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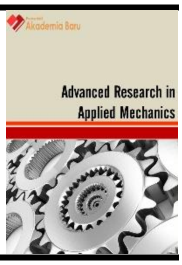

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Comparison of Bending Stresses for Different Face Width of Helical Gear Using AGMA and ANSYS

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ABSTRACT

Helical gears are widely used in industry where the power transmission is required at heavy loads with smoother and noiseless operation. The design of helical gear is complex requiring superior software skills for modelling and analysis. This problem is resolved by using Pro/Engineer and ANSYS environment that provide equivalent results to the AGMA. In this paper, helical gear was modelled using Pro/Engineer wildfire 4.0 and stress analysis of the helical gear was carried out with ANSYS 11.0. The results of FEM and AGMA procedureswere compared. The results indicated that as the face width of the gear increased the bending stress decreased. At face width of 80 mm and 100 mm bending stresses were 7.1941 MPa and 7.0256 and 5.7553 MPa and 5.4856 MPa for AGMA and FEM respectively. The differences in prediction of these stresses between the AGMA and FEM were 2.3% and 4.7% respectively.

Keywords:
Helical Gear, Modeling, Pro/Engineer, Bending Stress, Face Width, AGMA, FEM

1. Introduction

Gearing is one of the most effective methods for transmitting power and rotary motion from the source to point of application with or without change of speed or direction [1]. Gears will prevail as a critical machine element for transmitting power in future machines due to their high degree of reliability and compactness [2]. At present helical gears are being used as a power transmitting element owing to their relatively smooth and silent operation, large load carrying capacity and higher operating speed [3]. Designing highly loaded helical gears for power transmission systems that are good in strength and low level in noise necessitate suitable analysis methods that can easily be put into practice and give useful information on contact and bending stresses [4]. The finite element method (FEM) is proficient to furnish this information but the time required to generate proper model is a large amount [5]. Therefore, to reduce the modeling time a preprocessor technique that builds up the geometry required for FEM analysis might be used, such as Pro/Engineer (Pro/E). Pro/E can generate three-dimensional models of gears. In Pro/E, the generated model is saved as a file and then transferred to ANSYS for analysis [6]. Gear analysis can be performed using analytical methods

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which required a number of assumptions and simplifications which aim at getting the maximum stress values only but gear analyses are multidisciplinary including calculations related to the tooth stresses and failures like wear [7]. Due to the advancement in computer technology, many researchers tended to use numerical methods to develop theoretical models to calculate the effect of whatever is studied [8]. Numerical methods are capable of providing more solution that is truthful since they require very less restrictive assumptions [9]. However, the developed model and its solution method must be selected attentively to ensure that the results are more acceptable and its computational time is reasonable [10].

In 1992, Rao and Muthuveerappan [11] explained that the geometry of helical gears by simple mathematical equations. A parametric study was made by differing the face width and the helix angle to look into their effect on the root stresses of helical gears. Chengand Tsay [12] studied the contact and the bending stresses of helical gear set with localization contact by means of finite element analysis (FEA). Mathematical models of the complete teeth geometry of the pinion and the gear have been derived based on the theory of gearing. Accordingly, a mesh generation program was as well developed for finite element stress analysis. The computerized design, methods for generation, simulation of meshing as well as enhanced stress analysis of modified involute helical gears was conducted by Litvin *et al*., [13]. The methods proposed for modification of traditional involute helical gears were based on conjugation of double – crowned pinion with a conventional helical involute gear. Zhang et al. [14]in their research on analysis with varying mesh stiffness, proposed a new approach to analysis of the loading and stress distribution for spur and helical gears. Face width and helix angle are significant geometrical parameters in determining the state of stresses during the design of gears. Zhang and Fang [15] presented an approach for the analysis of teeth contact and load distribution of helical gear with crossed axis. The approach was based on tooth contact model that accommodate the influence of tooth profile modifications, gear manufacturing errors and tooth surface deformation on gear mesh quality. Hedlund and Lehtovaara [16] presented a study that focused on the modeling of helical gear contact with tooth deflection. They introduced a mathematical model for the computation of tooth deflection that include foundation flexibility, shearing and tooth bending. The proposed helical gear set comprises an involute pinion and double crowned gear. Thus, in this work a parametric study is conducted by varying the face width to study their effect on the bending stress of helical gear using Pro/E and ANSYS for modelling and stress analysis respectively.

