Steering System Design for an All Terrain Vehicle

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Abstract

Steering system for an all terrain vehicle needs to be very robust due to the ups and downs of a rough terrain. Rack and pinion geometry was set in LOTUS software by taking into consideration the suspension chosen for the vehicle. The Design torque was calculated for rack and pinion design with the help of mechanics and vehicle characteristics. All the components for the steering system were recognized and then designed and modelled in Autodesk Inventor. All the components were then analysed for failure in Autodesk Nastran software.

Keywords—Rack, Steering System, Pinion, LOTUS software, Autodesk Nastran, Autodesk Inventor.

I. Introduction

This Rack pinion is a basic mechanism used as a steering system. Many other mechanisms like hydraulics, pneumatics have sprung up in recent times which are more user friendly and accurate but they require a power source and are complex mechanisms. Rack and pinion is most basic arrangement but yet effective.

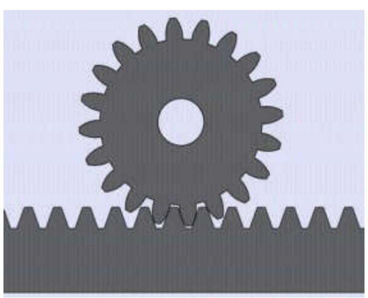


fig1. Rack and Pinion

It is a gear arrangement in which rotary motion is converted into translatory motion. It is not a torque converter but a motion transmission mechanism. It consists of a rotary gear which may be spur or helical and linear gear bar called rack.

Steering system consist of rack and pinion arrangement, casing and tie rods with ball joints at both, rod end and knuckle end. But its design is a little different for an All Terrain Vehicle than for a passenger vehicle. It is made more robust and long lasting for an A.T.V. An A.T.V or an all terrain vehicle is built to be run in rough environments like dessert or mountainous terrain. This puts a lot of stress on the system and hence vehicle dynamics are an important consideration while designing a steering system for an A.T.V.

In this papert, the steering system designed is for an A.T.V for Baja competition. Hence, all suspension geometries and roll cage design specifications were taken into consideration while designing the system.

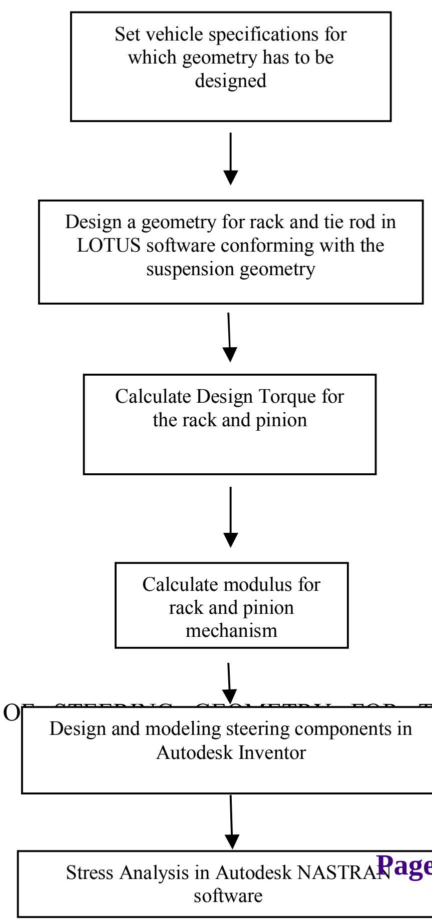
VEHICLE SPECIFICATIONS:

The steering system is designed for the vehicle of weight 270kg inclusive of the weight of the driver. The track width of the vehicle is 44 inch and wheel base is 56 inch.

The suspension choosen for the vehicle is Wishbone suspension.

METHODOLOGY

A procedure is followed while designing the steering system which is represented by following flow chart:-



DESIGN OF STEERING GEOMETRY FOR THE VEHICLE:-

Vehicle Dynamics:

Vehicle Dynamics is the study of the response of the vehicle to driver inputs. It is done to design geometries of systems like steering and suspension. These geometries must ultimately collaborate to keep the vehicle stable during operating conditions for which it is being designed. Following is steering geometry which was designed for the project.

