Analytical and Modeling Study of Composite Material Drive Shaft for an Automobile

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Abstract*: The advanced composite materials such as Graphite, Carbon, Kevlar and Glass with suitable resins are widely used because of their high specific strength (strength/density) and high specific modulus (modulus/density). The weight reduction of the drive shaft can have a certain role in the general weight reduction of the vehicle and is a highly desirable goal, if it can be achieved without increase in cost and decrease in quality and reliability. It is possible to achieve design of composite drive shaft with less weight to increase the natural frequency of the shaft, decrease the bending stresses, and maximize the torque transmission and torsional buckling capabilities. Substituting composite material structure for conventional metallic structure has many advantages because of high specific strength and stiffness of composite material. This work deals with the replacement of conventional two-piece drive shaft with a single piece E-glass/Epoxy, high strength carbon/ epoxy, high modulus carbon/ epoxy composite drive shaft for an automotive application. The design parameters were optimized with the objective of minimizing the weight of composite drive shaft. The design optimization also showed significant potential improvement in the performance of drive shaft.*
Key words*: Bernoulli Euler theory, Static Analysis, Modal analysis, Buckling analysis, ANSYS*

I.   Introduction

The advanced composite materials such as Graphite, Carbon, Kevlar and Glass with suitable resins are widely used because of their high specific strength (strength/density) and high specific modulus (modulus/density). Advanced composite materials seem ideally suited for long, power driver shaft (propeller shaft) applications. Their elastic properties can be tailored to increase the torque they can carry as well as the rotational speed at which they operate. The drive shafts are used in automotive, aircraft and aerospace applications. 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.

Deborah D.L. Chung (2015) gives carbon fiber composites in depth review. Carbon fibers refer to fibers which are at least 92 wt. % carbons in composition. They can be short or continuous; their structure can be crystalline, amorphous, or partly crystalline.

G.J. Withers et al. (2015) modified surface nanoclay reinforced epoxy glass-fiber composite is evaluated for properties of mechanical strength, stiffness, ductility and fatigue life, and compared with the pristine or epoxy glass-fiber composite material not reinforced with nanoclays

Daniel Lawrence et al. (2014) examine the result of fiber orientation angles and stacking sequence on the torsional stiffness, natural frequency and buckling strength of composite drive shaft. The weight reduction of the drive shaft can have a certain role in the general weight reduction of the vehicle and is a highly desirable goal

Ravi Arun (2014) worked for weight reduction of the drive shaft which have a certain role in the general weight reduction of the vehicle and is a highly desirable goal, if it can be achieved without increase in cost and decrease in quality and reliability. It is possible to achieve design of composite drive shaft with less weight to increase the first natural frequency of the shaft and to decrease the bending stresses using various stacking sequences.

S V Gopals Krishna et al. (2013) examine that almost all automobiles (at least those which correspond to design with rear wheel drive and front engine installation) have transmission shafts. The weight reduction of the drive shaft can have a certain role in the general weight reduction of the vehicle and is a highly desirable goal. Substituting composite structures for conventional metallic structures has many advantages because of higher specific stiffness and strength of composite materials.

1.1 Purpose of the Drive Shaft
The torque that is produced from the engine and transmission must be transferred to the rear wheels to push the vehicle forward and reverse. The drive shaft must provide a smooth, uninterrupted flow of power to the axles. The drive shaft and differential are used to transfer this torque.

1.2 Functions of the Drive Shaft
1. First, it must transmit torque from the transmission to the differential gear box.
2. During the operation, it is necessary to transmit maximum low-gear torque developed by the engine.
3. The drive shafts must also be capable of rotating at the very fast speeds required by the vehicle.
4. The drive shaft must also operate through constantly changing angles between the transmission, the differential and the axles. As the rear wheels roll over bumps in the road, the differential and axles move up and down. This movement changes the angle between the transmission and the differential.
5. The length of the drive shaft must also be capable of changing while transmitting torque. Length changes are caused by axle movement due to torque reaction, road deflections, braking loads and so on. A slip joint is used to compensate for this motion. The slip joint is usually made of an internal and external spline. It is located on the front end of the drive shaft and is connected to the transmission.

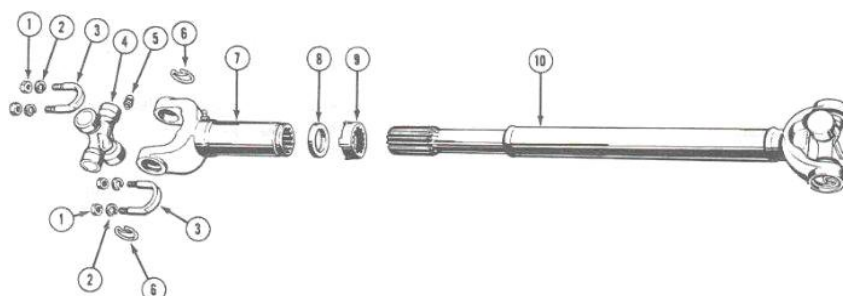### 1.3 Drive Shaft Arrangement in a Car Model
Conventional two-piece drive shaft arrangement for rear wheel vehicle driving system is shown in fig. 1 below.
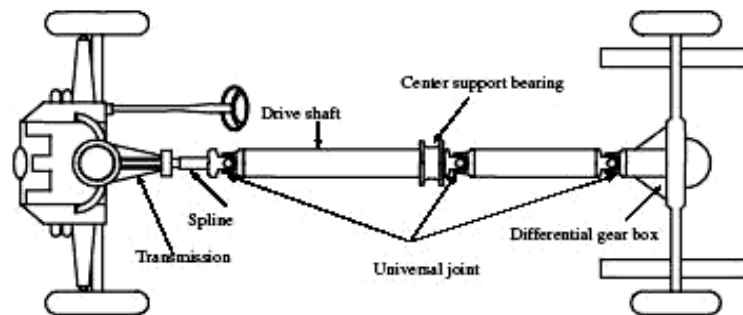
Fig.1 Conventional two-piece drive shaft arrangements for rear wheel vehicle driving system

1.4 Parts of Drive Shaft and Universal Joint
Parts of drive shaft and universal joint are shown in fig. 2 below;

Fig. 2 Parts of drive shaft and universal joint.

