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Bending Stress Analysis of Natural Fibre Reinforced Composite Gears

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ABSTRACT

Spur gear is an essential element in mechanical power transmission systems. While the gears have been made of metals in most application, the gears in recent years have been made of composite materials to take advantage of the specific strength and low weight of the composites. Due to harsh loading condition at contact line of the gear, it is subjected to failure due to bending and contact stress. In this study, bending stress analysis was conducted on composite spur gears using finite element method (FEM). SolidWorks software was used to design the 3D models of gears before the models were exported to the FEM software of ANSYS. Within ANSYS Workbench, bending stress analysis was firstly conducted on ANSI 1040 steel gear where the results were then validated with the stress values obtained using the American Gear Manufacturing Association (AGMA) formulation and past results. The validation study showed an excellent agreement between the three values of the finite element analysis (FEA) result, the AGMA formulation and the past result. Following this, stress analysis was conducted on composite gear with Epoxy resin while applying 3 kinds of fibre namely Kenaf, Pineaple leaf fibre (PALF) and E-Glass. The effect of the face width of the gear on the bending stress was studied by varying the face width values to 30, 35 and 40 mm. The results indicated that each three fibre type gave similar values of maximum bending stress. However, the ultimate tensile strength failure will first go to Kenaf, followed by PALF and E-Glass fibre. Furthermore, the values of stress can be seen to decrease with the increase of face width.

1. Introduction

Gear system is anticipated as the most efficacious way of power transmission in future machines as a result of their high reliability and compactness. A gear is a rotating circular machine part having teeth that mesh with another toothed part. It is to transmit an amount of rotational power, determined by adjusting the torque and rotational speed of the gear. While there are four primary types of gear i.e. helical, worm, spur and bevel that cover a wide range of applications, spur gear is the most widely used gear as it comes with high accuracy and produced through relatively easy production processes. An important design aspect of a spur gear is to avoid failure due to excessive stress that may come from bending and contact stresses. While failure due to contact stress or pitting

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is a surface fatigue failure due to repetitions of high contact stresses at the same point on the gear for a long period, this study focused on the bending stress that is due to the act of the tangential force at the contact line of the gear.

A spur gear has been treated as a cantilever by Lewis in deriving the basic bending stress formula for spur gear where tangential load, W^t is assumed to act at the end of the cantilever such as shown in Figure 1. This formula is the basis for the commercially used American Gear Manufacturing Association (AGMA) stress formula that adds many factors to the formula in order to make the formula closed to reality. These factors include overload, dynamic and size factors. On the other hand, in the last decades finite element method (FEM) has been conveniently used to deliver the task of determining the bending stress in the gear as the method can always handle complicated structures. Sabri *et al.*, [2] conducted a study on the bending and pitting stress of a helical gear used in a 5-speed car transmission system using FEM. The helical gear was modeled in 3D involute form. The results of the FEM modeling and the AGMA formulations were found to be in good agreement. A study on bending stress analysis of a spur gear compared the results from Lewis equation and finite element analysis (FEA) [3]. It was found that the change of pinion material C15 steel to C45 steel while maintaining CI 30 as the gear material gave lower value of bending stress. Furthermore, the results showed that the bending stress reduces with increase in module of the gear system.

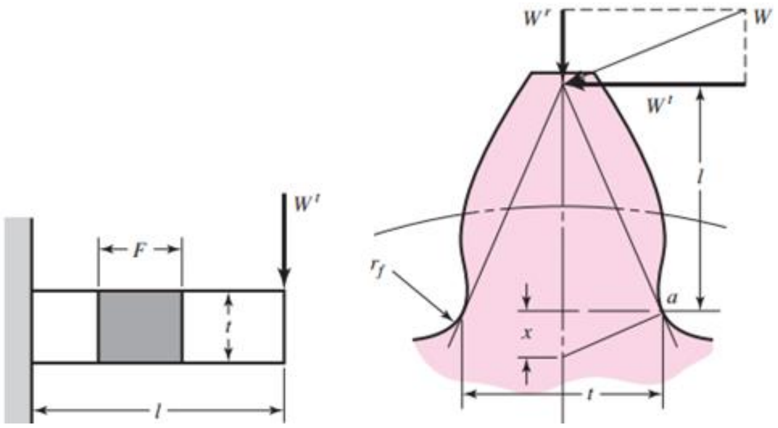


Fig. 1. A gear tooth treated as a cantilever [1]