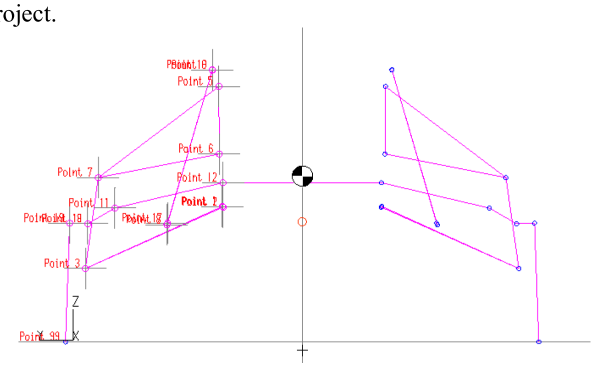


fig2. Steering geometry

In figure 2, points are listed for only right side of the vehicle and in the centre crossed black and white circle denotes origin. Line joining point 12 on both the side of origin denotes rack length which equals to 381 mm and line joining point 12 and point 11 denotes tie rod which equals to 390 mm.

There are various factors with which a steering systems are judged upon which were studied and designed in LOTUS SHARK software. These factors are listed below and represented in form of graphical variation with respect to rack travel.

1) Ackerman percentage:-

It indicates how much steering geometries conform to Ackerman steering mechanism for perfect steering condition. If it is more than 100% (oversteer), this means tire turns more than it is supposed to and if less (understeer) then it turns less than it is supposed to, for perfect steering of the vehicle. In both cases, rubbing of tire takes place but for better gripping at cornering a bit of oversteer is preferred than understeer. Following indicates graphical representation of variation of Ackerman percentage against rack travel:-

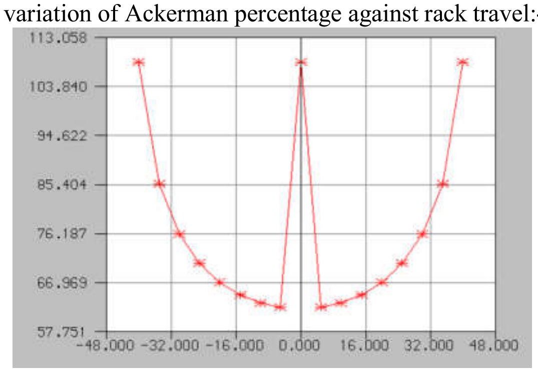


fig3. Ackerman percentage vs Rack travel

Ackerman percentage at maximum turning angle is 106% which results in minimum stresses on tie rod.

2) Toe angle

It is the angle tire makes with vertical in top view of the vehicle. It basically denotes angle turned by the tire. Outer Toe angles should be between 40° to 50° and inner toe angles should be between 25° to 35°. The curve also must be smooth with no sudden leaps or dives. Following is the graph of toe angle variation against rack travel for the steering geometry.

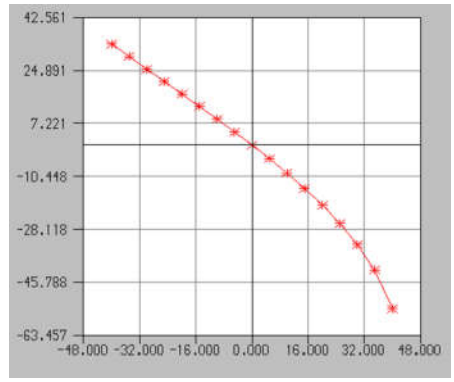


fig4. Toe angle vs rack travel

Maximum Inner turning angle was 46° and maximum outer turning angle was 35°. This resulted in minimum turning radius of 2.3 metres.

Conforming these geometry to suspension geometry and characteristics result in following vehicle specification:-

1) Rack length: 381 mm
2) Rack travel (lock to lock): 80 mm
3) Centre of gravity height: 406.4 mm
4) Static castor angle: 5°
5) Static Steering axis inclination: 8°

Torque Calculation:

Depending upon vehicle geometries, maximum alignment torque experienced by wheel was calculated at maximum turn and it is equal to 66.44 Nm.

