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Reducing the Root Fillet Stress in Helical Gear Using Internal Stress Relieving Features

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Abstract

Gear is the most effective component in mechanical power transmission system. In the gear design, the bending stress and surface stress of the gear tooth are considered to be one of the main contributions for the failure of the gears in the gear set. Thus the analysis of the stresses has become popular on area of research on gears to minimize or to reduce the failures and for optimal design of gears. To estimate bending stress, three dimensional solid models for different face widths, helix angle are generated using pro-E. And numerical solution is done by ANSYS, which is a finite element analysis package. A three dimensional gear according to AGMA standards will be modeled using pro-E. A similar model with single hole or combination of holes in the test gear segment and another model with more ductile material within the holes will be modeled using pro-E. The three models will be subjected to finite element analysis by assuming a constant force at the pitch diameter along the line of contact. The results obtained from the analysis of three gear models will be compared. Gear profile are created in pro engineering using the relation and equation modeling procedure, depending on certain key features like number of teeth, diametral pitch, pressure and helix angle. Parametric study is conducted by varying the face width and helix angle to study their effect on the bending stress of helical gear. Life time (no of cycles) of gear is also calculated based on the bending stress.

Keywords

Helical Gear, Bending Stresses, Helix Angle, Face Width, Life Time Etc.

I. Introduction

One of the best methods of transmitting power between the shafts is gears. Gears are used to transmit torque and angular velocity in a wide variety of application. There are also a wide variety of gear types to choose from. Spur gear designed to operate on parallel shafts and having teeth parallel to the shaft axis, and helical gears designed to operate on parallel shafts [7]. Other gear types such as bevel and worm can accommodate nonparallel shafts. Gears are standardized by AGMA (American Gear Manufacturers Association) depending on size and tooth shape. Gears may be classified into three types based on the axes of the shaft. The axes of Parallel, Intersecting, Non-Intersectingand Non-parallel. A primary requirement of gears is most remain constant of angular velocities or proportionality of position transmission Constant velocity (i.e., constant ratio) motion transmission is defined as "conjugate action" of the gear tooth profiles. Law of gearing ids defined as a common normal to the tooth profiles at their point of contact must, in all positions of the contacting teeth, pass through a fixed point on the line-of-centers called the pitch point".

Any two curves or profiles engaging each other and satisfying the law of gearing are conjugate curves. When the tooth profiles, or cams, are designed so as to give a constant angular-velocityratio during meshing these are said to be conjugate action. Involute and cycloidal profile teeth satisfy law of gearing. The curve farming face and flank is called profile. If the gear tooth is not an involute, then the centerdistance will violatethe fundamental law of gearing and there will be variations in the output velocity.So output is not equal to input.With an involute tooth for the center distance ensures do not affect the velocity ratio. This is the advantage.The pressure angle is affected by the change in center distance.Center distance is directly proportional to pressure angle. Velocity ratio of involute gear is fixed by the ratio of the base circular diameter which are unchanged once the gear is cut.

A. Helical Gear Geometry

Fig. 1 showsthe geometry of a basic helical rack. The teeth from the helix angle ψ with the "axis" of the rack. The teeth are cut at this angle and the tooth from is then in the normal plane. Lines AB and CD are the centre lines of the two adjacent helical teeth taken on the pitch. The distance between AC is the transvers circular in the plane of rotation.The distance between BD is normal circular pitch. The normal pitch p_n and the normal pressure angle ϕ_n are measured in this plane [6].

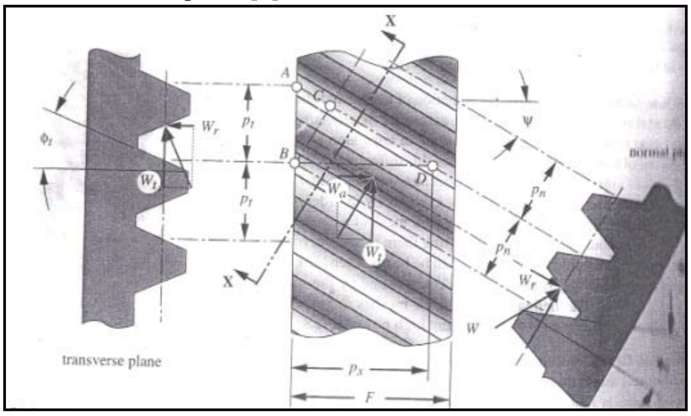

Fig. 1: Portion of Helical Rack

The transverse pitch p_t and the transverse pressure angle ϕ_t are measured in the transverse plane. These dimensions are related to one another by the helix angle. The transverse pitch is the hypotenuse of the right triangle ABC.

$p_t = p_n/\cos\psi$

Substituting (P=1/m) in the above expression, we have $m_n = m \cos\Psi$

Where m_n =Normal module (mm)

m = Transverse module (mm)

An axial pitch p_x can also be defined as the hypotenuse of the right triangle BCD.

