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Contact Stress Analysis of Helical Gear System

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Abstract -- **This research investigates the characteristics of an in volute helical gear system mainly focused on contact stresses induced between two gears using analytical and finite element analysis. To estimate the contact stress, three-dimensional solid models for different helix angles (20, 23˚, 26˚, 29˚, 32˚ and 35˚) are generated by SolidWorks software 16.0 that is a powerful and modern solid modelling software and the numerical solution is done by ANSYS. The results obtained from ANSYS are presented and compared with theoretical values. The analytical investigation is based on AGMA stress formula. The main objective of this research has to analyse the stresses induced between two gears by changing six different helix angles (20˚, 23˚, 26˚, 29˚ 32˚ and 35˚). Contact stresses analysis is also carried out for three different materials (AISI 5160 OQT 400, Stainless and Structural Steel) and the results are compared.**

Indexed Terms- - **different materials, helical gear, involute, synchromesh, von-Mises**

I. INTRODUCTION

Gears are used to transmit power and motion from one shaft to another. There are also a wide variety of gear types to choose from [2]. 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. Helical gears have a smoother operation than the spur gears because of a large helix angle that increases the length of the contact lines. Designing highly loaded helical gears for power transmission systems that are good in strength and low level in noise necessitate suitable analysis methods that can easily be put into practice and also give useful information on contact and bending stresses**.** Gears are used to change the speed, magnitude, and direction of a power source. Gears are being most widely used as the mechanical elements of power transmission. When two gears with unequal numbers of teeth are combined, a productive output is realized with both the angular speeds and the torques of the two gears differing through a simple relationship [9].

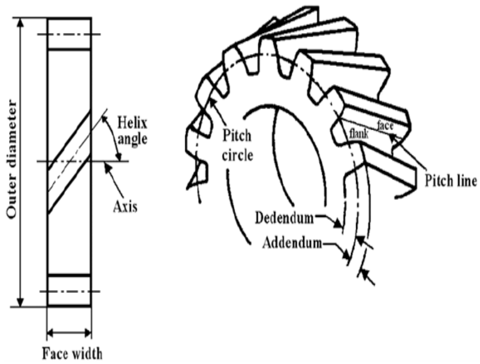

Figure 1: Nomenclature of Helical Gear [4]

1) Module: Module of a gear is defined as ratio of diameter to number of teeth. m=d/N.
2) Face Width: The width along the contact surface between the gears is called the face width.
3) Tooth Thickness: The thickness of the tooth along the pitch circle is called the tooth thickness.
4) Addendum: The radial distance between the pitch circle and the top land of the gear is called the addendum.
5) Dedendum: The radial distance between the pitch circle and the bottom land of the gear is called the dedendum.
6) Pressure Angle: The angle between the line joining the center of the two gears and the common tangent to the base circles.
7) Helix Angle: It is a constant angle made by the helices with the axis of rotation.
8) Axial Pitch: It is the distance, parallel to the axis, between similar faces of adjacent teeth. It is the same as circular pitch and is therefore

denoted by pc. The axial pitch may also be defined as the circular pitch in the plane of rotation or the diametric plane.

9) Normal Pitch: It is the distance between similar faces of adjacent teeth along a helix on the pitch cylinders normal to the teeth. It is denoted by p_n. The normal pitch may also be defined as the circular pitch in the normal plane which is a plane perpendicular to the teeth. Mathematically, normal pitch, p_n = pc cos.

II. THEORETICAL CALCULATIONS

Table I shows the parameters considered for design a helical gear. Pinion and gear are same material and so pinion is weaker. So based design on pinion.