Power required at wheel to keep wheel turning against alignment torque=P_{wh}=T_{wh}(Torque) X $d\theta_{wh}$ (angle turned)

Power required at pinion to supply torque at wheel=P_p=T_p X $d\theta_p$

But,

Power supplied = Power required

$$P_p = P_{wh}$$

$$T_p \text{ X } d\theta_p = T_{wh} \text{ X } d\theta_{wh}$$

$$T_p = T_{wh} \text{ X } \frac{d\theta wh}{d\theta p}$$

But $\frac{d\theta wh}{dSr}$ is available from graph in figure 4 where S_r is rack travel

Therefore relation between pinion turn and rack travel was found out by reiterative design process which is as follows-

$$d\theta_p = \frac{20}{3\pi} dS_r$$

Hence Torque equation becomes,

$$T_p = T_{wh} \text{ X } \frac{3\pi}{20} \text{ X } \frac{d\theta wh}{dSr}$$

But $\frac{d\theta wh}{dSr}$ value from graph of toe angle vs rack travel (fig4) at maximum outer angle is 0.86 and T_{wh} at maximum turn is 66.44 Nm.

Substituting these values,

$\mathbf{T_p=27\ Nm}$

Therefore torque at pinion is 27 Nm.

II. Component Design and Modeling

Problem Definition:

To Design Mechanical components of steering system subjected to 27 Nm torque at pinion. Mechanical components must include rack and pinion arrangement, tie rod with ball joint at both, rack and knuckle ends. Rack length is 381 mm and tie rod length is 390mm.

Design:-

1. Design torque = M_t = 27Nm
2. Assumptions:-
   1. Profile is involute and pressure angle(α) is 20°
   2. It is precision cut
   3. Rack is a rotary gear with gear ratio equal to 1
3. Minimum n.o of teeth-

   $Z_1= \frac{2\frac{a}{m}}{sin^2\alpha}$ ....$\frac{a}{m} = 1$ and α = 20°

   =18

   $Z_2=Z_1=18$ since i=1
4. Lewis Form Factor:-

   for pinion, $y_1=0.154 - \frac{0.912}{Z1} = 0.103$

   for rack, $y_2=0.154 - \frac{0.912}{Z2} = 0.103$
5. Material Selected:-

Component	Material	Density (kg/m^3)	Yield strength (N/mm^2)	tensile strength (N/mm^2)	Brinell Hardness N.o.	Poisson's Ratio
Pinion	En9(C55)	8.8 X 10^3	310	600	201	0.3
Rack	20MnCr5	7.7X10^3	1034	1158	335	0.27

6. Checking Weaker Element:-

   $fs_1 = y_1 X \sigma_{y1} = 106.502$

   $fs_2 = y_2 X \sigma_{y2} = 31.93$

Hence, Pinion is the weaker element and it will be designed.

7. Design Criteria

Pinion will be designed on the basis of static strength and checked for wear. Pinion and rack do not undergo dynamic loads and hence will not be checked under the same.

