Design and Development of Major Components for Gearbox

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Abstract-- *This Project deals with understanding of the gear transmission system principles with its design and working. Different types of gears are used in automobiles. Gears have teeth which mesh with each other to transmit the drive. The components of gear box have also been developed according to the theoretical calculations to validate the design with FEA and an brief study of speed reducer.*

Key Word: Design of gearbox, Developments of gearbox

1.1 INTRODUCTION

A transmission is a machine in a power transmission system, which provides controlled application of the power. Often the term transmission refers simply to the gearbox that uses gears and gear trains to provide speed and torque conversions from a rotating power source to another device.

1.2 GIVEN TECHNICAL DATA

Inclination angle α= 15

Friction between tire and road surface μ = 0.35

Total weight of vehicle with driver and fruits W = 365 + 25 = 390 Kg

Maximum vehicle speed V = 10 Km/hr

Diameter of wheel D = 800 mm = 0.8 m

1.3 REQUIRED POWER CALCULATION

- **Vehicle speed in m/s V= 10×103 /3600 = 2.78 m/s**
- **Total rpm required of wheel to achieve speed of 10 Km/hr**

$$V = (\pi \times D \times N) / 60$$

$$2.78 = (\pi \times 0.8 \times N)/60$$

$$N = 66.32 \text{ rpm}$$

- **Required total tractive force to propelled vehicle at wheel Fw**

Fw = Rolling resistance + Gradient resistance

Rolling resistance Ft $= W \times g \times \mu$ where $g = 9.81$

$= 390 \times 9.81 \times 0.35$

$= 1339.065$ N.

Gradient resistance Rg $= W \times g \times \sin\alpha$

$= 390 \times 9.81 \times \sin 15$

$= 990.22$ N.m

Fw = 1339.065 N. + 990.22 N.

= 2329.28 N.

- **Total required torque at single wheel Tw**

$Tw = (Fw \times r)/2 = 2329.28 \times (0.8/2) / 2$

$= 931.7$ N.m

- **Required power of motor**

$P = 2\pi NT/60000 = (2\pi \times 66.32 \times 931.7/60)$

= 6.5 KW

1.4 REQUIRED GEAR RATIO

Motor rpm = 932

Required wheel rpm = 66.32

Gear ratio G = 932/66.32 = 14

If we reduced motor rpm 14 times than output torque is = 67 × 14 = 938 N.m

And required torque at wheel is = 931.7 N.m

That means it is sufficient

1.5 SELECTION OF GEAR TRAIN

Simple gear train is not possible because of large gear reduction so Compound gear train is suitable by considering gear ratio is 14.

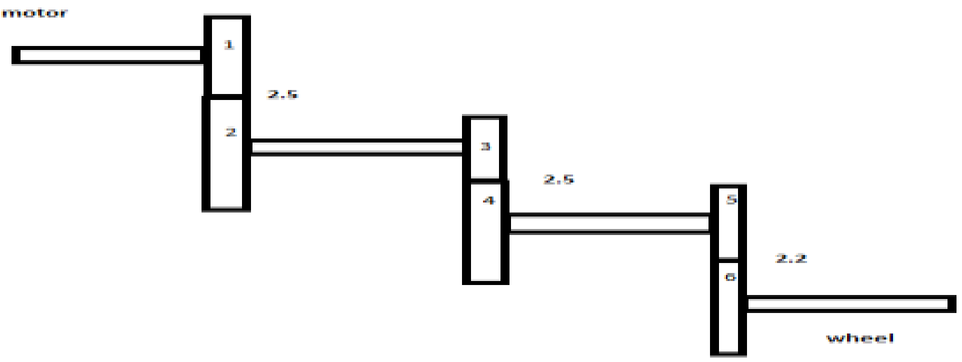

Figure 1 Proposed schematic diagram of compound gear train

Approximate over all Gear ratio of selected train is = 2.5 × 2.5 × 2.2 = 13.75

Here total six gears are required for compound gear train. So many types of gears are available like spur gear, bevel gear, helical gear, spiral gear having different pressure angle. Standard gear of 20^{o} full depth involute spur gear is selected for gearbox development because of easy manufacturing and cheaper compare to other gears.

