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Research Article

Static Analysis of Bending Stresses on Spur Gear Tooth Profile by using Finite Element Analysis & Photo Elastic Technique

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Abstract

Gears are one of the most basic components of any machine or mechanism. Spur gear is the most common means of transmitting power in the modern mechanical engineering world. The main objective of this research paper is to analyze the bending stresses occur on the gear tooth profile when subjected to loading condition with the help of FEM and Photo-elastic technique. Photo elastic stress analysis is a technique that provides stress distribution over an entire object/structural member of interest it is based on the property of some transparent material to exhibit colorful pattern when viewed with polarized light. These patterns occur as the result of alternation of the polarized light by the internal stresses into two waves travel at different velocities. In this work 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 Photo elastic technique. For this work parametric modeling is done using Pro-e WF 5.0 and for analysis ANSYS 12.0 workbench is used. For photo elastic validation optical polariscope is used .This work helpful to conclude effect of bending stress on gear tooth profile by variation of gear root radius it also give the comparison of FEM method with photo-elastic technique of stress analysis .

Keywords: *Spur gear, stress, FEA, & Pro/E, Ansys 12.0*

1. Introduction

Gears are essential for transfer motion with constant velocity ratio used in nearly all the applications where power transfer is required such as automobile, Industrial equipment. Spur gear can be used transmit large power. Usually failure of gear is occurred due to exceed of bending stresses on gear tooth profile. In this paper effect of bending stress on spur gear tooth profile by variation of gear root radius of gear are studied with the help of FEA this result then compared with the stresses calculated by Photo elastic technique.

There are several kinds of stresses present in loaded and rotating gear teeth. We have to consider all the possibilities, so that the gears all the stresses with in the design limit. Generally stresses calculated in the gear design formula are not necessary true stress, can make it difficult to get correct answer on gear-tooth stresses, because it may not be known whether loaded uniformly distributed across the width and whether properly shared by the two or more pairs of the teeth that are in mesh at same time. So we have to make right assumption that will allow for thing like stress concentration, residual stress, misalignment and tooth error, this means that the calculated stress is properly a not true stress. Each gear tooth may be considering as a cantilever beam, when it transmits the load, it subjected to loading (Shinde *et al.*)

Designing highly loaded spur gear for power transmission system that are both strong and quite requires analysis method that can easily be implemented and also provide information on contact and bending stresses. The finite element method is capable of providing this information, but the time required this model is large.

In order to reduce the modeling time, a preprocessor method that creates the geometry needed for finite element analysis may be used such that provided by Pro/Engineer. Using API toolkit of Pro/Engineer one can generate model of three dimensional gear easily. In Pro/E, the geometry is saved as a file and then it can be transferred from Pro/E to Ansys (Konstandinos *et al.*)

Literature survey contains the research work on stress analysis of the gear with the help of the FEM method and photo elastic method. According to the literature survey scope to analyze in static condition effect of bending stress on gear tooth profile with variation of gear parameter and validate with the help of the photo elastic method.

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Konstandinos *et al*. presented the calculation of maximum stress at gear tooth root when the meshing gears are loaded at their most unfavorable contact point (highest point of single-tooth contact-HPSTC), using both numerical and experimental methods. Finite Element Method (FEM) is used for the numerical stress analysis and photo elasticity is applied for the experimental investigation of the stress field.

Shinde *et al*. presented the spur gears to resist bending failure of the teeth, as it affects transmission error. Finite element models and solution methods for accurate calculation using ANSYS then result are compared with results obtained from existing theoretical methods. Prashant *et al*. shows the stress pattern by using three dimensional Photo elasticity techniques. Experimentally obtained results are verified with finite element analysis results. Finally the effect of bending stress is discussed. Sunil Kumar *et al*. shows the CAD model of spur gear using Pro/Program toolkit of Pro-E and investigated spur gear design by determining contact and bending stresses with the help of ansys 10.0. Atanasovska, *et al*. presented the development of the finite element model for simultaneously monitoring the deformation and stress state of teeth flanks, teeth fillet and parts of the helical gear during the tooth pair meshing period. Ananda Kumar *et al* studied the spur gear crack propagation of path using the finite element method. Pravin *et al* presented the stress analysis of the gear box used in sugar industry. They used the ANSYS for stress analysis to find out the cause of failure of the gear and improve the life of the gear they found the reason of failure the gear is wear at gear tooth edge. Dhavale & *et al*. analyze the higher stresses zone on gear tooth profile the stresses at that zone is minimized by introducing stress relief features at stress zone. Result that stressed reducing feature produces the good effect when they situated at most beneficial location. Vivek *et al*. presents the stress analysis of mating teeth of spur gear to find maximum contact stress in the gear teeth. The results obtained from Finite Element Analysis (FEA) are compared with theoretical Hertzian equation values. For the analysis, steel and grey cast iron are used as the materials of spur gear. Their study they conclude that the difference between maximum contact stresses obtained from Hertz equation and Finite Element Analysis is very less and it is acceptable.

