AN INVESTIGATION INTO LAMINATED COMPOSITE AUTOMOTIVE DRIVE SHAFTS

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Abstract - Composite materials can be used in design and fabrication of automotive drive shafts to reduce the weight of the structure. Hence, an optimum design of a composite drive shaft satisfying the design criteria, can be an appropriate substitution of metallic drive shafts. The aim of this study is to design and analyze a composite automotive drive shaft with high specific strength and low weight satisfying the design criteria. Tsai-Wu criterion is chosen as the failure criterion. Various designs with different lay-ups and materials are investigated based on the design requirements and finally, an optimum design satisfying the design criteria is chosen based on the weight and cost considerations.

Keywords - Bending Natural Frequency, Composite Drive Shaft, Peak Torque, Torsional Buckling.

I. INTRODUCTION

An automotive drive shaft is a rotating shaft which transmits power from the engine to the differential gear of rear wheel drive vehicles. Through constantly changing angles between the transmission and axle of the vehicle, a drive shaft operates. High quality steel is usually used in fabrication of drive shafts [1]. Steel drive shafts are commonly fabricated in two pieces to increase the fundamental bending natural frequency, because this frequency is proportional to the square root of specific modulus and inversely proportional to the square of beam length. As shown in Fig.1, the two-piece steel drive shaft consists of a center supporting bearing, three universal joints, and a bracket, leading to the increase of the total weight of a vehicle. However, we can improve the power transmission rate through the reduction of the inertial mass and weight of the drive shaft. It is also known that the energy conservation is one of the most important objectives in vehicle design, and the reduction of weight is one of the most effective measures to obtain it. Composite materials in comparison with conventional metals have many advantages because of their higher specific stiffness and higher specific strength [2]. Composite materials can be designed and fabricated to efficiently satisfy the design requirements of strength and stiffness. Advanced composite materials have arisen intense research interests in design of drive shafts because of their excellent properties such as high specific modulus (modulus/density), high specific strength (strength/density), and weight less than steel or aluminum of similar strength [2]. When torque peaks occur in the driveline, a composite driveshaft, due to its lower elasticity modulus, acts as a shock absorber and leads to the reduction of stress on the drive train. The advanced composite materials with Graphite, Carbon, Kevlar, and Glass fibers seem ideally suited for long power drive (propeller shaft) applications [1]. Elastic properties of a composite drive shaft can be tailored to increase the torque that it can carry as well as the rotational speed at which it operates. An optimum design of a composite drive shaft provides a lower weight of the shaft which leads to the increase of the first natural frequency, the torque transmission, and the torsional buckling capability of the shaft. Therefore, advantages of composite drive shafts can be summarized as: high specific modulus and strength, reduced weight and making less fuel consumption, high damping capacity and hence producing less vibration and noise, high corrosion resistance, high impact resistance; higher torque capacity than steel and aluminum shafts, and longer fatigue life than steel and aluminum shafts. The objective of this study is to design and analyze a laminated composite automotive drive shaft with high specific strength and low weight satisfying the design criteria. To this purpose, the stress-strain relationship in composite materials are derived and the torque transmission capacity, the torsional buckling capacity, and the bending natural frequency of the shaft for various designs are calculated. Tsai-Wu failure criterion is used to calculate the shear strength of a laminated composite driveshaft. The procedural steps in design of a composite drive shaft are formulated to ensure the design requirements for the drive shaft such as the torque transmission capacity, the torsional buckling capacity, and the bending natural frequency are satisfied. To obtain an optimum design of the drive shaft, different designs with various composite laminate stacking sequences and materials are investigated. Finally, an optimum design among various designs satisfying the design criteria is chosen based on the weight and cost considerations. The results of this study can be used in design and fabrication of laminated composite automotive drive shafts.

II. DESIGN PROCEDURE

A. Assumption

In this study, it is assumed that the shaft rotates with a constant speed about its longitudinal axis and it is

perfectly balanced with a circular cross-section, and all damping and nonlinear effects are excluded. The stress-strain relationship for composite materials is linear and elastic, hence, Hooke's law is applicable for composite materials. Since lamina is thin and no out-of-plane loads are applied, it can be considered as a plane stress model.