$p_x = p_n/\sin\psi$

p_t Corresponds to circular pitch p_c measured in the pitch plane of a circular gear. Diametral pitch is more commonly used to define tooth size and is related to circular pitch by

$$p_d = \frac{N}{d} = \frac{\pi}{p_c} = \frac{\pi}{p_t}$$

Where N is number of teeth and d is pitch diameter.
The diametral pitch in the normal plane is
$p_{nd} = p_d/\cos\psi$
The pressure angles in the two planes are related by
$\tan \phi_t = \tan\phi = \tan\phi_n/\cos\psi$
The Pitch Circle Diameter d of the helical gear is given by

$$d = \frac{zp}{\pi} = \frac{z}{P} = zm = \frac{zm_n}{\cos\Psi}$$

II. Literature Review

VIjayaragan and Ganesan [1] presented a static analysis of composite helical gears system using three dimensional finite element methods to study the displacements and stresses at various points on a helical gear tooth. The results of the FEM was tested by the root stress for C-45 steel material gear and comparing the result with obtained from conventional gear design equation. The paper composite helical gears by companion of with thatcomposite helical gears by companion of with thatsubmitted also the evaluation of the performance of of the conventional carbon steel gear. It is observed from the result that composite materials can be used safely for power transmission helical gears but the face width has to be suitably increased.

Rao and Muthureerapp [2] an explained about the geometry of helical gears by simple mathematical equations, the load distribution for different positions of the contact line and the stress analysis of helical gears using the three dimensional finite element methods. A computer program has been developed for the stress analysis of the gear. Root stresses are evaluated for different positions of the contact line when it moves from the root to the tip. To examine the validity of the developed program, the changes in thetrend of the maximum root stress values at various places of the tooth along the face width were compared with the experimental results a parametric study was made by varying the face width and the helix angle to study their effect on the root stresses of helical gears. Based on their study the effect of helix angle and face width on the toot stresses of helical gears was clarified for different positions of the contact line.

Chen and Tsay [3] investigate the contact and the bending stresses of helical gear set with localized bearing contact by means of Finite Element Analysis (FEA). The proposed helical gear set comprises an involute pinion and double crowned gear. Mathematical models of the entire teeth geometry of the pinion and the gear have been derived based on the theory of gearing. Correspondingly, a mesh generation program was also developed for finite element stress analysis. The gear stress distribution is studied using the commercial FEA package, ABAQUS standard. Additionally, several examples are presented to demonstrate the influences of the gear's design parameters and the contact positions on the stress distribution.

In 1992 Srinivasalu [4] experimented by placing the holes in relatively low stress neutral axis of the bending of the gear tooth. Even if bending stress reductions were not obtained from this placement, the aided flexibility of the gear tooth reduced the contact stress and improved the fatigue life of the gear. The theory that themaximum tensile stress can be reduced by placement of holes in the stressed area is based on the idea that stress will be relived and displaced away from the critical area. This would finally change the stress distribution in the gear and would most likely transfer more of the stress to the Compressive side of the gear teeth.

Jianfenget al[5] proposed a method namely the normal stiffness matrix along contact line (NSMCL) for analyzing cylindrical gears. The method established three dimensional finite element models for spur and helical gears; external and internal hobbling and slotting, various parameters and materials can be analyzed using these models. Results such as load distribution along the contact lines, deformations and stiffness at any point, and contact stresses are presented. The calculated results show that the trend of gear tooth deformation coincides with the tested one s using the dynamic speckle photography method.

III. Problem Definition and Theoretical Design Calculation For Helical Gear

A. Problem Definition

Gears which are very rigid develop high stress concentration at the root and contact point when subjected to loads. Due to these high stresses at the root and contact point there is a higher chance of fatigue failure at these locations. As the contact point shifts along the profile of the tooth a surface fatigue failure is more likely but the quick shifting of the load and enhanced material properties due to surface treatments means that this failure is not as critical. The repeated stress that occurs on the fillets is practically found to be the fatigue failure of the gear tooth. There is scope of improved design of gears by an introduction of stress relief features. These reduce stiffness which causes amore distributed lower stress at the fillets increases gear life. Stress relief features provide flexibility and help in reducing thepoints of stress concentration. It is proposed to investigate the effect of stress relief features of different size, shape, locationand number. A study is donefor helical gear with involute profile shapes for effects of stress relief feature of hole. The work done by Fredette L. and Brown M [8] is to be studied and furthered in this work.

Table 1: Helical Gear Parameters

S.No	Description	Value
1	Pressure angle	20^0
2	Helix angle	15^0
3	Face width	260
4	No. of Teeth	126
5	Pitch diameter(mm)	2502.5
6	Transmitted load	19624.38N
7	Revolutionsper Minute	3500
8	Normal Module	18
9	Modulus of elasticity	2
10	Poisson's ratio	0.3

Theoretical Design Calculation for Helical Gear
Input Parameters
Power = p =9000KW = 9000×10^3
Gear ratio = 1
Helix angle (α) = 25^0
Normal Pressure angle ϕ = 20
Material used = 40ni2cr1m028 steel
Properties = BHN =225
Minimum tensile strength = 900 n/mm²
Young's modulus = 2×10^5 N/mm²
Compressive stress = σ_c=11000 kgf /cm²
Bending stress=σ_b=4000kgf /cm² =392265.9 N/mm²

