Design and Analysis of Steering Gear and Intermediate Shaft for Manual Rack and Pinion Steering System

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Abstract- Manual rack and pinion steering systems are commonly used due to their simplicity in construction and compactness. The main purpose of this paper is to design and analyze the rack and pinion steering system. In this paper analyzed the two components of the steering system. Firstly, this paper investigates the characteristics of a rack and pinion gear system mainly focused on bending and contact stresses of the pinion gear and rack bending stress using analytical and finite element analysis. To estimate the contact stress, the-dimensional solid models for different materials are generated by SolidWorks software and the numerical solution is done by ANSYS, which is a finite element analysis package. The analytical investigation is based on Lewis stress formula. This paper also considers the study of contact stresses induced between two gears. Present method of calculating gear contact stress uses AGMA equation. To determine the contact stresses between two mating gears the analysis is carried out on the equivalent contacting cylinders. The results obtained from ANSYS are presented and compared with theoretical values. This paper also deals with the stress analysis of the rack. By using FEM a stress analysis has been carry out. Steering rack deflection and bending stresses are found. This stresses are compared with analytical result. Secondly, Fatigue analysis of intermediate steering shaft is done to find the life of the intermediate steering shaft in cycles and determined the factor of safety of the shaft. The Software results, mathematical and logical calculation implementation in a research will increase the performance and efficiency of a design.

Index Terms- Rack and Pinion Steering Gear, Contact Stress, Rack Bending Stress, Steering Intermediate Shaft, Life Cycle, Safety Factor, ANSYS Software.

I. Introduction

Two main types of steering systems are used on modern cars and light trucks: the rack-and-pinion system and the conventional, or parallelogram linkage, steering system. On automobiles, the conventional system was the only type used until the 1970s. It has been almost completely replaced by rack-and-pinion steering.

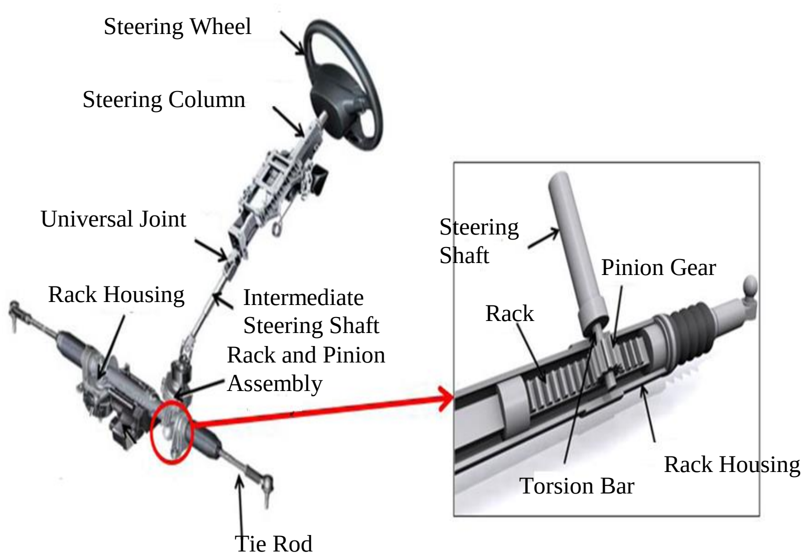

Figure 1. A Simplified Rack-and-pinion Steering System

Rack-and-pinion steering is a simple system that directly converts the rotation of the steering wheel to straight line movement at the wheels. The steering gear consists of the rack, pinion, and related housings and support bearings. Turning the steering wheel causes the pinion to rotate. Since the pinion teeth are in mesh with the rack teeth, turning the pinion causes the rack to move to one side. The rack is attached to the steering knuckles through linkage, so moving the rack causes the wheels to turns. All steering systems contain several common parts. Every steering system, no matter what type, will have a steering wheel, a steering shaft and column, universal joints, steering tie rod, and steering arm.

II. Theoretical Calculation of Steering System Components

A. Theoretical calculation of Steering Gear Design

Table I shows the parameters considered for design a rack and pinion gear.

TABLE I

PARAMETERS CONSIDERED FOR DESIGN A HELICAL PINION GEAR

Design Parameter	Specification
Applied Force	2610N
Pinion Speed (N_p)	15~8 rpm
Helix angle (ψ)	23°
Pressure angle (ϕ)	20°
Modulus of Elasticity (E)	207 GPa
Ultimate Strength (S_u)	1950 MPa
Yield Strength (S_y)	1570 MPa
Brinell Hardness (BHN)	555
Number of teeth of pinion (n_p)	6
Number of teeth of rack (n_R)	28

Pinion and rack are same material and so pinion is weaker. So based design on pinion.
Unknown diameter case
1. Calculation of Diameter of teeth,
Module selected from the standard module series.

$$D_p = n_p \times m_n \tag{1}$$

2. Calculation of Torque (M_t)

$$M_t = F_t \times D_p \tag{2}$$

3. Calculation of pitch line velocity (V)
The pitch line velocity can be calculated by

$$V = \frac{\pi \times D_p \times N_p}{60} \tag{3}$$

$$n_f = \frac{n_p}{\cos^3 \psi}$$

$y_p = 0.175\text{-}0.841/n_f$

4. Calculation of allowable stress, S_{all}
Allowable stress can be calculated by

$$S_{all} = S_0 \times \left[\frac{3}{3+V}\right], \text{ for } V < 10\text{m/sec} \tag{4}$$

5. Calculation of endurance stress, S_o

$$S_0 = \frac{S_u}{3} \tag{5}$$

6. The actual induced stress can be calculated by using Lewis equation.

$$S_{ind} = \frac{2M_t}{m^3 k\pi^2 y_p n_p \cos\psi} \tag{6}$$

7. Strength Check,
Compare S_{all} and S_{ind} (7)

If $S_{all} > S_{ind}$, Design is satisfied.

If not so, keeping on calculating by increasing the module until it is satisfied need to be done.
8. Calculation of the face width of helical gear, b
The face width of helical gear can be calculated as

$$b_{min} = k_{red} \times \pi \times m_n \tag{8}$$

$$b_{max} = k \times \pi \times m_n \tag{9}$$

$$k_{red} = k_{max} \times \frac{S_{ind}}{S_{all}} \tag{10}$$

After determining the design from strength point of view, it is necessary to check the dynamic effect.
9. Dynamic Check,
The transmitted load in (N) can be calculated as

$$F_t = \frac{2M_t}{D_p} \tag{11}$$

10. Calculation of dynamic load, F_d

$$F_d = F_t + \frac{21V(bC\cos^2\psi + F_t)\cos\psi}{21V + \sqrt{(bC\cos^2\psi + F_t)}} \tag{12}$$

11. Calculation of limiting endurance load, F_0

$$F_0 = S_0 b y_p \pi m \cos(\psi) \tag{13}$$

12. Calculation of limiting wear load, F_w

$$F_w = \frac{D_p \times b \times K \times Q}{\cos^2\psi} \tag{14}$$

$$K = \frac{S_{es}^2 \times \sin\varphi_n}{1.4} \times \left[\frac{2}{E}\right]$$

The required condition to satisfy the dynamic check is F_0 ,F_w>F_d.
If not so, keeping on calculating by increasing the module until it is satisfied need to be done.

TABLE II
DESIGN RESULTS FOR PINION GEAR

	Symbol	Value	Unit
No. of teeth	n_p	6	-
module	m_n	2.5	mm
Pitch circle diameter	D_p	15	mm
Face width	b	47	mm
Applied Force	F_t	2610	N
Torsional Moment	M_t (T)	19.575	Nm

The design results for pinion gear are shown in Table II.

TABLE III
RESULT DATA OF TOOTH DIMENSION FOR RACK AND PINION

No	Item	Symbol	Formula	Result	
				Pinion	Rack
1.	Module	m_n		2.5	
2.	Pressure angle	α		20°	
3.	Number of teeth	n		6	28
4.	Height of Pitch Line	H		-	23
5.	Addendum	h_a	$h_a = 0.8m_n$	2	
6.	Pitch Diameter	D_p	$D_p = n_p \times m_n$	15	-
7.	Diametral Pitch	P_d	$P_d = n_p/D_p$	0.4	-
8.	Tooth Thickness	t	$t = 1.5708/P_d$	0.628	
9.	whole Depth	h_t	$h_t = 1.8m_n$	4.5	
10.	Clearance	C	$C = 0.2m$	0.5	
11.	Outside Diameter	D_0	$D_0 = D_p + 2h_a$	19	-
12.	Dedendum	h_d	$h_d = 1m_n$	2.5	
13.	Root Diameter	D_R	$D_R = D_p - 2h_d$	10	-
14.	Center Distance	a	$a = D_p/2 + H$	-	30.5

The result data of tooth dimension for rack and pinion are shown in Table III.

