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Stedile Filho, P., Almeida Jr, J. H. S., & Amico, S. C. (2018). Carbon-epoxy filament wound composite drive shafts under torsion and compression. *Journal of Composite Materials*, *52*(8), 1103-1111. https://doi.org/10.1177/0021998317722043

Published in:
Journal of Composite Materials

Document Version:
Peer reviewed version

CARBON/EPOXY FILAMENT WOUND COMPOSITE DRIVE SHAFTS UNDER TORSION AND COMPRESSION

Paulo Stedile Filho[a], José Humberto S. Almeida Jr.[b*], Sandro C. Amico[a]

[a]PPGE3M, Federal University of Rio Grande do Sul, Av. Bento Gonçalves, 9500, 91501-970. Porto Alegre/RS, Brazil

[b]Leibniz-Institut für Polymerforschung Dresden e. V., Department of Composite Materials, Hohestraße 6, 01067 Dresden, Germany

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Abstract

Composite cylinders to be used as half shafts must satisfy several requirements, such as critical speed, critical buckling torque and load carrying ability. This study focused on the investigation of carbon fiber reinforced epoxy composite cylinders produced by filament winding to be used as half shafts. A preliminary torsional test in a $[\pm45]_5$ cylinder was performed and three other laminates were chosen for the study: [±22/±45], [±89/±45] and [±45/±45]. Radial and longitudinal compression tests were performed. Mechanical analysis has been carried out using analytical and numerical approaches, and good correlation was found between them and the experimental values. The [±45/±45] cylinder showed the best performance under torsional loading, as expected, as well as radial and longitudinal compression, but not critical buckling torque.

Keywords: Composite shaft; torque; compression; FEA; filament winding.

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[*] *Corresponding author*: humberto@ipfdd.de; jhsajunior@globomail.com.

Phone: +49 351 4658 1423; Fax: +49 351 4658 362

1. Introduction

Composite materials have been increasingly used in structural applications mainly due to their intrinsic high specific mechanical properties and lack of corrosion [1,2]. There are innumerous applications of composite structures in aerospace, aeronautical, marine, civil and automotive sectors. Regarding automotive parts, high performance vehicles constantly demand lighter and stronger parts [3], sometimes driving the choice to composite structures. Metallic-based drive shafts, for instance, have been steadily replaced by composite cylinders, offering better vibration damping, wear reduction on drivetrain components and improvement on wheel traction [4].

Drive shafts transfer the power from the differential to the wheel, and for each driving wheel there is a shaft. The shafts usually consist of cylindrical steel bars, which are joined by universal joints, which articulate movements from the suspension [5]. As key requirements for a drive shaft, allowable torque per passenger must be higher than 3500 N.m and fundamental bending natural frequency higher than 9200 rpm to avoid whirling vibration [6]. Since the natural frequency of a metallic-based drive shaft is lower than 5700 rpm (for a length of around 1.5 m), a steel shaft is normally manufactured in two halves with universal joints, a center supporting bearing and a bracket, increasing overall weight and fuel consumption [7]. On the other hand, a carbon/epoxy composite shaft could be manufactured in a single piece and be used over 9200 rpm [11], with lower noise and vibration [8].

From the design point of view, the most common parameters are: natural frequency, critical buckling torque, torsional frequency and torque transmission strength [9]. Natural frequency, for example, should not be close to the rotating frequency, otherwise the shaft would present significant vibration, occasionally leading to premature failure.

Badie et al. [10] evaluated natural frequency of composite drive shafts for different fiber orientations and concluded that the fiber should be placed as close as possible to the longitudinal axis in order to reach the maximum natural frequency. Khoshravan and Paykani [11] compared metallic and carbon/epoxy composite drive shafts and reported that the composite shaft may be optimized by increasing cylinder length, improving its natural frequency.

Torsional frequency is directly related to torsional stiffness (T / φ), where φ is the torsional angle. When the drive shaft is submitted to a torsional load, it will eventually reach a critical buckling load/torque. Bauchau et al. [12] evaluated the buckling behavior of graphite/epoxy shafts under torsional loading through theoretical and experimental approaches, finding good agreement between these results. Kim et al. [8] studied [$0_{carbon}/(\pm 45)_{N,glass}$] filament wound hybrid carbon/glass composite shafts and concluded that the higher the number of ±45° layers, the higher the torsional performance. In addition, Montagnier and Hochard [13] pointed out that the need for hoop layers to maximize critical buckling torque, which is more related to the transverse direction than the longitudinal one.

Thus, considering that drive shafts operate under a combination of forces, the current work focuses on the investigation of carbon/epoxy composite cylinders with different stacking sequences for drive shafts based on analytical and numerical analyses concerning torque behavior, natural frequency, critical buckling torque and torque carrying capacity. Radial and axial compressive behavior of the cylinders are also explored through experimental and analytical approaches.

