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DESIGN & OPTIMIZATION OF AUTOMOTIVE COMPOSITE DRIVE SHAFT

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ABSTRACT

Substituting composite materials for conventional metallic structures has many advantages because of higher specific stiffness and strength of composite materials. Advanced composite materials seem ideally suited for long, power drive shaft applications. Their elastic properties can be tailored to increase the torque and the rotational speed at which they operate. This study has been carried out to investigate maximum torque; buckling torque transmission and critical speed of composite drive shaft. Main aim of this work is to investigate either replacing steel structure of drive shaft; for rear wheel drive passenger cars; by composite structures such as carbon/Epoxy and Glass/Epoxy materials will be convenient or not. For finding out the suitability of composite structures for automotive drive shaft application the parameters such as; ply thickness, number of plies and stacking sequence are optimized for carbon/Epoxy and Glass/Epoxy shafts using an optimization tool with the objective of weight minimization of the composite shaft which is subjected to constraints such as torque transmission, torsional buckling load and fundamental natural frequency.

KEYWORDS: *Composite material, Carbon-Epoxy- glass, Drive Shaft, weight optimization*

I. INTRODUCTION

An automotive drive shaft transmits power from the engine to the differential gear of a rear wheel drive vehicle. The conventional steel drive shaft is usually manufactured in two pieces to increase the fundamental bending natural frequency because the bending natural frequency of a shaft is inversely proportional to the square of beam length and proportional to the square root of specific modulus[16]. The two-piece steel drive shaft consists of three universal joints, a center supporting bearing and a bracket, which increases the total weight of an automotive vehicle and decreases fuel efficiency.

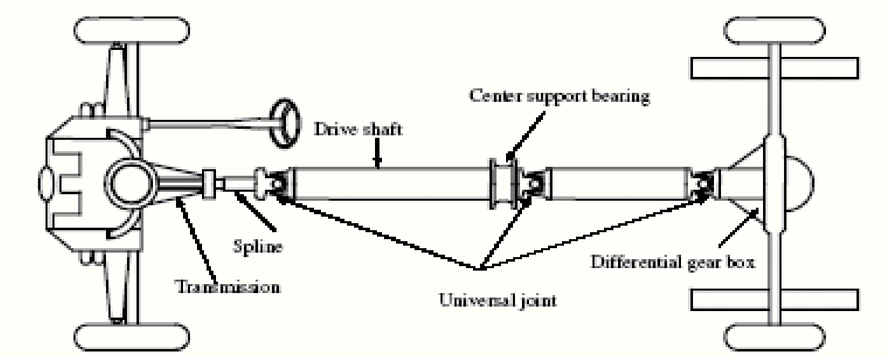

Figure 1: Conventional two-piece drive shaft arrangement for rear wheel vehicle driving system

Composite materials such as Graphite, Carbon, Kevlar and Glass with suitable resins can be used instead of steel to reduce the drive shaft weight. These materials have high specific strength (strength/density) and high specific modulus (modulus/density) [1]. The automotive industry is exploiting composite material technology for structural components construction in order to obtain the reduction of the weight without decrease in vehicle quality and reliability. It is known that energy conservation is one of the most important objectives in vehicle design and reduction of weight is one of the most effective measures to obtain this result. Actually, there is almost a direct proportionality between the weight of a vehicle and its fuel consumption, particularly in city driving. The fundamental natural frequency of the carbon fiber composite drive shaft can be twice as high as that of steel or aluminum because the carbon fiber composite material has more than 4 times the specific stiffness of steel or aluminum, which makes it possible to manufacture the drive shaft of passenger cars in one piece. A one-piece composite shaft can be manufactured so as to satisfy the vibration requirements. This eliminates all the assembly, connecting the two piece steel shafts and thus minimizes the overall weight, vibrations and the total cost.

II. SPECIFICATION OF THE PROBLEM

The fundamental natural bending frequency for passenger cars, small trucks, and vans of the propeller shaft should be higher than 6,500 rpm to avoid whirling vibration and the torque transmission capability of the drive shaft should be larger than 3,500 Nm. The drive shaft outer diameter should not exceed 100 mm due to space limitations. Here outer diameter of the shaft is taken as 90 mm. The drive shaft of transmission system is to be designed optimally for following specified design requirements as shown in Table 1 [18].

