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Investigation of Stress Analysis of Power Train Units for Light-Weight Car

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Abstract- Gears are one of the most critical components in mechanical power transmission systems. The bending and surface strength of the gear tooth are considered to be one of the main contributors for the failure of the gear in a gear set. The purpose of this research is to reduce the bending stress and contact stress happing often on gear by changing the different values of face width and three types of materials. This research investigates the characteristics of helical gear and bevel gear system mainly focused on bending and contact stresses using analytical and finite element analysis. Three-dimensional models of helical gear for different values of face width are generated by SolidWorks software and numerical solution is done by ANSYS 14.5 software. Lewis formula is used to estimate the bending stress and AGMA equation is used to determine the contact stresses between two mating gears. Helical gear and bevel gear used three types of material are analysed by ANSYS 14.5 software. Finally, the results obtained from ANSYS values are compared with theoretical values. In this analysis, the smallest face width is selected as the weight reduction and AISI 5160 OQT400 which has von-Mises stress 1114.3 MPa is the suitable material because it has lowest total deformation on gear.

Index Terms- Bending stress, Contact stress, Face width, Material, Lewis, AGMA

I. INTRODUCTION

Gearing is one of the most effective methods transmitting power and rotary motion from the source to its application with or without change of speed or direction. In automobile, highly reliable and lightweight gears are essential. Furthermore the best way to diminution of noise in engine requires the fabrication of silence gear system. Noise reduction in gear pairs is especially critical in the rapidly growing today's technology since the working environment is badly influenced by noise. The most successful way of gear noise reduction is attained by decreasing of vibration related with them. Helical gears are currently being used increasingly as a power transmitting gear owing to their relatively smooth and silent operation, large load carrying capacity and higher operating speed. Designing highly loaded helical gears for power transmission systems that are good in strength and low level in noise necessitate suitable analysis methods [12]. There are two basic types of helical gears parallel and crossed as shown in Figure 1. Helical gears are the modified forms of spur gears, in which all the teeth are cut at a constant angle, known as helix angle. Helical gears are also employed to transmit power between two shafts parallel to the axis [6]. Sarfraz Ali N.Quadri [2] have done his research work on "Contact Stress Analysis of Involute Spur Gear under Static Loading". In this paper, they have analysed a spur gear pair from lathe machine for contact stresses under static loading conditions using Hertz theory with the effect of various modules. The analytical method of calculating gear contact stresses are compared with calculated contact stresses using ANSYS. Finally, the higher the module, the lower the contact stress. S.K. Sureshkumar (2015) [3] investigated a research in which analysis of helical gear pairs. The author carried out an attempt to study the contact stresses among the helical gear pairs of different materials such as Steel, Cast Iron, and Aluminium under static conditions by using a 3D finite element method. J.Venkatesh [4] is analysed the helical gear based on structural analysis of high speed. In this paper, he modified the number of teeth to reduce the bending stress by using AGMA. In contact stress, the different value of helix angle is changed to minimize the contact stress. And then, two methods bending and contact stress results are compared with each other. Finally, the more the number of teeth, the more the bending stress. In contact stress, the more the helix angle, the less the stress. Puttapaka Nagaraju (2014) [6] presented a paper a suggest that, the bending stress of helical gear are analysed. In this study, bending stress were calculated with different face width by using Lewis equation and these results are compared with the results obtained in static analysis in ANSYS. The author concluded that the maximum bending stress decreases with increasing the face width of the gear. In 2014, at IJERA, Raghava Krishna Saneer.B [7] presented contact stress analysis of modified helical gear. The analytical study is based on Hertz's equation. This paper have analysed by varying the geometrical profile of the teeth and to find the change in contact stresses between gears. Pravin M.kinge (2012) [8] presented a paper to suggest that, three modifications in the design of gear were done and after that again stress analysis of the modified gear carried out. The three design modifications were done the edges of the gear teeth were tapered by an angle of 20, making groove in the gear wheel and making holes at the roots of the gear teeth. K. Rajesh has investigated the effect of pitch angle on bending stresses of bevel gears. In this paper, the author calculated the bending stresses for straight bevel gear tooth. The author analysed the bending stress by using three different material and five pitch angles. Finally, the analysis results obtained in ANSYS and theoretical results are compared. Form the obtained results, it was concluded that the factor of safety increases up to 35 degree pitch angle and then it decreases from 40 degree pitch angle. Cr-Ni steel was considered as the best material as compared PB102 based on factor of safety [1]. Nitin Kapoor did a research study in which design and stress strain analysis of composite differential gear box. In this paper, the author observed that Glass filled polyamide composite material is selected as best material for differential gear box and is found to suitable for different revolutions (2500rpm, 5000rpm, 750rpm) under static loading conditions. Comparisons of various stress and strain results using ANSYS-12 with Glass filled polyamide composite and metallic materials (Aluminum alloy, Alloy steel and Cast Iron) was also being performed and found to be lower for composite material. By observing these analysis results, Glass filled polyamide composite material was selected as a best material for differential gear box which in turn increases the overall mechanical efficiency of the system [5]. Gear tooth terminology is shown in Fig. 1.

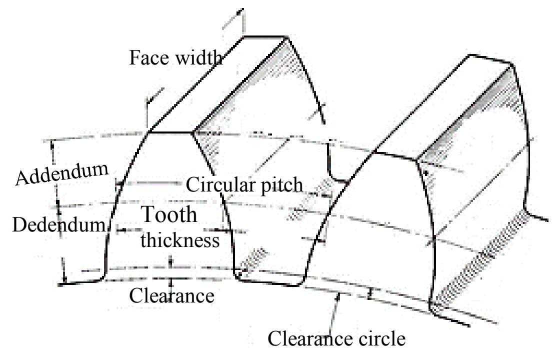

Fig.1 Gear tooth terminology [12]

II. Design Calculation of the Helical Gear pair and Bevel Gear

The design calculation of the gear pair has the following steps. The material for the gear pairs are taken as AISI 5160 OQT400. The parameters considered for design a helical gear is shown in Table I.

TABLE I
PARAMETERS CONSIDERED FOR DESIGN A HELICAL GEAR

Design Parameters	Value	Unit
Vehicle gross weight	700	kg
Power (P)	20	kW
R.P.M (N_p)	6500	rpm
Helix angle (ψ)	23	degree
Pressure angle (ϕ)	20	degree
Modulus of Elasticity (E)	207	GPa
Tensile Strength	2220	MPa
Yield Strength	1790	MPa
Brinell Hardness (BHN)	627	-
Number of teeth of pinion	11	mm
Number of teeth of gear	34	mm

When two mating gears are to be made of the same material, the smaller gear (pinion) will be the weaker. Both pinion and gear are same material.