2. Experimental

Towpregs from TCR Composites, comprised of Toray T700-12K-50C carbon fiber and an epoxy resin system (UF3369), were used in this work. A KUKA robot and MF Tech winding equipment were employed to manufacture the composite cylinders on top of a steel mandrel (l = 420 mm and ϕ = 25 mm). Robot programming was carried out with CADWIND software, which enables prior evaluation of pattern generation, layer thickness, actual winding angle in each position and deviation from geodesic path. After winding, a polyester-based shrink tape was used to wrap the laminate and aid compaction during curing. The system was then cured in an oven with air circulation at 130 °C for 4 h and later cooled at room temperature to allow extraction of the cylinders.

In order to validate the numerical models, a preliminary $[\pm 45]_5$ laminate with a nominal thickness of 3.15 mm was manufactured. This cylinder was adhesively bonded to metallic inserts and torsion experiments were carried out in a GIM equipment, where one extremity was fully clamped and the other one was submitted to torque loading until break.

For the other tests, three families of composite cylinders were manufactured, [±22/±45], [±89/±45] and [±45/±45]. These stacking sequences were chosen based on the parameters critical to the application, natural frequency, critical buckling torque, and torque transmission strength. The optimal angles for these parameters are, respectively, 0^o, 90^o and $\pm 45^o$. Considering that torque transmission strength is critical, $\pm 45^o$ layers were included in all laminates. Also, the $\pm 22^o$ and $\pm 89^o$ layers were selected intending to approach the optimal angles of 0 and 90, while respecting the limitations of the used FW set-up.

Optical micrographs (OM) were taken on untested samples to study laminate thickness and voids using a Carl Zeiss microscope Axio Scope, as shown in Figure 1 and Table 1. In FW process, voids appear due to fiber intercrossing, in-between bands (fiber positioning must be very accurate to avoid gaps between bands), high winding speeds and low winding tension during fiber deposition. The variation in thickness values measured with a caliper and by OM are due to uncertainties in the measurements and the chosen measurement location, since the outer surface of a FW structure may slightly vary, especially when a shrink tape is wound onto the structure, which is the case here.

<< Insert Table 1 >>

Radial compression tests (Figure 2(a)) were performed in an Instron Universal testing machine model 3382 following ASTM D2412-11. Five specimens (ϕ = 25 mm, l = 150 mm) were tested under a testing speed of 12.5 mm/min. Cylinder stiffness (CS), percentage cylinder deflection quota (P) and stiffness factor (SF) were calculated using Eqs. (1-3).

<< Insert Figure 1 >>

$$\boldsymbol{CS} = \frac{\boldsymbol{F}}{\boldsymbol{\Delta_y}} \tag{1}$$

$$\boldsymbol{P} = \frac{\boldsymbol{\Delta_y}}{\boldsymbol{d}} \times \mathbf{100} \quad (2)$$

$$SF = \frac{EI}{r^3} = 0.149r^3 \times TS \quad (3)$$

where: F is the load and Δ_y is the change in outside diameter of the cylinder in the load direction.

Axial compressive tests, shown in Figure 2(b), were performed in the same machine under the same conditions using five specimens (ϕ = 25 mm, l = 100 mm). Optical micrographs were taken to study the fractured samples from both radial and axial compressive tests.

<< Insert Figure 2 >>

A comprehensive characterization of the carbon/epoxy system herein used has been previously reported [14] and used as input in various numerical studies [15,16]. These properties (Table 2) were also used in the present study for all analytical and numerical predictions. The unidirectional lamina was regarded as transversely isotropic ($E_2 = E_3$, $\nu_{12} = \nu_{13}$ and $G_{12} = G_{13}$).

<< Insert Table 2 >>

3. Theoretical analysis

The drive shaft is considered as a simply supported beam undergoing transverse vibration, idealized as a pinned-pinned beam. Based on specimen deflection, the natural frequency is obtained, since there is no damping force on the structure. The half shaft was simulated under conditions similar to the drive shaft. Thus, natural frequency and moment of inertia of the cylinder were analytically calculated by Eqs. (4-5).

$$f_n = \frac{\pi}{2}\sqrt{\frac{g \cdot E_X \cdot I_X}{W \cdot L^4}} \quad (4)$$

$$I_X = \frac{\pi}{4}\left(r_0^4 - r_i^4\right) \approx \pi \cdot r^3 \cdot t \quad (5)$$

where: I_X is moment of inertia, L is length, g is gravity, E_X is longitudinal elastic modulus, W is weight and t is laminate thickness.

