# Optimal Design and FEA Analysis of Composite Drive Shaft

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***Abstract- The*advanced composites has resulted in remarkable achievements in many fields including marine and automobile engineering, prosthetics,medicine and sports, in terms of improved fatigue and corrosion resistances, high specific modulus and specific Strength and reduction in energy requirements owing to reduction in weight. The aim of this work is that to replace the conventional steelshaft of automobiles with an appropriate composite shaft. The conventional driveshafts are made in two pieces for reducing the bending natural frequency, whereas the composite shafts made as single-piece shafts, thus reducing the overall weight. E-Glass/Epoxy and Kevlar/Epoxy composites were design and analyze in terms of torsional strength,torsional buckling and bending natural frequency by compare them with the conventional steel driveshaft under the same grounds of design constraints and the best suited composite material was recommended. In this present work an attempt has been to estimate the deflection, stresses, and natural frequencies under subjected loads using FEA (Ansys).**

**Keyword: -propeller shaft,Drive shaft, optimization,composite material, composite drive shaft design etc.**

## I. INTRODUCTION

An automotive drive shaft transmits power from the engine to the differential gear of a rear wheel drive vehicle. The torque capability of the drive shaft for passenger cars should be larger than 3500 Nm and the fundamental bending natural frequency should be higher than 9200 rpm to avoid whirling vibration. Since the fundamental bending natural frequency of a one-piece drive shafts made of steel or aluminum is normally lower than 5700 rpm when the length of the drive shaft is around 1.5 m, the steel drive shaft is usually manufactured in two pieces to increase the fundamental natural frequency of bending because the natural frequency ofbending 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 center supporting bearing and a bracket, which increases the total weight of an automotive vehicle and decreases fuel efficiency. Since composite materials have more than four times specific stiffness (E= ρ) of Aluminum or steel materials, it is possible to manufacture composite drive shafts in one-piece without whirling vibration over 9200 rpm. The composite drive shaft reduced weight and less noise and vibration. However, because of the high material cost of E-Glass and Kevlar fiber epoxy composite materials, rather cheap aluminum materials may be used partly with composite materials such as in a hybrid type of aluminum/composite drive shaft, in which the aluminum has a role to transmit the require torque, while the E- Glass fiber epoxy composite increases the bending natural frequency above 9200 rpm.

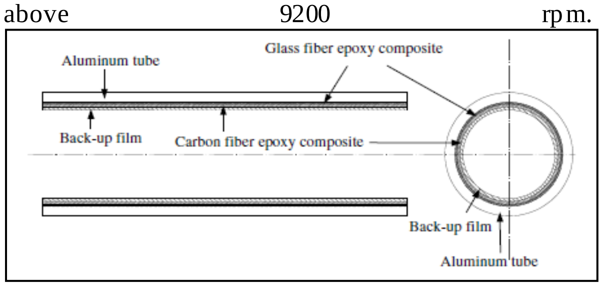



Fig.1 Schematic Diagram of the Co-Cured Aluminum/Composite Drive Shaft

## II. MATERIAL SELECTION

### *3.1 Selection of Reinforcement Fiber*

Fibers are available with widely differing properties. Review of the design and performance requirements usually dictate the fiber/fibers to be used. E-glass/ Kevlar fibers: Its advantages include high specific strength and modulus, low coefficient of thermal expansion, and high fatigue strength. Graphite, when used alone has low impact resistance. Its drawbacks include high cost, low impact resistance, and high electrical conductivity. Glass fibers: Its advantages include its low cost, high strength, high chemical resistance, and good insulating properties. The disadvantages are low elastic modulus, poor adhesion to polymers, low fatigue strength, and high density, which increase shaft size and weight. Also crack detection becomes difficult.

*3.2 Selection of Resin System*

The important considerations in selecting resin are cost, temperature capability, elongation to failure and resistance to impact (a function of modulus of elongation). The resins selected for most of the drive shafts are either epoxies or vinyl esters. Here, epoxy resin was selected due to its high strength, good wetting of fibers, lower curing shrinkage, and better dimensional stability.

TABLE-1
MATERIAL PROPERTIES OF STEEL(SM45C)

| SN | Mechanical Properties | Symbol | Unit | Value |
|---|---|---|---|---|
| 1 | Young's Modulus | E | GPa | 207.0 |
| 2 | Shear Modulus | G | GPa | 80.0 |
| 3 | Poisson's Ratio | υ | ----- | 0.3 |
| 4 | Density | P | Kg/m3 | 7600 |
| 5 | Yield Strength | Sy | MPa | 370 |
| 6 | Shear Strength | Sx | MPa | 275 |

## III. DESIGN OF DRIVE SHAFT

*3.1 Assumptions*

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

*3.2 Selection of Cross-Section*

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

i. The hollow circular shafts are stronger in per kg weight than solid circular.
ii. 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.