B. Contact Stress Analysis of Steering Gear by Using AGMA Equation

One of the main gear tooth failure is pittingwhich is a surface fatigue failure due torepetition of high contact stresses occurringin the gear tooth surface while a pair of teethis transmitting power [14Bab].

1. The contact stress equation is given as

$$\sigma_c = C_p \sqrt{\frac{F_t}{bdI}\left(\frac{\cos\psi}{0.95CR}\right) K_v K_0 \left(0.93K_m\right)} \quad (15)$$

2. The elastic coefficient factor equation is given as

$$C_p = 0.564 \sqrt{\frac{1}{\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}}} \quad (16)$$

3. The geometry factor I is given by

$$I = \frac{\sin\varphi\cos\varphi}{2}\frac{i}{i+1} \quad (17)$$

4. The speed ratio is given by

$$i = \frac{n_1}{n_2} = \frac{d_2}{d_1}$$

5. The contact ratio equation is given as

$$CR = \left[\frac{\sqrt{r_0^2 - r_R^2} + \frac{h_{ar}}{\sin\phi} - r_p \sin\phi}{\pi m_n \cos\phi}\right] \quad (18)$$

In the principle stress theory failure will occur when the principle stress in the complex system reaches the value of the maximum stress at the elastic limit in simple tension.

6.The principal stresses are determined by the following equation.

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \frac{1}{2}\sqrt{\left(\sigma_x - \sigma_y\right)^2 + 4\tau_{xy}^2} \quad (19)$$

With either yield criterion, it is useful to define an effective stress denoted as σ_v which is a function of the applied stresses. If the magnitude of σ_v reaches a critical value, then the applied stress state will cause yielding, in essence, it has reached an effective level. The von-Mises stress is calculated by the following equation:

7. The von-Mises stress is,

$$\sigma_v = \frac{1}{\sqrt{2}}[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2]^{1/2} \quad (20)$$

TABLE IV
THEORETICAL RESULT OF VON-MISES STRESSES AND STRAIN FOR GEAR PAIR

Parameter	Results
Von Mises Stress (max)	944.31 MPa
Strain (max)	4.04×10^{-3}

The stress and strain value for gear pair is shown in Table IV.

C. Theoritical Calculations of Rack Bending Stress And Deflection

Rack is subject to not only the axial load but also the vertical loading. The vertical load causes the bending stress and if the load is higher than critical load then it will be lead to breakage. Consider the vehicle load front axle weight of 740N. The rack vehicle load (W) come on rack end is calculated. The assembly is considered as cantilever beam. Pinion is fixed and then vertical load is applied the rack end. Refer Figure 2 for the rack assembly and loading conditions.

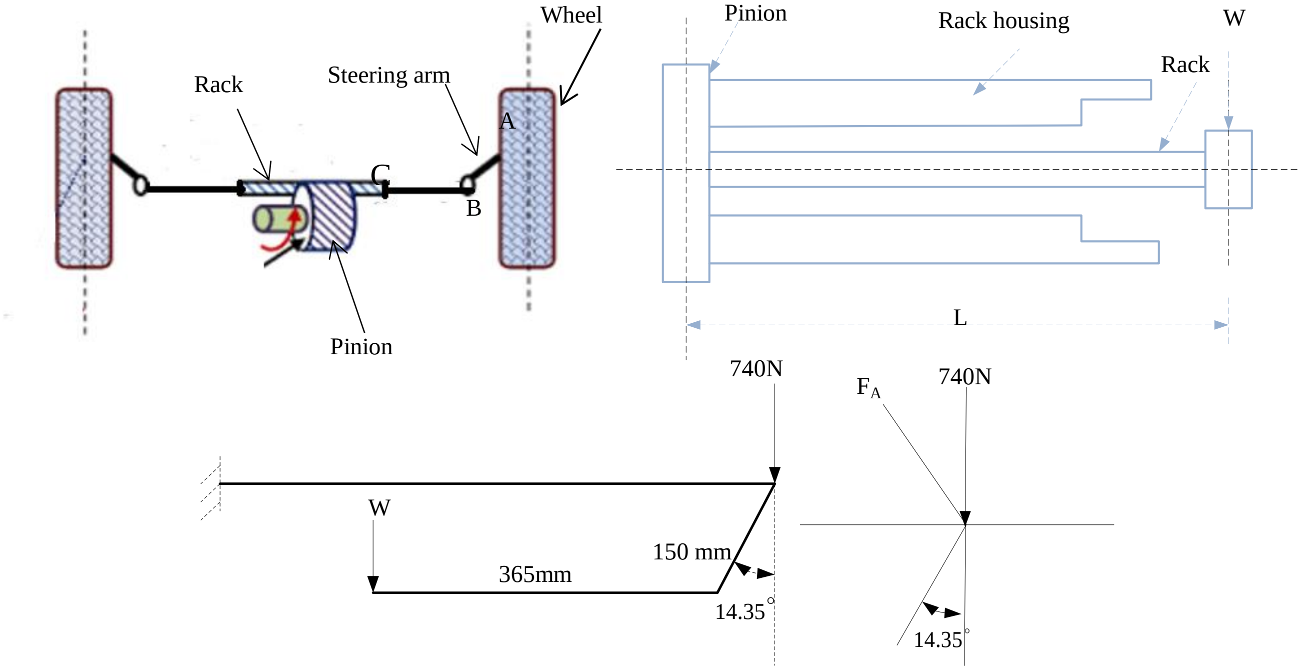

Figure 2. Rack Assembly and Loading Condition

Figure 3 shows the rack cross section view. The shows the minimum cross section of rack. The sector area is removed for making tooth on rack.

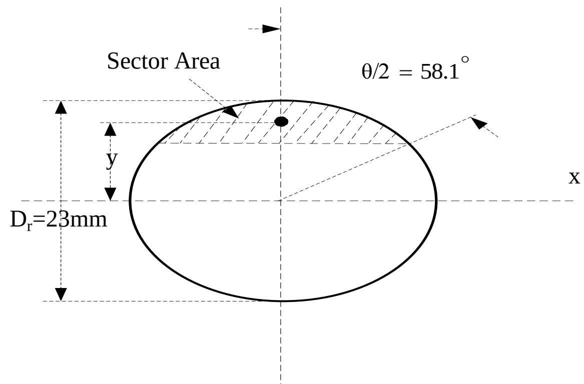

Figure 3. Rack Cross Section View

1. To find the center of gravity of sector area, y

$$y = \frac{4r}{3}\left(\frac{\sin^3\frac{\theta}{2}}{\theta - \sin\theta}\right) \tag{21}$$

2. To find the moment of inertia, I

$$I = \frac{r^4}{8}\left(\theta - \sin\theta + 2\sin\theta\sin^2\frac{\theta}{2}\right. \tag{22}$$

TableVshows specification data of rack and rack housing assembly.

TABLE V
SPECIFICATION DATA OF RACK AND RACK HOUSING ASSEMBLY

Parameter	Symbol	Value	Unit
Rack Diameter	D_r	23	mm
Center of Gravity of Sector Area	y	8.289	mm
Moment of Inertia	I	5299.80	mm^4
Rack Tip Max. Bend Load	W	294.35	N
Rack Length	L_r	210	mm
Young's Modulus	E	207	GPa

3. To find the Rack Plane coefficient, Z

$$Z = \frac{I}{y} \tag{23}$$

4. To find the Rack Stress, σ

$$\sigma = \frac{W.L}{Z} \tag{24}$$

5. To find the Deflection, δ

$$\delta = \frac{W.L^3}{3EI} \tag{25}$$

TABLE VI
THEORETICAL RESULT OF VON-MISES STRESSES AND DEFORMATION FOR RACK

Parameter	Results
Von Mises Stress (max)	96.782 MPa
Deformation (max)	0.828 mm

The stress and deformation value for rack in Table VI.

D. Design and Fatigue Analysis of Intermediate Steering Shaft

There are three steps in the design and fatigue analysis of rear axle shaft. They are (i) design calculation of intermediate steering shaft, (ii) stress analysis of intermediate steering shaft, and (iii) fatigue analysis of intermediate steering shaft.

Table VII shows the parameters considered for Intermediate Shaft.

