Design and Experimental Investigation of Composite Automotive Drive Shaft

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Abstract- *Substituting composite structuresfor conventional metal structures has many advantages because of greater specific stiffness and higher specific strength of composite materials. Composite drive shafts offer the potential of lighter and longer life drive train with higher critical speed. Now a day‘s two pieces steel shaft are used as drive shaft. However, the main advantage of the Current design is only one piece of composite drive shaft is possible that fulfil all the requirements of drive shaft. The elementary requirements considered here are torsional strength, torsional buckling and bending natural frequency. This work deals with the replacement of conventional two-piece steel drive shaft with single-piece carbon/epoxy. Design of composite drive shaft is done by using Classical Lamination Theory. Finite element analysis was used to design composite drive shafts incorporating carbon within an epoxy matrix. From experimental results, it was found that the manufactured one-piece automotive composite drive shaft had 64% mass reduction, 74% increase in torque capability compared with a two-piece steel drive shaft.*

Keywords- Composite, Shaft, FEA, Fibers, Carbon Epoxy.

I. INTRODUCTION

An automotive drive shaft transmits power from the engine to the differential gear of a rear wheel drive vehicle as shown in Fig 1.1 [1]. The torque capability of the drive shaft for passenger cars should be larger than 3500Nm and the fundamental bending natural frequency should be higher than 9200rpm to avoid whirling vibration [2]. Since the fundamental bending natural frequency of a one-piece drive shafts made of steel or aluminium is normally lower than 5700rpm when the length of the drive shaft is around 1.5m [2], the 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.

The two piece steel drive shaft consists of three universal joints, a centre supporting bearing and a bracket, which increases the total weight of automotive vehicle and decreases fuel efficiency. Since carbon fibre epoxy composite materials have more than four times specific stiffness (E/ρ) of steel or aluminium materials, it is possible to manufacture composite drive shafts in one-piece without whirling vibration over 9200rpm[3].

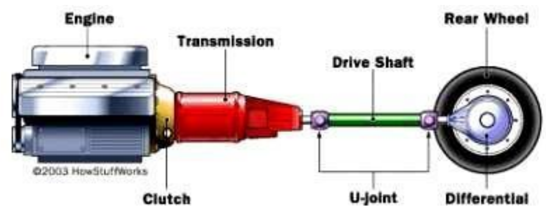


Fig 1.1 Schematic arrangement of Propeller Shaft

II. PROBLEM STATEMENT

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 shaft can have a certain role in the general weight reduction of the vehicle and is a highly desirable goal, if it can be achieve without increase in cost and decrease in quality and reliability. It is possible to achieve design of composite drive shaft with less weight to reduce the fuel consumption, increase the first natural frequency of the shaft. By doing the same, maximize the torque transmission and torsional buckling capabilities are also maximized.

This work deals with replacement of a conventional steel drive shaft with High strength carbon/Epoxy and High modulus Carbon/epoxy composite drive shaft for an automobile application.

2.1Project Objectives

a. To calculate Natural Frequency of the Propeller shaft.
b. To minimize the overall weight of composite shaft as compared to steel drive shaft
c. Modelling of the High strength carbon epoxy and Glass epoxy composite drive shaft using ANSYS.
d. Static, Modal and Buckling analysis are to be carried out on the finite element model of the High strength carbon epoxy and Glass epoxy composite drive shaft using ANSYS
e. To Validation of results with Finite element analysis.

f. Experimental validation of Torsion Strength of propeller shaft by using Torsion Test

2.2 Methodology

2.2.1. Analytical Solution

1. Torque transmission capacity of the steel drive shaft
2. Torsional Buckling Capacity of steel drive shaft
3. Fundamental bending natural frequency of steel drive Shafts.