TABLE-2
SPECIFICATION OF DRIVE SHAFT

| SN | Name | Notation | Unit | Value |
|---|---|---|---|---|
| 1 | Ultimate Torque | T | Nm | 3500 |
| 2 | Max. Speed of shaft | N | Rpm | 6500 |
| 3 | Length of shaft | L | mm | 1250 |
| 4 | Outer Diameter of shaft | do | mm | 92 |
| 5 | Inner Diameter | di | mm | 80 |
| 6 | Thickness of shaft | t | mm | 6 |

*3.3 Mass of Drive Shaft*

$$M = \rho AL = \rho\,(do2 - di2) \times \frac{L}{4}$$

Where, do = outer diameter (m)
di = inner diameter (m)

$$m = 8.58 \text{ Kg}$$

*3.4 Torque Transmission Capacity of Drive Shaft*

$$T = Ss\frac{\pi(do^4 - di^4)}{16do}$$

$$T = 123.33 \text{ X } 106 \times \frac{\pi(0.092^4 - 0.08^4)}{16 \text{ X } 3.32 \text{ X } 0.09}$$

Taking factor of safety as 3,

$$T = 4599.19 \text{ Nm}$$

*3.5 Torsional buckling capacity of the drive shaft*

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

For long shaft, the critical stress is given by,

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

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

$$\tau cr = \frac{4.39 \text{ E}}{(1-\nu^2)}(t/r^2)\sqrt{1 + 0.0257\,(1-\nu^2)^{\frac{3}{4}}}\,\frac{L^3}{(rt)^{1.5}}$$

$$\tau cr = 1119.65 \text{ N / mm2}$$

The relation between thecritical stress and torsional Buckling Capacity is given by,

$$T\ cr = \tau\ cr\ 2\pi r2t$$

$$Tcr = 43857.9 \text{ N-m}$$

*3.6 Lateral or Bending Vibration*

The shaft is considered as simply supported beam undergoes transverse vibration. Thus the Natural Frequency can be found by using two theories.

- *Bernoulli-Euler Beam Theory- Ncrbe*

It neglects the both transverse shear deformation and rotary inertia effects. Natural frequency is given by,

$$fnbe = \frac{\pi p^2}{2L^2}\sqrt{\frac{EI_x}{m_l}}$$

Where, p = 1, 2…

$$N\ crbe = 60f\ nbe$$

$$fnbe = 161.03\ \text{Hz}$$

$$N\ crbe = 9662.38\ \text{rpm}$$

- *Timoshenko Beam Theory-Ncrt*

It considers both transverse shear deformation and rotary inertia effects. Natural frequency based on the Timoshenko beam theory is given by,

$$fnt = Ks\left(\frac{30\pi p^2}{L^2}\right)\sqrt{\frac{Er^2}{2\rho}}$$

$$N\ crt = 60f\ nt$$

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

$$Ks = 0.964$$

fs= 2 for hollow circular cross-sections

The relation between Timoshenko and Bernoulli-Euler beam theories is given by,

$$fnt = Ks\ f\ nbe$$

$$fnt = 0.962\ X\ 161.03$$

$$fnt = 155.32\ \text{Hz}$$

$$N\ crt = 9319.98\ \text{rpm}$$

TABLE-3
MATERIAL PROPERTIES OF CARBON/EPOXY COMPOSITE AND GLASS EPOXY COMPOSITE

| S N | Properties | Symbols | Units | E-glass / Epoxy | Kevlar / Epoxy |
|---|---|---|---|---|---|
| 1 | Longitudinal Modulus | E11 | GPa | 134 | 170 |
| 2 | Transverse Modulus | E22 | GPa | 7.0 | 10.0 |
| 3 | Shear Modulus | G12 | GPa | 5.8 | 6 |
| 4 | Poisson's Ratio | υ | ---- | 0.3 | 0.3 |
| 5 | Density | P | Kg/m3 | 2200 | 1450 |
| 6 | Longitudinal tensile strength | St1 | MPa | 870 | 880 |
| 7 | Transverse tensile strength | St2 | MPa | 60 | 70 |
| 8 | Shear strength | Ss | MPa | 97 | 100 |