1.6 DESIGN OF GEARS 1

Factor of safety for gear is varies from 1.5 to 2

In practice optimum range of face width is varies from 8 module to 12 module

Minimum teeth require for 20 of full depth involute gear is 18

➢ **Find module (m) of gear1**

$$\mathbf{m} = \left[\frac{60 \times 10^{\wedge}6}{\pi} \times \left\{\frac{P \times Cs \times fs}{Zp \times n \times Cv \times \frac{b}{m} \times \sigma ut \times \gamma}\right\}\right]^{1/3} \quad \text{..............................(1)}$$

Assume teeth of gear1 Z1 = 18 and pitch line velocity is 5 m/s and b/m = 10

Constant Cv = 3/ (3+v) = 3/8

Lewi's form factor is γ = 0.308 for 18 teeth from table

Plain carbon steel of 40 C 8 material is selected for gear

Ultimate tensile stress is σut = 600 Mpa

Put all above value in equestrian (1)

Module m = 4.18

Select first preferred value of module as per standard m = 5

- **Check design of gear1**

m = 5

Zp1 = 18

Np1 = 392

dp1 = m×Zp1 = 90 mm

Tangential load on gear Pt1 = 2Tp1/dp1 = $2\times67\times10^3$ /90 = 1488.89 N

Velocity of gear v = (π×dp1×Np1)/60000 = 4.39 m/s

Service factor Cs = Max torque/Rated torque = 67/45 = 1.5

Effective load between two meshing gear Pef = (Cs × Pt1) / Cv = 5501.45 N

Beam strength (Sb) is maximum value of tangential force that the tooth can transmit without bending failure

Sb = m×b×(σut/3)×γ = 5×50×(600/3)×0.308 = 15400

Factor of safty Fs = Sb/Pef = 2.8

That means our selected gear is in safe condition

1.7 SPECIFICATION OF GEAR

Do design of all remaining gears according to above design process of gear 1.

Parameters	Gear 1	Gear 2	Gear 3	Gear 4	Gear 5	Gear 6
Module (m) in mm	5	5	5	5	5	5
Number of teeth (Z)	18	45	18	45	18	40
Pitch circle diameter = Dp (mm)	90	225	90	225	90	200
Pitch = (πD)/Z in mm	15.707	15.707	15.707	15.707	15.707	15.707
Addendum (a)= m (mm)	5	5	5	5	5	5
Addendum circle = m + 2a (mm)	100	235	82	235	82	210
Dedendum Dd= 1.25 m (mm)	6.25	6.25	6.25	6.25	6.25	6.25
Dedendum circle = D – 2(Dd)	77.5	212.5	77.5	212.5	77.5	187.5
Working depth = 2m (mm)	10	10	10	10	10	10
Whole depth = 2.25m (mm)	11.25	11.25	11.25	11.25	11.25	11.25
Tooth thickness = 1.5708 m (mm)	7.854	7.854	7.854	7.854	7.854	7.854
Rpm	392	372.8	372.8	149.12	149.12	40

1.8 FEA OF GEAR 1

Pinion is weaker compare to gear that means if pinion is in safe condition, gear is also safe. So do FEA of gear 1, gear 3 and gear 5 because those are pinion of our gearbox.

➢ Boundary condition

Apply fixed support at surface A and force of 1489 on teeth as at surface B. also apply rotation velocity at centre of gear as shown in figure 2