TABLE I PARAMETERS CONSIDERED FOR DESIGN A HELICAL GEAR

Design Parameter	Specification
Power (P)	20 kW
Pinion Speed (Np)	6500 rpm
Helix angle (ψ)	23˚
Pressure angle (ϕ)	20˚
Modulus of Elasticity (E)	207 GPa
Ultimate Strength (S_u)	2220 MPa
Yield Strength (S_y)	1790 MPa
Brinell Hardness (BHN)	627
Number of teeth of pinion (n_p)	11
Number of teeth of gear (n_g)	34
Poisson ratio (v)	0.3

1. Unknown diameter case
The actual induced stress can be calculated by using Lewis equation.

$$S_{ind} = \frac{2M_t}{m^3 k\pi^2 y_p n_p \cos\psi} \qquad (1)$$

2. Calculation of Torque (M_t)

$$M_t = \frac{9550 \times P}{N_p} \qquad (2)$$

Where, N_p = number of teeth for pinion

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

$$V = \frac{\pi \times D_p \times N_p}{60} \qquad (3)$$

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

Where, D_p = the diameter of pinion (mm)
k = 6 (for helical)

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

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

5. Calculation of endurance stress, S_o

$$S_0 = \frac{S_u}{3} \qquad (5)$$

Where, S_u = Ultimate strength (MPa)

6. Calculation of number of teeth,

$$n = \frac{D}{m} \qquad (6)$$

7. Strength Check,

$$\text{Compare } S_{all} \text{ and } S_{ind} \qquad (7)$$

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

$$b_{min} = k_{red} \times \pi \times m \qquad (8)$$

$$b_{max} = k \times \pi \times m \qquad (9)$$

$$k_{red} = k_{max} \times \frac{S_{ind}}{S_{all}} \qquad (10)$$

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

$$F_t = \frac{2M_t}{D_p} \qquad (11)$$

10. Calculation of dynamic load, F_d

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

Where , ψ = helix angle (degree)

C = Dynamic factor (N/m)

11. Calculation of limiting endurance load, F_0

$$F_0 = S_0 b y_p \pi m \cos(\psi) \qquad (13)$$

where, m= module (mm)

12. Calculation of limiting wear load, F_w

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

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

where, $S_{es} = (2.75BHN - 70)$

S_{es} = Surface endurance limit of a gear Pair (MPa)
BHN = Average brinell hardness number of gears

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

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

where, E = Young's modulus (GPa)
K = Load stress factor for fatigue (N/m^2)
Q = Ratio factor
D_g = Pitch diameter of gear (mm)

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 [12]. The design results for helical gear is shown in Table II.

TABLE II DESIGN RESULTS FOR HELICAL GEAR PAIR

	Symbol	Pinion	Gear	Unit
No. of teeth	n	11	34	-
Pitch circle diameter	D	28	86	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

III. CONTACT STRESS ANALYSIS OF HELICAL GEAR BY USING AGMA EQUATION

One of the main gear tooth failure is pitting which is a surface fatigue failure due to repetition of high contact stresses occurring in the gear tooth surface while a pair of teeth is transmitting power [14Bab]. The contact stress equation is given as

$$\sigma_C = C_P\sqrt{\frac{F_t}{bdI}\left(\frac{\cos\psi}{0.95CR}\right)K_v K_0\left(0.93K_m\right)} \qquad (15)$$

The elastic coefficient factor equation is given as

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

The geometry factor I is given by

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

The speed ratio is given by

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

The contact ratio equation is given as

$$CR = \frac{\sqrt{(r_1+a)^2 - r_{b1}^2} + \sqrt{(r_2+a)^2 - r_{b2}^2} - (r_1 + r_2)\sin f}{\pi m \cos f} \qquad (18)$$

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. The principal stresses are determined by the following equation.

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

where,
σ_1 = first principal stress (MPa)
σ_2 =second principal stress (MPa)

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:

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 III THEORETICAL RESULT OF VON-MISES STRESSES FOR VARIOUS HELIX ANGLES

Helix Angles	von-Mises Stress (MPa)	Effective Strain
20°	1215.83	0.00521
23°	1203.35	0.00516
26°	1189.08	0.00509
29°	1172.98	0.00503
32°	1155.03	0.00495
35 °	1135.18	0.00486

The theoretical result of von-Mises stresses is shown in Table III.

IV. STRESS ANALYSIS OF HELICAL GEAR

Finite element analysis (FEA) is the numerical solution of the behavior mechanical components that are acquired by discretizing the mechanical components into a small finite number of building blocks (known as elements) and by analyzing those mechanical components for their acceptability and reliability [8].

