

Evaluation of Face Width and Helix Angle Influence on the Performance of Helical Gears through FEM Technique

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Abstract

This work made an attempt to summarize about contact stresses developed in a mating helical gear which has involute teeth. Face width and helix angle are important geometrical parameters in determining the state of stresses during the design of gears. The analysis of contact stresses has become popular as an area of research on gears to minimize and reduce the failures. The present investigation is carried out to make use of helical gear, by analyzing the contact stresses for different helix angles (0°, 5°, and 15°) and (15 mm, 20 mm, and 25 mm) Face width. A Three-dimensional solid model is generated by Solid Works which is powerful and modern solid modeling software. The numerical solution is done by Ansys, which is a finite element analysis package. The analytical approach is based on contact stress equation, to determine the contact stresses between two mating gears. The results obtained from Ansys values are compared with Analytical results are in close agreement. Results advices that with increase in helix angle and face width, contact stresses decreases for a pair of helical gears. The present analysis is useful in quantifying the above said parameters that helps in safe and efficient design of the helical gear.

Keywords: *Contact Stress, Helix Angle, Face Width, Friction Coefficient, ANSYS-14.5, VB Interface.*

INTRODUCTION

Gears are essential to the global economy and are used in nearly all applications where power transfer is required, such as automobiles, industrial equipment, airplanes, helicopters, and marine vessels. The understanding of the behaviour when gears are in mesh is extremely important if one wants to perform system monitoring and control of the gear transmission system. Although there are large amount of research studies about various topics of gear transmission, the basic understanding of gears in mesh still needs to be confirmed. When a pair of gears mesh, localized Hertzian contact stress are produced along with tooth bending and shearing. This is a non-linear problem, and it can be solved by applying different types of contact elements and algorithms in finite element codes.

However, due to the complicated contact conditions, acquiring results in the meshing cycle can be challenging since some solutions may not converge. In any case, using quadrilateral elements seem to be useful in solving gear contact problems with finite element analysis. Furthermore, meshing stiffness is often being discussed when a pair of gears are in mesh. Meshing stiffness can be separated into Torsional

Mesh Stiffness and Linear Tooth Mesh Stiffness.

Gears are a critical component in the rotating machinery industry. Various research methods, such as theoretical, numerical, and experimental, have been done throughout the years regarding gears. One of the reasons why theoretical and numerical methods are preferred is because experimental testing can be particularly expensive. Thus, numerous mathematical models of gears have been developed for different purposes.

In loaded gears, the contact behavior along the path of contact changes significantly. This causes variation in gear contact conditions along the tooth flank, leading to failures such as surface wear such as surface wear, scuffing and pitting at different positions on the tooth surface. Pitting is a surface failure due to repetitive high contact stresses which occur during gear mesh. Hence, the contact stress behavior along the line of action is crucial information, but its evaluation for real gear conditions is complicated. By nature, contact phenomena always involve friction. If the exact contact behavior needs to be studied, then the frictional variations have to be included. The friction coefficient along the path of action

varies continuously, thus affecting the contact conditions and stress. Finite element method (FEM) applied on the involute spur gear is the most appropriate method to solve the nonlinear contact problem having different friction coefficients at the point of contact.

Nomenclature

A	transverse pressure angle, deg
a	addendum, mm
b	length of contact, mm
β	helical angle, deg
Cp	elastic coefficient factor
D	pitch diameter of pinion and gear, mm
E	Young's modulus, G Pa
Ft	tangential force on gear tooth, N
Fn	normal force on gear tooth, N
i	speed ratio
I	geometry factor
Kv	dynamic factor
Ko	overload factor
KM	load distribution factor
Ks	size factor
n1 & n2	number of teeth, suffix 1 for the pinion and 2 for the gear
σ_c	Contact stresses, M Pa
R	radial distance from centre to pitch circle, mm
r1 & r2	the pitch radii of pinion and gear, mm

μ	coefficient of friction
ν	Poisson's ratio
V	velocity of rotation, m/s
ZR	surface condition factor

LITERATURE REVIEW

Santosh S Patil, Saravanan Karuppanan, Ivana Atanasovska, Azmi Abdul Wahab: In this paper an attempt has been made to study the contact stresses among the helical gear pairs, under static conditions, by using a 3D finite element method. The helical gear pairs on which the analysis was carried out were 0° , 5° , 15° , 25° helical gear sets. The effect of friction was varied at the point of contact which made the problem nonlinear and complicated. To simplify the problem, a range of average static coefficients of friction, from 0 to 0.3, has been considered as the scope of study. The variation of contact stresses with helix angle and also with friction coefficients has been discussed. The commercial finite element software used was ANSYS and the results were compared with analytical calculations.

Ivana Atanasovska, Vera Nikolic-Stanojlovic, Dejan Dimitrijevic, Dejan Momcilovic: The paper also describes the Finite Element Method simulation of contact conditions for helical gears teeth with an involute profile. A suitable

analysis is performed in order to select a meshed gears model which is sufficiently economic and in same time sufficiently geometrically accurate. The special algorithm for the tooth involute profile drawing is developed and built in currently available software for Finite Element Analysis to assure drawing of real flanks contact geometry.

Zheng Li and Ken Mao: The present paper concentrates on sliding friction is one of the major reasons causing gear failure and vibration; the adequate consideration of frictional effects is essential for understanding gear contact behaviour accurately. An analysis of tooth frictional effect on gear performance in spur gear is presented using finite element method. Nonlinear finite element model for gear tooth contact with rolling/sliding is then developed. The illustrated bending stress and transmission error results with static and dynamic boundary conditions indicate the significant effects due to the sliding friction between the surfaces of contacted gear teeth, and the friction effect cannot be ignored. To understand the particular static and dynamic frictional effects on gear tooth contact analysis, some significant phenomena of gained results will also be discussed.