2.2.2. Modelling of the Drive shaft with the help of Catia/AutoCAD

2.2.3. Analysis by using Software ANSYS

1. Static Analysis
2. Buckling Analysis
3. Modal Analysis

2.2.4. Manufacturing By filament winding machine

2.2.5. Testing

1. To find out the Torsional Strength of shaft.
2. To find out the Tensile strength of shaft.

A. Specification of the problem

The drive shaft outer diameter should not exceed 100mm due to space limitations. The drive shaft of transmission system is to be designed optimally for following specified design requirements which are as follows. The following specifications are assumed which are based on literature and available standards of automobile drive shaft.

TATA 407 Pickup Specifications [1]

Model : TATA 497 SP TURBO INTERCOOLED
Max. Engine Output: 55.2 kW (75 PS) at 3050 rpm
Max. Torque : 225 N-m (24.9 Mkg) at 1400-1600 rpm

In this project based on the existing model of TATA 407 pickup, the outer diameter of the drive shaft is taken as 68 mm and length of the shaft is 1300 mm.

B. Design of conventional steel Drive shaft

The steel drive shaft should satisfy three design specifications such as torque transmission capability, buckling torque capability and bending natural frequency. Steel (SM45C) used for automotive drive shaft applications. The material properties of the steel (SM45C) are given in table.

Table 2.1 Mechanical Properties of the steel (SM45C)

Mechanical properties	Symbol	Units	Steel
Young's Modulus	E	GPa	210
Shear modulus	G	GPa	80
Poisson's ratio	μ	-	0.3
Density	ρ	Kg/m3	7800
Yield Strength	S_y	MPa	370

Gear Ratio for Tata 407 pickup:-

Table 2.2 Gear ratio for different gear position

Gear Position	Gear ratio
1st Gear	6.01:1
2nd Gear	3.46:1
3rd gear	1.97:1
4th Gear	1.37:1
Reverse Gear	5.69:1

So, the torque is maximum at 1st gear, when the speed of vehicle is low. Therefore the maximum torque will be,
Tmax = 225x6.01
Tmax = 1325.25N-m

C. Analytical results by using Classical lamination theory (CL)

Table 2.3 Analytical results of Composite drive shaft

Sr. No	Parameter	Steel shaft
1	Outer Diameter	68 mm
2	Thickness	2.4 mm
3	Applied Torque (T)	1352.25 N-m
4	Torsional Buckling (T_b)	18332.05 N-m
5	Natural Frequency (f_{nb})	111.52 Hz
6	Critical speed (N_{cr})	6931.2 rpm
7	Mass (m)	5.10 g

III. DESIGN OF COMPOSITE DRIVE SHAFT

a. Specification of Problem

The specification of composite drive shaft for an automobile is same as steel drive shaft. Classical lamination theory was used for design of composite drive shaft.

In the analysis of the composite drive shaft following assumptions are made.

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 stresses-strain relationship for composite material is linear and elastic, i.e. 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 to be in plane stress condition.

The shaft can be solid circular or hollow circular. Here hollow circular cross-section was chosen because:

1. The hollow circular shafts are stronger in per kg weight than solid circular shafts.
2. 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.

The high strength and high modulus Carbon/Epoxy materials are selected for composite drive shaft.

b. Features Of Carbon Epoxy Composites

Following are the features of carbon epoxy composite, the reason for which it is chosen.

1. Carbon epoxy composite gives high tensile strength, high modulus of rigidity as compared to other composites.
2. Carbon epoxy composite has unique damping characteristic.
3. Carbon epoxy composite has positive coefficient of thermal expansion i.e. tensile strength of this composite increases with temperature.
4. Carbon epoxy composite is fatigue, wear and corrosion resistant.