A. Modelling of Helical Gear:

In this work, module, pressure angle, helix angle, shaft diameter, face width, numbers of teeth of both the gears are taken as input parameters. SolidWorks software uses these parameters, in combination with its features to generate the geometry of the helical gear and all essential information to create the model. SolidWorks has model the involute profile helical gear geometry perfectly. The assembly of gear is done by considering the left hand helical gear and right hand helical pinion. Then the file is saved as IGES format. In this research, AISI 5160 OQT 400 is used as the helical gear materials. The material properties of AISI 5160 OQT 400 is given in the Table IV.

TABLE IV 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	1790 MPa
Tensile Ultimate Strength	2220 MPa

The procedure to model the gear of 34 number of teeth with the combination of the all above mentioned parameters in the SolidWorks software, other set of gears are modelled in the similar way [7].

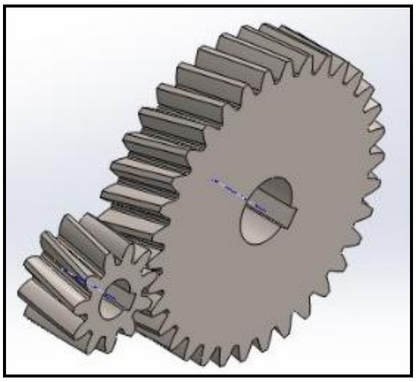

Figure 2. 3D Solid Model of Contact Gear Pair

B. *Meshing of Helical Gear*

Meshing is basically the division of the entire model into small cell so that at each and every cell the equations are solved. It gives the accurate solution and also improves the quality of solution [5]. Figure 3 shows meshed 3-D model of helical gear pair.

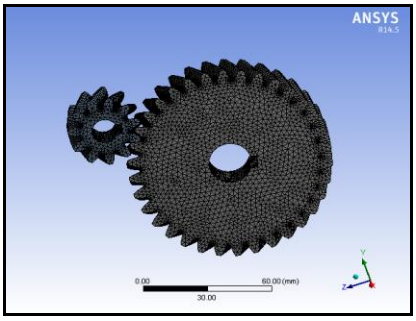

Figure 3. Meshed 3-D model of Helical Gear Pair

C. *Boundary Conditions*

Helical gear assembly was imported in ANSYS 14.5 and the boundary conditions were applied to the gear model. Frictionless support is applied on inner rim of the pinion gear to allow its tangential rotation but restrict from radial translation. Fixed support is applied on the inner rim of gear. Moment of 37054 N.mm is applied on the face of pinion in counter clockwise direction as a driving torque. In helical gear only 3-D analysis was performed because of the helical profile of its teeth. In this research, six different helix angles of helical gears are used to find out the contact stress.

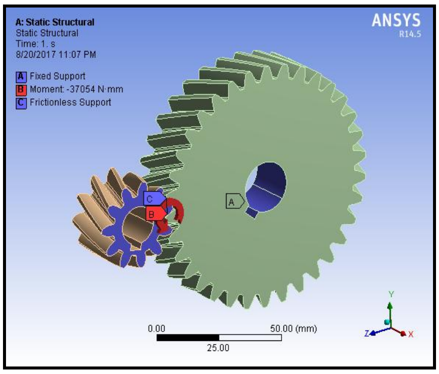

Figure 4. Boundary Condition of Helical Gear Pair

D. von-Mises Stress Analysis of Gear by Using ANSYS

Figure 5 shows minimum and maximum von-Mises stress of helical gear for Helix Angle 20˚ at first gear pair. The values are 0.0003 and 1227.9 N/mm^2 (MPa). Comparing yield strength value 1790 (MPa), the maximum value is less than yield strength. So the design is satisfactory.