S. Sai Anusha, P. Satish Reddy, P. Bhaskar, M. Manoj: The present investigation is carried out to make use of helical gear, by analyzing the contact stresses for different Pressure angles ($14.5^\circ, 16^\circ, 18^\circ, 20^\circ$) Helix angles ($15^\circ, 20^\circ, 25^\circ, 30^\circ$) and (80 mm, 90 mm, 100 mm, 110 mm, 120 mm) Face width. A Three-dimensional solid model is generated by Pro-E that which is powerful and modern solid modeling software. The numerical solution is done by Ansys, which is a finite element analysis package. The analytical approach is based on contact stress equation, to determine the contact stresses between two mating gears. The results obtained from Ansys values are compared with theoretical values are in close agreement. The present analysis is useful in quantifying the above said parameters that helps in safe and efficient design of the helical gear.

Dr M S Murthy, Pushpendra kumar Mishra: This paper presents a detailed study of different techniques proposed and used by various researchers to optimize and to calculate the stresses involved in the helical gear design. Several three dimensional solid models of gears of different specification have been developed by various researchers for analysis. They then used various analysis

tools like Ansys, Computer aided FEM; Pro-e software etc for analysis. In this work various parameters that can affect the gear tooth, i.e. variation in face width, helix angle etc. for the complex design problems is also discussed. After going through the literature it is suggested that Matlab Simulink could be used as one of the tools to effectively solve the problems which are taken up as future work.

Mr. Kishor N. Naik, Prof. Dhananjay Dolas: The main objective of this paper is to analyze the bending stresses occur on the gear tooth profile of gear used in gear box of special purpose machine also effect on bending stress by variation of the gear parameters. Face width and root radius are taken gear parameters, how stress redistribution are taken place by varying this parameter studied. The stresses are calculated with the help of the FEA this result are compared with the stresses calculated by Lewis equation.

Muhammad, A. A. Umar: This paper presents that there are two kinds of stresses in gear teeth, root bending stresses and tooth contact stresses. These two stresses results in the failure of gear teeth, root bending stress results in fatigue fracture and contact stresses results in pitting failure at the contact surface. But it

is impossible to design gears with infinite life against surface failure. In this paper one of the principal failure modes are studied based on the calculation of bending stress.

J. Venkatesh, P.B.G.S.N. Murthy: This paper presents the Design and Structural analysis of High speed helical gear using ANSYS. In the gear design the bending stress 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. Thus, the analysis of stresses has become popular as an area of research on gears to minimize or to reduce the failures and for optimal design of gears In this paper bending and contact stresses are calculated by using analytical method as well as Finite element analysis. To estimate bending stress modified Lewis beam strength method is used. Pro-e solid modeling software is used to generate the 3-D solid model of helical gear. Ansys software package is used to analyze the bending stress. Contact stresses are calculated by using modified AGMA contact stress method. Ansys software package is used to analyze the contact stress. Finally these two methods bending and contact stress results are compared with each other.

CONTACT MECHANICS

Study of deformation of solids under contact is called contact mechanics, comprising of mechanics of material and continuum mechanics. Contact mechanics provides the information for safe and energy efficient design of mechanical elements in contact, while continuum mechanics provides for analysis of the kinematics and the mechanical behaviour of materials modeled as a continuous mass rather than as discrete particles.

Contact between two continuous, non-conforming solids is initially a point or line. Under the action of a load the solids deform and a contact area is formed as shown in Figure 1. Hertz contact stress theory allows for the prediction of the resulting contact area, contact pressure, compression of the bodies, and the induced stress in the bodies. In 1880 Heinrich Hertz developed his theory for contact stress after studying Newton's rings with two glass lenses. He became concerned about the effect of contact pressure between the two lenses and set out to analyze the effects. The result was the first satisfactory theory for contact mechanics and is still in use today.

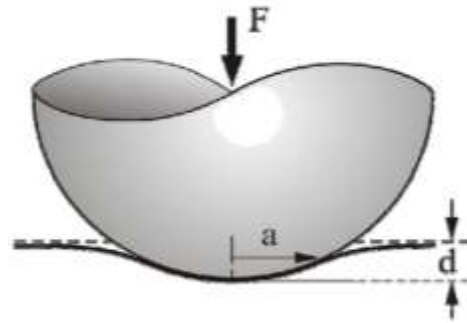


Figure 1 Depiction of contact area under applied load

In the course of developing his theory Hertz made some simplifying assumptions which are summarized as follows:

- a) Surfaces are continuous and non-conforming (i.e. initial contact is a point or a line)
- b) Strains are small
- c) Solids are elastic
- d) Surfaces are frictionless

Hertz contact stress (Involute Gear Tooth Contact Stress Analysis)

One of the main gear tooth failure is pitting which is a surface fatigue failure due to many repetition of high contact stresses occurring in the gear tooth surface while a pair of teeth is transmitting power.

Contact failure in gears is currently predicted by comparing the calculated Hertz contact stress to experimentally determined allowable values for the given material.

The method of calculating gear contact stress by Hertz's equation originally derived for contact between two cylinders.

$$\sigma_c = C_p \left[\frac{F_t}{dLI} \right]^{1/2}$$

$$C_p = \left[\frac{1}{\pi \left(\frac{1-V_1^2}{E_1} + \frac{1-V_2^2}{E_2} \right)} \right]^{1/2}$$

$$F_t = F_n * \cos \beta$$

$$I = \frac{\cos \alpha \sin \alpha m_g + 1}{2} m_g$$

The AGMA Contact stress equation is given by,

$$\sigma_H = C_p \sqrt{\frac{F_t}{bDI} \left(\frac{\cos \beta}{0.95CR} \right) K_v K_o (0.93K_m)}$$

Hertz Contact Stress

Contact pressures for two mating spur gears are calculated using hertz equation.