1.U-bolt nut	2. U-bolt washers	3. U-bolt
4.Universal joint journal	5.Lubrication fitting	6. Snap ring
7.Universal joint sleeve yoke	8. Spline seal	9. Dust cap
10.Drive shaft tube		

1.5 Material of Conventional Steel Drive Shaft
Steel (SM45C) used for automotive drive shaft applications. The material properties of the steel (SM45C) are given in Table 1

Table No. 1 Crankshaft Material Properties

Sr. No.	Mechanical properties	Symbol	Units	Steel
1	Young's Modulus	E	GPa	207.0
2	Poisson's ratio	υ	-----	0.3
3	Density	ρ	Kg/m^3	7600
4	Yield Strength	S_y	MPa	370
5	Shear Strength	S_s	MPa	--

1.6 Demerits of a Conventional Steel Drive Shaft

1. They have less specific modulus and strength.
2. Increased weight.
3. Conventional steel drive shafts 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.
4. Its corrosion resistance is less as compared with composite materials.
5. Steel drive shafts have less damping capacity.

1.7 Material for Composite Material Drive Shaft

The materials selected for composite drive shafts are E-Glass/Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy their properties shown in the Table 2

Table No. 2 Properties of E-Glass/Epoxy, High Strength Carbon/Epoxy and High Modulus carbon /Epoxy Material.

Sr. No.	Property	Units	E-glass / epoxy	HS carbon/Epoxy	HM carbon / epoxy
1	E_{11}	GPa	50.0	134.0	190.0
2	E_{22}	GPa	12.0	7.0	7.7
3	G_{12}	GPa	5.6	5.8	4.2
4	v_{12}	_	0.3	0.3	0.3
5	$S_1^t = S_1^c$	MPa	800.0	880.0	870.0
6	$S_2^t = S_2^c$	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

1.8 Merits of Composite Drive Shaft

1. They have high specific modulus and strength.
2. Reduced weight.
3. The fundamental natural frequency of the carbon fibre composite drive shaft can be twice as high as that of steel or aluminium because the carbon fibre 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. This eliminates all the assembly, connecting the two piece steel shafts and thus minimizes the overall weight, vibrations and the total cost.
4. Due to the weight reduction, fuel consumption will be reduced.
5. They have high damping capacity hence they produce less vibration and noise.
6. They have good corrosion resistance.
7. Greater torque capacity than steel or aluminium shaft.
8. Longer fatigue life than steel or aluminium shaft.
9. Lower rotating weight transmits more of available power.

II. Analytical Study

A one-piece composite drive shaft for rear wheel drive automobile was designed optimally by using analytical method 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.

2.1 Specification of the Problem
The torque transmission capability of the drive shaft for passenger cars, small trucks, and vans should be larger than 3500 Nm (Tmax) and fundamental natural bending frequency of the drive shaft should be higher than 6500 rpm (Nmax) 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. The drive shaft of transmission system was designed optimally to the specified design requirements as shown in following Table No. 3

Table No. 3 Design requirements and specification

Sr. No.	Name	Notation	Unit	Value
1	Ultimate Torque	Tmax	Nm	3500
2	Max. Speed of shaft	Nmax	rpm	6500
3	Length of shaft	L	mm	1250

2.2 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 centre coincides with the geometric centre.
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.

2.3 How to Derive Essential parameters for Drive Shaft
To design a drive shaft one must know about how much of torque it has to be transfer, torsional buckling capacity, length of drive shaft, diameter of drive shaft, bending natural frequency and mass. We can easily compute the required data by knowing some essential parameters as follows;

✓ **Torque Transmission Capacity**

$$T = \frac{S_S \pi \left(d_o^4 - d_i^4\right)}{16 \times d_o \times F.S}$$

Where,

Ss = Shear strength
do= Outer diameter of shaft
di = Inner diameter of shaft
$F.S$ = Factor of safety

✓ **Torsional Buckling Capacity**

$$T_b = 2\pi r^2 t(0.272).(E_x E_y^3)^{\frac{1}{4}}.\left(\frac{t}{r}\right)^{\frac{3}{2}}$$

Where,

r = mean radius of shaft
t = Thickness of shaft
Ex= Elastic modulus in axial direction
Ey= Elastic modulus in tangential direction

✓ **Bending Natural Frequency**

$$f_{nb} = \frac{\pi}{2L^2} \times \sqrt{\frac{E_x I_x}{m^{'}}}$$

Where,

$$I_x = \frac{\pi}{64} \times \left(d_o^4 - d_i^4\right)$$

$$m' = \rho\left(\frac{\pi}{4}\right) \times \left(d_o^2 - d_i^2\right)$$

Where,
fnb = Natural frequency
L = Length of shaft
Ix= Moment of inertia of cross section of shaft
m'= Mass of shaft per unit length

✓ **Mass**

$$m = m' \times L$$

✓ **Percentage of Reduction in Mass**

$$= \left(1 - \frac{m_{new}}{m_{old}}\right) \times 100$$

✓ **Design Constraints**
- Torque transmission capacity of the shaft T ≥T$_{max}$
- Buckling torque capacity of the shaft T$_{cr}$ ≥T$_{max}$
- Lateral fundamental natural frequency N ≥N$_{crt}$

2.3 Calculations for Steel drive shaft
a) **Torque transmission capacity of Steel Drive Shaft**

$$T = \frac{S_S \pi \left(d_o^4 - d_i^4\right)}{16 \times d_o \times F.S}$$

$\left(d_0^4 - d_i^4\right)$ = (T x 16 x do x F.S.)/Ss X π

= (3500 x 16 x 0.09 x4)/ (370x 10^6 x π)

$\left(d_0^4 - d_i^4\right)$ = 1.734 x 10^{-5}

$d_i^4 = d_o^4$ -1.734 x 10^{-5}

d_i = 0.0833510m

d_i = 83.35

r_i = 41.675mm

Hence, thickness of shaft

$t = r_0 - r_i$

t = 45 - 41.6755 ($r = \dfrac{r_i + r_o}{2}$; r = 43.337mm)

t = 3.324mm
t = 3.324 x 10^{-3} m.

