INDIAN JOURNAL OF SCIENCE AND TECHNOLOGY

RESEARCH ARTICLE

Whole Field Analysis of Asymmetric Helical Gear Using Photo Stress Method and FEA Method

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OPEN ACCESS

Received: 09-09-2022

Accepted: 28-11-2022

Published: 13-01-2023

Citation: Patil PJ, Patil AJ (2023) Whole Field Analysis of Asymmetric Helical Gear Using Photo Stress Method and FEA Method. Indian Journal of Science and Technology 16(2): 109-117. https://doi.org/10.17485/IJST/v16i2.1817

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Funding: None

Competing Interests: None

Published By Indian Society for Education and Environment (iSee)

ISSN
Print: 0974-6846
Electronic: 0974-5645

Abstract

Objectives: To carry out dynamic state analysis of asymmetrical helical gears for contact stresses using photo stress technique and suitable finite element software. **Methods:** Currently the Hertz contact stress equation is used to calculate the contact stress which is based on certain speculations. The value of load transmitted by a gear may be known but whether it is taken by a single or more pairs of teeth are based on some assumptions and stresses thus calculated may not be true stresses. To get precise evaluation of stress field in gear tooth it is decided to carry out experimental work on asymmetric helical gear in actual working condition. The photo stress method not only provide magnitude of stress but also gives idea about stress distribution in gear. Finite element method is used for asymmetrical helical gear to validate experimental results. **Findings:** The stress values obtained by photostress method and FEA method has minimum error of 2.17% and maximum of 8.89% respectively. The error values are less than 10% and within the limit. We can observe close agreement between these two methods. **Novelty:** This study has provided a more accurate result than theoretical method of stress calculation. Photostress analysis method is one of the best test methods for evaluation of contact stress of asymmetric helical gear with different torque condition in actual working that is dynamic condition. Maximum error of 8.89% has been found with photostress method.

Keywords: Asymmetric Gears; Contact Stress; Experimental Method; Reflection Polariscope; PhotoStress; Finite Element Analysis

1 Introduction

Gears are used to transfer power to the modern world of mechanical engineering. The primary motive of the gear machines is always transmitting the rotation and torque between the shafts. When in loading condition of gears, contact behavior of a gear changes greatly. This results into a variety of gear conditions, causing it to failures like wear and pitting in various areas in teeth surface. One such phenomenon is pitting. It is a local failure because of the repetition of high contact pressures occurring at the time of meshing [1]. Asymmetric gear teeth have teeth with different pressure angles on both driving side as well as coast side. Because of its geometry, different pressure angles on the driving side and the coast side can be used. The maximum pressure angle on the drive side results into higher load capacity, contact pressure, and vibration rate [2]. During the process of meshing of tooth many internal forces are developed due to the elastic deformation and manufacturing errors. Dynamic stress analysis can lead to adjustments during the design stage creating both economical and efficiency benefits [3]. Several assumptions have to be made to perform the calculations of stress in gear teeth. Current trends in engineering globalization require research to revisit various normalized standards to determine their common fundamentals [4]. Accurate assessment of stress in gear teeth especially in the dynamic state is essential. The complexity of gear profiles and changing load during any loading cycle add to the problems of determining both maximum bending stress and maximum contact stress [5]. The absence of dynamic stress analysis can have catastrophic effects, as were seen in the destruction of the European space Agency, Ariane rocket launcher in 1996 [6]. In photoelasticity use of white light is well known for analysis of isochromatic fringes in birefringent coating and for nondestructive evaluation of stress in mechanical components [7]. Photoelastic data are straightforward to obtain along solution region boundaries which are free surfaces or lines of symmetry. For such locations, the principal directions are known and it is not necessary to determine isoclinic angles [8]. The two methods, photoelasticity and finite elements, taken together provide greater confidence and understanding of the true stress state than does either method alone [9]. Complex engineering problems which are difficult to solve by conventional methods can be solve by Experimental element method and FE methods readily programmed for this purpose [10].

2 Methodology

2.1 Theoretical Calculations

For theoretical contact stress analysis asymmetric helical gears with same helix angle and different angle of pressure on both sides of gear which are drive side and coast side have been considered. Velocity factor method gives the approximate estimation of dynamic load in preliminary stages [11]. It's almost impossible to calculate accurate magnitude of a variable load in the early phases of gear formation. To prevail over the problem using the velocity factor (Cv) developed by Barth. The velocity factor method is used for determining effective dynamic load of helical gear in working condition.