2. Methodology

Five different sets of gears were modeled in Pro/E Wildfire [17], having the following parameters as showing in Table 1 below. The material for the gear is structural steel. The following steps show the procedure to model the gear of 30 numbers of teeth with the combination of the all the mentioned parameters in the Pro/E, other parts of the gear are modeled in similar methods.

PART parameters menu in PRO/E is a section where the basic parameters of modeling the gear are defined. These PART parameters determine all the other parameters that define the gear tooth profile, by using the Tools/Relation menu. Figure 1 shows the PART parameters menu.

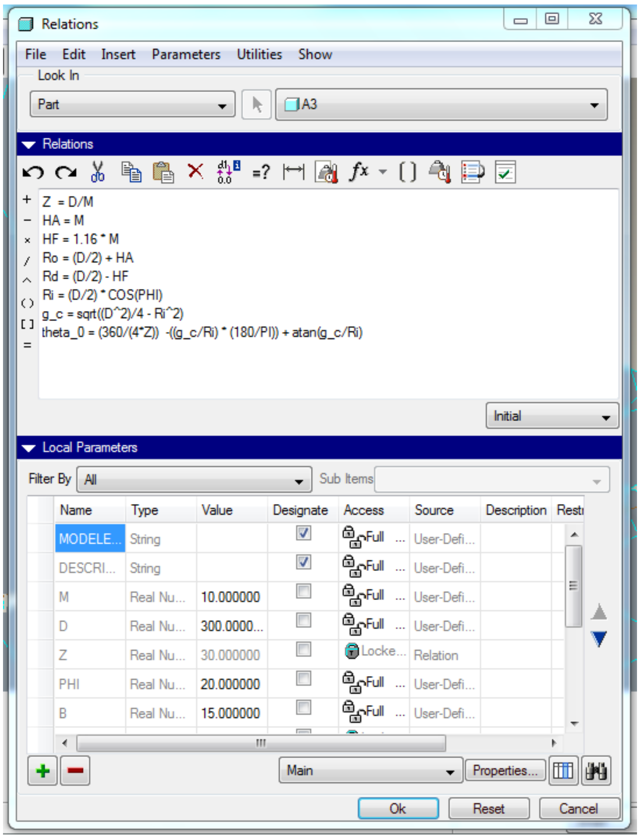

Fig. 1. Tools Relation/parameter menu

Table 1
Gear parameter

Description	Specifications
No. of teeth (Z)	30
Module (m)	10
Pitch circle diameter (D)	300
Face Width (b)	50
Pressure angle (α)	20°
Helix angle (β)	15°
Addendum (ha)	10
Dedendum (hf)	1.16*m
Modulus of elasticity (E)	2E+ 05 MPa
Poisson's ratio (υ)	0.3
Power (P)	15kW
Speed (N)	1200 rpm

Bending Stress calculation

The bending stress equation for helical gear teeth is given by [18]

$$\sigma_b = \frac{F_t}{bmJ} K_V K_O \ (0.93 K_m) \tag{1}$$

Introduction of constant 0.93 with the mounting factor reflects slightly lower sensitivity of helical gears to mounting conditions. All the calculations are done based on Eq. (1) as recommended by American Gear Manufacturers Association (AGMA) [19].
where,
F_t = Force transmitted = 1591.53 N
b =Face width = 100 mm
m= Normal module = 10 mm
J= Geometry factor = 0.56
Kv= Velocity Factor = 1.34
K_O = Overload factor = 1.25
Km = Load distribution factor = 1.3

Tangential force calculation
P= Power = 30 KW
Speed = 1200 RPM

$$Torque\ (T_P) = \frac{9549.3 \times P(kW)}{speed\ (RPM)} = 238.73\ N.$$

$$Tangential\ Force\ (Ft) \frac{2000 \times T_P}{Pitch\ diameter} = 1591.53N$$

Bending Stress Calculation (σ_b AGMA):

For face width *b*= 100 mm

$$\sigma_{b_{AGMA}} = \frac{1591.53}{100 \times 10 \times 0.56} \times 1.34 \times 1.25(0.93 \times 1.3) = \ 5.7553\ MPa$$