b) **Torsional Buckling Capacity of Steel Drive Shaft (it must satisfy the following condition)**

$$\frac{1}{\sqrt{1-v^2}}\frac{L^2 t}{(2r)^3} > 5.5,$$

It is for long shaft only.
As it satisfy the condition the next step is to calculate the torsional buckling Capacity
So,

[1/√(1-0.3^2)] x [(1250 x 10^{-3})^2 x 3.324 x10^{-3}]/ 2x 43.3377 > 5.5
= 8.3612 > 5.5

Condition satisfied.
Now, Torsional   Buckling Capacity

$T_{cr} = \tau_{cr} 2\pi r^2 t$ and

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

$$\tau_{cr} = 1.112361 \text{ x } 10^9 \text{ N/m}^2$$

and

$$T_b = \tau_{cr} (2\pi r^2 t)$$
$$T_b = 3733.44 \text{ Nm}$$

Thus $T_b > T$

c) Bending Natural Frequency of Steel Drive Shaft
According to Bernoulli- Euler beam theory
The Bending natural Frequency of rotating shaft is given by.

$$f_{nbe} = \frac{\pi p^2}{2L^2}\sqrt{EI_x/m_1}$$

Where,
p – No of frequency (1st, 2nd, 3rd ….)

$I_x = \pi/64 \ (do^4 - di^4)$

&

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

Hence,

$I_x = 8.5147 \text{ x } 10^{-7} \text{ m}^4$

And

$m' = 6.881 \text{kg/m}$

$f_{nb} = 160.89 \text{ Hz}$

d) Mass of Steel Drive Shaft

$M = m' \text{ x } L$
$= 6.881 \text{ x } 1.25$
$= 8.601 \text{ kg.}$

2.4 Calculations for E-Glass / Epoxy Drive Shaft
a) Torque Transmission capacity of E-Glass/Epoxy Drive Shaft

$$T = \frac{S_S \pi \left(d_o^4 - d_i^4\right)}{16 \times d_o \times F.S}$$

$\left(d_o^4 - d_i^4\right)$ = (T x 16 x d_o x F.S.) / S$_s$ x π

Hence d_i = 0.0764467 m

d_i = 76.44 mm

& r_i =38.22 mm

$t = r_o - r_i$

Hence t = 6.77 mm

& $r = \frac{r_i + r_0}{2}$

r = 41.61mm

b) Torsional Buckling Capacity of E-Glass/Epoxy Drive Shaft
Buckling Torque is given by,

$$T_{cr} = (2\pi r^2 t)(0.272)(E_x E_y^3)^{0.25}(t/r)^{1.5}$$
$$T_b = 3893.86 \text{ Nm}$$

As,

$T_b > $ T condition satisfied

c) Bending Natural Frequency of E-Glass/Epoxy Drive Shaft

$$f_{nbe} = \frac{\pi p^2}{2L^2}\sqrt{EI_x / m_1}$$

$$I_x = 1.5447 \text{ x } 10^{-6}$$
$$\text{\& m}^{'} = 3.545 \text{ kg/m}$$

Now $f_{nb} = (\pi \text{ x } 1^2/ 2 \text{ x } L^2)\ \sqrt{}\ (E_x I_x/m^{'})$

Hence $f_{nb} = 112.98$ Hz

d) Mass of E-Glass/Epoxy Drive Shaft

$$m = m^{'} \text{ x } L$$
$$= 4.261 \text{ kg}$$

2.5   Calculations for High Strength Carbon / Epoxy Drive Shaft

a) Torque transmission Capacity of High Strength carbon/Epoxy Drive Shaft

$$T = \frac{S_S \pi \left(d_o^4 - d_i^4\right)}{16 \times d_o \times F.S}$$

$$d_i = 0.0859 \text{ m}$$

$$d_i = 85.9 \text{ mm}$$

$$r_i = 42.95 \text{ mm}$$

$$t = 2.04 \text{ mm}$$

$$\& \qquad r = \frac{r_i + r_o}{2}$$

Hence $r = 43.97$ mm

b) Torsional Buckling Capacity of High Strength carbon/Epoxy Drive Shaft

$$T_{cr} = (2\pi\text{r}^2\text{t})(0.272)(\text{E}_x\text{E}_y^3)^{0.25}(t/r)^{1.5}$$

$$T_b = 4005.70 \text{ Nm}$$

As

$$T_b > \text{T condition satisfied}$$

c) Bending Natural Frequency of High Strength carbon/Epoxy Drive Shaft

$$f_{nbe} = \frac{\pi p^2}{2L^2}\sqrt{EI_x / m_1}$$

p – No of frequency (1st, 2nd, 3rd ….)

$$I_x = 5.4797 \text{ x } 10^{-7} \text{ m}^4$$
$$m^{'} = 0.90627 \text{ kg/m}$$
$$f_{nb} = (\pi \text{ x } 1^2/ 2 \text{ x } L^2)\ \sqrt{}\ (E_x I_x/m^{'})$$
$$f_{nbe} = 127.71 \text{ Hz}$$

Now,

d) Mass of High Strength carbon/Epoxy Drive Shaft

$$m = m^{'} \text{ x } L$$
$$= 1.0568 \text{ kg}$$

2.6 Calculations for High Modulus carbon / Epoxy Drive Shaft

a) Torque transmission capacity of High Modulus Carbon/Epoxy Drive shaft

$$T = \frac{S_S \pi \left(d_o^4 - d_i^4\right)}{16 \times d_o \times F.S.}$$

$$d_i = 0.0859 \text{ m}$$

$$d_i = 85.9 \text{ mm}$$

$$r_i = 42.95 \text{ mm}$$

$$t = 2.04 \text{ mm}$$

$$\& \quad r = \frac{r_i + r_o}{2}$$

Hence
$$r = 43.975 \text{ mm}$$

b) Torsional Buckling Capacity of High Modulus Carbon/Epoxy Drive shaft

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

$$T_b = 4365.54 \text{ Nm}$$

As

$$T_b > T \text{ condition satisfied}$$

c) Bending Natural Frequency of High Modulus Carbon/Epoxy Drive shaft

$$f_{nbe} = \frac{\pi p^2}{2L^2}\sqrt{EI_x / m_1}$$

p – No of frequency (1st, 2nd, 3rd ….)

$$I_x = 5.4797 \times 10^{-7} \text{ m}^4$$
$$m^{'} = 0.90627 \text{ kg/m}$$

$$f_{nb} = (\pi \times 1^2 / 2 \times L^2) \sqrt{(E_x I_x / m^{'})}$$

$$f_{nbe} = 157.68 \text{ Hz}$$

Now,

**d) Mass of High Modulus Carbon/Epoxy Drive shaft**

$$m = m^{'} \times L$$
$$= 1.0892 \text{ kg}$$

III. Design Analysis

The above result obtain from analytical calculation is to be compare with results obtained by design analysis in ANSYS, for confirmation. There is a small variation which is to be considered.