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Normal Module =m= 18

Gear ratio = $GR = \frac{T_G}{T_P} = \frac{N_P}{N_G} = \frac{D_G}{D_P}$

No of teeth on gear = T_G=126

Calculations:

(i). $m_t = \frac{m_n}{\cos\Psi} = \frac{18}{\cos(25)} = 19.86 \text{ mm}$

(ii) Diameter of gear $=\frac{z_p m_n}{\cos\Psi} = \frac{126*18}{\cos(25)} = 2502.43\text{mm}$

(iii) Center distance = $\frac{2502.43+2502.4}{2} = 2502.43$

iv) Normal pitch $P_N = P_c \cos\alpha$

v) circular pitch= $Pc = \frac{\pi D}{T} = \frac{\pi(D_G+D_P)}{T_G+T_P} = \frac{\pi 2592}{144} = 56.52$

P_N=56.52cos 25=51.22

vi) Normal pressure angle = ϕN

$\tan\phi N = \tan\phi \times \cos\alpha$ (ϕ=20)

$\tan\phi N = \tan 20 \times \cos 25 = 0.328$

$\phi N = \tan^{-1}(0.328) = 18.210$

Face Width of Helical Gears

Usually recommended that the overlap should be 15 percent of the circular pitch

vii). b tanα=1.15p_c

$$b = \frac{1.15p_c}{\tan\alpha} = \frac{64.998}{\tan 25} = 139.388$$

viii) The maximum face width may taken as 12.5 m to 20m

b = 20m = 360

viii) Formative or equivalent no of teeth for helical gears =

$$\text{gears} = T_E = \frac{T}{\cos^3\alpha}$$

Equivalent no of teeth on gear =

$$= T_{EG} = \frac{T_G}{\cos^3\alpha} = \frac{126}{\cos^3 25} = 169.42$$

xi) Tooth form factor for gear for 20^0 full depth involute

$$Y^1G = 0.154 - \frac{0.912}{T_{EG}} = 0.154 - \frac{0.912}{169.428} = 0.1486$$

x) Addendum = m_n=19.86 mm

xi) Dedendum= 1.25m_n = 1.25*19.86=24.83 mm

xii) Minimum total depth = addendum+dedendum

xiii) Minimum clearance 0.25m_n =5 mm

xiv) Thickness of tooth = 1.5708M =28.2744

xv) Strength of Helical Gears:

$S_b = m_n b\sigma_b Y$

S_b= Beam strength

σ_b= allowablestaticstress

S_b = face width

m_n = Normal module

Y = tooth form factors

a) The allowable static stress (σ_0) for steel gears is approximately one third of the ultimate tensile strength = $\sigma_0 = \frac{\sigma_u}{3}$

σ_0 = 516.666 N/mm^2

σ_u=1500 N/mm^2

b) Peripheral speed = V $=\frac{\pi D_P N_P}{60\times1000} = \frac{3.14\times2502.5\times3500}{60\times1000} = 458.61\frac{m}{s}$

c) The value of velocity factor C depending upon peripheral velocities greater than 20 m/s is given by $\frac{0.56}{0.56+\sqrt{V}}$

d) $C_V = \frac{0.56}{0.56+\sqrt{V}} = \frac{0.75}{0.75+458.61} = 0.207$

$S_b = m_n b\sigma_b Y$

=18*260*516.67*0.452

=1092.94 KN

Tangential load in helical gear,

T=P/2Πn =9000*1000*60/2*π*3500=24555 Nm

Pt=2T/d=2*24555*1000/2502.5=196.24 KN

Effective load on gear teeth,P_{eff}= $\frac{C_S P_t}{C_V}$=1.5*1092/0.207=142.205 K N

In order to avoid failure of gear tooth due to bending, $S_b > P_{eff}$

So the Design is Safe

xvi) The dynamic tooth load on the helical gear is given by

$$W_D = W_T + \frac{21V(bC\cos^2\alpha + W_T\cos\alpha)}{21V + \sqrt{b \times C\cos^2\alpha + W_T}}$$

Where v, b, c have usual meaning as discussed in spur gears

xvii) C = deformation factor = $\frac{k\times e}{\frac{1}{E_P}+\frac{1}{E_G}}$

K =0.111 for 20^0 full depth involute system

E_G and E_P are young's Modulus of gear and pinion (N/mm^2)

e=Tooth error action (0.032)

C= $\frac{0.111\times0.032}{\frac{1}{2\times10^5}+\frac{1}{2\times10^5}}$ =355.2 N/mm

xviii) $W_D = W_T + \frac{21V(bc\cos^2\alpha+W_T\cos\alpha)}{21V+\sqrt{b\times c\cos^2\alpha+W_T}} = 55457.68 + \frac{21\times415.422(360\times355.2\cos^2 25+55457.68\cos 25)}{21\times415.422+\sqrt{360\times355.2\cos^2 25+55457.68}} = 20394.3958\text{N}$

xix) The static tooth load or endurance strength of the tooth for gear is given by