TABLE VII

PARAMETERS CONSIDERED FOR DESIGN A HELICAL PINION GEAR

Design Parameter	Specification
Material	ASTM A36
Ultimate Strength (S_u)	400
Yield Strength (S_y)	250
Modulus of Elasticity (E)	207 GPa
Speed of Shaft (N)	15~8 rpm
Outside Diameter of Shaft (D_0)	19 mm
Torque (T)	6 Nm

(i) Design calculation of Intermediate Steering Shaft
The axle shaft is a rotating member, in general, has a circular cross-section and is used to transmit power. The shaft is generally acted upon by torsion.
The torque can be computed from the known power transmitted and the rotational speed.

1. Maximum power, $P = \frac{T \times N_{max}}{9550}$ (26)

2. Maximum Torque, $T_{max} = \frac{60P}{2\pi N_{minn}}$ (27)

3. Minimum Torque, $T_{min} = \frac{60\ P}{2\pi N_{max}}$ (28)

4. Mean Torque, $T_m = \frac{T_{max} + T_{min}}{2}$ (29)

5. Alternative Torque, $T_a = \frac{T_{max} - T_{min}}{2}$ (30)

6. Maximum Shear Stress, $\tau_{max} = \frac{S_u}{SF}$ (31)

SF = 15 (Maximum Safety Factor)

7. Inside Diameter of Intermediate Shaft, $\tau_{max} = \frac{16T_{max}}{\pi(D_0{}^3 - D_i{}^3)}$ (32)

(ii) Stress analysis of Intermediate Steering Shaft

1. Mean torsional shear stress; $\tau_m = \frac{16T_m}{\pi(D_0{}^3 - D_i{}^3)}$ (33)

2. Alternating torsional shear stress; $\tau_a = \frac{16T_a}{\pi(D_0{}^3 - D_i{}^3)}$ (34)

If τ_m and $\tau_a < \tau_{max}$, Design is satisfied.

3.The principal stresses are determined by the following equation.

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \frac{1}{2}\sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}{}^2} \quad (35)$$

4. The von-Mises stress is,

$$\sigma_v = \frac{1}{\sqrt{2}}[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2]^{1/2} \quad (36)$$

(iii) Fatigue Analysis of Intermediate Steering Shaft

1. Estimating the Theoretical Endurance Limit

$$S'_e = 0.504S_u \text{ for } S_u \le 1400\text{MPa} \quad (37)$$

2. The equation can be written to give corrected endurance limit (fatigue strength), S_e as follows:

$$S_e = k_a \times k_b \times k_c \times k_d \times k_e \times k_f \times S'_e \quad (38)$$

The coefficients are detailed below,

3. Surface Factor (k_a)

$$k_a = aS_u^b \quad (39)$$

Where, a and b are constants, they are to be found in Table.

4. Size Factor (k_b)

$$k_b = \left(\frac{d}{7.62}\right)^{-0.1133} \qquad 2.79\text{mm} \le d \le 51\text{mm} \quad (40)$$

5. Loading Factor (k_c)

$$k_c = \begin{cases} 1 & \text{bending} \\ 0.85 & \text{axial} \\ 0.59 & \text{torsion} \end{cases} \quad (41)$$

6. Temperature Factor (K_d)

$$k_d = \frac{S_T}{S_{RT}} \tag{42}$$

7. Reliability Factor (k_e)

Assume reliability factor is 99.99%, ke =1 (43)

8. Miscellaneous-effects Factor (k_f)

$k_f = 1$ (44)

9. Calculation of Number of Cycles to Failure

$$S_f = \frac{\sigma'_a}{1 - \frac{\sigma'_m}{S_u}} \tag{45}$$

Number of cycles to failure (*N)* can be expressed by using the fatigue strength of material; *f* is the fatigue strength fraction as shown in Table A-3 [2].

$$N = \left(\frac{S_f}{a}\right)^{\frac{1}{b}} \tag{46}$$

$$a = \frac{(f\, S_u)^2}{S_e} \tag{47}$$

$$b = -\frac{1}{3}\log\left(\frac{f\, S_u}{S_e}\right) \tag{48}$$

10. Fatigue factor of safety *(*n_f)

Fatigue factor of safety *(*n_f) can be calculated by using Modified Goodman Method

$$n_f = \frac{S_e S_u}{\sigma'_a S_u + \sigma'_m S_e} \tag{49}$$

TABLE VIII
THEORETICAL RESULT OF VON-MISES STRESSES AND SAFETY FACTOR FOR INTERMEDIATE SHAFT

Parameter	Results
Von Mises Stress (max)	39.092 MPa
Fatigue Safety Factor	3.706

Table VIII shows thetheoretical results for von-Mises stress and fatigue safety factor value.

III. ANALYSIS OF THE STEERING SYSTEM COMPONENTS

Analysis is process of analyzing the components by applying external factors such as loads, temperature, pressure etc. and obtaining the values such as stresses (bending, tangential and normal), deformations etc. in order to determine the safety of the components when implemented in practical use. It gives optimum result of the safety of components and very easy to understand various factors applicable in the process. These Analyses gives optimum result of safety of components and minimize the chances of failure. There are various packages in market to carry out these simulations on computer such as ANSYS, SolidWorks, HYPERWORKS, and FLOTRAN etc. In this project we have used ANSYS as the software to analyze the safety of our components under various loading conditions.

Two major analyses carried out in this project are:

1) Deformation analysis
2) Stress analysis

Various components analyzed in this project are:

1) Steering Gear
2) Steering Rack

3) Intermediate Steering Shaft

Process for performing the analysis:

1) Making or importing the geometry to software interface (GUI).
2) Defining the field.
3) Applying the material properties.
4) Meshing the components with appropriate element size.
5) Applying the actions such as load, pressure etc. on the body.
6) Applying the boundary conditions such as fixed supports (constraints).
7) Solving using the solver.
8) Obtaining required reactions such as stresses, deformations etc.

A. Gear Tooth Strength Analysis of Steering Pinion Gear

The material properties of AISI 5160 OQT 400 is given in the Table IX.

TABLE IX

MATERIAL PROPERTIES OF AISI 5160 OQT 400

Material Properties	value
Young modulus	207 GPa
Poisson ratio	0.3
Density	7850 kg/m3
Coefficient of Thermal Expansion	1.15e-05 C^-1
Tensile Yield Strength	1570 MPa
Tensile Ultimate Strength	1950 MPa

(i) Simulation Result (Boundary Condition)

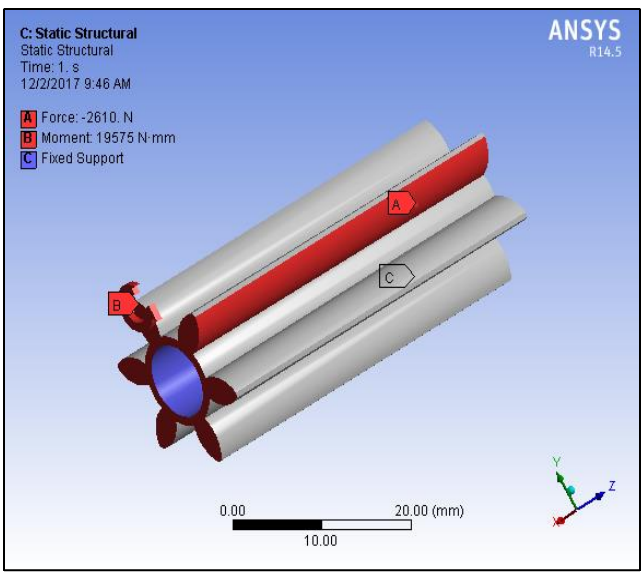

Figure 4. Fixed and forced Condition of Pinion Gear

Figure 4 shows the fixed and forced conditions for structural analysis of pinion gear. There are three boundary conditions for structural; fixed support, moment and force.

(ii) Simulation Result (Von- Mises Stress and Strain)

Stress analysis is used to determine equivalent stress and strain of the pinion gear. The numerical results of stress analysis are carried out by using ANSYS 14.5 software.