3.1 Comparison of Finite Element Analysis and Analytical Study

The general procedure of finite elements analysis is as shown in fig.3. It starts with modelling and meshing as shown in fig.4 & fig.5 respectively to specify geometric condition. As most of the commonly used element geometries have straight sides, it is generally impossible to include the entire physical domain in the element mesh if the domain includes curved boundaries. Where a curved-boundary domain is meshed (quite coarsely) using square elements. A refined mesh for the same domain is using smaller, more numerous elements of the same type. Note that the refined mesh includes significantly more of the physical domain in the finite element representation and the curved boundaries are more closely approximated. (Triangular elements could approximate the boundaries even better.) Suppose solid cylinder fixed at one end and subjected to a tensile load at the other end. Assuming the displacement at the point of load application to be of interest, a first approximation is obtained by considering the cylinder to be uniform, having a cross-sectional area equal to the average area
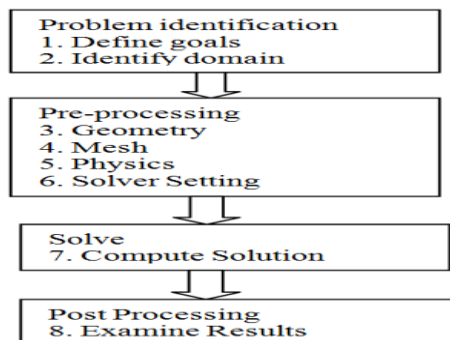
Fig. 3 FEA Steps

3.2 Modeling And Meshing of Composite Drive Shaft
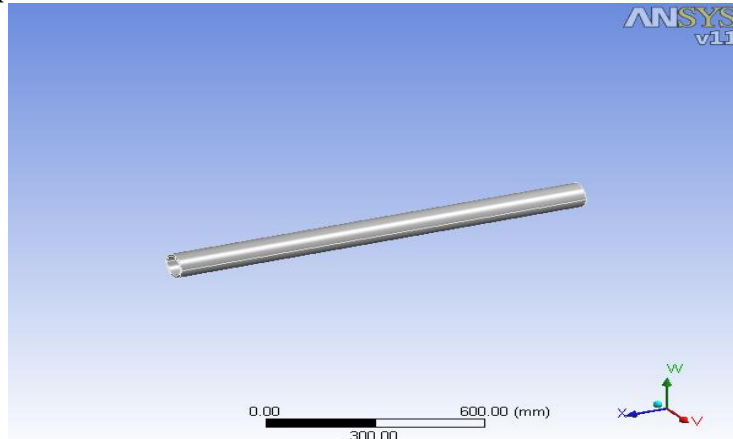**a) Model for Composite Drive Shaft**

Figure 4 Models for Composite Drive Shaft

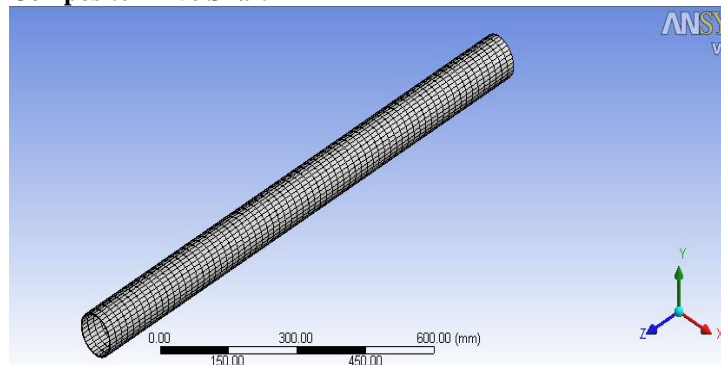**b) Meshing for Composite Drive Shaft**

Figure 5 Meshed E-Glass Epoxy Drive Shaft

3.3 Analysis

Static analysis as shown in Fig. 6 deals with the conditions of equilibrium of the bodies acted upon by forces. A static analysis can be either linear or non-linear. All types of non-linearities are allowed such as large deformations, plasticity, creep, stress stiffening, contact elements etc. this chapter focuses on static analysis. A static analysis calculates the effects of steady loading conditions on a structure, while ignoring inertia and damping effects, such as those carried by time varying loads. A static analysis is used to determine the displacements, stresses, strains and forces in structures or components caused by loads that do not induce significant inertia and damping effects. A static analysis can however include steady inertia loads such as gravity, spinning and time varying loads.
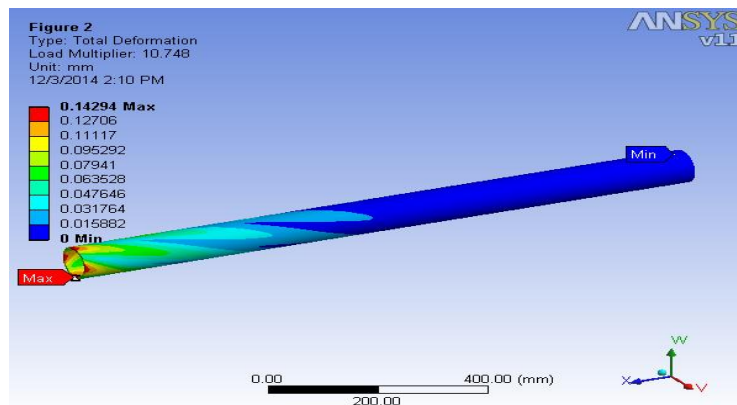

