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GENETIC ALGORITHM AS AN OPTIMISATION TOOL FOR AUTOMOTIVE COMPOSITE DRIVE SHAFT DESIGN

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Research Article

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ABSTRACT

Because composite materials have better specific stiffness and strength than traditional metallic constructions, there are several benefits to replacing them. It seems that lengthy, power drive shaft applications are best suited for advanced composite materials. The torque and rotational speed at which they function may be increased by adjusting their elastic characteristics. The purpose of this research is to examine the composite drive shaft's maximum torque, buckling torque transfer, and critical speed. The primary goal of this investigation is to determine if it will be easy to replace the steel drive shaft structure in rear-wheel drive passenger automobiles with composite constructions made of carbon/epoxy and glass/epoxy materials. Using a genetic algorithm as an optimization tool, parameters like ply thickness, number of plies, and stacking sequence are optimized for carbon/epoxy and glass/epoxy shafts in order to determine whether composite structures are suitable for automotive drive shaft applications. The goal is to minimize the weight of the composite shaft, which is subject to constraints like torque transmission, torsional buckling load, and fundamental natural frequency.

INTRODUCTION

The link between the car's back axle and gearbox is called a driveshaft. Because of their high specific strength (strength/density) and high specific modulus (modulus/density), sophisticated composite materials including boron, graphite, carbon, Kevlar, and glass with appropriate resins are commonly employed. For lightweight shafts in drivelines for the automobile industry, polymer matrix composites were suggested.

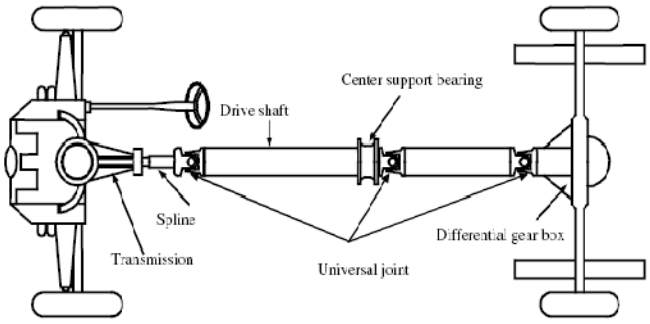


Fig 1: The conventional two-piece steel drive shaft for a rear wheel driving vehicle.

The car's complete driveline is made up of many parts, each of which has a spinning mass. As a general rule, the rotational mass of the drive train loses 17–22% of the engine's power. Because spinning heavier pieces requires more energy, power is wasted. Reducing the amount of rotational mass may help minimize this energy loss. Aluminum and carbon-fiber drive shafts, riffle-drilled axels, lightweight flywheels and transmission gears, and aluminum hubs are a few examples of replacement or modified components used to lower the amount of spinning mass.

To prevent whirling vibration, the drive shaft for passenger automobiles should have a torque capacity

more than 3500 Nm and a fundamental bending natural frequency greater than 9200 rpm [2]. Since the fundamental bending natural frequency of a single steel or aluminum drive shaft is typically less than 5700 rpm when the drive shaft length is approximately 1.5 m [2], the steel drive shaft is typically manufactured in two pieces to increase the fundamental bending natural frequency. This is because the bending natural frequency of a shaft is proportional to the square root of a particular modulus and inversely proportional to the square of beam length. The two-piece steel drive shaft, which adds weight to an automobile and reduces fuel economy, is made up of three universal joints, a bracket, and a central supporting bearing.
The authors used a genetic algorithm (GA) to optimize the design of steel leaf springs in their earlier work [12]. While many researchers have focused on design optimization of steel springs and composite leaf springs, few of these efforts have used genetic algorithms as optimization tools.
The current study attempts to assess the appropriateness of composite materials, such as carbon/epoxy and E-glass/epoxy, for automobile gearbox applications. When designing a one-piece composite drive shaft for a rear-wheel drive car, GA is used to minimize the weight of the shaft while taking into account restrictions like torque transfer, torsional buckling strength capabilities, and natural bending frequency.