Micromechanical Analysis of Lamina

a. Volume fraction of fibre – 0.7 (70%)
b. Volume fraction of matrix – 0.3 (30%)
c. Volume of composites – 1 (100%)

Using the formula of micromechanical analysis of lamina

Following properties are calculated which is shown in Table 3.1

Table 3.1 Properties of composite lamina (carbon fibre and epoxy resin)

Sr. No	Composite Properties	Symbol	Value	Unit
1	Young's Modulus(Longitudinal direction)	E_1	170.3	GPa
2	Young's Modulus(Transverse direction)	E_2	9.7	GPa
3	Major Poison's ratio	μ_{12}	0.3	-
4	Minor Poisson's ratio	μ_{21}	0.3	-
5	Shear Modulus	G_{12}	3.72	GPa
6	Ultimate longitudinal strength	$(\sigma_1^T)_{ult}$	2910	MPa
7	Ultimate transverse strength	$(\sigma_2^T)_{ult}$	25.81	MPa
8	Ultimate longitudinal compressive strength	$(\sigma_1^C)_{ult}$	83.7	MPa
9	Ultimate transverse compressive strength	$(\sigma_2^C)_{ult}$	20.21	MPa
10	Minimum fiber volume fraction	$(V_f)_{min}$	0.52	%
11	Critical fiber volume fraction	$(V_f)_{cr}$	0.53	%
12	Shear strength	τ	12.85	MPa

c. Composite Ply Orientation

Only 0^0, $\pm45^0$ and 90^0 were consider for the ply orientations, due to their higher advantages. Such as $\pm45^0$ plies increases the torsional strength/stiffness, 90^0 plies increase the critical torsional buckling load, and the 0^0 plies increase the natural frequency of the drive shaft. [1]

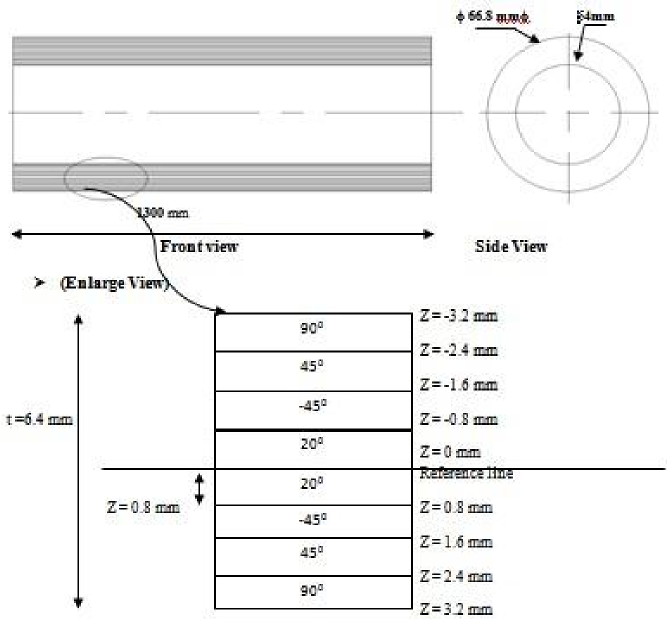


Fig 3.1 Thickness and coordinate location of the Eight-ply laminates [90 / 45 / 45 / 20]s

d. Torsional Strength of Shaft

Assuming that the drive shaft is a thin, hollow cylinder, an element in the cylinder can be assumed to be a flat laminate. The only nonzero load on this element is the shear force, N_{xy} if the average shear stress (τ_{xy}) average the applied torque then is [1]

$$T = (\tau_{xy})_{average}\Pi(r_0^2 - r_i^2)r_m$$

then

$$N_{xy} = (\tau_{xy})_{average} \times t$$

$$N_{xy} = \frac{T}{2\Pi r^2_m}$$

The mean radius, r_m is

$$r_m = r_0 - \frac{t}{2}$$

$$= 33.4 - \frac{6.4 \times 10^{-3}}{2}$$

$$r_m = 30.2\ mm$$

$$N_{xy} = \frac{1352.25}{2\Pi(0.0302)^2}$$

$$N_{xy} = 614.4 \times 10^6 N/m$$

The average shear stress is given by

$$N_{xy} = (\tau_{xy})_{average} \times t$$

$$(\tau_{xy})_{average} = \frac{N_{xy}}{t}$$

$$(\tau_{xy})_{average} = \frac{614.4 \times 10^6}{6.4 \times 10^{-3}}$$

$$(\tau_{xy})_{average} = 96.1 \times 10^9$$

$$(\tau_{xy})_{average} = 96.1 GPa$$

e. Torsional Buckling Capacity of Shaft

When a hollow shaft is subjected to torsion, at a certain amount of torsional load instability occurs. This is called the torsional buckling load. An orthotropic thin hollow cylinder will buckle torsionally, if the applies torque is greater than the critical torsional buckling load given by [3]

