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Design and Analysis of Helical Elliptical Gear using ANSYS

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ABSTRACT: The characteristics of a helical elliptical gear system including bending stresses and the contact stresses are examined using 2-D Finite Element Method (FEM) models. The bending stresses in the tooth root were examined using a 3-D FEM model.

Keywords: Helical gears, Elliptical gears, Bending stress, Contact stress, FEM, 2-D modeling, 3-D modeling

I. INTRODUCTION

Non circular gears are used to transmit variable motions in oil industries, textile industries, printing presses etc. Non –circular gear replaces cam and linkages due to following advantages:
-These are more compact as compared to cam and linkages.
-These are nonlinear and balanced.
-These can produce continuous unidirectional cyclic motion
-Cams offer only reciprocating motion.
The Helical elliptical gears which fall under the category of non –circular gear are used, where irregular rotational motion requirements are involved. Cams and linkages can provide these special motion requirements, but helical elliptical gears often represent a simpler, more compact and more accurate solution.Irregular rotational motion is characterized by a repetitive increase and decrease inrotational speed of output shaft for each revolution. These gears can accurately and economically produce irregular rotational motion. The crossed link mechanism is kinematically equivalent to an elliptical gear. All the undesirable features found in crossed link mechanisms are eliminated when an elliptical gear is used.

II. DESIGN EQUATION OF HELICAL ELLIPTICAL GEAR BASED ON BENDING FATIGUE FAILURE

The fig. 1 illustrates the conjugate pair of identical ellipses.

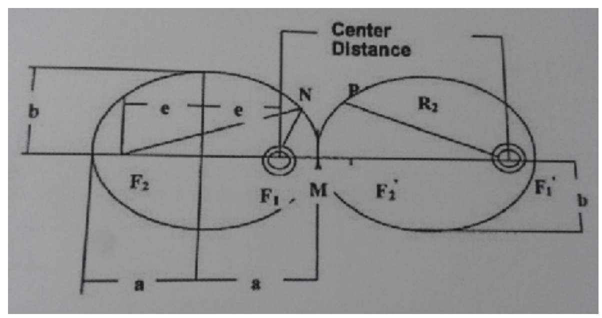

Fig. 1. Pair of identical ellipses.

They rotate in opposite directions, one around its focus F1 and the other around corresponding focus F1 when the gear rotate through angle Θ_1 and Θ_2 respectively, all points on arc MN come in successive contact with points of the equally long and identically shaped arc MP. Basic rule is the R_1+R_2 must constant. From symmetry F_1P (which is R_2) equals F_2N; therefore distance F_1N+F_2N is constant, which is the definition of ellipse. From this condition the instantaneous angular velocity$\frac{d\Theta 2}{dt}$ of output gear can easily be found if the constant input velocity ω is given. The instantaneous velocity ratio $\frac{d\Theta 2}{dt}$ i.e. ω is determined by the inverse ratio of the instantaneous radii vectors.

$$\frac{d\Theta 2}{dt}=\omega\frac{R1}{R2}$$

From the polar equation of ellipse

$$R1=a\frac{(1-\epsilon 2)}{1+\cos\Theta 1}$$

Where ϵ (the numerical eccentricity of the ellipse) is defined as

$$\epsilon = \frac{\sqrt{a^2 - b^2}}{a}$$

This equation can be written as

$$\epsilon = \frac{e}{a}$$

Where, e is the distance from focus to the center of the ellipse.

From $R_1 + R_2 = 2a$

It is found that

$$R_2 = 2a - \frac{a(1-\epsilon 2)}{1+\epsilon \cos \Theta 1}$$

$$R_2 = a\frac{1+2\epsilon \cos \Theta 1 + \epsilon 2}{1+\epsilon \cos \Theta 1}$$

$$\frac{d\Theta 2}{dtmin} = \omega \frac{1-\epsilon 2}{1+\epsilon 2 + 2\epsilon \cos \Theta 1}$$

