Design of Composite Drive Shaft and its Coupling for Automotive Application

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Abstract

This paper presents design method and vibration analysis of a carbon/epoxy composite drive shaft. Designing of the composite drive shaft is divided into two main sections: design of the composite shaft and design of its coupling. Some parameters such as critical speed, static torque, fiber orientation and adhesive joints are studied. Tsai-Hill failure criterion is implemented to control the rupture resistance of the composite shaft and then its critical speed analysis and modal analysis are carried out using ANSYS. The behavior of materials is considered nonlinear isotropic for adhesive, linear isotropic for metal and orthotropic for composite shaft. The results show significant points about the optimum design of composite drive shafts. The substitution of composite shaft has resulted in considerable weight reduction about 72% compared to conventional steel shaft. Furthermore, results reveal that the orientation of fibers has great influence on the dynamic characteristics of the composite shaft.

Keywords: Adhesive joint; Composite material; Drive shaft; Modal analysis; Natural frequency

1. Introduction

Nowadays, composite materials are used in large volume in various engineering structures including spacecrafts, airplanes, automobiles, boats, sports' equipments, bridges and buildings. Widespread use of composite materials in industry is due to the good characteristics of its strength to density and hardness to density. The possibility of increase in these characteristics using the latest technology and various manufacturing methods has raised application range of these materials. Application of composite materials was generally begun only at aerospace industry in 1970s, but nowadays after only three decades, it is developed in most industries. Meanwhile, the automotive industry considered as a mother one in each country, has benefited from abilities and characteristics of these advanced materials. Along with progress in technology, metallic automotive parts are replaced by composite ones. One of them is drive shaft (propeller shaft), which numerous researches have been carried out on it in recent decades. Drive shafts are usually made of solid or hollow tube of steel or aluminum. Over than 70% of single or two-piece differentials are made of multi-piece propeller shaft that result in a rather heavy drive shaft [1]. Figure 1 shows a photographic view of two-piece steel and a sample composite drive shaft. Graphite/carbon/fiberglass/aluminum driveshaft tube was developed as a direct response to industry demand for greater performance and efficiency in light trucks, vans and high performance automobiles. The main reason for this was significant saving in weight of drive shaft; the results showed that the final composite drive shaft has a mass of about 2.7 kg, while this amount for steel drive shaft is about 10 kg.

Numerous studies have been carried out to investigate the optimal design and analysis of composite drive shafts with different materials and fibers orientation. Pollard [2] studied different applications of composite drive shafts for automotive industry. He compared the advantages and disadvantages of them at various conditions. Rangaswamy and et al. [3] optimized and analyzed a one-piece composite drive shaft using genetic algorithm and ANSYS. They found that the use of composite materials lead to the significant reduction in weight compared to steel drive shaft. They also reported that the fiber orientation of a composite shaft strongly affects the buckling torque. Rastogi [4] implemented a FEA approach to design and analyze a composite drive shaft with its couplings in different conditions. Badie et al. [5] conducted a finite element analysis to study effects of design variables on the drive shaft critical mechanical characteristics and fatigue resistance. They found that stacking sequence has an obvious effect on the fatigue
resistance of the drive shaft. Kumar [6] performed an optimum design and analyzed a composite drive shaft for automobile application using a genetic algorithm approach. He optimized design parameters with the objective of minimizing the weight of composite drive shaft. Chowdhuri et al. [7] replaced a two-piece composite shaft by a one-piece steel shaft. They proposed two different designs consisting graphite/epoxy and aluminum with graphite/epoxy.

An efficient design of composite drive shaft could be achieved by selecting the proper variables, which can be identified for safe structure against failure and to meet the performance requirements. As the length and outer radius of drive shafts in automotive applications are limited due to spacing, the design variables include the inside radius, layers thickness, number of layers, fiber orientation angle and layers stacking sequence. In optimal design of the drive shaft these variables are constrained by the lateral natural frequency, torsional vibration, torsional strength and torsional buckling. In the present work an effort has been made to design a HM-Carbon/Epoxy composite drive shaft. A one-piece composite drive shaft for rear wheel drive automotive application is designed and analyzed using ANSYS software.