B. Design Constraints

The drive shaft must carry the applied torsional load, thus, it must be designed to have enough torsional strength to carry the applied torque without failure. Furthermore, the torsional buckling possibility must be considered in design of a thin-walled tube. As the third major design requirement, the bending natural frequency of the drive shaft should be sufficiently high [1]. An optimum design of the drive shaft is desirable, which is the cheapest and lightest but meets all the above design requirements. Based on some reliable collected data, the above three load-carrying requirements are summarized in Table 1. Physical dimensions of the shaft to be designed are given in Table 2.

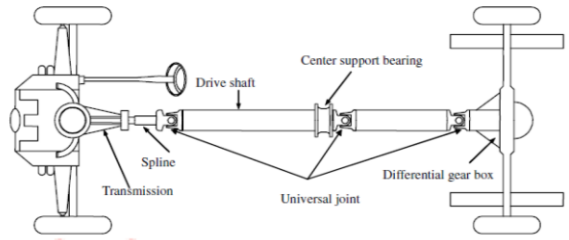


Fig. 1. Schematic of a conventional two-piece drive shaft for rear wheel drive vehicle [3].

Requirement	Value
Minimum bending natural frequency	90 Hz
Minimum buckling torque	2030 N-m
Peak torque for 100,000 cycle reversed fatigue	678 N-m

Table 1. Load requirements for the drive shaft design.

Length of the drive shaft (L)	173 cm
Mean radius of the drive shaft (r_m)	50.8 mm

Table 2. Physical dimensions for the drive shaft design.

C. Selection of Cross Sections

In this study, hollow circular shaft is chosen because it isstronger in per kg weight than solid circular and for a hollow shaft, the stress variation is smaller than that for a solid shaft.

D. Selection of Materials

Composite materials are made up of two phases: matrix and reinforcement phases, where resins and fibers are usually used for the matrix and the reinforcement phases, respectively. The important considerations in selecting the matrix phase (resin) are elongation to failure, resistance to impact (a function of modulus of elongation), temperature capability, and cost. The resins commonly used for most of the drive shafts are either epoxies or vinyl esters [4]. In this study, epoxy resin is selected due to its high strength, good wetting of fibers, lower curing shrinkage, and better dimensional stability. Fibers with various properties are available. Based on the design requirements, Carbon/Graphite fiber,Kevlar fiber, and Glass fiber are chosen in this study.

Thus, in this study, Carbon/Epoxy, Kevlar/Epoxy, and E-Glass/Epoxy composite materials are investigated for design of a laminated composite drive shaft. The properties of Carbon/Epoxy, Kevlar/Epoxy, and E-Glass/Epoxy unidirectional composites are presented in Table 3. In this study, the lamina thickness is considered 0.125 mm.

E. Composite Lay-Up

In this study, symmetric composite laminate is considered. Different fiber angles and stacking sequences of composite laminates, made of Carbon/Epoxy, Kevlar/Epoxy, and E-Glass/Epoxy unidirectional composites, are investigated and compared with the minimum design requirements. Laminates combined of aluminum and Carbon/Epoxy unidirectional composite are also investigated. Thus, in this study various lay-up designs are proposed and finally, an optimum design is chosen among the various lay-up designs satisfying the design requirements based on mass and cost considerations. The various designs are also compared with a steel drive shaft. The properties of metals used in this study are presented in Table 4.

III. MECHANICAL MODELING

A. Lamination Theory

A thin fiber-reinforced laminated composite hollow shaft is shown in Fig.2. Coordinate x is considered for the axial direction, θ for the circumferential direction, and z for the radial (thickness) direction. The $x-\theta$ plane is equidistant from the top and bottom surfaces of the laminate and is called the mid-plane or the reference plane. Fig.3 shows the material principle coordinate system $(1,2,3)$ with respect to the cylindrical coordinate system (x,θ,z) . Material principle axes along and transverse to the fiber direction for kth layer are presented by 1, 2, and 3, respectively. The angle φ is measured positive counterclockwise from the $x-$axis to the 1-axis. The classical lamination theory is used to analysis the material properties of the composite drive shaft. The lamina is thin and if no out-of-plane loads applied, it is considered as the plane stress problem. Hence, it is possible to reduce a 3-D problem into a 2-D problem. From the properties of the composite materials along and transverse to the fiber direction, components of the principle reduced stiffness matrix, Q_{ij} $(i,j = 1,2,6)$, in terms of engineering constants are given by [5]-[9],