$$T_c = (2\Pi r^2_m t)(0.272)(E_x E^3_y)^{\frac{1}{4}}(\frac{t}{r_m})^{\frac{3}{2}}$$

$$E_x = 47.05 GPa$$

$$E_y = 58.64\ GPa$$

Because lamina thickness is 0.8 mm, the thickness of the eight ply [90 / 45 / 45 / 20]s, laminate, t is

t 8 0.8
t 6.4 mm

$$T_c = (2\Pi r^2_m t)(0.272)(E_x E^3_y)^{\frac{1}{4}}(\frac{t}{r_m})^{\frac{3}{2}}$$

$$T_c = (2\Pi \times (0.0302^2) \times (6.4 \times 10^{-3})(0.272)[(22.58 \times 10^9)(106 \times 10^9)^3]^{0.25}(\frac{6.4 \times 10^{-3}}{0.0302})^{\frac{3}{2}}$$

$$T_c = 69933.88 N-m$$

This value is greater than the applied torque of 1352.25 N-m, thus the composite drive shaft is safe in buckling.

f. Natural frequency

Let us find out the minimum natural frequency of the drive shaft, which is given by

$$f_{nt} = \frac{\pi}{2}\sqrt{\frac{EI_x}{mL^4}} \quad \ldots\ldots\ldots\ldots$$

$$f_{nt} = \frac{\pi}{(2)}\frac{\sqrt{(47.05 \times 10^9)(5.60 \times 10^{-7})}}{(1.88)(1.3^4)}$$

$$f_{nt} = 110.03\ HZ$$

Because the minimum natural frequency is required to be 50 Hz, this requirement is also meet by the [90 / 45 / 45 / 20]s laminate.

g. Mass Saving

1. Mass of steel drive shaft = 5.10 kg

2. Mass of Composite drive shaft

$$m = \Pi(r_0^{\,2} - r_i^{\,2}) L \rho$$

$$m = \Pi(0.0334^2 - 0.027^2)(1.3)(1550.5)$$

$$m = 1.88\ kg$$

3. Percentage of mass saving over steel is

$$\frac{5.10 \quad 1.88}{5.10} \quad 100$$

$$63.13\ \%$$

h. Results and Discussion

Table 3.2 Comparison between steel and composite drive shaft

Sr. No	Parameter	Steel shaft	Composite shaft
1	Outer Diameter	68 mm	66.8 mm
2	thickness	2.4 mm	6.4 mm
3	Applied Torque (T)	1352.25 N-m	1352.25N-m
4	Torsional Buckling (T_b)	18332.05 N-m	69933.88N-m
5	Natural Frequency (f_{nb})	111.84 Hz	110.03 Hz
6	Critical speed (N_{cr})	6710.4 rpm	6601.8 rpm
7	Mass (m)	5.10 kg	1.88 kg
8	Percentage of mass saving	-	63.13 %

Table 3.2 shows that the comparison between steel and composite drive shaft ,in which applied torque is same for steel and composite drive shaft but we can see that the composite driveshaft is having higher torsional buckling strength as compare to steel drive shaft. Minimum requirement of natural frequency is 50 Hz which can be achieved by both steel and composite drive shaft. Composite shaft is having very low weight of 1.8 kg whereas steel shaft of weight is 5.10 kg.