$W_S = \sigma_e \times b \times \pi \times m \times Y^1$

Where σ_e=1.75xBHN=446.25Mpa (BHN=225) =flexural endurance limit

$$W_S = 446.25 \times 360 \times 3.14 \times 18 \times 0.1163 = 1055996.789$$

xx) Wear load= $W_W = \frac{D_P\times b\times Q\times K}{\cos 2\alpha}$

Where D_P, b, Q and K have usual meanings as discussed in spur gears in this case

$$K = \text{load stress factor} = \frac{(\sigma_{es})^2 \sin\phi N}{1.4}\left(\frac{1}{EP}+\frac{1}{EG}\right)$$

σ_{es}=2.8×BHN-70=644 N/mm²

$$K=\frac{(644)^2 \sin 18.210}{1.4}\left(\frac{1}{2\times10^5}+\frac{1}{2\times10^5}\right) = 0.925$$

$$Q=\frac{2\times VR}{VR+1} = 1 \quad (VR=1)$$

$$\text{Wear load} = W_w = \frac{2268\times360\times1\times0.925}{\cos^2 25} = 919466.3943\ \text{N}$$

Creating Estimated S-N Diagram

i). $S_m = 0.9S_{ut}$
$=0.9\times1500=1350\text{N/mm}^2$

ii) $S_e = 446.25\ \text{N/mm}^2$

iii) $b = -\frac{1}{3}\log\left(\frac{1350}{446.25}\right) = -0.16025$

$\log a = \log S_m - 3b$

iv) $= \log(1350) + 3\times0.16025$

$= 3.611083768$

a = 4083.98

v) $S_n = aN^b$
$=4083.98\times N^{-0.16025}$
$446.25=4083.98\times N^{-0.16025}$

a) When S_n=446.25
log(446.25)=log(4083.98)-0.16025 log(N)
2.65=3.61-0.16025 logN
logN=6
$N=10^6$

b) When S_n=190.83
$S_n=aN^b$
$190.83= 4083.98\times N^{-0.16025}$
log(190.83)=log(4083.98)-0.16025 log(N)
2.28065=3.61-0.16025 logN
logN=1.33043
$N=10^{8.30}$

c) When S_n=47.299
$S_n=aN^b$
log47.299=log4083-0.16025 logN
1.67485=3.61-0.16025 logN
$N=10^{12.083}$

d) When S_n=47.976
$S_n=aN^b$
log47.976=log4083-0.16025 logN
1.68102=3.61-0.16025 logN
$N=10^{12.04}$

IV. Modeling of Helical Gear

A solid modeling is done with Pro E 5.0 and then meshing and Analysis is done with ANSYS Workbench 14.5

