FINITE ELEMENT ANALYSIS OF CONTACT AND BENDING STRESSES IN PAIR OF HELICAL GEAR

Himanshu Srivastava[1], Prashant Kumar Srivastava[2]

[1]M.Tech.(CAD CAM) Scholar, Mechanical Engineering department, Ambalika Institute of Management & Technology Lucknow, India

[2]Assistant Professor, Mechanical Engineering department, Ambalika Institute of Management & Technology Lucknow, India

[1]*Email-himsri1511@gmail.com* [2]*Email-pksri.hbti@gmail.com*

Abstract: *In gear design, excessive tooth contact stresses and bending stresses are one of the prime gear failure factors; therefore, its analysis is very important in order to shorten thepossibility of gear tooth failure. In the present work, an attempt is made to find the tooth bending stresses and contact stresses in a helical gear pair is calculated by using AGMA theory and finite element analysis (FEA) method. The modelling of helical gear pair is carried out in CREO and ANSYS is used for FEA.*

The stress analysis of helical gear pair is done by using AGMA theory considering different helix angle and different pressure angle. Here taking annealed AISI 9310 material for pinion and annealed AISI 8620 material for gear. It is observed that the bending stresses and contact stresses, both decreases with an increase in the helix angle if pressure angle remains constant. However, the error in the calculation by AGMA and FEA is higher for the bending stresses than the contact stresses. During analysis it's found that validation b/w AGMA and FEA for contact stresses is about 0.74% and for bending stresses 1.67% for bending stresses.

Keywords: Modelling of helical gear, Finite Element Analysis, FEA, Stresses in gears.

1. INTRODUCTION

In 27th centennial BC, the origin of Gear is able to stay traced and they may be taken as individual of the oldest pieces of apparatus proven near mankind. Referring to Chinese south-pointing Chariot in that generation, wherein wheels of chariot with movable indicator always pointed southwards no matter the way wherein it turned. Mechanical engineer Ma Jun designed the chariot with rotating wheels which became geared automatically to place the indicator in southwards path without the usage of magnets [1].

In 4th century B.C, soonest description of gear got here in technique by means of Aristotle. He described when one gear wheel drives every other gearwheel is reverse rotation. Various Greek scholars applied the use of gear within the clocks. Drawings of diverse gears which had been used inside the 3rd century BC can be observed in notebooks of Leonardo da Vinci[1].

After those research it could be found that there's massive gap no such major alternate as much as 17th century utilization of the involute curve indicated the first actual strive that provide speed ratio consistent. A rotating machinery is taken into consideration Gears have existed considering the fact that then. Gears are used for taking hefty loads because of the belongings of force-multiplying properties.

Early gears have been consisted of wood having cylindrical pegs and cogs were lubricated with the help of grease consisting animal fats. Gears have been used in water and wind wheel equipment for imparting rotational speed in lowering or increasing way to pumps and strength driven machines. A latest association of gear utilized in electricity fabric equipment is defined inside the following figure1.1. The rotational velocity of water or horse drawn wheel changed into usually too slow to use, so a hard and fast of wood gears had to be use to enhance the speed at a selected degree.

In the eighteenth century, the economic revolution in Britain saw an explosion inuse of steel gearing. Development of gear layout and manufacture hastily thru innineteenth century.

In nineteenth century, English inventor Whitworth noticed the first use of rotating cutters and form cutters in 1835, patented first the technique of gear hobbing. Many different patents followed till 1897, Herman Pfauter of Germany invented the first hobbing system that's capable of slicing both the spur and helical gears.

In twentieth century, various types of machines evolved. Andin 1975 the following main step got here when the Pfauter Company in Germany added the primary NC hobbing machine and 6 axis machines became also introduced in 1982.

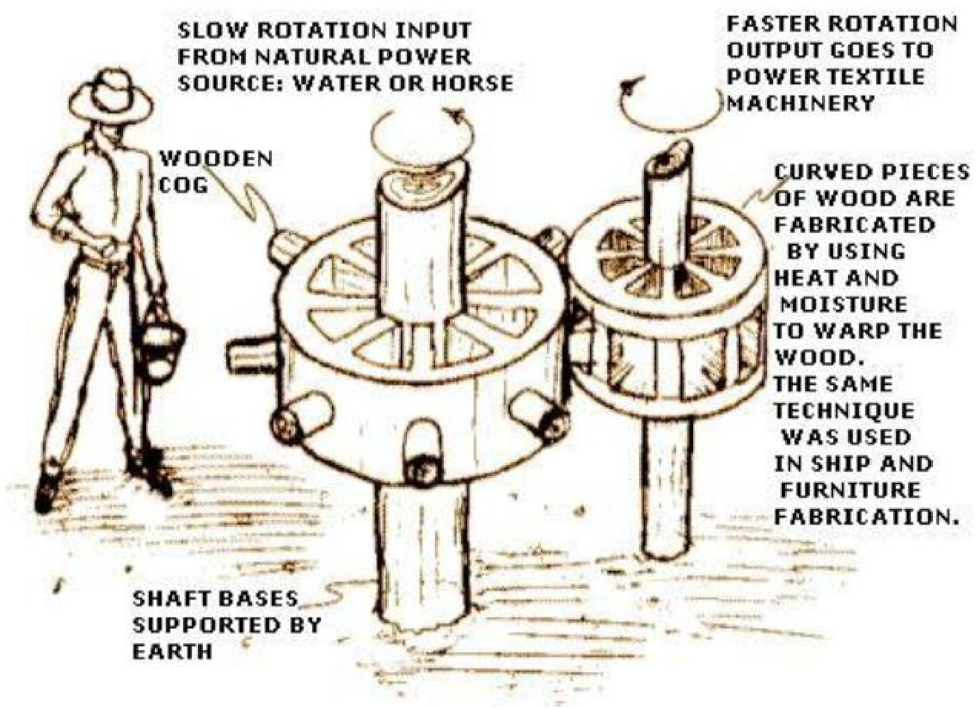

Figure 1.1: Application of Gears for Powering Textile Machinery[1]

Today, the most noteworthy new gear improvements are within the discipline of materials. Present day the useful existence of automotive and commercial gears metallurgy has enormously increased, and electronics has driven plastic gearing to new level of grease-loose reliability and calm operation.

Any Mechanical design upon the choice of types geometry and materials of the component which fulfill its beneficial needs. The design methodology needs to reduce the most critical undesirable impact and to amplify the most noteworthy desirable impact. The strategy of design optimization is achieved by whole considerations of numerous particular optimum design studies. Gears have been considered over ages andhave been effectively utilized in different machines. One such successful application has been in automobiles. They are utilized for transmission of energy from engines to wheels.