While the stress in the gear is determined by its cross-sectional properties and the amount of the applied load, the strength of the gear depends on the material used. In the last decades, as the issues of environment and sustainability loom, the consideration on advanced and suitable materials that includes smart material [4], carbon nanotube [5] and natural fibre reinforced composite [6,7] to be applied in industries such as energy [8-11], aerospace [12] and manufacturing [13] has been accentuated. While various types of steel and synthetic fibre reinforced composites have been widely used for gear material, the use of natural fibre reinforced composites that offer low environmental impact, high specific strength (low weight) and price competitiveness has increasingly important in the realm of engineering.

While there are researchers that studied the improvement made by replacing steel gear with carbon fibre reinforced composite gear [14,15], several researchers have also been dedicated to study gears made from natural fibres including bamboo [16], sisal [17], flax [17], birch (wood) [18], nettle [19] and coconut coir [20]. A fully bamboo made gear was proposed and its characteristics such as its hot press molding conditions, the tooth bending strength, root strain and vibration due to meshing teeth were studied [16]. One of the finding was the hot pressing of the large and small bamboo fibers could be performed a wide range of temperatures with an appropriate molding time.

A study on the bending strength of a gear tooth made of coir fibre reinforced composite was conducted using FEA [20]. An experimental study by Dhanushkudi and Arunagri [17] on glass, sisal and flax fiber reinforced composite gears showed that the natural fiber-reinforced gears can be used for smooth and quiet operation especially in intermittent applications.

In this study, bending stress analysis was conducted on natural fibre reinforced composite spur gear using FEM. Two natural fibres used in this study were Kenaf and Pineapple leaf fibre (PALF) while E-glass fibre was used for comparison purpose. The 3D model of the gears was produced using SolidWorks software before the models were exported to the FEM software of ANSYS. The bending stress analysis was firstly conducted on the high carbon ANSI 1030 steel gear for validation purpose. Validation was conducted by comparing the three maximum bending stress values produced from the present FEA analysis, AGMA formulation and past result. Following this, the analysis was further conducted on natural fibre reinforced composite where comparison to bending static strength was made for each gears and calculated. Furthermore, a parametric study was then conducted in studying the effect of face width of the gear on the maximum bending stress of the gears.

2. Methodology

2.1 Geometry of the Gear

The chosen geometry of the gear with specifications is shown in Table 1. Following this, the gear system of pinion and gear designed through SolidWorks software is then shown as in Figure 2.

Table 1
The spur gear geometric specification

Parameter	Pinion	Gear
Pitch diameter (mm)	54	150
Module, m	3	3
Pressure angle (°)	20	20
No of teeth	18	54
Face width, b	30, 35, 40	30, 35, 40

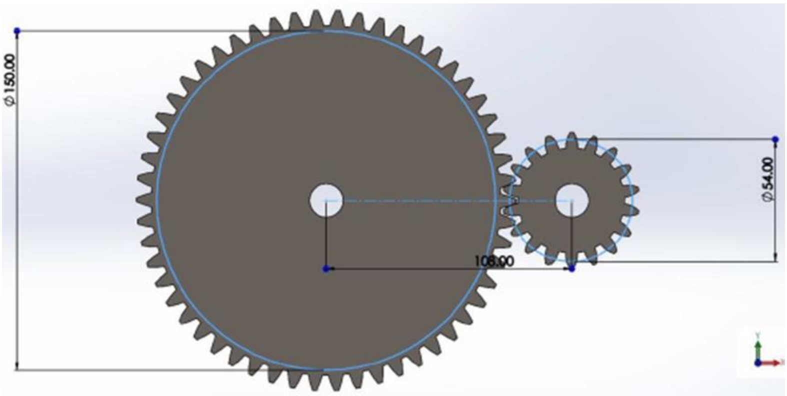


Fig. 2. The gear system

2.2 Material Properties

In this study, a steel and composite gears were used. The study applied 3 types of fibres namely Kenaf, PALF and E-glass while Epoxy was used as the resin. Furthermore, the steel was of the high carbon ANSI 1040 type. The relevant properties of each of material constituent are shown in Table 2. The occurrence of fracture failure due to the applied stress reaches over the ultimate tensile strength of the composite was estimated for Kenaf-Epoxy composite. The tensile strength properties of the Kenaf-Epoxy composites were taken from an experimental study that used two parameters Weibull distribution to give high reliability values of tensile strength of 15 % and 45 % kenaf reinforced epoxy composites at 60 MPa and 102 MPa, respectively such as given in Table 3.