The maximum torque transmission was calculated analytically by classical theory of laminates and using finite element (FE) method. The numerical models were built in Abaqus 6.14 FE package.

As boundary conditions (BC), the shaft was clamped axially and radially at one end and submitted to torsion loading at the other extremity. The resulting shear stress is in-plane (τ_{xy}) and analytically defined by Eq. (6):

$$\tau_{xy} = \frac{T}{2\pi r^2} \tag{6}$$

where: T is the torque.

A similar analysis has been carried out using the Autodesk Simulation Composite Design for comparison purposes. Through this approach, maximum shear stresses throughout the cylinder and along the cylinder length have been determined, along with stiffness and displacement.

A numerical model has been developed to evaluate torque response, where a torque load of 650 N.m was applied at the free extremity of the cylinder. The structure has been modeled using linear quadrilateral shell elements with equivalent single layer (ESL) formulation with reduced integration (S4R) and three integration points in each layer (only in-plane stresses and strains are considered). The mesh used and the BCs can be viewed in Figure 3. The mesh had 3,640 elements (5-mm long each) and 3,666 nodes, defined after a preliminary convergence analysis (also shown in Figure 3).

<< Insert Figure 3 >>

Critical buckling torque was also determined using analytical and numerical approaches. The Eigensolver used was Lanczos, which is an algorithm to determine Eigenvalues in a matrix, disregarding non-linearity in the analysis. As boundary conditions, one extremity of the half shaft was clamped, restricting displacements and rotations in all directions. A torsion load was applied at the other extremity, yielding a torsion moment. The analytical method [17] used to determine critical torque (T_{cr}) is shown in Eq. (7):

$$T_{cr} = (2\pi r^2 t) \cdot (0.272) \cdot (E_X \cdot E_Y^3)^{1/4} \cdot \left(\frac{t}{r}\right)^{3/2} \tag{7}$$

where: E_y is the elastic modulus in the transverse direction.

The same mesh and BCs of the previous analysis were employed in the model. This model differed from the previous one just for the applied load, i.e. the buckling load was replaced by a torque of 400 N.m, which is the estimated in-service torque load of the drive shaft. Tsai-Wu failure criterion was used to estimate ultimate torsional load of the cylinder.

4. Results and discussion

4.1 Torque carrying capacity

Three specimens were submitted to torsion tests, but two of them failed at the metallic inserts. Only one test was validated, which presented no evidence of buckling or slipping. The maximum experimental torque obtained for the $[\pm45]_5$ laminate was 780 N.m, shown in Figure 4, and this result lies between analytical and numerical predictions, being the analytical one more conservative.

A drive shaft normally operates under a maximum in-service torque of 400 N.m, and the current cylinder $[\pm45]_5$ reached a 95% higher torque load. Taking into account that the weight is critical in high performance parts, subsequent analyses were performed using fewer laminas. Torque stiffness and angular displacement were calculated for the [±22/±45], [±89/±45] and [±45/±45] laminates and are shown in Table 3. This table also presents numerical and analytical predictions for torque strength and the [±45/±45] cylinder showed the best performance. This is expected since this is the optimum angle for cylindrical structures subjected to torsional loading, with the highest shear strength [18].

<< Insert Figure 4 >>

<< Insert Table 3 >>

4.2 Critical buckling torque

The critical buckling torque has been calculated (using Eq. 7 and Autodesk software) and numerically by FEM. Figure 5 and Table 4 indicate that [±89/±45] is the stiffest laminate, justified by the presence of a hoop layer of higher stiffness in the circumferential direction. This also reduces geometrical instabilities and, as a consequence, this laminate requires higher torque load to produce significant buckling. The other laminates are more unstable and prominent to buckling. The buckling patterns presented in Figure 5 consist of helical waves homogeneously spread along the length of the cylinder.

<< Insert Figure 5 >>

In Table 4, only the [±89/±45] laminate exceeded the maximum operating torque (400 N.m), as can be seen in all numerical and analytical buckling moment predictions (Table 4). In addition, the numerical values of buckling loads (also presented in Figure 5) are higher than the analytical ones. The analytical models are usually more conservative and, for example, disregard stress distribution along the cylinder length, which is accounted for in the numerical approach. Some discrepancies can be attributed to the nature of the constitutive equations in each method, and the numerical approach is thought to produce more realistic results since the problem can be more detailed developed, including more parameters.

<< Insert Table 4 >>

4.3 Natural frequency

The natural frequency results obtained for the laminates are also presented in Table 4. Assuming that the in-service natural frequency is about 120 Hz, all laminates presented values much above the minimum required. The best performance was obtained for the [±22/±45] cylinder due to the presence of a layer more closely oriented in the longitudinal direction which increases bending modulus and, consequently, natural bending frequency.