Gear is defined as a mechanical system detail that transfers power and rotating movement from one

shaft to other. The necessity for designing of multistage gear drives has been increasing with the frequent use of gear drives for small area and high-velocity application Gears are manufactured in various size ranges[1].
For enhancing the efficiency and performance of gear optimized helical gear parameters to be vital. While noteworthy research efforts have evolved for fewer constrained structures, layout for over-confined such as without lubrication, negligible vibrations, etc have not been studied as extensively. A suitable design should consider for all the constraints and should prioritize them depend on application, ease in manufacturing, cost etc. Removing the blockages and satisfying the constraints will show the results in minimization of cost with increase in effectiveness.

A. SCOPE OF THE PRESENT WORK

Maintaining the power requirement can be usually a layout challenge while reducing theweight of the aspect. The major complexity in designing of gears against fracture in high strength material is that presence of cracks/holes can modify the local stress to such an extent that the elastic stress analyses done so suspiciously by the designers are inadequate. But the other side saving of material and reducing the weight is also the prime importance for the designers while maintaining the strength of material
In the present work, an attempt has been made to calculate the contact and bending stresses of helical gear pair by using AGMA techniques and then finding the optimum set of value after analysis to check its safety. The whole work will have carried out step by step in following manner

- The theoretical analysis of contact stresses and bending stresses of a pair ofhelical gear.
- Calculation of contact stresses and bending stresses based on AGMA theory.
- Helical gear pair modelling using CREO software.
- Finite element analysis for contact and bending stresses using ANSYS software.
- The validation of analytical and FEA results.
- Analysis of contact and bending stresses for different helix angles and pressure angles.

2. LITERATURE REVIEW

In the following paragraphs, the research carried out on the effects of contact and bending stresses are as follows:

S. Vijayarangan and N. Ganeshan[3]studied, the overall performance of graphite/epoxy and Kevlar epoxy material for helical gears compared with carbon metal helical gears using 3-d FEM. After analysis they concluded that composite material for helical gears behave similarly to carbon steel for helical gears but graphite/epoxy helicalgear behavior may be very near that of carbon metallic helical gears, except for the marginally large displacements on the tip.

Ch. Rama Rao and G. Mutthuveerapan [5] studied on the geometry of helical gear through using mathematical equation beneath the weight distribution for numerous functions. They explain the geometry of helical gears via simple mathematical equations, the load distribution for numerous positions of the contact line and the stress analysis of helical gears the use of the three-dimensional finite element method. A computer application has been evolved for the stress analysis of the gears. In this paper loading aspects is used for geometrical consideration.

J. Lu, F. L. Litvin and J. S. Chen [4] studied the gear contact and stresses for double circular-arc helical gear drives. The proposed technique depends on use of automatic simulation of meshing and get in contact with stacked gear drives, and the finite element technique. Load proportion between the neighboring pairs of tooth is based totally on the evaluation of function errors resulting from surface mismatch and elastic deformation of tooth.

Anders Flodin, Soren Andersong[5] studied on wear prediction of helical equipment. The author determines the wear simulation in rolling and sliding contact. They use simplified put on version for spur and helical tools. During calculation they observed that over teeth surface pressure is redistributed due to wear and grow to be unevenly dispensed. They locate that sliding distance is most important element in put on fee.

Osman Asi [8] studied on failure evaluation of helical equipment utilized in bus gearbox. They used AISI 8620 metal as gear material. They study failure sector through the prevent of scanning electron microscope together with EDX facility. After analysis they found simplified wear simulation approach for helical gears were developed. Its consciousness is how the influence of simplification inside the simulation effect the wear.

K. Mao [9] give attention to the gear fatigue put on reduction via micro- geometry modification technique. An accurate non-linear finite element approach could be hired to provide a quantitative knowledge of gear tooth contact behavior. A research has been done on automobile transmission gear surface failure because of shaft misalignment and meeting deformations. The solution for the wear is based totally on equipment micro-geometry amendment approach.

Juha Hedland and ArtoLehtovarra [10] studied awareness on modelling of helical equipment contact with device deflection. The surface profile of helical gear constructed via gear tool geometry by using hobbing technique. The version is primarily based on avoid massive meshes. In this paper version is three-D Finite element. In this paper pseudo interface method is used. The method is primarily based on wide set of numerical calculation. The model is tested towards Hertz system in case of circular and elliptical contact. In this paper mesh length and form effect the end result.

B. Venkatesh, V. Kamala and A. M. K. Prasad [11] studied focus on decreasestress and deflection for secure function of engine. For exceptional cloth stress and deflection is analyzed. The end result is examined between theoretical analysis and finite element analysis. For marine application venture contain design modelling and manufacturing of helical equipment.

R. Devraj [14] study on gear failure causes in line with him bending stress and make contact with pressure is the of equipment failure. According to him contact stress tooth failure is more than bending stress. They say that to minimize failure stress evaluation is key vicinity. In this paper author makes use of ANSYS 12.0 for the Finite Element Analysis and 3-D version software.

B. Venkatesh et.al [15] have a look at on gear parameter combined impact on various pressure inclusive of Tangential Force and Dynamic teeth load. In this paper they use Buckingham equation and foremost layout for the tools. They want to find module impact on dynamic tooth load and tangential pressure. MATLAB application is used for helical gear numerical analysis. Author used metal alloy fabric for the helical gear and want locate numerous gear parameter impact which include helix angle, face width, tooth ratio and module on dynamic tooth load. After evaluation they find, face width, module and helix angle are constant. Dynamic tooth load also constant when tooth ratio is changed. But effect of helix angle tooth ratio module and face width is considered for tangential pressure.

3. ANALYSIS OF HELICAL GEAR

To investigate the effects of helix angle on the bending stresses and contact stresses in helical gear pair based on the AGMA (American Gear Manufacturing Association) theory and Finite Element Analysis (FEA).

Input Data

The following input has been considered for the contact stress of helical gear pair

- Power to be transmitted(P) = 20 kW
- Pinion Speed = 1440rpm
- Gear Speed = 360rpm,
- Normal Pressure angle ($\emptyset_n$) = 20
- Helix angle (Ψ)= 30
- Material is Ductile Iron
- Normal Module (m_n) - 4 mm

Design Variables

The parameters which affect the objective function are called variables. For helical gear volume minimization, the module, number of teeth on pinion/gear, face width, and helix angle are important variables

4. FEM ANALYSIS

For modeling of helical gear pair use the optimum set of parameters. After drawn the model of helical gear pair stress analysis is performed through the ANSYS for its safe design. The stress generated at the contact of helical gear pair should always be less than the material permissible stress. The gear tooth profile can be outstanding as involute and trochoid fillet curves. First of all, it's miles vital to encompass the gear tooth that has the involute equipment teeth profile. If the fundamental specs such as the stress attitude, module, wide variety of enamel and shift coefficient are given, the involute gear tooth profile can be embodied the usage of the gear specification and involute characteristic

A. MODELLING OF HELICAL GEAR PAIR

Modeling is a group of standards used for mathematical and pc modeling of 3 dimensional solids. Solid modeling is famous from related areas of geometric modelling and laptop pictures through its importance on its physical reliability. Together, the concepts of geometric and strong modeling form the muse of laptop aided layout (CAD) and in preferred help the creation, visualization, simulation, interrogation and annotation of virtual modes of bodily objects.