Table 2
Material properties of the gear material constituents [21]

Material	Density (g / cm^3)	Young's modulus (GPa)	Shear modulus (GPa)	Poisson's ratio	Tensile strength (MPa)
Kenaf	1.5	4.3	1.63	0.32	223
Palf	1.07	4.405	2.20	0.3	126.6
E-Glass	2.6	7.3	29.92	0.22	2000
Epoxy	1.15	3.76	1.35	0.39	52
ANSI 1040	7.854	190-210	80	0.27-0.3	620

Table 3
Tensile strength of Kenaf reinforced composite [22]

Material	Shape parameter	Scale parameter (MPa)	Mean tensile strength (MPa)	
			Weibull distribution	Normal distribution
Pure Epoxy	18.29	33.79	32.82	32.90
15 % kenaf / epoxy composites	7.54	59.69	56.05	56.21
45 % kenaf / epoxy composites	9.46	101.83	96.64	96.93

2.3 Gear Design

SolidWorks was applied to model the pinion and gear in 3D. The decision to use 3D model comes after it was decided to use 3D solid element in the FEA to be described in the next sub-section. The modelling process began with the design of each gear tooth's 2D involute curve. After completing both models, they were integrated into SolidWorks' assembly drawing and meshed together. The mate procedure in SolidWorks was used to mesh both pinion and gear in order to rotate both gears while maintaining contact between their teeth. To guarantee that no errors occurred during the meshing process, the SolidWorks software included a function that allowed researchers to check whether there was little interference between the gears and a motor that can be attached to the pinion gear to verify that both gears were meshed and rotated smoothly. The SolidWorks created 3D models and are saved in either STEP or IGES format before being exported into ANSYS Workbench. In this study, all 3D models were kept in STEP format so that Direct Modeler module in ANSYS could access them. The 3D gear models provided the geometric modelling of the gear system which was necessary prior to the creation of the FEM modelling in ANSYS.

2.4 Stress Analysis

FEM was used through ANSYS software to determine the maximum bending stress of the gears. FEA required the division of the developed geometric model into so called elements. In this study, it was decided to use 3D solid element. Inside ANSYS, upon inserting the values of material properties, boundary conditions and loading, a FEM governing equation was developed for each element and later all equations produced from each element were assembled to form a global governing equation of the form in Eq. (1),

$$\{F\} = [K]\{d\} \tag{1}$$

where, $\{F\}$ and $\{d\}$ are vectors for forces and displacements, respectively and $[K]$ is the stiffness matrix of the structure that represents the geometry and material properties distribution throughout the structure. Upon solving this equation in the so called Static analysis in ANSYS, the displacement vector, $\{d\}$ can be determined and as such in the post-processing stage, stress and strain for each node can be obtained.

In the Design Modeler of the ANSYS workbench, once the geometric model has been imported and loaded into the static structural tab of the new mechanical analysis tab, the next step was to define contacts between two involute tooth profiles. The computer-aided design (CAD) application automatically detects the associated geometry for predetermined contact. The contact among two teeth was stated as having no separation in the connections section.

In the meshing step, the blending of mesh was necessary as the ANSYS's default meshing option was inadequate for generating exact results. The importance of refined meshing was critical in determining the most accurate results. Refined meshing or mesh quality was indicated by the element's skewness which can be determined by selecting the mesh metrics type of skewness by clicking the statistics option beneath the mesh tab. The skewness value reflects a face's or cell's proximity to the ideal. For the meshing developed in Figure 3, the average skewness range derived from the study resulted in ranges between 0.25 and 0.40, resulting in a relatively high cell quality meshing process.

At applying boundary condition stage, frictionless support was applied to the inner rim of the pinion and gear. The load in the form of a moment of 136.42 Nm was applied to the surface body of the gear in the clockwise direction, as shown in Figure 4.