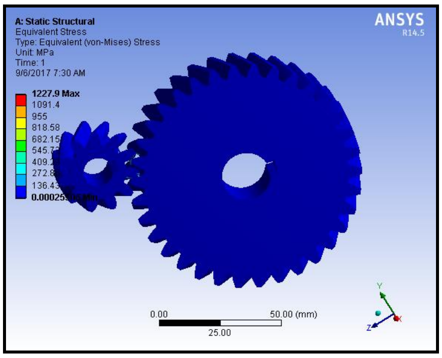

Figure 5. von-Mises Stress of Helix Angle 20˚ at First Gear Pair

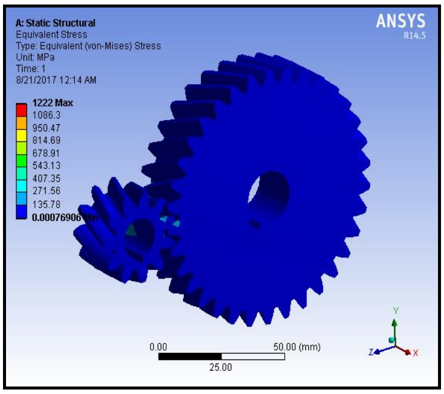

Figure 6. von-Mises Stress of Helix Angle 23˚ at First Gear Pair

Figure 6 shows minimum and maximum von-Mises stress of helical gear for Helix Angle 23˚ at first gear pair. The values are 0.0007 and 1222 N/mm^2 (MPa). Comparing yield strength value 1790 (MPa), the maximum value is less than yield strength. So the design is satisfactory.

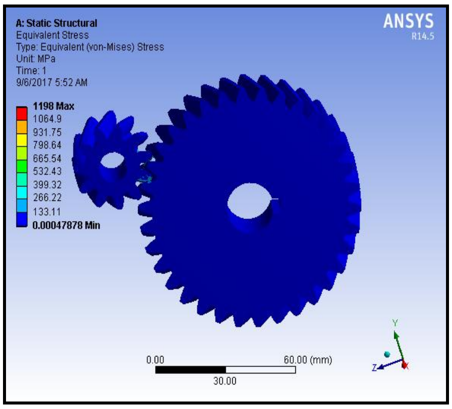

Figure 7. von-Mises Stress of Helix Angle 26˚ at First Gear Pair

Figure 7 shows minimum and maximum von-Mises stress of helical gear for Helix Angle 26˚ at first gear pair. The values are 0.0005 and 1198 N/mm^2 (MPa). Comparing yield strength value 1790 (MPa), the maximum value is less than yield strength. So the design is satisfactory. Figure 8 shows minimum and maximum von- Mises stress of helical gear for Helix Angle 29˚ at first gear pair. The values are 0.0004 and 1182 N/mm^2 (MPa). Comparing yield strength value 1790 (MPa), the maximum value is less than yield strength. So the design is satisfactory.

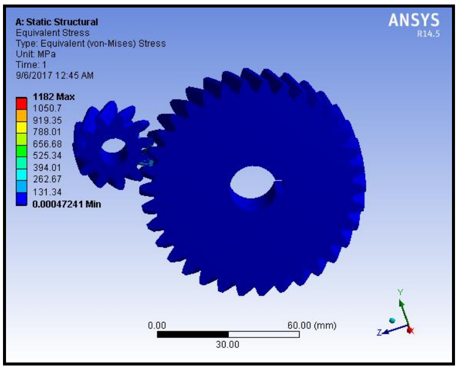

Figure 8. von-Mises Stress of Helix Angle 29˚ at First Gear Pair

Figure 9 shows minimum and maximum von-Mises stress of helical gear for Helix Angle 32˚ at first gear pair. The values are 0.03 and 1163 N/mm^2 (MPa). Comparing yield strength value 1790 N/mm^2 (MPa), the maximum value is less than yield strength. So the design is satisfactory.

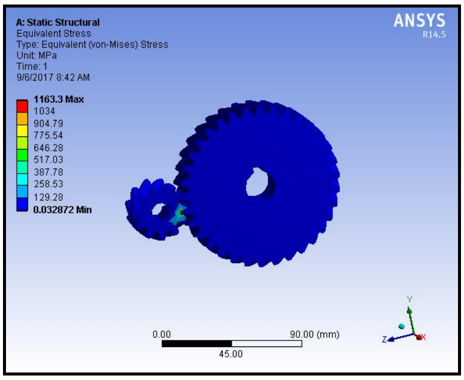

Figure 9. von-Mises Stress of Helix Angle 32˚ at First Gear Pair

Figure 10 shows minimum and maximum von-Mises stress of helical gear for Helix Angle 32˚ at first gear pair. The values are 0.0005 and 1155.8 N/mm^2 (MPa). Comparing yield strength value 1790 (MPa), the maximum value is less than yield strength. So the design is satisfactory. Table V shows the comparison of maximum von-Mises stress results at first gear pair.

