

# Investigation of Effect of Helix Angle and Pressure Angle on Bending Stress of Helical Gear

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**Abstract**— Ad-hoc mobile networking is a active research area. They can be used in agriculture for monitoring and control of environment parameters in the form of wireless sensor network. The aim of this project is performance evaluation of protocol of this Ad-hoc network. The performance evaluation is done by normal AODV protocol. Quos stands for Quality of Service. The research proposed some enhancement to the AODV protocol for Rural & Urban Area to provide Quos. This project focuses on different parameters namely Throughput, Good put, Instantaneous Delay, Packet Delivery Ratio, Packet Loss. Performance of the AODV Protocol for Quos is Evaluated with respect to above parameters using network simulator 3 without downgrading its main functionality & generated graphs for Rural, Urban area using APP Tool Master Kit.

**Keywords:** Ad Hoc Networks, AODV, Quality of Service, NS3

## I. INTRODUCTION

Gears are the most common means used for power transmission. They can be applied between two shafts which are Parallel, Collinear, Perpendicular and intersecting, Perpendicular and nonintersecting, Inclined at any arbitrary angle.

Gears are made to high precision Purchased from gear manufacturers rather than made in house however it is necessary to design for a specific application so that proper selection can be made. Used to be called toothed wheels dating back to 2600 b.c.

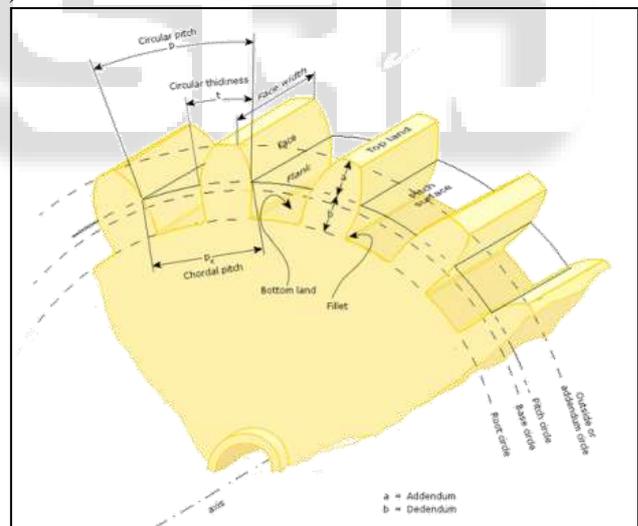
### A. Gear terminology:

- 1) Clearance (c): is the addendum minus is the addendum minus dedendum. ,,
- 2) Working depth: Working depth: is the distance that one tooth of a is the distance that one tooth of a meshing gear penetrates into the tooth space.
- 3) Base circle: is an imaginary circle about which is an imaginary circle about which the tooth the tooth involutes profile is developed.
- 4) Fillet: is the radius that occurs where the flank of is the radius that occurs where the flank of the tooth meets the dedendum circle.
- 5) Module: replaces diametric pitch in metric pitch in metric system.

A gear is a rotating machine part having cut teeth, which mesh with another toothed part to transmit torque. Gearing devices can change the speed, torque, and direction of a power source. Gears almost always produce a change in torque, creating a mechanical advantage, through their gear ratio, and thus may be considered a simple machine. The teeth on the two meshing gears all have the same shape. Two or more meshing gears, working in a sequence, are called a gear train or a transmission. A gear can mesh with a

linear toothed part, called a rack, thereby producing translation instead of rotation. The gears in a transmission are analogous to the wheels in a crossed, belt pulley system. An advantage of gears is that the teeth of a gear prevent slippage. When two gears mesh, if one gear is bigger than the other, a mechanical advantage is produced, with the rotational speeds, and the torques, of the two gears differing in proportion to their diameters. In transmissions with multiple gear ratios—such as bicycles, motorcycles, and cars—the term "gear" as in "first gear" refers to a gear ratio rather than an actual physical gear. The term describes similar devices, even when the gear ratio is continuous rather than discrete, or when the device does not actually contain gears, as in a continuously variable transmission. An external gear is one with the teeth formed on the outer surface of a cylinder or cone. Conversely, an internal gear is one with the teeth formed on the inner surface of a cylinder or cone. For bevel gears, an internal gear is one with the pitch angle exceeding 90 degrees. Internal gears do not cause output shaft direction reversal.