1. Calculation of Torque (M_t)

The torque of gears in (Nm) can be calculated as

$$M_t = \frac{9550 \times P}{N_p} \tag{1}$$

2. Calculation of pitch line velocity (V)

The pitch line velocity can be calculated by

$$V = \frac{\pi \times D_p \times N_p}{60} \tag{2}$$

3. Unknown diameter case

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

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

$k = 6$,

$n_f = \frac{n}{\cos^3\psi}$ $\rightarrow y_p = 0.088$ (from the table)

4. Calculation of allowable stress,

Allowable stress can be calculated by

$$S_{all} = S_0 \times \left[\frac{5.6}{5.6 + \sqrt{V}}\right] \tag{4}$$

5. Calculation of number of teeth,

The number of teeth can be calculated as

$$n = \frac{D}{m} \tag{5}$$

6. Strength Check,

Compare S_{all} and S_{ind} (6)

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

If not so, keeping on calculating by increasing the module until it is satisfied need to be done

7. Calculation of the face width of helical gear (b)

The face width of helical gear in (mm) can be calculated as

$$b_{min} = k_{red} \times \pi \times m \tag{7}$$

$$b_{max} = k \times \pi \times m \tag{8}$$

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

8. Dynamic Check,

a. Calculation of transmitted load (F_t)

The transmitted load in (N) can be calculated as

$$F_t = \frac{2M_t}{D_p} \tag{9}$$

b. Calculation of dynamic load (F_d)

The dynamic load can be determined as

$$F_d = F_t + \frac{21V(bC\cos^2\psi + F_t)\cos\psi}{21V + \sqrt{(bC\cos^2\psi + F_t)}} \tag{10}$$

c. Calculation of limiting endurance load (F_0)

The limiting endurance load can be determined as

$$F_0 = S_0 b y_p \pi m \cos(\psi) \tag{11}$$

d. Calculation of limiting wear load (F_w)

The limiting wear load can be determined as

$$F_w = \frac{D_p \times b \times K \times Q}{\cos^2\psi} \tag{12}$$

where $S_{es} = (2.75BHN - 70)$ MN/m^2

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

$$Q = \frac{2 \times D_g}{D_g + D_p}$$

$\tan\phi_n = \tan\phi\cos\psi$

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 RESULT DATA FOR HELICAL GEAR

	Symbol	Pinion	Gear	Unit
No. of teeth	n	11	34	-
Pitch circle diameter	D	28	85	mm
Outside diameter	D_0	32	89	mm
Root diameter	D_R	21	78	mm
Face width	b	21	21	mm
module	m	2.5	2.5	mm
Speed	N	6500	5000	rpm

The design result data for helical gear is shown in Table II.
Bending stresses and contact stresses are analysed by increasing face width which are obtained from the result data.

TABLE III
PARAMETERS CONSIDERED FOR DESIGN OF BEVEL GEAR

Design Parameter	Value	Unit
Vehicle gross weight	700	kg
Power (P)	20	kW
R.P.M (N_p)	6500	rpm
Modulus of Elasticity (E)	207	GPa
Tensile Strength	2220	MPa
Yield Strength	1790	MPa
Brinell Hardness (BHN)	627	-
Number of teeth of pinion	10	mm
Number of teeth of gear	16	mm
Transmission ratio	1.6	-

The parameters considered for design a bevel gear is illustrated in Table III.

1. Pitch cone angle of pinion (δ_1)

Pitch cone angle of pinion can be calculated as

$$\tan\delta_1 = \frac{n_p}{n_g} \tag{13}$$

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

$$S_{ind} = \frac{2M_t}{m^3 b y_p \pi n_p} \times \left(\frac{L_c}{L_c - b}\right) \tag{14}$$

$$n_{fp} = \frac{n_p}{n_g}\sqrt{n_p^2 + n_g^2} \rightarrow y_p = 0.079 \text{ (from the table)}$$

3. Calculation of cone length (L_c)

Cone length can be calculated by

$$L_c = \frac{n_p m}{2}\sqrt{1 + V.R^2} \tag{15}$$

4. Calculation of face width (b)

Face width can be calculated as

$$b = \frac{L_c}{3} \tag{16}$$

5. Strength Check,

Compare S_{all} and S_{ind} (17)

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

If not so, keeping on calculating by increasing the module until it is satisfied need to be done

6. Dynamic Check,

a. Calculation of dynamic load (F_d)

The dynamic load can be determined as

$$F_d = F_t + \frac{21V(bC + F_t)}{21V + \sqrt{(bC + F_t)}} \tag{18}$$

b. Calculation of limiting endurance load (F_0)

The limiting endurance load can be determined as

$$F_0 = S_0 b y_p \pi m\left(\frac{L_c - b}{L_c}\right) \tag{19}$$

c. Calculation of limiting wear load (F_w)

The limiting wear load can be determined as

$$F_w = \frac{0.75 \times D_p \times b \times K \times Q}{\cos^2\psi} \tag{20}$$

The required condition to satisfy the dynamic check is F_0 ,$F_w > F_d$.

Allowable stress and tangential force equation have been expressed in the design of helical gear. If not so, keeping on calculating by increasing the module until it is satisfied need to be done.

TABLE IV
DESIGN RESULT DATA FOR BEVEL GEAR

Description		Nomenclature	Dimensions	Units
Pitch cone angle	Pinion	δ_1	32.01°	degree
	Gear	δ_2	57.99°	degree
Tip circle diameter	Pinion	d_{a1}	46.7836	mm
	Gear	d_{a2}	68.2405	mm
Pitch circle diameter	Pinion	D_p	40	mm
	Gear	Dg	64	mm
Gearing Arrangement		Σ	90°	degree
Module		m	4	mm
Cone distance		L	37.7311	mm
Face width for pinion and gear		b	13	mm

The theory results for contact stress by using AGMA equation is shown in Table IV.

III. LEWIS AND AGMA EQUATION USED TO CALCULATE BENDING AND CONTACT STRESSES

A. Bending Stress Calculation by using Lewis Equation

The analysis of bending stress in gear tooth was done by Lewis equation. The tangential component (F_t) occurs at the tooth of the pinion.

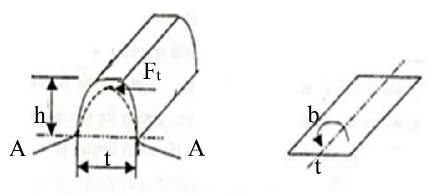

Fig.2 Force analysis of gear [12]

The force analysis of gear is shown in Fig.2.

Bending Stress Calculation by using Lewis Equation

$$\frac{M}{I} = \frac{\sigma_b}{y} \tag{21}$$

where $I = \frac{bt^3}{12}$

$$F_t = \frac{\sigma_b bt^2}{6h} = \sigma_b b\left(\frac{t^2}{6hP_c}\right) \times P_c = \sigma_b b y_p P_c$$

$$\sigma_b = \frac{F_t}{b y_p P_c} \tag{22}$$

B. Contact Stress Calculation by using AGMA Equation

$$\sigma_c = C_p \sqrt{\frac{F_t\left(\frac{\cos\psi}{0.95CR}\right)K_v K_0 (0.93K_m)}{bdI}} \tag{23}$$

where,

$$I = \frac{\sin\phi\cos\phi}{2} \times \frac{i}{i+1}$$

$$C_p = \sqrt{\frac{1}{\pi\left(\frac{1-\upsilon_1^2}{E_1} + \frac{1-\upsilon_2^2}{E_2}\right)}}$$

$$\text{CR}=\left(\frac{\sqrt{(r_1+a)^2-r_{b1}^2}+\sqrt{(r_2+a)^2-r_{b2}^2}+(r_1+r_2)\sin\phi}{\pi m\cos\phi}\right)$$

C. Calculation of von-Mises Stress

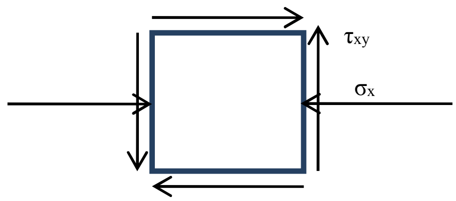

Fig.3 Stresses in the x-y plane

Shear stress τ_{xy},

$$\tau_{xy}=\frac{T\times R}{J} \tag{24}$$

$$=\frac{37.054}{(\frac{\pi}{32})\times 28^4}\times\frac{28}{2}=0.0085\ \text{MPa}$$

$$\sigma_{1,2}=\frac{1}{2}(\sigma_x+\sigma_y)\pm\frac{1}{2}\left[(\sigma_x-\sigma_y)^2+4\tau_{xy}^2\right]^{\frac{1}{2}} \tag{25}$$

By using von-Mises Criterion equation,

$$\bar{\sigma}=\frac{1}{\sqrt{2}}\left[(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2\right]^{\frac{1}{2}} \tag{26}$$

TABLE V
THEORETICAL RESULT OF BENDING STRESS

Face Width (mm)	von-Mises stress for Lewis (MPa)
21	182.35
22	174.06
23	166.49
24	159.55
25	153.17
26	147.28
27	141.83
28	136.76

The theoretical results of von-Mises stresses is shown in Table V.