Those two mating gears are taken from lathe gear box as shown in Figure. The details of these gears shown below

The AGMA Contact stress equation is given by,

$$\sigma_H = C_p \sqrt{\frac{F_t}{bDI} \left(\frac{\cos \beta}{0.95CR} \right) K_v K_o (0.93K_m)}$$

Hertz Contact Equation is given by,

$$\sigma_c = C_p \sqrt{w^t K_o K_v K_s \frac{K_m C_f}{d_p F I}}$$

Hertz contact stresses for different parameters like module, power, are easily calculated by making an interface in VISUAL BASIC (VB) shown in Fig 2.

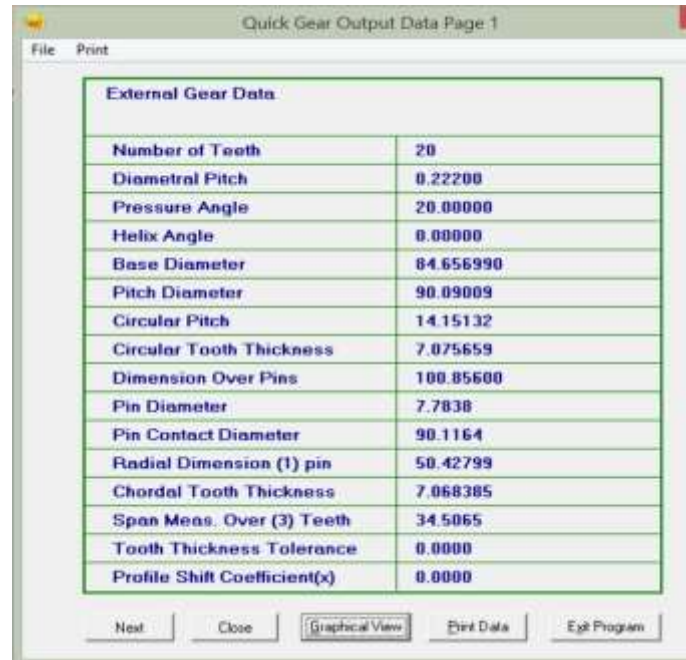
VB makes an easy way to calculate much iteration for calculating contact pressure and stresses.



Figure 2 Main Interface

DESIGN AND MODELLING OF GEAR

Specifications of gear calculated by using Quick Gear-2 Interface shown in Fig 3



External Gear Data	
Number of Teeth	20
Diametral Pitch	0.22200
Pressure Angle	20.00000
Helix Angle	0.00000
Base Diameter	84.656990
Pitch Diameter	90.09009
Circular Pitch	14.15132
Circular Tooth Thickness	7.075659
Dimension Over Pins	100.85600
Pin Diameter	7.7838
Pin Contact Diameter	90.1164
Radial Dimension (1) pin	50.42799
Chordal Tooth Thickness	7.068385
Span Meas. Over (3) Teeth	34.5065
Tooth Thickness Tolerance	0.0000
Profile Shift Coefficient(x)	0.0000

Figure 3 Trial Data used for gear construction

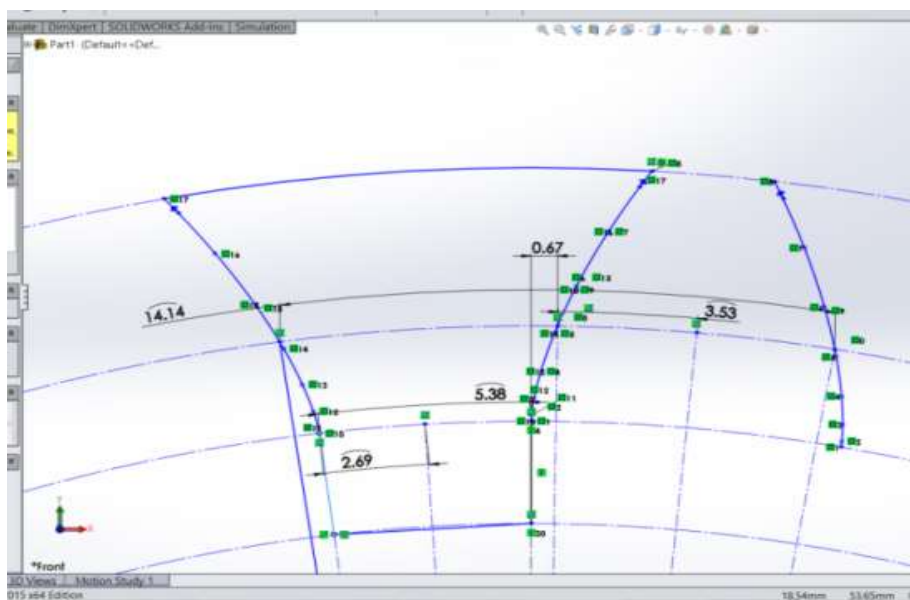


Figure 4. Involute Profile

The finite element models have been prepared for nine sets of gear pairs. They are spur, 0° , 5° and 15° helical gear pairs.

The specifications of these nine gear sets are represented in Table 1. The material used for all the gear sets was structural

steel, grey cast iron and its properties are listed in Table 2. The involute of the standard gear profile was created using a Solid Works Modeling package shown in Fig 4.

The involute standard gear profile was copied three times to create a three-tooth segment of pinion and similarly that of gear shown in Fig 5. Further, they are brought together to form a meshed model shown in Fig 6.

