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Dynamic Analysis Of Helical Gear To Evaluate Bending Stress At Root Of Tooth Using Photoelastic Coating And Finite Element Method

Prof.P.J.Patil[1], Amol A.Patil[2], Santosh S. Nulke[3]

[1] *Project Guide, Department of Mechanical Engineering , TKIET Warananagar,(India)*

[2] *PG Student, Department of Mechanical Engineering , TKIET Warananagar,(India)*

[3] *PG Student, Department of Mechanical Engineering ,SETI Panhala ,(India)*

ABSTRACT

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. Because these stresses differ so greatly in nature, they deserve to be considered separately. The stresses produced under dynamic loading conditions in machine member and structural elements differ considerably from those produced under static loading. In practice, one encounters dynamic loading conditions more frequently than static loading situations. Mathematical analysis of dynamic problems are in generally involved more than those of static problems. When the design member under consideration is complex in geometry, the analytical approach to dynamic problems become too involved. In such situation, an experimental approach may give a direct and more appropriate solution.

Keywords : Bending stress, Dynamic Analysis, Photoelastic stress analysis, PS-I, Reflection Polariscope.

I. INTRODUCTION

The increasing demand for quite power transmission in machines, vehicles and generators, created a growing demand for a more precise analysis of the characteristics of gear systems. In automobile industry, the largest manufacture of gears, higher reliability and lighter weight gears are necessary as lighter automobile continue to be in demand. In addition, the success in engine noise reduction promotes the production of quieter gear pairs for further noise reduction.

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. Because these stresses differ so greatly in nature, they deserve to be considered seperately.The stresses produced under dynamic loading conditions in machine member and structural elements differ considerably from those produced under static loading. In practice, one encounters dynamic loading conditions more frequently than static loading situations. Mathematical analysis of dynamic problems are in generally involved more than those of static problems. When the design member under consideration is complex in geometry, the analytical approach to dynamic problems become too involved. In such situation, an experimental approach may give a direct and more appropriate solution.

The tooth of gear is subjected to fluctuating bending stresses as it comes in contact with the meshing tooth. Since in gear drives, teeth are subjected to fluctuating stresses. Endurance limit stress is the criterion for design. Therefore maximum bending stress is equal to the endurance limit stress of the gear tooth. For gear tooth the endurance limit stress depends upon terms such as,

1. Surface finish of gear tooth.
2. Size of gear tooth.
3. Stress concentration in gear tooth.
4. Directionality of gear rotation.
5. Reliability used in design.

As, in practical sense, it is difficult to quantify the above parameter for each and every case of gear design, we go with Earle Buckingham suggestion and assume the value of endurance limit stress to be one third of the ultimate tensile strength of gear material. In gear design the maximum tangential force due to maximum torque is the criterion. The value of the tangential component depends upon the rated power and rated speed. In practical applications, the torque developed by the source of power varies during the work cycle. Also torque required by the driven machine also varies. When gears rotate at very low speed say almost at zero velocity, the transmitted load (P_t) can be considered to be the actual force presented between two meshing teeth. However, in most of the cases, gears rotate at an appreciable speed and thus it is necessary to consider the dynamic force. There are two methods to account for the dynamic load. The velocity factor method is an empirical relationship derived by past experience. This method has certain disadvantages such as:

1. Dynamic load depends upon a number of factors such as the mass of gear, properties of gear material like modulus of elasticity. The velocity factor method neglects these factors. It assumes that dynamic load depends upon pitch line velocity.
2. Use of velocity factor is restricted to a limited range of pitch line velocities. It is not possible to extrapolate the values.

Dynamic load calculated using second Earle Buckingham method is far more than the corresponding load calculated by velocity factor method; very often it is three to four times the load (P_t) due to power transmission. In most of practical cases, the dynamic load is less than that of calculated by Buckingham method. This method is mainly applicable to large gears rotating at moderate speeds. The actual dynamic load is less in cases such as:

1. Small gears transmitting large power.
2. High speed light load gears.
3. Small gears on small shafts.
4. Gear shafts up to 50 mm diameter.
5. Gears transmitting power less than 15 kW.

For improved reliability and higher endurance we must have precise and clear knowledge of the gear tooth stress field during meshing. 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.Dynamic structural analysis is an essential aspect of engineering design, but one that is often difficult to perform. Standard finite element method can predict areas of failure to a certain degree of accuracy, but these methods often make assumptions that may not be applicable to a real life situation. Dynamic stress analysis can lead to a adjustments during the design stage creating both economical and efficiency benefits for all aspects of engineering.