The extremes occur at $\Theta_1 = 0°$ and 180°, with $\cos\Theta_1 = -1$, Thus by substituting for ϵ, the minimum velocity at $\Theta_1 = 0$:

$$\frac{d\Theta 2}{dtmin} = \omega \frac{a-e}{a+e}$$

And for $\Theta_1 = 180$;

$$\frac{d\Theta 2}{dtmax} = \omega \frac{a+e}{a-e}$$

Maximum and minimum angular output velocities are the reciprocal of each other. If K is the ratio of maximum to minimum, then

$$K = \frac{d\Theta 2/dtmax}{d\Theta 2/dtmin}$$

$$k = \frac{(a+e)2}{(a-e)2}$$

It is good practice to keep k under 5 to ensure smooth running without "whip". Design usually starts with the choice of Centre distance 2a, in other words, with the size of gears. If the major semi axis a, is given and a certain overall ratio k is desired, the minor axis b can be found from:

$$b = \frac{2a \times k(1/4)}{1 + k(1/2)}$$

where a = major semi axis, b = minor semi axis, e = distance from focal point to geometric Centre of ellipse, ϵ = numerical eccentricity, F = focal point and Centre of rotation of ellipse, k = ratio of velocity change of output gear, N = number of teeth, R= instantaneous radius, Θ = instantaneous angle of rotation, p = radius of approximating contour of ellipse, ω = constant angular velocity of input subscript 1 and 2 relates to input and output elliptical gear respectively.

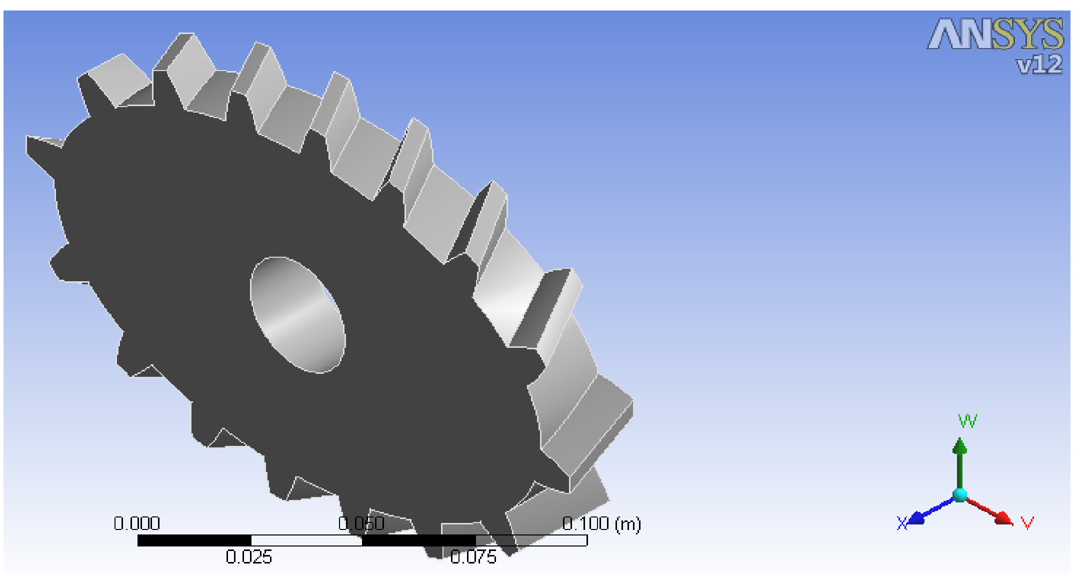

Fig. 2. 3-D Model of the gear having 16 teeth.