2. Methodology

In this work gear used in special purpose machine is consider. Regular failure of gear occurs due to the bending stresses which occur on gear tooth profile during working. So we carried out the work to find out value of root radius of gear which produces minimum bending stress during working condition and life of gear get increases. Spur gear specifications are given in Table 01.

Table 1 Spur Gear specification

Number of teeth(Z)	24
Module (mm)	4.5
Pitch Circle Diameter (D) (mm)	108
Base circle diameter (mm)	100
Pressure angle (degree)	20
Addendum circle diameter(mm)	112.5
Circular pitch(mm)	12.56
Face width (mm)	10

Spur gear Material is steel
Modulus of elasticity E= 210000 MPa
Poisson ratio = 0.3.
In this work considering the Gear box transmitting = 4 KW power at 318 RPM,

Calculate the Tangential loadW_T

Power in (KW) = $\frac{2\,\pi NT}{60\ X 10^6}$

Torque (T) = $\frac{P(\mathrm{kw}) X 60\ X 10^6}{2\,\pi N}$ (1)

Tangential load (W_T) = $\frac{T}{\frac{D}{2}}$ (2)

Torque (T) = $\frac{4X60\ X10^6}{2\ X\pi X\ 318}$ = 120177.8 Nm

Tangential load (W_T) = $\frac{120177.8}{\frac{108}{2}}$ = 2225.5 N

2.1 Geometric Modeling

In this work parametric geometric modeling of spur gear is done by using the software Pro-e WF5. Modeling is done by defining the relation in terms of basic gear parameter i. e Number of teeth Z, Module M and The Pressure angle Θ.

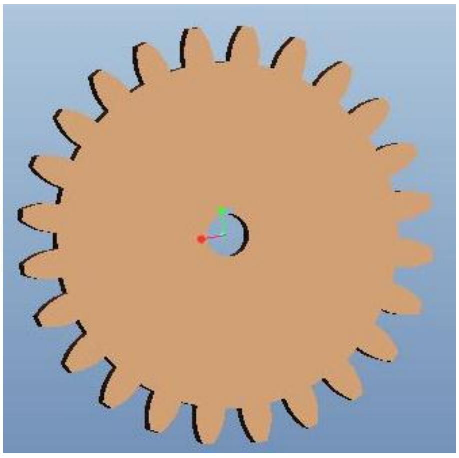


Fig 1 3- D modeling in Pro/E WF5

2.2 Finite Element Analysis

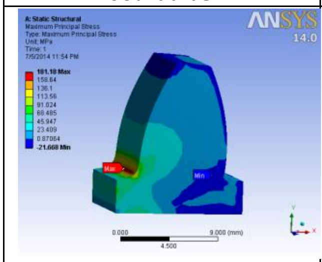
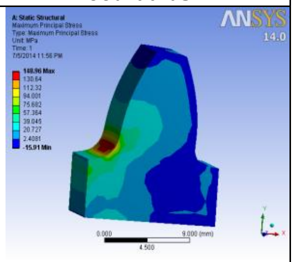
Spur gear only tooth section is considered for Finite element analysis. The total analysis is done on the

Ansys 12.0 workbench. A point load 2503 N is applied at tip of the gear.

The basic procedure steps as follow.

1. Import the geometric model in the form of IGES.
2. Define the properties of material
3. Mesh the model.
4. Apply the boundary constrained and point load on model.
5. Solve for the Maximum Principle stresses.

As per the root radius variable gear parameters are same procedure is done by changing the root radius. We get the results are shown in Table 2. These results are shows the Ansys result of Max Principle stresses for different root radius. Fig 2 shows the Ansys FEA result of gear tooth having different root radius.

Root radius 1	Root radius 1
	
Max Principle Stress.- 181.18	Max Principle Stress- 148.96

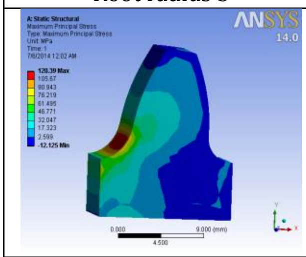
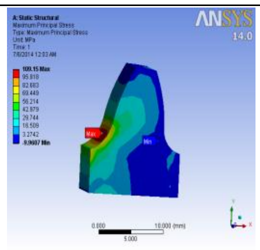
Root radius 3	Root radius 4
	
Max Principle Stress- 120.39	Max Principle Stress- 109.15

Fig.2 Ansys result on different roots radius.