Table 1: Design requirements and specifications

Sr. No.	Name	Notation	Unit	Value
1	Ultimate Torque	T_{max}	Nm	3500
2	Max. Speed of shaft	N_{max}	rpm	6500
3	Length of shaft	L	mm	1250

III. DESIGN OF STEEL DRIVE SHAFT

Steel used for automotive drive shaft applications. The material properties of the steel are given in Table 2 [19]. The steel drive shaft should satisfy three design specifications such as torque transmission capability, buckling torque capability and bending natural frequency.

Table 2 : Mechanical properties of Steel

Mechanical Properties	Symbol	Units	Steel
Young's Modulus	E	Gpa	207.0
Shear Modulus	G	GPa	80.0
Poisson's Ratio	N	-	0.3
Density	P	Kg/m^3	7600.0
Yield Strength	S_y	Mpa	370.0
Shear Strength	S_s	MPa	-

3.1 Torque Transmission capacity of the Drive Shaft

Torque transmission capacity T of a steel drive shaft is given by,

$$T = S_s \frac{\pi(d_o^4 - d_i^4)}{16td_o} \quad ....(1)$$

Where S_s is the shear strength, d_o and d_i represent outside and inside diameter of the steel shaft.

3.2 Torsional Buckling Capacity of the Drive Shaft

$$\text{If } \frac{1}{\sqrt{1-v^2}} \frac{L^2 t}{(2r)^3} > 5.5 \quad ....(2)$$

It is called as Long shaft otherwise it is called as Short & Medium shaft [20].
For long shaft, the critical stress is given by,

$$\tau_{cr} = \frac{E}{3\sqrt{2}(1-v^2)^{3/4}}(t/r)^{3/2} \qquad ....(3)$$

For short & medium shaft, the critical stress is given by,

$$\tau_{cr} = \frac{4.39E}{(1-v^2)}(t/r)^2 \sqrt{1 + 0.0257(1-v^2)^{3/4}\frac{L^3}{(rt)^{1.5}}} \qquad ....(4)$$

Where E, and ν represent steel properties. L, t and r are the length, thickness and mean radius of the shaft respectively.
The relation between the torsional buckling capacity and critical stress is given by,

$$T_{cr} = \tau_{cr} 2\pi r^2 t \qquad ....(5)$$

3.3 Lateral or Bending Vibration

The shaft is considered as simply supported beam undergoing transverse vibration or can be idealized as a pinned-pinned beam. Natural frequency is calculated using Timoshenko beam theory [21]. Natural frequency based on the Timoshenko beam theory is given by,

$$f_{nt} = K_s \frac{30\pi p^2}{L^2}\sqrt{\frac{Er^2}{2\rho}} \qquad .....(6)$$

Where f_{nt} is the natural frequency and p is the first natural frequency. E and ρ are the material properties of the steel shaft, and K_s is given by:

$$N_{crt} = 60 f_{nt} \qquad .....(7)$$

$$\frac{1}{K_s^2} = 1 + \frac{n^2\pi^2 r^2}{2L^2}\left[1 + \frac{f_s E}{G}\right] \qquad .....(8)$$

$f_s = 2$ for hollow circular cross-sections.

IV. DESIGN OF A COMPOSITE DRIVE SHAFT

4.1 Assumptions

1. The shaft rotates at a constant speed about its longitudinal axis.
2. The shaft has a uniform, circular cross section.
3. The shaft is perfectly balanced, i.e., at every cross section, the mass center coincides with the geometric center.
4. All damping and nonlinear effects are excluded.
5. The stress-strain relationship for composite material is linear & elastic; hence, Hooke's law is applicable for composite materials.
6. Acoustical fluid interactions are neglected, i.e., the shaft is assumed to be acting in a vacuum.
7. Since lamina is thin and no out-of-plane loads are applied, it is considered as under the plane stress.