Similarly, we can calculate the Torque transmission capacity, Torsional buckling capacity, Frequency for composite shaft. We get the design solution as,

TABLE-4
THE TORQUE TRANSMISSION CAPACITY OF THE SHAFT

| Material | Steel | E-glass / Epoxy | Kevlar / Epoxy |
|---|---|---|---|
| Torque, T (N-m) | 4599.19 | 5260.18 | 5570.26 |

TABLE-5
EFFECT OF TRANSVERSE SHEAR ON THE FUNDAMENTAL NATURAL FREQUENCY

| Material | Steel | E-glass / Epoxy | Kevlar / Epoxy |
|---|---|---|---|
| Ncrbe (rpm) | 9662.3 | 9461.65 | 7663.31 |
| Ncrt (rpm) | 9319.9 | 9270.28 | 7495.42 |

## IV. ANALYSIS OF DRIVE SHAFT USING ANSYS

### *4.1 Modeling and simulation*

In this section 3D FE Models along with the loads and boundary conditions will be presented.

*Step1-* 3D PROE Model Creation is based on Specifications and design consideration from passenger car, van specification.

*Step 2-* 3D FE Model Creation The 3D FE model for drive shaft was created by using Finite Element modeling software. The mesh has been generated using relevance in ANSYS 16.0 APDL.

*Step 3-* using model with boundary conditions in ansys required results are predicted.

*Step 4-* By applying boundary conditions and loading conditions we will comparedobtained resultssuggestedthe suitable material which gives less torsional value and frequency nearer to steel.

### *4.2 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. The kinds of loading that can be applied in static analysis includes, Externally applied forces, moments and pressures Steady state inertial forces such as gravity and spinning Imposed non-zero displacements. If the stress values obtained in this analysis crosses the allowable values it will result in the failure of the structure in the static condition itself. To avoid such a failure, this analysis is necessary. Boundary conditions The finite element model of E- glass / Epoxy shaft is shown in Figure .One end is fixed and torque is applied at other end.

*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 natural frequencies and the mode shapes are important parameters in the design of a structure for dynamic loading conditions. Modal analysis 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. 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.

*4.4 Buckling Analysis*

Buckling analysis is a technique used to determine buckling loads (critical loads) at which a structure becomes unstable, and buckled mode shapes. 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.

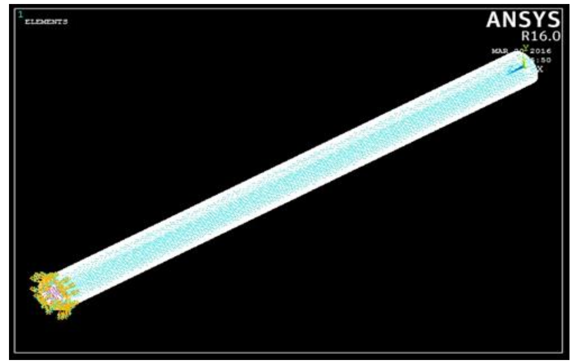
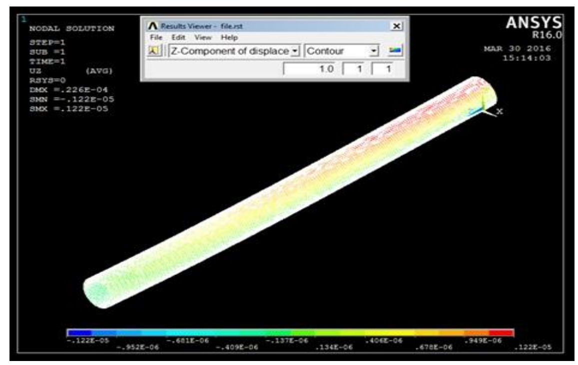
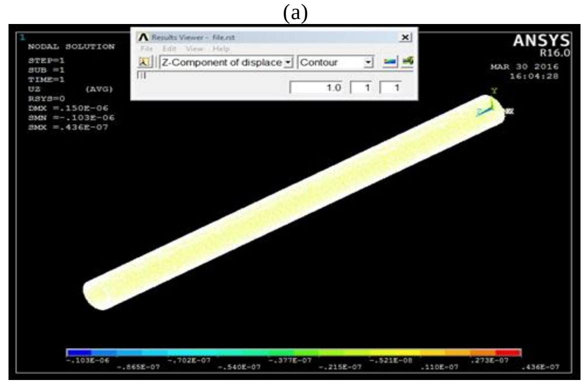