PROBLEMS ASSOCIATED WITH THE CONVENTIONAL STEEL DRIVE SHAFT

The torque transmission capability of the drive shaft for passenger cars, small trucks, and vans should be larger than 3500 Nm (T_{max}) and fundamental natural bending frequency of the drive shaft should be higher than 6500 rpm (N_{max}) to avoid whirling vibration. The drive shaft outer diameter *do* should not exceed 100 mm due to space limitations. Here outer diameter of the shaft is taken as 90 mm. For the purpose of experimentation the composite drive shaft of 200 mm length and d_o= 32 mm amd d_i= 22 mm was manufactured. Conventional steel drive shafts ; having less specific modulus and strength; are 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. Therefore the steel drive shaft is made in two sections connected by a support structure, bearings and U-joints and hence over all weight of assembly will be more.
While on the other hand the fundamental natural frequency of the carbon fiber composite drive shaft can be twice as high as that of steel or aluminium because the carbon fiber composite material has more than 4 times the specific stiffness of steel or aluminium, 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. Lower rotating weight transmits more of available power. This eliminates all the assembly, connecting the two piece steel shafts and thus minimizes the overall weight, vibrations and the total cost.
Due to the weight reduction, fuel consumption will be reduced. Composite materials have high damping capacity and hence they produce less vibration and noise with the ability of good corrosion resistance. Composite structures have longer fatigue life than steel or aluminium shaft.

DESIGN OF COMPOSITE DRIVE SHAFT

While designing the composite drive shaft some assumptions are made such as the shaft rotates at a constant speed about its longitudinal axis and has uniform circular cross section. All damping and nonlinear effects are excluded and since lamina is thin and no out-of-plane loads are applied, it is considered as under the plane stress. The stress-strain relationship for composite material is linear & elastic; hence, Hook's law is applicable for composite materials.
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 and the stress distribution in case of solid shaft is zero at the centre and maximum at the outer surface while in hollow

shaft stress variation is smaller. In solid shafts the material close to the centre are not fully utilized.

Table 1: Mechanical properties of E-glass / epoxy and HM carbon / epoxy

Property	Glass / epoxy	Carbon / epoxy
E_{11} (GPa)	50.0	190.0
E_{22} (GPa)	12.0	7.7
G_{12} (GPa)	5.6	4.2
v_{12}	0.3	0.3
$\sigma_1^T = \sigma_1^C$ (MPa)	800.0	870.0
$\sigma_2^T = \sigma_2^C$ (MPa)	40.0	54.0
τ_{12} (MPa)	72.0	30.0
ρ (kg/m^3)	2000.0	1600.0
V_f	0.6	0.6

Table shows the properties of the E-glass / epoxy and high modulus carbon / epoxy materials used for composite drive shafts. E_{11}, E_{22}, G_{12}, σ^T_1, σ^C_1, σ^T_2 and σ^C_2 represent lamina properties in longitudinal and transverse directions (Fig. 2) respectively. v_{12}, τ_{12}, ρ and V_f are the Poisons ratio, shear stress and fiber volume fractions. Since, composites are highly orthotropic and their fractures were not fully studied the factor of safety is taken as 2

Torque transmission of the composite drive shaft

Since the lamina is thin and no out-of-plane loads are applied, it is considered as the plane stress problem and 3-D problem can be reduced into 2-D problem. For unidirectional 2-D lamina, the stress-strain relationship in terms of physical material direction for unidirectional is given by [11],

$$\begin{Bmatrix} \sigma_1 \\ \sigma_2 \\ \tau_{12} \end{Bmatrix} = \begin{bmatrix} Q_{11} & Q_{12} & 0 \\ Q_{12} & Q_{22} & 0 \\ 0 & 0 & Q_{66} \end{bmatrix} \begin{Bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \gamma_{12} \end{Bmatrix},$$

where σ, τ, γ and ε represent stresses and strains in material directions. The matrix Q is referred as the reduced stiffness matrix for the layer and its terms are given by [11]:

$$Q_{11} = \frac{E_{11}}{}, \qquad Q_{12} = \frac{v_{12}E_{22}}{} ;$$

$1-v_{12}v_{21}$ $1-v_{12}v_{21}$

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

STRESS-STRAIN RELATIONSHIP FOR ANGLE-PLY LAMINA

The relation between material co-ordinate system and X-Y-Z co-ordinate system is shown in figure 2 below. Co-ordinate 1-2-3 are principle material directions co-ordinate X-Y-Z are transferred or laminate axis