4.2 Selection of Cross-Section

The drive shaft can be solid circular or hollow circular. Here hollow circular cross-section was chosen because:
• The hollow circular shafts are stronger in per kg weight than solid circular.
• The stress distribution in case of solid shaft is zero at the center and maximum at the outer surface while in hollow shaft stress variation is smaller. In solid shafts the material close to the center are not fully utilized.

4.3 Selection of Materials

Based on the advantages discussed earlier, the E-Glass/Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy materials are selected for composite drive shaft. The Table 3 shows the properties of the E-Glass/Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy materials used for composite drive shafts. E_{11}, E_{22}, G_{12}, S^t_1, S^c_1, S^t_2 and S^c_2 represent lamina properties in longitudinal and transverse directions respectively. ν, τ_{12} and ρ are the Poisons ratio, shear stress and material mass density.

Table 3: Properties of E-Glass/Epoxy, HS Carbon/Epoxy and HM Carbon/Epoxy

Sr. No.	Property	Units	E-Glass/Epoxy	HS Carbon/Epoxy	HM Carbon/Epoxy
1	E11	Gpa	50.0	134.0	190.0
2	E22	Gpa	12.0	7.0	7.7
3	G12	Gpa	5.6	5.8	4.2
4	ν	-	0.3	0.3	0.3
5	$S^t_1=S^c_1$	Mpa	800.0	880.0	870.0
6	$S^t_2=S^c_2$	Mpa	40.0	60.0	54.0
7	S_{12}	Mpa	72.0	97.0	30.0
8	ρ	Kg/m^3	2000.0	1600.0	1600.0

The designer must take into account the factor of safety when designing a structure. Since, composites are highly orthotropic and their fractures were not fully studied the factor of safety was taken as 2.

4.4 Torque Transmission Capacity of the Shaft

4.4.1 Stress-Strain Relationship for Unidirectional Lamina

The lamina is thin and if no out-of-plane loads are applied, it is considered as the plane stress problem. Hence, it is possible to reduce the 3-D problem into 2-D problem.
For unidirectional 2-D lamina, the stress-strain relationship is given by,

$$\begin{Bmatrix}\sigma_1\\ \sigma_2\\ \tau_{12}\end{Bmatrix} = \begin{bmatrix}Q_{11} & Q_{12} & 0\\ Q_{12} & Q_{22} & 0\\ 0 & 0 & Q_{66}\end{bmatrix}\begin{Bmatrix}\varepsilon_1\\ \varepsilon_2\\ \gamma_{12}\end{Bmatrix} \quad ....(9)$$

Where σ, τ, γ and ε represent stresses and strains in material directions. The matrix Q is referred to as the reduced stiffness matrix for the layer and its terms are given by:

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

$$Q_{22} = \frac{E_{22}}{1 - v_{12}v_{21}} \; Q_{66}=G_{12} \qquad Q_{21}=Q_{12} \quad ....(10)$$

4.4.2 Stress-Strain Relationship for Angle-ply Lamina

The relation between material coordinate system and X-Y-Z coordinate system is shown in Fig 2.Coordinates 1, 2, 3 are principal material directions and coordinates X, Y, Z are transformed or laminate axes.

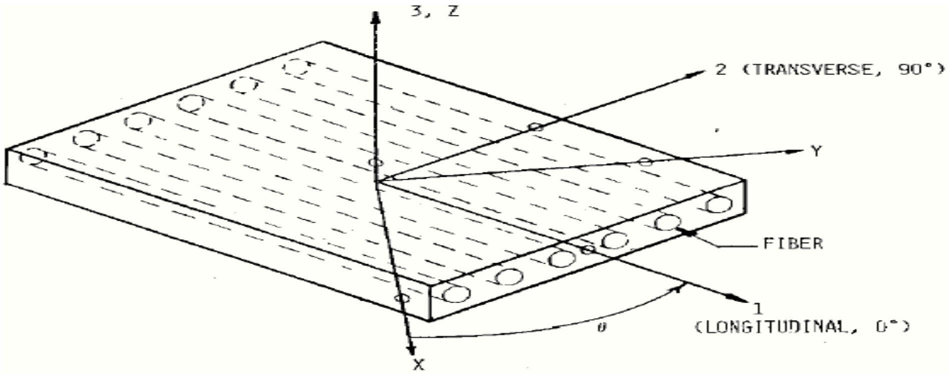

Figure 2: Relation between material coordinate system and X-Y coordinate system

For an angle-ply lamina where fibers are oriented at an angle with the positive X-axis (Longitudinal axis of shaft), the effective elastic properties are given by,