TABLE VI
THEORETICAL RESULT FOR CONTACT STRESS

Face Width (mm)	von-Mises stress for AGMA (MPa)
21	1043.97
22	1019.97
23	997.55
24	976.55
25	956.82
26	938.24
27	920.70
28	904.11

The theoretical results of von-Mises stresses is shown in Table VI.

IV. Modelling of Gear

This gears are modelled by the results obtained from theory in SolidWorks software. The material for the gear is AISI 5160 OQT400. The gears are modelled with the combination of the all above mentioned parameter, other gear set are modelled in same way. The helical gear and bevel gear were generated by using SolidWorks software.

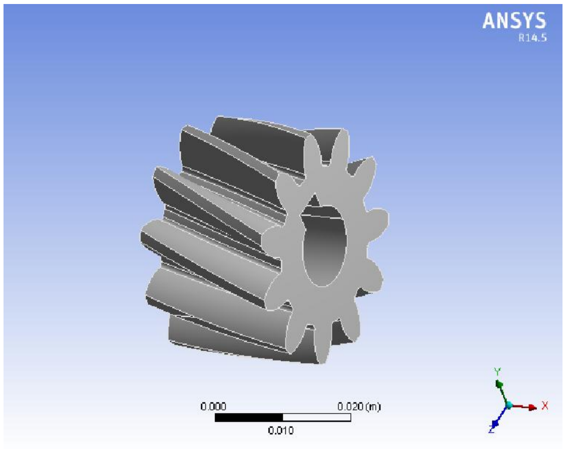

Fig.4 Solid model of helical gear

The solid model of helical gear is shown in Fig. 4.

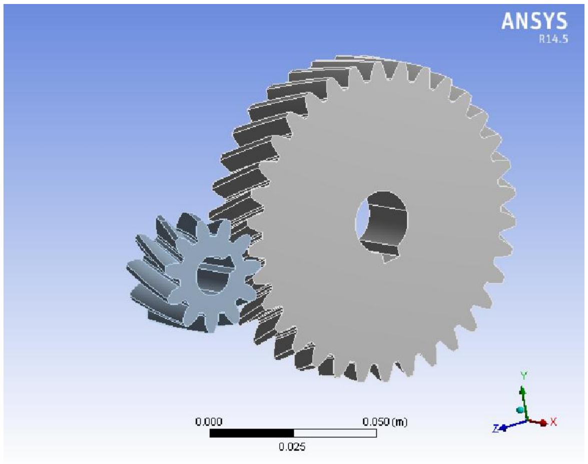

Fig.5 Solid model of helical gear pair

The solid model of helical gear pair is shown in Fig. 5.

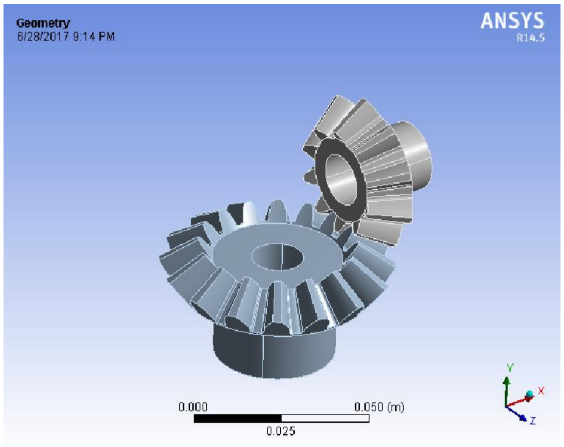

Fig.6 Solid model of bevel gear pair

The solid model of bevel gear pair is shown in Fig. 6.

A. FEM Analysis

The structural analysis of helical gear and bevel tooth model are carried out in SolidWorks and imported in ANSYS software. The fixed support is given at the inner rim of the pinion. The tangential load 2646.7142 N is applied at the tooth of the helical gear. Torque is applied to the helical pinion in clockwise direction.

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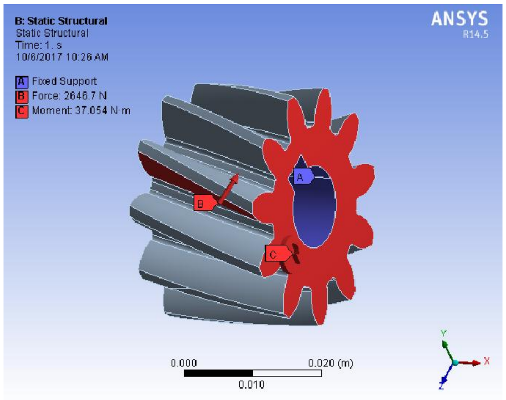

Fig.7 Boundary condition of helical gear

The boundary condition of helical gear is shown in Fig. 7.

The fixed support is given at the inner rim of the pinion and torque which the value is 37.054Nm is applied to the helical pinion in clockwise direction. The frictionless support is given at the inner rim of the pinion.

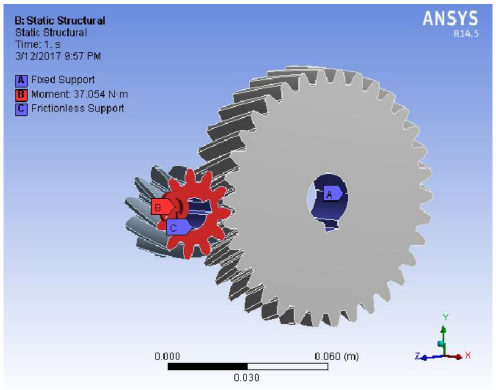

Fig.8 Boundary condition of helical gear pair

The boundary condition of helical gear pair is shown in Fig. 8.

The fixed support is given at the inner rim of the bevel gear and torque which the value is 37.054Nm is applied to the bevel pinion in counter clockwise direction. The frictionless support is given at the inner rim of the bevel pinion.

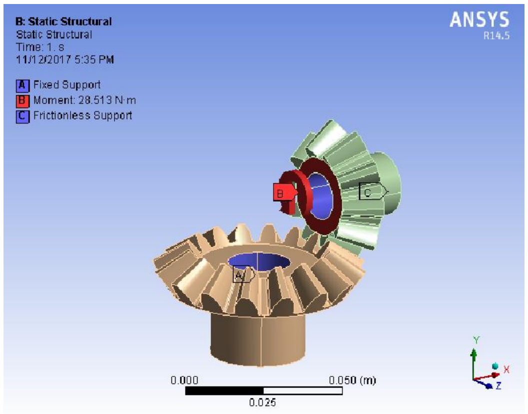

Fig.9 Boundary condition of bevel gear pair

The boundary condition of bevel gear pair is shown in Fig. 9.