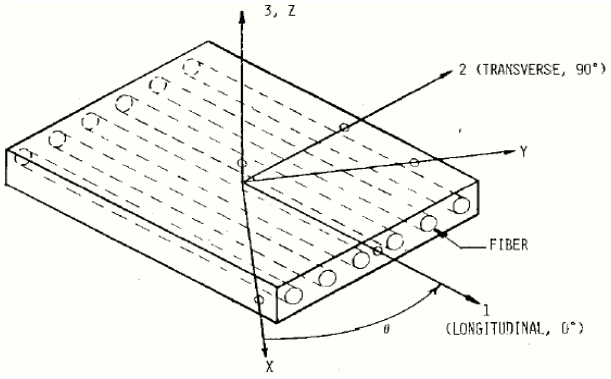


Fig 2: Relation between material coordinate system and X-Y coordinate system

For an angle-ply lamina, where fibbers are oriented at an angle with the positive X-axis (Longitudinal axis of shaft), the effective elastic properties are given by [11],

$$\frac{1}{E_x} = \frac{1}{E_{11}} C^4 + \left[\frac{1}{G_{12}} - \frac{2v}{E_{11}}\right] S^2C^2 + \frac{1}{E_{22}} S^4$$

$$\frac{1}{E_y} = \frac{1}{E_{11}} S^4 + \left[\frac{1}{G_{12}} - \frac{2v}{E_{11}}\right] S^2C^2 + \frac{1}{E_{22}} C^4$$

$$\frac{1}{G} = 2\left[\frac{2}{E_{11}} + \frac{2}{E_{22}} + \frac{2v}{E_{11}} - \frac{1}{G_{12}}\right] S^2C^2 + \frac{1}{G_{12}} [C^2 + S^2]$$

The Stress strain relationship for an angle-ply lamina is given by;

$$\begin{bmatrix} \sigma_x \\ \{\sigma_y\} \\ \tau_{xy} \end{bmatrix} = \begin{bmatrix} \bar{Q}_{11} & \bar{Q}_{12} & \bar{Q}_{16} \\ \bar{Q}_{12} & \bar{Q}_{22} & \bar{Q}_{26} \\ \bar{Q}_{16} & \bar{Q}_{26} & \bar{Q}_{66} \end{bmatrix} \begin{bmatrix} \varepsilon_1 \\ \{\varepsilon_2\} \\ \gamma_{12} \end{bmatrix};$$

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 stiffness. Its terms are individually given by [11]:

$\bar{Q}_{11} = Q_{11}c^4 + Q_{22}s^4 + 2(Q_{12} + 2Q_{66})s^2c^2$;

$\bar{Q}_{12} = (Q_{11} + Q_{22} - 4Q_{66})s^2c^2 + Q_{12}(c^4 + s^4)$;

$\bar{Q}_{16} = (Q_{11} - Q_{12} - 2Q_{66})c^3s - (Q_{22} - Q_{12} - 2Q_{66})cs^3$;

$\bar{Q}_{22} = Q_{11}s^4 + Q_{22}s^4 + 2(Q_{11} + 2Q_{66})s^2c^2$;

$\bar{Q}_{26} = (Q_{11} - Q_{12} - 2Q_{66})cs^3 - (Q_{22} - Q_{12} - 2Q_{66})c^3s$;

$\bar{Q}_{66} = (Q_{11} + Q_{22} - 2Q_{12} - 2Q_{66})s^2c^2 + Q_{66}(s^4 + c^4)$; with C = cosθ and S = sinθ.

TORSIONAL BUCKLING CAPACITY:

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 is expressed below [3]:

$$T_{cr} = 2\pi r^2 t \; 0.272 \; (E_x E_y)^{0.25} \; t^{1.5} \qquad (-)$$

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, the torsional buckling capability of composite shaft is strongly dependent on the thickness of composite shaft and the average modulus in the hoop direction.