IV. FINITE ELEMENT ANALYSIS

Finite element analysis is a computer based analysis technique for calculating the strength and behaviour of structures. In the FEM the structure is represented as finite elements. These elements are joined at particular points which are called as nodes. The FEA is used to calculate the deflection, stresses, strains temperature, buckling behaviour of the member. In our project FEA is carried out by using the ANSYS 14.5. Initially we don't know the displacement and other quantities like strains, stresses which are then calculated from nodal displacement.

4.1. Static Analysis

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. In static analysis loading and response conditions are assumed, that is the loads and the structure responses are assumed to vary slowly with respect to time.

4.1.1 Static Analysis of Steel Drive Shaft

The below image is showing shear stress value with the help of color bar. Color bar is used to determine the value ranges on object. Maximum shear stress in the steel drive shaft is 88.272 MPa which are within permissible limit

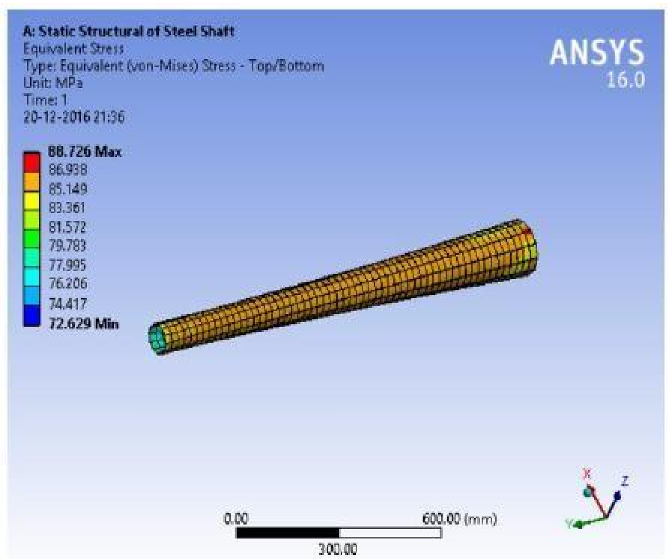


Figure 4.1.1 Maximum shear stress in the Steel drive shaft

4.1.2 Static Analysis of Composite Drive Shaft

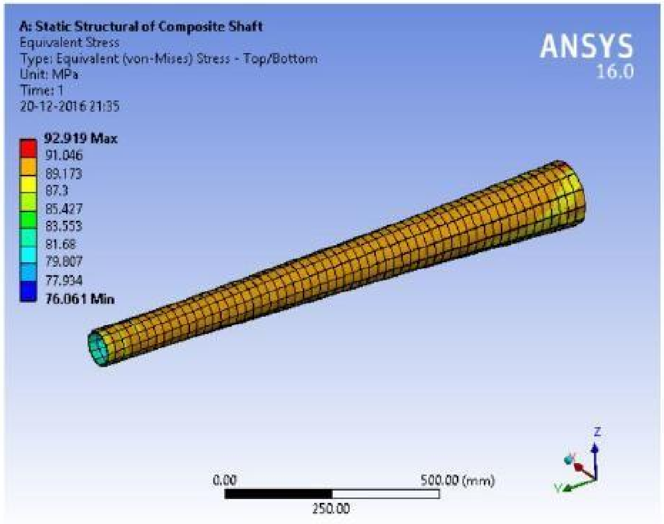


Figure 4.1.2 Maximum shear stress in the composite drive shaft

The above image is showing shear stress value with the help of color bar. Color bar is used to determine the value ranges on object. Maximum shear stress in the composite drive shaft is 92.91 MPa which are within permissible limit.

4.2 Boundary Condition

The finite element model of HS Carbon fibre / Epoxy shaft has one end is fixed and torque is applied at other end. The torque of 1352 N-m (for carbon shaft and steel shaft) is applied at the other end which is free. As there are drawbacks in manufacturing method of both shafts taking the experimental values for analysis may not yield the best possible results.