Static structural Analysis

The structural analysis of the helical gear tooth model was done using the finite elements analysis in ANSYS 11.0. The load was applied at the tooth of the helical gear. The mesh with the default settings is not adequate to get the accurate results. Therefore, both gears were finely meshed with 'Sizing' option in the menu. The element size chosen was 0.001 and it was then refined at the bending surfaces to get the finer mesh and continuous stress values. Figure 2 below shows the meshed gear according to the size selected. The mesh looks fine enough for the analysis. By applying the force over the tooth that is facing equivalent load, the stress distribution was obtained in the numeric as well as in the form of color scheme. By varying, the face width and keeping the other parameters constant various models of the helical gear were created. Figure 3 and Figure 4 represent the finite element analysis for the various models created by varying face width.

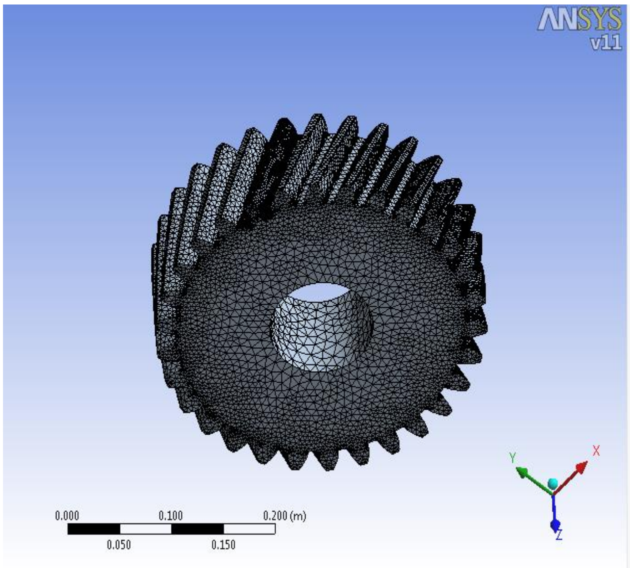

Fig. 2. Meshed 3-D Model of Helical gear

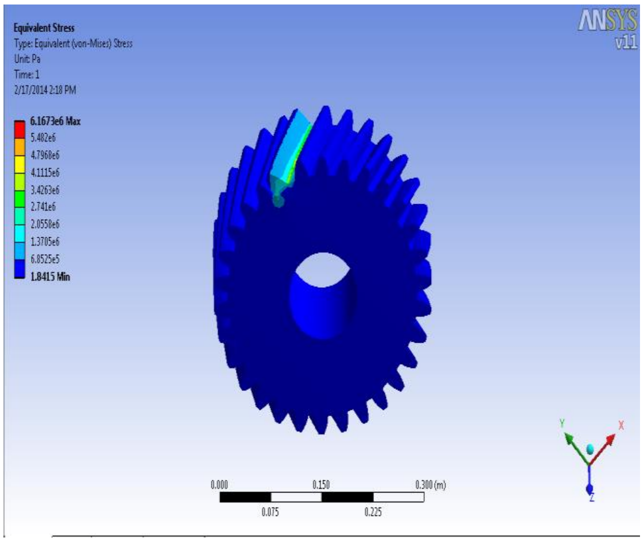

Fig. 3. Static analysis of gear with 30 teeth face width = 95 mm

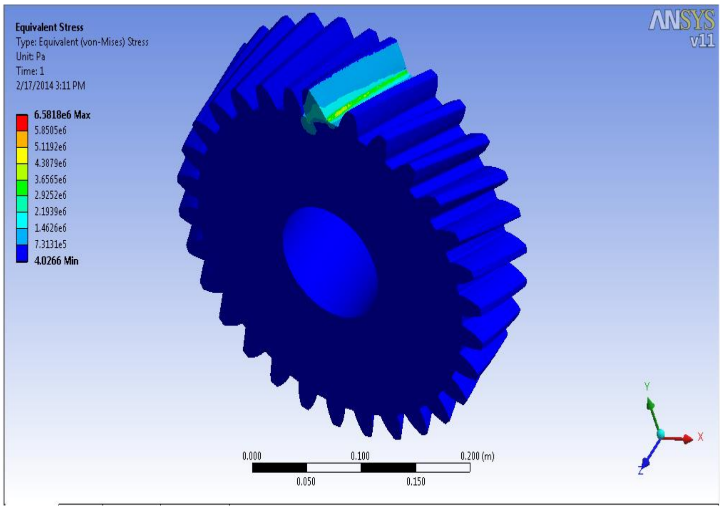

Fig. 4. Static analysis of gear having 30 teeth (face width =85 mm)