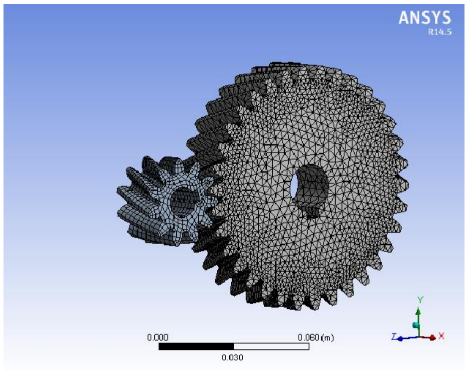

Fig.10 Meshed gear model of helical gear pair

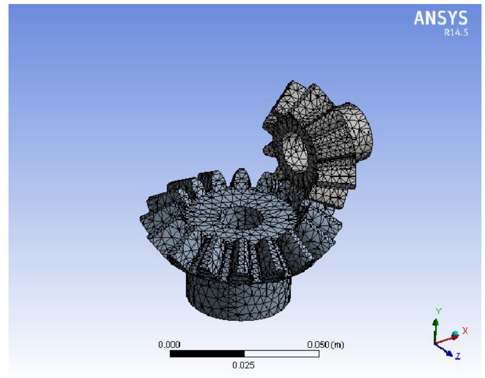

Fig. 11 Meshed gear model of bevel gear pair

The generated mesh of helical gear pair and bevel gear pair are shown in Fig. 10 and Fig. 11. The generated mesh is done by fine position to obtain the good quality of mesh.

V. FEM Equation used to Calculate Bending Stress and Contact stress

The model were analysed by using ANSYS software. The different value of face width are used to find out the bending stress and contact stress for helical gear.

A. Bending Stress for Different Values of Face Width

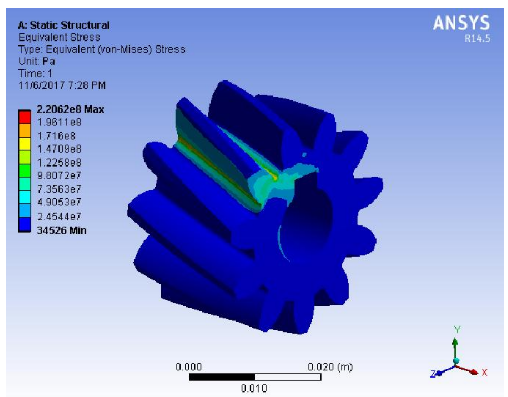

Fig.12 von-Mises stress of 21 mm face width modelled gear

The von-Mises stress of 21 mm face width modelled gear is shown in Fig. 12. The magnitude of maximum stress is 220.6 MPa which occur at the root of the teeth.

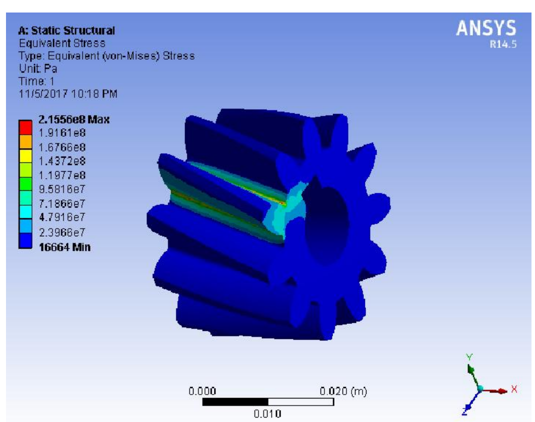

Fig. 13 von-Mises stress of 22 mm face width modelled gear

The von-Mises stress of 22 mm face width modelled gear is shown in Fig. 13. The magnitude of maximum stress is 215.56 MPa which occur at the root of the teeth.

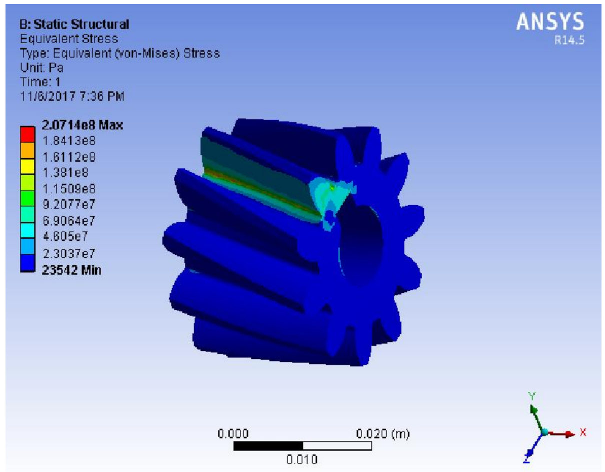

Fig. 14 von-Mises stress of 23 mm face width modelled gear

The von-Mises stress of 23 mm face width modelled gear is shown in Fig. 14. The magnitude of maximum stress is 207.14 MPa which occur at the root of the teeth.

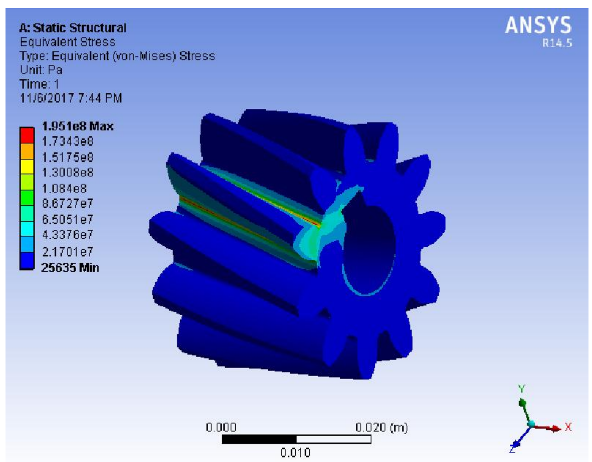

Fig. 15 von-Mises stress of 24 mm face width modelled gear

The von-Mises stress of 24 mm face width modelled gear is shown in Fig. 15. The magnitude of maximum stress is 195.1 MPa which occur at the root of the teeth.

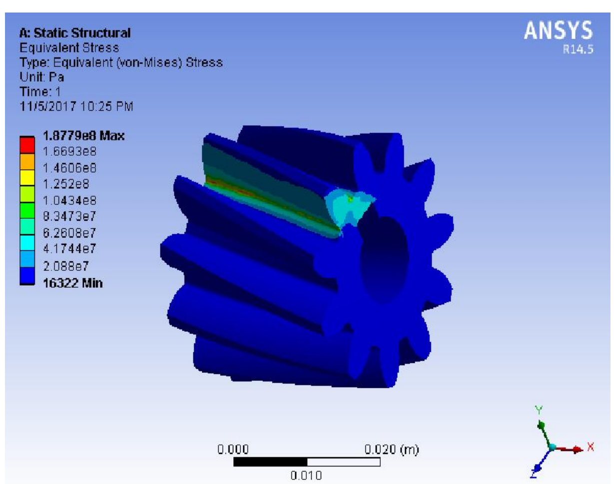

Fig. 16 von-Mises stress of 25 mm face width modelled gear

The von-Mises stress of 25 mm face width modelled gear is shown in Fig. 16. The magnitude of maximum stress is 187.79 MPa which occur at the root of the teeth.