$$\frac{1}{E_{xlamina}} = \frac{1}{E_{11}}C^4 + \left[\frac{1}{G_{12}} - \frac{2v_{12}}{E_{11}}\right]S^2C^2 + \frac{1}{E_{22}}S^4 \quad ....(11)$$

$$\frac{1}{E_{ylamina}} = \frac{1}{E_{11}}S^4 + \left[\frac{1}{G_{12}} - \frac{2v_{12}}{E_{11}}\right]S^2C^2 + \frac{1}{E_{22}}C^4 \quad ....(12)$$

$$\frac{1}{G_{xylamina}} = 2\left[\frac{2}{E_{11}} + \frac{2}{E_{22}} + \frac{2v_{12}}{E_{11}} - \frac{1}{G_{12}}\right]S^2C^2 + \frac{1}{G_{12}}[C^4 + S^4] \quad ....(13)$$

The stress strain relationship for an angle-ply lamina is given by,

$$\begin{Bmatrix}\sigma_x\\ \sigma_x\\ \tau_{xy}\end{Bmatrix} = \begin{bmatrix}\overline{Q_{11}} & \overline{Q_{12}} & \overline{Q_{16}}\\ \overline{Q_{12}} & \overline{Q_{22}} & \overline{Q_{26}}\\ \overline{Q_{16}} & \overline{Q_{26}} & \overline{Q_{66}}\end{bmatrix}\begin{Bmatrix}\varepsilon_x\\ \varepsilon_y\\ \gamma_{xy}\end{Bmatrix} \quad ....(14)$$

Where σ and ε represent normal stresses and strains in X, Y and XY directions respectively and bar over Q_{ij} matrix denotes transformed reduced stiff nesses. Its terms are individually given by:

$$\begin{aligned}
\overline{Q_{11}} &= Q_{11}C^4 + Q_{22}S^4 + 2(Q_{12} + 2Q_{66})S^2C^2\\
\overline{Q_{12}} &= (Q_{11} + Q_{22} - 4Q_{66})S^2C^2 + Q_{12}(C^4 + S^4)\\
Q_{16} &= (Q_{11} - Q_{12} - 2Q_{66})C^3S - (Q_{22} - Q_{12} - 2Q_{66})CS^3\\
Q_{22} &= Q_{11}S^4 + Q_{22}S^4 + 2(Q_{11} + 2Q_{66})S^2C^2\\
Q_{26} &= (Q_{11} - Q_{12} - 2Q_{66})CS^3 - (Q_{22} - Q_{12} - 2Q_{66})C^3S\\
Q_{66} &= (Q_{11} + Q_{22} - 2Q_{12} - 2Q_{66})S^2C^2 + Q_{66}(S^4 + C^4)
\end{aligned} \quad ....(15)$$

$$\begin{Bmatrix}N_x\\ N_y\\ N_{xy}\end{Bmatrix} = \begin{bmatrix}A_{11} & A_{12} & A_{16}\\ A_{12} & A_{22} & A_{26}\\ A_{16} & A_{26} & A_{66}\end{bmatrix}\begin{Bmatrix}\epsilon_x^0\\ \epsilon_y^0\\ \gamma_{xy}^0\end{Bmatrix} + \begin{bmatrix}B_{11} & B_{12} & B_{16}\\ B_{12} & B_{22} & B_{26}\\ B_{16} & B_{26} & B_{66}\end{bmatrix}\begin{Bmatrix}k_x\\ k_y\\ k_{xy}\end{Bmatrix} \quad ....(16)$$