3. Results

For determining the stresses at any stage during the design of gears, face width is an important parameter. To determine the stress variation with the face width, various models of helical gear were made by keeping other parameters *i.e.* number of teeth, helix angle etc. constant. Table 2 shows the results of bending stress of both AGMA and ANSYS with the variation of the face width of the helical gear tooth.

Throughout the analysis each gear is studied for five different face width (b=80mm, 85mm, 90mm, 95mm, 100mm) the other parameters and the applied load were kept constant. The maximum bending stresses of the modelled gear as shown in Figure 2 have been compared with those in Table 2, and the values calculated from AGMA Eq. (1). The results showed consistency between the numerical (AGMA), predicted (FEM). Table 2 and Figure 5 clearly show variations in face width from 80 mm to 100 mm, and there is continuous decrement in the value of stress at the tooth of the helical gear as the face width increased. For constant load and speed, the greater face width is suitable.

Table 2
Bending Stress (MPa)

S/N	Facewidth(mm)	AGMA (MPa)	ANSYS (MPa)	Differences (%)
1	80	7.1941	7.0256	2.34
2	85	6.7709	6.5818	2.80
3	90	6.3948	6.4873	1.44
4	95	6.0582	6.1673	1.80
5	100	5.7553	5.4856	4.70

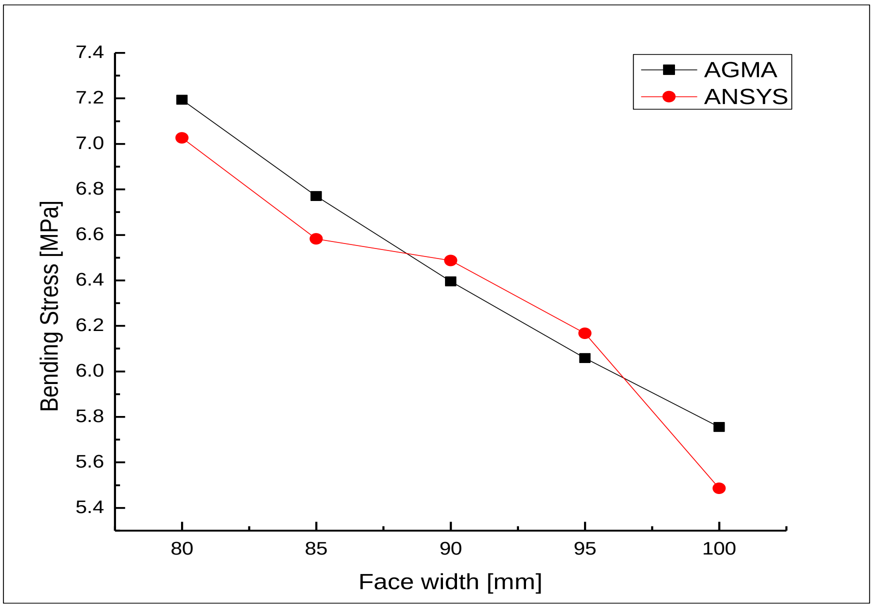

Fig. 5. Graph of Bending Stress [MPa] against Face width [mm]

4. Conclusions

The results obtained from ANSYS are close to the results obtained from AGMA procedure. From the results, it is justified that ANSYS can be used for predicting the values of bending stress at any required face width, which is much easier to use to solve complex design problems.

We can conclude that the results showed consistency between the numerical (AGMA) and predicted (FEM). Table 2 and Figure 5 clearly show variations in face width from 80 mm to 100 mm, and there is continuous decrement in the value of stress at the tooth of the helical gear as the face width increased. Therefore, for constant load and speed, the greater face width is suitable.

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