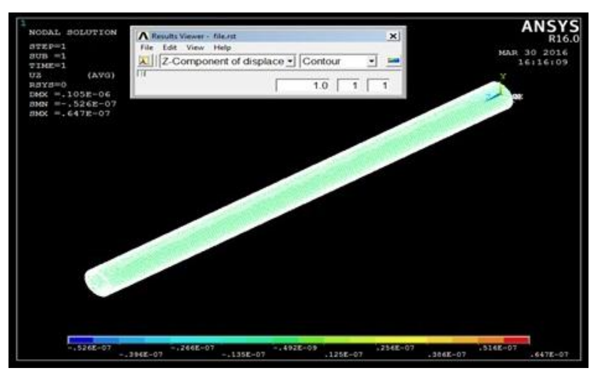

Fig. 2 Boundary Conditions for the Modal Analysis




(a)

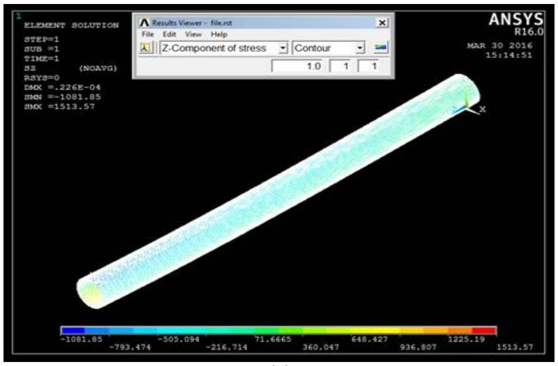



(b)

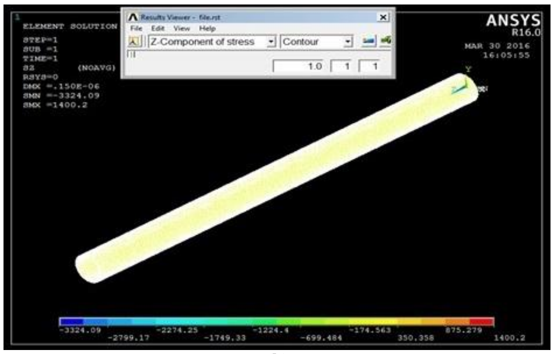



(c)

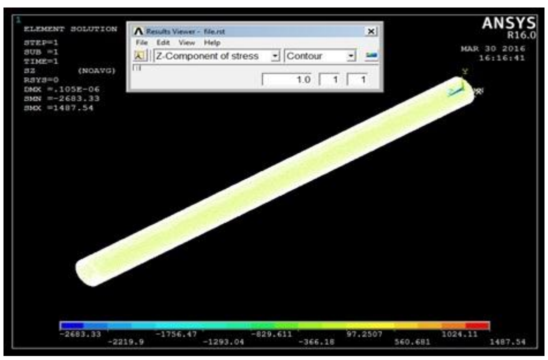
Fig.3 Maximum Deformation (a) Steel (b) E- Glass/ Epoxy (c) Kevlar/ Epoxy



(a)



(b)



(c)

Fig.4 Maximum Stess (a) Steel (b) E- Glass/ Epoxy (c) Kevlar/ Epoxy

## V. DESIGN OPTIMIZATION

A simple Genetic Algorithm (GA) is used to obtain the optimal number of layers, thickness of ply and fiber orientation of each layer. All the design variables are discrete in nature and easily handled by GA. A one-piece composite drive shaft for rear wheel drive automobile was designed optimally by using genetic Algorithm for E glass / Epoxy and Kevlar / 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.

A simple Genetic Algorithm is designed to optimize the weight of the drive shaft by using the MATLAB 7.10.0. The different M files are written in MATLAB.

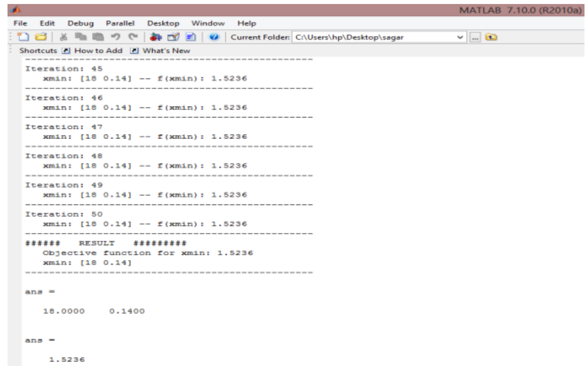



Fig.5 Output of Genetic Algorithm for E-glass / Epoxy

TABLE-6
SUMMARY OF GA RESULT

| Parameters | Steel | E-glass / Epoxy | Kevlar / Epoxy |
|---|---|---|---|
| do (mm) | 90 | 90 | 90 |
| L (mm) | 1250 | 1250 | 1250 |
| tk (mm) | 6 | 0.42~0.5 | 0.43~0.5 |
| Optimum no. of Layers | 1 | 4.2~4 | 3.9~4 |
| t (mm) | 6 | 2+(steel 1mm) | 2+(steel 1mm) |
| Weight (kg) | 15 | 1.5+2.5 | 1.01+2.5 |
| Weight saving (%) | - | 73% | 78% |