Timoshenko beam theory (N_{crt}):

Timoshenko beam theory considers both transerverse shear deformation as well as rotary inertia. Natural frequency f_{nt} based on the Timoshenko beam theory is given by:

$$f_{nts} = K \frac{30}{L^2}\sqrt{\frac{p\pi^2 E x r^2}{2\rho}};$$

$$\frac{1}{K_S^2} = 1 + \frac{p^2\pi^2 r^2}{2L^2}\left[1 + \frac{f_{sEs}}{G_{xy}}\right],$$

where f_{nt} and p are the natural and first natural frequency. K_s is the shear coefficient of the natural frequency (< 1), f_s is a shape factor (equals to 2) for hollow circular cross-sections [7].

Critical speed:

$N_{crt} = 60 f_{nt}$.

DESIGN OPTIMIZATION 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 but due to their high prices a hybrid of layers of carbon-epoxy and E-glass-epoxy could be utilized. 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 angles from the shaft axis, on the other hand, the angle of ±45º orientation realizes the maximum shear strength and 90º is the best for buckling strength [4]. 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 [4].

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 [4]. The variable of the laminate thickness has a big effect on the buckling strength and slight effect on bending natural frequency.

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:

$m = \rho A L$,

Or

$m = \rho \frac{\pi}{4}(d_o^2 - d_i^2)L$,

Design Variables:

The design variables of the problem are

Number of plies [n];

Stacking Sequence [θ_k];

Thickness of the ply [t_k].

The limiting values of the design variables are; 1] $n \geq 0$

2] $-90 \leq \theta_k \leq 90$

3] $0.1 \leq t_k \leq 0.5$

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.

Design Constraints:
Torque transmission capacity of the shaft: $T \geq T_{max}$
Bucking torque capacity of the shaft: $T_{cr} \geq T_{max}$
Lateral fundamental natural frequency: $N_{crt} \geq N_{max}$
The constraint equations may be written as:

$C_1 = (1 - \frac{T}{T_{max}})$, If $T < T_{max}$
= 0 Otherwise;

$C_2 = (1 - \frac{T_{cr}}{T_{max}})$, If $T_{cr} < T_{max}$
= 0 Otherwise;

$C_3 = (1 - \frac{N_{crt}}{N_{max}})$, If $N_{crt} < N_{max}$
= 0 Otherwise

$C = C_1 + C_2 + C_3$.

A new algorithm has been developed using MATLAB R2007b for optimum design of automobile composite drive shaft for following design specification which should be sustained by both steel and composite drive shafts.

Table 2: constraints of design optimization of composite drive shaft

Material	Steel (SM45C)	Glass/Epoxy and Carbon Epoxy
Length	200mm	200 mm
Inner Diameter	22	24 mm
Outer Diameter	32	34 mm
Possible angle combination-(Stacking sequence)	-	-45/0/45/90 (in terms of θ)
Maximum number of plies	1	10
Maximum torque transmission capacity (T_{max})	1350 Nm	1350 Nm
Maximum buckling torque transmission capacity ($T_{cr\ max}$)	1350 Nm	1350 Nm
Critical Speed (N_{crt})	4500 Nm	4500 rpm

RESULTS AND DISCUSSION

Following table 3 & 4 shows the optimized results for Carbon/Epoxy and Glass/Epoxy composite drive shaft;

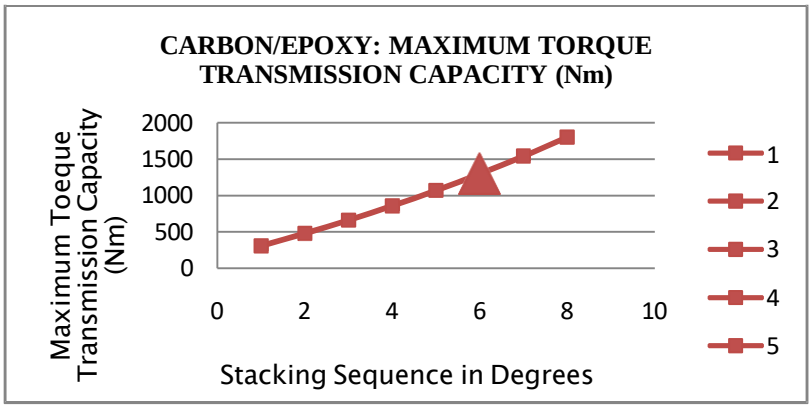
Table 3: Optimization of carbon/epoxy drive shaft