### a) General nomenclature



Photoelastic stress analysis technique based on concept of material birefringence is a non destructive and highly efficient method of structural analysis. The possibility of using photo elasticity to visually represent stress pattern during operation has the potential to create significant efficiency and economical benefits in all aspect of engineering's and industry. Stress freezing photo elasticity is a responsive and inexpensive tool for stress analysis in the industrial environment. Pressure angle of gear plays a vital role in the design of gear, because it simultaneously affects the base circle radius of the involutes profile and minimum number of teeth to avoid interference varies. In past 14.5° pressures angle is used but the most common pressure angle now in use is 20°, for aircraft gears

pressure angle more than  $20^{\circ}$  is often used. Because as  $14^{\circ}$  gears are weak in load carrying capacity. The  $30^{\circ}$  pressure angle gears have strong and thicker at root of teeth so they generally run with more noise. Less than  $20^{\circ}$  pressure angle may be used to get smooth and quitter running. Moreover the Lewis form factor that is used in Lewis equation to account for bending stress at tooth root and incremental dynamic load depends upon pressure angle.

## II. LITERATURE REVIEW

Considerable research work in the area of design and optimization of spur gear has been carried out. It is seen that a very little work has been carried out in the area of helical gear. The gears are used for power transmission in different applications. One of the reasons of failure of gear may be due to failure of teeth. A failure occurs due to excessive bending stress and tension at a critical section giving rise to stress concentrations. With the use of photo elastic and finite element analysis many researchers have worked on the problems of analyzing stresses in different types of machine components and to determine the stress concentration factor to be applied in design. Three dimensional photo elastic technique and FEA finds application in engineering. Before undertaking present work, the literature survey has been carried out. A brief review of some selected references is presented here.

Paul Wyluda et.al [1] have carried out an elastic-plastic finite element analysis of the quasi-static loading of two acetal copolymer gears in contact. According to the International Standard for calculation of load capacity of spur and helical gears, the tooth root tensile stress has relevance to plane strain conditions.. Load verses rotation of the gear set is compared to actual experimental results. The results indicate that to optimize a gear set, a non-linear analysis is required to be performed. Only under low loads and deformation can a linear elastic approach be suitable. Clearly combining computer simulations with material and component testing has led to a far better understanding of copolymer acetal gear design; this understanding could not be achieved by either simulation or testing alone. It is envisioned that with a few more material tests, the torque-displacement response of the gear pair can be simulated with confidence thus advancing the technology of copolymer acetal gear applications.

Cornwill R.W [2] has presented work on compliance and stress sensitivity of spur gear teeth using strain gauges. For measurement of maximum bending stress at tooth root, the gauges were placed at the  $30^{\circ}$  tangency location.

Jose L F Freire et.al [3] has explained details of photo elasticity technique. Photo elasticity indicates not only the most loaded areas of the observed component, but also provides accurate stress values at any critical point. Photoelasticity is a branch of Photomechanics. It employs models constructed from materials transparent to the light being used. These materials display birefringence under applied stress and are observed under polarized light using an instrument called a polariscope.

Donald Berghaus [4] has combined photoelastic data with finite element data for stress solutions over regions

partially bounded by free surfaces and axes of symmetry. Least square solution has been provided without presumed values of applied forces at element nodes.

Fred B. Oswald et al. [5] carried out strain gauge analysis of low contact ratio spur gear to measure dynamic load using NASA gear noise rig. For analysis strain gauges were installed in the tooth root fillets of gear.

