DESIGN OF A REAL-SIZED COMPOSITE DRIVE SHAFT AND CRITICAL POINTS FROM BEGINNING TO END

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Abstract

In this study, the replacement of a metal drive shaft by a composite counterapart was investigated. The drive shaft considered here is a thin walled tube with an internal diameter of 60 mm and a length of 1200 mm. These dimensions correspond to the dimension of a regular metal drive shaft. Filament winding method was used to manufacture the composite shafts. The number of plies and the orientation angle are selected as variables to minimize the mass of the composite shaft under constraints of torsional stiffness, strength, vibration and buckling. Torsional behavior of composite drive shaft was simulated using Finite element method (FEM). The shafts manufactured were tested by using torsion test set-up. The torsional stiffness of the composite shafts was higher as compared to the metal shaft. In the composite shafts, adhesive failure occurred at the interface between the composite tubes and the steel inserts before the failure of the composite shaft itself. Thus, connector geometries between the shaft and the inserts was also investigated to prevent premature failure.

1. Introduction

Automotive industry tries to reorient its way to have lighter, less polluting and more fuel-efficient vehicles, which enforce to manufacturers to use lighter materials. Fiber reinforced composite materials (FRCMs) is one of the options to fulfill this demand. During the last decade, automobile manufacturers have started to use FRCMs for the vehicle components such as floor panel, bumper beam, spoiler, etc. The drive shaft of a vehicle can be another component to be manufactured using FRCM.

The choice of reinforcement material, manufacturing method and joining methods of the composite to the yokes have great importance in composite drive shaft applications. Hybrid glass/carbon fiber reinforced [1-3] and hybrid aluminum/composite drive shafts [4, 5] have been studied as an alternative material combination to carbon or glass composite drive shafts [6]. In order to increase the ductility of drive shafts, Sevkat et al. [1] considered mixed glass and carbon reinforcement for hybrid shafts. Badie et al. [2] investigated the effect of fiber orientation and stacking sequence of glass and carbon layers on the torsional stiffness, natural frequency, buckling strength, fatigue life and failure modes of composite tubes. Similar findings were reported that carbon fibers have the major advantage over glass fibers in improving the torsional stiffness. In addition, the orientation angle of 45^{o} was found as the best angle that increase the torsional stiffness. In the optimization study of Montagnier and Hochard [7] reported same comment of that 45^{o} plies maximize the torque resistance and 0^{o} plies maximize the axial stiffness and minimize the axial damping. Filament winding and vacuum infusion have been considered methods for manufacturing of composite drive shafts. Braided fabric, fabric wrapping are used in the vacuum infusion method [8]. In the study of Lee et al. [4], a carbon fiber epoxy composite layer was co-cured on the inner surface of an aluminum tube, which eliminates the possibility of the impact damage and the moisture absorption. Maximum torsional stiffness of 4320 Nm was measured and buckling of aluminum was observed without the delamination failure of composite and the failure of protrusions on the press fitted joint. Joining methods of the composite shafts and the yokes have been investigated in addition to the design of composite shaft itself. Mechanical joining and adhesive bonding are mainly used joining methods. Choi and Lee [9] and Kim et al. [10] investigated tubular lap joints as a joining method. Kim et al. found that bonding length should be larger than 16 mm to have the torque capacity of 3500 Nm as the diameter of the shaft was 90 mm. In addition to experimental studies, numerical models were developed to predict the torsional stiffness [1, 2], natural frequencies [1, 2], and torsional buckling [2, 6, 7]. In the models, the stress-strain relationship was assumed to be linear and elastic. Tsai-Wu and maximum stress criterion were considered in the study of Montagnier and Hochard [7] and it was found that the maximum stress criterion show better agreement with experimental data than those obtained using Tsai-Wu criterion when calculating the failure strength. In order to investigate the torque transmission capability of tubular joint, Kim et al. [10] developed a model using finite element method to predict the bonding length and yoke thickness.