NO. OF LAYERS	**STACKING SEQUENCE** ***(In θ and from Inner layer to outer layer)***	**MAXIMUM TORQUE TRANSMISS-ION CAPACITY (T_{msx})** ***(Nm)***	**BUCKLING TORQUE TRANSMISSI-ON CAPACITY (T_{crt})** ***(Nm)***	**Critical Speed (N_{crt}) (rpm)**	**MASS (m)** ***(KG)***
1	0	340.32	507.55	19187	0.101
2	0/0	528.65	1029.01	27285	0.131
3	0/0/0	730.23	1564.02	27936	0.271
4	90/0/0/0	947.78	2112.25	28534	0.418
5	45/90/0/0/0	1179.58	2317.90	29087	0.573
6	***-45/45/90/0/0/0***	***1429.5472***	***2526.4331***	***29595***	***0.612***
7	0/-45/45/90/0/0/0	1698.96	3093.27	30067	0.894
8	90/0/-45/45/90/0/0/0	1985.24	3672.69	30495	1.082

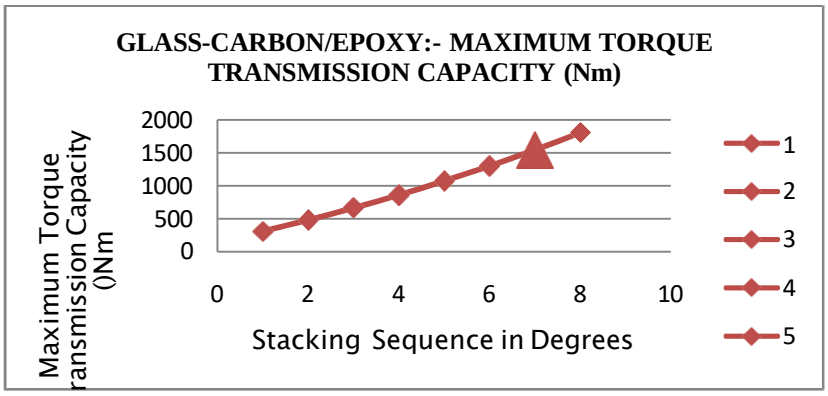
Table 4: Optimization of glass-carbon/epoxy drive shaft

NO. OF LAYERS	STACKING SEQUENCE *(In θ and from Inner layer to outer layer)*	MAXIMUM TORQUE TRANSMISS-ION CAPACITY (T_{msx}) *(Nm)*	BUCKLING TORQUE TRANSMISSI-ON CAPACITY (T_{crt}) *(Nm)*	Critical Speed (N_{crt}) (rpm)	MASS (m) *(KG)*
1	0	309.52	463.8619	8747	0.105
2	0/90	480.49	1244.0453	8486	0.131
3	0/90/90	663.29	2044.4965	8887	0.271
4	0/90/90/90	860.13	2864.7149	9013	0.418
5	-45/0/90/90/90	1072.09	3085.8766	10376	0.573
6	45/-45/0/90/90/90	1299.57	3309.3942	8281	0.635
7	***90/-45/45/0/90/90/90***	***1543.5649***	***4145.5415***	***9325***	***0.829***
8	0/90/-45/45/0/90/90/90	1804.76	4705.2754	9410	1.082

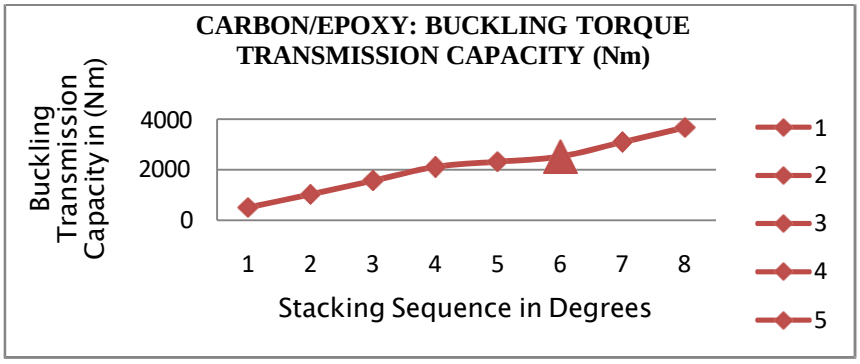
Graph: Maximum torque transmission capacity of Carbon/Epoxy Shaft