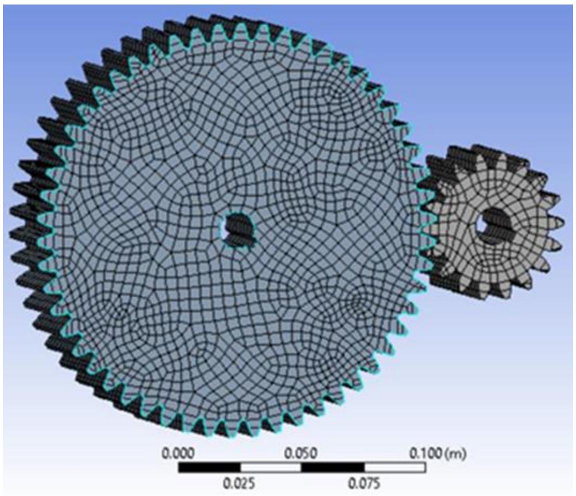


Fig. 3. The meshing of the gears

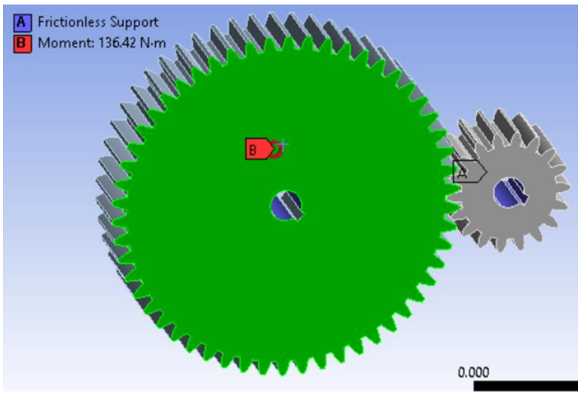


Fig. 4. The loading and boundary condition

2.5 The AGMA Formula

The results of the maximum bending stress, σ_b from the FEA was compared with the formula provided by the AGMA as in Eq. (2),

$$\sigma_b = W^t\, K_o K_v K_s \,\frac{1}{bm_t}\frac{K_H K_B}{Y_J} \tag{2}$$

where, b and m_t are the face width of the narrower member and the metric number, respectively, W^t is the tangential load (N), K_o, K_v and K_s are the overload, dynamic and size factor. K_H, K_B and Y_J are the load-distributed, rim-thickness and the geometric factor, respectively. For the values of these factors, readers were referred to the book of Budynas and Nisbett [1].

3. Results and Discussion

3.1 The Bending Stress of Steel Gear

A bending stress study was conducted on steel gear having face width of $b = 40$. Results from the FEA were compared with the results applying the AGMA formula and past result [23]. Table 4 showed that the result from the current study was very close to those of the AGMA and the past result.

Table 4
The maximum stress from 3 analyses

Face width, ***b***	AGMA (MPa)	Khudhair [23] (MPa)	Present (MPa)	% difference
40	24.42	25.64	25.93	0.201