4.3.Modal Analysis

When an elastic system free from external forces can disturbed from its equilibrium position and vibrates under the influence of inherent forces and is said to be in the state of free vibration. It will vibrate at its natural frequency and its amplitude will gradually become smaller with time due to energy being dissipated by motion. The main parameters of interest in free vibration are natural frequency and the amplitude. The Mode shapes and Natural frequencies are important parameters in the design of a structure for dynamic loading conditions. Modal analysis is used to determine the vibration characteristics such as mode shapes and natural frequencies of a structure or a machine component while it is being designed. Modal analysis is used to determine the mode shapes and natural frequencies of a structure or a machine component. The rotational speed is limited by lateral stability considerations.

4.3.1 First Natural frequency of composite drive shaft

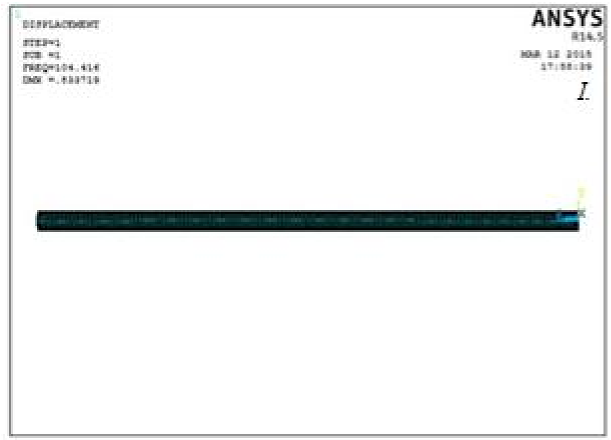


Figure 4.3.1 1st Mode shape of carbon composite shaft

4.3.2 First Natural frequency of Steel drive shaft

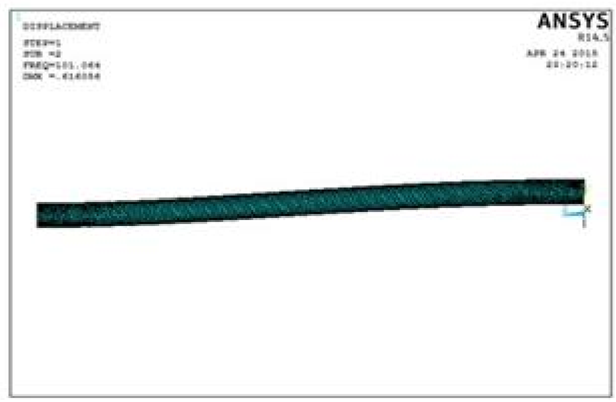


Figure 4.3.2 1st Mode shape of Steel shaft

4.3.3 Results & Discussion

Table 4.1 Finite element Method Result

Analysis Type	Composite shaft	Steel shaft
	Modal Analysis	
1 st Frequency	104.41 Hz	101.06 Hz
2 ndFrequency	388.28 Hz	533.26 Hz
3 rd Frequency	917.59 Hz	1073.07 Hz
	Static Analysis	
Maximum Shear Stress	92.91 MPa	88.27 MPa
	Dynamic Analysis	
Maximum Shear Stress	98.30 MPa	87.38 MPa

V. MANUFACTURING METHODOLOGY

Unlike most conventional materials, there is a very close relation between the manufacture of a composite material and its end use. The manufacture of the material is often part of the fabrication process for the structural element or even the complete structure. The selection of a fabrication process obviously depends on the constituent materials in the composite, with the matrix being the key (i.e. the process for polymer matrix, metal matrix and ceramic matrix composites are generally quite different).

I. Filament winding Machines

The simplest winding machines have two axes of motion, the mandrel rotation and the carriage travel (usually horizontal). Two axis machines are best suited to the manufacture of pipes only. For pressure vessels such as LPG or CNG containers (for example) it is normal to have a four axis winding machine. A four-axis machine additionally has a radial (cross-feed) axis perpendicular to carriage travel and a rotating fibre payout head mounted to the cross-feed axis. The payout head rotation can be used to stop the fibre band twisting and thus varying in width during winding.