Specifications of Gear

1. Material - EN 8 steel
2. Module (m) - 3 mm
3. Number of teeth in pinion, Zp - 18
4. Number of teeth in gear, Zg - 72
5. Face Width(w) - 35 mm
6. Helix Angle (φ) - 20°
7. Drive side angle (ϕ) - 25°
8. Coast side angle (ϕ) - 20°

Sample Calculation for case 1-

Speed = 32 rpm

Torque =1500 Nmm

Helix Angle (φ) = 20°

Drive-side pressure Angle (ϕ) = 25°

Pitch Circle Diameter,

$$\mathrm{d}p = \frac{m \times \mathrm{Zp}}{\cos\varphi} \tag{1}$$

$$\mathrm{d}p = \frac{3 \times 18}{\cos 20}$$

$$\mathrm{d}p = 57.47 \text{ mm}$$

Virtual No. of teeth,

$$z'p = \frac{\text{zp}}{(\cos\varphi)3} \tag{2}$$

$$Z'p = \frac{18}{(\cos 20)3}$$

$$z'p = 22$$

Tangential component of force,

$$Pt = \frac{2\times T}{dp} \tag{3}$$

$$Pt = \frac{2\times 1500}{57.47}$$

$$Pt = 52.20\ \text{N}$$

Pitch line velocity,

$$\text{V} = \frac{\pi\times \text{dp}\times N}{60000} \tag{4}$$

$$\text{V} = \frac{\pi\times 57.47\times 32}{60000}$$

$$\text{V} = 0.096\ \text{m/s}$$

Velocity Factor,

$$Cv = \frac{5.6}{5.6+\sqrt{v}} \tag{5}$$

$$Cv = \frac{5.6}{5.6+\sqrt{0.096}}$$

$$Cv = 0.947$$

Effective Load,

$$Peff = \frac{\text{Cs}\times\text{Pt}}{\text{Cv}} \tag{6}$$

Service factor (Cs) = 1.1,

$$\text{Peff} = \frac{1.1\times 52.20}{0.947}$$
$$\text{Peff} = 60.63\ \text{N}$$

Ratio Factor,

$$\mathrm{Q} = \frac{2 \times dg}{dg + dp} \tag{7}$$

$$Q = \frac{2 \times 229.86}{57.47 + 229.86}$$

$$Q = 1.6$$

Load Stress Factor,

$$\mathrm{K} = \frac{\mathrm{Peff} \times (\cos\varphi)2}{b \times Q \times dp} \tag{8}$$

$$\mathrm{K} = \frac{60.63 \times (\cos 20)2}{35 \times 1.6 \times 57.47}$$

$$\mathrm{K} = 0.017$$

Also,

$$\mathrm{K} = \frac{\sigma_c^2 \times \sin\phi \times \cos\phi \times \left(\frac{1}{E1} + \frac{1}{E2}\right)}{1.4} \tag{9}$$

$$0.017 = \frac{\sigma_c^2 \times \sin 25 \times \cos 25 \times 10^{-5}}{1.4}$$

$$\sigma_c^2 = 6213.73$$

$$\sigma_c = 78.82\ \mathrm{N/mm^2}$$

The effective load and contact stress for different torques evaluated by using velocity factor method are as follows and corresponding stresses are listed in Table 1 .

Table 1. Effective load and contact stress for different torques

Case. No.	Speed (RPM)	Torque (Nmm)	Contact Stress (N/mm^2)
Case I.	32	1500	78.82
Case II.	60	2250	95.59
Case III.	80	3000	111.478

2.2 Photo-stress analysis

Photostress method is used in this study to experimentally find the contact stresses in asymmetric helical gear. This method not only gives magnitude of stress but also the stress distribution. Photoelastic stress analysis, a technique based on the concept of material birefringence, is a non- destructive and highly-efficient method of structural analysis with great potential to be used in dynamic analysis. This research explores the possibility through the synchronization of a strobe light and a digital camera. Using this synchronization, it may be possible to capture progressive images of the stress distribution throughout a turbine model rotating at variable and increasing speeds. Utilizing a rapid prototyping technology called stereolithography, the model can be created such that it can be both prepared and analyzed by the method of photoelasticity.