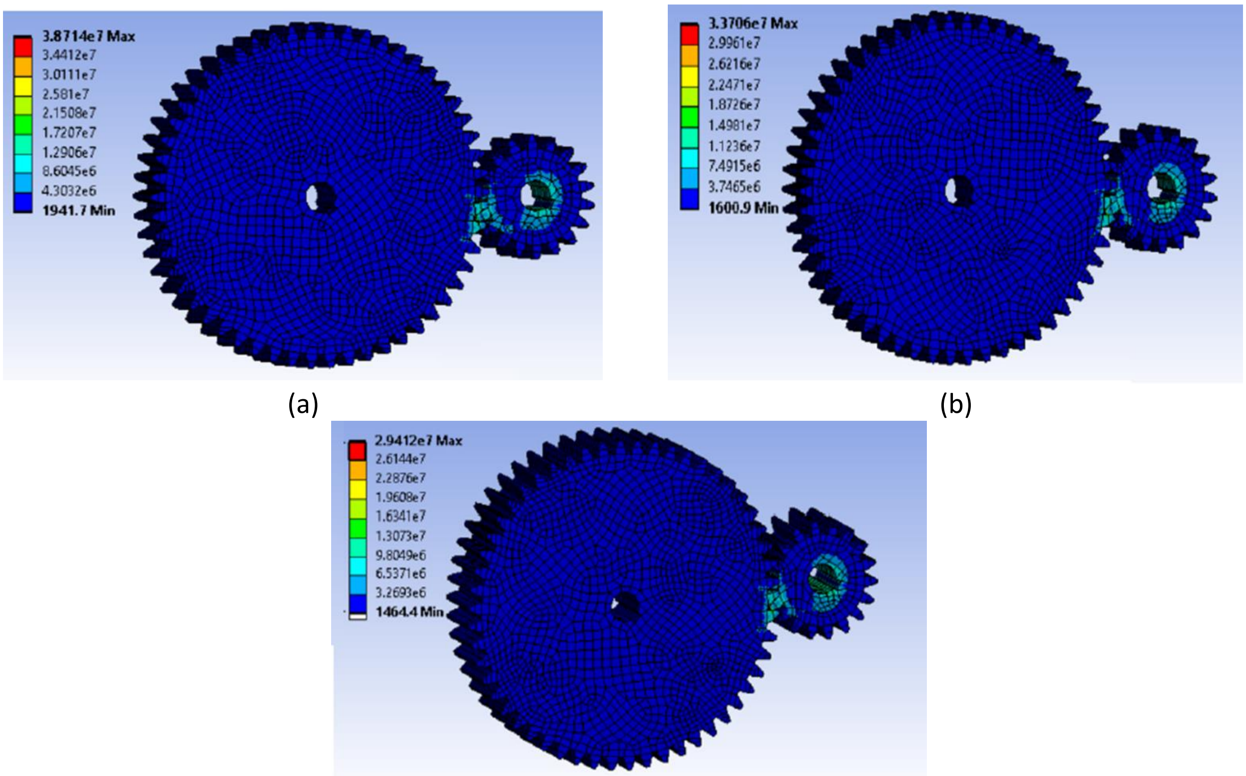
3.2 The Bending Stress of Composite Gears

The bending stress analysis of fibre reinforced composite gears with epoxy resin was conducted using FEA. Three types of fibres used were Kenaf, PALF and E-glass fibres. Table 5 shows the results of the maximum bending stresses correspond to the 3 reinforced fibres for 3 values of face widths. The table shows very similar results for each fibre type which shows that the effect of material on bending stresses was minimum. On the other hand, it can be seen that the stress values for each fibre type decreased as the face width was increased. This can be explained using the AGMA formula of bending stress that the stress is indirectly proportional to the face width.

Table 5
The maximum stress from analysis of three face width values

Face width, ***b***	Kenaf-Epoxy (MPa)	PALF-Epoxy (MPa)	E-Glass-Epoxy (MPa)
30	38.71	39.02	38.84
35	33.71	33.96	33.81
40	29.41	29.66	29.56

Similar distributions of stress on the Kenaf-Epoxy gears upon application of load can be seen in Figure 5 (a) - (c) which correspond to face width, b=30, 35 and 40 mm, respectively. Obviously, the maximum stress cannot be seen from the diagram as it occurs at the contact point between the two gears.



(a) (b)

(c)

Fig. 5. Bending stress distribution of Kenaf-Epoxy composite gear with face width, b as (a) 30 mm (b) 35 mm and (c) 40 mm

The bending stress and its corresponding factor of safety (FS) of Kenaf-Epoxy composite for each face width are given in Table 6. FS was calculated as FS = Strength / Stress. It showed that the FS can reach up to 3.46 for 45 % Kenaf-Epoxy having 40 mm face width.

Table 6
Factor of safety for Kenaf-Epoxy composites

Face width, ***b*** (mm)	Bending Stress (MPa)	FS for 15 % Kenaf	FS for 45 % Kenaf
30	38.71	1.54	2.63
35	33.71	1.77	3.02
40	29.41	2.03	3.46

4. Conclusions

Bending stress analysis was conducted on fibre-Epoxy composite gears using FEA. Three types of fibres applied were Kenaf, PALF and E-glass. For validation purpose, the bending stress analysis was conducted on ANSI 1040 steel gear. The result of the maximum bending stress using FEA was in excellent agreement with the maximum stress values coming from the AGMA formulation and past results. The stress results corresponding to the 3 types of fibres were much closed to each other confirming that material has no or little effect on stress value. At the same time, the effect of face width can be seen to be indirectly proportional to the maximum bending stress of the gear.

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