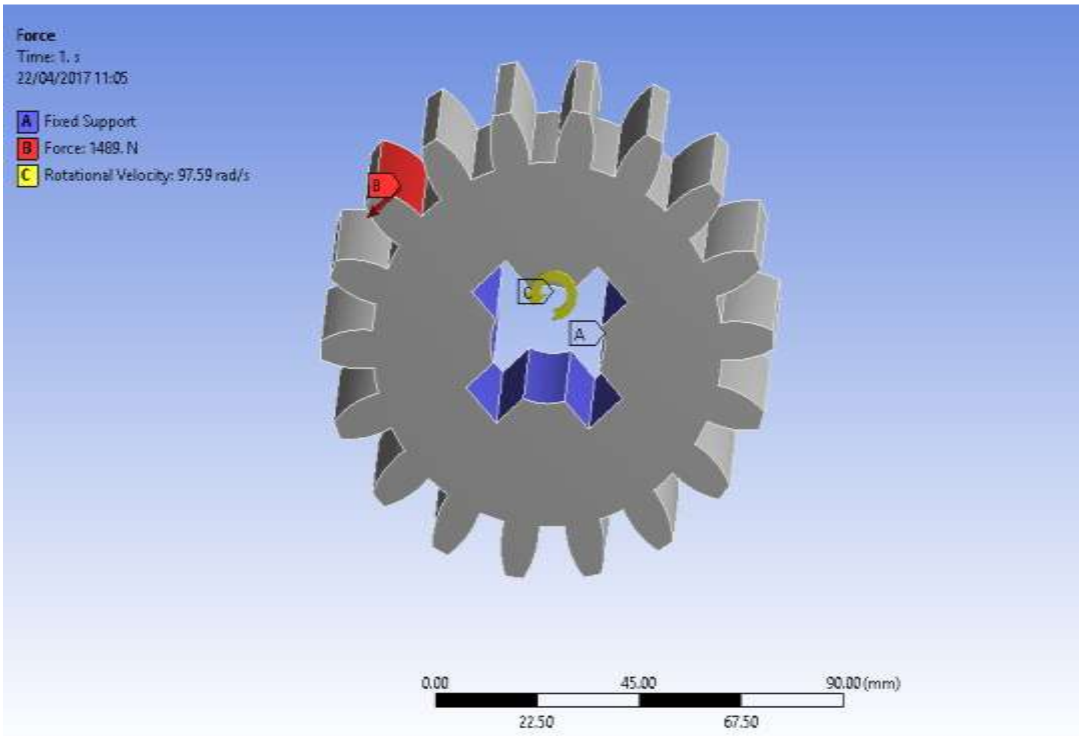

Figure 2 Boundary condition of gear 1

➢ Equivalent stress

Here Von-mises stress is only 18 Mpa which is safer than permissible limit of 105 Mpa as shown in figure 3