## III. PROBLEM IDENTIFICATION

### A. Concluding Remarks:

From the extensive review of the literature work carried out, it is seen that considerable research in the field of stress analysis of spur gear but very few works is available in the field of helical gear. There is no any experimentation work that studies the effect of pressure angle on bending stress at root of tooth of helical gear. A very few pertinent references are available on analysis of helical gear. From the extensive review of the published research work on stress analysis at root of tooth of gears, it is seen that the main issues in the helical gear as follows:

- 1) It is seen that mathematical modeling and design and manufacturing of gear analysis are the ticklish problems.
- 2) The complex helical gear geometry and variable length of tooth contact lines during meshing period results in complex load distribution in gear mesh.
- 3) Experimental methods are necessary for validation of theoretical and numerical analysis. Even though these methods are complicated, laborious, expensive, these can be applied for special cases like analysis of gear teeth.
- 4) Current trends in engineering globalization require research to revisit various normalized standards to determine their common fundamentals and those approaches need to identify "best practices" in industries. In this regard, some work on helical is required.

### B. Problem Definition:

In Shivprasad Industries, Kolhapur, Helical gears are manufactured these gears are used in Printing machines these machines are used to Printing technology. Positive infinite Variable (PIV) gear box is used to transmit to power to belt conveyor of spreading machine .This PIV gear box consist of helical gears. The power transmitted by gear box is 1 kW at (150 to 200) rpm. In working condition of gear box it has been observed that pinion helical gear mounted on motor shaft fails due to load coming on teeth, It seems that the failure is due to stress concentration and bending stresses at the tooth root of gear. The crack is initiated just near to high stress concentration region of gear that is at the root of tooth.



Fig. 3.1: Damaged Helical Gear of Gearbox

C. Objective:

- 1) To carry out the theoretical static analysis of bending stress at root of tooth of helical gears of different pressure angle.
- 2) To carry out the 3D photoelastic analysis of bending stress at root of tooth of helical gears of different pressure angle in static condition.
- 3) To carry out the bending stress analysis at root of tooth of helical gears with different pressure angle in static condition by using analysis FE software like ANSYS.

D. Work Constraints:

The proposed work will be carried out with following steps.

1) Phase I- Literature Survey

In this phase Literature Survey of stress analysis, experimentation and FE study of helical gear will be carried out by referring standard journals.

2) Phase II;

a) Experimental work:

We are going to carry out experimental study by using 3D photo elasticity as follows.

- 1) To prepare patterns of gears as per specifications
- 2) To prepare photoelastic model of gears.
- 3) To design and develop loading frame.
- 4) To carry out stress frizzing operation.
- 5) To evaluate fringe order at root of gear tooth.
- 6) To evaluate Stress in photoelastic model.
- 7) To determine stress at root of helical gear tooth.

b) FE Analysis

- 1) To model gears of different pressure angle and helix angle using modeling software.
- 2) To carry out FE analysis using suitable software.

IV. METHODOLOGY

In 3D photoelastic analysis, the material fringe values are estimated from disc to calibrate photo elastic material. These findings are applied to determine the actual stresses developed in a model and by scaling the model to prototype. The results obtained from this method depend largely on photo elastic model. Therefore, the preparation of the model, itself has its own significance in entire process of photo elastic stress analysis to handle the particular problem. Similarly, in strain gauge analysis, important parameters that should be considered to maintain accuracy in the results are selection of strain gauges and their mounting locations. In the process of gear design pressure angle is an important

parameter as base circle radius of tooth profile and minimum number of teeth to prevent interference depends upon it. Asymmetric gears have different pressure angles on drive side and coast side. The critical thickness of tooth increases due to this asymmetry. Due to low weight to torque ratio asymmetric gears are extensively used in automobile industry, wind turbine industry and aerospace industry. The virtual number of teeth and pitch circle radius vary with helix angle. In this work three different methods have been used to analyze the effect of pressure angle and helix angle on bending stress at critical section of the tooth of helical gear. In theoretical method bending stress has been evaluated using Lewis equation, experimental analysis have been carried out by using 3D photo elasticity. For FE analysis ANSYS software has been used. Entire analysis process involves helical pinion with three different pressure angles i.e full depth 14.00, 200, 300 each subjected to three different torques with three different helix angles (100, 1500, 200) for each of them. Specifications of helical gears in mesh are identical except pressure angle and helix angle, the gear ratio being 1:1.