### *3.7 Deflection*

Deflection of a shaft is calculated as follows: -

For steel shaft :-

$$\theta = \frac{32 \text{ X } 10^3 \text{ X T X L}}{\pi \text{ X G X} (do^4 - di^4)}$$

$$\theta = \frac{2 \text{ X } 103 \text{ X } 3723.4 \text{ X } 1250}{\pi \text{ X } 80 \text{ X } 10^3 \text{ X } (90^4 - 83.36^4)}$$

$$\theta = 0.03422 \text{ radians or } 1.96^{o}$$

For E-Glass / Epoxy Drive Shaft: -

$$\theta = \frac{32 \text{ X } 10^3 \text{ X T X L}}{\pi \text{ X G X} (do^4 - di^4)}$$

$$\theta = \frac{2 \text{ X } 103 \text{ X } 4023.5 \text{ X } 1250}{\pi \text{ X } 20.02 \text{ X } 10^3 \text{ X } (90^4 - 85.92^4)}$$

$$\theta = 0.230 \text{ radians or } 13.20^{o}$$

For Kevlar / Epoxy Drive Shaft: -

$$\theta = \frac{32 \text{ X } 10^3 \text{ X T X L}}{\pi \text{ X G X} (do^4 - di^4)}$$

$$\theta = \frac{2 \text{ X } 103 \text{ X } 3875.3 \text{ X } 1250}{\pi \text{ X } 34.20 \text{ X } 10^3 \text{ X } (90^4 - 84.96^4)}$$

$\theta$ = 0.1068 radians or 6.125°

TABLE-7
DEFLECTION OF DRIVE SHAFTS

| Material | Deflection, $\theta$ (Radians) |
|---|---|
| Steel | 0.3422 |
| E-glass / Epoxy | 0.1068 |
| Kevlar / Epoxy | 0.230 |

## VI. CONCLUSION

- Taking into account the weight saving, deformation, shear stress induced and resultant frequency it is evident that composite has the most encouraging properties to act as replacement to steel
- Considerable weight benefits have been obtained to the extent of a minimum of 40%.This study indicates that carbon fiber shafts can replace conventional steel drive shaft.
- The present work was aimed at reducing the fuel consumption of the automobiles in particular or any machine, which employs drive shaft, in general. This was achieved by reducing the weight of the drive shaft with the use of composite materials. This also allows the use of a single drive shaft (instead of a two piece drive shaft) for transmission of power to the differential parts of the assembly.
- Analysis of both drive shaft shows that the composite drive shaft has capability to transmit more torque, has more buckling torque transmission capability and has much higher fundamental natural bending frequency which provides better margin of safety than the conventional composite drive shaft.
- Apart from being lightweight, the use of composites also ensures less noise and vibration.
- If we consider cost of glass/epoxy composite, it is slightly higher than steel but lesser than carbon/epoxy.
- The composite drive are safer and reliable than steel as design parameter are higher in case of composite.
- The composite are recyclable so they can be reuse.
- Apart from being lightweight, the use of composites also ensures less noise and vibration.
- So in comparison of mass, cost, safety and recycling steel shaft can be replaced by composite drive shaft.
- Natural frequency using Bernoulli-euler beam theory and Timoshenko's beam theory are compared. The frequency calculated by using Bernoulli-euler beam theory is high as it neglects rotary inertia and transverse shear.
- The successful application of the present design can make a huge improvement in automotive industry.

## VII. FUTURE SCOPE

- This study leaves wide scope for future investigations. It can be extended to newer composites using other reinforcing phases and the resulting experimental findings can be similarly analyzed.
- Tribological evaluation of glass/carbon fiber reinforced epoxy resin composite has been a much less studied area. There is a very wide scope for future scholars to explore this area of research. Many other aspects of this problem like effect of fiber orientation, loading pattern, weight fraction of ceramic fillers on wear response of such composites require further investigation.

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