In this study, a real sized one-piece composite drive shaft was manufactured with carbon fiber epoxy and tested. The tubular joint was considered to join the shaft and the inserts to prevent premature failure.

2. Experimental Study

2.1. Materials, manufacturing and testing

The 24k Carbon filaments (HS carbon fiber-24k) and medium viscosity resin were used to manufacture the composite drive shafts bu filament winding. The properties of the Carbon fiber and resin are given in Table 1. The filament passed through the resin bath and then wound on the aluminum mandrel, as shown in Figure 1(a). The wound filament then was vacuum bagged and cured according to the cure cycle given in Figure 1(b). The internal diameter and the length of the shafts manufactured were 60 mm and 1200 mm, respectively.

Table 1. Properties of the resin and the carbon fiber.

	Resin	HS Carbon fiber (24K)	Adhesive
Viscosity (mPas)	750 ±150		120
Density (g/cm^3)	1.10 ±0.05	1.79	0.97
Tensile strength (Mpa)	75-80	4900	22.6
Elastic modulus (Gpa)	3.2 -3.7	250	1.55

Two shafts were manufactured to investigate the effect of number of plies on the torsional stiffness of the shafts. The number of plies selected was 13 and 17 of ±45 fiber orientation. Two steel inserts were manufactured and bonded to both ends of the shafts using high strength epoxy based commercial grade adhesive. A small hole was drilled on the ends of composite shafts to inject the adhesive between the insert and the inner surface of the shaft. The thickness of the adhesive film was 2 mm. The shaft manufactured and assembly are shown in Figure 2.

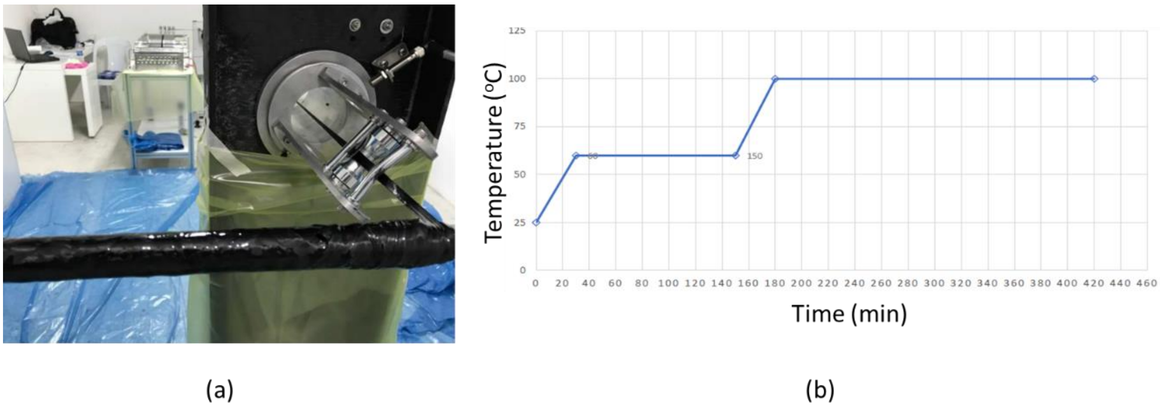

Figure 1. (a) flament winding method, (b) cure cycle.

Figure 2. (a) the composite shaft manufactured, (b) insert for testing, (c) final assembly with regular yokes.

2.2. Torsion tests

The composite shaft-insert assembly was mounted to the test set-up shown in Figure 3. One end of the shaft was fixed to the test set-up using three bolds and the other hand was twisted by an arm. Rotation and the torsion values were recorded in time. Two shafts manufactured (13-ply and 17-ply) and the original metal shaft were tested.

Figure 3. Torsion test set-up.