$$Q_{11} = \frac{E_{11}}{1-\nu_{12}\nu_{21}}, \qquad Q_{12} = Q_{21} = \frac{\nu_{12}E_{22}}{1-\nu_{12}\nu_{21}},$$

$$Q_{22} = \frac{E_{22}}{1 - v_{12}v_{21}}, \quad Q_{66} = G_{12}, \quad (1)$$

and

$$v_{21} = \frac{E_{22}}{E_{11}}v_{12}. \quad (2)$$

The transformed reduced stiffness matrix, Q_{ij} $(i, j = x, \theta, s)$, as a function of the principal lamina stiffnesses, are given by [5]-[9],

$$\begin{aligned}
Q_{xx} &= m^4Q_{11} + n^4Q_{22} + 2m^2n^2Q_{12} + 4m^2n^2Q_{66},\\
Q_{\theta\theta} &= n^4Q_{11} + m^4Q_{22} + 2m^2n^2Q_{12} + 4m^2n^2Q_{66},\\
Q_{x\theta} &= m^2n^2Q_{11} + m^2n^2Q_{22} + (m^4 + n^4)Q_{12} - 4m^2n^2Q_{66},\\
Q_{xs} &= m^3nQ_{11} - mn^3Q_{22} - mn(m^2 - n^2)Q_{12} - 2mn(m^2 - n^2)Q_{66},\\
Q_{\theta s} &= mn^3Q_{11} - m^3nQ_{22} + mn(m^2 - n^2)Q_{12} + 2mn(m^2 - n^2)Q_{66},\\
Q_{ss} &= m^2n^2Q_{11} + m^2n^2Q_{22} - 2m^2n^2Q_{12} + (m^2 - n^2)^2Q_{66}, \quad (3)
\end{aligned}$$

where $m = \cos\varphi$, and $n = \sin\varphi$.

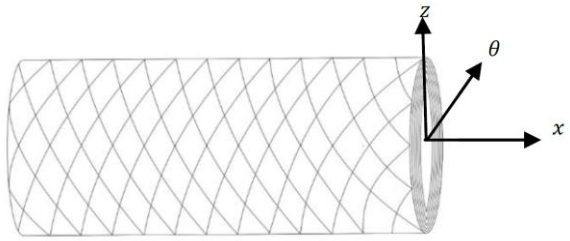


Fig. 2. Configuration of a fiber-reinforced laminated composite hollow shaft.

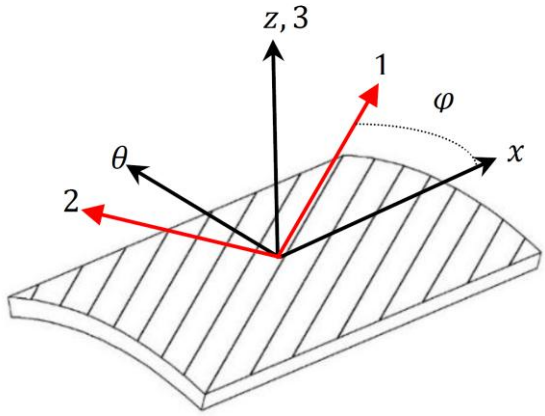


Fig. 3. Material principle and cylindrical coordinate systems.

The in-plane elastic constants for a balanced symmetric composite shaft, with total thickness t are given by,

$$\begin{aligned}
E_{xx} &= \frac{1}{t}\left(A_{xx} - \frac{A_{x\theta}^2}{A_{\theta\theta}}\right), \quad E_{\theta\theta} = \frac{1}{t}\left(A_{\theta\theta} - \frac{A_{x\theta}^2}{A_{xx}}\right),\\
G_{x\theta} &= \frac{A_{ss}}{t}, \quad (4)
\end{aligned}$$

where E_{xx} and $E_{\theta\theta}$ are the average moduli in the axial and the circumferential (hoop) directions, respectively, $G_{x\theta}$ is the shear modulus, t is the total wall thickness, and A_{ij} $(i, j = x, \theta, s)$ are components of the extensional stiffness matrix, [A], which are the summation of product of the transformed reduced stiffness matrix and the thickness of each layer as,

$$A_{ij} = \sum_{k=1}^{n} Q_{ij}^{k}(z_k - z_{k-1}) \quad (i, j = x, \theta, s), \quad (5)$$

where n is the total number of composite layers and z_k and z_{k-1} are the z −coordinate of the upper and lower surfaces of layer k, respectively from the reference plane.