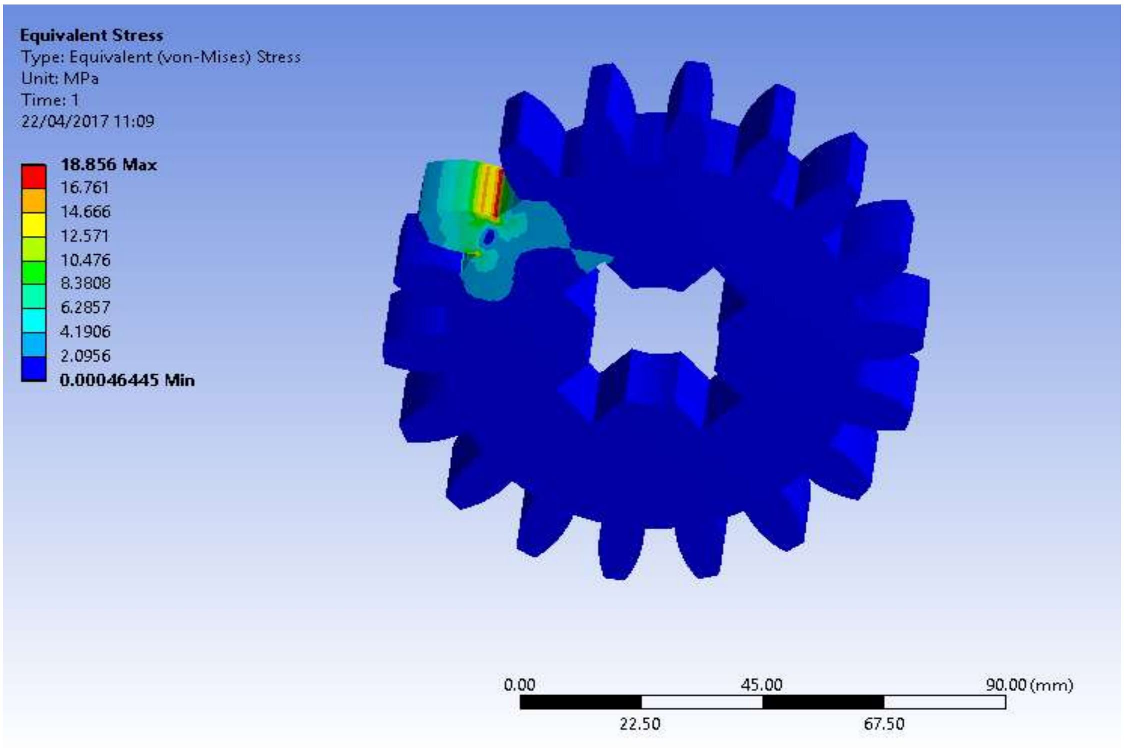

Figure 3 Equivalent stress of gear 1

- Total deflection

Maximum teeth deflection is 0.0023 mm is safe condition as shown in figure 4

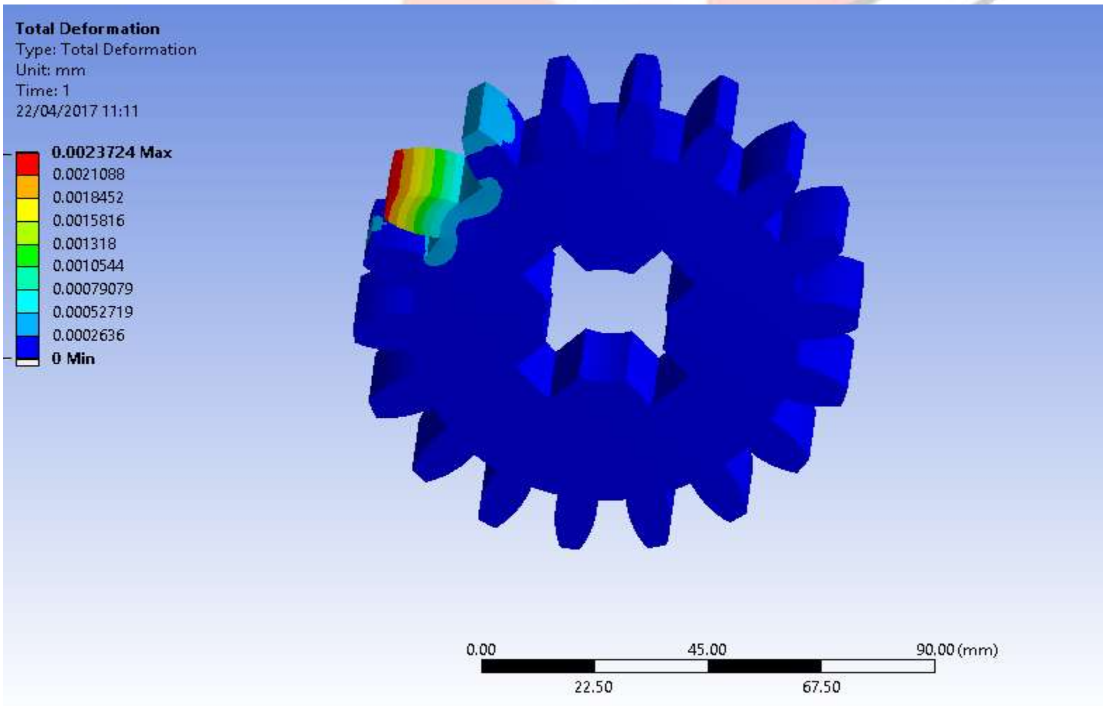

Figure 4 Total deflection of gear 1

1.9 FEA OF GEAR 3

- Boundary condition

Apply fixed support at surface A and force of 3722.2 on teeth as at surface B. also apply rotation velocity at centre of gear as shown in figure 5