Gear pair volumes were discretised (meshed) using linear finite element SOLID185. The contact analysis is a

complex nonlinear problem and the use of parabolic finite elements.

At the same time, the number of nodes increases and makes the meshed model more complex and uses more computer resources for matrix formation and calculations.

The contact stress result analysis with parabolic elements type suitable for determining the deformations, stress and strain with an appropriate utilisation of time and computer resources. The meshing details of each gear pair are shown in Table 3.

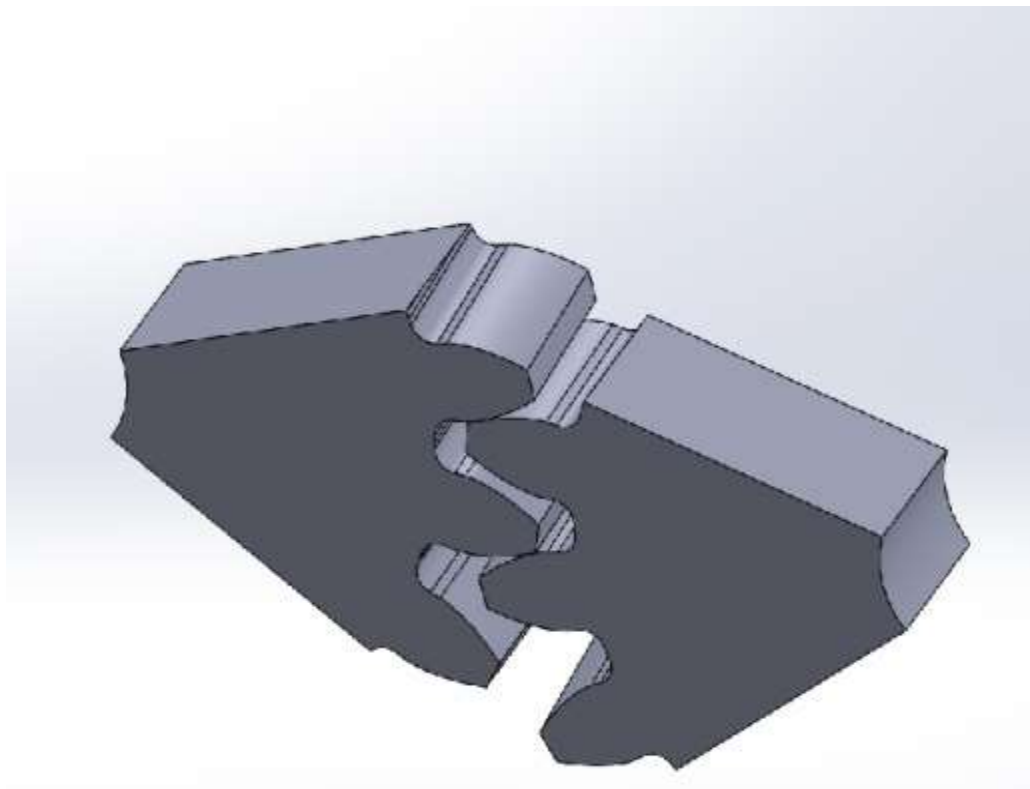


Figure 5. Final Three teeth Gear Profile assembly

Table 1: Data used for Gear Construction

Parameter	Spur gear		5° Helical gear		15° Helical gear	
P – Pinion G- Gear	P	G	P	G	P	G
No. of teeth	20	20	20	20	20	20
Normal module	4.5	4.5	4.5	4.5	4.5	4.5
Normal pressure (deg)	20	20	20	20	20	20
Helix angle (deg)	0	0	5	5	15	15
Pitch Dia (mm)	90	90	90	90	90	90
Face Width (mm)	15 mm in Case-1					
Face Width (mm)	20 mm in Case-2					
Face Width (mm)	25 mm in Case-3					
Contact ratio	1	1	1	1	1	1
Torque (N-m)	302					
Power (K-w)	31.1077					
Speed (rpm)	1000					
Contact force (N)	6601.25					

Table 2: Material Properties of all gear sets

Parameter	Material Properties Common to Pinion and Gear	
	Structural Steel	Grey Cast Iron
Young's modulus (G Pa)	210	110
Poisson's ratio	0.3	0.211
Density kg/m ³	7860	7200

Table: 3 Mesh Information of gear pairs

Helix angle (°)	Face width (mm)	Nodes	Elements
0	15	6558	1140
5	15	7813	4310
15	15	7556	4182
0	20	7550	1332
5	20	9733	5445
15	20	7556	4182
0	25	8614	1554
5	25	12648	7206

15	25	12608	7125
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The gear is considered as the master surface and the pinion as the slave. The boundary condition is represented as Fig 7. A torque is applied directly to the ends of the rigid bars that are connected with the teeth of the gear. The pinion rim is fixed, and torque is applied on the gear. The inner rim of the pinion and all the degrees of freedom in the vertical section of the pinion are

constrained, and the inner side of the rim of the gear is connected to the center of the gear by using beam elements. Also, the degrees of freedom in the center of the gear are constrained, excluding the z-directional rotation. Then a torque of 302 N m is applied to the center of the gear. This is shown in Fig 7.