Face width 260mm

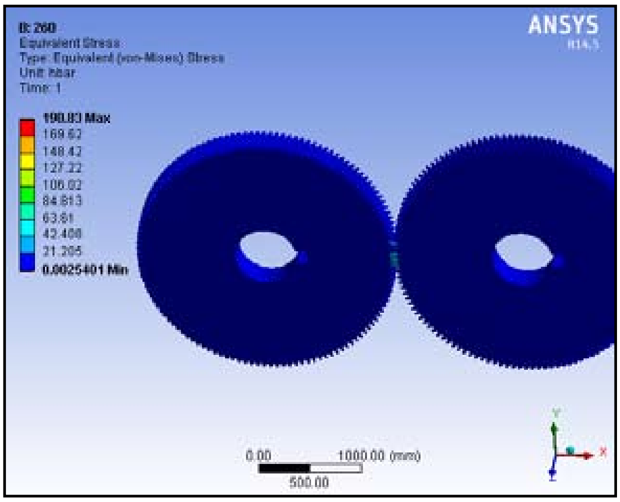

Fig. 2: The Above Image Shows Von-Misses Stress Value 190.83 N/mm²

Face width 300mm

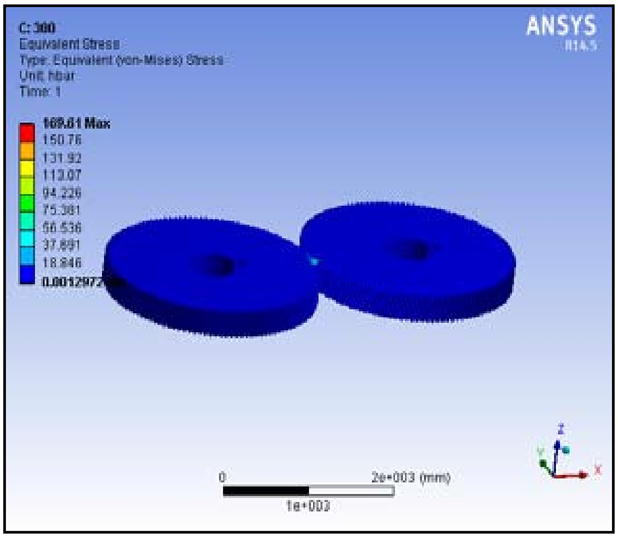

Fig. 3: The Above Image Shows Von-Misses Stress Value 169.61 N/mm

Face width 360mm

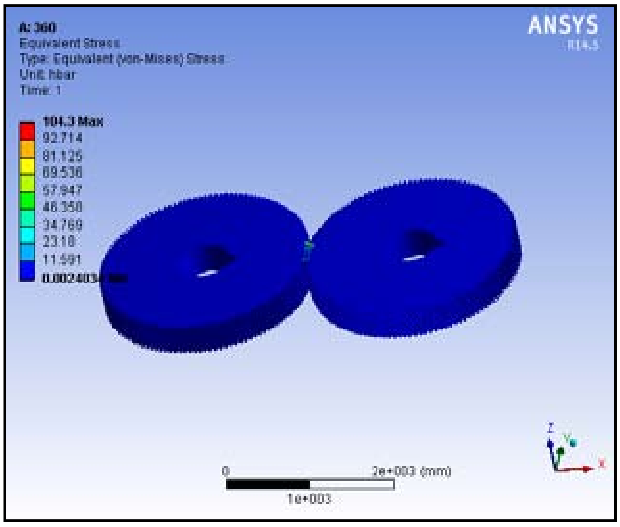

Fig. 4: The Above Image Shows Von-Misses Stress Value 104.3 N/mm²

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Face width 400mm

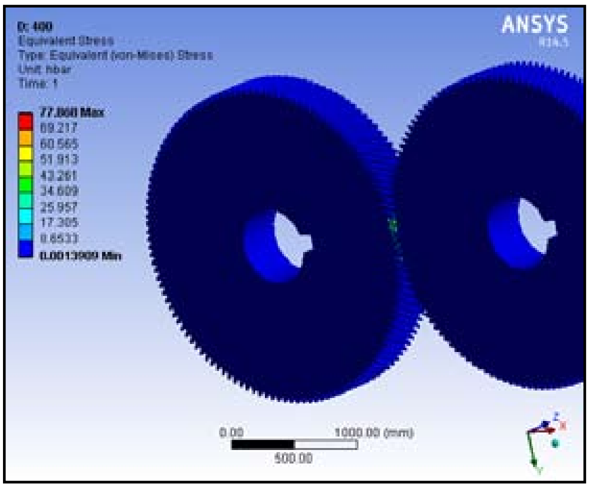

Fig. 5: The Above Image Shows Von-Misses Stress Value 77.868 N/mm²

Helix angle -15°

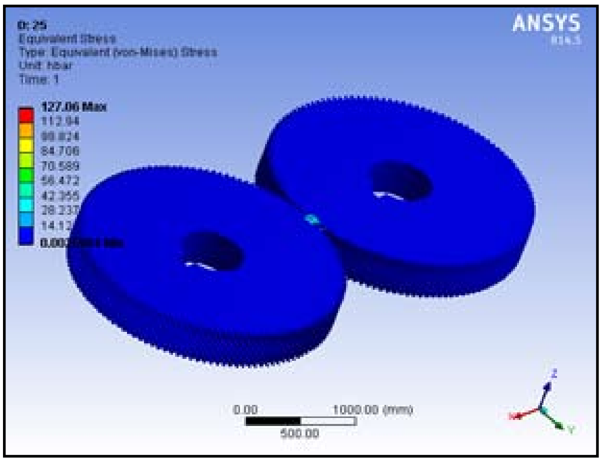

Fig. 6: The Above Image Shows Von-Misses Stress 127.06 N/mm²

Helix angle -250

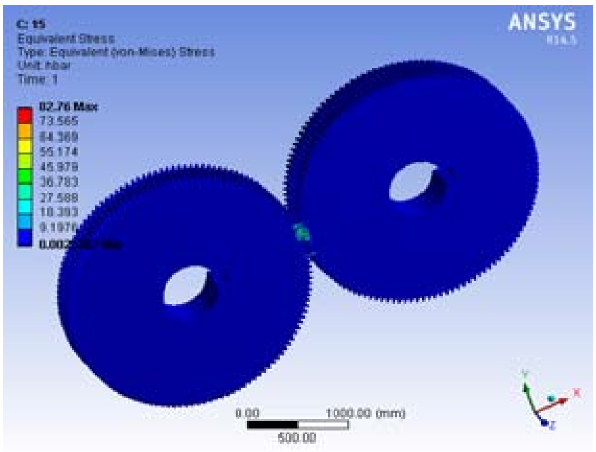

Fig. 7: The Above Image Shows Von-Misses Stress 82.76 N/mm²

Helix angle -300

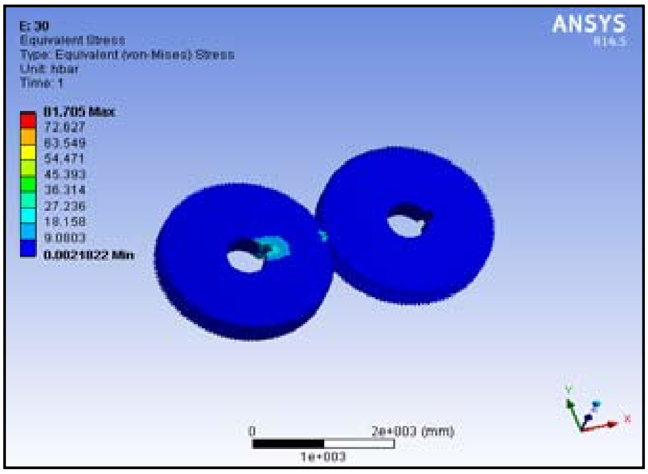

Fig. 8: The Above Image Shows Von-Misses Stress 81.705 N/mm²

Single row holes

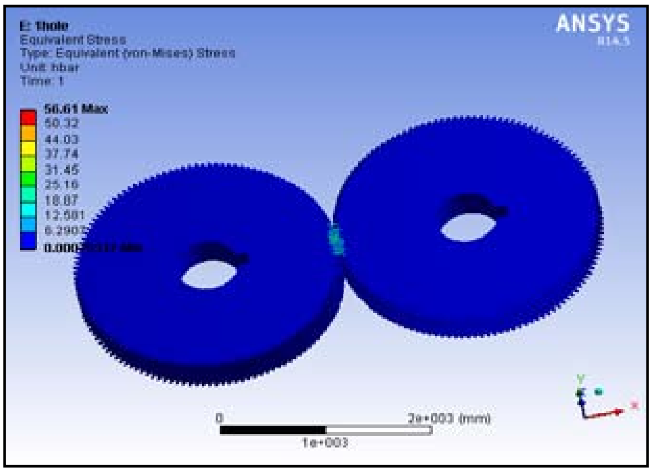

Fig. 9: The Above Image Shows Von Misses Stress 56.61 N/mm²

Double row holes

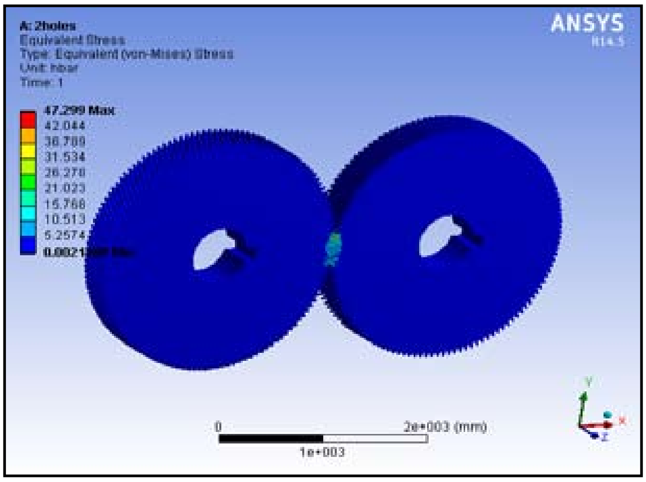

Fig. 10: The Above Image Shows Von-Misses Stress 47.299N/mm²

Holes between the adjacent holes