Figure 6 Static Analysis for Composite Drive Shaft

Modal analysis as shown in fig. 7 is used to determine the vibration characteristics such as natural frequencies and mode shapes of a structure or a machine component while it is being designed. It can also be a starting point for another more detailed analysis such as a transient dynamic analysis, a harmonic response analysis or a spectrum analysis. Modal analysis is used to determine the natural frequencies and mode shapes of a structure or a machine component. The rotational speed is limited by lateral stability considerations. Most designs are sub critical, i.e. rotational speed must be lower than the first natural bending frequency of the shaft. The natural frequency depends on the diameter of the shaft, thickness of the hollow shaft, specific stiffness and the length
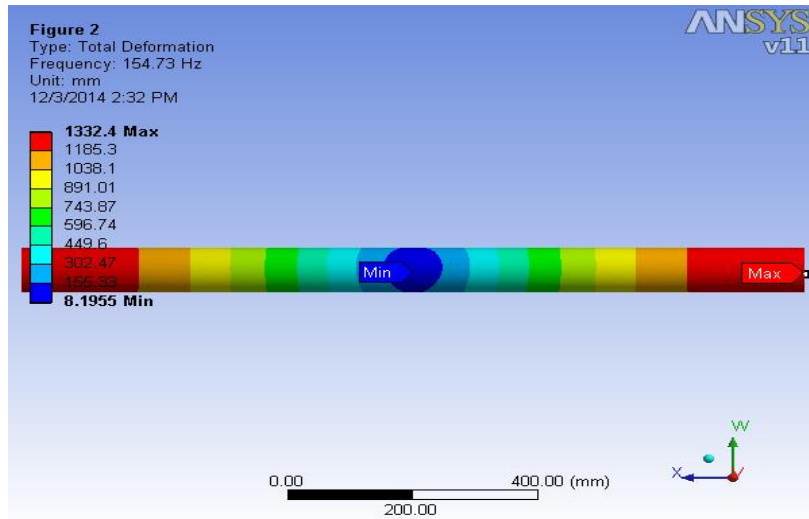

Figure 7 Modal Analyses for Composite Drive Shaft

Buckling analysis as shown in fig. 8 is a technique used to determine buckling loads (critical loads) at which a structure becomes unstable, and buckled mode shapes (The characteristic shape associated with a structure's buckled response). For thin walled shafts, the failure mode under an applied torque is torsional buckling rather than material failure. For a realistic driveshaft system, improved lateral stability characteristics must be achieved together with improved torque carrying capabilities. The dominant failure mode, torsional buckling, is strongly dependent on fibre orientation angles and ply stacking sequence.
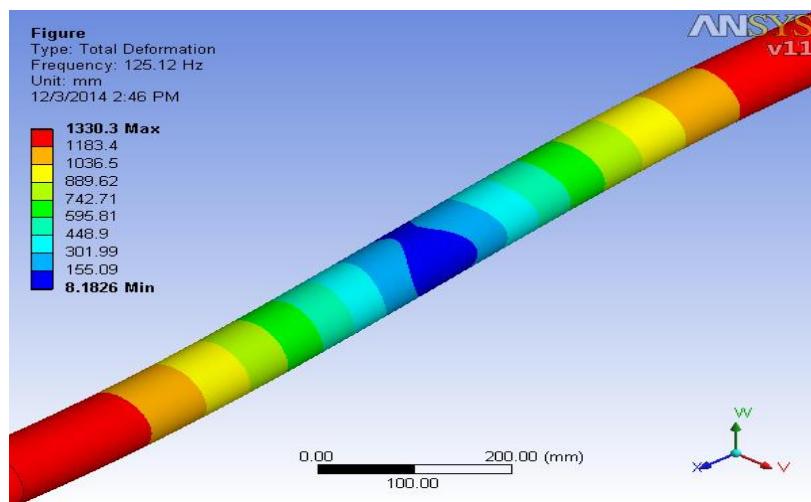

Figure 8 Buckling Analysis for Composite Drive Shaft

IV. Results And Discussion

A comparative study of three different materials was conducted to choose the best "suited materials. Steel (SM45 Steel) was chosen for reference and the rest of three composites were analysed at ± 45 degrees ply orientation. The material properties of all materials considered from design considerations. The analysis carried out by using ANSYS 11.0
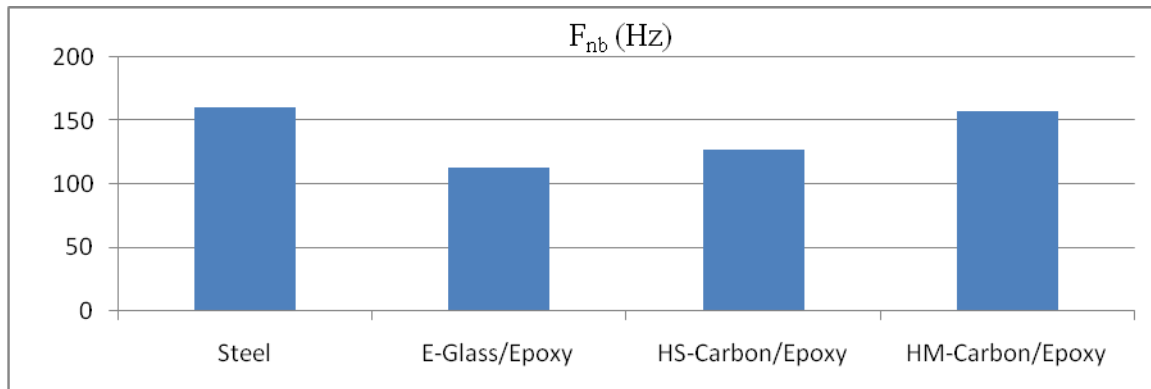
4.1 Results obtained by Analytical Study