8. module calculation

   On the basis of static strength,

   $$m \geq 1.26\sqrt[3]{\frac{Mt}{\sigma b\ Y\ \psi\ z}}$$

   1. M_t=27Nm=27 X 10^3 Nmm
   2. $\sigma_b=\frac{1.4Kbl}{nK}\sigma_{-1}$

      where K_{bl}=1

      K_σ = 1.4

      n=2.5

      $\sigma_{-1}=0.35\ \sigma_u+1200$

      =0.35 X 6000+1200

      =3300 kgf/cm^2

      σ_b=1320 kgf/cm^2=132 N/mm^2
   3. Z=18
   4. Y=πy=0.323
   5. ψ=10.....assumed

   Substituting above obtained values in module equation,

   m=1.912≈2

   Checking for Wear:-

   F_w = d x Q x K x b

   d=mz

   $Q=\frac{2i}{i+1} = 1$

   $K=(\sigma_c)^2 \frac{\sin\alpha}{1.4}(\frac{1}{E1}+\frac{1}{E2})$

   σ_c = 2.8BHN-70

   = 2.8 x 201-70

   =492.8 N/mm^2

   α=20

   $E_1=E_2=2.1x10^5$N/mm^2

   K=0.565

   b=ψxm

   $M_t = F_w \times \frac{d}{2}$

   Simplifying the above equation we get,

   $M_t=\frac{m^3 z^2\psi K}{2}$

   Substituting value of m=2, we get

   $M_t=7.3224x10^3$Nmm < $27x10^3$Nmm

   Hence this module fails under wear load, therefore new module for M_t=27Nm is:

   $m=\sqrt[3]{\frac{2xMt}{z^2\psi K}}$

   Substituting the torque with other values, we get

   m=3

9. Gear Proportions

	Pinion	Rack
Pitch Circle Diameter	54mm	-
No. of teeths	18	10
pitch	9.428mm	9.428mm

10. Shaft design

    Rack Rod:-

Shaft diameter=30mm

Force acting on the centre of the shaft= weight of the torso of the driver + shear thrust applied pinion on rack

$=30kgf + \frac{2Mt}{d}tan\alpha$

= 300 +364

= 664N

Distance of pinion from pivot on rack = 114mm= 0.114m

Bending Moment at the centre = $\frac{664\ x0.114}{2}$ = 37.848Nm

Applying bending moment equation:-

$$\frac{Mt}{I} = \frac{\sigma b}{y}$$

we get, $\sigma b = M_t \times \frac{y}{I} = 37.848\text{x}10^3\text{x} \frac{32}{\pi d^3}$ $= 37.848\text{x}10^3\text{x}\frac{32}{\pi x 30^3} = 14.27\text{N/mm}^2$

but material $\sigma_y = 1034$ N/mm^2 > σ_b=14.27N/mm^2

Hence Shaft is safe under bending load.

Tie rod:-

Shaft diameter=12mm

Compressive force acting along the rod=1000N

Compressive stress = $\frac{1000}{\frac{\pi}{4}d^2}$ = 8.84194N/mm^2<<$\sigma_{u=}$1158N/mm^2

Hence shaft design is safe.

11.

Ball joint Design:-

Resisting area for a ball to compressive load is circle with maximum diameter.

Therefore, checking for compressive failure at ball joint at rack end:-

ball diameter = 18mm

Compressive strength = $\frac{1000}{\frac{\pi}{4}d^2}$ =3.92N/mm^2<< $\sigma_{u=}$1158N/mm^2

Therefore, checking for compressive failure at ball joint at knuckle end:-

ball diameter = 12mm

Compressive strength = $\frac{1000}{\frac{\pi}{4}d^2}$ =8.84N/mm^2<< $\sigma_{u=}$1158N/mm^2

MODELLING:-

Modelling was done in Autodesk Inventor 2017. Spur gear was modelled using design generator present in the software. The involute profile was later exported from the spur gear model and was used to model rack rod. Other components were modelled using feature based command present in the software.

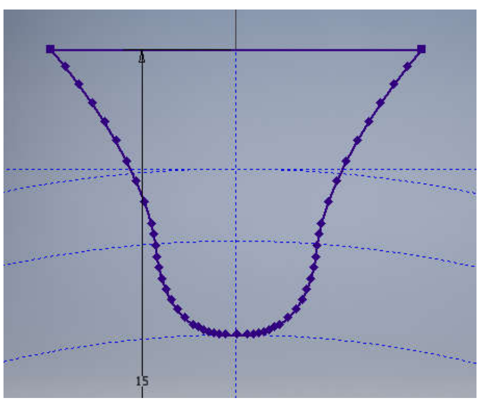


Fig5. Involute Profile

Figure 5 shows involute profile exported from spur gear and used to model rack teethes.