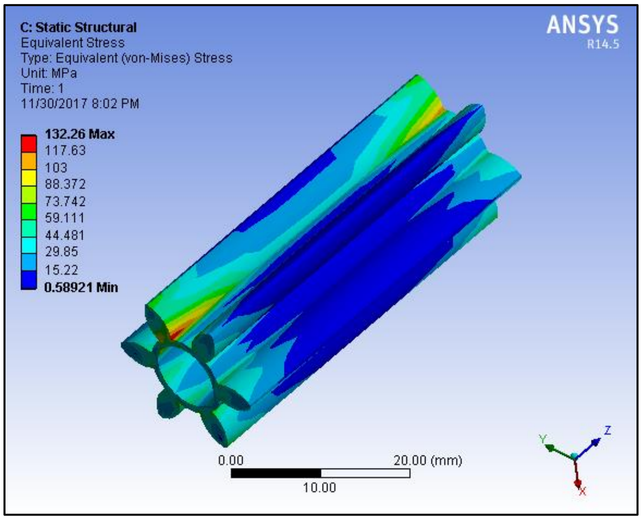

Figure 5.Equivalent (von-Mises) Stress on Pinion Gear

Figure 5 shows minimum and maximum von-Mises stress. The values are 0.58921 N/mm^2 (MPa) and 132.26 N/mm^2 (MPa). Comparing Yield strength value 1570N/mm^2 (MPa), the maximum value is less than yield strength. So the design is satisfied. The steering pinion will work safely at this stress.

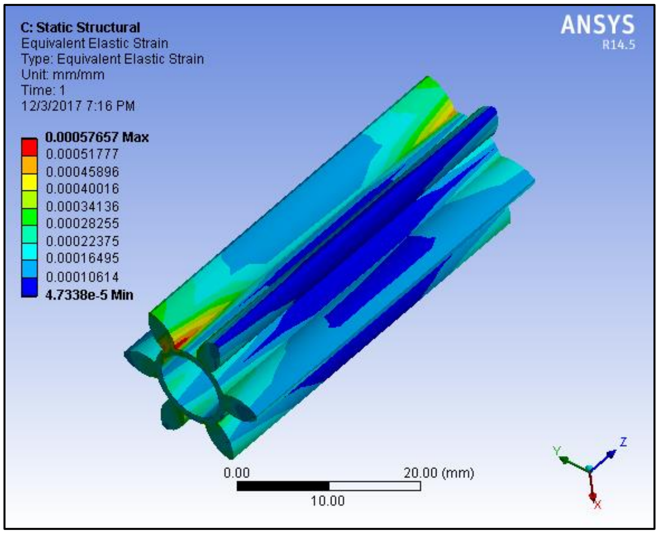

Figure6.Equivalent Strain of Pinion Gear

Figure 6 shows the equivalent strain values. The minimum value is 4.7338×10^{-5} and the maximum value is 0.00057657.

B. Stress Analysis of Steering Shaft for Pinion Gear

Stress analysis was analyzed by ANSYS software and material properties of AISI 4340 OQT 400 is given in the Table IX.

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TABLE X

MATERIAL PROPERTIES OF AISI 4340 OQT 400

Material Properties	value
Young modulus	207 GPa
Poisson ratio	0.3
Density	7850 kg/m3
Tensile Yield Strength	1570MPa
Tensile Ultimate Strength	1950MPa

Table XIshows specification data of steering shaft for pinion gear.

TABLE XI

SPECIFICATION DATA OF STEERING SHAFT FOR PINION GEAR

Description	Symbol	Values
Torque (Nm)	T	6
Tangential Force (N)	F_t	2610
Axial Force (N)	F_a	1108
Radial Force (N)	F_r	950
Diameter of Shaft (mm)	D	8
Length of Shaft (mm)	L	105

(i) Simulation Result (Boundary Condition)

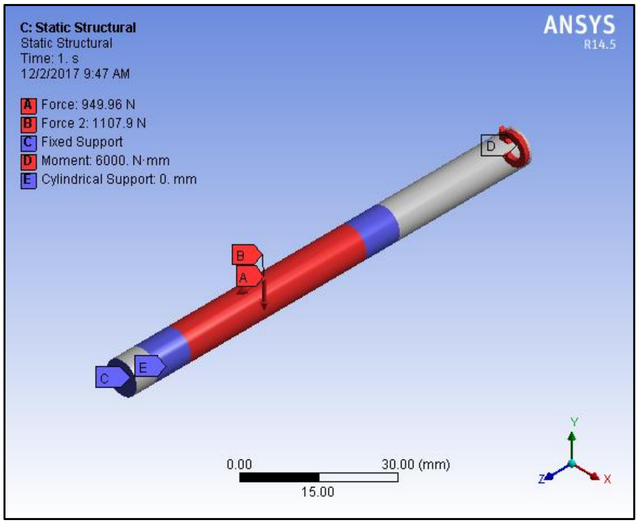

Figure 7. Fixed and forced Condition of steering shaft for pinion gear

Figure 7shows the fixed and forced conditions for structural analysis of shaft. There are four boundary conditions for structural; fixed support, cylindrical support, moment and force.

(ii) Simulation Result (Von- Mises Stress and Strain)

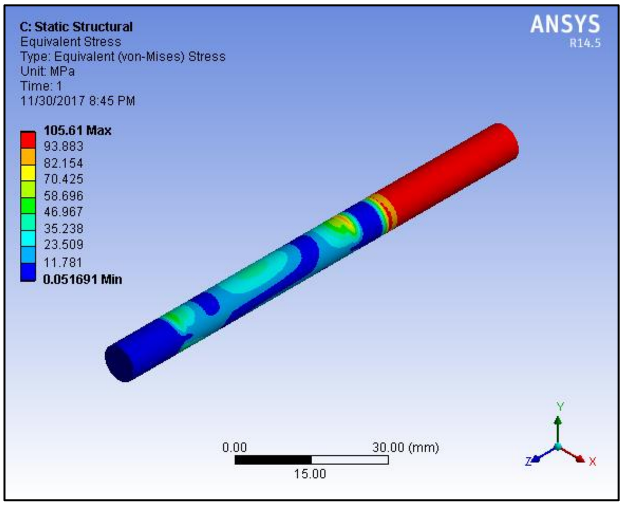

Figure 8.Equivalent (von-Mises) Stress Analysis of Shaft

Figure 8 shows minimum and maximum von-Mises stress. The values are 0.051691 N/mm^2 (MPa) and 105.61 N/mm^2 (MPa), the maximum value is less than yield strength. So, shaft design is satisfied for pinion gear of steering system.

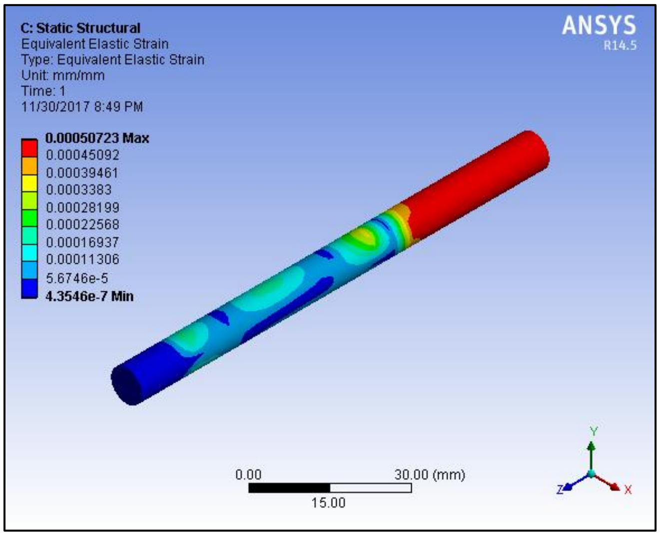

Figure 9.Equivalent Strain of Shaft

Figure 9 shows the equivalent strain values. The minimum value is 4.3546×10^{-7} and the maximum value is 0.00050723.

C. Stress Analysis of Steering Shaft and Pinion Gear Assembly

Structural analysis of shaft and gear assembly was analyzed by ANSYS software.

(i) Simulation Result (Boundary Condition)

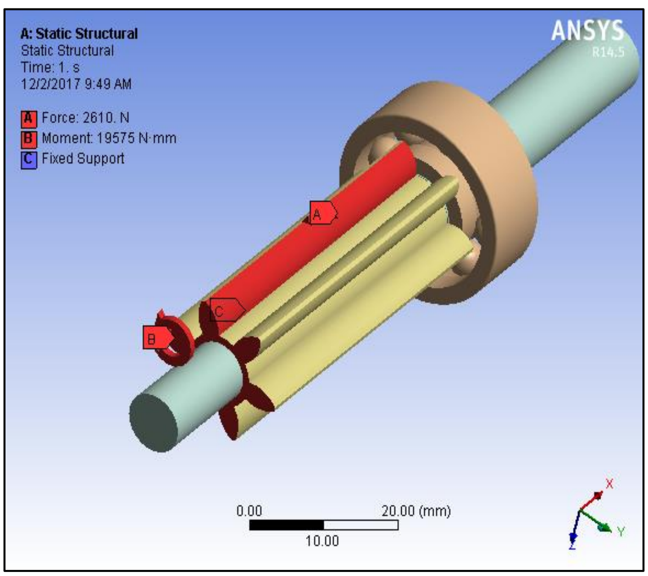

Figure 10. Fixed and forced Condition of Shaft and Pinion Gear Assembly

Figure 10shows the fixed and forced conditions for structural analysis of shaft and pinion gear assembly. There are three boundary conditions for structural; fixed support, moment and force.