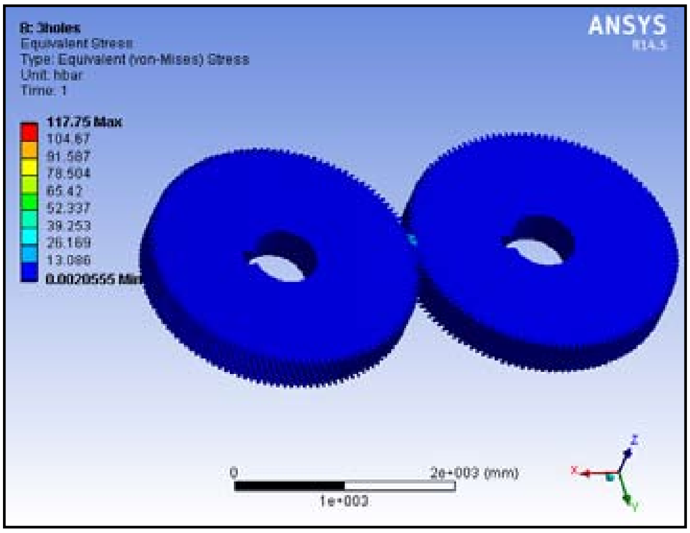

Fig. 12: The Above Image Shows Von-Misses Stress 117.75 N/mm^2

Holes Filled With Material

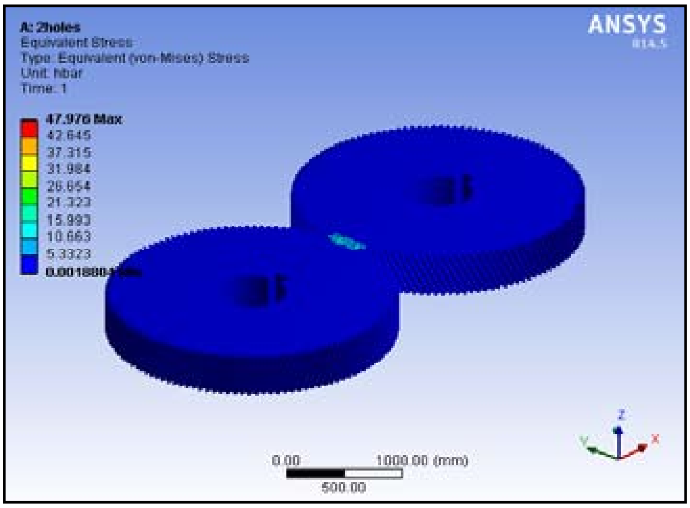

Fig. 13: The Above Image Shows Von-Misses Stress 47.976 N/mm^2

V. Results and Discussion

A. Effect of Face Width on Helical Gear Tooth

The effect of face width on von-misses stress (maximum bending stress) is studied by varying the face width for four different values which are 260mm, 300mm, 360mm and 400mm. The magnitude of stresses obtained for these face widths has been displayed below in the table.

Table 2: Effect of Face Width on Maximum Bending Stress

Face width	Displacement (mm)	Strain	Stress (N/mm^2)
260	1.5305	0.0095413	190.83
300	1.6458	0.0086248	169.61
360	0.71624	0.0052151	104.3
400	0.40622	0.0038934	77.868

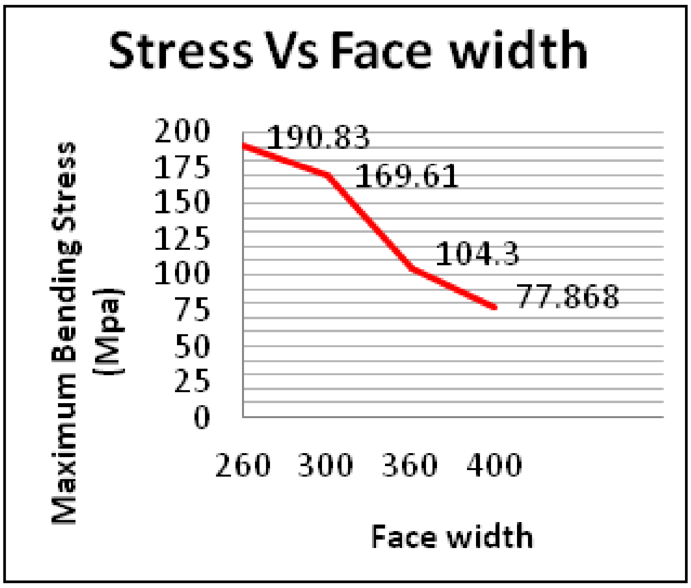

Fig. 14: Graphical Representation of Effect of Face Width on Maximum Bending Stress

B. Effect of Helix Angle on Helical Gear Tooth

The effect of helix angle on maximum bending stresses is studied by varying helix angle for three different values which are 150, 250 and 300 . Apart from the constant number of teeth, module, pressure angle and the face width also kept constant. The variations in the magnitude of stresses with respect to helix angle are displayed below in the table.