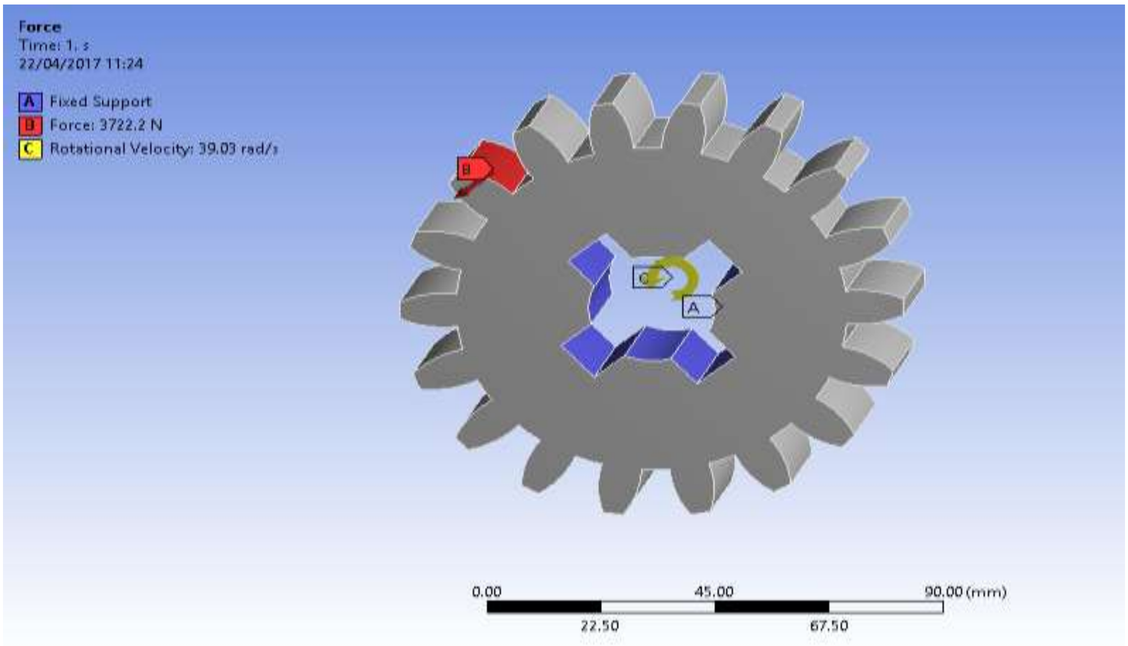

Figure 5 Boundary condition of gear 3

- Equivalent stress

Here Von-mises stress is only 47 Mpa which is safer than permissible limit of 105 Mpa as shown in figure 6