Models of mechanical components are presented below:-

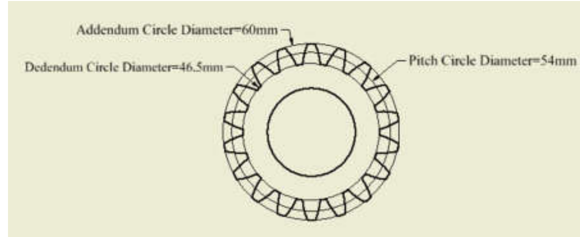


Fig6. Spur Gear Front View

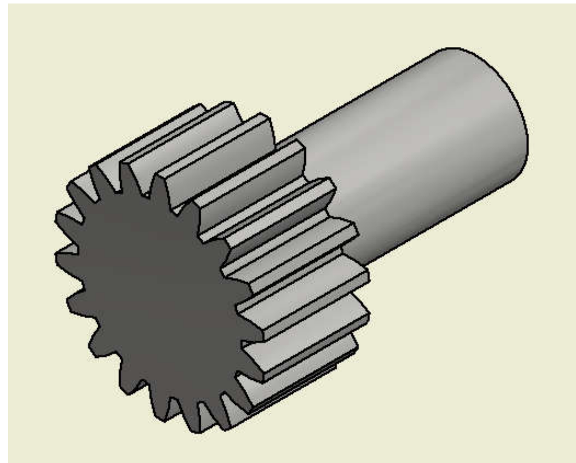


Fig7.Spur Gear Isometric view

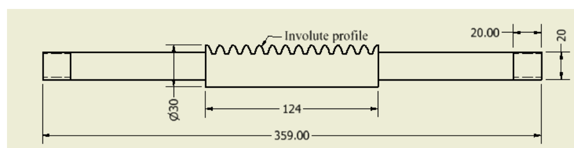


Fig8.Rack Front View

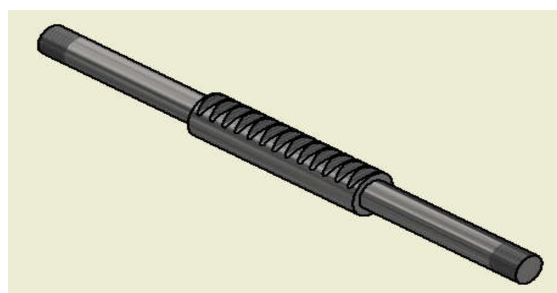


Fig9.Rack Isometric View

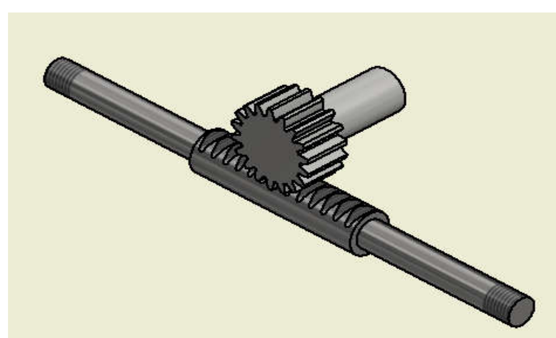


Fig10.Rack and Pinion arrangement

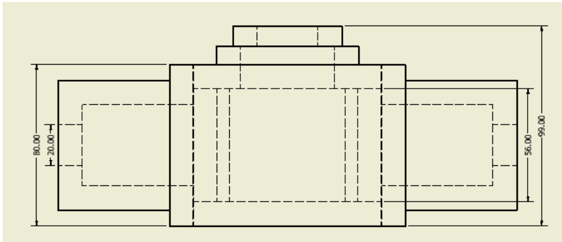


Fig11Casing Top View

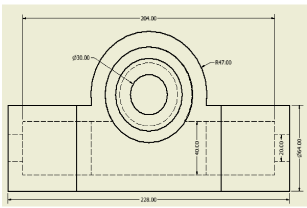


Fig12.Casing front view

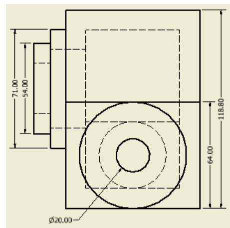


Fig13 Casing Side View

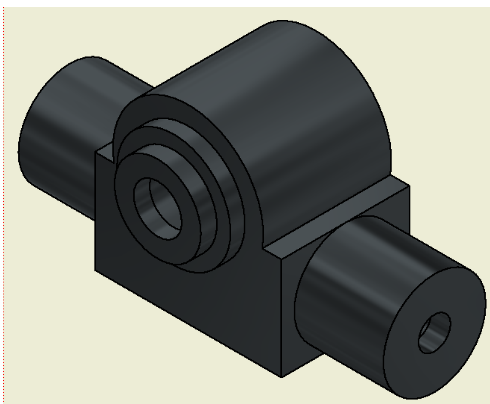


Fig14. Casing Isometric View

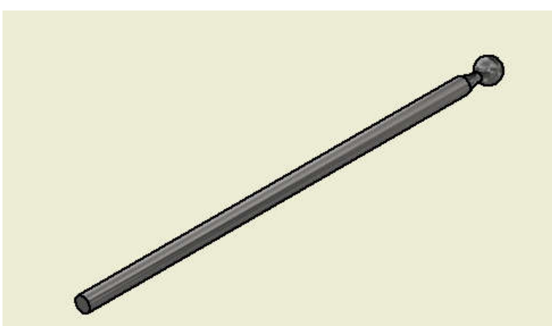


Fig15.Tie Rod Isometric View

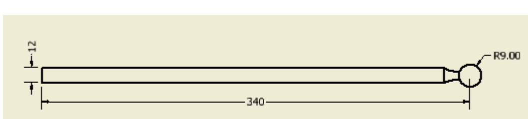


Fig16.Tie Rod Front View

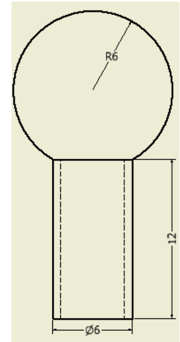


Fig.17 Isometric View of Ball Screw