Figure 6. Meshed model

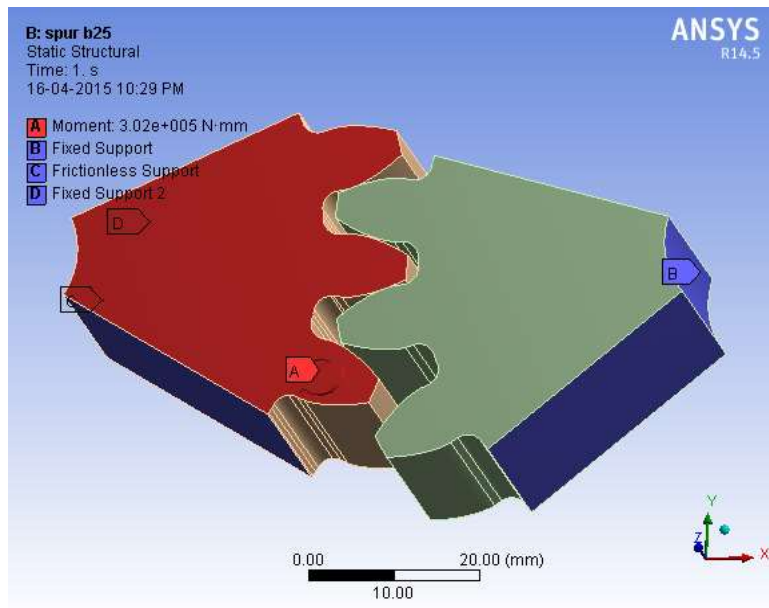


Figure 7. Boundary Conditions

RESULTS AND DISCUSSION

The contact stress for the spur gear pair of face width 20 mm was first calculated analytically by the Hertz and AGMA equations. The maximum contact stresses obtained by the Hertz and AGMA equations were 1297.65 MPa and 1410.76 MPa, respectively. Since, AGMA contact equation was derived from Hertz's theory of contact between two cylinders, they also include all the safety factors, namely the size factor K_s , overload factor K_o , Dynamic factor K_v , etc. Hence, the AGMA contact stress is higher when compared to Hertz's contact stress. Neither, the Hertz equation nor the AGMA equation includes the coefficient of friction between the contacts. The

results below represent the contact pressures acting on helical gears with helix angles 0° , 5° , 15° and face widths 15 mm, 20mm, and 25mm. The frictionless maximum contact stresses of face width 20 mm of structural steel material for the spur gear pair, 5° , and 15° helical gear pairs were 1245.7 MPa, 1242.1 MPa and 1226.5 MPa, respectively, Similar FE model is also generated for other sets of helix angles and face width. The sample contact stress distribution for spur, 5° , and 15° helical gears are presented in Figs. 8-14, respectively. The analysis outcome i.e. the maximum contact stress results for various other face widths and different materials were summarized in Table 4 and Table 5.

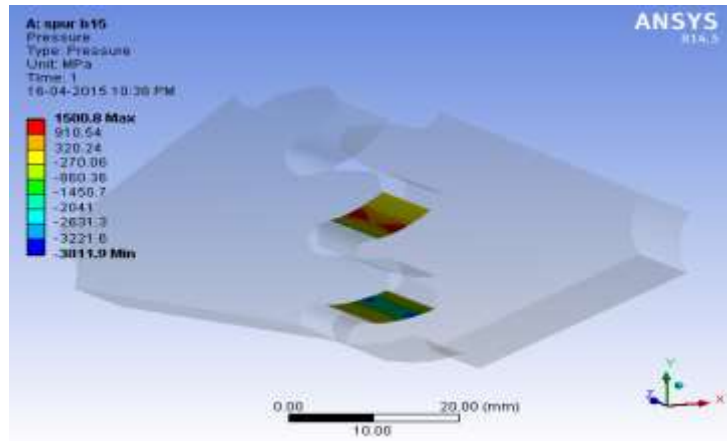


Figure 8. Sample contact stress distribution for face width 15 mm and $\beta = 0^\circ$

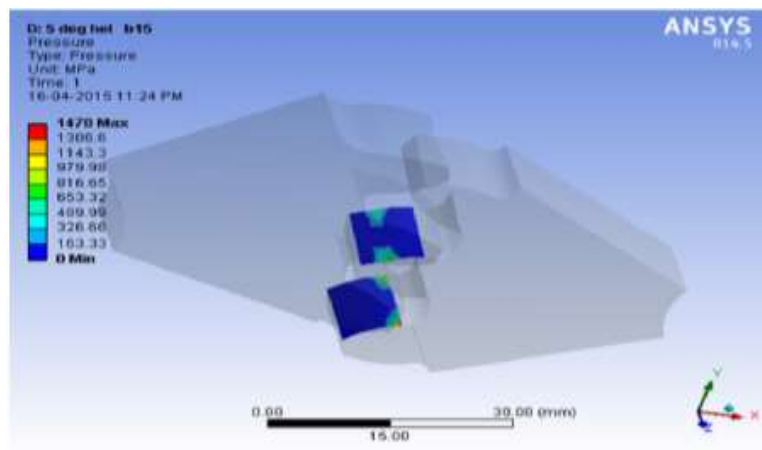


Figure 9. Sample contact stress distribution for face width 15 mm and $\beta = 5^\circ$

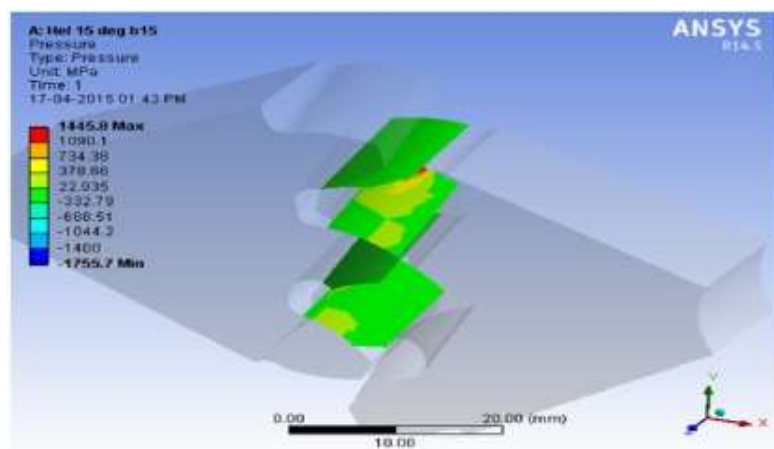


Figure 10 Sample contact stress distribution for face width 15 mm and $\beta = 15^\circ$