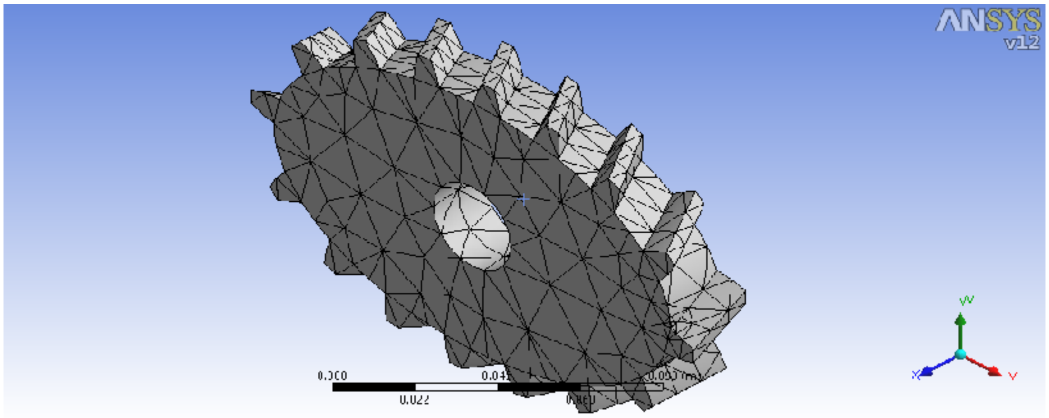

Fig. 3. 3-D FEA Model for 16 teeth Gear.

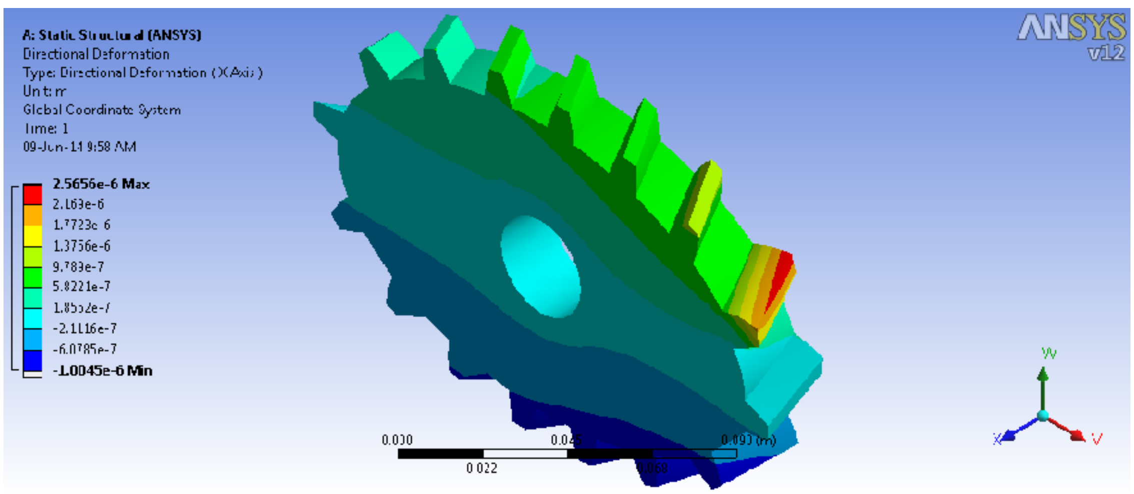

Fig. 4. Directional Deformation.