The following family members are used for the duration of the modeling of helical equipment and pinion

$$Transverse\ module\ (m) = normal\ module\ (m_n)/\cos\ (helix\ angle) \tag{4.1}$$

$$Normal\ circular\ pitch = pi \times normal\ module \tag{4.2}$$

$$Transverse \text{ circular pitch} = pi \times transverse\ module \tag{4.3}$$

$$Transverse\ pressure\ angle = tan^{-1}\left(\frac{tan(normal\ pressure\ angle)}{cos(helix\ angle)}\right) \tag{4.4}$$

$$Outer\ diameter = pitch\ diameter + 2 \times (normal\ module) \tag{4.5}$$

$$Root\ diameter = pitch\ diameter - (2 \times 1.25 \times normal\ module) \tag{4.6}$$

$$Base\ diameter = pitch\ diameter \times cos(transverse\ pressure\ angle) \tag{4.7}$$

$$Normal\ tooth\ thickness = (\pi \times normal\ module)/2 \tag{4.8}$$

$$Transverse\ tooth\ thickness = (\pi \times transverse\ module)/2 \tag{4.9}$$

$$Lead = \pi \times pitch\ diameter/\ tan\ (helix\ angle) \tag{4.10}$$

3D version of helical gear pair with involute profile using the equation attract
CREO 4.0 designing software. Basic designing software CREO 4.0 always choice to draw a tool have to start with basics: the involute curve. An involute is defined as the direction of a factor on a immediately line known as the generatrix, as it rolls along the convex base curve (the involute). The involute curve is typically used as the idea for theprofile of a gear tooth.

Several techniques can be used to create the involute teeth profile of helical gearin CREO. In the existing work, modeling of helical gear pair is executed the use of the involute curves equation as given below. This approach generates the accurate involute curve profile based on genuine geometric equations. The generated profile of helical equipment pair can be rather bendy. It also allows the user to pick cylindrical or Cartesian coordinate systems for the era of involute teeth profile.

$$Base\ radius = Base\ diameter/2 \tag{4.11}$$

$$Angle = t \times 90 \tag{4.12}$$

$$Cir\_len = (\pi \times base\ radius \times t)/2 \tag{4.13}$$

$$X\_PNT = Base\ radius \times \cos(Angle) \tag{4.14}$$

$$Y\_PNT = Base\ radius \times \sin(Angle) \tag{4.15}$$

$$X = X\_PNT + (Cir\_len \times \sin(Angle)) \tag{4.16}$$

$$Y = Y\_PNT + (Cir\_len \times \cos(Angle)) \tag{4.17}$$

$$Z = 0 \tag{4.18}$$

Procedures to Create helical gear and pinion with an Involute tooth profile

The above teeth geometries have been first modelled the usage of Pro/E after which later analyzed.

Initially, modelling of the equipment changed into performed in Pro/E. The model become completed by means of sketching the bottom circle using members of the family and parameters and after the extrude part is generated the curve is created and the sweep alternative is accomplished to acquire the teeth profile.

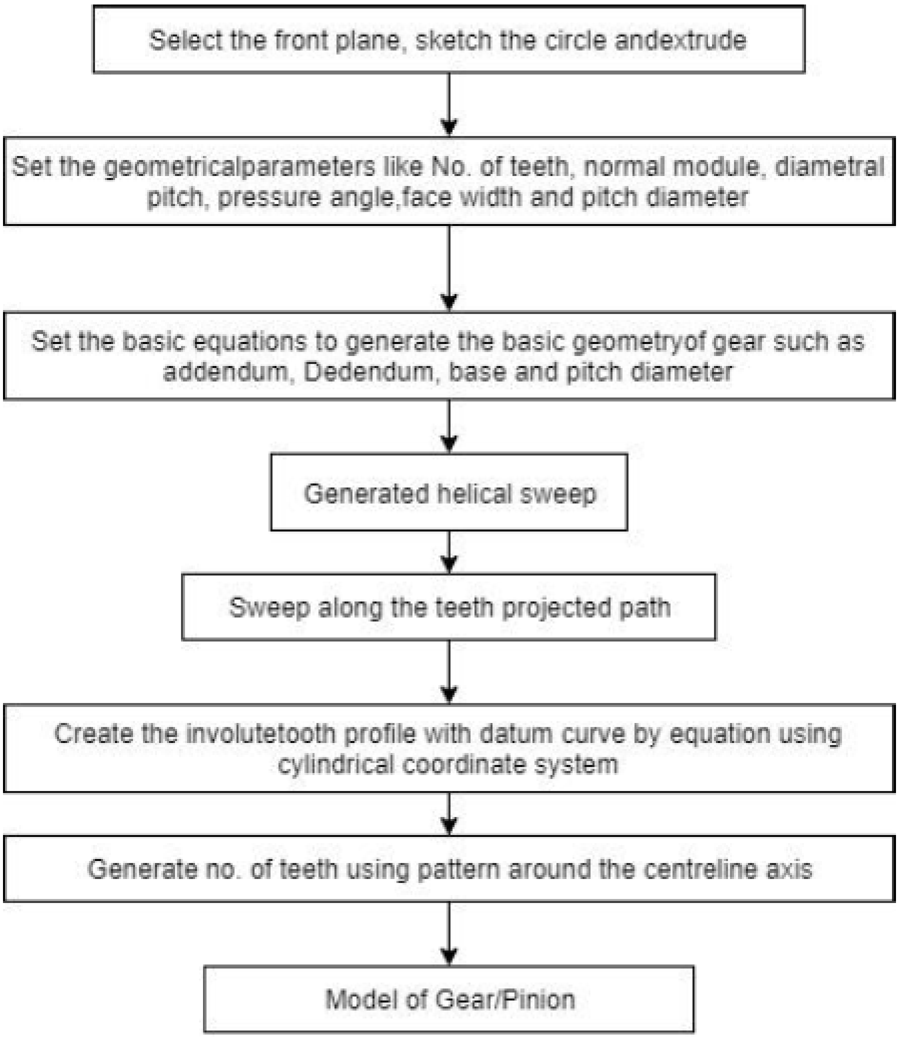

Figure 4.1: Steps to assemble the helical gear and pinion

a. Modeling of helical pinion

Geometrical parameters for pinion- these are the geometrical parameters which are usedto draw the model of pinion.
No. of Teeth –18,
Normal Module -4 mm,
Pitch Diameter – 83.2 mm,
Normal Pressure Angle – 20°,
Helix Angle – 30°
Width – 30.16 mm, & Side - Right hand helix