(ii) Simulation Result (Von- Mises Stress and Strain)

Figure 11 shows minimum and maximum von-Mises stress. The values are 4.4846×10^{-9} N/mm^2 (MPa) and 392.61 N/mm^2 (MPa). Comparing Yield strength value 1570 N/mm^2 (MPa), the maximum value is less than yield strength. So the design is satisfied.

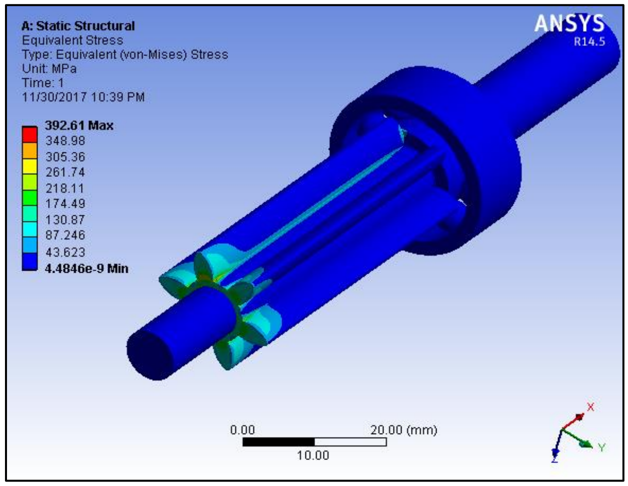

Figure 11.von-Mises Stress of Pinion Gear and Shaft Assembly

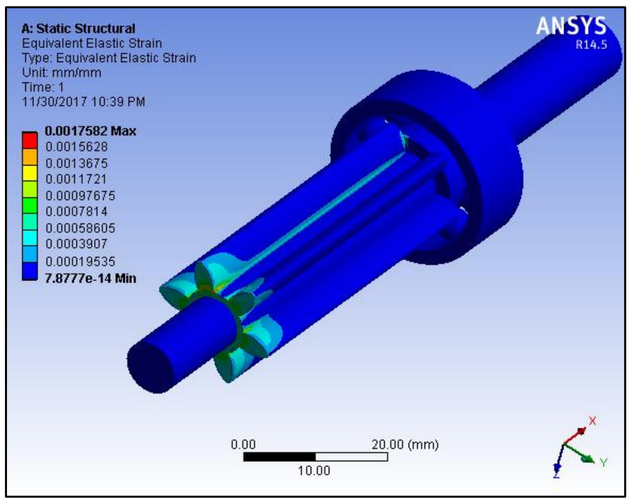

Figure12.Equivalent Strain of Pinion Gear and Shaft Assembly

Figure 12 shows the equivalent strain values. The minimum value is 7.8777×10^{-14} and the maximum value is 0.0017582.

D. Contact Stress Analysis of Steering Rack and Pinion Gear

(ii) Simulation Result (Boundary Condition)

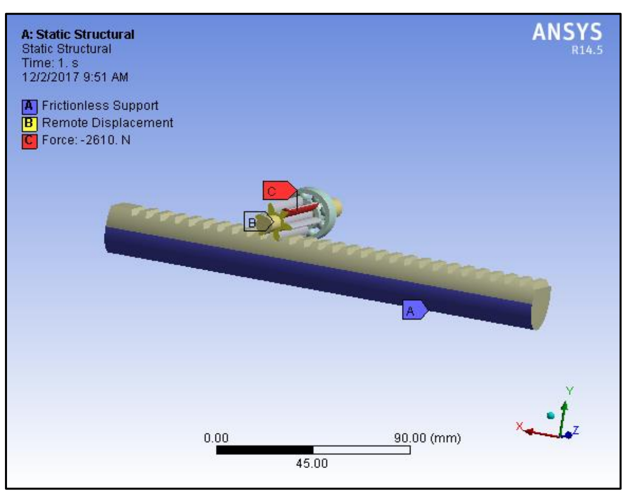

Figure 13. Fixed and forced Condition of Rack and Pinion Gear Assembly

(ii) Simulation Result (Contact Stress)

Stress analysis is used to determine equivalent stress and the strain of the rack and pinion gear assembly contact point. The numerical results of stress analysis are carried out by using ANSYS 14.5 software. The numerical results of von-Mises stress and strain are compared with different materials of rack and pinion gear (AISI 4340 steel, aluminum alloy and gray cast iron).

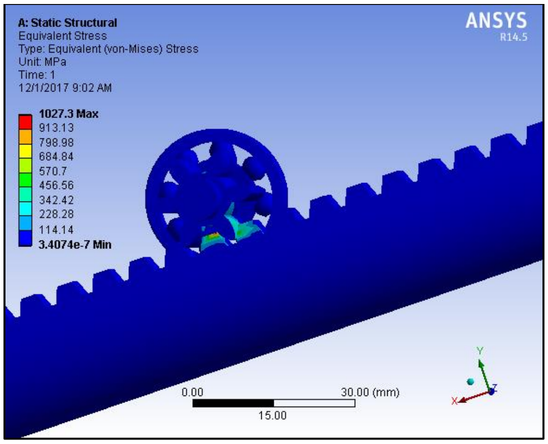

Figure 14.Equivalent (von-Mises) Stress on Rack and Pinion Gear Assembly using AISI 4340 Steel

Figure 14 shows the equivalent (von-Mises) stress on rack and pinion gear assembly using AISI 4340 steel. The maximum equivalent (von-Mises) stress on contact point is 1027.3 MPa and location of maximum stress is at the meshing area of the rack and pinion while the yield strength of the structural steel is 1570 MPa. The steering gear will work safely at this stress.

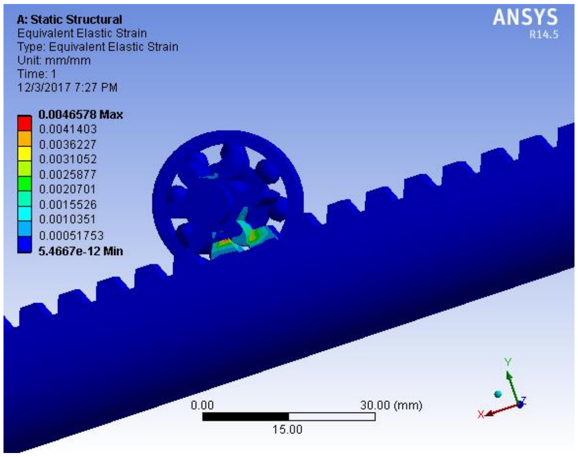

Figure15.Equivalent Strain of Rack and Pinion Gear Assembly using AISI 4340 steel

The equivalent strain on rack and pinion gear pair using AISI 4340 steel is 0.0046578 and occurs at meshing area of the rack and pinion as shown in Figure 15.

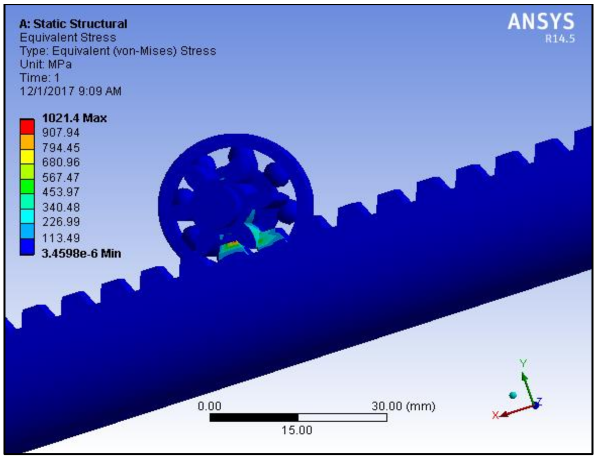

Figure 16.Equivalent (von-Mises) Stress on Rack and Pinion Gear Assembly using Aluminum Alloy

Figure 16 shows the equivalent (von-Mises) stress on rack and pinion gear assembly using Aluminum. The maximum equivalent (von-Mises) stress on contact point is 1021.4 MPa and location of maximum stress is at the meshing area of the rack and pinion while the yield strength of the structural steel is 225 MPa. The steering gear will not work safely at this stress.