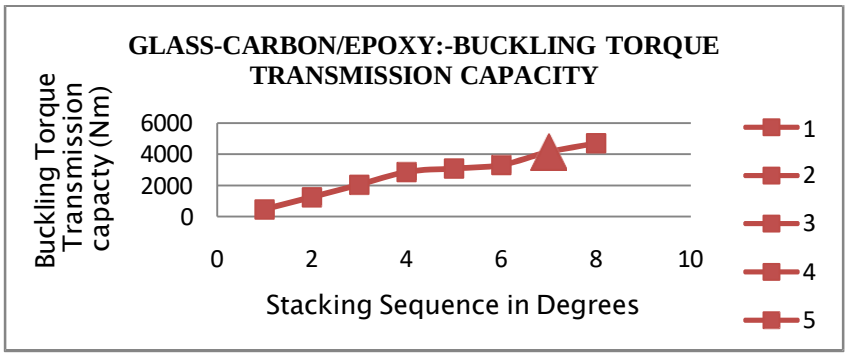
Graph: Maximum torque transmission capacity of Glass-Carbon/Epoxy Shaft



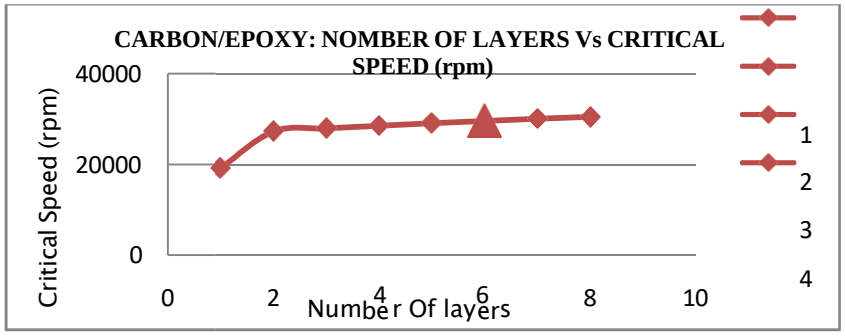
Graph: Buckling torque transmission capacity of Carbon/Epoxy Shaft



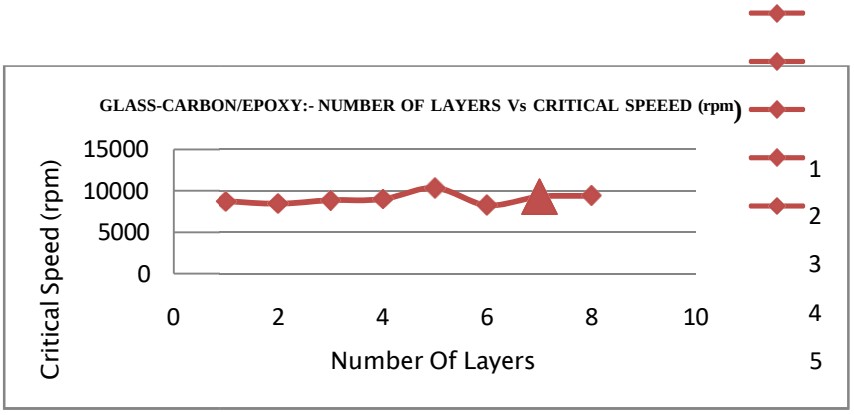
Graph: Buckling torque transmission capacity of Glass-Carbon/Epoxy Shaft