Table 2 Ansys result for different face width & root radius

Sr. No.	Module (mm)	Root radius (mm)	Max Principle Stress
1	4.5	1	181.18
2	4.5	2	148.96
3	4.5	3	120.39
4	4.5	4	109.15

We have conclude that spur gear having radius 4 produces lower bending stress so that we have cross check this result with photo elastic experimental method.

3. Experimental Method

Photoelasticity is an experimental method to determine the stress distribution in a material where mathematical methods become quite cumbersome. Unlike the analytical methods of stress determination, photoelasticity gives a fairly accurate picture of stress distribution even around abrupt discontinuities in a material. The method serves as an important tool for determining the critical stress points in a material and is often used for determining the stress concentration factors in irregular geometries.

3.1 Gear Casting

- Firstly prepared the mould by using the acrylic sheet [6]. The dimension of the mould is 200×200.
- The volume of sheet is 144 Cubic Capacity for every 100 Cubic Capacity of araldite 10 c.c. of hardener is to be mixed.
- 144ml araldite and 14 ml of hardener and mixed with each other.
- The mixture should be stirred in one direction continuously for 15 minutes till it is transparent.
- The mixture is ready to pouring in the mould for preparation of the sheet.
- This is the gear model after the machining is use for the experimentation.

3.2 Scaling Model to Prototype Stresses

In the analysis of a photoelastic model fabricated from a polymeric material, the question of applicability of the result is often raised since prototype is usually fabricated from metal. Obviously, the elastic constant of the photoelastic model are greatly different from those of the metallic prototype. However, the stress distribution obtained for a plane-stress or plain-strain problem by a photoelastic analysis is usually independent of the elastic constant and the results for an elastic analysis are applicable to a prototype constructed from any material. Since photoelastic model may differ from the prototype in respect to scale, thickness and applied load as well as elastic constant, it is important to extend this treatment to include the scaling relationship. The literature abounds with scaling relationship employing dimensionless ratios and the Buckingham π theory. For instance, for a model with an applied load P, the dimensional less ratio for stress is

$$\frac{\sigma m}{\sigma p} = \frac{Pm}{Pp} \times \frac{hp}{hm} \times \frac{lp}{lm} \qquad (3)$$

Fig.3 Casted Gear Model

3.3 Specifications of Polari scope

1. Representation of mechanical distribution of stress in photo elastic experiments.
2. Two plane polarization filters as polarizer and analyzer.
3. Two quarter wave filters to generate circular polarized light
4. All filters with 360° angle scale and marking of the main optical axis
5. White light generated using fluorescent tube andtwo incandescent lamps
6. Monochromatic light (color yellow) generate during a sodium vapour lamp
7. Filters roller bearing mounted and rotating
8. Frame cross-arms height-adjustable
9. Generation of compression or tension forces by means of a threaded spindle

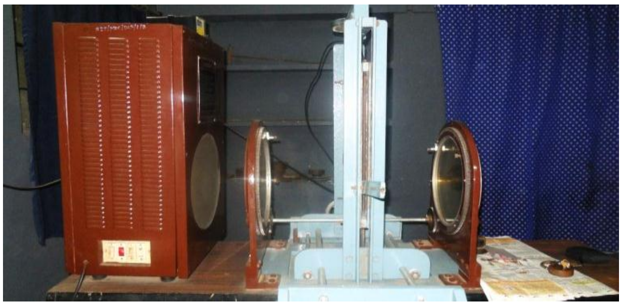


Fig.4 Transmission polariscope

3.4 Material Fringe Value

The material fringe value $F\sigma$ is defined as number of fringes produced per unit load. The material fringe value is the property of the model material for a given wavelength (λ) and thickness of the model (h). Here the circular disk subjected to dimensional compressive load is employed as a calibration model.

3.5 Procedure to Find Material Fringe Value

The circular disc of diameter 50 mm and thickness 6mm was used to find material fringe value. This circular disc was loaded under compression by special fixture. Acompressive load of 167.68 N was applied to find materialfringe value. This circular disc was also subjected to samestress freezing cycle as that for the model.

Viewing through dark-field of circulated Polar scope, locked isochromatic fringe pattern was observed as shown in fig.

