= 50mm
Base circle dia = d1cosα
= 46*cos(20)
= 43.22mm
Tooth thickness on Pitch circle = ∏m/2
= 3.14mm.

3.1.4. STEER CONDITION:

(Wf/Cxf) -(Wr/Cxr) = K (K=understeer gradient**)**
Wf = 882.9N Wr = 1569. 6N
Cxf = 155. 75N/deg Cxr = 267N/deg =
(882.9/155.75) - (1569.6/267).
= - 0.21 (negative).

Since Wf < Wr (oversteer condition).

In this case, the lateral acceleration at the CG causes the slip angle on the rear wheel to increase than the front wheels. The outward drift at the rear of the vehicle turns the front wheel inward, thus diminishing the radius of turn.

3.1.5. CRITICAL SPEED:

In oversteer case, a critical speed will exist above which the vehicle will be directionally unstable.

$$V=\sqrt[2]{-57.3\,Lg/K}$$

= 65m/s.

Long wheelbase vehicles will have a higher critical velocity than the shorter wheelbase vehicles.

3.1.6. YAW VELOCITY GAIN:

Yaw velocity r, is the rate of rotation in the heading angle

r = 57. 3V/R
Where **V**-forward speed.
R-radius of turn.
r = (57.3*65*5)/(18*1765.3)
= 0.586deg/Sec.

Yaw velocity gain is defined as the ratio of steady state yaw velocity to the steer angle.

3.1.7. CURVATURE RESPONSE GAIN:

CR= (1/L) /(1+K*v2/Lg)
= (1/1651)/(1+0.21*65^2/1651*9.81)
= 0. 0121deg/Sec

IV. UPRIGHT DESIGN AND ANALYSIS:

4.1. Steering Upright:

The Steering Upright is designed in such a way that Provision is given for mounting the control arms, Tie rod, Brake calipers and space for housing the wheel assembly with the help of Stub axle. The important feature in this upright is offsetting of stub axle mounting point from its center. The main objective is to reduce the static inclination in the front side of the vehicle. This reduces the roll center height. The rear side roll center height is higher than that of the front. While cornering the vehicle, due to the elastic and geometric load transfer in the front and rear side of the vehicle tends to increase the slip angle in the rear. Hence over steer is achieved.

4.1.1 Analysis by Ansys:

1. Steering Upright

Step 1: Generating mesh for the upright:

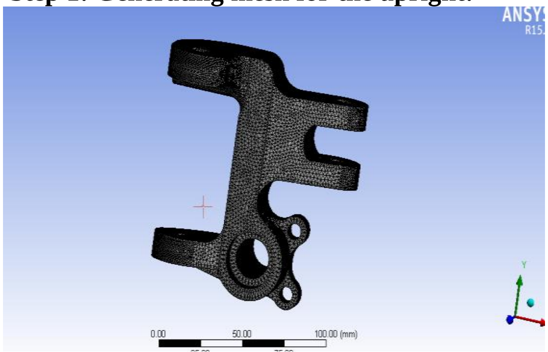

Fig 8: Meshing of upright

Step 2: Fixed support is given at two points:

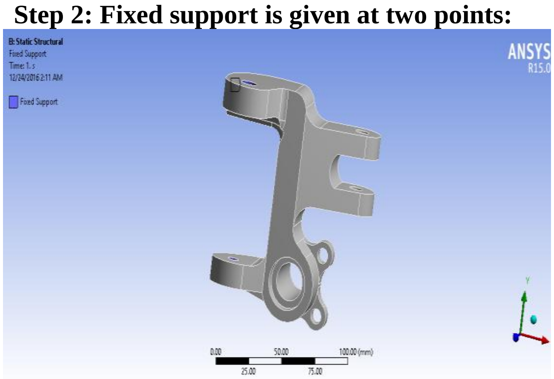

Fig 9: fixed support is given

Step 3: Bearing load of 4500N is applied at the steering arm:

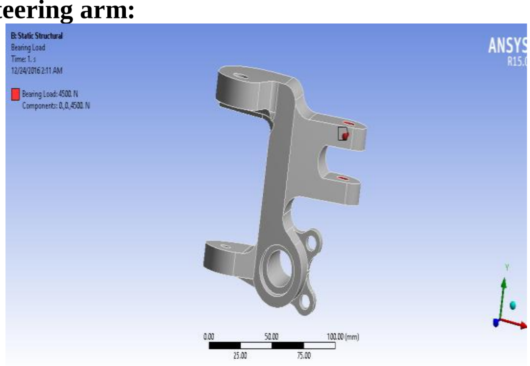

Fig 10: bearing load is applied at steering arm

Step 4: Bearing load of 18000N is applied to the spindle mount point considering one wheel landing:

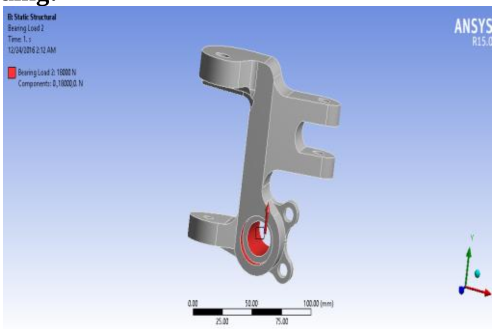

Fig 11: bearing load applied at spindle mounting point.

Step 5: A moment of 140 Nm is applied at the caliper mounting point:

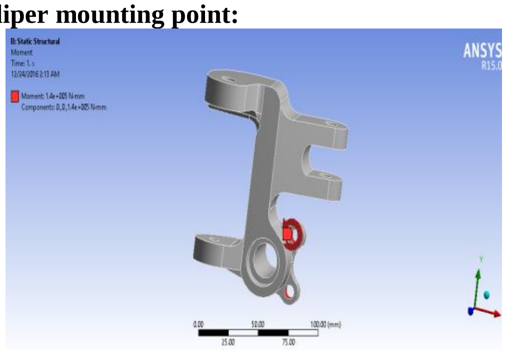

Fig 12: load applied at caliper mounting point.

Step 6: The total deformation is found out by solving the upright:

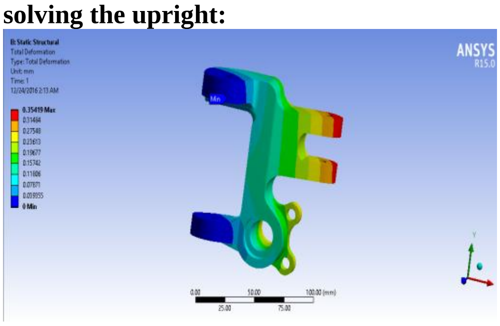

Fig 13: total deformation of knuckle

Step 7: Equivalent stress is found out:

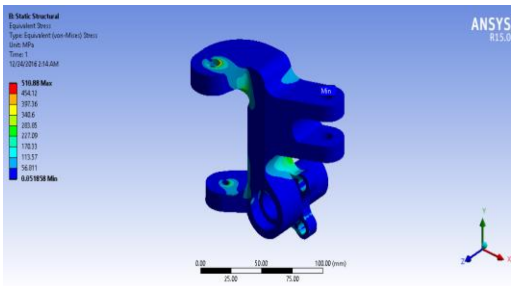

Fig 14: Equivalent stress

Step 8: Finally safety factor is checked:

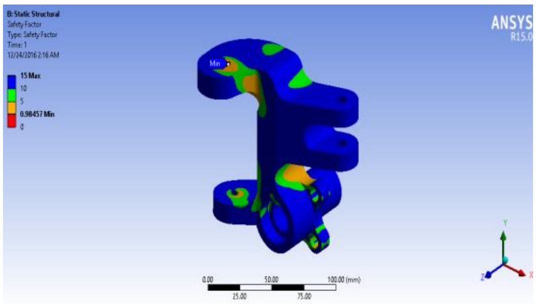

Fig 15: safety factor

2. Analysis for rack and pinion

Step 1: Remote point is applied at the surface of the pinion

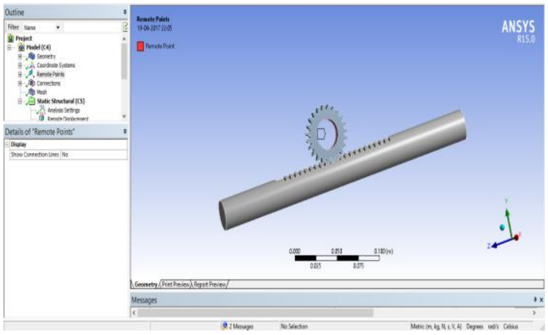

Fig16: Applying remote point at the rack and pinion

Step 2: Meshing of Rack and pinion:

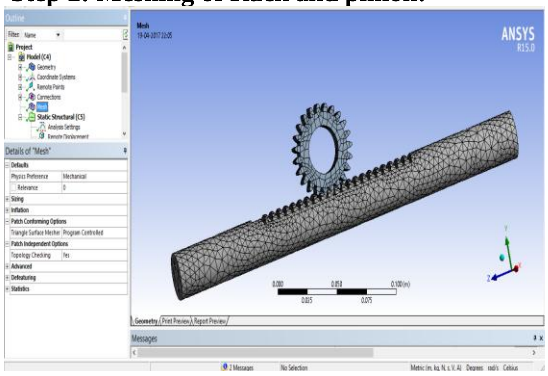

Fig17: meshing of rack and pinion

Step 3: Remote displacement is applied:

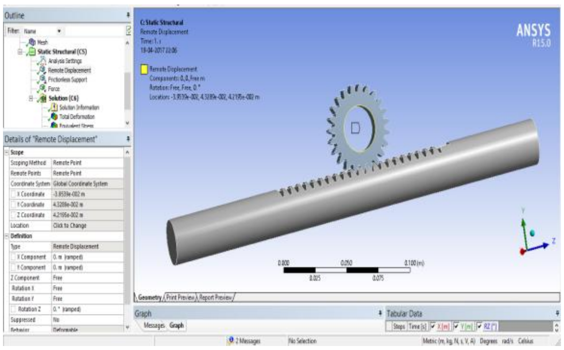

Fig18: Remote displacement is applied at the pinion surface

Step 4: No separation is applied along the surface of the rack

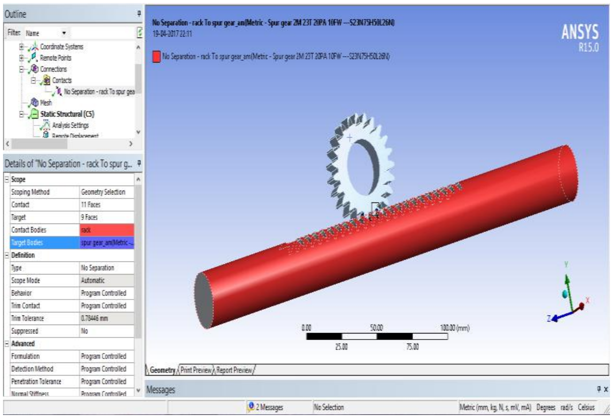

Fig19: No separation is applied along the rack surface

Step 5: Frictionless support is applied at the rack surface:

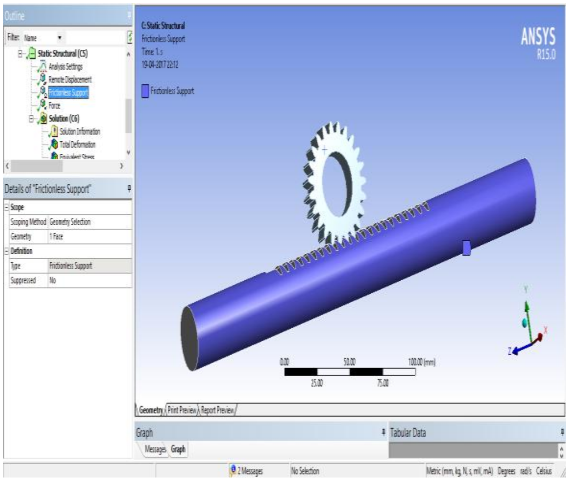

Fig20: Frictionless support is given at the rack surface

Step 6: Force is applied along the z-axis to the rack:

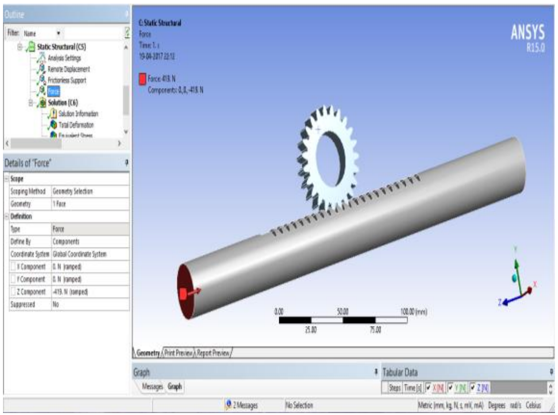

Fig21: Force applied along the z-axis to the rack

Step 7: Equivalent stress

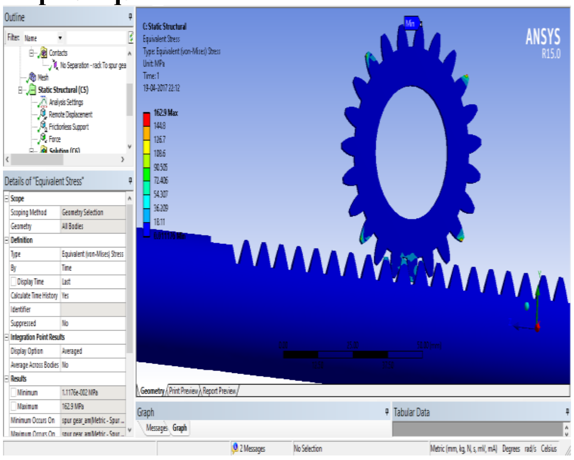

Fig22: Equivalent stress of rack and pinion

Step 8: Factor of safety

Fig23: Factor of safety

V. LOTUS SHARK ANALYSIS:

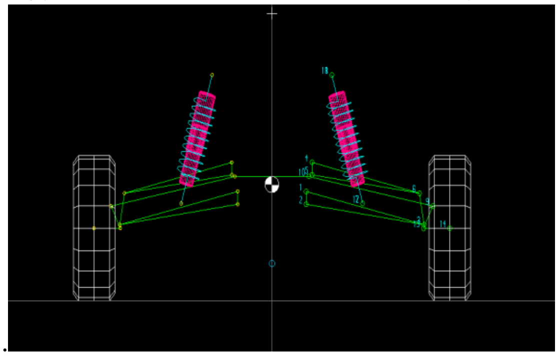

Fig 16: Steering geometry

Fig 7 shows the geometry of steering with double wishbone with lower damper and simulated for 300mm arms travel for better camber, toe and caster angle variations

5.1.1 Lotus Shark Points-Front:
Double wishbone Damper to Lower Wishbone - Static values.

Points.	**X(mm)**	**Y(mm)**	**Z(mm)**
Lower wishbone front pivot	-1003.4	-124.4	-549.4
Lower wishbone rear pivot	-759.6	-124.8	-588.9
Lower wishbone outerball joint	-844.2	-552.8	-651.7
Upper wishbone front pivot	-985.9	-145.8	-459.7
Upper wishbone rear pivot	-748	-145.85	-498.2
Upperwishbone outerball joint	-828.5	-533.8	-554.8
Damper wishbone end	-837.8	-329.7	-586.5
Damper body end	-916.7	-584.5	-594.5
Outer track rod ball joint	-954.3	-134.4	-503.2
Inner track rod ball joint	-773.8	-217.8	-190.1
Upper spring pivot point	-837.9	-329.7	-586.5
Lower spring pivot point	-889.5	-551.1	-662.9
Wheel spindle point	-889.5	-646.5	-662.9

Table 3: Lotus shark point.