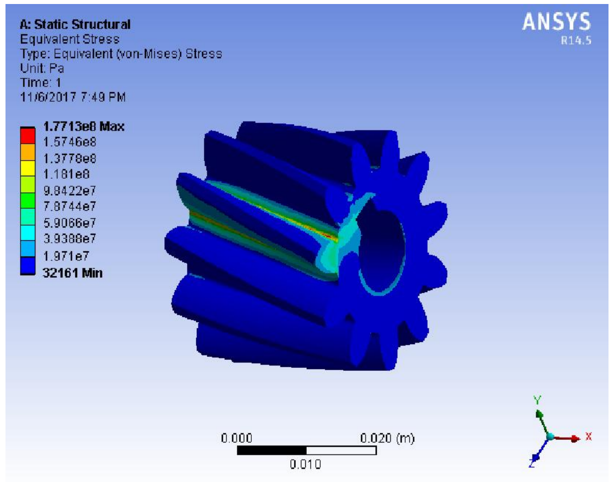

Fig. 17 von-Mises stress of 26 mm face width modelled gear

The von-Mises stress of 26 mm face width modelled gear is shown in Fig. 17. The magnitude of maximum stress is 177.13 MPa which occur at the root of the teeth.

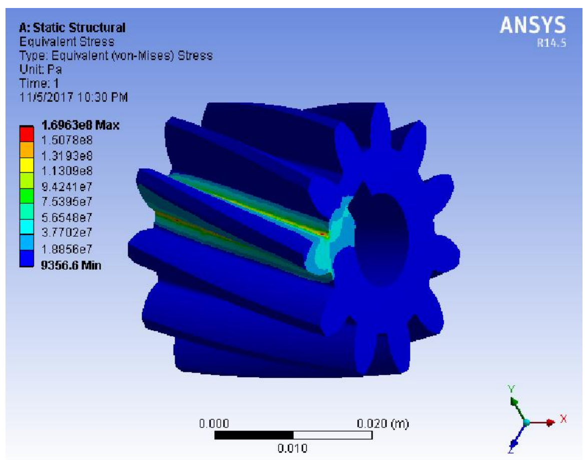

Fig. 18 von-Mises stress of 27 mm face width modelled gear

The von-Mises stress of 27 mm face width modelled gear is shown in Fig. 18. The magnitude of maximum stress is 169.6 MPa which occur at the root of the teeth.

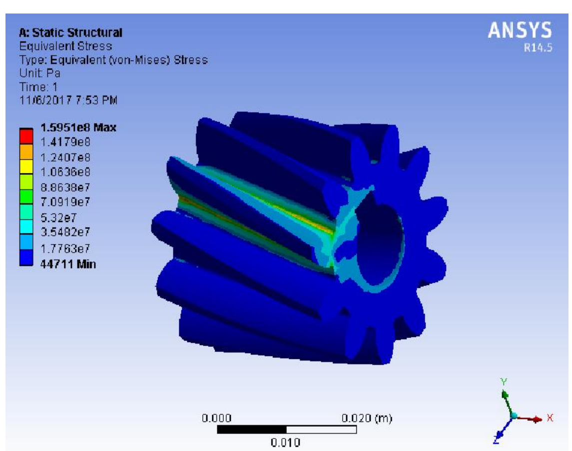

Fig. 19 von-Mises stress of 28 mm face width modelled gear

The von-Mises stress of 28 mm face width modelled gear is shown in Fig. 19. The magnitude of maximum stress is 159.51 MPa which occur at the root of the teeth.

B. Contact Stress for Different Values of Face Width

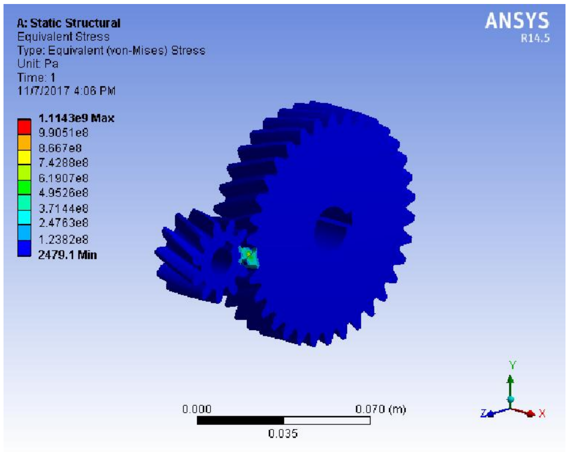

Fig. 20 von-Mises stress of 21 mm face width modelled gear

The von-Mises stress of 21 mm face width modelled gear is shown in Fig. 20. The magnitude of maximum stress is 1114.3 MPa which occur at the contact point.

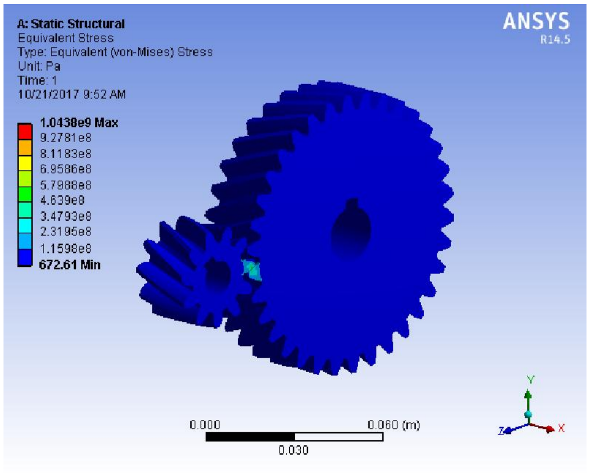

Fig. 21 von-Mises stress of 22 mm face width modelled gear

The von-Mises stress of 22 mm face width modelled gear is shown in Fig. 21. The magnitude of maximum stress is 1043.8 MPa which occur at the contact point.

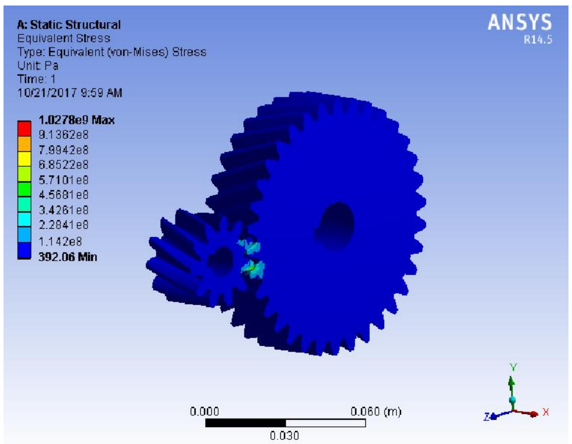

Fig. 22 von-Mises stress of 23 mm face width modelled gear

The von-Mises stress of 23 mm face width modelled gear is shown in Fig. 22. The magnitude of maximum stress is 1027.8 MPa which occur at the contact point.

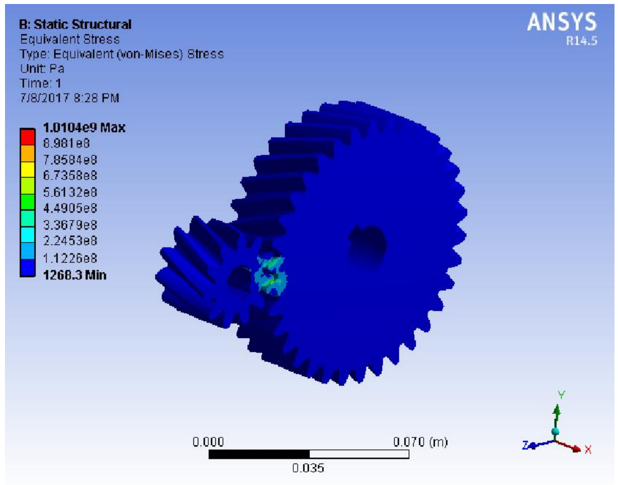

Fig. 23 von-Mises Stress of 24 mm face width modelled gear

The von-Mises stress of 24 mm face width modelled gear is shown in Figure 23. The magnitude of maximum stress is 1010.4 MPa which occur at the contact point.