Specifications:

Normal Module ( $m_n$ ) : 3mm

No. of teeth (Z) : 32

Three different cases of working conditions have been considered. Each of the cases have again three sub cases with three different pressure angles.

Case I	Power (P)	= 1 kW
	Speed (n)	= 150 rpm
	Torque I	= 63661.97 N mm
Case II	Power (P)	= 1 kW
	Speed (n)	= 180 rpm
	Torque II	= 53051.65 N mm
Case III	Power (P)	= 1 kW
	Speed (n)	= 200 rpm
	Torque III	= 45472.84 N mm

In the present research, Taguchi L27 standard orthogonal array has been utilized to design the experiments.

Taguchi Orthogonal Array Design

L27 (3×3)

Factors : 03

Level : 03

Runs : 27

Range of response variables and their levels are selected and presented in table 3.1; accordingly twenty seven experiments were performed on helical gear.

A. Theoretical Analysis

1) Force Analysis in Helical Gear:

Helical gears force analysis can be done in similar manner as is done in spur gear. Helix angle causes creation of additional force component. It is seen as an axial force resulting in axial thrust on the bearings. The pictorial view of tooth forces in helical gear is as shown in figure 3.1. In a helical gear, tooth force  $P_N$  is normal to the tooth surface making pressure angle with horizontal plane. This tooth force has three components acting at right angle to one another.

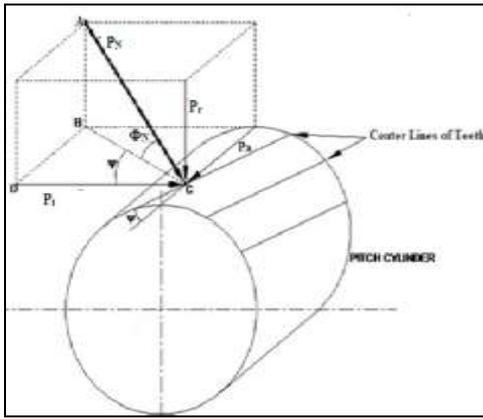


Fig. 4.1: Forces in helical gear

2) Lewis Equation for Bending Stress of Gear:

Bending stress at critical section of gear tooth can be evaluated by using Lewis equation developed by Wilfred Lewis in 1892. Lewis equation is based on various assumptions and gear tooth is considered as cantilever beam as shown in figure 3.2.

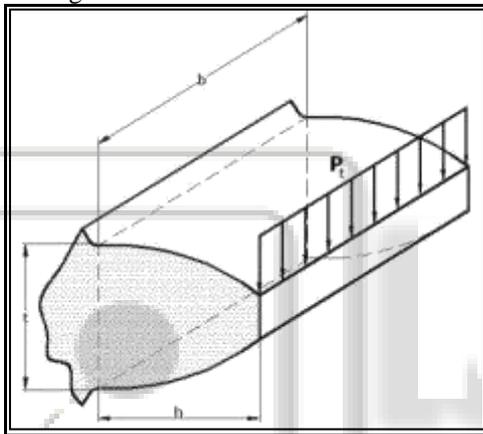


Fig. 4.2: Gear tooth as cantilever beam

The tangential component of resultant force due to transmitted torque produces the bending moment about the base of gear tooth. In Lewis equation the effect of the radial component of resultant force as well as stress concentration has not been taken into account. At any instant only one pair of teeth is assumed to be in contact and total load is subjected to it.