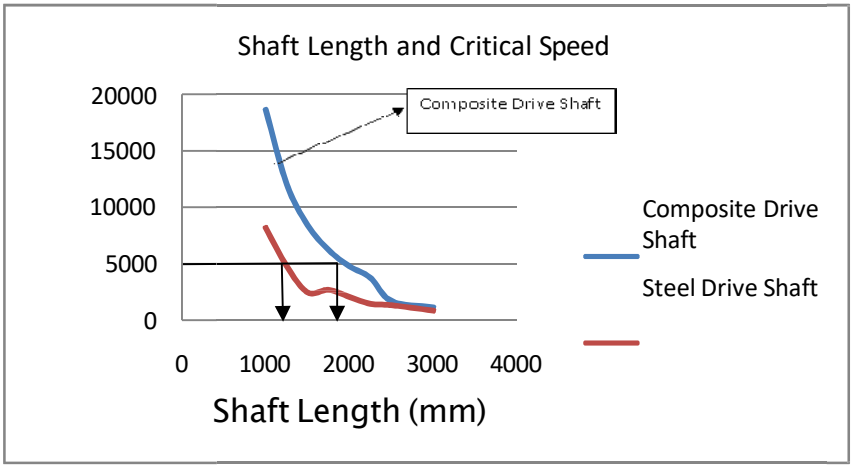
Graph: Carbon/Epoxy: Number of layers Vs Critical speed



Graph: Glass-Carbon/Epoxy: Number of layers Vs Critical speed



Graph: Critical speed analysis in composite drive shafts



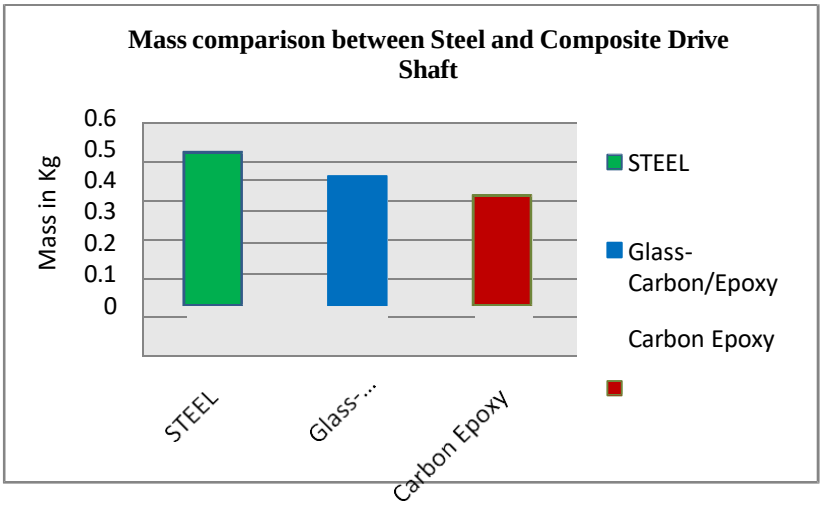
Graph shows that for the steel drive shaft having about 6000 revolutions per minute can be manufactured of length about 1m to 1.5m while on the other hand for composite drive shaft it is possible to manufacture a shaft of length 1.5m to 2 m for same revolutions.

Mass Comparison between Steel and Composite drive Shafts

Following table gives the comparison of masses of conventional steel drive shaft, Carbon/Epoxy and Glass-Carbon/Epoxy composite drive shaft.

Material	**Weight (Kg)**	**Weight Reduction (%)**
Steel	0.489	-
Glass-Carbon/Epoxy	0.412	15.75
Carbon Epoxy	0.352	28.01

Graph: Mass Comparison of three shafts



CONCLUSION

The following conclusions are drawn from initial tests and research on physical characteristics such as weight, material combination, torque transmitting capacity, etc.

• A one-piece composite drive shaft made of Carbon/Epoxy and Glass-Carbon/Epoxy is designed optimally with Genetic Algorithm as optimization tool with the objective of minimization of weight of drive shaft which is subjected to constraints such as Maximum torque transmission capacity, Buckling torque transmission capacity and critical speed.

• About 28.01 % of weight saving is achieved with Carbon/Epoxy shaft with increase in critical speed

enabling manufacturing of shaft of length 1.8m to 2 m; as compared to steel shaft; by experimentation.
• About 15.75% weight saving is achieved with Glass-Carbon/Epoxy composite shaft with increase in critical speed enabling manufacturing of shaft of length 1.7 m to 2m; as compared to steel shaft; by experimentation.
• The findings show that the orientation of the fibers has a significant favorable impact on the dynamic properties of the composite material shafts.
• A genetic algorithm is recommended as a useful optimization method.

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