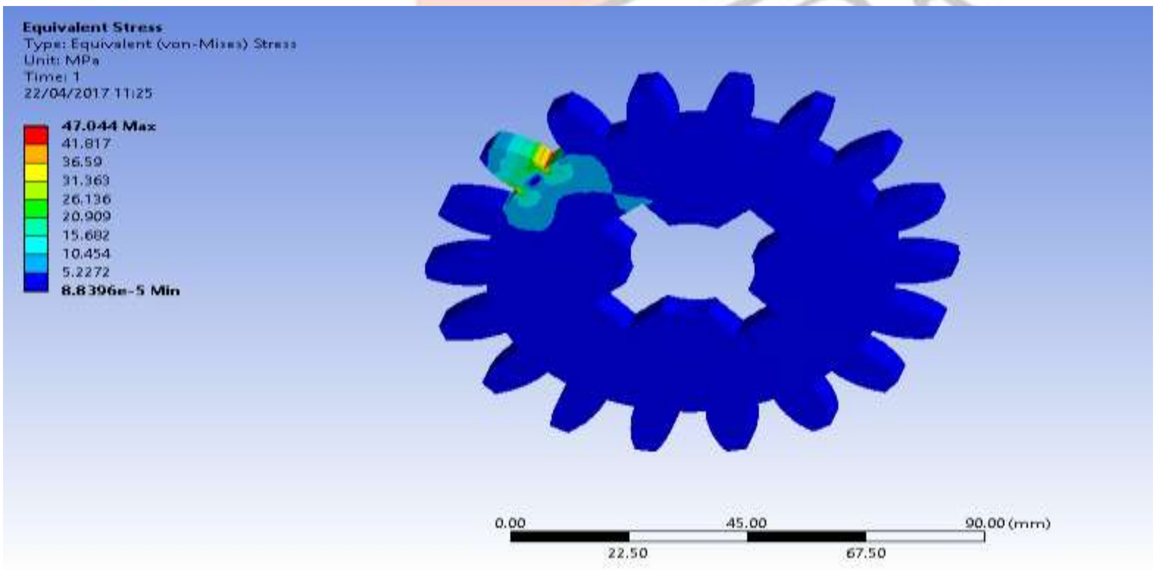

Figure 5.6 Equivalent stress of gear 3

➢ Total deflection

Maximum teeth deflection is 0.0059 mm is safe condition as shown in figure 5.7

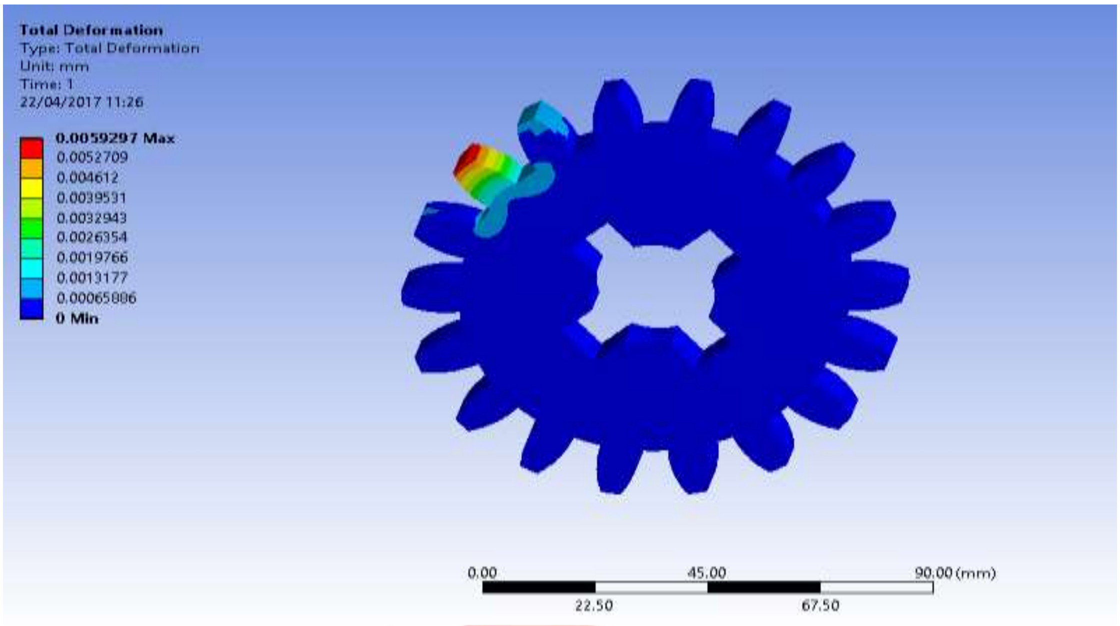

Figure 5.7 Total deflection of gear 3

1.10 FEA OF GEAR 5

➢ Boundary condition

Apply fixed support at surface A and force of 8188.9 on teeth as at surface B. also apply rotation velocity at centre of gear as shown in figure 8