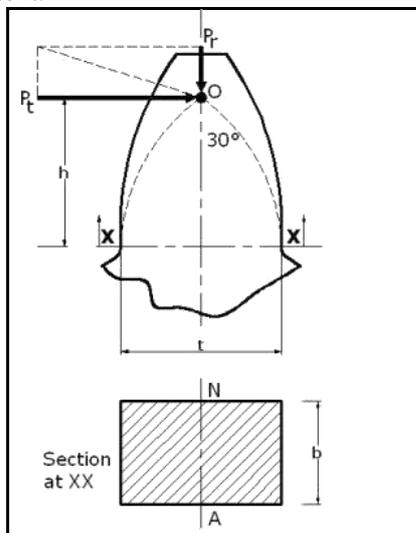


Fig. 4.3: Gear tooth as parabolic beam

B. Beam Strength of Helical Gear:

In order to determine beam strength, helical gear is treated as a formative spur gear. The formative gear means a spur gear which is imagined to be in a plane perpendicular to the tooth element. The beam strength of formative spur gear is given by equation 3.11,

$$S_b = m \times b \times \sigma_b \times Y \quad (3.11)$$

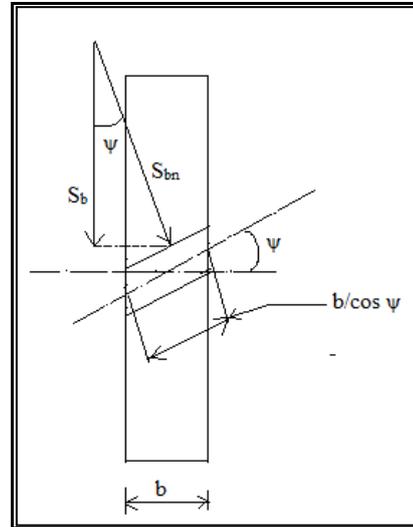


Fig. 4.4: Geometrical interpretation of beam strength in helical gear

C. Experimental Analysis

1) Introduction to Photo Elasticity:

Photoelasticity is a non-destructive and whole-field technique widely used for the stress analysis. In this technique, components to be analyzed are made up of a transparent polymer which exhibits an optical property called birefringence. When the specimen made up of such polymer is loaded and observed against an ordinary light source, specimen shows fringe patterns which can be used to estimate principal stresses generated in a plane perpendicular to direction of the light propagation. The technique was utilized initially to determine two dimensional stresses in components subjected to two dimensional loading conditions. Technique was further modified and extended to estimate the three dimensional stresses in specimen with aid of stress freezing method.

The steps in the experimental procedure of Photoelastic technique include

- 1) Development of specimen/ model of polymers
- 2) Subjecting it to the given loads
- 3) Freezing of stresses in the model
- 4) Observing the fringes under polariscope
- 5) Determination of the material fringe value of a photoelastic material
- 6) Evaluation of the stresses developed in a model and scaling model to prototype.

The accuracy of results obtained from photo elastic model depends upon the preparation of the model. Therefore it has its own importance in the whole problem of photo elastic stress analysis. In other words, manufacturing of model for the photo elastic technique is an art of combination of experience and practice which needs great care.

A variety of materials are used to manufacture photo elastic models, starting from the ordinary glass to modern epoxy resins.

V. DESIGN AND DEVELOPMENT OF LOADING FIXTURE

To have the actual loading conditions of prototype on model, suitable loading frame is appropriate. The design of a loading frame is such that it should affect the actual circumstances in which the circling pinion is operating. chiefly helical gears are used to transmit power or torque between two parallel shafts. By alert this working accustom the loading fixture has been arrange as shown in add 3..

Both the gear pivot are on the seated haft and shaft ends are fixed in ball bearing. The bearing development are connected to the frame with the help of bolts adjustment. The hole is created on the pinion gear cylinder to fit a lever into it. The lever is used to carry the load to be applied on the arm. To get the actual working condition which is to be assumed, the angular of driver gear is restricted. The instrument is provided for this purpose. The real time loading frame with gears, load and loading arm used for empirical work are shown in figure 3.. An arrangement of constraining action torque is made to avoid the orbit of gear