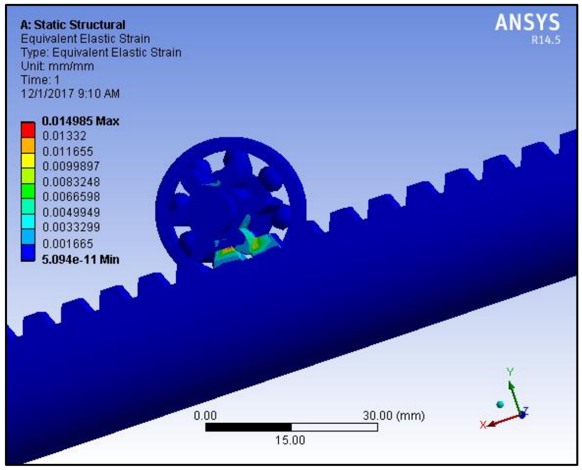

Figure17.Equivalent Strain of Rack and Pinion Gear Assembly using Aluminum Alloy

The equivalent strain on rack and pinion gear pair using Aluminum is 0.014995 and occurs at meshing area of the rack and pinion as shown in Figure 17.

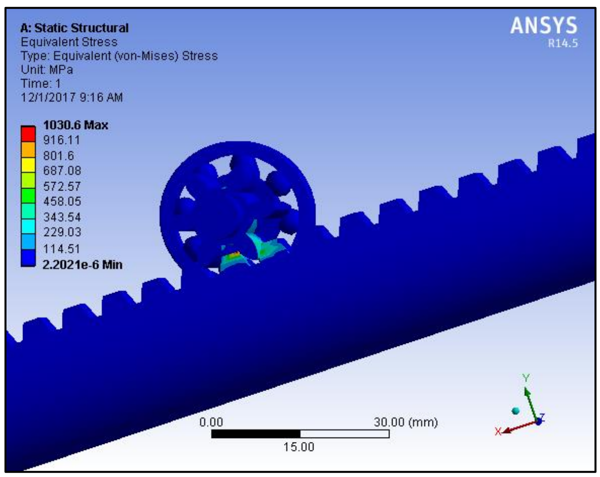

Figure 18.Equivalent (von-Mises) Stress on Rack and Pinion Gear Assembly using Gray Cast Iron

Figure 18 shows the equivalent (von-Mises) stress on rack and pinion gear assembly using Gray Cast Iron. The maximum equivalent (von-Mises) stress on contact point is 1030.6 MPa and location of maximum stress is at the meshing area of the rack and pinion while the yield strength of the structural steel is 170 MPa. The steering gear will not work safely at this stress.

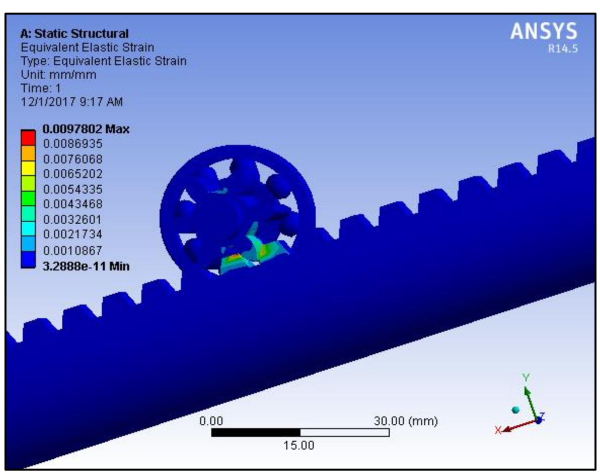

Figure19.Equivalent Strain of Rack and Pinion Gear Assembly using Gray Cast Iron

The equivalent strain on rack and pinion gear pair using Gray Cast Iron is 0.0097802 and occurs at meshing area of the rack and pinion as shown in Figure 19.

TABLE XII

COMPARISON OF THEORETICAL AND SIMULATION RESULT FOR GEAR ASSEMBLY

Parameter	Results		
	Theoretical calculation	Simulation	% Error
Von Mises Stress (max)	944.031 MPa	1027.3 MPa	8.1%
Strain (max)	4.04×10^{-3}	4.65×10^{-3}	13.1%

Table XII shows thecomparison of theoretical and simulation results for von-Mises stress and strain value of the gear pair.

(iii) Comparison of Von-Mises Stress and Strain on the Rack and Pinion Gear Assembly with Three Types of Materials

Comparative study on contact stress analysis has been carried out on the rack and pinion gear assembly made of different materials namely AISI 4340, Aluminum Alloy and Gray Cast Iron which are suitable for rack and pinion gear pair.

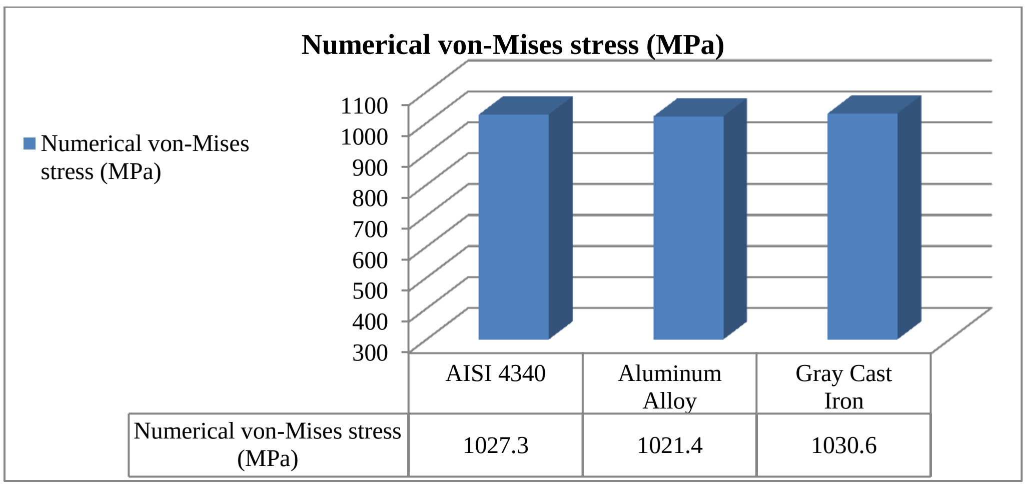

Figure 20.Comparison of Numerical Result for Equivalent (von-Mises) Stress

Figure 20 show von-Mises stress on rack and pinion gear assembly with three types of materials by using ANSYS 14.5.

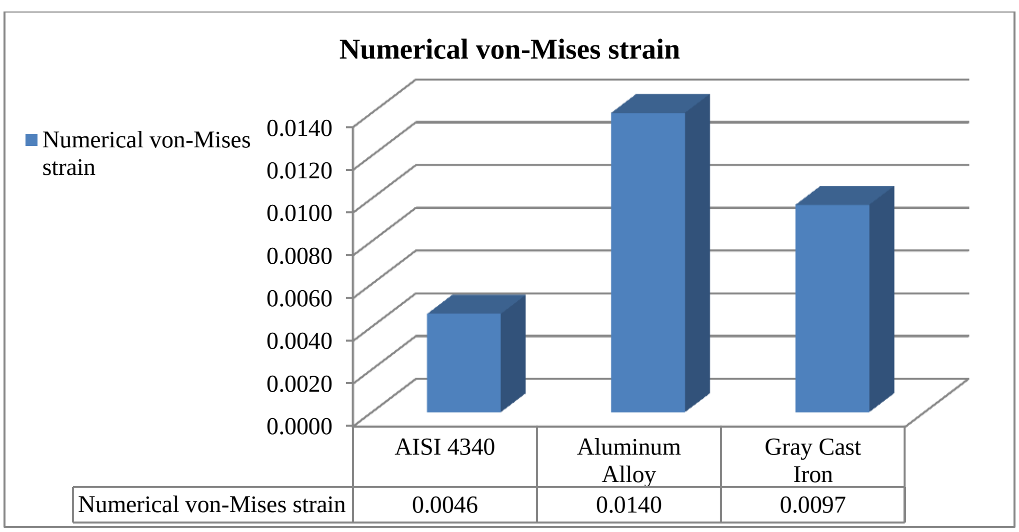

Figure 21.Comparison of Numerical Result for Equivalent (von-Mises) Strain

Figure 21 show von-Mises strain on rack and pinion gear assembly with three types of materials by using ANSYS 14.5. The resulted data for static structural analysis with three types of materials are compared with von-Mises stress and strain. Static analysis reveals that maximum stress on rack and pinion gear pair are 1027.3 MPa, 1021.4 MPa and 1030.6 MPa respectively and maximum strain on rack and pinion gear pair are 0.0046 mm, 0.014 mm and 0.0097 mm. The maximum von-Mises stresses on rack and pinion gear pair are occurred at the meshing area of the rack and pinion gear. From the structural contours of ANSYS, it can be observed that the maximum strains are also occurred at the meshing area of the rack and pinion gear. The von-Mises stresses on gear pair using different materials are nearly the same. The analysis shows that the strain has minimum with AISI 4340 steel. Therefore, AISI 4340 steel is suitable material in this designed steering gear pair.