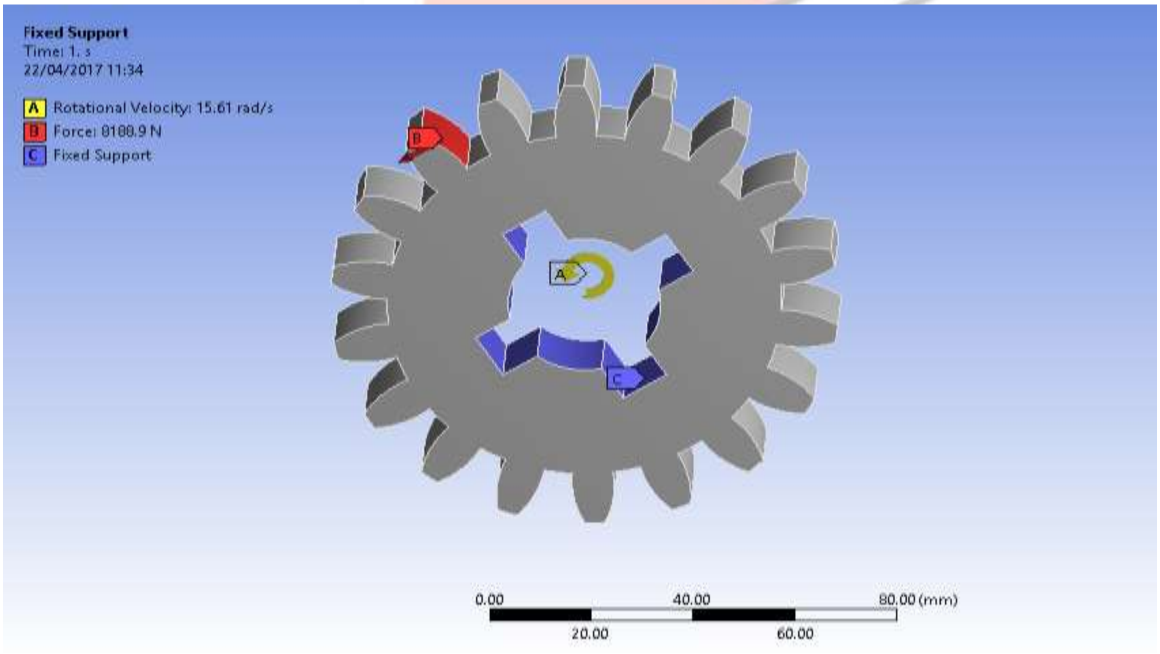

Figure 8 Boundary condition of gear 5

- Equivalent stress

Here Von-mises stress is only 45.5 Mpa which is safer than permissible limit of 105 Mpa as shown in figure 9