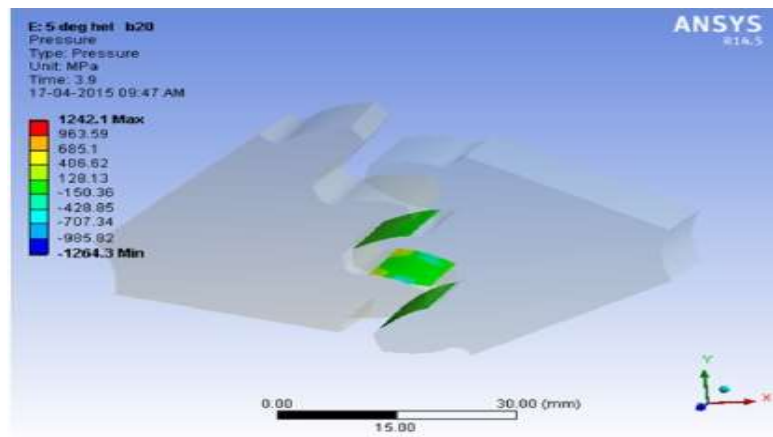


Figure 11 Sample contact stress distribution for face width 20 mm and $\beta = 0^\circ$

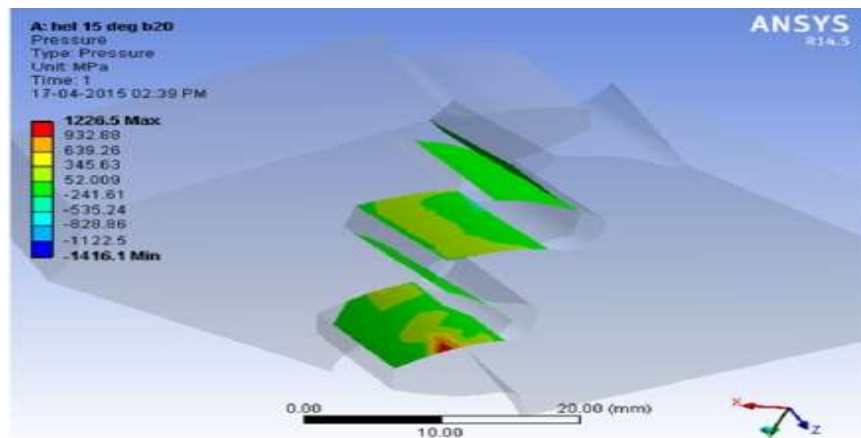


Figure 12 Sample contact stress distribution for face width 20 mm and $\beta = 5^\circ$

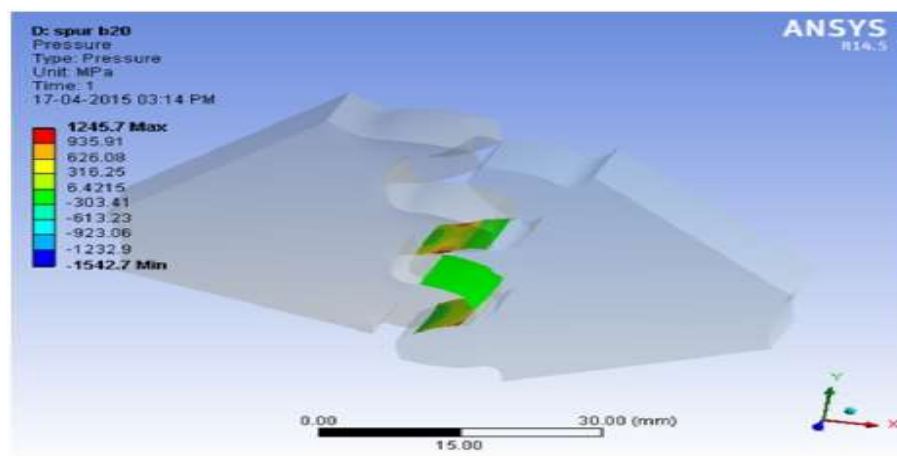


Figure 13 Sample contact stress distribution for face width 20 mm and $\beta = 15^\circ$

Table: 4 contact stress results for structural steel

Helix angle (°)	Face Width (mm)	Hertz Contact Stress (Analytical results) MPa	Hertz Contact Stress (Ansys results) MPa
0	1	1498.34	1500
5	15	1492.64	1470
15	15	1447.28	1445.8
0	20	1297.65	1245.7
5	20	1292.66	1242.1
15	20	1253.38	1226.5
0	25	1160.69	1150.3
5	25	1156.19	1142.6
15	25	1121.06	1119.3

Fig. 14, 16 shows the stress distribution plots for different gears sets at their containing face widths. From the Figure one can observe the decreasing pattern of maximum contact stress with regards to helix angle and face width. The stresses

decrease with helix angle for a constant face width. The finite element results were also compared with the analytical calculations for frictionless case. The results showed a decreasing trend, hence validating the FE model results.

Table: 5 Contact stress results for grey cast iron

Helix angle (°)	Face Width (mm)	Hertz Contact Stress (Analytical results) MPa	Hertz Contact Stress (Ansys results) MPa
0	15	1084.37	1064.7
5	15	1080.24	1067

15	15	1047.42	1027.1
0	20	939.09	959.57
5	20	935.52	940.56
15	20	907.09	916.45
0	25	839.95	831.95
5	25	836.75	839.49
15	25	811.33	827.71

Thus the finite element model generated for the spur gear pair of face width 20 mm at zero frictional coefficient give compatible result which was verified by hertz and AGMA equations.