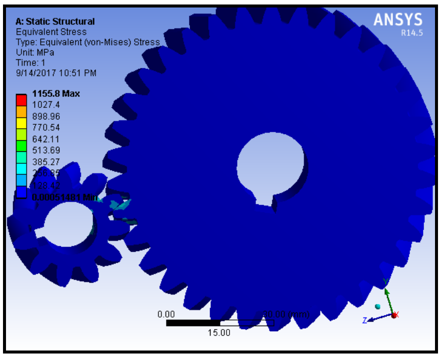

Figure 10. von-Mises Stress of Helix Angle 35˚ at First Gear Pair

TABLE V COMPARISON OF MAXIMUM VON-MISES STRESS RESULTS AT FIRST GEAR PAIR

Helix Angles	von-Mises Stress (Theoretical) (MPa)	von-Mises Stress (ANSYS) (MPa)	% Deviation	Yield Stress (MPa)
20°	1215.83	1227.3	0.93	1790
23°	1203.35	1222	1.53	
26°	1189.08	1211.7	1.87	
29°	1172.98	1194.8	1.83	
32°	1155.03	1163	0.69	
35°	1135.18	1155.8	1.72	

E. Specification of Materials for Helical Gear

TABLE VI SPECIFICATION OF MATERIALS FOR HELICAL GEAR

Material Properties	AISI 5160 OQT 400	Stainless Steel	Structural Steel
Young's Modulus	207 GPa	200 GPa	200 GPa
Poisson Ratio	0.3	0.31	0.3
Density	7850 kg/m3	7750 kg/m^3	7850 kg/m^3
Tensile Yield Stress	1790 MPa	586 MPa	460 MPa
Ultimate Stress	2220	207	250

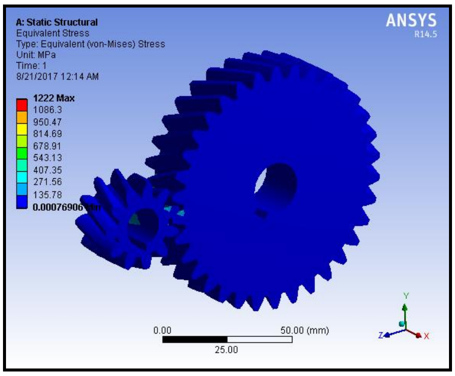

Figure 11. von-Mises Stress for AISI 5160 OQT 400

Finite element analysis of helical gear is done by means of ANSYS workbench 14.5. von-Mises stress of helical gear is found for deferent materials (AISI 5160 OQT 400, Stainless Steel and Structural Steel). As shown in the Figure 11, von-Mises stress for AISI 5160 OQT 400 material is 1222 MPa. Figure 12 shows the simulation result of first gear assembly with Stainless Steel material at helix angle 23 degree. Figure 13 shows the simulation result of first gear assembly with Structural Steel material at helix angle 23 degree.