3. Finite element method

Finite element models were developed to predict the torsional stiffness and natural frequency of the composite drive shafts. Finite element analysis was performed using Abaqus software to find the torsional stiffness, strength, and natural frequencies. The composite drive shaft was considered as a thin-walled orthotropic tube. A 8-node doubly curved thick shell element (S8R) was selected. The shaft was subjected to torsional load of 3800 Nm at one end and the other end was constrained in axial, radial, and hoop direction. The meshing and loading condition are shown in Figure 4. Coupling constraint were assigned to the red region in the figure which represent the tubular joint region. Linear and elastic relationship were assumed in the model. Rule of mixture was used to calculate the elastic properties of the composite. The thickness of each layer was 0.441 mm and the fiber orientation was ±45. Two analysis were done for 13- and 17-ply shafts.

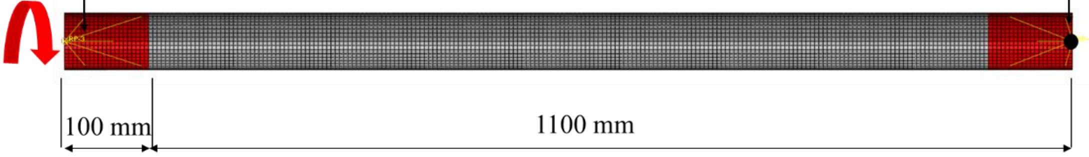

Figure 4. Meshing, loading and boundary conditions.

4. Results and Discussion

4.1. Torsional stiffness

The torque capacity of a regular metal drive shaft for passenger cars should be larger than 3500 Nm [11] so that the composite drive shaft was designed considering this value without torsional failure. In-plane principal stress values is shown in Figure 5. The maximum stress value occurred was 203.7 MPa on the outer surface of the shaft. Torque –twisting angle curves calculated using FEA and those measured in the torsion test are given in Figure 6. The calculated torsional stiffness values using FEM were higher than measured ones. The reason might be due to improper joint method, or due to the inaccuracy of the material properties used in FEA. There was an adhesive failure between the composite and the inserts before the first ply failure occurred in the composite. The decrease in the slope of the experimental curves that can be seen in Fig. 6 may be due to yielding of the adhesive layer. The joint failure is shown in Figure 7. The torsional stiffness of the composite drive shafts was larger than metal drive shaft but they did not reach the torque value of 3500 Nm due to joint failure.

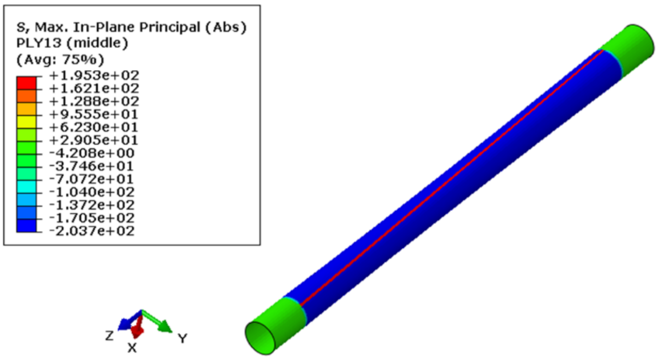

Figure 5. Stress values occurred at the end of the analysis for 13 plies configuration.

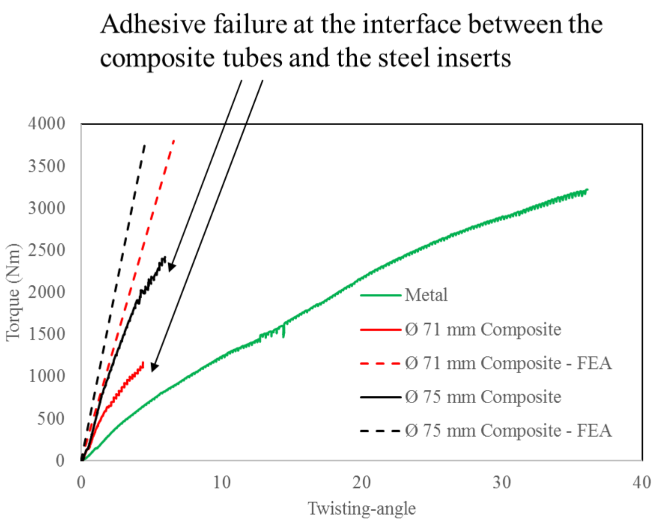

Figure 6. Torque versus twisting angle.