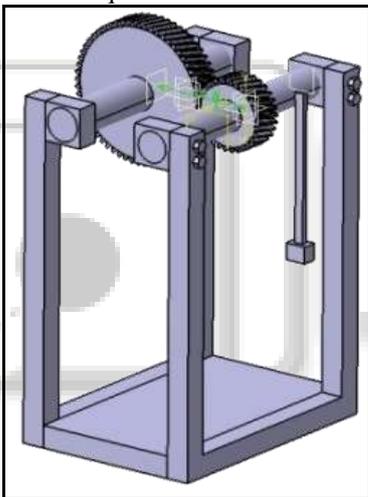


Fig. 5.1: CAD model of loading frame / fixture

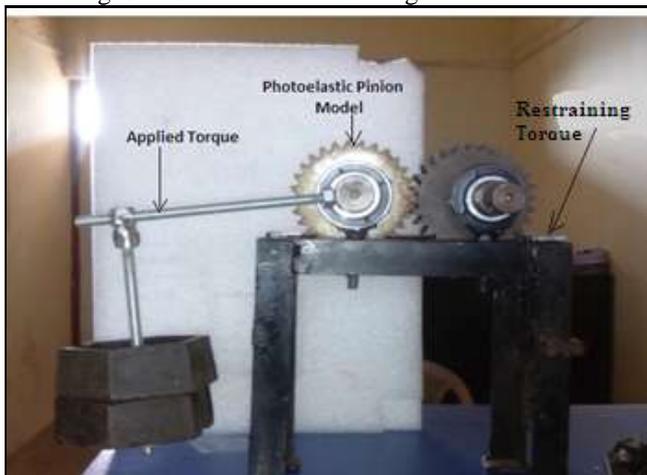


Fig. 5.2: Loading frame for static analysis

1) Slicing:

To observe optical brink in the model, the slicing process is appropriate. The slices are cut on even milling machine at

around 1500 rpm and with a aperture saw of 1mm thickness. The lengthwise slices of model are desirable to cut on the apparatus. This is because model has symmetrical shape and symmetrical loading condition in longitudinal control. Therefore cross and cross slices are not considered for this study. Figure 3. shows a classic slicing process.



Fig. 5.3: Slicing of a stress frozen photoelastic pinion

For continuous cutting of model without disturbing the frozen stresses, enough amount of acerbic oil is used as coolant. Slice density is kept 2 and 3mm. SAE- 40 oil is used as a oating during slicing process. After carve on machine, the surface of each slice is cleaned and gleaming manually by using zero number emery cardboard.

2) Evaluation of Material Fringe Value:

To estimate the certain stress by observing the fringes obtained on slices, the material fringe value ( $f_\sigma$ ) is appropriate. The circular disc of diameter 50 mm and density 5mm subjected to compressive load of 2 kg was used to find material fringe value. In the circular disc weight are frozen during same stress freezing cycle as that of exemplary. Figure 3. shows fringe pattern of a stress chilled disc when observed under circular periscopes

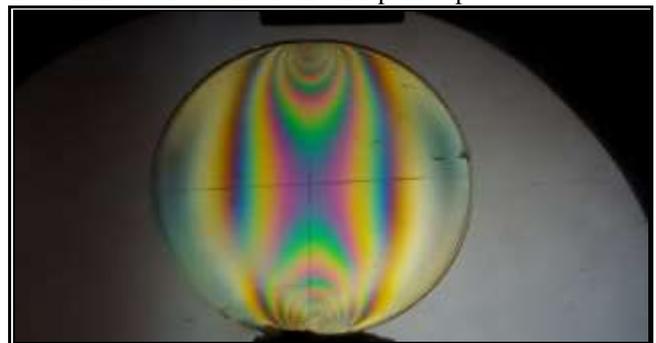


Fig. 5.4: Fringes in calibration disc

The value of  $f_\sigma$  is calculated by using equation 3.22,

$$P = \pi \times N \times D \times f_8$$

Rearrange the equation 3.23 for  $f$ , we get  $W = \pi \times N \times D \times f_8$

$$f = 8 \times F \pi \times D \times N \tag{3.23}$$

Where,

F = Applied Load on the pinion = 2Kg = 19.62 N,

D = Diameter of disc = 50mm,

N = Fringe order observed at the center of the disc = 2.66

Substituting the vales of above terms in equation 3.23, we get,

$$f_\sigma = 0.37 \text{ N/mm}$$

Therefore, material fringe value of considered totalistic material is 0.37 N/mm.