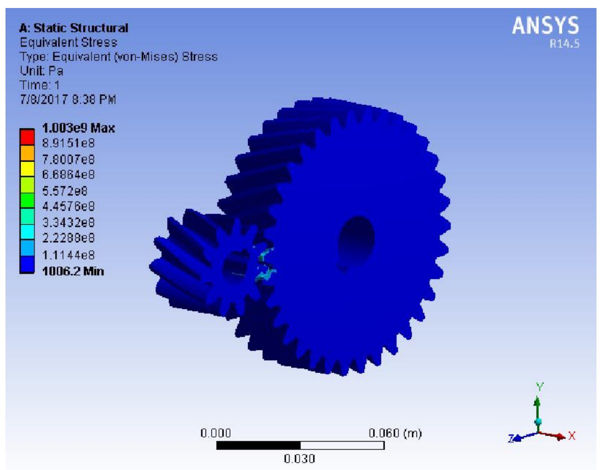

Fig. 24 von-Mises stress of 25 mm face width modelled gear

The von-Mises stress of 25 mm face width modelled gear is shown in Fig. 24. The magnitude of maximum stress is 1003.00 MPa which occur at the contact point.

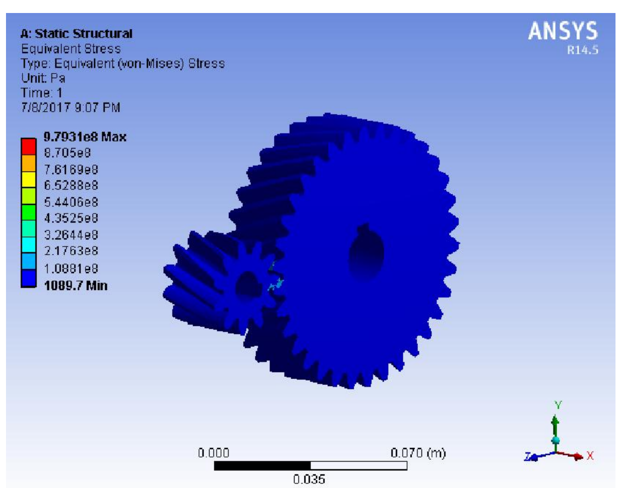

Fig. 25 von-Mises stress of 26 mm face width modelled gear

The von-Mises stress of 26 mm face width modelled gear is shown in Fig. 25. The magnitude of maximum stress is 979.31 MPa which occur at the contact point.

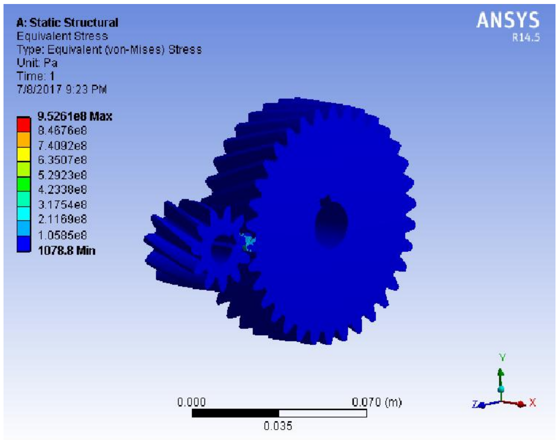

Fig. 26 von-Mises stress of 27 mm face width modelled gear

The von-Mises stress of 27 mm face width modelled gear is shown in Fig. 26. The magnitude of maximum stress is 952.61 MPa which occur at the contact point.

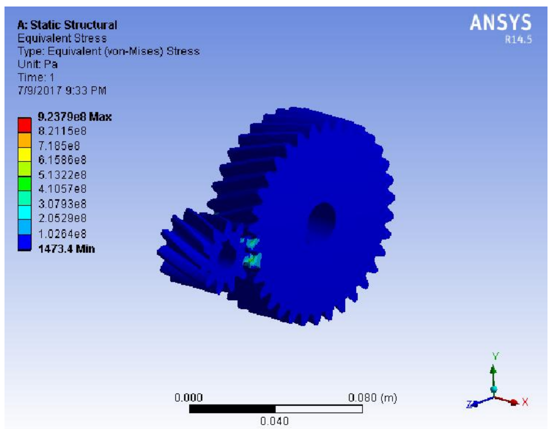

Fig. 27 von-Mises stress of 28 mm face width modelled gear

The von-Mises stress of 28 mm face width modelled gear is shown in Fig. 27. The magnitude of maximum stress is 923.79 MPa which occur at the root of the teeth which contact between pinion and gear.

C. Contact Stress for Different Three Types of Materials

The following figures show von-Mises stress for different properties of materials which obtained from analysis (FEA) result. The different properties of materials are AISI 5160 OQT 400, AISI 4150 OQT 400 and AISI 1030.

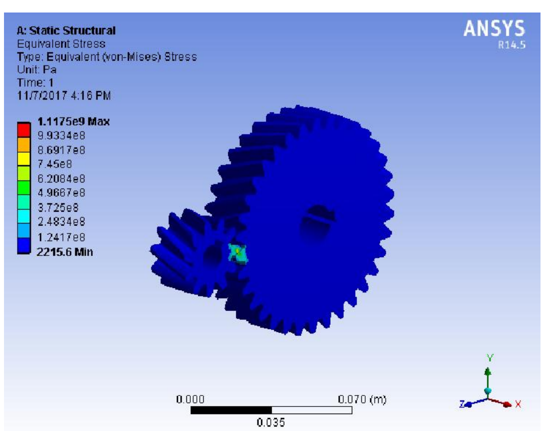

Fig. 28 von-Mises stress of AISI 4150 OQT 400 modelled gear

The von-Mises stress of AISI 4150 OQT 400 modelled gear is shown in Fig. 28. The magnitude of maximum stress is 1117.5 MPa which occur at the root of the teeth which contact between pinion and gear.

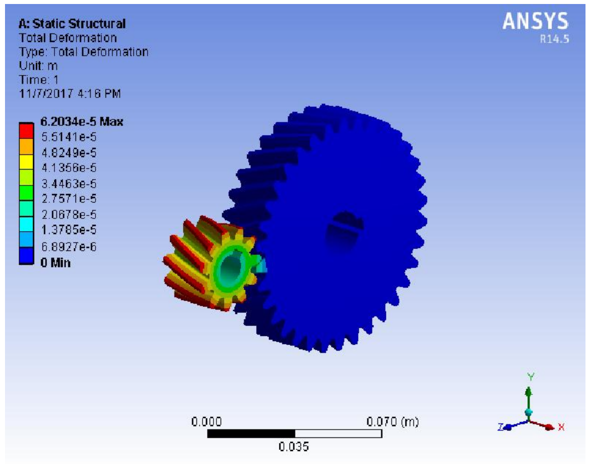

Fig. 29 Total deformation of AISI 4150 OQT 400 modelled gear

Total deformation of AISI 4150 OQT 400 modelled gear is shown in Fig. 29.