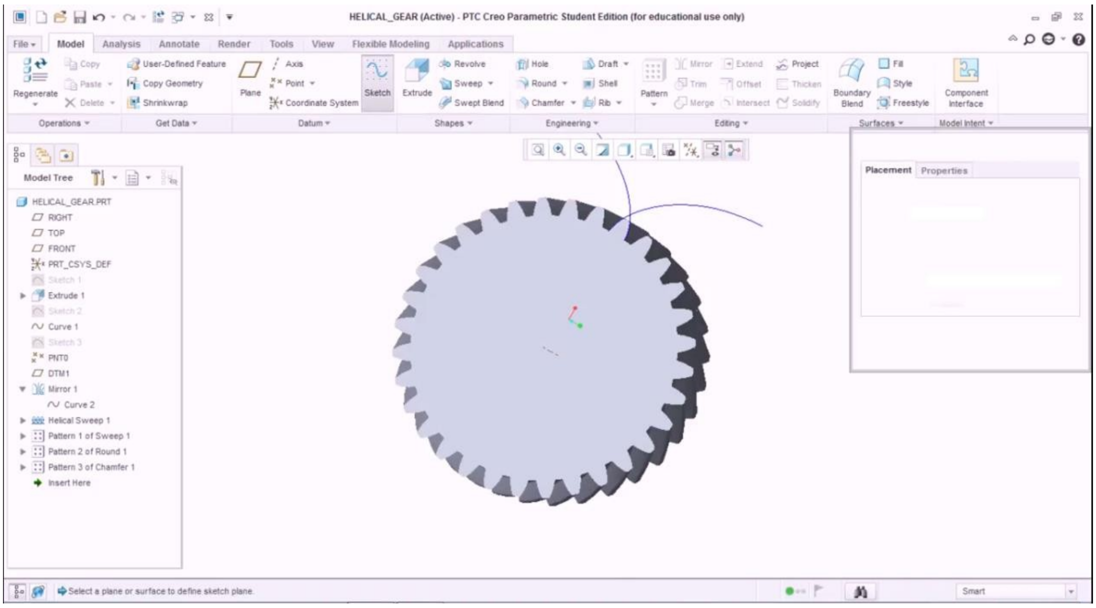

Figure 4.2: Model of helical pinion

b. Modeling of helical Gear

The following geometrical parameters are used for modeling the helical gear
No. of Teeth – 72,
Normal Module -4 mm,
Pitch Diameter – 332.64 mm,
Normal Pressure Angle – 19°,
Helix Angle – 29°
Width – 30.16 mm, & Side - Left hand helix
.