Fig18. Front View of Ball Screw

Other connecter components for rack and pinion assembly

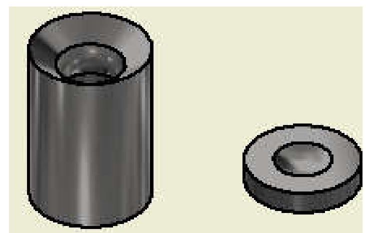


Fig19 Ball Socket for tie rod and rack rod

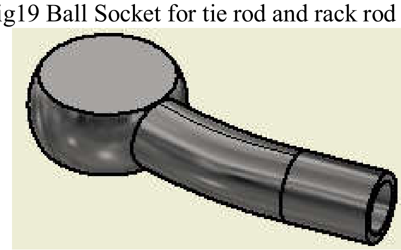


Fig20 Ball Socket connector for Ball Screw and Tie Rod

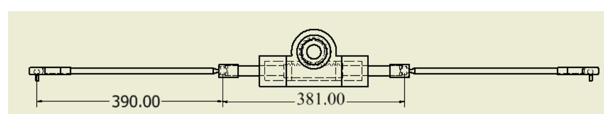


Fig21 Front View of Total Assembly of all the components in complete stretched position

Hence, it can be seen that modelled assembly conforms with the steering geometry of rack length of 381mm and tie rod length of 390mm.

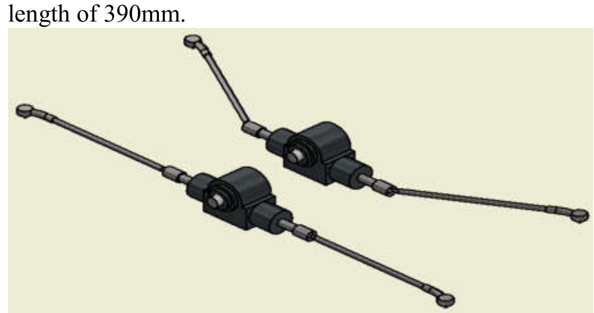


Fig22. Isometric View of Assembly of Steering System(Stretched And Unstretched)

Analysis

Rack and pinion form the most critical component of the system and hence must be analyzed for failure. A structural stress analysis is carried out on rack and pinion in Autodesk Nastran In Cad software.