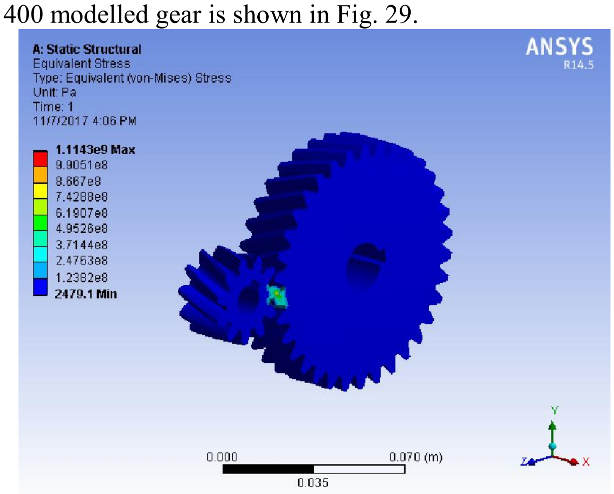

Fig. 30 von-Mises stress of AISI 1030 modelled gear

The von-Mises stress of AISI 1030 modelled gear is shown in Fig. 30. The magnitude of maximum stress is 1114.3 MPa which occur at the root of the teeth which contact between pinion and gear.

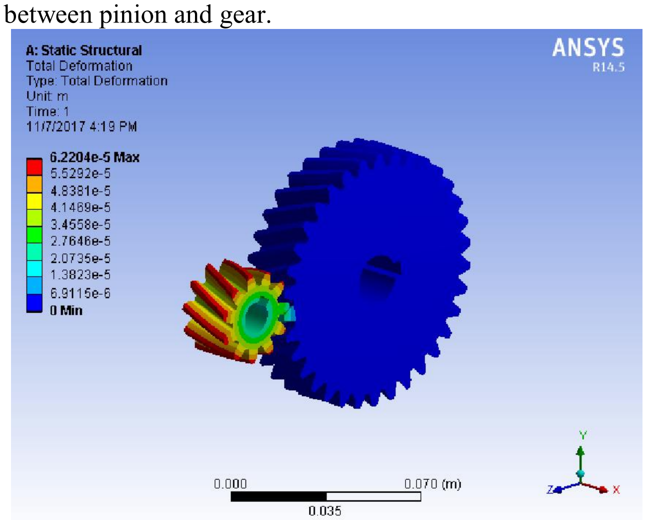

Fig. 31 Total deformation of AISI 1030 modelled gear

Total deformation of AISI 1030 modelled gear is illustrated in Fig.31.

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D. Contact Stress for Different Three Types of Material for Bevel Gear

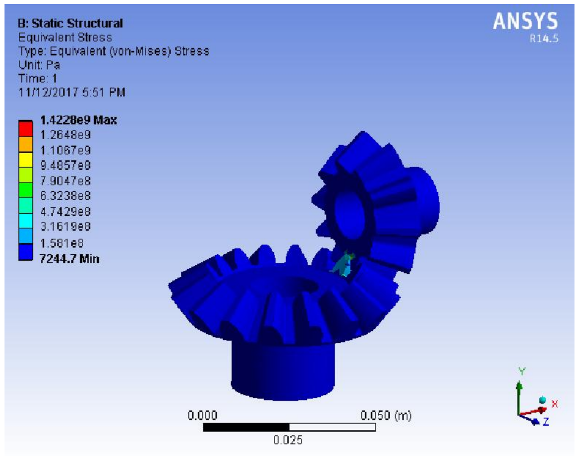

Fig. 32 von-Mises stress of AISI 5160 OQT 400 modelled gear

The von-Mises stress of AISI 5160 OQT 400 modelled gear is shown in Fig. 32. The magnitude of maximum stress is 1422.8 MPa which occur at the root of the teeth which contact between bevel pinion and bevel gear.

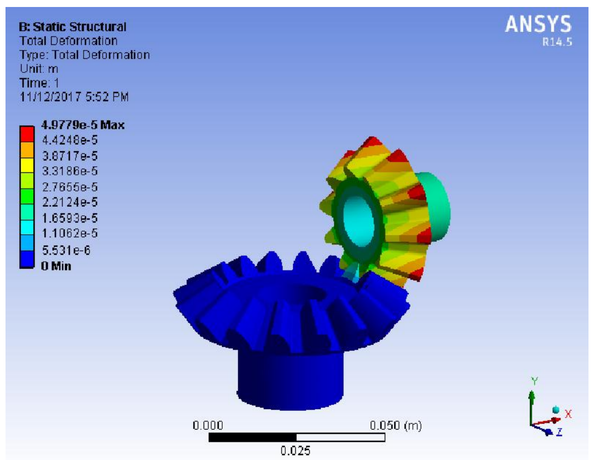

Fig. 33 Total deformation of AISI 5160 OQT 400 modelled gear

Total deformation of AISI 5160 OQT 400 modelled gear is shown in Fig. 33.

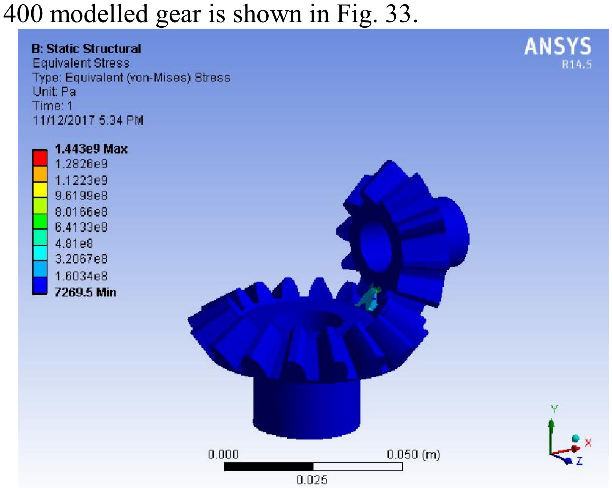

Fig. 34 von-Mises stress of AISI 4150 OQT 400 modelled gear

The von-Mises stress of AISI 4150 OQT 400 modelled gear is shown in Fig. 34. The magnitude of maximum stress is 1443 MPa which occur at the root of the teeth which contact between pinion and gear.

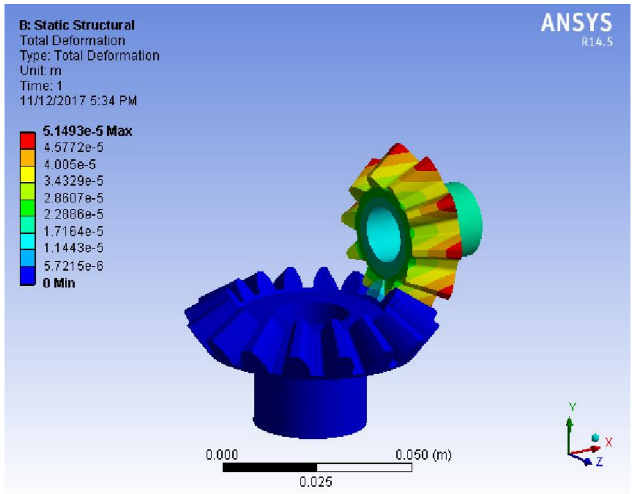

Fig. 35 Total deformation of AISI 4150 OQT 400 modelled gear

Total deformation of AISI 4150 OQT 400 modelled gear is expressed in Fig. 35.