The axial buckling force of a drive shaft is the load on which the first natural frequency (herein presented) of the drive shaft becomes null. Moreover, when a beam is submitted to axial loading, the first natural frequency of the first bending mode decreases by increasing the applied load [19]. That is, when the composite drive shaft is subjected to torque loading, the natural torque frequency does not become less than the natural bending frequency. So, the critical speed of a drive shaft is based on the first natural bending frequency, just as herein presented [20].

4.4 Compressive behavior

Figure 6 displays representative curves for each type of cylinder under radial compression. The [±22/±45] cylinder sustains up to ~3.6 kN, and no abrupt failure is noticed, only a slight decrease, which is classical buckling behavior. This cylinder has a stable post-buckling behavior, in which it keeps sustaining load in the buckled state. At this point, the cylinder might be considered as imperfect, and the structure can have stable or unstable post-buckling behavior depending whether the loading increases or decreases after the bifurcation point [21]. These imperfections arise from the manufacturing process, mainly voids and fiber waviness.

The [±89/±45] cylinder supports a higher critical load, ~10 kN with a displacement of around 1 mm, but slightly different from the [±22/±45] cylinder, the structure still supports a great portion of the critical buckling load. That is, after reaching the bifurcation point, the structure does not collapse (no buckling) and sustains most of the load, which still slightly increases. This is a typical post-buckling behavior of a “perfect” structure, in other words, imperfections from manufacturing did not significantly affect the compressive behavior of the cylinder. The [±45/±45] cylinder behaves differently, and after it reaches the critical load, with a sharp peak at ~4.5 kN, it starts failing progressively, with multiple load peaks typical of delaminations. The fiber orientation at ±45° leads to high shear stresses within the laminas throughout the composite thickness.

<< **Insert Figure 6** >>

Table 5 displays mean critical load, cylinder stiffness, stiffness factor and percentage cylinder deflection values. The [±89/±45] cylinder is stiffer than the other cylinders due to the presence of the hoop layer. This cylinder also has the lowest percentage deflection, i.e. a more fragile behavior. The percentage cylinder deflection is associated with cylinder stiffness, and less stiff cylinders tend to deflect more [16]. The stiffness factor follows the same trend, being higher for the tube with a hoop layer. Stiffness factor is a useful design parameter since it is related to flexural modulus and wall thickness and it directly reflects cylinder deflection.

<< Insert Table 5 >>

Optical micrographs of the samples fractured under radial compression are shown in Figure 7. It is possible to find prominent cracks in all specimens, especially in [±22/±45], at the fiber direction in all layers. The same is observed in the other specimens, where cracks are found mainly in the fiber direction, along with delaminations.

<< Insert Figure 7 >>

Regarding axial compression, representative load *vs.* displacement curves are presented in Figure 8. The [±22/±45] and [±89/±45] cylinders presented similar behavior when under axial compression, with a sudden drop in load, although the latter supports around 33% more load than the former. On the other hand, the [±45/±45] cylinder started buckling prior to reaching the critical load, and after that, started delaminating, with subsequent small load drops.

<< Insert Figure 8 >>

Observation of the samples fractured under axial compression (Figure 9) reveals no evidence of sharp cracks for the [±22/±45] and [±89/±45] cylinders, with only small cracks along the specimen. However, for the [±45/±45] cylinder, several delaminations (large horizontal cracks) were seen, along with some resin rich regions and voids.

<< Insert Figure 9 >>

5. Conclusions

The mechanical behavior of carbon/epoxy composite drive shafts manufactured by filament winding under torsion and radial/axial compression has been investigated by means of analytical, numerical and experimental approaches. The torque carrying capacity showed good correlation between analytical and numerical predictions, and the [±45/±45] laminate displayed the best performance due to its higher shear strength. The [±89/±45] laminate presented the best critical buckling torque behavior, being considerably higher than the in-service load of 400 N.m. In this case, the analytical and numerical predictions were about 50% higher than that. The results for first natural frequency were also very satisfactory, and all laminates presented frequency levels much above the minimum required (i.e. 120 Hz).

In the experimental radial compression tests, only the [±45/±45] laminate did not buckle, delaminating above the critical load. However, the [±22/±45] laminate buckled and kept sustaining part of the buckling load, presenting a post-buckling behavior. In the axial compressive tests, none of the structures buckled, and the [±22/±45] cylinder showed the best behavior due to the layers with angles closer to the axial direction. In all, the [±45/±45] filament-wound composite cylinder presented the most satisfactory overall behavior, which could represent a weight reduction of up to 47% compared to a steel-based drive shaft based on the parameters herein considered for the design of composite cylinders as half shafts.

Acknowledgments

The authors are grateful to CNPq and AEB for the financial support.

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