Table 3: Effect of Helix Angle on Maximum Bending Stress

Helix Angle	Displacement (mm)	Strain	Stress (N/mm^2)
150	0.43498	0.0053834	127.06
250	1.1032	0.0066889	82.76
300	1.9229	0.0042926	81.705

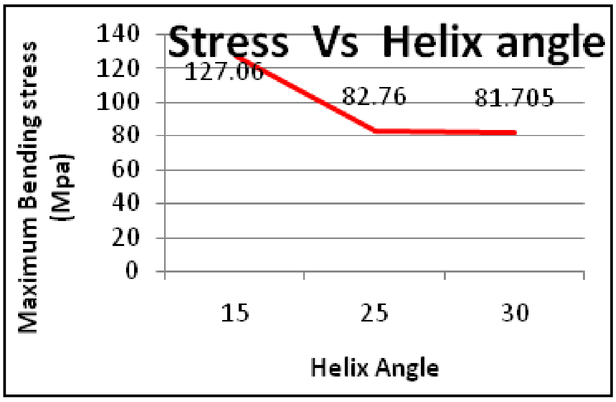

Fig. 15 Graphical Representation of Effect of Helix Angle on Maximum Bending Stress

C. Effect of Stress Relieving Features on Helical Gear Tooth

It is found that the von-misses stress at the root fillet is 190.83 N/mm2with out any relieving feature.Stress reliving features did not improve gear stresses. Three cases were used in this model one hole, two holes, and three holes .The first model was performed by creating holes in the gear body with a diameter of 10mm in the direction of axis, along the circumference of base circle. The second model was also performed by creating holes as a row in the direction of axis, along the circumference of the base circle. The third model was performed by creating holes in between the two adjacent holes. Forsecond case, it was found that the best result was obtained when two row holes as a stress relieving feature.