B. Torsional Buckling Torque

A drive shaft is long, thin, and hollow, thus, there is a possibility of buckling. The expression of the critical torsional buckling torque for a thin-walled orthotropic tube is given by [10],

$$T_b = (2\pi r_m^2 t)(0.272)[E_{xx}E_{\theta\theta}^3]^{0.25}\left(t/r_m\right)^{1.5}, \quad (6)$$

where r_m is the mean radius. It is obvious that the stiffness modulus at hoop direction ($E_{\theta\theta}$) has a great importance in increasing the buckling resistance.

C. Bending Natural Frequency

A drive shaft should have a critical speed (60 times the frequency) that is higher than the rotational speed. The drive shaft is idealized as a simply supported beam (pinned-pinned beam) for which the bending natural frequency expression is given by [11],

$$f_n = \frac{\pi}{2L^2}\sqrt{\frac{E_{xx}J}{m_l}}, \quad (7)$$

where L is the length of the shaft, m_l is the mass per unit length which is given by,

$$m_l = \rho(2\pi r_m t), \quad (8)$$

where ρ is the density, and J is the polar moment of inertia of the thin-walled composite shaft (tube) which is given by,

$$J = \frac{\pi}{2}(r_o^4 - r_i^4), \quad (9)$$

where r_i and r_o are the inner and outer radii of the shaft, respectively. The mean radius is $r_m = \frac{r_i + r_o}{2}$ and the wall thickness is $t = r_o - r_i$, thus, the polar moment of inertia (J) can be written as,

$$\begin{aligned}
J &= \frac{\pi}{2}\left(\left(r_m + \frac{t}{2}\right)^4 - \left(r_m - \frac{t}{2}\right)^4\right)\\
&= 2\pi(r_m^3 t + t^3 r_m). \quad (10)
\end{aligned}$$

Property	Carbon/ Epoxy	Kevlar/ Epoxy	E-Glass/ Epoxy
Density, $\rho\ (Kg/m^3)$	1600	1380	1970
Longitudinal modulus, $E_{11}\ (GPa)$	147	80	41
Transverse modulus, $E_{22}\ (GPa)$	10.3	5.5	10.4
In-plane shear modulus, $G_{12}(GPa)$	7	2.2	4.3
Major Poisson's ratio, v_{12}	0.27	0.34	0.28
Minor Poisson's ratio, v_{21}	0.02	0.02	0.06
Longitudinal tensile strength, $F_{1t}\ (MPa)$	2280	1400	1140
Transverse tensile strength, $F_{2t}\ (MPa)$	57	30	39
In–plane shear strength, $F_6\ (MPa)$	76	49	89
Longitudinal compressive strength, $F_{1c}\ (MPa)$	1725	335	620
Transverse compressive strength, $F_{2c}\ (MPa)$	228	158	128

Table 3. Properties of Carbon/Epoxy, Kevlar/Epoxy, and E-Glass/Epoxy unidirectional composites (two-dimensional) [2].

Property	Aluminum (2024 T3)	Steel (AISI 1025)
Density, $\rho\ (Kg/m^3)$	2800	7800
Young's modulus, $E\ (GPa)$	73	207
Shear modulus, $G\ (GPa)$	26.6	79
Poisson's ratio, v	0.33	0.3
Tensile strength, $F_t\ (MPa)$	414	394
Compressive strength, $F_c\ (MPa)$	414	394
Shear strength, $F_6\ (MPa)$	248	248

Table 4. Properties of typical structural metals [2].