3) *Observation of slices under periscopes*

The all able chop are attended under the bombsight the is client and sonar fringes are access using plane andular circ periscopes preparations [37]. The points of interest are focused during this and the apportioned brink order is recorded using tads method of aid. Figures 3., 3. and 3. represent fringe patterns attended observed during the study of lengthwise slices under communication type periscopes

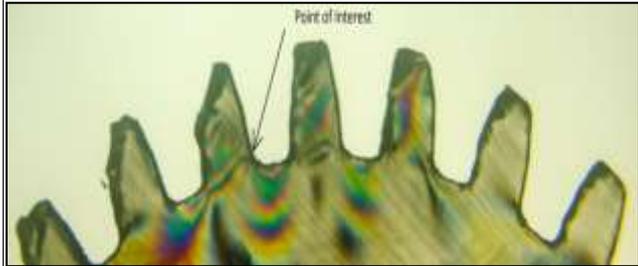


Fig. 5.5: Isochromatic fringe pattern for gear with 14.5° pressure angle, 10° Helix angle and torque I

4) *Evaluation of bending Stress at root of tooth:*

Some arithmetic are required to get the magnitudes of the stresses advanced at the root of marked tooth from fringe arrangement. It is borne out for every slice.

Specimen Calculations for gear with Helix angle 10° pressure angle of 14.5° and torque I:

$$\text{Fractional fringe order} = N = N' + 180 \quad (3.24)$$

Where,

$N$  = Fringe order of the nearest fringe with respect to point of interest = 1

= Analyzer rotation to pass the nearest fringe through that point = 80°.

Put this value in equation 3.18, we get fractional fringe order at root of tooth,

$$N = 1.44$$

In the slice, stresses developed is calculated by equation 3.25,

$$1 - 2 = f_{\sigma} \times N \times h \quad (3.25)$$

where,

$\sigma_{1/2}$  = Maximum and minimum Principal stresses

$N$  = Exact fringe order at the point of interest = 1.44

$f_{\sigma}$  = Material fringe value = 0.37 N/mm

$h$  = Slice thickness = 2 mm.

As  $\sigma_2 = 0$  at the boundary and  $\sigma_1$  = bending stress developed in model

Therefore from equation 3.25,

Bending stresses developed in model

$$\sigma_1 = \sigma_m = 0.266N / \text{mm}^2$$

5) *Scaling model stress to prototype:*

In photo flexible analysis of stress and strain, model is manufactured by using polymeric material to chill the stress into it. Then results of models are correlated to predict the stresses in prototype. But the mock-up is made up of metal. There is large change in elastic constants value of photo elastic material model and metallic mock-up. But, this analysis technique up of metal. alike if the prototype is of large size, scale and applied load, then model can be prepared with smaller size and extent. In this case, it is important to extend this treatment through proper scaling relationship. There is well entrenched theory about the

scaling and model testing in physics. It is achieved by applying dimensional ratios and the big-name  $\pi$  theory. for plane stress does not depend upon the value of elastic consistent of material. Thus stress circulation obtained during this method can be used to correlate the stress distribution of mock-up even though it is made up of metal. alike if the prototype is of large size, scale and applied load, then model can be prepared with smaller size and extent. In this case, it is important to extend this treatment through proper scaling relationship. There is well entrenched theory about the scaling and model testing in physics. It is achieved by applying dimensional ratios and the big-name  $\pi$  theory.

VI. RESULT AND DISCUSSION

For enhanced accuracy and higher endurance of gear, clear knowledge of gear tooth stress all along cobweb is necessary. As such in this essay, bending stress analysis at critical section of helical gears have been carried out. By seeing the contrasting torques, squeeze angles and helix angles for helical gear, bending stress behavior were observed through the empirical method, FEA method and abstract method. The results are esoteric compile in the table number 4.1.

The results shows that there is appropriately good agreement between bending stresses at critical area under static condition by all design methods. The bending stresses at demanding area by theoretical method and finite element analysis are very close to each other, recorded error in reach 8.19%. The maximum error recorded between experimental and finite element analysis results is 8.22%, whereas between experimental and theoretical results is 9.34%.