2. Design procedure of composite drive shaft

First, fibers are selected to provide the best stiffness and strength beside cost consideration. It is the best selection, indeed, to use carbon fibers in all layers. Since the fiber orientation angle that offers the maximum bending stiffness which leads to the maximum bending natural frequency is to place the fibers longitudinally at zero angle from the shaft axis, on the other hand, the angle of ±45° orientation realizes the maximum shear strength and 90° is the best for buckling strength. The main design goal is to achieve the minimum weight while adjusting the variables to meet a sufficient margin of safety, which is translated in a critical speed (natural frequency) higher than the operating speed, a critical torque higher than the ultimate transmitted torque and a nominal stress (the maximum at fiber direction) less than the allowable stress after applying any of the failure criteria like the maximum stress criteria. Due to the physical geometry (larger radius) of the drive shafts used in the mentioned applications including automotive applications, the shear strength which specify the load carrying capacity, is of minor design importance since the failure mode is dominated by buckling, therefore the main design factors are the bending natural frequency and the torsional buckling strength, which are functions of the longitudinal and hoop bending stiffness, respectively. The variable of the laminate thickness has a big effect on the buckling strength and slight effect on bending natural frequency. From the properties of the composite materials at fibers direction, the first step is the construction the reduced stiffness matrix. The expressions of the reduced stiffness coefficients $Q_{ij}$ in terms of engineering constants are as follows:

$$Q_{11} = \frac{E_1}{1 - \nu_{12} \nu_{21}}, \quad Q_{22} = \frac{E_2}{1 - \nu_{12} \nu_{21}}, \quad Q_{12} = \nu_{12} E_2$$

$$Q_{66} = G_{12}$$

$$v_{21} = \frac{E_2}{E_1} v_{12}$$

The second step is to construct the extensional stiffness matrix $[A]$. This matrix is the summation of the products of the transformed reduced stiffness matrix $\overrightarrow{[Q]}$ of each layer and the thickness of this layer as:

$$[A] = \sum_{k=1}^{N} \overrightarrow{[Q]}^k (z_k - z_{k-1})$$

The $A$ matrix is in (Pa.m) and the thickness of each ply is calculated in reference of their coordinate location in the laminate. The $A$ matrix is used to
calculate $E_x$ and $E_h$, which are the average module in the axial and hoop directions, respectively from:

$$E_x = \frac{1}{t}[A_{11} - \frac{A_{12}^2}{A_{22}}]$$

$$E_h = \frac{1}{t}[A_{22} - \frac{A_{12}^2}{A_{11}}]$$

(3)

### 2.1. Buckling torque

Since the drive shaft is long, thin and hollow, there is a possibility for it to buckle. The expression of the critical buckling torque for thin-walled orthotropic tube is given as [8]

$$T_{cr} = (2\pi^2\nu)(0.272)\left[\frac{E_x}{E_h}\right]^{1/4}\left(\frac{t}{\rho}\right)^{3/2}$$

(4)

Where $r$, is the mean radius and $t$ is the total thickness. It is obvious that the stiffness modulus at hoop direction ($E_h$) plays the big role in increasing the buckling resistance. The factor of safety is the ratio of the buckling torque to the ultimate torque.

### 2.2. Lateral bending natural frequency

The main point that attracts manufacturers to use composite materials in the drive shafts is that they make it possible to increase the length of the shaft. The relationship between shaft’s length and the critical speed for both types of drive shafts are shown in Figure 2. It is evident that for a specific application where the critical speed is about 8000 rev/min, the longest possible steel shaft is 1250 mm, while the composite one can have a length 1650 mm [2].

![Fig. 2. The effect of shaft length on critical speed][2]

Critical speed of a shaft is obtained through following equation:

$$N_{cr} = \frac{\pi^2}{P} \sqrt{\frac{E_t}{\rho A}}$$

$$f = \frac{\pi}{2} \sqrt{\frac{E_t}{mL^2}}$$

(5)

Where $N_{cr}$ is the critical speed and $f$ is the bending frequency. Considering that the natural frequency of a shaft according to the above equation is inversely proportional to the square of shaft’s length and is proportional to the square root of Young’s modulus, conventional steel drive shafts are made of two pieces to increase the natural frequency of the shaft, which results in overall increase in weight of shaft. So, in order to increase the natural frequency, the length of shaft should be reduced or $E/\rho$ ratio should be increased. Despite the space limitations that confines outer diameter of the shaft, the only way to increase the critical speed is to increase $E/\rho$ ratio (Specific module) [3].