$$\begin{Bmatrix}M_x\\ M_y\\ M_{xy}\end{Bmatrix} = \begin{bmatrix}B_{11} & B_{12} & B_{16}\\ B_{12} & B_{22} & B_{26}\\ B_{16} & B_{26} & B_{66}\end{bmatrix}\begin{Bmatrix}\epsilon_x^0\\ \epsilon_y^0\\ \gamma_{xy}^0\end{Bmatrix} + \begin{bmatrix}D_{11} & D_{12} & D_{16}\\ D_{12} & D_{22} & D_{26}\\ D_{16} & D_{26} & D_{66}\end{bmatrix}\begin{Bmatrix}k_x\\ k_y\\ k_{xy}\end{Bmatrix} \quad ....(17)$$

$$\begin{aligned}
A_{ij} &= \sum_{k=1}^{n}\left(\overline{Q_{ij}}\right)_k (h_k - h_{k-1}) \qquad & A_{ij} &= \sum_{k=1}^{n}\left(\overline{Q_{ij}}\right)_k t_k\\
B_{ij} &= \frac{1}{2}\sum_{k=1}^{n}\left(\overline{Q_{ij}}\right)_k (h_k^2 - h_{k-1}^2) \qquad & D_{ij} &= \frac{1}{3}\sum_{k=1}^{n}\left(\overline{Q_{ij}}\right)_k (h_k^3 - h_{k-1}^3)
\end{aligned} \quad ....(18)$$

Where i, j = 1 ,2, 6.

[A], [B], [D] matrices are called the extensional, coupling, and bending stiffness matrices respectively. By combining the equations 16 and 17,

$$\begin{Bmatrix}N_x\\ N_y\\ N_{xy}\\ M_x\\ M_y\\ M_{xy}\end{Bmatrix} = \begin{bmatrix}A_{11} & A_{12} & A_{16} & B_{11} & B_{12} & B_{16}\\ A_{12} & A_{22} & A_{26} & B_{12} & B_{22} & B_{26}\\ A_{16} & A_{26} & A_{66} & B_{16} & B_{26} & B_{66}\\ B_{11} & B_{12} & B_{16} & D_{11} & D_{12} & D_{16}\\ B_{12} & B_{22} & B_{26} & D_{12} & D_{22} & D_{26}\\ B_{16} & B_{26} & B_{66} & D_{16} & D_{26} & D_{66}\end{bmatrix}\begin{Bmatrix}\epsilon_x^0\\ \epsilon_y^0\\ \gamma_{xy}^0\\ k_x^0\\ k_y^0\\ k_{xy}^0\end{Bmatrix} \quad ....(19)$$

For symmetric laminates, the B matrix vanishes and the in plane and bending stiffness are uncoupled. For a symmetric laminate,

$$\begin{Bmatrix}N_x\\ N_y\\ N_{xy}\end{Bmatrix} = \begin{bmatrix}A_{11} & A_{12} & A_{16}\\ A_{12} & A_{22} & A_{26}\\ A_{16} & A_{26} & A_{66}\end{bmatrix}\begin{Bmatrix}\epsilon_x^0\\ \epsilon_y^0\\ \gamma_{xy}^0\end{Bmatrix} \quad ....(20)$$

$$\begin{Bmatrix} M_x \\ M_y \\ M_{xy} \end{Bmatrix} = \begin{bmatrix} D_{11} & D_{12} & D_{16} \\ D_{12} & D_{22} & D_{26} \\ D_{16} & D_{26} & D_{66} \end{bmatrix} \begin{Bmatrix} k_x^0 \\ k_y^0 \\ k_{xy}^0 \end{Bmatrix} \quad ....(21)$$

$$\begin{Bmatrix} \epsilon_x^0 \\ \epsilon_y^0 \\ \gamma_{xy}^0 \end{Bmatrix} = \begin{bmatrix} a_{11} & a_{12} & a_{16} \\ a_{12} & a_{22} & a_{26} \\ a_{16} & a_{26} & a_{66} \end{bmatrix} \begin{Bmatrix} N_x \\ N_y \\ N_{xy} \end{Bmatrix} \quad ....(22)$$

$$\begin{Bmatrix} k_x^0 \\ k_y^0 \\ k_{xy}^0 \end{Bmatrix} = \begin{bmatrix} d_{11} & d_{12} & d_{16} \\ d_{12} & d_{22} & d_{26} \\ d_{16} & d_{26} & d_{66} \end{bmatrix} \begin{Bmatrix} M_x \\ M_y \\ M_{xy} \end{Bmatrix} \quad ....(23)$$