Analysis Procedure is same for both the components except the loads are applied on their surfaces in opposite directions which will illustrated as follows:-

1) Material Selection

Since material properties of rack and pinion are known from the design procedure, they are fed into the software. Data which was required was for density, young's modulus, ultimate tensile strength, yield strength and poison's ratio.

2) Element Specification

Element type selected is solid type. Tetrahedron Solid 45 is the only type of Solid Type element available in Nastran.

3) Meshing Specification

Meshing element size is selected as 7 mm for rack and 4 mm for pinion. Order of the element selected is parabolic. First, confirm that you have the correct template for your paper size. This template has been tailored for output on the A4 paper size. If you are using US letter-sized paper, please close this file and download the Microsoft Word, Letter file.

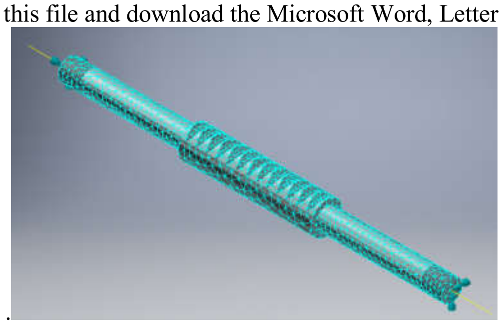


fig 23 Meshed Model Of Rack

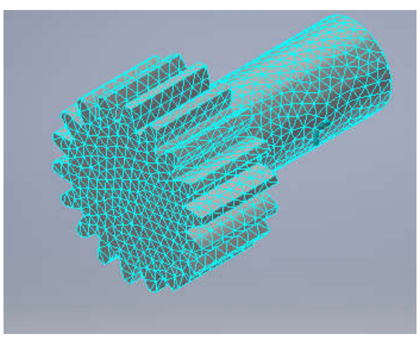


fig 24 Meshed model of pinion

No. of nodes are 12990 for rack and 34764 for pinion. No of elements are 7659 for rack and 23863 for pinion.

4) Constraints:-

For pinion, the shaft connected to the pinion gear was constrained in all the directions as shown below:-

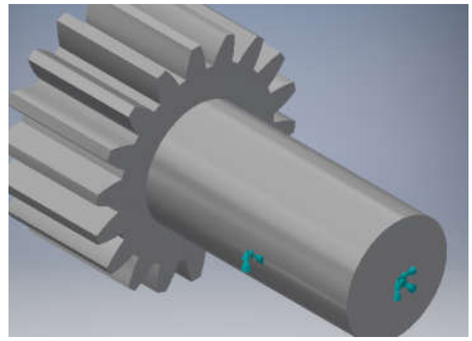


fig 25constraints on pinion

For Rack, both the rack ends of rack rod were constrained in all directions as shown in the following figure:-

fig 26 constraints on rack

5) Loads

For both the components, load value is same but the directions are opposite. The loads are applied on the surface of the teeth such that load passes through pitch plane for rack and pitch circle for pinion.