2.2.1 Selection of Material

In order to get sufficient optical response and to make measurement easy the coating material has sufficient thickness and high strain optical coefficient. The surface of gear to be coated is flat. So considering above points PS-1 pre-manufactured sheet with thickness 3.05 mm [12] has been selected as photoelastic coating for analysis.

2.2.2 Development of Loading Fixture

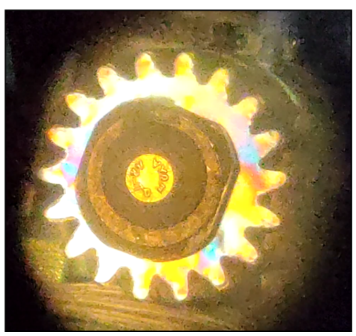
There are no customary hobs present for making asymmetric involute helical gear, so a specialized hob has to be made. In place of the regular method of designing gear. A 3 Hp DC motor (2.237 kW) is selected as the main motive and speed variation is achieved using a 12 ampere DC dimmer. For maximum torque-designed torque, a rope dynamometer is selected [13]. Gear mounted shafts are supported by bearings. A weighing pan is hanged to wire cord and a pulley with a diameter of 300 mm is used for the intended torque.

2.2.3 Coating Preparation

The coating installation process is achieved through following operations, Preparation of Coating, Surface construction, Adhesive construction, Bonding Process. Flat pre manufactured photostress sheet of PS-1 plastic material has been selected for dynamic analysis of Asymmetrical helical gear. The sheet is trimmed properly using cutting machine. The surface of the pinion to be coated was cleaned by washing it down with isopropyl alcohol and conditioned. As we have used PS-1 material for coating so PC-1C is taken as adhesive.

2.2.4 Measurement of fringe order

After successful installation of photostress coating on surface of pinion, coated pinion along with its mating gear was mounted on loading fixture. The strains in the coating produce proportional optical effects which appear as isochromatic fringes when viewed through a reflection polariscope as shown in Figure 1 . As rotating pinion is exposed to white light and observed through reflection polariscope, colorful isochromatic fringes were observed in the coating attached to pinion surface. These colorful fringe patterns were captured by using a high-speed camera in dynamic state. For full field interpretation of fringe pattern stroboscope is used to make rotating gear appear stationary during measurement of fringe order.



Color	Approximate Relative Retardation nm	Approximate Relative Retardation in x 10^{-6}	Fringe Order N
Black	0	0.0	0.0
Pale Yellow	345	14.0	0.60
Dull Red	520	20.0	0.90
Red/Blue Transition	575	22.7	1.00
Blue-Green	700	28.0	1.22
Yellow	800	32.0	1.39
Rose Red	1050	42.0	1.82
Red/Green Transition	1150	45.4	2.00
Green	1350	53.0	2.35
Yellow	1440	57.0	2.50
Red	1520	60.0	2.65
Red/Green Transition	1730	68.0	3.00
Green	1800	71.0	3.10

Fig 1. Fringe Patternobserved

2.2.5 Measurement of Contact Stress

The fringe patterns captured in photostress coatings equate to the difference between the principle stress or the principle strains on the coating or over the helical pliers [14]. The relation mentioned above is expressed as below:

$$\sigma_x - \sigma_y = \frac{E}{1+\gamma} \times \mathrm{N}f \tag{10}$$

Where;

$\sigma_x - \sigma_y$= Principal stresses difference

E = Modulus of elasticity

γ = Poisson's ratio

N = Fringe order at contact region of gear tooth

f = PS-I coating material fringe value

If the stress level is uniaxial, only one major stress is present in the local plane of the test component and this can be found in equation 10.

$$\sigma_x = \sigma_c = \frac{E}{1+\gamma} N f \tag{11}$$

The fringe value is obtained from,

f = $\frac{}{2tk}$ 12

Where,

t = Thickness of coating which is 3.05 mm

l = Wavelength of passage tint in white coloured light, which is 575 nm

K = Photoelastic plastic strain-optic coefficient = 0.15

Putting the values of above mentioned l, k and t in the equation 12, value of f,

f = 6.28×10^{-4}

As the gear material is taken steel [EN8], For the Steel:

E = 2×10^5 N/mm^2 and $\gamma = 0.3$

Puttinσ_c ab$\frac{2\times10^5}{1+0.3}$ve values o^f γ, f and E in equation 11, value of σ_c,

$$\sigma_c = \frac{2\times 10^5}{1+0.3} \times 6.28 \times 10^{-4} \times \mathrm{N} \tag{13}$$

Stress Calculations:

Captured fringe order at contact of tooth is N= 0.7, from eq 4,

$$\sigma_c = \frac{2\times 10^5}{1+0.3} \times 6.28 \times 10^{-4} \times 0.7$$
$$\sigma_c = 67.63\ \mathrm{N/mm^2}$$

Fringe orders and contact stresses at contact of gear tooth by photostress technique for different torques and loads are given in Table 2 .

Table 2. Captured fringe order at contact of tooth is N= 0.7,from eq 4,

Sr. No.	Torque (Nmm)	Fringe Order	Contact Stress (N/mm^2)
1.	1500	0.7	67.63
2.	2250	0.8	77.30
3.	3000	0.9	86.97

2.3 Finite element analysis

The FE model is designed in such a way as resembles actual gear system. The gear models of driven and driver gears as specified have been modeled using the software CATIA V5 R17. For the dynamic analysis it is decided to carry out the analysis in ANSYS 2021 R1 [(15)]. To simulate actual working conditions explicit dynamics is used. Importing the assembly file to ANSYS workbench in IGS format as shown Figure 2 . A local coordinate system is defined on the gear, it is then changed to cylindrical coordinate system and all other movements are restricted except the rotation in Y direction. Define pinion as flexible and gear as rigid element. The velocity is applied on gear in Rad/sec and additional load has been applied on tooth flank in vertical direction. The results are shown in Figure 3 .

Material Properties:

Defining material properties for rigid element and flexible element as below,

Material Steel [EN8]

Young's Modulus (E) = 2 X 10^5 N/ mm^2,

Density = 7850 Kg/ m^3

Tensile Stress = 750 N/mm^2 ,

Poisson's ratio (γ) = 0.3

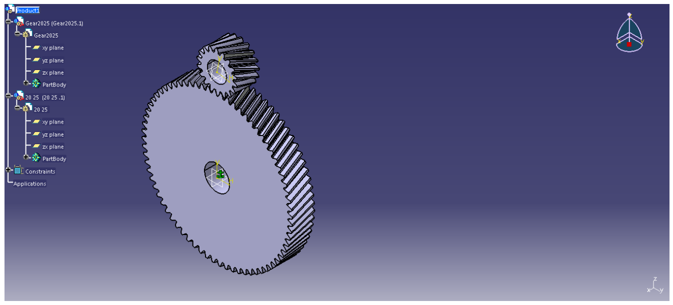


Fig 2. CATIA assembly of helical gear pair

Yield stress = 340 N/mm^2
Mesh Properties:
Number of element failing in Warpage, Contact element: CONTA 174.
Mesh size – 1 X 1. Hexa and Tetra element, Element order – Linear

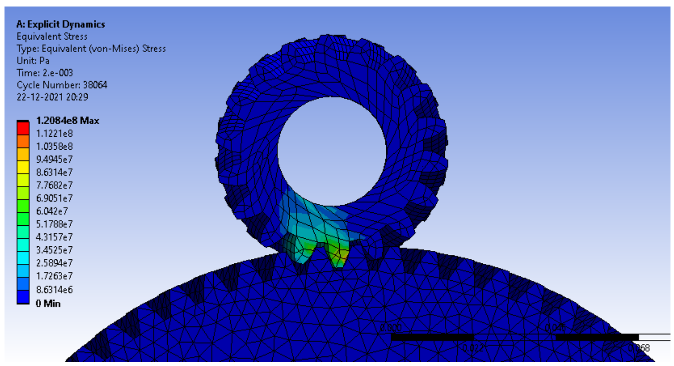


Fig 3. Stress distribution at torque 3000 nm

Following Table 3 shows the contact stresses for various conditions by FEA method.