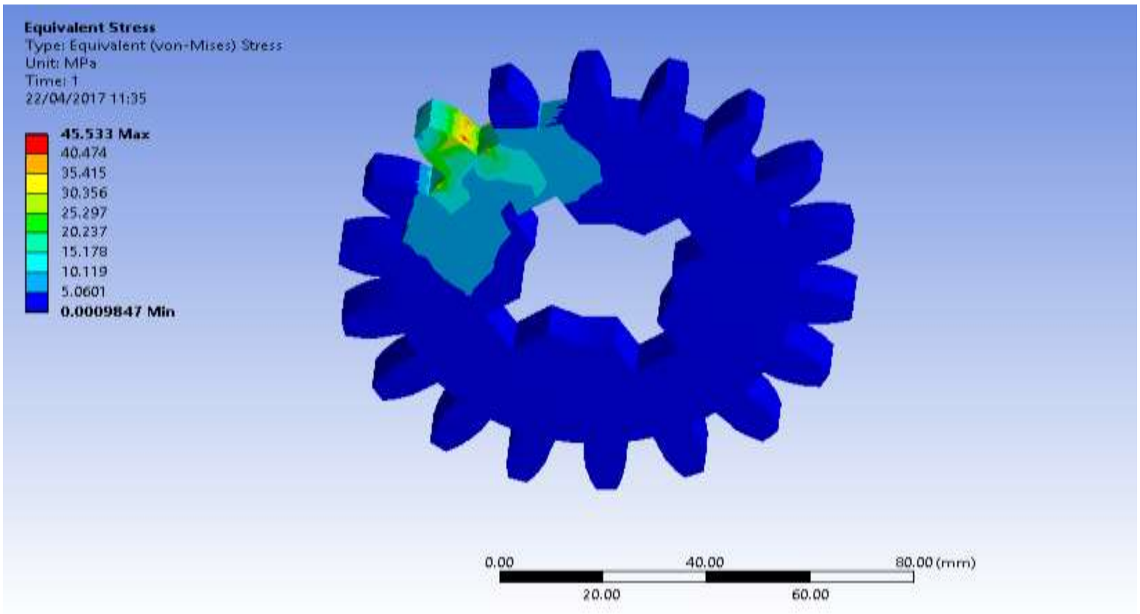

Figure 9 Equivalent stress of gear 5

- Total deflection

Maximum teeth deflection is 0.0088 mm is safe condition as shown in figure 10

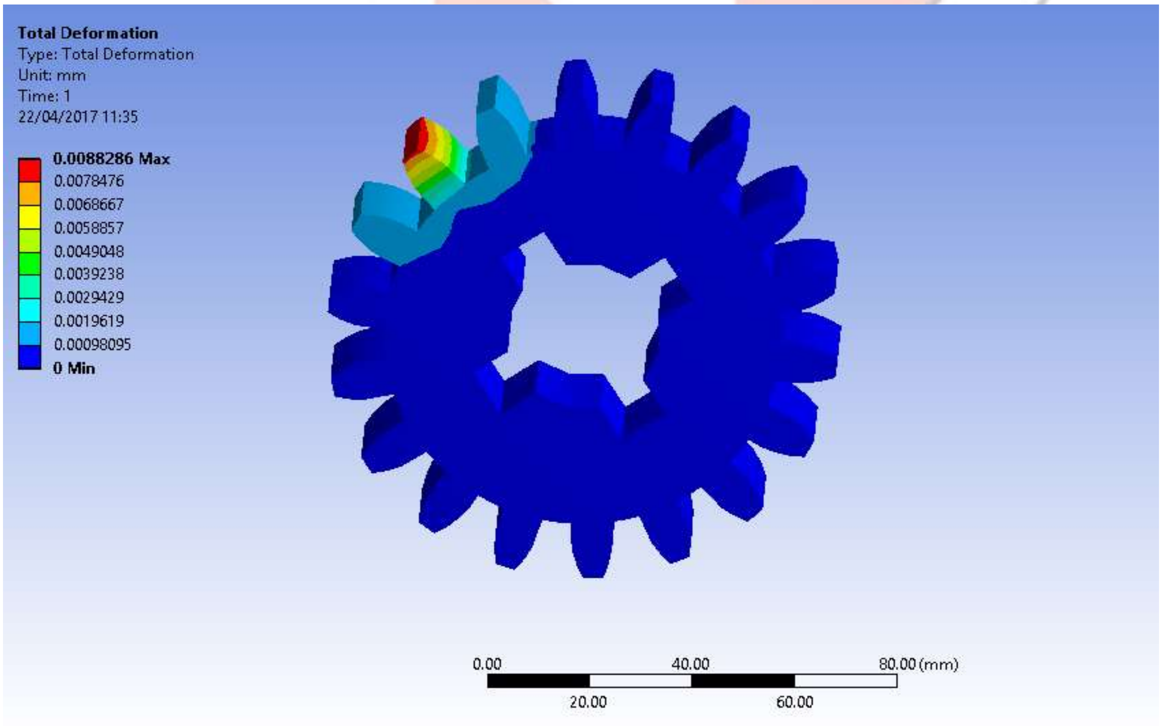

Figure 10 Total deflection of gear 5

1.11 Result and discussion for gears

Specification	Gear 1	Gear 2	Gear 3	Gear 4	Gear 5	Gear 6
Tangential Load in N	1488.89	1488.89	3722.22	3722.2 2	8188.89	8100
Velocity in radian/second	97.59	39.03	39.03	15.61	15.61	7.02
Von-Mises stress in Mpa	18.85		47.04		45.53	
Deformation	0.0023		0.0059		0.0088	

CONCLUSION.

- Thus we conclude that the Gearbox designed developments is satisfactory and meets the requirements specified at the outset of the project
- We would also like to state that this project can be further improved by further study research and Design.

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