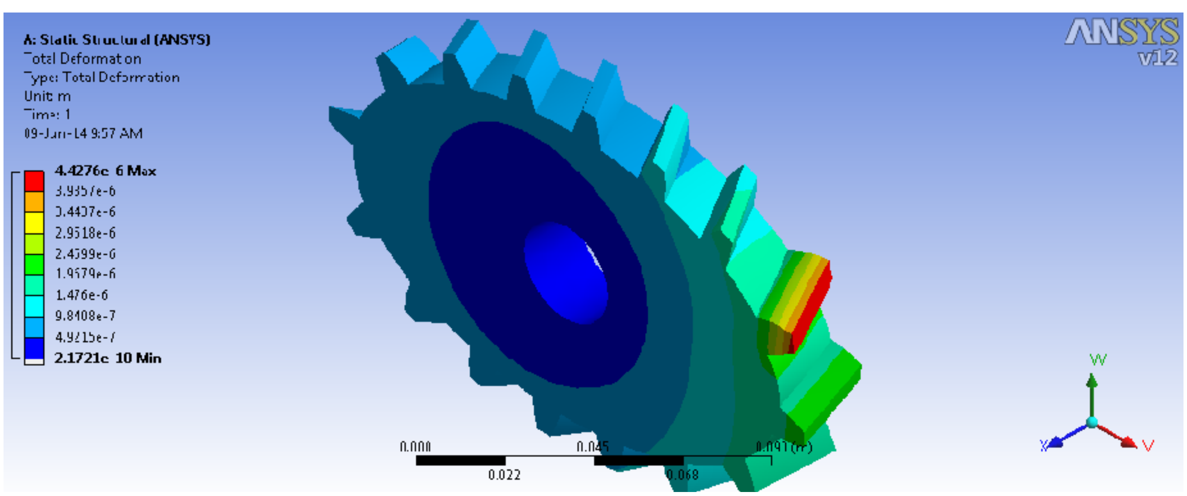

Fig. 5. Total Deformation.

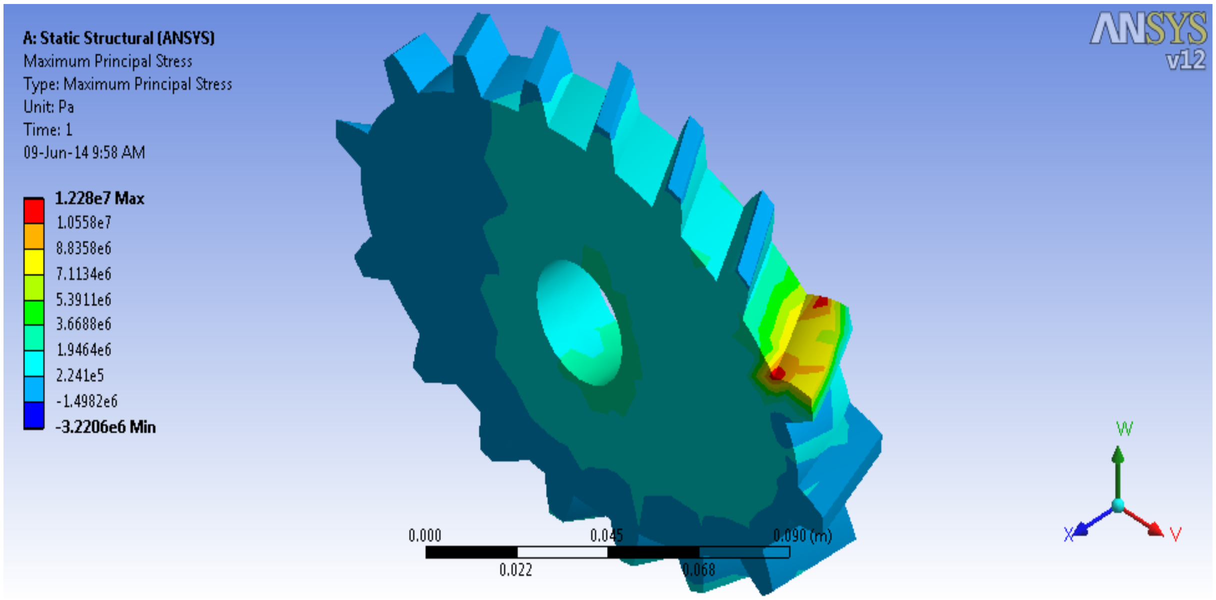

Fig. 6. Maximum Principal Stress.