A shaft has to be design to meet the stringent design requirements for automobiles
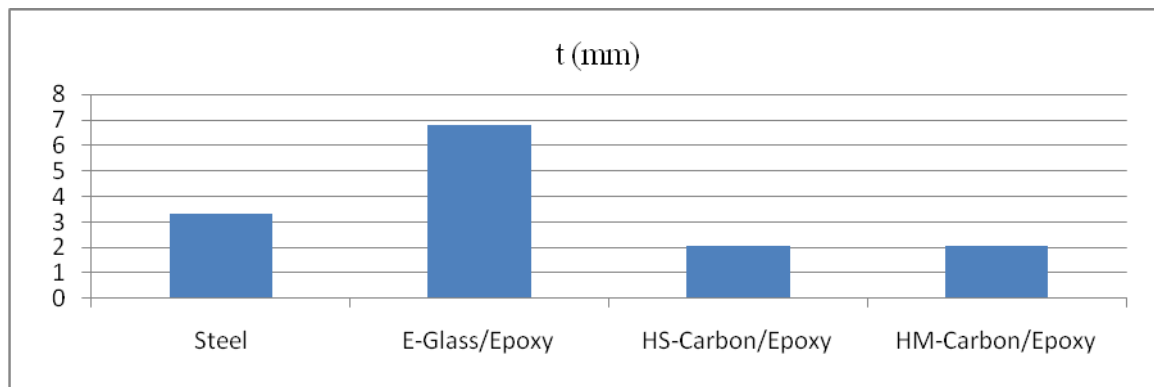
Table No.4 Results obtained by Analytical Study

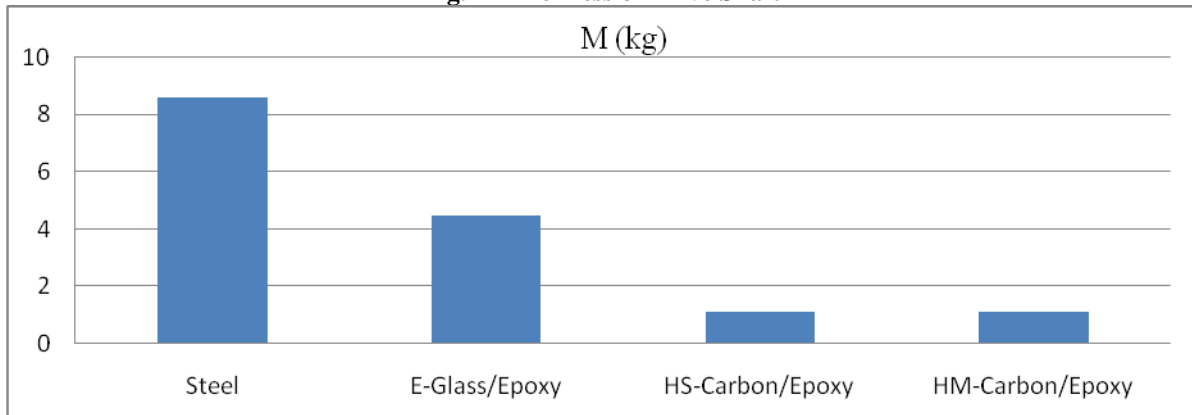| Sr. No.	Material	Steel	E-Glass/ Epoxy	HS Carbon/ Epoxy	HM Carbon/ Epoxy
1	T (Nm)	3733.44	3893.86	4005.70	4365.54
2	Fnb(Hz)	160.89	112.98	127.71	157.68
3	t (mm)	3.318	6.80	2.04	2.04
4	Mass (kg)	8.604	4.443	1.1328	1.1328
5	% Saving	-	48.36 %	86.9 %	86.89 %
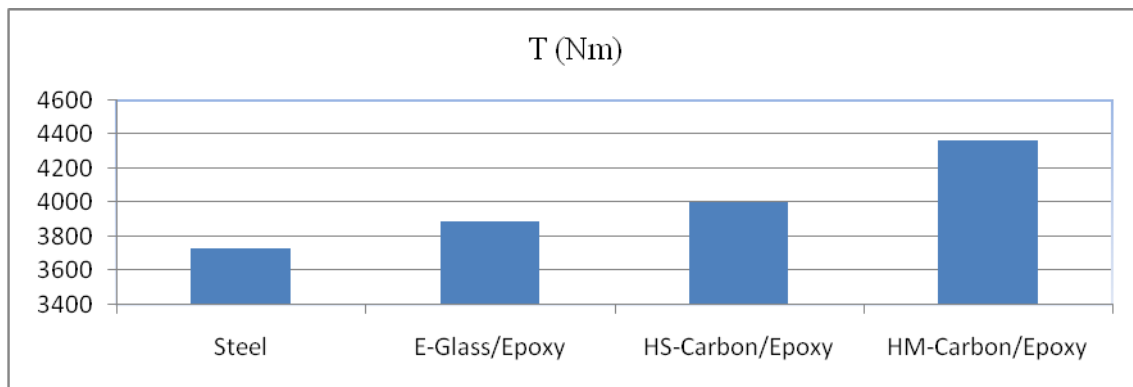
Fig.9 Torsional Buckling Capacity of Drive Shaft

Fig.10 Bending Natural Frequency of Drive Shaft

Fig.11 Thickness of Drive Shaft

**Fig.12 Mass of the Drive Shaft**

4.2 Results comparison obtained by Analytical and Modeling study for drive Shaft
        A comparison between the results obtained based on theoretical calculations and the results obtained from the ANSYS 11.0.work bench has been carried out. For this purpose, the drive shaft of the automobile has been idealized as a hollow cylindrical shaft. It is then subjected to same load theoretically and in finite element solver. The comparisons of results shows a very close range (Table No. 5) conformance, which has been plotted

Table No.5 Results comparison obtained by Analytical and Modeling study for drive Shaft

Material	Analytical Study			Modeling Study		
	T(Nm)	f_n(hz)	M(Kg)	T(Nm)	f_n(hz)	M(Kg)
Steel	3733.44	160.89	8.604	3815.94	158.25	8.504
E-Glass/Epoxy	3893.86	112.98	4.261	4087.77	111.03	4.511
HS-Carbon/Epoxy	4005.70	127.71	1.0868	4189.12	125.12	1.1283
HM-Carbon/Epoxy	4365.54	157.68	1.0561	4450.38	154.72	1.1272