For a thin-walled shaft $t \ll r_m$, thus, the last term of "(10)", ($t^3 r_m$), is negligible and we approximate to,

$$J \approx 2\pi r_m^3 t. \qquad (11)$$

D. Load Carrying Capacity

Aim of this study is to design a composite drive shaft to carry the torque without failure. The torque at which the shaft fails or the torsional strength (T_{max}) is given by [12],

$$T_{max} = \frac{\tau_{all} J}{c}, \qquad (12)$$

where τ_{all} is the laminate shear strength and for a thin-walled shaft, $J \approx 2\pi r_m^3 t$ and $c = r_m$, thus, the in-plane shear strength (τ_{all}) and the torsional strength (T_{max}) are obtained as,

$$\tau_{all} = \frac{T_{max}}{2\pi r_m^2 t}, \qquad (13)$$

$$T_{max} = 2\pi r_m^2 t \tau_{all}. \qquad (14)$$

Therefore, the resultant shear force (shear flow), $N_{x\theta}$ in (N/m), due to the applied torque is given by [12],

$$N_{x\theta} = \tau_{x\theta} t = \frac{T}{2\pi r_m^2}, \qquad (15)$$

where torque T is the peak torque if the design involves fatigue consideration. Since the laminate is assumed to be failed at the failure of the first ply, the Tsai-Wu failure criterion can be used after finding the in-plane stresses at each ply to specify a factor of safety for torque transmission capacity. The first step is to construct the inverse of the extensional stiffness matrix [A] and after solving for the overall strains, the stresses in each layer can be examined by transforming these stresses to the direction of fiber at each layer. The $[A]^{-1}$ matrix multiplied by the through thickness resultant forces matrix gives the resultant strain as follows,

$$\begin{Bmatrix} \varepsilon_{xx} \\ \varepsilon_{\theta\theta} \\ \gamma_{x\theta} \end{Bmatrix} = A^{-1} \begin{Bmatrix} N_{xx} \\ N_{\theta\theta} \\ N_{x\theta} \end{Bmatrix} = A^{-1} \begin{Bmatrix} 0 \\ 0 \\ N_{x\theta} \end{Bmatrix}, \qquad (16)$$

where the axial force $N_{xx} = 0$, the circumferential force $N_{\theta\theta}$ is neglected, and $N_{x\theta}$ is the resultant shear force. The resultant strains are transformed to the fiber direction by multiplying these strain matrices by the transformation matrix. Then, the plane stresses can be obtained. The resultant strains and stresses in the principle material axes are given, respectively by,

$$\begin{Bmatrix} \varepsilon_{11} \\ \varepsilon_{22} \\ \gamma_{66} \end{Bmatrix} = \begin{bmatrix} m^2 & n^2 & mn \\ n^2 & m^2 & -mn \\ -2mn & 2mn & m^2 - n^2 \end{bmatrix} \begin{Bmatrix} \varepsilon_{xx} \\ \varepsilon_{\theta\theta} \\ \gamma_{x\theta} \end{Bmatrix}, \qquad (17)$$

$$\begin{Bmatrix} \sigma_{11} \\ \sigma_{22} \\ \tau_{66} \end{Bmatrix} = \begin{bmatrix} Q_{11} & Q_{12} & 0 \\ Q_{21} & Q_{22} & 0 \\ 0 & 0 & Q_{66} \end{bmatrix} \begin{Bmatrix} \varepsilon_{11} \\ \varepsilon_{22} \\ \gamma_{66} \end{Bmatrix}. \qquad (18)$$

E. Tsai-Wu Failure Criterion

In this study, Tsai-Wu failure criterion is used to calculate the shear strength of each proposed laminate design. For a two-dimensional state of stress ($\sigma_{11}, \sigma_{22}, \tau_{66}$), the Tsai-Wu failure criterion is reduced to the most familiar form [2],

$$f_1\sigma_{11} + f_2\sigma_{22} + f_{11}\sigma_{11}^2 + f_{22}\sigma_{22}^2 + f_{66}\tau_{66}^2 + 2f_{12}\sigma_{11}\sigma_{22} = 1, \qquad (19)$$

where

$$f_1 = \frac{1}{F_{1t}} - \frac{1}{F_{1c}}, \quad f_2 = \frac{1}{F_{2t}} - \frac{1}{F_{2c}}, \quad f_{11} = \frac{1}{F_{1t}F_{1c}},$$
$$f_{22} = \frac{1}{F_{2t}F_{2c}}, \quad f_{66} = \frac{1}{F_6^2}, \quad f_{12} = -\frac{1}{2}\sqrt{f_{11}f_{22}}, \qquad (20)$$

where F_{1t} and F_{1c} are respectively the longitudinal tensile and the compressive strength, F_{2t} and F_{2c} are respectively the transverse tensile and the compressive strength, and F_6 is the shear strength which are given in Table 3.