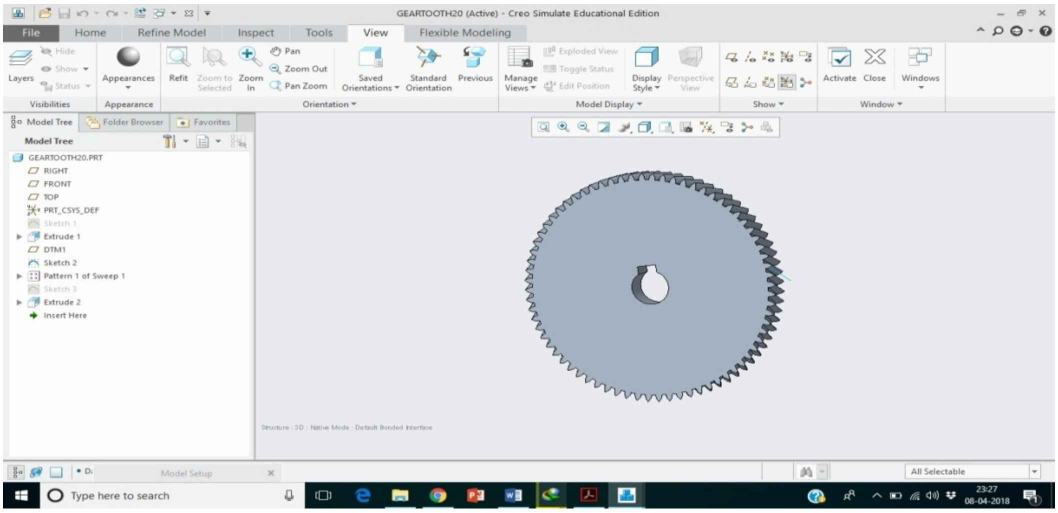

Figure 4.3: Model of helical gear

c. Modeling of helical gear pair

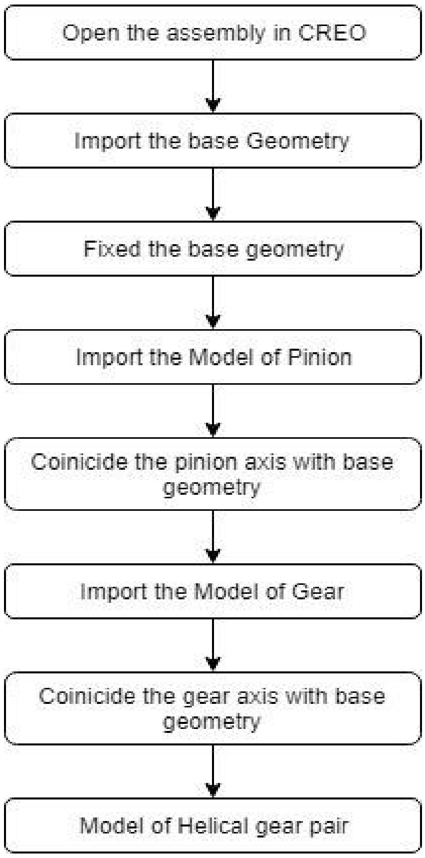

Figure 4.4: Steps for modelling of helical gear and pinion

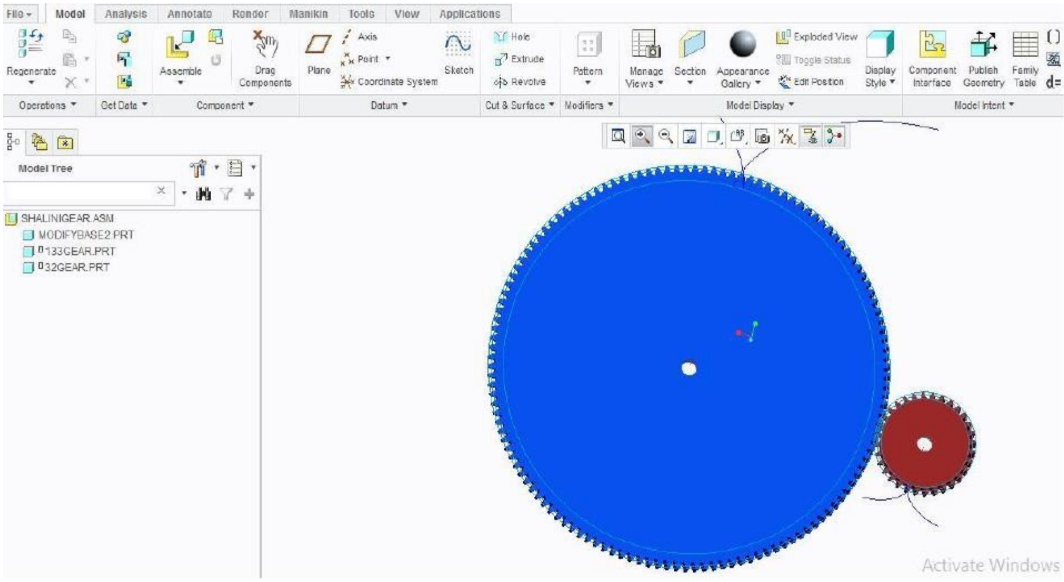

Figure 4.5: Model of helical gear pair

B. FINITE ELEMENT ANALYSIS OF HELICAL GEAR PAIR

ANSYS workbench is used for the Finite element analysis. The step file of helical gearpair, modeled in CREO software is generated and imported into the ANSYS workbenchsoftware and meshing is dome in the ANSYS. The material properties is given to thewhich is imported from ANSYS material library.

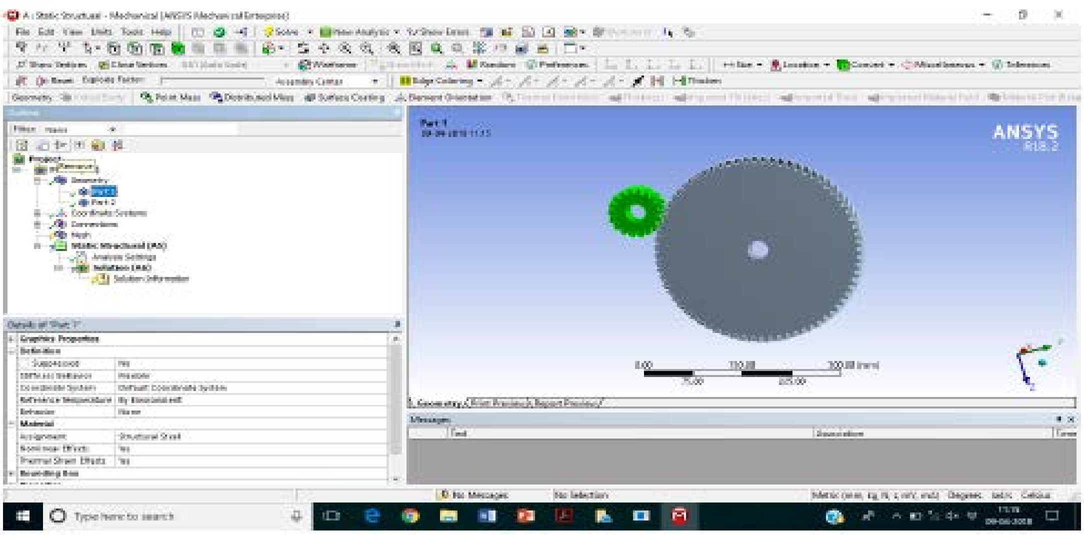

Figure 4.6: Import the geometry in ANSYS

a. Meshing of helical gear pair

The next stage is to generate the mesh in 3D helical gear pair model. The meshing of helical gear pair is shown in fig. through the 20-noded hexahedral elements (373709). Mesh convergence is a partially automated for CAD-based, auto meshed models and forstatic stress with linear model analysis.
Element shape

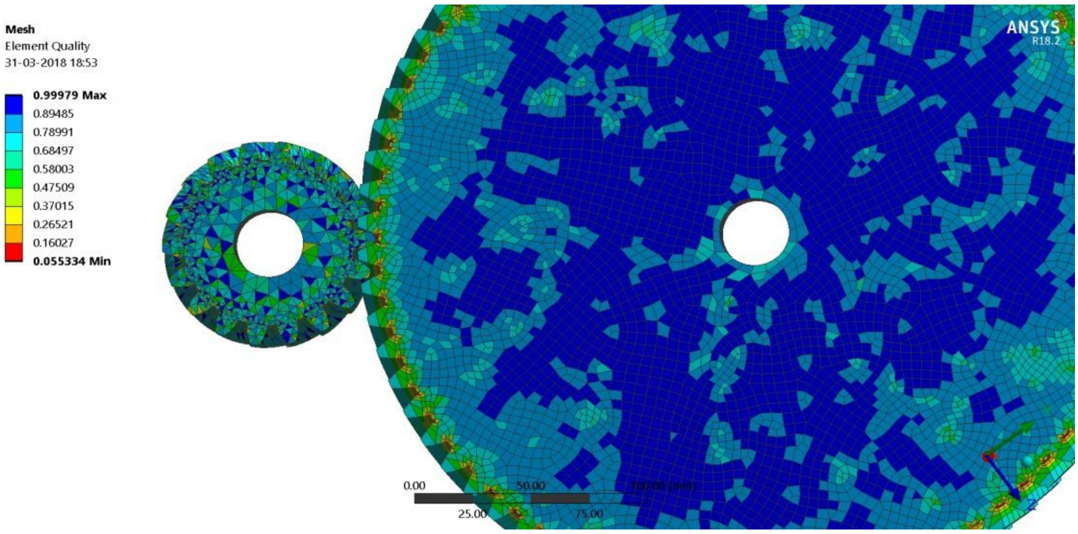

Figure 4.7: Meshing of helical gear pair

Mess convergence

Finite element meshes are generated using the hexahedral elements with various element lengths from 10 mm to 5 mm in a step of 1 mm. The equivalent stress, which stress calculated exceed the limit are checked for convergence at every location. At the mess size of 5 mm and 6 mm, the stress value shows error less than 0.1 %. So, the convergence has been achieved for entire range of elemental length.