In this present finite element model, the contact stress calculation includes coefficient of friction and was calculated.

The sample contact stress distribution for spur gear including friction coefficient 0.04, 0.1 and 0.5 are presented in Figs. 17-19. The analysis outcome i.e. the maximum contact stress results for various

frictional coefficient were summarized in Table 6. The result of spur gear set in Fig. 20 showed an increase in the maximum contact stress with the increase in coefficient of friction values.

A 2.19% increase of contact stress was recorded for spur gear set when friction coefficient was increased from 0 to 0.5.

Hence, the present 3D finite element model showed that the percentage rise of the contact stress was 2.19% more than the previous result in frictionless case.

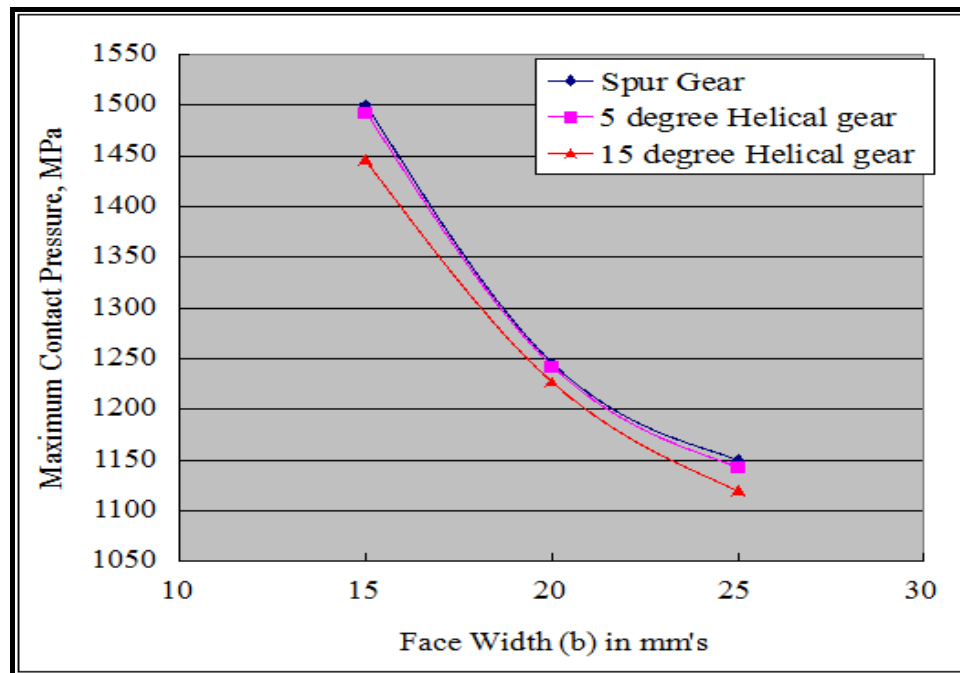


Figure 14 Maximum contact stresses for Different face width and helix angular gear sets (Structural steel)

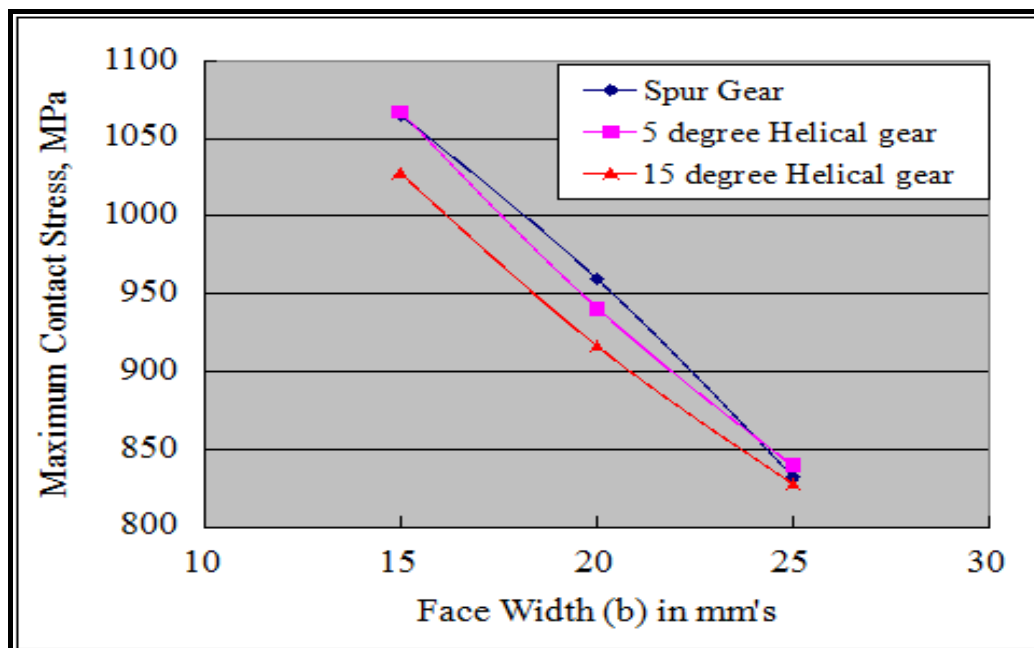


Figure 15 Maximum contact stress for Different face width and helix angular gear sets (Grey cast iron)