The safety factor S_f for a given two-dimensional state of stress $(\sigma_{11}, \sigma_{22}, \tau_{66})$ is a multiplier that is applied to all stress components to produce a critical or failure state as defined by the Tsai-Wu criterion. Thus, for a given state of stress $(\sigma_{11}, \sigma_{22}, \tau_{66})$, the state of stress at failure is $(S_f\sigma_{11}, S_f\sigma_{22}, S_f\tau_{66})$. Substitution of the critical stress in the Tsai-Wu criterion in "(19)" yields,

$$f_1 S_f\sigma_{11} + f_2 S_f\sigma_{22} + f_{11} S_f^2\sigma_{11}^2 + f_{22} S_f^2\sigma_{22}^2 + f_{66} S_f^2\tau_{66}^2 + 2f_{12} S_f^2\sigma_{11}\sigma_{22} = 1, \quad (21)$$

or

$$aS_f^2 + bS_f - 1 = 0, \quad (22)$$

where

$$a = f_{11}\sigma_{11}^2 + f_{22}\sigma_{22}^2 + f_{66}\tau_{66}^2 + 2f_{12}\sigma_{11}\sigma_{22}, \quad b = f_1\sigma_{11} + f_2\sigma_{22}. \quad (23)$$

Thus, the problem of determining the safety factor is reduced to that of solving "(22)". The roots of "(22)" are,

$$S_{fa} = \frac{-b + \sqrt{b^2 + 4a}}{2a} \text{ (actual state of stress)}, \quad S_{fr} = \frac{-b - \sqrt{b^2 + 4a}}{2a} \text{ (actual state of stress)}. \quad (24)$$

To obtain the shear strength, F_s, referred to the $x-$ and $\theta-$ axes, following stresses in the cylindrical coordinate system are considered,

$$\sigma_{xx} = 0, \quad \sigma_{\theta\theta} = 0, \quad \tau_{x\theta} = 1. \quad (25)$$

From "(25)", σ_{11}, σ_{22}, and τ_{66}, referred to the principle axes, are obtained as,

$$\sigma_{11} = 2mn, \quad \sigma_{22} = -2mn, \quad \tau_{66} = m^2 - n^2. \quad (26)$$

Substituting "(26)" into "(22)" yields the following shear strength,

$$F_s^{(+)} = S_{fa} \text{ (positive shear strength)}, \quad F_s^{(-)} = S_{fr} \text{ (negative shear strength)}. \quad (27)$$

IV. DESIGN SPECIFICATIONS

Based on the design requirements of this study, given in Tables 1 and 2, minimum bending natural frequency and minimum buckling torque are 90 Hz and $2030\ N-m$, respectively. Thus, for each composite lay-up design, the critical torsional buckling torque (T_b), obtained from (6), and the bending natural frequency (f_n), obtained from (7) should be higher than $2030\ N-m$ and 90 Hz, respectively.

According to the design requirements, the peak torque for 100,000 cycle reversed fatigue is 678 N-m, thus, the shear load capacity ($N_{x\theta}$) is obtained by substituting the peak torque value into (15) as $41814.066\ N/m$. In this study, this peak load is considered as the first-ply failure load. The shear strength (or the shear load capacity) of a proposed design should be higher than this load ($41814.066\ N/m$). Tsai-Wu failure criterion is used to calculate the shear strength for each lamina from the first-ply failure load of the laminate.

V. RESULTS AND DISCUSSIONS

A. Design I: Carbon/Epoxy Laminate

For the first design, composite drive shafts made of Carbon/Epoxy laminate with lay-ups of $[\pm 45]_s$, $[90_2/0_2]_s$, and $[90_2/\pm 45/0_2]_s$ are considered. Table 5 shows a comparison between the results obtained from the above lay-ups and the design requirements of the drive shaft. As shown in Table 5, for a composite drive shaft made of Carbon/Epoxy with an angle-ply laminate of $[\pm 45]_s$, the shear load capacity ($N_{x\theta}$) and the bending natural frequency (f_n) satisfy the design requirements, while the torsional buckling torque (T_b) is lower than the minimum requirement. The shear load capacity ($N_{x\theta}$) and the bending natural frequency (f_n) of a composite drive shaft with cross-ply laminate of $[90_2/0_2]_s$ meet the minimum design requirements, while the torsional buckling torque (T_b) of this lay-up is not sufficient, however, it is higher than that of the angle-ply lay-up $[\pm 45]_s$. Thus, it can be concluded that by a cross-ply lay-up, higher torsional buckling torque can be obtained in comparison to an angle-ply one. Hence, a design which is a combination of angle-ply and cross-ply lay-ups may satisfy all design requirements. Accordingly, a composite drive shaft with a lay-up of $[90_2/\pm 45/0_2]_s$ is proposed. As shown in Table 5, this design has a shear load capacity ($N_{x\theta}$) of $282987.426\ N/m$, a torsional buckling torque (T_b) of $2220.477\ N-m$, and the bending natural frequency (f_n) of 171.435 Hz. Also, its mass is 1.325 Kg. Thus, a composite drive shaft made of Carbon/Epoxy with a lay-up of $[90_2/\pm 45/0_2]_s$ can satisfy all design requirements.