Table 4: Effect of Stress Relieving Features on Maximum Bending Stress

Stress relieving features	Displacement (mm)	Strain	Stress (N/mm²)
Single row Hole	0.39472	0.0032897	56.61
Double row Holes	0.38183	0.0044217	47.299
Holesbet'n adjacent	0.93304	0.0061604	117.75

D. Effect of Stress Relieving Features (With Filling Material) on Helical Gear Tooth

The holes on the gear model is through holes .The dense mesh around the circle represents the stress at the hole surface which were at tooth root fillet before the holes were made. By making holes in the gears, stresses from tooth root fillet are distributed to the surface of the holes. The holes have to be filled with a material more ductile than gear material. So that, the maximum stress from the critical areas like tooth root fillet radius are shifted to material in the holes. Gold, silver, platinum, copper, iron, aluminum are the best examples of ductile materials.

Table 5: Effect of Stress Relieving Features (With Filling Material) on Maximum Bending Stress

With Filling material	Displacement (mm)	Strain	Stress (N/mm²)
2 Holes	0.362	0.00245	47.976

Table 6: Effects of Face Width,Helix Angle, Stress Relieving Features and With Filling Material on Maximum Bending Stress

		Displacement (mm)	Strain	Stress (N/mm²)
Face width	260	1.5305	0.0095413	190.83
Helix Angle	150	1.1032	0.0066889	127.06
Stress relieving features	2 Holes	0.38183	0.0044217	47.299
With Filling material	2 Holes	0.362	0.00245	47.976

E. Effect of Stress on Life of the Gear (No of Cycles)

Table 7: Effect of Stress on Life of the Gear (No of Cycles).

TYPE OF STRESS	STRESS VALUE	NO OF CYCLES
Ultimate stress	1350	103
Endurance stress	446.25	106
Bending stress	190.83	108.30
Bending stress	47.299	1012.08
Bending stress (With filler material)	47.976	1012.04

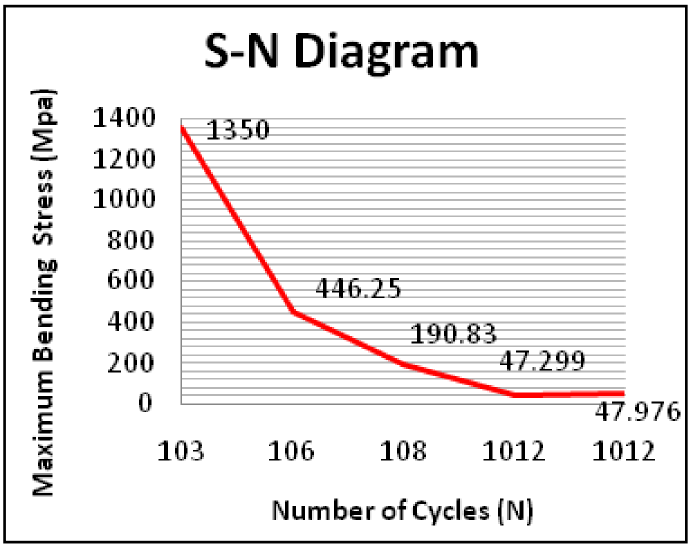

Fig. 16: Graphical Representation of Effect of Stress on Life of The Gear (No of Cycles)

Stresses can be reduced by a large amount while deflection increased due to loss of rigidity. This give exponential increase in the life of gears The above graph shows the life of gears (no of cycles).The life of gear without hole is about 108.30 and that with a hole of 10 mm diameter is 1012.08 an increase of 6066 times in the life a gear. The holes are filled with material more ductile than gear material. So the stresses from critical areas are shifted to the material in the holes. In this way we can avoid loss of rigidity. The life of the gear with filler material is about 1012.04.

VI. Conclusions and Future Scope

The aim of this study was to create stress relieving features to reduce the root fillet stress in helical gear. In addition to pilot model, two extra models have been used in this study. The finite element modelling was performed using Ansys.

Firstly, a pilot model with no holes was analysed to predict stresses at the root fillet of the gear by varying face width and helix angle. From the result and graph von-misses stress decreases with increase of face width and helix angle.

The second model has been performed by creating holes in the gear body. Three cases were used in this model; single row holes, double row holes, and placing a hole in between the two adjacent holes. The Consequently, the second model were constructed to examine the effect of creating holes in the gears body as stress relieving features on root fillet stresses. Increasing the number of holes resulted in higher stress reduction compared to pilot model. For double row hole case, it was found that the best results were obtained when double row hole as a stress reliving feature was created along the axis direction. But gear rigidity in this case was questionable. On the contrary, stress increase because of the reduction in contact area.

We know that with the increase of stress, life of gear decreases. With best result which was obtained in second model second case, exponential increase in the life of the gear has been calculated.

Scope of Further Work

- The bending stresses can be analysed for other type of gears such as bevel, spur and worm etc by varying face width, helix angle and number of teeth.
- Different locations, number of holes and shapes of hole can also be studied for relieving stress.
- Study can be conducted by varying the depth of a groove

feature only.

- Other failure criteria can be used for the study other than fatigue.(vibrational analysis, wear analysis, noise reduction, election analysis)
- Study can be conducted on the whole gearbox with all elements in the system including gear casing and bearing.
- Study can be conducted to find the effect of bending stress for helical gear tooth with spokes

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