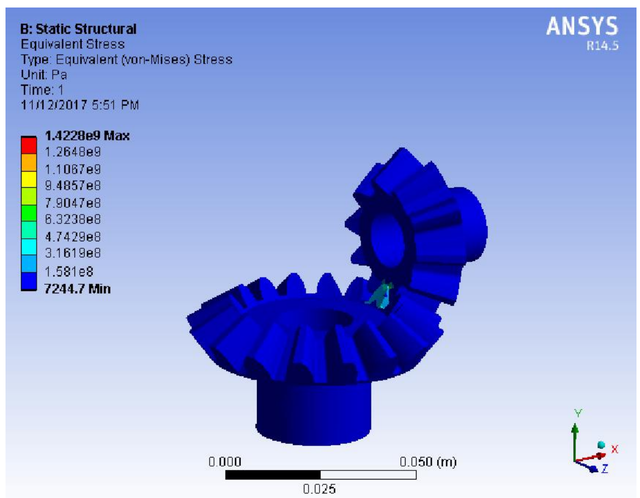

Fig. 36 von-Mises stress of AISI 1030 modelled gear

The von-Mises stress of AISI 1030 modelled gear is shown in Fig. 36. The magnitude of maximum stress is 1402.0 MPa which occur at the root of the teeth which contact between pinion and gear.

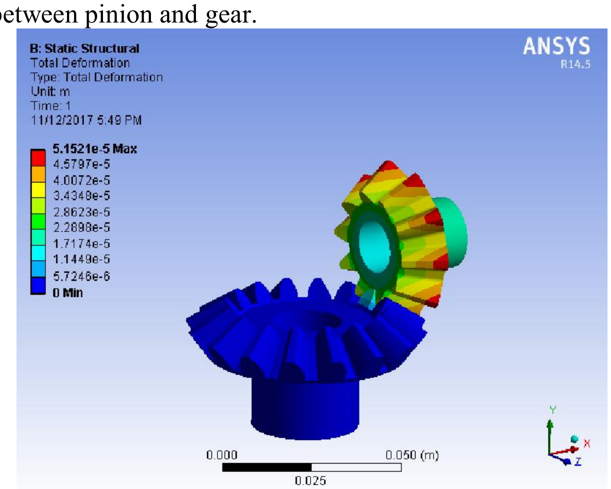

Fig. 37 Total deformation of AISI 1030 modelled gear

Total deformation of AISI 1030 modelled gear is illustrated in Fig.37.

VI. Results and Discussions

TABLE VII
COMPARISON OF LEWIS THEORY AND FEM FOR HELICAL GEAR

Face Width (mm)	Theoretical Stress (MPa)	Simulation Results (MPa)	Yield Strength (MPa)
21	182.35	220.62	1790
22	174.06	215.56	
23	166.49	207.14	
24	159.55	195.1	
25	153.17	187.79	
26	147.28	177.13	
27	141.83	169.63	
28	136.76	158.93	

The result data for theory and FEM is shown Table VII.

TABLE VIII
COMPARISON OF AGMA THEORY AND FEM FOR HELICAL GEAR PAIRS

Face Width (mm)	Theoretical Stress (MPa)	Simulation Results (MPa)	Yield Strength (MPa)
21	1043.97	1114.30	1790
22	1019.97	1043.80	
23	997.55	1027.80	
24	976.55	1010.40	
25	956.82	1003.00	
26	938.24	979.30	
27	920.70	952.60	
28	904.11	923.70	

The result data for theory and FEM is shown Table VIII.
The above table show that the von-Mises stresses are inversely proportional to the face width of the helical gear, as the face width increases the bending stresses and contact stresses decreases. The effect of face width on bending stress and contact stress is studied by varying the face width. It can be observed that the variation in the magnitude of stress to face width. When comparing, Lewis and AGMA values are a little lower than the ANSYS values.

TABLE IX
TOTAL DEFORMATION AND VON-MISES STRESS OF BEVEL GEAR PAIRS FOR THREE TYPES OF MATERIAL

Materials	AISI 5160 OQT400	AISI 4150 OQT 400	AISI 1030
Total deformation (displacement) (m)	0.0497	0.0514	0.0515
von-Mises stress (MPa)	1422.8	1443	1422.8

The result data for different properties of material is shown in Table IX shows.
From the obtained results, three types of material have nearly the same total deformation. The suitable material is AISI 5160 OQT 400 because it has lowest von-Mises stress on gears.

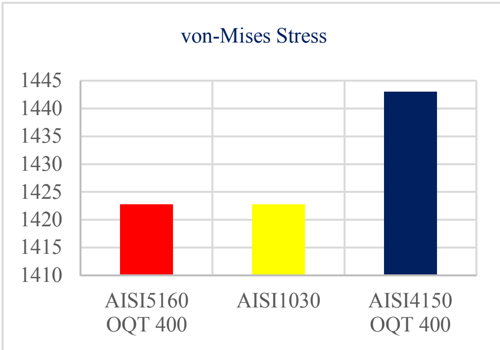

Fig. 38 Simulation results of von-Mises stress

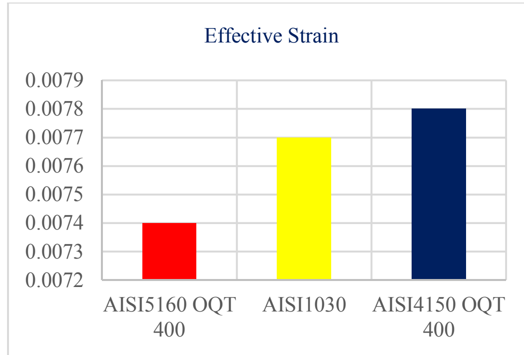

Fig. 39 Simulation results of effective strain

Simulation results of von-Mises stress and effective strain for different three types of material are shown in Fig. 38 and Fig. 39. According to the results, the values of AISI 4150 OQT 400 is the largest in the figure of von-Mises stress. In the figure of effective strain, AISI 5160 OQT 400 has the smallest value. That is why AISI 5160 OQT 400 is preferred.

TABLE X
THEORETICAL AND SIMULATION RESULTS OF VON-MISES STRESS AND EFFECTIVE STRAIN FOR BEVEL GEAR (AISI 5160 OQT 400)

	Theoretical Result	Simulation Result	Percentage Error
von-Mises stress (MPa)	1529	1422.8	6.9
Effective Strain	0.0065	0.0074	12.2

Theoretical and simulation results of von-Mises stress and effective strain for straight bevel gear (AISI 5160 OQT400) are shown in Table X.

TABLE XI
WEIGHT OF HELICAL GEAR PAIRS

Face Width (mm)	Weight of helical gear pairs (N)
21	10.2152
22	10.7615
23	11.2392
24	11.7279
25	12.2165
26	12.7052
27	13.1939
28	13.6825

The weight of helical gear pairs is shown in Table XI.
Weight reduction is very important criterion in automobile. Therefore, face width 21 mm is selected for reducing the weight of car.

VII. CONCLUSIONS

Face width is an important geometrical parameters in determining the state of stresses during the design of gears. The effect of face width on von-Mises stress is studied by varying the face width which are from 21mm to 28mm calculated from theory. Both Lewis and AGMA theory of stresses as well as FEM method of stresses give almost identical results with a mere percentage difference. The results are indicated that as the face width increases the bending stress and contact stress decreases. By observing the analysis results, the stress values obtained are less than the yield strength (1790 MPa). So, it can conclude that the design is safe under working condition. And then, weight reduction is a very important criterion in the helical gears. So, face width 21mm is preferred. And then, Comparative study on static structure analysis has been carried out the helical and bevel gear made of different materials. From the obtained results, it is clear that the suitable material is AISI5160 OQT 400 because three types of gear material have nearly the same von-Mises stress on gear but AISI 5160 OQT 400 has lowest total deformation on gears. Therefore, AISI 5160 OQT 400 is suitable for helical gear and bevel gear.

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