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 photoelasticity to visually represent stress progression during dynamic operation has the potential to create significant efficiency and economical benefits in all aspect of engineering and industry.

In this paper, bending stress analysis at root of tooth of helical gear was studied experimentally and using finite element method.

II.LITERATURE REVIEW

There has been great deal of research on gear analysis and a large body of literature on gear analysis has been published. The gear stress analysis, prediction of gear dynamic loads, gear noise and the optimal design for gear sets are always major concerns in gear design. In the past, some works on experimental analysis of gear and photoelasticity field have been reported in the literature review. Less work has been done on dynamic analysis of gear. The literature review in these area was carried out. Reputed journals and the transactions of the professional societies referred for this purpose.

1) **I M Allison** et.al [1] (1980) have carried out an extensive two-dimensional photoelastic study of the basic deformation behavior of gear teeth under load. Limitations of existing bending-strength design procedures are used to formulate a test program which considers the magnitude of the critical root-fillet stresses, the effects of varying the load position on the flank, the effects of friction forces at contact and the relationship between bending and shearing deformations.

2) Jose L F Freire et.al [3] (1985) has explained details of photo elasticity technique. It is a branch of photo mechanics .the photo elastic response consists of two families of fringes – isochromatic and isoclinic. Photoelasticity may be applied to models in the laboratory or to prototype in the field, as well as to 2D and 3D studies .It is whole field technique. Photo elasticity indicates not only the most loaded areas of the observed component, but also provide accurate stress values at any critical point.

3) **Prashant Patil** et.al [31] (2011) carried out bending stress analysis of helical gear used in PIV gear box using 3D photoelastic experimental method and FEA method. The accurate evaluation of stress at root of tooth is complex task. Photoelastic stress analysis technique using stress freezing is the best method for experimental stress analysis. Also low temperature epoxy mixture is the good material for casting photoelastic models and only longitudinal slices are helpful for determining bending stress pattern at tooth root.

4) **F. Trebu!a** et.al [37] (2012) have provide a new approach of use of software in Photostress technique. software application PhotoStress makes the analysis of principal strain and principal normal stress directions and magnitudes on photoelastically coated objects faster. The foundations of this improvement lie in automatic

processing of photoelastic entities such as isoclinic fringes, singular points and isochromatic fringes. The application is currently subjected to some improvements regarding little imperfections that arise during automatic projection of isostatic curves and recognition of colourful isochromatic fringes or surfaces.

III.PROBLEM STATEMENT

Dynamic analysis by conventional method has certain disadvantages. Velocity factor and buckinghams equation method causes over design of the component . So to find reliable approach for dynamic analysis.

Gear specifications :

Module (m) = 3mm

Pressure Angle (Φ) = 20^o

No. of teeth (t) on Gear and Pinion = 35

Helix Angle (ψ) = 10^o

IV.THEORETICAL DYNAMIC ANALYSIS

The analysis of bending stress in gear tooth was done by Mr. Wilfred Lewis known as Lewis equation. In the Lewis analysis, the gear tooth is treated as a cantilever beam and the tangential component (P_t) causes the bending moment about the base of tooth. The Lewis analysis is based on the following assumptions:

- The effect of radial component (P_r) is neglected.
- The effect of stress concentration is neglected.
- At any time only one pair of teeth is in contact and takes the total load.

Case I

Gear Specifications:

Module (m) = 3mm

Pressure Angle (Φ) = 20^o

No. of teeth (t) = 32

Helix Angle (ψ) = 10^o

Power (P) = 1 Kw

Speed (n) = 150 rpm

S_b= m b σ_b Y ……………Lewis equation

1) Torque Transmited

$$M_t = \frac{60 \times 10^6 (kw)}{2\pi n}$$

$$M_t = \frac{60 \times 10^6 (1)}{2\pi (150)}$$

$$M_t = 63661.97 \text{ Nmm}$$

2) Tangential Component on tooth (P_t) is given by

$$P_t = \frac{2M_t}{d_p}$$

$$d_p = \frac{32(3)}{\cos(10)}$$

$$d_p = 97.48 \text{ mm}$$

$$\therefore \quad P_t = \frac{2 \times 63661.97}{97.48}$$

$$P_t = 1306.15 \text{ N}$$

3) Effective load (P_{eff}) on gear tooth is given by :