For pinion, load applied:- Fx = -1000N and Fy = 364N

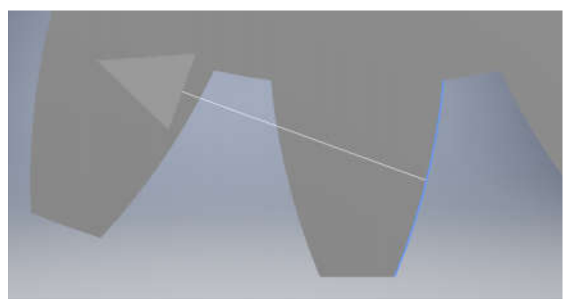


fig 27 load application

For rack, load applied:- Fx = 1000N and Fy = -364N

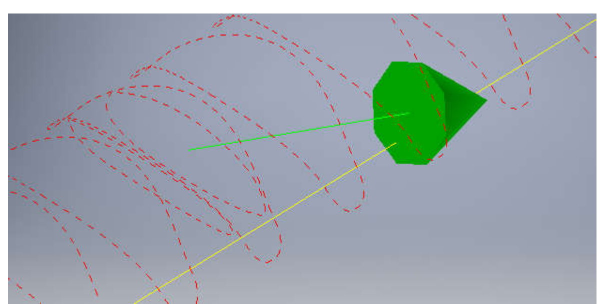
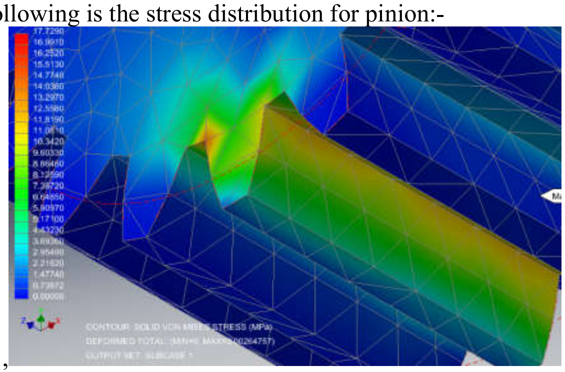


fig 28 load application on rack

Solution:-
Command is given to solve the analysis in Nastran code.
Following is the stress distribution for pinion:-



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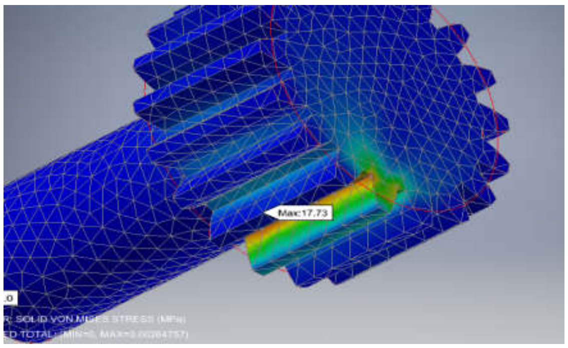
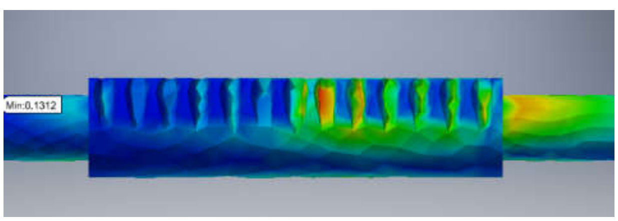


fig 29 stress distribution for pinion

Maximum stress =17.73 Mpa which is much below yield stress for the material. Hence it is safe for operation

Following is stress distribution for Rack:-



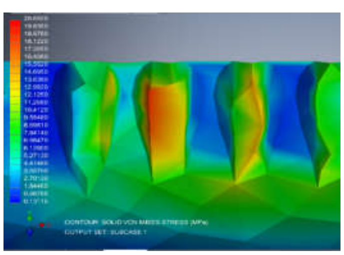


fig 30 stress distribution for rack

Maximum stress = 20.69 Mpa which is much below yield stress for the material. Hence it is safe for operation.

Result and Conclusion

Steering system was successfully designed for the All Terrain Vehicle. The components which go through maximum load were analyzed for failure and proved safe. Rack and pinion proves to be simple in design and setup but hydraulic system though complex and expensive provide better comfort in operation and response. Hence in future, design of hydraulic system can be considered for the steering mechanism. In Baja completion, weight reduction is an important characteristic for the design. Hence Clevis joint at rack and tie rod end can be put which not only is less in weight but also better in maneuverability and resistant to dust clogging.

Particulars	RESULTS
Turning radius	2300 mm
rack travel	80 mm
Inside turning angle	46°
Outside turning angle	35°
wheel base	58.5"
track width	44"
Module for Rack and Pinion	3

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