Figure 7. Joint failure.

4.2. Torsional buckling

The possibility of the torsional buckling of the composite shaft was checked by the expression for the torsional buckling load T_{cr} of a thin walled orthotropic tube, which was expressed below [12],

$$T_{cr} = (2\pi r^2 t)(0.272)(E_x E_y^3)^{0.25} (t/r)^{1.5}. \quad (1)$$

where E_x and E_y are the Young's modulus of the composite shaft in axial and hoop direction, r and t are the mean radius and thickness of the composite shaft. The calculated E_x and E_y equals to 8321 MPa and the calculated critical torsional moment is 8022 Nm for 13-ply and 15686 Nm for 17-ply shafts, which are higher than 3500 Nm. These two configurations are in the safe region under torsional buckling consideration. The critical torque value for 9 plies was 3200 Nm so that the number of plies should be larger than 9 plies.

4.3. Natural frequency analysis

The fundamental bending natural frequency of a one-piece drive shaft should be higher than 9200 rpm without whirling vibration [11]. The critical natural frequencies are estimated by performing eigenvalue analysis using finite element method. The natural frequencies for the first four mode shapes were calculated and given in Figure 8. The analysis was done for both shafts and the minimum frequency calculated was 152360 rpm for 17-ply shaft, which is higher and is not in the range of 9200 rpm.

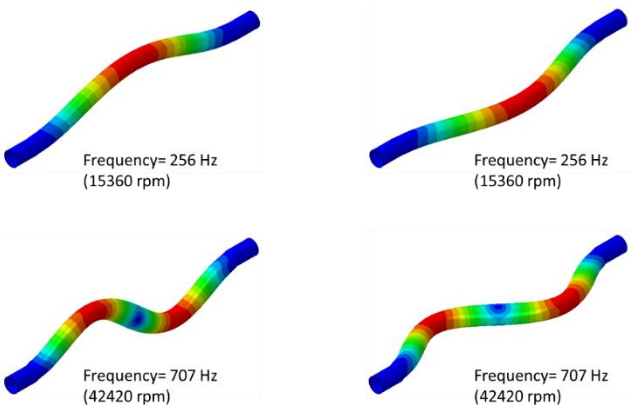

Figure 8. Set of the first four mode shapes and the corresponding natural frequency values for the 17-ply shaft.

4.4. Proposed joint method and future work

Due to premature joint failure a new joint design is proposed, which is polygonal shape. The shape and dimensions of the drive shaft are shown in Figure 9. In the analysis, pentagon (5), hexagon (6), heptagon (7), and octagon (8) polygons were evaluated and torsional stiffness analysis was done. The model is same as the previous model except the end sections. The results of the analysis are given in Figure 10.

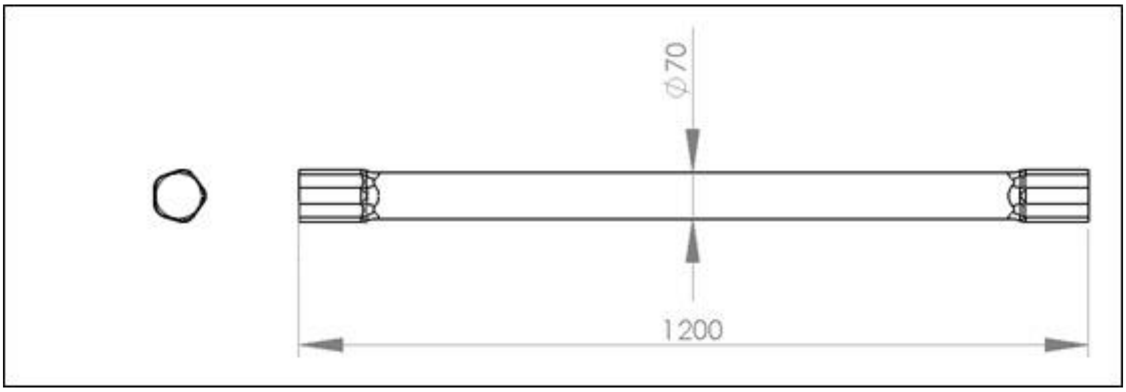

Figure 9. The end shape and dimensions of proposed design of the composite drive shaft.