Nature of graph (fig. 13, 14, 15) for above comparison is as shown below:
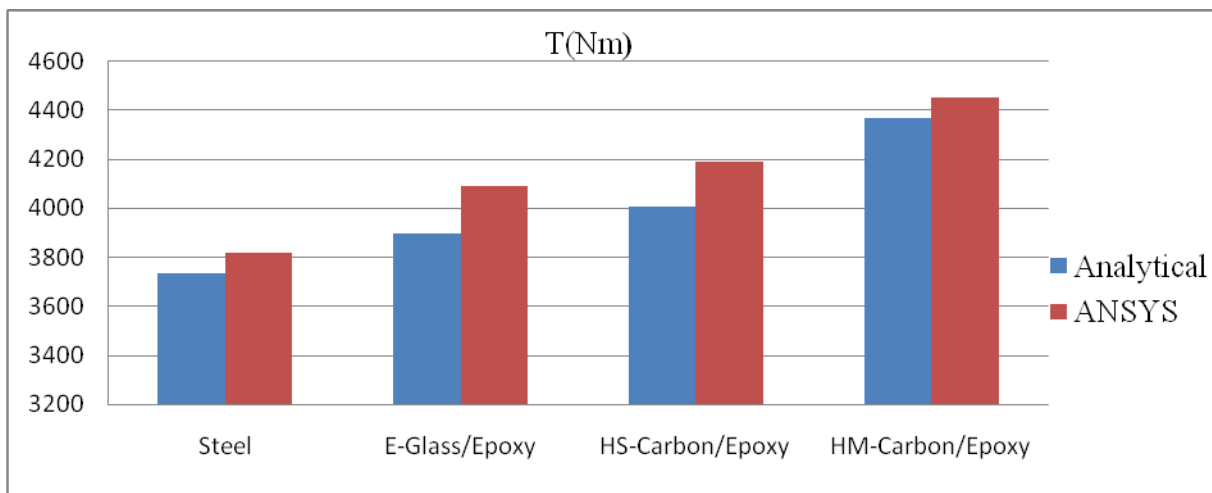
Fig.13 Comparative results of Analytical study and Modeling study for Torsional Buckling Capacity Drive Shaft
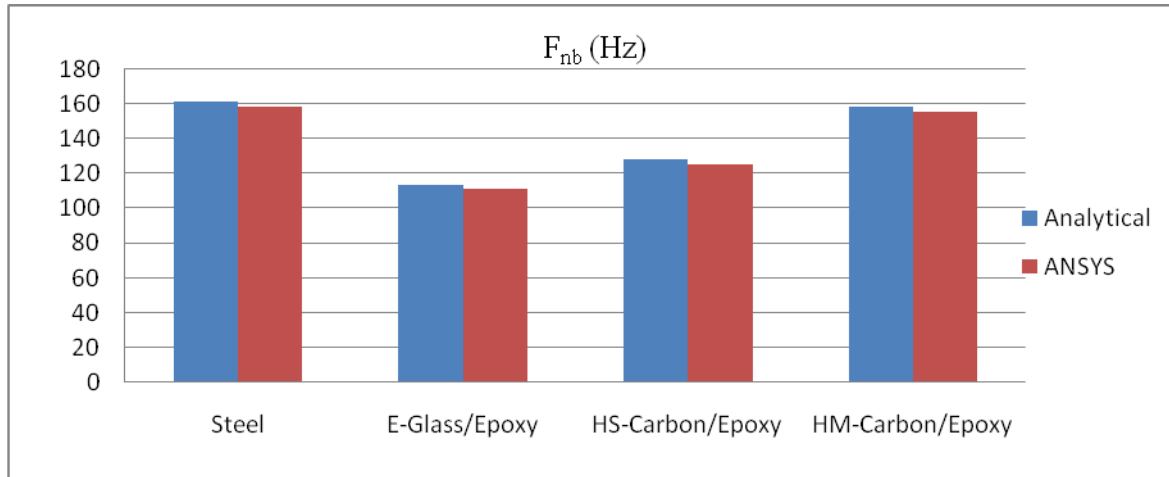
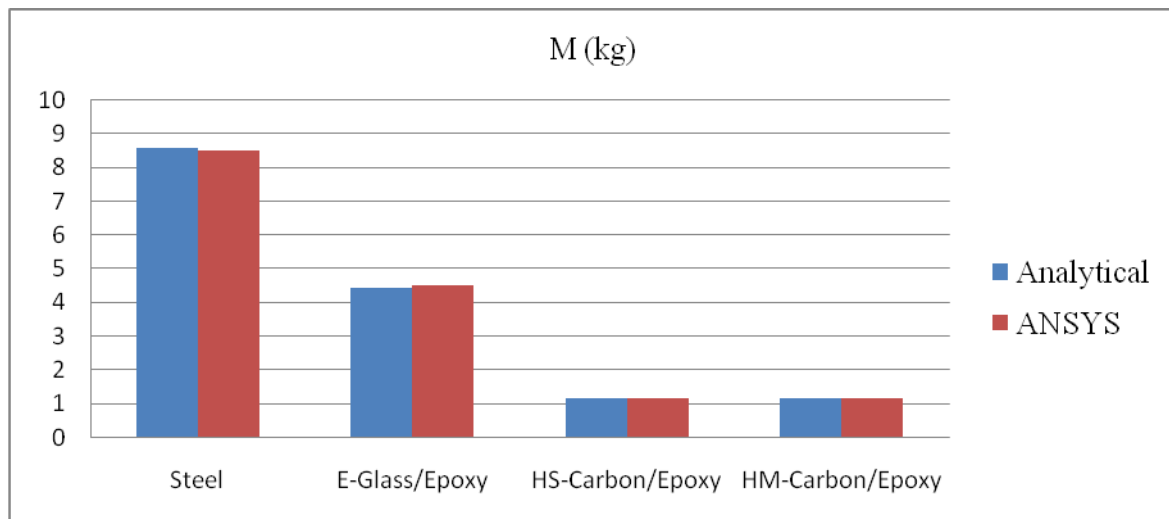**Fig.14 Comparative results of Analytical study and Modeling study for Bending Natural Frequency of Drive Shaft**

Fig.15 Comparative results of Analytical study and Modeling study for Mass of Drive Shaft