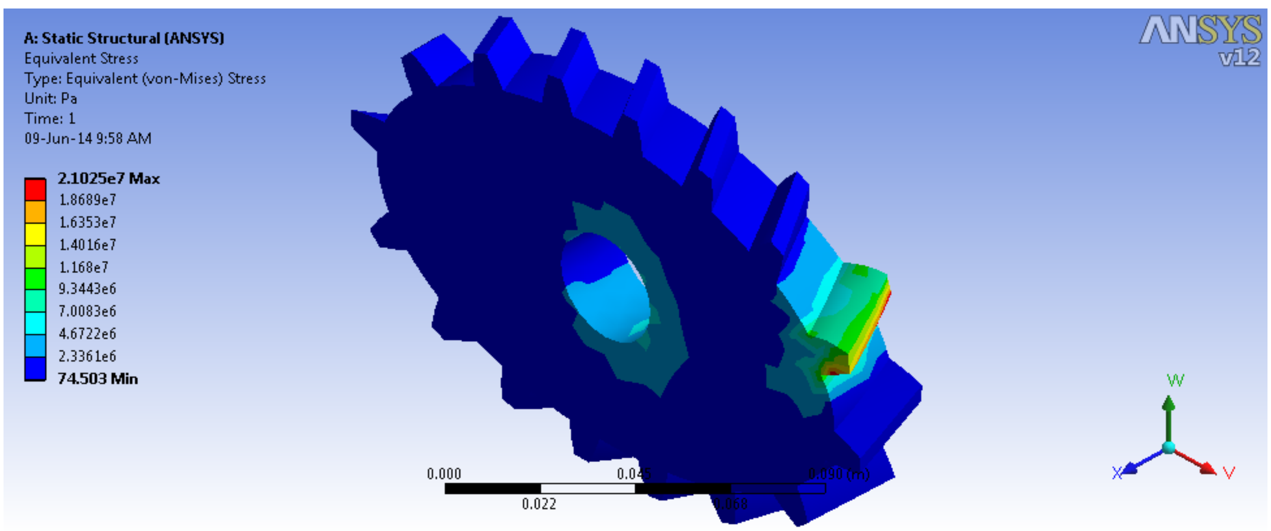

Fig. 7. Equivalent Von-Mises Stress.

III. CALCULATION OF STRESS USING LEWIS EQUATION

The stresses are calculated by Lewis formula using standards of AGMA is carried out, and the other coefficients, such as the dynamic factor, are set at 1.1. Here analysis of gears with different numbers of teeth is carried out. First, the number of teeth is 16. The meshing helical gear has pitch radii of 60.5mm and a pressure angle of 20 degree. The face width = 10 mm, transmitted load is 1020 N.

$$P_d = \frac{N}{d} = \frac{16}{121} = 0.132\, mm^{-1}$$

$$\sigma_t = \frac{F_t \times P_d \times K_a \times K_s \times K_m}{b \times y_j \times K_v}$$

$$= \frac{1020 \times .132 \times 1.1 \times 1.1 \times 1.13}{10 \times 0.5 \times 1.5} = 24.54 MPa$$

K_a = 1.1 (application factor), K_s = 1.1 (size factor), K_m = 1.13 (load distribution factor), K_v = 1.5 (dynamic factor), F_t = 1020N (Normal tangential load), Y_j = 0.5 (geometry factor)

Table1: Comparison of stresses calculated by Lewis Equation and ANSYS.

Number of Teeth	Stress 3D (ANSYS)	Stress (LEWIS EQ.)	Difference in Stress
16	21	24.5	3.5

IV. CONCLUSION

The FEA model is used to simulate contact between two bodies accurately by verification of contact stresses between two helical elliptical gears in contact and comparison is made with the results of AGMA analysis. The average difference of results between ANSYS and AGMA approach are very small and equal to 3.5%.It was shown that FEA model could be used to simulate bending between two bodies accurately. Effective methods to estimate the tooth contact stress using a 2D contact stress model and to estimate the root bending stress using 3D FEA models are proposed.

REFERENCES

[1]. Igor Zrebski, Tadeusz Salanciski "Designing of Non-circular Gear" Warsaw University of Technology, Faculty of Production Engineering; Narbutta Str.8502524 Warsaw, Poland.

[2]. Leon Kowalczyk, Stanislaw Urbanek, "The geometry and Kinematics of a toothed gear of variable motion", Technical University of Lodz Department of Mechanics of Textile Machines Poland.

[3]. Professor R.A. Khan, "Optimum synthesis of Non Circular gear incorporating tolerances", Department of Mechanical Engineering Jamia Millia Islamia New Delhi.

[4]. Faydor L. Litvin, "Design and investigation of Gear drives with Noncircular gears applied for speed variation and generation of functions" Department of Mechanical and Industrial engineering, University of Illinois at Chicago, United States.