Octagon end shape drive shaft gave the best results in terms of the stress concentration occuring between polygonal region and circular region.

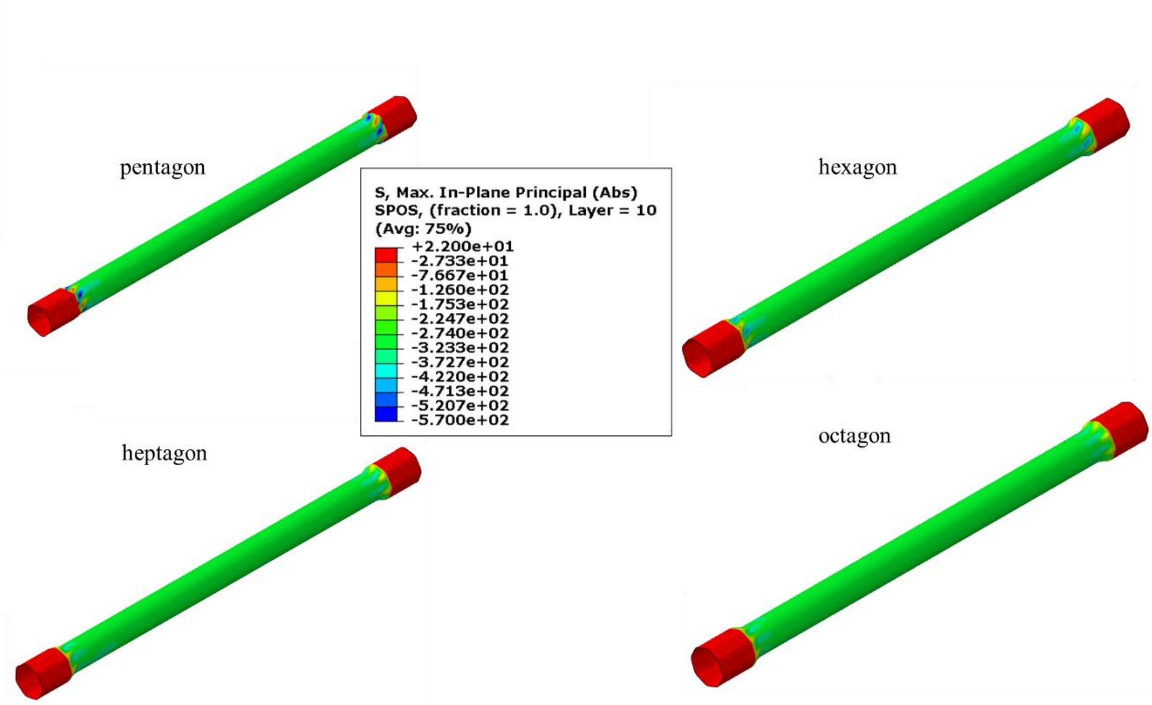

Figure 10. Max principal stress distribution on the 10-ply composite shafts with various connector ends.

5. Conclusions

Real sized composite drive shafts were designed and manufactured using a filament winding method. The shafts manufactured were tested using a torsion test set-up. A finite element method was developed to predict the torsional stiffness of the shafts. The calculated torsional stiffness was higher than measured ones. The reason might be due to improper joint method. There was an adhesive failure between the composite and the inserts before the first ply failure occurred in the composite. The torsional stiffness of the composite drive shafts was larger than metal drive shaft but they did not reach the torque value of 3500 Nm due to premature joint failure. Thus, the joint design is understood to be one of the critical points during designing and manufacturing of composite driveshafts.

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