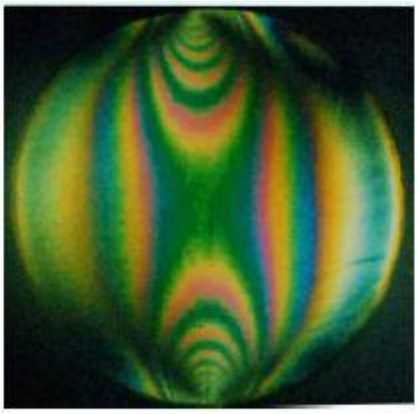


Fig.5 Fringing Pattern

By using following equation, the material fringe value $F\sigma$, at critical temperature were found out.

$$F\sigma=\frac{8P}{\pi DN}$$

Where,
P= Load applied= 167.68 N
D= Diameter of Disc= 30.8
N= Fringe order observed at the centre of disc=1.115
Substituting these values in above equation

$$F\sigma=\frac{8X167.68}{3.14X30.8X1.115}, F\sigma= 12.433 \quad (4)$$

3.6 Photo elasticity experiment

Specimens of gear tooth were manufactured having the root radius 1,2,3,4, respectively of 1:4 scale. Loading arrangement is done as shown in fig. the maximum stresses were determined experimentally for each tooth less than 2kg of loading.

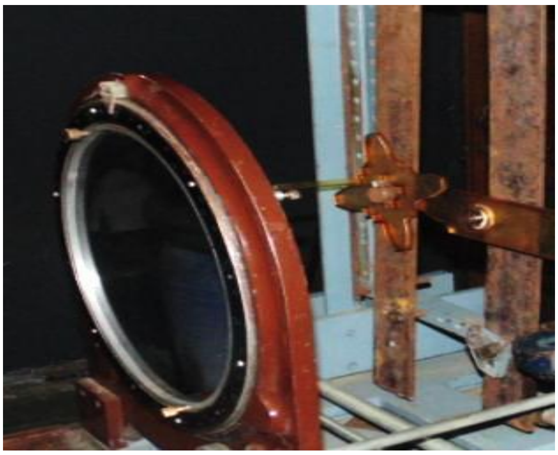


Fig.6 Loading arrangement of gear

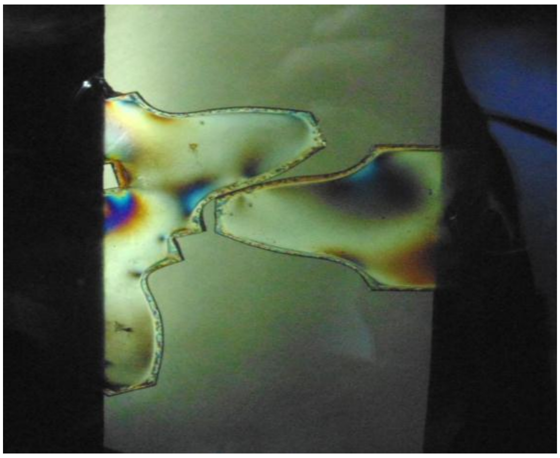


Fig.7 Fringe pattern

When the load applied on the model then the fringes are formed on gear tooth profile as shown in fig corresponding Fringe value is noted down.

Following Table shows the corresponding fringe value for each root radius specimen.

Table 3 Result comparisons

Sr .No.	Gear Module	Root Radius	Fringe Value (N)
1	4.5	1	0.41
2	4.5	2	0.35
3	4.5	3	0.30
4	4.5	4	0.27

By using equation (3) Calculate the stresses on model.

$$\frac{\sigma m}{\sigma p} = \frac{Pm}{Pp} \times \frac{hp}{hm} \times \frac{lp}{lm}$$

Where,

$$\sigma m = \frac{N \times F\sigma}{h} F\sigma = 12.41 \quad \text{(From...eq 4)}$$

$$= \frac{0.27 \times 12.41}{7} = 0.53$$

$$\frac{\sigma m}{0.53} = \frac{2225}{52.32} \times \frac{10}{7} \times \frac{40.60}{10.15} \quad \text{(from..... eq 3)}$$

$$\sigma m = 115.62$$

Similarly we calculate the σm for different fringe value we get the following table.

$F\sigma$= 12.41

Table 3 Max Principle Stress

Sr. No.	Root Radius (mm)	Fringe Value (N)	Max Principle Stress σm
1	1	0.41	175.57
2	2	0.35	149.88
3	3	0.30	128.02
4	4	0.27	115.62

4. Result & Discussion

Comparisons between the Ansys result with Photo elastic technique shown in Table 4. The result shows the corresponding Max Principle stress value nearly closed.

Table 4 Result comparisons

Sr. No.	Root Radius	ANSYS Calculated	Photo-elastic Experimental	% Difference
		Max Principle stresses		
1	1	181.18	175.57	3.12
2	2	148.96	149.88	0.62
3	3	120.39	128.02	6
4	4	109.15	115.62	5.37

Conclusions

Comparisons of results of Ansys and photo-elastic method. Spur gear having the root radius 4 produces minimum bending stress and result shows that Bending stresses at contact region decreases with increase the root radius. Also conclude that Ansys result is close agreement with result come from by Photo-elastic technique and hence Photo elastic method is cost effective technique of the stress analysis.

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