One of the interesting properties of metals is that although there is a clear difference in their density, their specific modulus is almost constant. By applying fiber-reinforced composites, fiber orientation arrangement becomes possible in the shaft; therefore, bending modulus will be high. Also their relative density is low leading to the desirable specific modulus and increases the critical speed [9]. The natural frequency of the shaft was obtained through Timoshenko theory as following equation:

$$f_n = K_t \frac{30\pi^2 \rho^2}{E_t} \sqrt{\frac{E_t}{2\rho}}$$

(6)

Where $f_n$ is the natural frequency, $\rho$ is the first natural frequency and $\rho, E$ are properties of the steel shaft. $K_t$ is given by following equation:

$$\frac{1}{K_t} = 1 + \frac{p^2 \pi^2 \rho^2}{2L^2} \left[1 + \frac{f_n E}{G}\right]$$

(7)

Where the $G$ is modulus of rigidity of steel shaft and $f_n$ is equal to 2 for hollow cross sections. Then critical speed is obtained in following way [3]:

$$N_{cr} = 60 f_n$$

(8)

### 2.3. Load carrying capacity
The composite drive shaft is designed to carry the torque without failure. The torsional strength or the torque at which the shaft fail, is directly related to the laminate shear strength through

\[ T_s = \frac{2\pi^2 r_s^2}{r_m^2} \tau_l \]  

(9)

where \( T_s \) is the failure torque, \( \tau_l \) is the in-plane shear strength of the laminate, \( r_m \) is the mean radius and \( t \) is the thickness.

### 2.4. Tsai-Hill failure criterion

By using Tsai-Hill failure criterion, it would be possible to calculate the dimension for failure. With the thickness of 2.03 millimeters and the applied loads, the 0° fibers will not be ruptured. With the thickness of 2.2 millimeters and the applied loads, the 90° fibers will not be ruptured. Due to the torsion of the shaft, the buckling is negligible.

(10)

### 3. Modal analysis of composite drive shaft using ANSYS

In this study, finite element analysis is conducted using ANSYS commercial software. The 3-D model is developed and typical meshing is generated by using Shell 99 element. The shaft is fixed at both ends and is subjected to torque at the middle. The torque transmission capability of the drive shaft is taken as 3000 N.m, the length and the outer diameter here are considered as 2 meters and 120 millimeters, respectively. The shaft rotates at a constant speed about its longitudinal axis. The shaft has a uniform, circular cross section. The shaft is perfectly balanced, all damping and nonlinear effects are excluded. The stress-strain relationship for composite material is linear and elastic; hence, Hook’s law is applicable for composite materials. Since lamina is thin and no out-of-plane loads are applied, it is considered as under the plane stress. The HM Carbon/Epoxy material with fiber volume of 60% is selected for composite drive shaft. Table 1 shows the mechanical properties of each layer of the laminate. Considering the equations and design correlations the optimum fiber arrangement of the composite drive shaft is obtained as [90° / 0° / ±45°].

| Property          | HM Carbon/Epoxy |
|-------------------|-----------------|
| \( E_{11}(Gpa) \) | 190             |
| \( E_{22}(Gpa) \) | 7.7             |
| \( G_{12}(Gpa) \) | 4.2             |
| \( v_{12} \)      | 0.3             |
| \( \sigma_{11} \) | 870             |
| \( \sigma_{22} \) | 540             |
| \( \tau_{13} \)  | 30              |
| \( \rho(kg/m^3) \) | 1600           |
| \( V_f \)        | 0.6             |

Table 1. Mechanical properties for each lamina of the laminate

Figure 3 shows the domain of finite element mesh. Once the finite element mesh and the layers are created, orientation of materials is defined for the shell element and layer materials for each of these elements are being allocated. The other steps include placing the boundary conditions and selecting appropriate solvers. The shaft rotates with maximum speed so the design should include a critical frequency. If the shaft rotates at its natural frequency, it can be severely vibrated or even collapsed. The modal analysis is performed to find the natural frequencies in lateral directions. The mode shapes for all material combinations are obtained to their corresponding critical speeds. A number of fundamental modes, which all are critical frequencies, are obtained. The dynamic analysis shows that the first natural frequency is 169.64 Hz, and according to it the critical speed is equal to 10178 rpm, which is much more than the critical rotational speed of 4000 rpm. According to the equations obtained in previous section, natural frequency of a specific composite drive shaft is 4570.2 rpm. This value is very different from the initial one because the correlations used to obtain the values associated with the shaft, were in case of considering some assumptions.