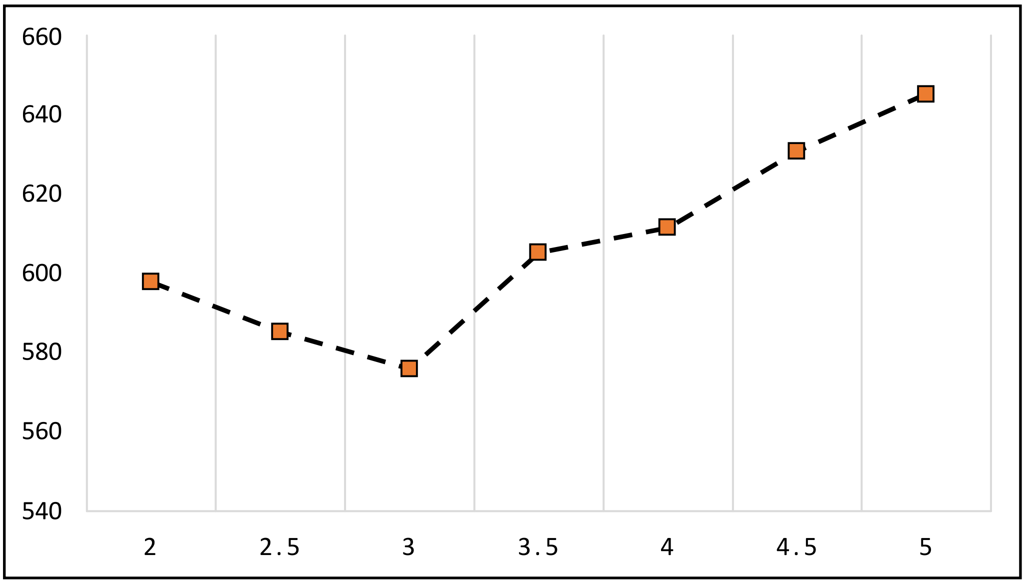

Figure 4.8: Mess convergence

Boundary conditions

For analysis boundary condition play very vital role. After the creation of mesh apply the boundary condition to evaluate the stress. Helical gears pair analysis can be done by fixed support in one gear and apply the frictionless support and moment on another gear.

We know that the gear and pinion made of same material. So, the design is based onpinion.

Velocity**,**

Velocity,

$$v = \frac{\pi d n}{60}$$

Speed of pinion,

$$n_p = 1440\ rpm$$

Tangential moment,

$$\boldsymbol{m_t} = \frac{20(KW) \times 10^6 \times 60}{2 \times \pi \times 1440}$$

$$m_t = 1.3269 \times 10^5 \text{N.mm}$$

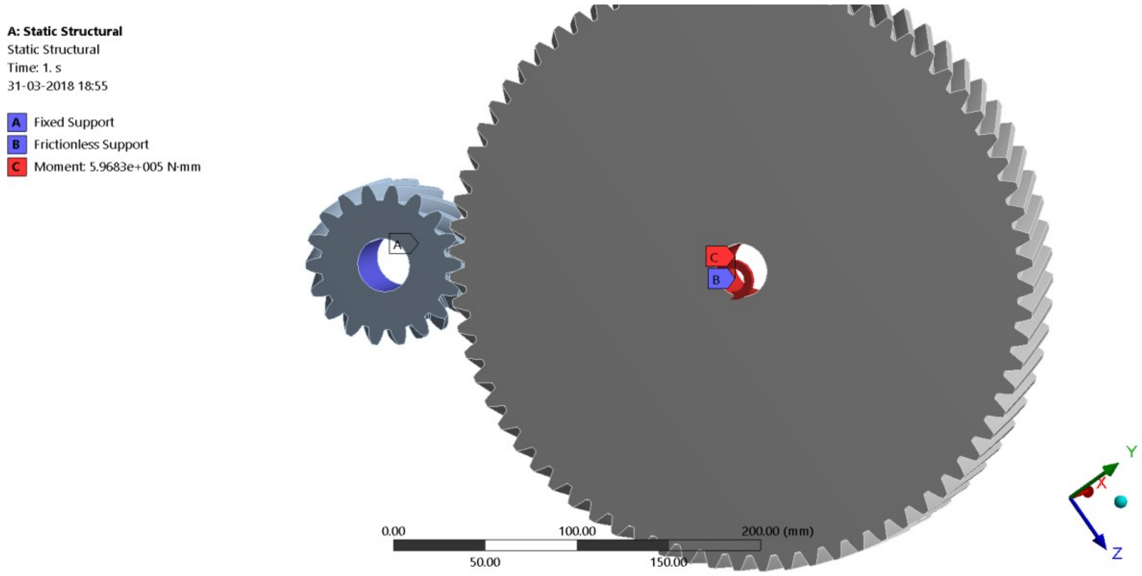

Figure 4.9: Boundary conditions

5. RESULT AND DISCUSSION

Bending stresses and contact stresses are considered as vital role to achieve an efficient, compact and high power transmitting helical gear pair. The Von-mises stress is calculated on the basis of distortion energy theory of failure. The bending stresses and contact stresses are calculated for specific values of helix angles (15˚, 20˚, 25˚and 30˚) with exclusive pressure angle (14.5˚, 20˚).

Both bending stresses and contact stresses in helical gear pair depends on the helix angle, material and face width of gear. In the present work, the bending stresses and contact stresses are calculated using AGMA theory and FEA. Both stress decreases with an increase in the helix angle. Hence result from FEA is different than the allowable stress. The value of contact stress is depending upon both helix angle and pressure angle. The error in bending stresses between AGMA and FEA is maximum (16.4%) for 15^o helix angle; however, it is minimum (14.33%) for 30^o helix angle and the contact stresses also decrease with an increase within the helix angle however the error among AGMA and FEA could be very small (1.2%) for 25˚ helix angle maintaining the pressure angle fixed at 14.5˚ however the errors among AGMA and FEA could be very small (0.74%) for 25˚ helix angles maintaining pressure angle 20˚.

A. Bending Stress Analysis

The bending stresses are calculated for different values of helix angles (15˚, 20˚, 25˚and 30˚). It can be observed that the bending stresses decreases gradually with an increase in helix angle. The error in bending stresses between AGMA and FEA is maximum (16.367%) for 15^o helix angle; however, it is minimum (14.33%) for 30^o helix angle. Fig. 5.1 shows the variation of bending stresses at different helix angles, keeping the pressure angle fixed at 20^o.

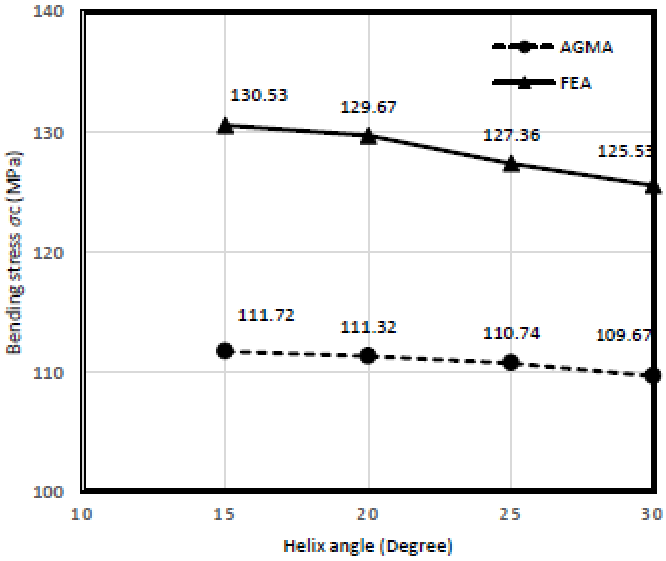

Figure 5.1: Effect of helix angle on bending stresses

B. Contact Stress Analysis

Contact stresses are calculated for different values of helix angles (15˚, 19˚, 25˚and 29˚). Fig. 5.2 shows the variation of contact stress at different helix angles, keeping thepressure angle fixed at 14.5^o and Fig. 5.3 shows the variation of contact stress at different helix angles, observance the pressure angle fixed at 19^o. The contact stresses also decrease with an increase in the helix angle but the error between AGMA and FEA is very small 1.37% for 29^{o}helix angle and 1.85% for 15^0 helix angle observance pressure angle 14.5^0 and 0.74% for 29^0 helix angle observance pressure angle 19^0. For 14.5^0 pressure angle value of von-mises stress at 29^{0}helix angles 601.89 MPa and at 15^0 helix angles 664.41 MPa and for 19^0 pressure angle value of Von-Mises stress for 29^0 helix angle 585.25 MPa and at 15^0 helix angle 631.86 MPa.