Where

$$\begin{bmatrix} a_{11} & a_{12} & a_{16} \\ a_{12} & a_{22} & a_{26} \\ a_{16} & a_{26} & a_{66} \end{bmatrix} = \begin{bmatrix} A_{11} & A_{12} & A_{16} \\ A_{12} & A_{22} & A_{26} \\ A_{16} & A_{26} & A_{66} \end{bmatrix}^{-1} \quad ....(24)$$

$$\begin{bmatrix} d_{11} & d_{12} & d_{16} \\ d_{12} & d_{22} & d_{26} \\ d_{16} & d_{26} & d_{66} \end{bmatrix} = \begin{bmatrix} D_{11} & D_{12} & D_{16} \\ D_{12} & D_{22} & D_{26} \\ D_{16} & D_{26} & D_{66} \end{bmatrix}^{-1} \quad ....(25)$$

$$E_x = \frac{1}{a_{11}}\frac{1}{t} = \text{Young's Modulus of the shaft in axial direction} \quad ....(26)$$

$$E_y = \frac{1}{a_{22}}\frac{1}{t} = \text{Young's Modulus of the shaft in hoop direction} \quad ....(27)$$

$$G_{xy} = \frac{1}{a_{66}}\frac{1}{t} = \text{Rigidity Modulus of the shaft in xy plane} \quad ....(28)$$

$$\begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \gamma_{xy} \end{Bmatrix} = \begin{Bmatrix} \epsilon_x^0 \\ \epsilon_y^0 \\ \gamma_{xy}^0 \end{Bmatrix} + h \begin{Bmatrix} k_x^0 \\ k_y^0 \\ k_{xy}^0 \end{Bmatrix} \quad ....(29)$$

When a shaft is subjected to torque T, the resultant forces in the laminate by considering the effect of centrifugal forces are

$$N_x = 0 \qquad N_y = 2\rho t r^2 \omega^2 \qquad N_{xy} = \frac{T}{2\pi r^2} \quad ....(30)$$

Stresses in the K^{th} ply are given by,

$$\begin{Bmatrix} \sigma_x \\ \sigma_x \\ \tau_{xy} \end{Bmatrix}_k = \begin{bmatrix} \overline{Q_{11}} & \overline{Q_{12}} & \overline{Q_{16}} \\ \overline{Q_{12}} & \overline{Q_{22}} & \overline{Q_{26}} \\ \overline{Q_{16}} & \overline{Q_{26}} & \overline{Q_{66}} \end{bmatrix}_k \begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \gamma_{xy} \end{Bmatrix} \quad ....(31)$$

$$\begin{Bmatrix} \sigma_1 \\ \sigma_2 \\ \tau_{12} \end{Bmatrix}_k = \begin{bmatrix} C^2 & S^2 & 2CS \\ S^2 & C^2 & -2CS \\ -CS & CS & C^2 - S^2 \end{bmatrix} \begin{Bmatrix} \sigma_x \\ \sigma_x \\ \tau_{xy} \end{Bmatrix}_k \quad ....(32)$$

Knowing the stresses in each ply, the failure of the laminate is determined by using the First Ply Failure criteria. That is, the laminate is assumed to fail when the first ply fails. Here maximum stress theory is used to find the torque transmitting capacity.

4.5 Torsional Buckling Capacity (Tcr)

Since long thin hollow shafts are vulnerable to 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.

$$T_{cr} = (2\pi r^2 t)(0.272)\left(E_x E_y^3\right)^{0.25}(t/r)^{1.5} \quad ....(33)$$

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.