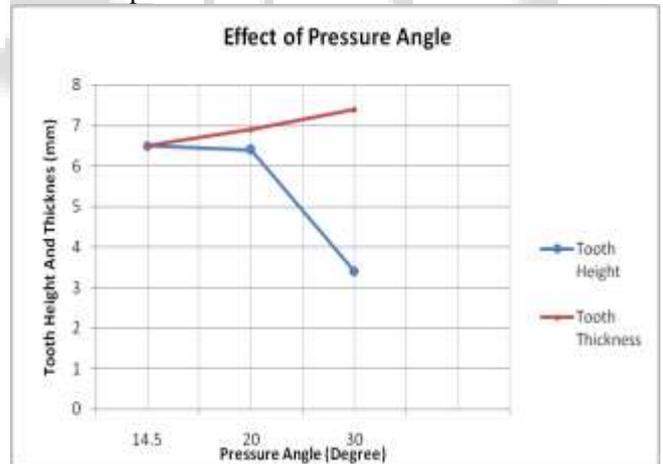


Fig. 4.1: Effect of pressure angle on the tooth height and the tooth thickness

The amount number 4.1 shows the effect of burden angle on the tooth height and the tooth density of differing gears used for this study. Why the bending stresses are decreased with inbound in bend the pressure angles of gear? And how does the congestion angle play a vital role in gear design? strength. These questions can be acknowledge by the above graph. It shows that the tooth thickness at critical area increases and the tooth height decrease with burden angle which advance the tooth strength.

Helix Angle (Degree)	Pressure Angle (Degree)	Torque (N-mm)	Theoretical Bending Stress (N/mm <sup>2</sup> )	Experimental Bending Stress (N/mm <sup>2</sup> )	FE Bending Stress (N/mm <sup>2</sup> )
10	14.5	I	55.81	53.27	54.73
		II	49.49	44.55	45.74
		III	39.84	37.08	38.68
10	20	I	49.02	46.72	48.52
		II	40.83	37.85	40.64
		III	35	32.01	34.64
10	30	I	34.80	32.74	33.81
		II	28.83	26.77	28.1
		III	24.71	22.75	24.6
15	14.5	I	54.23	52.29	52.7
		II	45.88	43.84	43.87
		III	38.75	37.26	36.44
15	20°	I	47.44	45.79	45.84
		II	39.52	36.96	38.27
		III	33.89	32.39	31.67
15°	30	I	33.50	32.55	32.3
		II	27.90	27.28	26.9
		III	23.93	22.71	22.54
20	14.5	I	51.79	51.49	50.12
		II	43.17	44.23	41.63
		III	37.00	35.32	34.67
20	20	I	44.93	43.92	43.29
		II	37.55	36.96	36.27
		III	32.10	31.14	29.69
20	30	I	31.97	30.87	30.65
		II	26.65	25.8	25.65
		III	22.84	22.01	21.14

Table 4.1: Bending Stress Result

For consistent pressure angle, the helix angle is directly comparable to the tooth thickness i.e. as the angle value increases the tooth thickness also increases. It leads to more amount sharing circulation area of tooth at root and result in increasing stress absorbing ability. Thus greater value of the helix angle will increase the stress carrying capacity of the gear. In this contact, for higher value of helix angle, the bending stress generated at the root of tooth is less. The experiments were attend for different gears in which the gear with pressure angle 30° and helix angle 20°, is having higher tooth thickness around 7.8 mm, while gear with pressure angle 14.5° and helix angle 10°, is having less thickness of 6.7mm.

Results indicates the effect of helix angle on arching stress achieve at critical section of the tooth. The bending stresses are decreased with increase in the compel angle for constant helix angle. The bending repeat are also decreased with the increasing value of the helix angle as number of choppers in contact increases with boost in helix angle.

## VII. CONCLUSION

From theoretical calculations it is come to know that, Pressure angle increases bending stress decreases like that helix angle increases bending stress decreases. That is

Pressure angle and helix angle is inversely proportional to the bending stress.

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