$$P_{eff} = \frac{C_s P_t}{C_v}$$

Now,

v = pitch line velocity

$$v = \frac{\pi d_p n_p}{60 \times 10^3}$$

$$v = 0.765 \text{ m/s}$$

Since,

$$C_v = \frac{5.6}{5.6 + \sqrt{v}}$$

$$C_v = \frac{5.6}{5.6 + \sqrt{0.765}}$$

$C_v = 0.864$ and $C_s = 1.1$

$$\therefore \quad P_{eff} = \frac{1.1 \times 1306.15}{0.86}$$

$$P_{eff} = 1670.59 \text{ N}$$

But, $S_b = P_{eff} \times f(s)$

$$m_n b \sigma_b Y = P_{eff} \times f(s)$$

$$\therefore \quad \sigma_b = \frac{P_{eff} \times f(s)}{m_n b Y}$$

Now,

Virtual no. teeth,

$$z'_p = \frac{z_p}{\cos^3 \varphi}$$

$$z'_p = \frac{(32)}{\cos^3(10)}$$

$$z'_p = 33.5$$

Y = 0.369 .. Lewis form factor from table

$$\sigma_b = \frac{1670.89 \times 2}{3 \times 24 \times 0.369}$$

$$\sigma_b = 62.87\ \mathrm{N/mm^2}$$

The same procedure has been carried out for other cases and results are tabulated in result table.

V.PHOTO STRESS ANALYSIS

5.1 Selection of Material :

The selection of photostress coatings and their proper application to the test part are most essential to the successes of photostress analysis. A wide range of coating materials is available in both flat sheet and liquid form for application to metals, concrete, plastics and other materials.An ideal photostress coating should have

1. High strain optical coefficient.
2. Linear stress- strain and strain- fringe relationship.
3. Easy bondability to various materials.
4. Sufficient pliability to permit use on curved surfaces of intricate components.

You will not have all these properties available in a single material, so you have to do compromise. For whole field analysis of gear we select the material PS-I (Having K = 0.15 and E =2.5 Gpa) which is available in pre manufactured sheet as surface of test part to be coated is flat. We selected the coating thickness 3 mm that was necessary for producing fringes for easy measurement and also there is no problem of reinforcement effect as material of test specimen is steel.

5.2 Design and Development of Loading Fixture :

As aim is to analyze the bending stress at root of gear tooth in actual working condition that is whole field analysis. We developed the loading fixture which simulate the actual working condition of gear pairs, the developed loading fixture is as shown in fig. no. 5.2.1. We applied the torque with the help of pulley rope dynamometer. DC 1 kW motor was selected as a prime mover and speed was vary with the help of DC dimmer. For easy assembly and disassembly the gear shafts were fixed in pillow bearings. The shaft diameter was 32 mm and keyway was 6mm × 3mm.

FIG.NO. 5.2.1 LOADING FRAME

5.3 Bonding of Coating :

5.3.1 Sawing :

First we prepared the templates of all pinion profiles from gear models. Then with the help of using these templates we cut the photoelastic coating in particular gear shape using saber saw. Fig. no.4.3.1.1 shows photoelastic coating cut in gear profile shape. A set of coarse, medium and fine files including round and half round geometries are used for finishing. After coating has machined to its final shape it was thoroughly cleaned. The cleaning was carried out with isopropyl alcohol using a sponge.

FIG.NO.5.3.1.1 SAWING PHOTOSTRESS SHEET

5.3.2 Surface Preparetion:

The intent of surface preparation is to develop a chemically clean surface with a roughness sufficient to promote good adhesion.

i) Degreasing and preliminary cleaning:

The surface which to be coated was cleaned by washing it down with acetone or isopropyl alcohol.

FIG.NO.5.3.2.1 CLEANING OF GEARS

ii) Abrading and Conditioning:

In this step the surface was conditioned by lapping lightly with abrasive paper of 120 or 220 grit or scotch bright wetted with conditioner. After the surface was prepared it was washed with acetone and wiped dry using paper towels.

iii) Adhesive Preparation:

Preparation of adhesive for bonding the photoelastic coating sheet to the test part required very careful attention. In our case we selected PC-1C adhesive for bonding PS-1 photoelastic sheet. The amount of adhesive required was calculated in advance , generally one gram of mixed adhesive will cover approximately 10 cm^2 area. For PC-1C proportion was 10 gram of hardener for 100 gram of resin that is 10 pph.