Table 3. Contact stresses by FEA

Sr. No.	Torque (Nmm)	Contact Stress (N/mm^2)
1.	1500	66.19
2.	2250	81.05
3.	3000	94.95

3 Result and Discussion

To get improved performance and quality of a gear, clear knowledge of stress acting on the gear while meshing is very crucial [16]. This study is performed to find more precise and optimum method for the analysis of the gear under dynamic conditions. To understand the real time stress variation in the gears whole field or dynamic state analysis was carried out as an important objective of the study. By considering the different torques, contact stress behavior in a dynamic state were evaluated through velocity factor method, Photostress method and FEA method are given in Table 4 .

Comparatively very large variation is observed between experimental and theoretical results because in theoretical method, the effective load is based on tangential component of load. For estimation of tangential load, theoretically, it is assumed that total load is shared by single tooth as in static analysis but in actual the total transmitted load is shared between the meshing tooth pairs. Also because of induction of velocity factor and service factor in velocity factor method the value of effective load increased. In actual working condition, more than one tooth is always in contact which causes the amount of effective load with respect to single tooth to reduce. Moreover, it is obvious fact that stress on single static tooth is always more than on the same tooth set in motion. It emphasizes the need of use of the advanced method to get the true idea about the working conditions at engaged gear tooth. It can be interpreted that the experiment photostress method gives contact stress closer to the true value in dynamic conditions. From all the results, it can be concluded that in dynamic state, the contact stress at contact point increases with increase in torque.

Table 4. Contact stress for different torques and percentage errors

Sr. No.	Contact Stress by Theoretical Method (N/mm^2)	Contact Stress by Photo stress Method (N/mm^2)	Contact Stress by FEA Method (N/mm^2)	Percentage error between Theoretical & FEA	Percentage error between Photo stress & FEA
1.	78.82	67.63	66.19	19.08 %	2.17 %
2.	95.59	77.30	81.05	17.93 %	4.85 %
3.	111.48	86.97	94.95	17.4 %	8.4 %

This research work also proposes a modification factor for velocity factor method which is also an important output of this study. The contact stresses calculated by velocity factor method is more than actual contact stress developed during working of asymmetric helical gear. A modification factor of 0.85, 0.80, 0.78 for different torques 1500 Nmm, 2250 Nmm, 3000 Nmm, respectively should be multiplied to the theoretical results so as to get accurate results.

Figure 4 is the graphical representation of contact stress results obtained by various methods.

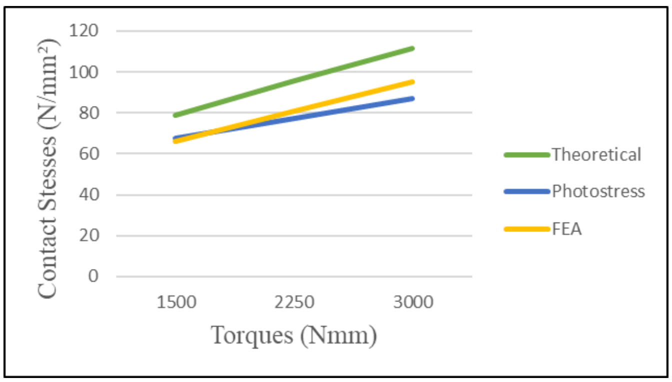


Fig 4. Stress distribution comparison

4 Conclusion

The following are the conclusions obtained from the whole field analysis or contact stress analysis of asymmetric helical gear by using theoretical, experimental and FEA methods.

1. In actual working condition that is dynamic state there is very good agreement between results obtained from experimental photo stress method and Finite element method for contact stress with maximum error of 8.4 %.
2. In dynamic state a large variation has been observed when contact stress of gear tooth obtained by velocity factor method is compared with contact stress by FEA method. The maximum error recorded is 19.08 %.
3. The modification factor of 0.85 to 0.78 has been suggested to get the more accurate results of contact stress. The contact stress goes on increasing with increase in torque value.
4. The results obtained of experimental and theoretical method varies because theoretical method is based on different assumptions leading to overdesign. Hence Photostress method is best method for precise whole field stress analysis in gear teeth under dynamic state.

5 Acknowledgement

The authors thank Prof. Mirza, RIT, Islampur for permitting to use Polariscope instrumentation facilities.

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