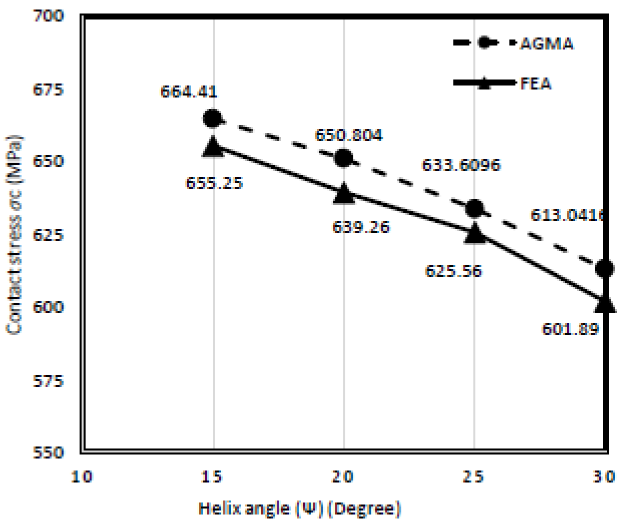

Figure 5.2: Effect of helix angle on contact stresses for 14.5˚ pressure angle

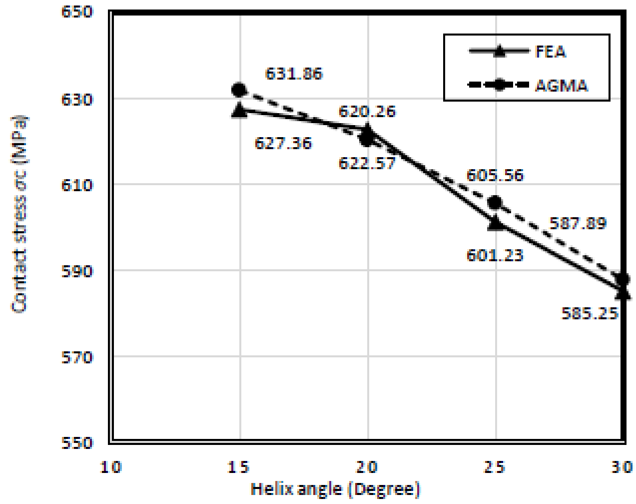

Figure 5.3: Effect of helix angle on contact stresses for 20˚ pressure angle

C. FEA Result

Contact and bending stresses are considered as critical position to gain a compact efficient and excessive power transmitting helical gear pair. Basis of distortion strength theory of failure consider for Von mises stress calculation. However, the designs are carried out for contact and bending stresses