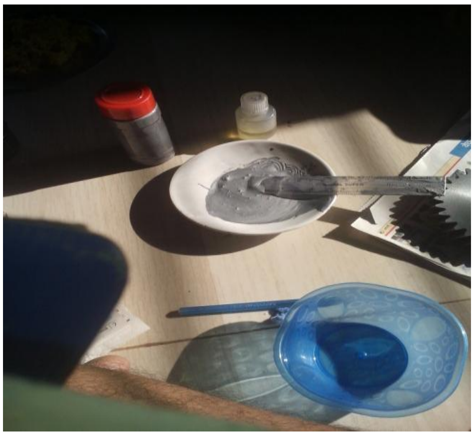

FIG.NO. 5.3.2.2 ADHESIVE PREPARATION

5.3.3 Bonding :

a)Pour the mixed adhesive onto the cleaned test part surface and spread it over the entire surface with a brush as shown in fig. no.5.3.3.1. Brushing is best accomplished with a mildly stiff brush, which serves to wet the surface as well. The adhesive layer should be at least 1 mm thick.

FIG.NO.5.3.3.1 APPLICATION OF ADHESIVE

b). Carefully position the flat sheet and allow one edge to contact the spread. Press down on the contacting edge with moderate finger pressure and slowly work additional contact toward the opposite edge as shown in Fig.no.5.3.3.1. This technique allowed the air to flow out along with the excess adhesive. After full contact was made, additional finger pressure wae applied. Start near one edge and slowly progress across the sheet to bleed additional adhesive from beneath. Re-peat this several times, taking care to brush excess adhesive in contact along all edges of the sheet.

FIG.NO.5.3.3.3 APPLYING FINGER PRESSURE

c) After all of the excess adhesive is squeezed out, apply a thin coating around all edges of the plastic (including holes that may have been drilled) to provide a seal against moisture absorption

d) Allowed the adhesive to cure for 12 Hrs. at room temperature.Fig. no.5.3.3.2 shows coating bonded to steel gear.

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FIG.NO.5.3.3.4 COATING BONDED TO GEAR

5.4 Measurement Of Fringe Order :

Reflection polariscope (Model No.) with white light was used for fringe measurement. The experimental set up is as shown in fig. no. X. For full field interpretation of fringe pattern stroboscope is used to made rotating gear teeth stationary during measurement of fringe order. For different values of torque we recorded the fringe order at root of tooth by using model 832 compensator. The fringe order recorded was as shown in fig. no X and listed below .

Case I:0.70

Case II: 0.59

Case III: 0.51

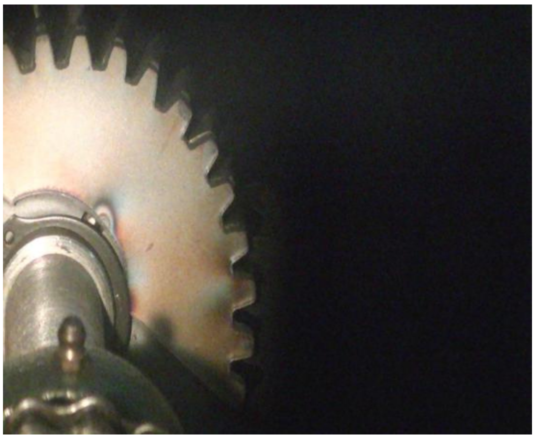

FIG.NO. 5.4.1 EXPERIMENTAL SET UP **FIG.NO.5.4.2 FRINGE ORDER OBSERVED**

5.5 Measurement Of Bending Stress at Tooth Root :

The fringe orders observed in PhotoStress coatings are proportional to the difference between the principal strains or principal stresses in the coating or in the surface of the test part and calculated using equation as below

$$\sigma_x - \sigma_y = \frac{E}{1+\upsilon} N_f$$

Where,

σ_x- σ_y = principal stresses in test part

E = Modulus of elasticity of test part material

ν = Poisson's ratio of test part material

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N = Fringe order

f = fringe value of the plastic coating, (m/m) per fringe

In our case the value of stress in y direction is zero that is $\sigma_y = 0$

$\sigma_x = \sigma_b = \frac{E}{1+v} \times N \times f$ …… Eqn.1

As modulus of elasticity of coating material is very less as compared with modulus of elasticity of gear material also thickness of test gear is large as compared with thickness of coating there were no need of correction factor.