E. Stress Analysis of Steering Rack and Rack Housing

Stress analysis of steering rack and rack housing assembly was analyzed by ANSYS software. Rack material is used as height tensile steel, as standard material AISI 4340 steel. Mechanical properties of all the materials are in Table XIII.

TABLE XIII

MECHANICAL PROPERTIES FOR STEERING RACK AND RACK HOUSING

Part Name	Material	Young's Modulus (GPa)	Poisson's Ratio	Tensile Strength (MPa)	Yield Strength (MPa)
Pinion	Steel AISI 4340	207	0.3	1950	1570
Rack	Steel AISI 4340	207	0.3	1950	1570
Housing	Aluminum Alloy	71	0.33	273	225

(i) Simulation Result (Boundary Condition)

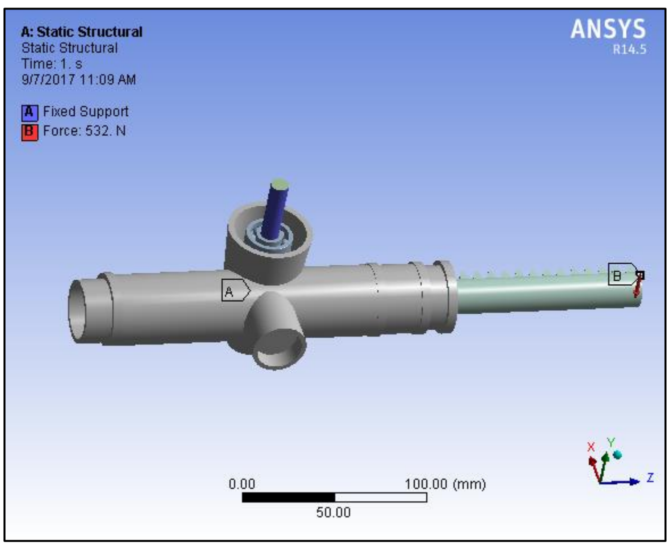

Figure 22. Fixed and Forced Condition of Rack and Rack Housing Assembly

Figure 22 shows the fixed and forced conditions for structural analysis of rack and rack housing assembly. There are two boundary conditions for structural; fixed support and force.

(ii) Simulation Result

Stress analysis is used to determine equivalent stress and the total deformation of the rack. The numerical results of stress analysis are carried out by using ANSYS 14.5 software.

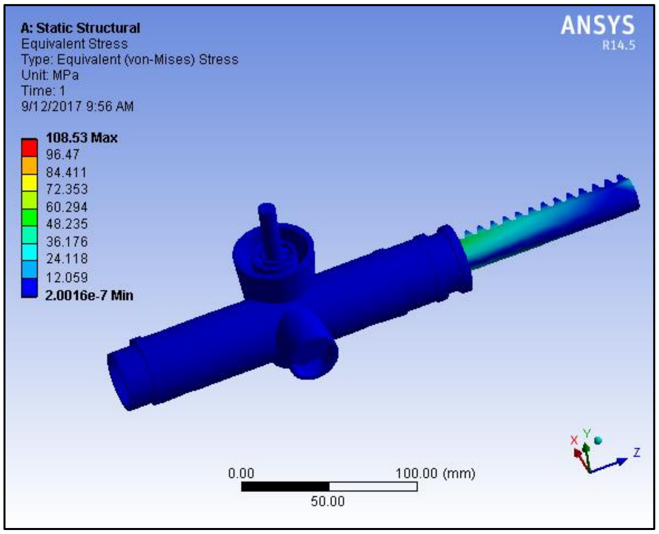

Figure. 23. Equivalent (von-Mises) Stress on Rack and Rack Housing Assembly

Figure 23 shows the equivalent (von-Mises) stress on rack and rack housing assembly. The maximum equivalent (von-Mises) stress is 108.53 MPa while the yield strength of the structural steel is 1570 MPa. The steering rack will work safely at this stress.

(iii) Deformation Result

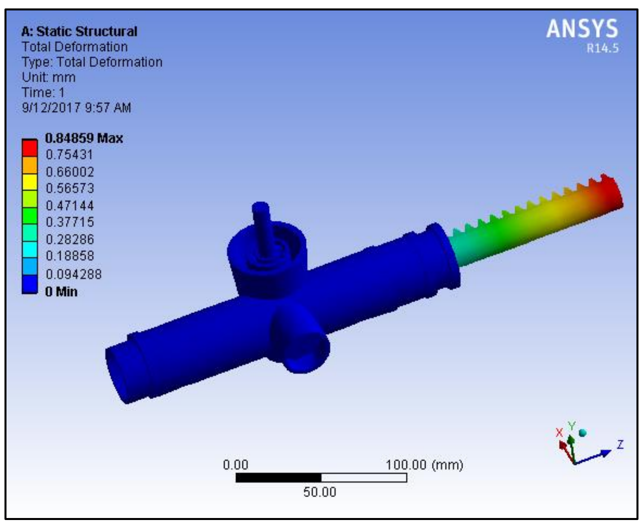

Figure 24. Deformation on Rack and Rack Housing Assembly

Figure 24 shows the deformation values. The minimum value is 0 and the maximum value is 0.84859 mm.

TABLE XIV

COMPARISON OF THEORETICAL AND SIMULATION RESULT FOR RACK AND RACK HOUSING ASSEMBLY

Parameter	Results		
	Theoretical calculation	Simulation	% Error
Von - Mises Stress (MPa)	96.782	108.53	10.8%
Deformation (mm)	0.828	0.849	2.5%

Table XIV shows thecomparison of theoretical and simulation resultsfor von-Mises stress and deformation value of the rack.

F. Stress Analysis and Fatigue Analysis of Intermediate Steering Shaft

Stress analysis of intermediate steering shaft was analyzed by ANSYS software. Intermediate steering shaft material is used as steel, as standard material ASTM A36 steel. Mechanical properties of the materials are in Table XV.

TABLE XV

MECHANICAL PROPERTIES FOR INTERMEDIATE SHAFT

Properties	
Name	ASTM A36 steel
Model Type	Linear Isotropic
Default Failure Criterion	Max Von Mises Stress
Yield Strength	250MPa
Tensile Strength	400MPa
Elastic Modulus	200GPa
Poisson's Ratio	0.3
Mass Density	7850kg/m^3

The static structural shaft SolidWorks model was added to the geometry in ANSYS Workbench. This geometry model was meshed with high smoothing. This meshed model was imported to static structural for static structural analysis of the intermediate steering shaft. Firstly, give the input conditions to the model, which applied the torque 6000 N-mm at end of steering intermediate shaft. Then, supports are fixed at another end as shown in Figure 25. The engineering data for type of material uses the structural steel for testing material.

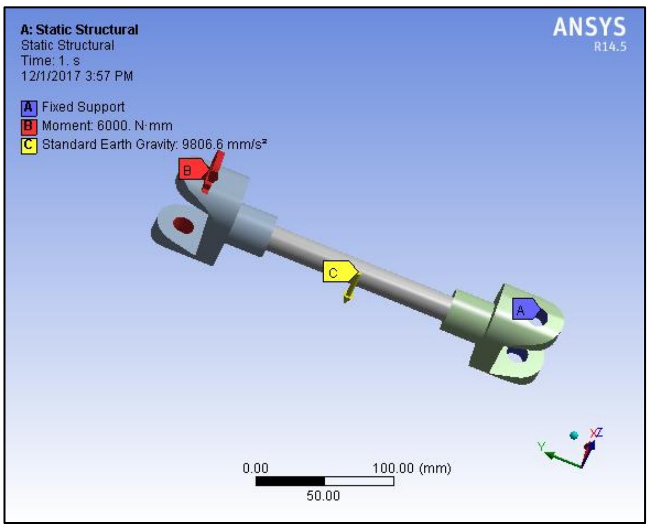

Figure 25.Loading Condition of Intermediate Steering Shaft

After finishing set up the boundary conditions on intermediate steering shaft, run the solution and get equivalent (von-Mises) stress and deformation.