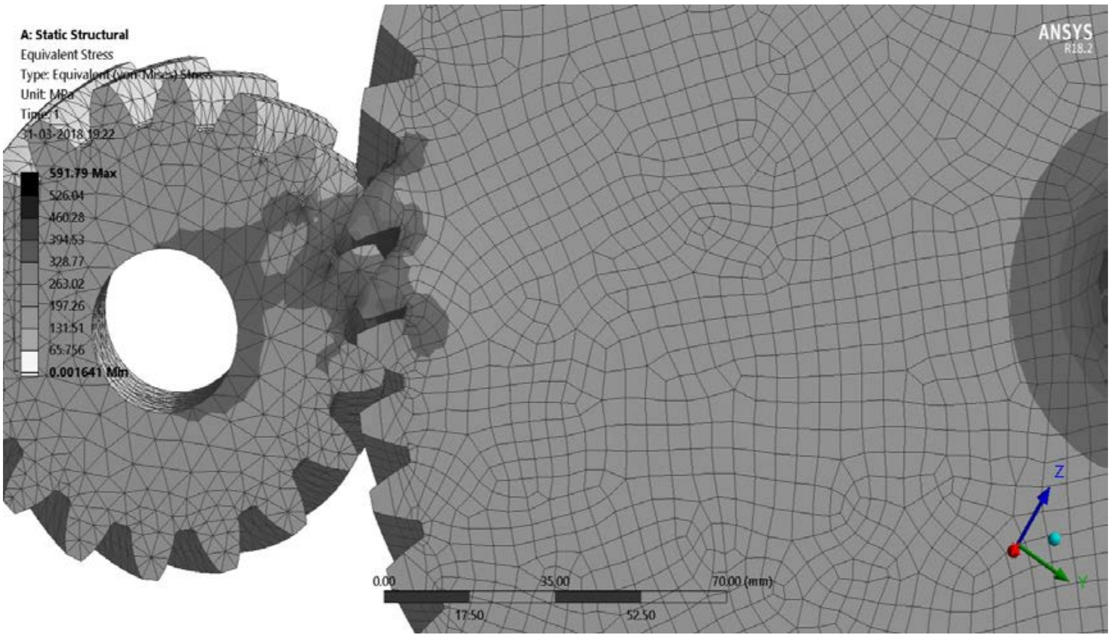

Fig. 5.4: Stress analysis

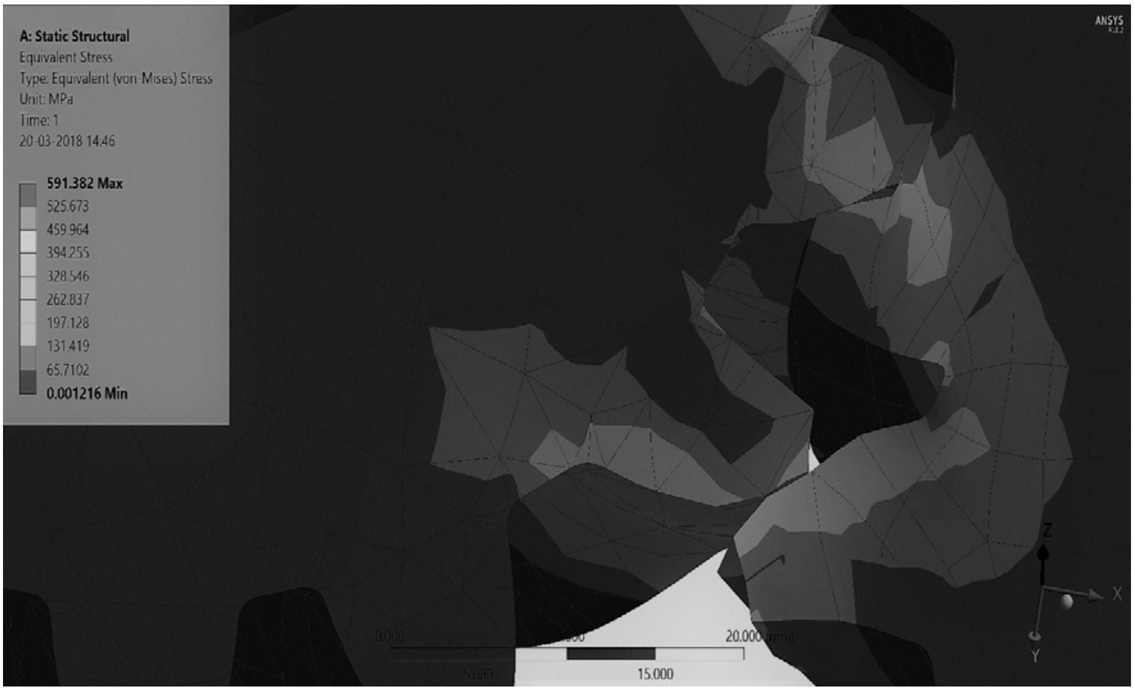

Fig. 5.5: Maximum stress region

6. CONCLUSION

The strength of helical gear is an important parameter for its design as it decides the force and power to be transmitted. If various factors like pressure angle, helix angle, contact ratio, module, face width, are done considering their combined effects, it will certainly enhance the performance and effectiveness of helical gear.

Both bending stresses and contact stresses in helical gear pair depends on the angle, material and countenance width of gear. In my work, the flexible stresses and contact stresses are calculated using AGMA. Both stress decreases with an increase in the angle. It is concluded that the error in the estimation of contact stresses using AGMA theory and FEA is approximately 1.67%, keeping pressure angle at 19^o

A. Effect on Bending Stress

Bending Stress analysis is done for different helix angle and different pressure angle using AGMA method. A validation of result made between AGMA value and FEA value. During analysis following point is find

- The value of bending stress depends upon the helix angle, it means bending stress decrease with increase of helix angle.
- Bending stress for gear is not much affected by pressure angle because when we change value of pressure angle there is minor change in bending stress which is less than 0.1%
- The error in bending stresses between AGMA and FEA is maximum (16.367%) for 15^o helix angle; however, it is minimum (14.33%) for 30^o helix angle keeping the pressure angle fixed at 20^o.

B. Effect on contact stress

For the analysis of helical gear contact stress using AGMA method for different helix angle and different pressure angle. The value of contact stress depends on both helix angle and pressure angle. The value of stress decrease with increase of helix angle and pressure angle.

- The value of contact stress depends upon the helix angle, it means bending stress decrease with increase of helix angle.
- Contact stress for gear affected by pressure angle because it means bending stress decrease with increase of pressure angle.
- At 20˚ pressure angle the value of contact stress using AGMA theory for 30˚ helix angle is 587.87 MPa and for 15˚ helix angle 631.86 and at 14.5˚ pressure angle the value of contact using AGMA theory for 30˚ helix angle is 664.42 MPa and for 15˚ helix angle is 613.04 MPa.
- The contact stresses also decrease with an increase in the helix angle but the error between AGMA and FEA is very small 1.37% for 30^{o}helix angle and 1.85% for 15^{0} helix angle keeping pressure angle 14.5^{0} and 0.74% for 30^{0} helix angle keeping pressure angle 20^{0}

REFERNECES

[1] A. Flodin and S. Andersson, "A simplified model for wear prediction in helical gears," *Elsevier Science*, p. 285–292, 2001.

[2] M. G. Khosroshahi and A. M. Fattahi, "Three Dimensional Stress Analysis of a Helical Gear Drive with Finite Element Method," *MECHANIKA*, vol. 23, no. 5, pp. 630-638, 2017.

[3] S.-C. Hwang, J.-H. Lee, D.-H. Lee, S.-H. Han and K.-H. Lee, "Contact stress analysis for a pair of mating gears," *Mathematical and Computer Modelling*, p. 40– 49, 2013.

[4] S. S. Patila, S. Karuppanan, I. Atanasovska and A. A. Wahab, "Contact Stress Analysis of Helical Gear Pairs, Including," *International Journal of Mechanical Sciences*, 2014.

[5] J. Venkatesh and P. B. G. S. N. Murthy, "Design and Structural Analysis of High Speed Helical Gear Using," *International Journal of Engineering Research and Applications*, vol. 4, no. 3, pp. 01-05, 2014.

[6] P. R. P. M. M. S.Sai Anusha, "Contact Stress Analysis of Helical Gear by Using AGMA and ANSYS," *International Journal of Science Engineering and Advance Technology*, vol. 2, no. 12, pp. 1012-1016, 2014.

[7] A. S. Achar, R. P. Chaitanya and S. Prabhu, "A Comparison of Bending Stress and Contact Stress of a Helical Gear as Calculated by AGMA Standards and FEA," *International Journal of Emerging Technology and Advanced Engineering*, vol. 4, no. 5, pp. 38-43, 2014.

[8] H. H. Htet San, H. H. Win and M. Thein, "Design and Contact Stress Analysi of Helical Gear for Light-Weight Car," *International Journal of Mechanical AndProduction Engineering,* vol. 5, no. 8, pp. 7-12, 2017.

[9] R. Devraj, "Contact Stress Analysis of a Helical Gear," *Applied Mechanics and Materials,* Vols. Vols. 766-767,, no. ISSN: 1662-7482, pp. 1070-1075, 2015.

[10] K. Naresh and C. Chandrudhu, "Design And Analysis Of Helical Gear," *International Journal Of Professional Engineering Studies,* vol. 6, no. 4, pp. 194- 203, 2016.

[11] B. Venkatesh, V. Kamala and A. Prasad, "Design, Modelling and Manufacturing of Helical Gear," *INTERNATIONAL JOURNAL OF APPLIED ENGINEERING RESEARCH,* vol. 1, no. ISSN 0976-4259, pp. 103-114, 2010.

[12] J. Hedlund and A. Lethovaara, "Modeling of helical gear contact with tooth deflection," *Tribology International 40,* p. 613–619, 2007.

[13] I. Atanasovaska, V. N. Stanojlovic, D. Dimitrijevic and D. Momcilovic, "Finite Element Model For Stress Analysis And Nonlinear Contact Analysis Of Helical Gears," *Scientific Technical Review,* pp. 61-69, 2009.

[14] O. Asi, "Fatigue failure of a helical gear in a gearbox," *Engineering Failure Analysis,* p. 1116–1125, 2006.

[15] K. Mao, "Gear tooth contact analysis and its application," *Wear 262,* p. 1281–1288, 25 July 2006.