Fringe value, $f = \frac{\lambda}{2tk}$ ……. Eqn.2

Here,

t = coating thickness = 3mm

λ = Wavelength of tint of passage in white light, taken as 575 nm

K = strain-optic coefficient of the photoelastic plastic (dimensionless) = 0.15

Put these values of t, λ and k in equation number 2,

We get,

$f = 638.88 \times 10^{-6}$

For Steel : $E = 2\times10^5$ N/mm^2 and $v = 0.3$

For case I: N= 0.51

Therefore, from equation no. 1

$\sigma_b = \frac{2\times10^5}{1+0.3} \times 638.88 \times 10^{-6} \times N$

$\sigma_b = 49.78$ N/mm^2

For case II: N= 0.43

$\sigma_b = 41.64$ N/mm^2

For case III: N= 0.37

$\sigma_b = 35.67$ N/mm^2

VI. FINITE ELEMENT ANALYSIS

Finite Element Analysis is done in LS-DYNA.

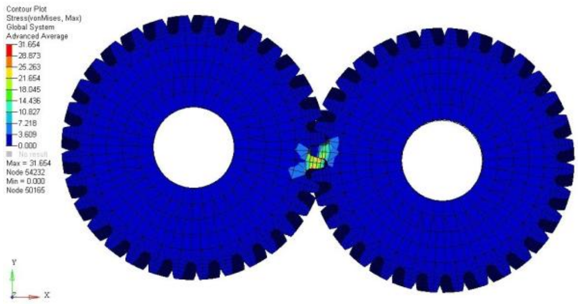


FIG.NO. 5.4 STRESS DISTRBUTION AT FULL ENGAGEMENT

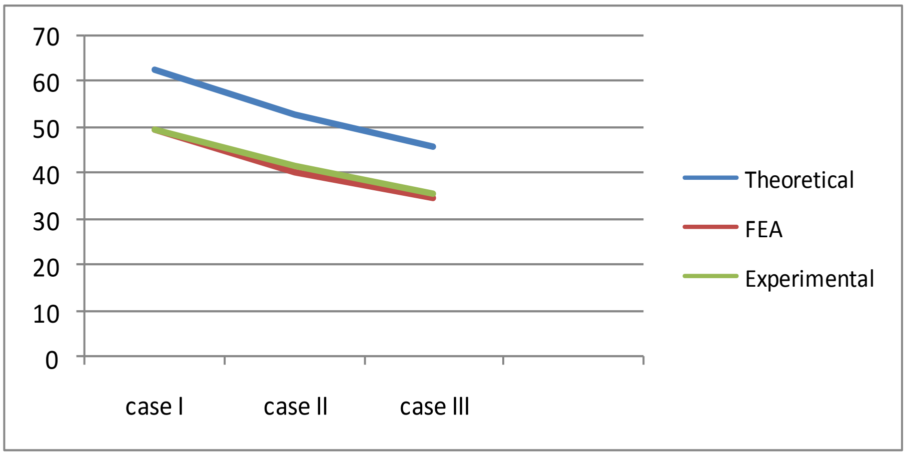
VII.RESULT

The results obtained by theoretical, FEA and Experimental approach for all three cases is tabulated in the result table.

Case no.	Theoretical	FEA	Expt.	% variation in Theoretical and FEA	% variation in FEA and Experimental	% variation in Theoretical and Experimental
I	62.87	48.52	49.78	22.82	2.53	20.82
II	53.00	40.49	41.64	23.60	2.76	21.43
III	45.96	34.64	35.67	24.63	2.88	22.38

Table 7.1 RESULT TABLE

If we compare Experimental and FEA bending stress at root of helical gear tooth. There is fairly good agreement between experimental and FEA results. The maximum error observed is 3%. While error between FEA and Theoretical is 24%. The error between Theoretical and Experimental is 22%. The graph is ploted of Torque Vs bending stress.



VIII. CONCLUSION

The Photostress method is reliable method for dynamic analysis of bending stress at root of tooth of gears. PS-I material of Vishay precision is best suitable material for Photostress analysis of steel component. If we compare FEA and Experimental results with theoretical results. The actual bending stress is far less than bending stress calculated by theoretical method.

In actual working condition bending stress developed at root of tooth of gear is less than those calculated by theoretical method because the tangential component develops less stress when tooth is in dynamic condition than tooth is in static condition. The bending stress is maximum at the start of engagement between two meshing teeth. We can optimize the width of gear as bending stress level is far less than in dynamic condition. It is possible to reduce weight to torque ratio of gear box for same power transmission capacity.

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