B. Design II: Kevlar/Epoxy Laminate

Based on the results obtained from the lay-up of Carbon/Epoxy, in the first design, composite drive shafts made of Kevlar/Epoxy laminate with lay-ups of $[90_2/\pm 45/0_2]_s$ and $[90_2/\pm 45/0_4]_s$ are considered as the second design. A comparison between the results obtained from shafts made of Kevlar/Epoxy with the above lay-ups and the design requirements is presented in Table 6. The results indicate that a composite drive shaft made of Kevlar/Epoxy with a lay-up of $[90_2/\pm 45/0_4]_s$ can satisfy all design requirements. It is obvious that a heavier composite drive shaft made of the Kevlar/Epoxy laminate with a mass of 1.524 Kg is needed to satisfy the design requirements compared to that of the Carbon/Epoxy with a mass of 1.325 Kg. It is due to lower mechanical properties of the Kevlar/Epoxy material such as elastic moduli than the Carbon/Epoxy material.

C. Design III: E-Glass/Epoxy Laminate

For the third design, the E-Glass/Epoxy lay-up is considered for the composite drive shaft. Table 7 shows a comparison between the minimum design requirements and the results obtained from a composite drive shaft made of E-Glass/Epoxy laminate with lay-ups of $[90_2/\pm45/0_2]_s$, $[90_3/\pm45/03s$, and $904/\pm45/04s$. By increasing the number of layers to 20 and the mass of 2.719 Kg, the design requirements can be satisfied. It is obvious that a heavier composite drive shaft made of E-Glass/Epoxy laminate is needed to satisfy the design requirements compared to that of Carbon/Epoxy and Kevlar/Epoxy due to its lower mechanical properties.

D. Design IV: Hybrid Laminate

Final proposed design is a hybrid laminate made of Carbon/Epoxy and aluminum. A lay-up of $[90/0/AL/0/90$ is considered for the hybrid drive shaft and the obtained results are compared with the minimum design requirements. As shown in Table 8, a hybrid laminate of Carbon/Epoxy and aluminum with a lay-up of $[90/0/AL/0/90]$ and with a mass of 1.987 Kg satisfies all design requirements of the shaft. Although a drive shaft made of pure aluminum satisfy all design requirements but the mass will be higher than the hybrid laminate one. Table 8 also shows the obtained results for a drive shaft made of steel and aluminum. The results indicate that weight of a steel drive shaft is much higher than the weight of composite laminate, hybrid laminate, and aluminum drive shafts.

E. Evaluation of the Proposed Design

To determine an optimum design among various proposed lay-ups and materials, two considerations of weight and cost are necessary. Table 9 shows a comparison between the various designs satisfying the design requirements. According to the proposed designs, the mass of Aluminum, AL/Carbon/Epoxy, E-Glass/Epoxy, Kevlar/Epoxy, and Carbon/Epoxy drive shafts satisfying the design requirements are 2.319 Kg, 1.987 Kg, 2.719 Kg, 1.524 Kg, and 1.325 Kg, respectively. The weight savings of pure Aluminum, AL/Carbon/Epoxy, E-Glass/Epoxy, Kevlar/Epoxy, and Carbon/Epoxy drive shafts are equal to 46.157%, 53.865%, 36.87%, 64.615%, and 69.236% of the weight of the steel drive shaft, respectively. Therefore, if weight is the main consideration, the proposed lay-up of Carbon/Epoxy is better design than other proposed designs. On the other hand, the cost per unit weight of Aluminum is lower than, E-Glass/Epoxy, Kevlar/Epoxy, and Carbon/Epoxy. Thus, if cost is the main consideration, the hybrid laminate made of AL/Carbon/Epoxy can be better choice.