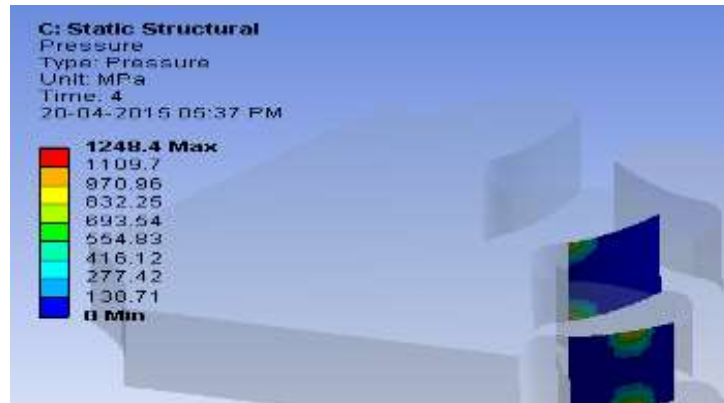


Figure 16 Sample contact stress distribution for spur gear pair with $\mu = 0.04$

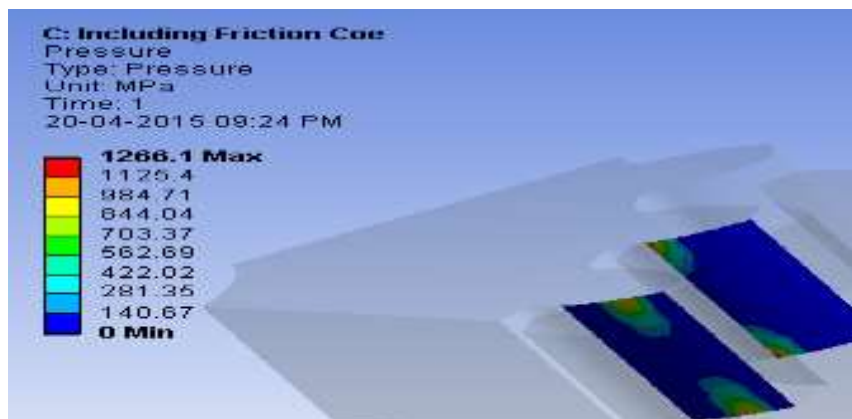


Figure 17 Sample contact stress distribution for spur gear pair with $\mu = 0.1$

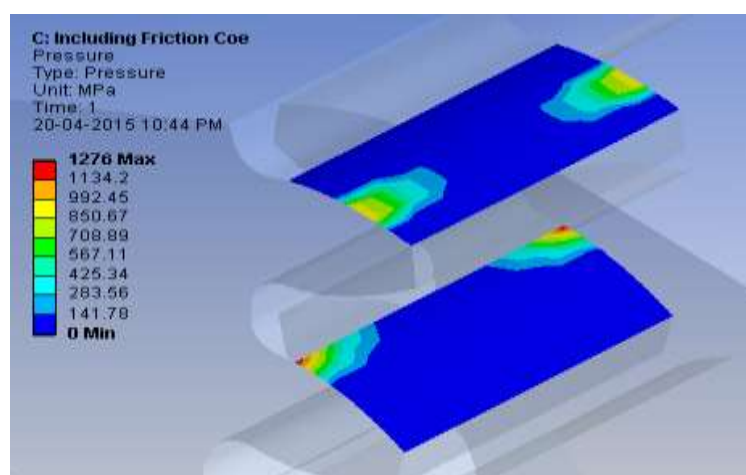


Figure 18 Sample contact stress distribution for spur gear pair with $\mu = 0.5$

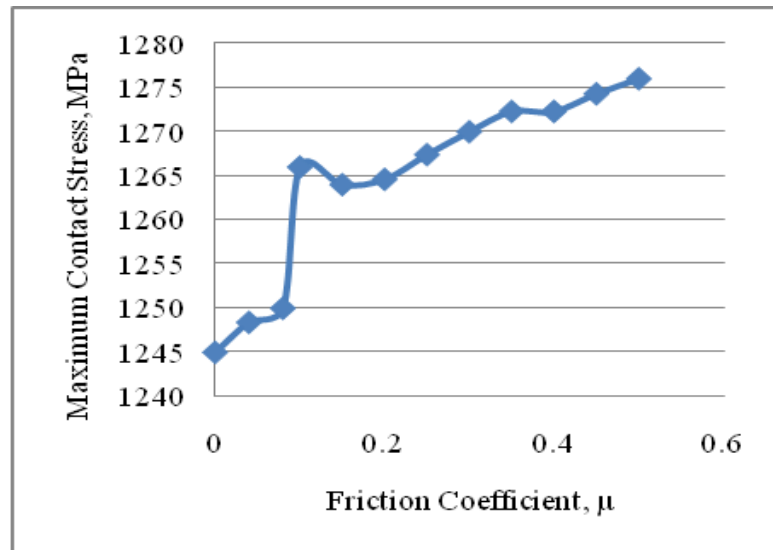


Fig. 19 Variation in Contact Stress with Frictional Coefficient

Table: 6 Contact stress results for grey cast iron

Friction Coefficient (μ)	Contact Stress, MPa
0	1245
0.04	1248.4
0.08	1250
0.1	1266
0.15	1264
0.2	1264.6
0.25	1267.4
0.3	1270
0.35	1272.3
0.4	1272.3

CONCLUSIONS

Based on the study made on the nine sets of gear pairs, the results obtained have been analysed and summarized below.

- 1) The FE models of gear pairs are compatible in evaluating the contact stresses and the results obtained are in good agreement with analytical calculations.
- 2) These results reveal that the contact stresses on the helical gear decreases with the increase in the helix angle and the face width.
- 3) For a set of Spur gear with 20 mm face width, the contact stress calculated for coefficient of friction showed that the contact stresses increased with the increasing value of static coefficient of friction. A 2.19% rise has been observed when the friction coefficient value was increased from 0 to 0.5.

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