3.1.8. Graphs after simulation:

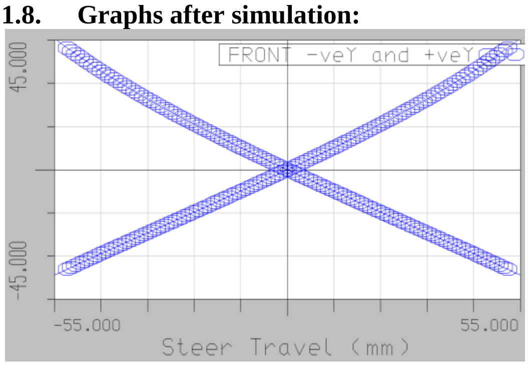

Fig 17: Steer travel Vs Toe angle

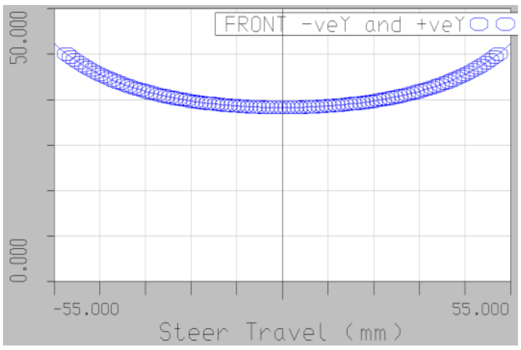

Fig 18: Steertravel vs Ackerman%

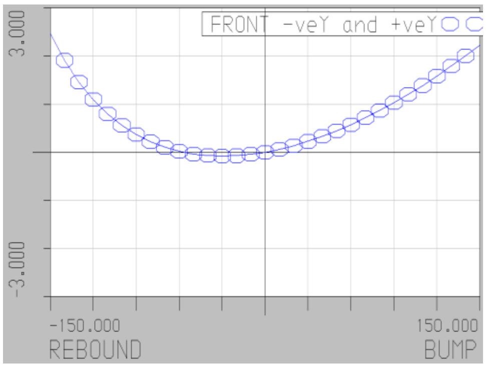

Fig 19: Bump steer vs toe angle

1. The first graph represents the steer travel vs toe angle. The inner wheel turns at an angle of 45deg and outer wheel turns at angle of 35deg during full-lock.The high steering angle of 45deg is used to aid in the sharp turns.It is also made sure that the tie rod does not hit the upright,suspension arms and the wheel rim during these manuers.It is also ensured that brake hose does not bend beyond certain limit thus brake hose having only very minimal damage.
2. 45% Ackermann is obtained during full lock of steering wheel.A configuration in between parallel steering configuration and pure Ackermann configuration is used as there is considerable difeerence in the slip angle of the inner and the outer wheels. The percentage of Ackermann and the amount of bump steer were also determined for different rack travels.

Bump steer will be low when the suspesion arms and the tie rod are parallel.In our case tie is not exactly parallel to the suspension arms.The outer tie rod end is made to move in arc with inner tie rod end as its Locus.Analysis were made using Lotus software for minimal change in toe when the vehicle hits the bump.The graph shows that the angle is 3deg for 150mm of bump travel.

VI. CONCLUSION

The main motivation behind this project is to rectify the faults experienced and to enhance the steering characteristics of a BAJA buggy used for engineering competitions.

In order to enhance the steering characteristics, a Steering gear box and steering upright is designed. The designed parts are analysed and finally simulation is carried out.
The values obtained after the design process i.e, such as analysis and simulation is desierd to run the BAJA buggy efficiently.

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