VI. CONCLUSION

In this study, various designs of composite drive shaft made of Carbon/Epoxy, Kevlar/Epoxy, E-Glass/Epoxy and a hybrid shaft made of AL/Carbon/Epoxy laminate have been proposed. Different designs were investigated and analyzed to find an optimum design minimizing the weight of the shaft and satisfying the design requirements including the torque transmission capacities, the torsional buckling capacities, and the bending natural frequency. If weight is the main consideration, the shaft made of the Carbon/Epoxy laminate gives better performance and if cost is the main concern, the hybrid one can be better option.

Laminate lay- up	Number of plies	Laminate thickness (mm)	$N_{x\theta}\ (N/m)$	$T_b\ (N-m)$	$f_n\ (Hz)$	$M\ (Kg)$
$[\pm45]_s$	4	0.5	509239.947	51.486	103.067	0.441
$[90_2/0_2]_s$	8	1	76000	961.745	187.292	0.883
$[90_2/\pm45/0_2]_s$	12	1.5	282987.426	2220.477	171.435	1.325
Design requirements	-		41814.066	2030	90	-

Table 5. Comparison between the results obtained from laminated composite drive shafts made up of Carbon/Epoxy and the minimum design requirements.

Laminate lay- up	Number of plies	Laminate thickness (mm)	$N_{x\theta}\ (N/m)$	$T_b\ (N-m)$	$f_n\ (Hz)$	$M\ (Kg)$
$[90_2/\pm45/0_2]_s$	12	1.5	278450.833	1182.142	134.688	1.143
$[90_2/\pm45/0_4]_s$	16	2	302950.833	2202.661	154.655	1.524
Design requirements	-	-	41814.066	2030	90	-

Table 6. Comparison between the results obtained from laminated composite drive shafts made up of Kevlar/Epoxy and the minimum design requirements

Laminate lay- up	Number of plies	Laminate thickness (mm)	$N_{x\theta}\ (N/m)$	$T_b\ (N-m)$	$f_n\ (Hz)$	$M\ (Kg)$
$[90_2/\pm45/0_2]_s$	12	1.5	209307.179	756.85	90.2	1.631
$[90_3/\pm45/0_3]_s$	16	2	253807.179	1615.467	91.977	2.175
$[90_4/\pm45/0_4]_s$	20	2.5	298307.179	2884.329	92.984	2.719
Design requirements	-		41814.066	2030	90	-

Table 7. Comparison between the results obtained from laminated composite drive shafts made up of E-Glass/Epoxy and the minimum design requirements.

Laminate lay- up	Number of plies	Laminate thickness (mm)	$N_{x\theta}\ (N/m)$	$T_b\ (N-m)$	$f_n\ (Hz)$	$M\ (Kg)$
Pure Steel (AISI 1025)	1	1	248000	2521.453	137.35	4.307
Pure AL (2024 T3)	1	1.5	372000	2450.369	136.135	2.319
$[90/0/AL/0/90]$	5	1.5	305300	2568.796	150.554	1.987
Design requirements	-		41814.066	2030	90	-

Table 8. Comparison between the results obtained from laminated composite drive shafts made up of carbon/epoxy and aluminum and the minimum design requirements.

Parameter	Steel (AISI 1025)	Aluminum (2024 T3)	Hybrid AL/Carbon/Epoxy	E-Glass/Epoxy	Kevlar/Epoxy	Carbon/ Epoxy
Ply thickness (mm)	1	1.5	AL: 1 Carbon/Epoxy: 0.125	0.125	0.125	0.125
Number of plies	1	1	5	20	16	12
Laminate thickness (mm)	1	1.5	1.5	2.5	2	1.5
Stacking sequence	-	-	$[90/0/AL/0/90]$	$[90_4/\pm45/0_4]_s$	$[90_2/\pm45/0_4]_s$	$[90_2/\pm45/0_2]_s$
Weight (Kg)	4.307	2.319	1.987	2.719	1.524	1.325
Weight saving (%)	-	46.157	53.865	36.87	64.615	69.236

Table 9. Comparison between the various designs satisfying the design requirements.

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