Figure 4 depicts the deformation rate change for composite drive shaft at the first natural frequency. Figures 5 and 6 show the displacement rate of composite drive shaft in different directions at first mode. The natural frequencies of composite drive shaft are given in Table 2.

![Fig. 3. The mesh configuration of composite shaft](image)
3.1. Effect of fiber angles

The fibers orientation angle has a big effect on the natural frequency of the drive shaft. It is clear that from Figure 7, the fibers must be oriented at zero degree to increase the natural frequency by increasing the modulus of elasticity in the longitudinal direction of the shaft. This explains why the carbon fibers, with their high modulus saved to be oriented at zero angle. The drive shaft loses 38% of its natural frequency when the carbon fibers oriented in the hoop direction at 90º instead of 0º. The cost factor plays a role in selecting only one layer of carbon/epoxy. The stacking sequence has no effect on the natural frequency since there is no load applied in defining the natural frequency.

3.2. Design of adhesive joints in composite drive shafts

The joints used for connecting composite materials can be metallic or non-metallic. Steel fasteners due to the possibility of galvanic corrosion with carbon-epoxy materials, are mainly made of titanium or stainless steel. Other alloys such as aluminum or steel can be used provided that no contact with the surface is occurred. Joints are divided to metal screws and rivets. Non-metallic connectors are created from reinforced thermostet or thermoplastic resins. By using this connection, structural weight reduces and corrosion problems disappear [10]. In this part, first the thickness of the adhesive and length of adhesive bond are computed.

Then, a finite element analysis of this type of bond is performed using ANSYS software. The "Araldite" adhesive was used in this study. The following correlations were used to calculate the required parameters:
\[ \tau_{\text{max}} = \frac{a}{\tanh a} \times \tau_w \]  
\[ a = \sqrt{\frac{G_f l^2}{2G_{\text{ee}}}} \]  

(11)

Where \( l \) is the length of adhesive bond. For obtaining reasonable results the only possible way is to increase the value of \( e_c \), so, the thickness of the adhesive and the length of the adhesive bond are obtained 12 millimeters and 4.5 centimeters, respectively. The details of the bond are given in Table 3.

Table 3. Mechanical properties of the adhesive

| Property | Value  |
|----------|--------|
| \( E \)  | 2.5 Gpa |
| \( G \)  | 1 Gpa  |
| \( e_c \) | 0.25  |
| \( l \)  | 45 mm  |
| Layers orientation | [±45/0/±45]_t |

For analysis, a FE model was applied using the 3-D linear solid elements. A suitable mesh for finite element modeling of adhesive layer is needed. The shear distribution stress of the adhesive is demonstrated in Figure 8. Application of appropriate adhesive, results in decreased maximum shear stress in adhesive at the edges of the connection; however, if the stresses remain same in the middle connection, start of failure will depend on the relative shear strength values within the adhesive.

4. The weight comparison between composite and steel drive shafts

The entire vehicle drive shaft is consisted of several rotating masses. About 17-22% of the power generated by the engine is wasted due to rotating mass of power train system. Power is lost because a lot of energy is needed to rotate heavy parts. This energy loss can be reduced by decreasing the amount of rotating mass. In Figures 10, a mass comparison between steel and composite drive shafts has been done. The substitution of composite shaft has resulted in considerable weight reduction about 72% compared to conventional steel shaft.

5. Conclusions

In this paper a one-piece composite drive shaft is considered to be replaced a two-piece steel drive shaft. Its design procedure is studied and along with finite element analysis some important parameter are obtained. The composite drive shaft made up of high modulus carbon/epoxy multilayered composites has been designed. Modal analysis is conducted to obtain natural frequencies of the composite shaft. Effect of changing the carbon fiber orientation angle on natural frequency is also studied. The replacement of composite materials has resulted in considerable amount of weight reduction about 72% when compared to conventional steel shaft. Also, the results show that the orientation of fibers has great influence on the dynamic characteristics of the composite shafts.

Figure 9. Shear stress distribution in the adhesive bond connection

Figure 10. Mass comparison between steel and composite shaft (Kg)

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