This equation has been generated from the equation of isotropic cylindrical shell and has been used for the design of drive shafts. From the equation 33, the torsional buckling capability of composite shaft is strongly dependent on the thickness of composite shaft and the average modulus in the hoop direction.

4.6 Lateral or Bending Vibration

The shaft is considered as simply supported beam undergoing transverse vibration or can be idealized as a pinned-pinned beam. Natural frequency based on the Timoshenko beam theory is given by,

$$f_{nt} = K_s \frac{30\pi p^2}{L^2}\sqrt{\frac{E_x r^2}{2\rho}} \quad ....(34)$$

$$N_{crt} = 60 f_{nt} \quad ....(35)$$

Where K_s = shear coefficient of the lateral natural frequency (<1)

$$\frac{1}{K_s^2} = 1 + \frac{n^2\pi^2 r^2}{2L^2}\left[1 + \frac{f_s E_x}{G_{xy}}\right] \quad ....(36)$$

f_s = 2 for hollow circular cross-sections

V. DESIGN OPTIMIZATION

Optimization of an engineering design is an improvement of a proposed design that results in the best properties for minimum cost. Most of the methods used for design optimization assume that the design variables are continuous. In structural optimization, almost all design variables are discrete. A optimization is done to obtain the optimal number of layers, thickness of ply and fiber orientation of each layer. With reference to the middle plane, symmetrical fiber orientations are adopted.

5.1 Objective Function

The objective for the optimum design of the composite drive shaft is the minimization of weight, so the objective function of the problem is given as

Weight of the shaft,

$$m = \rho A L$$

$$m = \rho\pi/4\left(d_o^2 - d_i^2\right)L$$

5.2 Design Variables

The design variables of the problem are

- Number of plies
- Thickness of the ply
- Stacking Sequence

The limiting values of the design variables are given as follows,

1. $n \geq 0$
2. $0.1 \leq t_k \leq 0.5$
3. $-90 \leq \theta_k \leq 90$

Where k =1, 2 … n and n = 1, 2, 3,…32

The number of plies required depends on the design constraints, allowable material properties, thickness of plies and stacking sequence. Based on the investigations it was found that up to 32 numbers of plies are sufficient.

5.3 Design Constraints

1. Torque transmission capacity of the shaft, $T \geq T_{max}$
2. Bucking torque capacity of the shaft, $Tcr \geq T_{max}$
3. Lateral fundamental natural frequency, $N \geq N_{crt}$

VI. RESULTS

A one-piece composite drive shaft for rear wheel drive automobile was designed optimally for E-Glass/ Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy composites with the objective of minimization of weight of the shaft which is subjected to the constraints such as torque transmission, torsional buckling capacities and natural bending frequency.
The Optimization results are shown in Table 4

Table 4: Optimization Results

Parameters (s)	Steel	E-Glass/Epoxy	HS Carbon/Epoxy	HM Carbon/Epoxy
d_o(mm)	90	90	90	90
L (mm)	1250	1250	1250	1250
t_k (mm)	3.318	0.4	0.12	0.12
Optimum No of Layers	1	17	17	17
t (mm)	3.318	6.8	2.04	2.04
Optimum Stacking Sequence	-	[46/-64/-15/-13/39/ -84/-28/20/-$\overline{27}$]$_S$	[-56/-51/74/ -82/67/70/13/-44/-$\overline{75}$]$_S$	[-65/25/68/-63/36/ -40/-39/74/-39]$_S$
Weight (kg)	8.604	4.443	1.1273	1.1274
Weight Saving (%)	-	48.36	86.90	86.90

VII. CONCLUSIONS

1. A method to design composite shaft is presented.
2. Drive shafts of material E-Glass/ Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy have been designed.
3. Optimization of the designed drive shaft is done with the objective of minimization of weight of the shaft which was subjected to the constraints such as torque transmission capacities, torsional buckling capacities and natural bending frequency.
4. The weight savings of the -Glass/ Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy drive shafts were equal to 48.36%, 86.90% and 86.90% of the weight of steel drive shaft respectively.

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