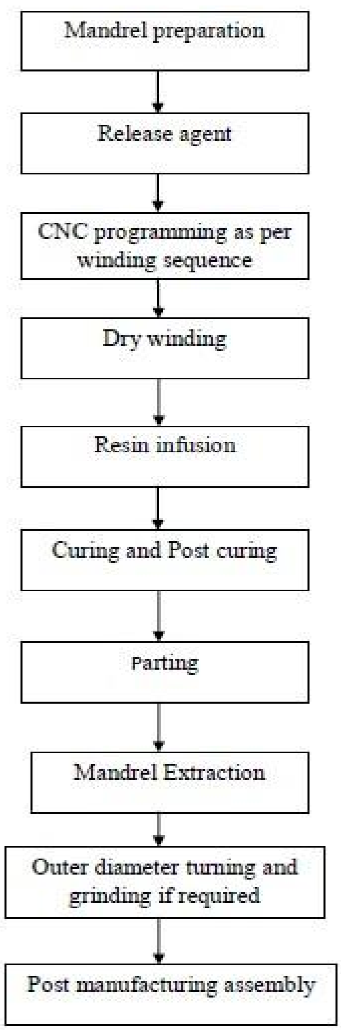


Fig. 5.1.1 Steps used in manufacturing of filament wound product

II. Torsion Test Setup

Generally, torsion occurs when the twisting moment or torque is applied to a member according to fig. 1. The torque is the product of tangential force multiplied by the radial distance from the twisting axis and the tangent, measured in a unit of N-m. In torsion testing, the relationship between torque and degree of rotation is graphically presented and parameters such as ultimate torsional shearing strength (modulus of rupture), shear strength at proportional limit and shear modulus (modulus Of rigidity) are generally investigated. Moreover, fracture surfaces of specimens tested under torsion can be used to determine the characteristics of the materials whether it would fail in a brittle or a ductile manner.

Fig.5.2.1 Torsion testing machine

The data that was collected during the torsion testing lab was Torque (kg-m), and angle of twist. Plot a graph of applied Torque 'T' against angle of twist as a base for the elastic region.

Fig. 5.2.2 Twisted Composite Shafts

Plot a graph of applied torque against angle of twist of the specimen as a base, for the complete test to destruction.

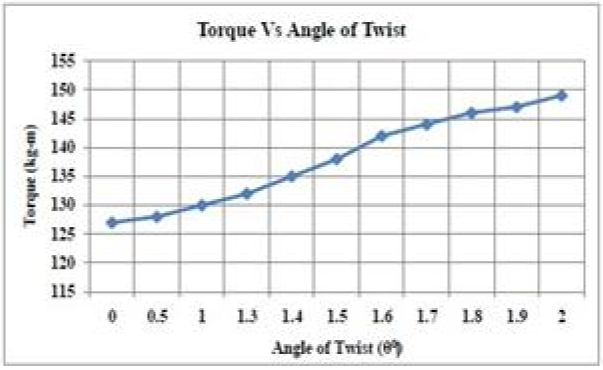


Fig.5.2.3. Graph of Torque Vs Angle of Twist

The above results are showing a linear increase in deflection with increasing torque. In this torsion testing we apply torque up to 1500 N-m (150 kg-m) on the composite drive shaft. Here we want to transmit the torque up to 1352.25 N-m and the shaft is slightly twisted at 1500 N-m. This is an expected output and it confirms with the theoretical behaviour of a shaft subjected to torsional loading. Thus theoretically speaking, there exists a complete linear relationship between the applied torque and the angular deflection as shown in the graphs.

III. FFT Analyzer Setup

The FFT analyzer allows real-time, multi-channel FFT spectrum analysis, whether you want to perform mobility measurements, vibration diagnostics, or narrow-band analysis of acoustic signals. For characterizing your noise or vibration signals, there is an overall level analyzer, which performs a broadband analysis. When measuring sound, this analyzer is equivalent to a sound level meter and fulfils selected, relevant requirements of IEC 651, IEC 61672 and IEC 60804 for a class 1 instrument.