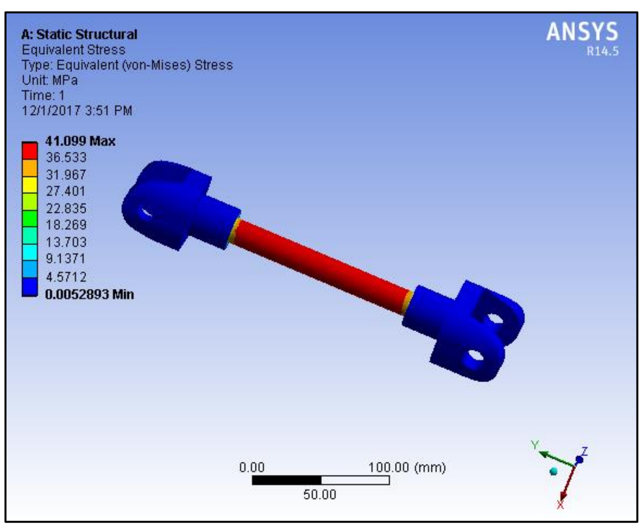

Figure 26.Equivalent (von-Mises) Stress of Intermediate Steering Shaft

Figure 26 shows the equivalent (von-Mises) stress on intermediate steering shaft. The maximum equivalent (von-Mises) stress is 41.099 MPa while the yield strength of the structural steel is 250 MPa. The steering shaft will work safely at this stress.

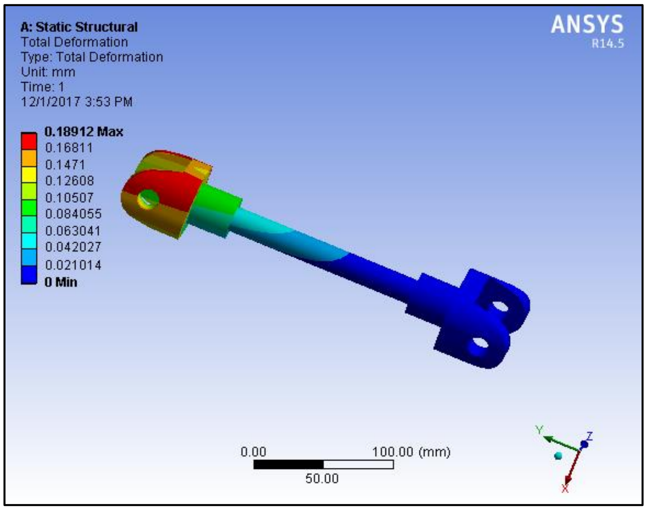

Figure 27.Total Deformation of Intermediate Steering Shaft

Figure 27 shows the deformation values. The minimum value is 0 and the maximum value is 0.18912 mm.

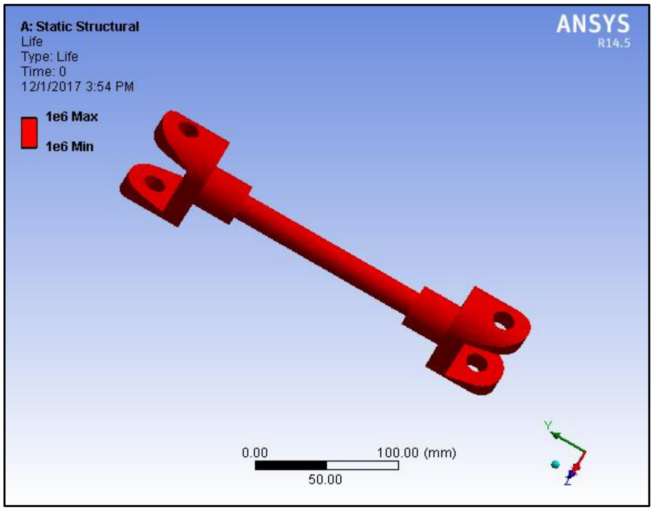

Figure28.Life Cycle of Intermediate Steering Shaft

In the fatigue tool bar, the life cycles of intermediate steering shaft can be solved. Figure 28 shows the results of life cycles structural steel intermediate steering shaft.

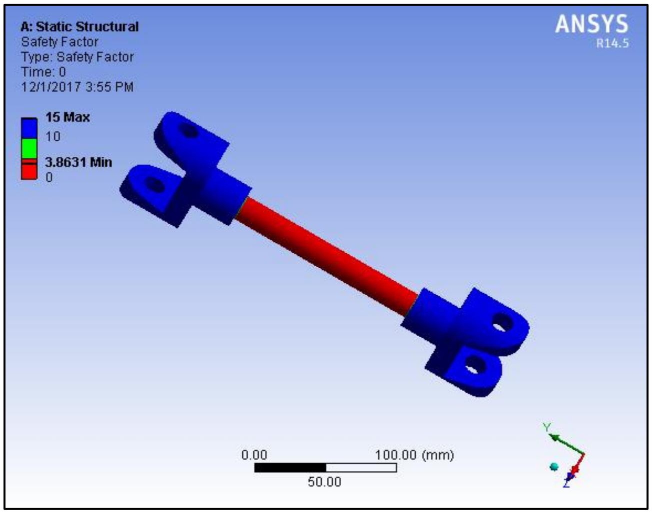

Figure 29.Factor of Safety on Intermediate Steering Shaft

Then, the factor of safety on steering intermediate steering shaft is also shown in Figure 29. The maximum and minimum factor of safety is 15 and 3.8631.

TABLE XVI

COMPARISON OF THEORETICAL AND SIMULATION RESULT FOR INTERMEDIATE STEERING SHAFT

Parameter	Results		
	Theoretical calculation	Simulation	% Error
von-Mises Stress (MPa)	39.092	41.099	4.8%
Fatigue Safety Factor	3.706	3.863	4.1%

Table XVI shows thecomparison of theoretical and simulation resultsfor von-Mises stress and fatigue safety factor of the steering intermediate shaft.

IV. CONCLUSION

Manual rack and pinion steering system is suitable for solar car. In steering gear design, the diameter of pinion 15mm and face width 47mm and module 2.5mm was satisfied. The von Mises stress and strain of steering shaft and pinion gear have been compared with theoretical and simulation result. The contact stress of steering rack and pinion gear pair have been compared in theoretical and simulation result with different materials. From analysis of rack and pinion gear pair, the von-Mises stress for steel was 1027.3 MPa, aluminum alloy was 1021.4 MPa and gray cast iron was 1030.6 MPa. From analysis of rack and pinion steering gear pair, strain results for steel were 0.0046, aluminum alloy was 0.014 and gray cast iron was 0.0097. In structural analysis, steel rack and pinion steering gear pair is having least strain value. Hence steel rack and pinion steering gear pair was safe for design.

This research also analyzed the stress of the steering rack. By using Finite Element method, a stress analysis has been carried out. Steering Rack Deflection and Bending stresses are found. These stresses are compared with analytical results. Modeling has been done by SolidWorks and Analysis has been done by ANSYS software. From analysis of rack, the von-Mises stress was 108.53MPa and a deformation result was 0.849mm. From analytical result, the von-Mises stress was 96.782MPa and a deformation result was 0.828mm. Error Percent was 10.8% for von-Mises stress and 2.5% for deformation.

This research has been studied the stress and fatigue analysis for intermediate steering shafts. This research focused on the stress analysis. It is caused by torsion. The models of intermediate steering shafts are also drawn by SolidWorks software. In the numerical analysis, stress and fatigue analysis of intermediate steering shafts are considered base on ANSYS software. Fatigue analysis of intermediate steering shaft is done to find the life of the intermediate steering shaft in cycles and determined the factor of safety of the shaft. From analysis of intermediate shaft, the von-Mises stress was 41.099 MPa and a safety factor was 3.863. Maximum stress occurs at the corner points of the circular hole. From analytical result, the von-Mises stress was 39.092MPa and a safety factor was 3.706. Error Percent was 4.8% for von-Mises stress and 4.1% for deformation.

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