4.3 Stress and Strain distribution along

It's been found that the stresses developed in E-Glass/Epoxy drive shaft are under permissible limit. The values of stresses developed within the shaft with its permissible limit are shown in Table No. 6 below;

Table No. 6 Stress and Strain distribution along thickness of E-Glass/Epoxy drive Shaft

Material	Allowable Stress(MPa)	Predicted Stress (MPa)	Design is OK/NOT
E-Glass/Epoxy	$S^t_1 = 400$	119.78	OK
	$S^t_2 = 20$	19.38	OK
	$S_{12} = 36$	23.45	OK
	$S^c_1 = -400$	-87.34	OK
	$S^c_2 = -20$	-18.11	OK
	$S_{12} = -36$	-26.08	OK

It's been found that the stresses developed in HS Carbon/Epoxy drive shaft are under permissible limit. The values of stresses developed within the shaft with its permissible limit are shown in Table No. 7 below;

Table No.7 Stress and Strain distribution along thickness of HS Carbon/Epoxy drive Shaft

Material	Allowable Stress(MPa)	Predicted Stress (MPa)	Design is OK/NOT
HS Carbon/Epoxy	$S^t_1 = 440$	378.36	OK
	$S^t_2 = 30$	22.24	OK
	$S_{12} = 48.5$	35.06	OK
	$S^c_1 = -440$	-392.38	OK
	$S^c_2 = -30$	-6.43	OK
	$S_{12} = -48.5$	-38.25	OK

It's been found that the stresses developed in HM Carbon/Epoxy drive shaft are under permissible limit. The values of stresses developed within the shaft with its permissible limit are shown in Table No. 8 below;

Table No.8 Stress and Strain distribution along thickness of HM Carbon/Epoxy drive Shaft

Material	Allowable Stress(MPa)	Predicted Stress (MPa)	Design is OK/NOT
HM Carbon/Epoxy	$S^t_1 = 435$	406.54	OK
	$S^t_2 = 27$	13.47	OK
	$S_{12} = 15$	12.09	OK
	$S^c_1 = -435$	-343.63	OK
	$S^c_2 = -27$	-8.07	OK
	$S_{12} = -15$	-14.08	OK

4.4   Deflection among E-Glass/Epoxy, HS Carbon/Epoxy and HM Carbon/Epoxy Drive Shafts

The deflection of Steel, E-Glass/Epoxy, HS Carbon/Epoxy and HM Carbon/Epoxy drive shafts are shown in Table 9

Table No. 9 Deflection among E-Glass/Epoxy, HS Carbon/Epoxy and HM Carbon/Epoxy Drive Shafts

Sr. No.	Material	Deflection(mm)
1	Steel	0.012407
2	E-Glass/Epoxy	0.025262
3	HS Carbon/Epoxy	0.019288
4	HM Carbon/Epoxy	0.012919

4.5 Elastic Constants of the Composite Drive Shafts

The elastic constants of E-Glass/Epoxy, HS Carbon/Epoxy and HM Carbon/Epoxy drive shafts are shown in Table 10

Table No. 10 Elastic Constant of E-Glass/Epoxy, HS Carbon/Epoxy and HM Carbon/Epoxy Drive Shafts

Sr. No.	Material	E_x (MPa)	E_y (MPa)	G_{xy} (MPa)	v_{12}
1	E-Glass/Epoxy	28.99	20.69	9.94	0.35
2	HS Carbon/Epoxy	26.69	71.68	20.02	0.2
3	HM Carbon/Epoxy	40.69	62.14	34.20	0.39

4.6 The Effect of Centrifugal Forces on the Torque Transmission Capacity

Torque transmission capacities of the drive shafts by considering and neglecting the effect of centrifugal forces are shown in the Table 11 It was observed that centrifugal forces will reduce the torque transmission capacity of the shaft.

Table No. 11 Torque Transmission Capacity of E-Glass/Epoxy, HS Carbon/Epoxy and HM Carbon/Epoxy Drive Shafts

Sr. No.	Material	E-Glass/Epoxy	HS-Carbon/Epoxy	HM-Carbon/Epoxy
1	T(Nm)with centrifugal force	3763.25	3901.43	4198.83
2	T(Nm)without centrifugal force	3893.56	4005.7	4365.54

The above result obtain from analytical calculation and ANSYS, there is a small variation which is to be considered. The FEA analysis is done to validate the analytical calculations of the work. The results of the work are encouraging and suggesting to replace the conventional two piece steel drive shaft by one piece composite material drive shaft with following major key observations;

✓ The replacement of conventional drive shaft results in reduction in weight of automobile.
✓ The finite element analysis is used in this work to predict the deformation of shaft.
✓ The areas like Torsional Buckling Capacity, Bending Natural Frequency, Weight, durability, Damping Capacity or Continuous and smooth power transmission, High Modulus Carbon/Epoxy is the best drive shaft among all.
✓ Natural frequency using Bernoulli – Euler and Timoshenko beam theories was compared. The frequency calculated by Bernoulli – Euler theory is high because it neglects the effect of rotary inertia & transverse shear.
✓ Hence the single piece High Strength Carbon / Epoxy or High Modulus Carbon/ Epoxy composite drive shaft has been proposed to design to replace the two piece conventional steel drive shaft of an automobile.

IV. Conclusions

The following conclusions are drawn from the present work.

1) A one-piece composite drive shaft for rear wheel drive automobile has been designed optimally by using FEA minimization of weight and increase life of the shaft which was subjected to the constraints such as torsional buckling capacities and natural bending frequency.

2) The weight savings of the E-Glass/ Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy shafts were equal to 49.48%, 87.21% and 87.23% of the weight of steel shaft respectively.

3) By using CLT, the variations of the stresses and strains along thickness of the E-Glass/ Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy composite drive shafts were plotted CLT. It has been observed that all the stresses were within the allowable limit.

4) The torsional buckling capacity of E-Glass/ Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy shafts were 3893.86, 4005.70, 4365.54 (N-m) respectively

5) The average deviation between Analytical study and modelling study is 3.10%, 2.53%, and 3.14% for mass, bending natural frequency and torque transmission capacity of drive shaft respectively.

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