RT Pro-PHOTON+ is a ultra-portable, real-time analyzer designed for making fast and accurate vibration measurements with post-processing capabilities. It offers the convenience and speed of the industry standard USB 2.0 interface, and comes with two to four analog input channels, plus a waveform source/tachometer channel.

The PHOTON+ All-In-One System accelerates the time from data acquisition to analysis. Dynamic switching between multiple measurement projects lets you navigate easily between data recording, data post processing and analysis, and real-time analysis. With just one instrument you can use it as an FFT analyzer, vibration analyzer, or signal analyzer, with recorded data interpreted in a wide variety of ways without spending time translating the data file format or changing between different software programmers.

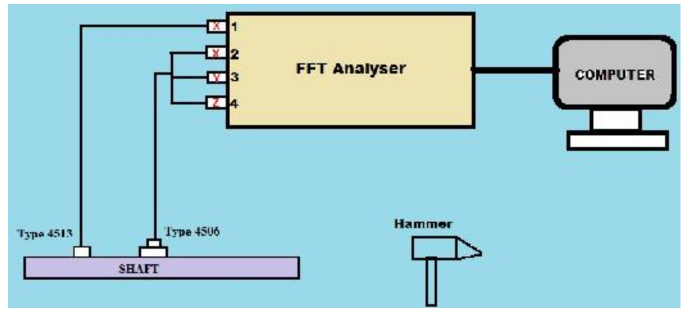
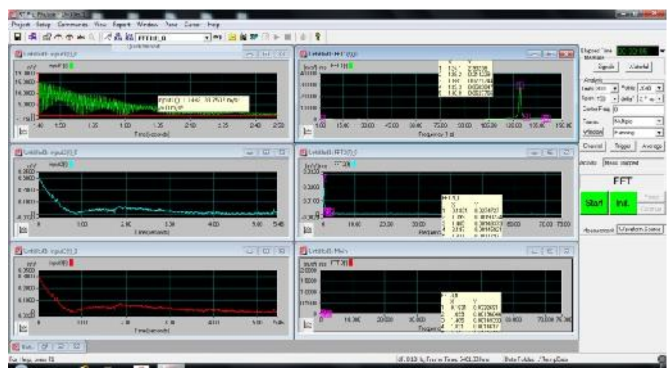


Figure 7.3.1.Schematic diagram of FFT analyzer

a. Triaxial Delta Tron Accelerometer Type 4506
Observation No. 1

First up all we choose Triaxial Delta Tron Accelerometer Type 4506 sensor which is mounted on the shaft which is as shown in fig., then we follow the above procedure and we get the Following results.

Project File Name: Type 4506



The above screenshot taken for the 4506 type side sensor to show all the result, in which right hand graph is plotted frequency Vs acceleration value and the left hand side graph is Time Vs acceleration data

Experimental Results:

Table 5.1 Experimental Result

Sr No.	Sensor	Natural Frequency
1	Type 4506	124 Hz
2	Type 4513	127.5 Hz

VI. CONCLUSION

In this thesis, firstly the Design of Composite drive shaft is done by using Classical lamination Theory. Then its finite element modal analysis was carried out with help of FE software ANSYS. After the manufacturing, its experimental modal analysis and Torsion Testing was performed. From above study, some conclusions can be made as: The following conclusions are drawn from the Present work.

I. The replacement of conventional drive shaft results in 64 %
II. reduction in weight of automobile shaft.

III. Classical lamination theory was used in this work to find out torsional buckling strength, natural frequency of the shaft

IV. Composite shaft in comparison with the steel shaft with the same geometry have lower buckling torque.

V. The fibre orientation of composite shaft strongly changes the buckling torque.

VI. Bernoulli-Euler theory was used to know the natural frequency of shaft

APPENDIX

Journal Paper submitted

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