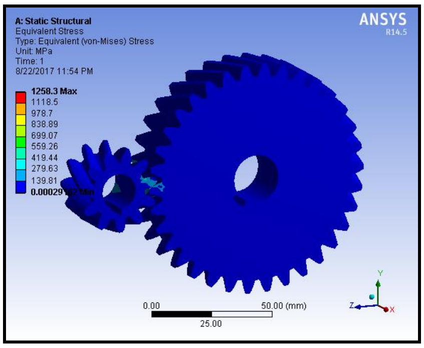

Figure 12. von-Mises Stress of Helix 23˚ with Stainless Steel

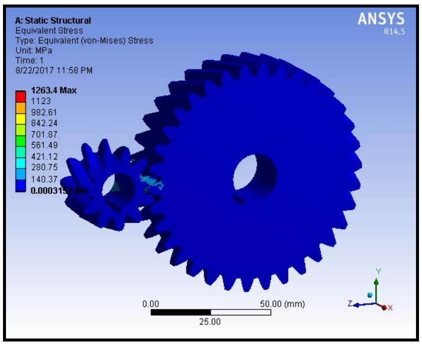

Figure 13. von-Mises Stress of Helix 23˚ with Structural Steel

V. RESULTS AND DISCUSSIONS

From the above figures, it was found that the maximum von-Mises stress and strain values of AISI 5160 OQT 400 are less than other two materials. According to above figures' results, the AISI 5160 OQT 400 material should be chosen for helical gear to be reduced stress. The structural stress analysis of the helical gear tooth model is carried out using the FEA in ANSYS 14.5. The contact stress is analysed by simulating the real contact region between the two mating gears. For determining the stresses at any stage during the design of the gear helix angle is an important parameter. To determine the stress variation with the helix angle, various models of helical gear are made by keeping other parameters i.e. number of teeth, face width etc constant. Table V clearly shows the results of the variation in helix angle from 20˚ to 35˚, there is continuous decrement in the value of the stress at the tooth of the helical gear. Results of theoretical, and ANSYS are less than the yield strength, therefore the design are accepted.

VI. CONCLUSION

The helix angle ψ is always measured on the cylindrical pitch surface. It ranges between 10˚ and 45˚. Commonly used values are 15, 23, 30 or 45˚. Lower values give less end thrust. Higher values result in smoother operation and more end thrust. Above 45˚ is not recommended .In single helical gears, the helix angle ranges from 20° to 35°, while for double helical gears (herringbone gears), it may be made up to 45°. The effect of helix angle on contact stress is studied for six different angles are 20˚, 23˚, 26˚, 29˚, 32˚ and 35˚. The von-Mises values of helical gear (1215.83 Mpa, 1203.35 Mpa, 1189.08 Mpa, 1172.98 Mpa, 1155.03 Mpa and 1135.18 Mpa) were calculated by using AGMA equation. By observing the analysis results, the stress values obtained are less than their yield stress (1790 MPa). So the design is safe under working conditions. The helix angle values of helical gear obtained using equation ANSYS is higher than AGMA values. During the contact of gear and pinion, the contact stress is decreased with the increase of helix angle. In this research, helix angle 23° is chosen for helical gear according to the range of helix angles, light weight and it is commonly used for helical gear. In this research, stress analysis of helical gear is also done for three different materials (AISI 5160 OQT 400, Stainless Steel and Structural Steel). von-Mises stress is found out by FEA software package ANSYS 14.5. By comparing the stress analysis results, the maximum von-Mises stress and strain values of AISI 5160 OQT 400 are less than other two materials.

REFERENCES

[1] Parth J. Bhatt et.al, 'Material optimization of high speed single helical gear by using fea approach 'PG student, department of mechanical engineering Noble group of institute Junagadh - 362 001, India, 2016.

[2] Chetan E. Kolambe et al. 'Study of Helical Gear Analysis Using FEA Software' Savitribai Phule Pune University, India, 2016.

[3] Dadi vijay et.al, 'Design and Structural Analysis of High Speed Helical Gear Using ANSYS' Department of Mechanical

Engineering Kakinada Institute Of Technology And Science, Divili, 2016.

[4] J. Venkatesh, P. B. G. S. N. Murthy, 'Design & Structural analysis of High Speed Helical Gear by using Ansys, IJERA., India, vol.4,Issue 3, pp. 01–05, 2014.

[5] B. Venkatesh et.al, 'Effect of Bending Stress on Steel Alloy Of Helical Gear For High Speed Applications' Department of Mechanical Engineering, Vardhaman College of Engineering, Hyderabad, India, 2014.

[6] Babita Vishwakarma et al. 'Finite Element Analysis of Helical Gear Using Three-Dimensional Cad Model', Government Engineering College, Jabalpur-482004, MP, India, 2014.

[7] R.S. Khurmi and J.K. Gupta, 'A Text Book of Machine Design (SI Unts), EURASIA PUBLISHING HOUSE (PVT.) LTD. RAM NAGAR, NEW DELHI-110 055, 2005.