

Design and Analysis of Automotive Composite Propeller Shaft Using FEA

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ABSTRACT: The last few years have seen the increasing use of composite materials in many fields of engineering applications. Polymer composites are today widely used to design the automobile components in view of their outstanding specific stiffness and strength properties. Composite shafts for automotive applications are among the most current areas of investigation. Weight reduction can be primarily achieved by the introduction of better material. The conventional system uses metallic shaft, has inherent limitations like heavy weight, corrosion, flexibility problems, vibrations, bearing and manufacturing problems, which magnifies with increase in shaft diameter. Advanced composite materials offer the potential to improve propulsion shafting, by reducing weight, bearing loads, alignment problems, life-cycle cost by using strategic materials, by increasing allowable fatigue stress, flexibility, and vibration damping characteristics.

This paper is related to investigations on Carbon Reinforced plastics (CFRP) and Glass Fiber Reinforced Plastics (GFRP) composite hollow shafts for automobiles. Failure analysis has been carried out using maximum stress criteria and it is found that the failure torque is well above the design torque level.

For accurate design solution, the propeller shaft was analyzed using FEM techniques (ANSYS package). The propeller shaft was geometrically modeled using FEM “3D-shell99 element and solid46 layered element”. To check all failure modes, linear static analysis, vibration Eigen value analysis, buckling analysis and harmonic analysis were done.

KEYWORDS: Propeller shaft, Composite shafts, Automotive applications, etc.

I. INTRODUCTION

Composite materials can be defined as a macroscopic combination of two or more materials having a recognizable interface between them. Composite materials typically have a fiber or particle phase that is stiffer and stronger than the continuous phase. Now day's people are using composite materials for many number of application in various fields, some of them are aerospace, automotive, construction etc. In the case of automotive application people are using the composite materials for the car door panels, bonnet construction, coming to the transmission system in the form of FRP composite propeller shafts. The extensive application of the composite materials is due to their superior properties over the conventional materials.

An automotive propeller shaft transmits power from the engine to the differential gear of a rear wheel drive vehicle. The power is delivered to the shaft by torque (or say, twisting moment) set up within the shaft, which permits the power to be transmitted to various machines, linked to the shaft, depending on the applications. The conventional materials used for ordinary shaft are steel, generally 35C8, 45C8, 55C8 etc. When high strength is required alloy steel such as nickel, nickel-chromium or chrome-vanadium steel is used. The fundamental natural frequency of the carbon fiber

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composite propeller shaft can be twice as high as that of steel or aluminum because the carbon fiber composite material has more than 4 times the specific stiffness of steel or aluminum which makes it possible to manufacture the propeller shaft of passenger cars on one piece. The composite propeller shaft has many other benefits such as reduced weight and less noise and vibration

The static torque transmission capability of the composite shaft was defined in this work as the torque value at which the first ply of the composite shaft failed. Since long thin hollow shafts are vulnerable to torsional buckling.

II. OBJECTIVE & SCOPE

The main objective of this work is to analyze the composite propeller shaft using finite element analysis. To develop an adequate technology base required to successfully design, specify, and manufacture composite shafting in future automobiles and to demonstrate the feasibility of this technology. Validate with user-friendly computer software for the design of composite shafts and using empirical formulae.

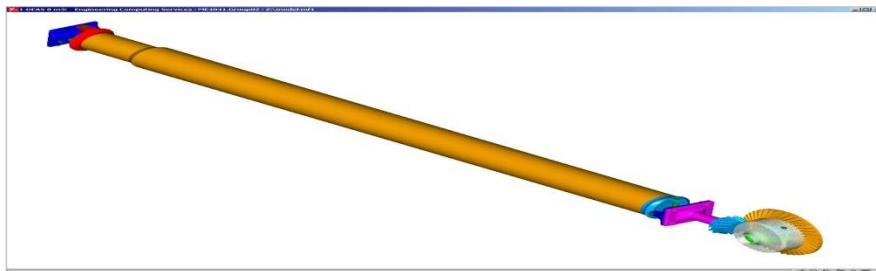


Fig.1 Schematic diagram of Propeller shaft arrangement

III. MODELING USING ANSYS

The propeller shaft was modeled in ANSYS using the macro in which the parameters mean radius of the shaft, length of the shaft, thickness of hollow shaft, thickness of each fiber, fiber orientation angle in each layer. For geometric modeling shaft was idealized as cylindrical tube and then layers were defined from bottom to top. The '3D laminated composite general shell99 element', (KEYOPT1=8) was selected for meshing. The shape of the element was 'quadrilateral'. After creating FEM mesh of nodes and elements, nodes were merged. The geometric model was verified for geometric check, bad element check. The material used is glass fiber reinforced in epoxy matrix and carbon fiber reinforced in epoxy matrix.

DIMENSIONS OF THE PROPELLER SHAFT

Length of the shaft	: 1730 mm
Mean radius of the shaft	: 40 mm
Thickness of the hollow shaft	: 4.578 mm (Carbon/epoxy shaft)
	: 5.110 mm (Glass/epoxy shaft)

BOUNDARY CONDITIONS

As the figure 2 shows one end of the propeller shaft is fixed to a shaft, which is connected to gearbox. Hence all the degrees of freedom at that end are arrested as shown in the figure 2. Another end of the shaft is connected to the differential gear shaft. As the propeller shaft is subjected to the twisting moment, a moment is applied about x-axis as shown in the figure.

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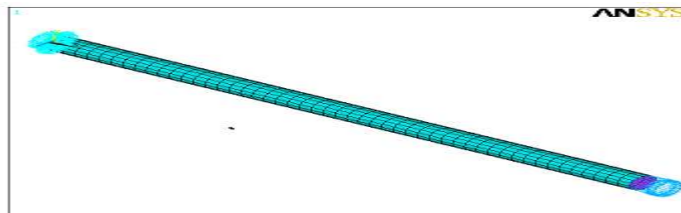


Fig. 2 FEA model of the composite propeller shaft

Structural static analysis was carried out on the shaft. After doing the structural static analysis for a torque of 2030 N-m, the optimized parameters were found with mean radius, thickness of each layer as the design variables, and the maximum allowable shear stress, maximum twist as the state variables and the total volume as the objective function which is to be optimized (minimization). The results of static analysis i.e., the maximum twist of the shaft and the maximum shear stress developed were checked and those are found to be within the allowable values.

IV. DESIGN THEOREMS

A. DESIGNING FOR TORSION

The torsional load can be carried most effectively by fibers placed at $\pm 45^\circ$ to the angle direction. The 0° and 90° orientations are required for other purposes, such as axial strength, and their contribution to torsional strength, although not negligible, is comparatively small.

B. DESIGNING FOR TORSIONAL BUCKLING

An expression for the torsional buckling of a thin-walled orthotropic tube is given as follows:

where t is the thickness of the tube, E_x and E_θ are the average moduli in the axial and hoop directions respectively. If E_x and E_θ are known, the critical torque for buckling, T_c , is easily calculated.

$$T_c = (2\pi r^2 t)(0.272) [E_x E_\theta]^{1/4} (t/r)^{9/2}$$

C. DESIGNING FOR NATURAL FREQUENCY RESISTANCE

The drive shaft can be idealized as a pinned-pinned beam. The expression for the lowest natural frequency (f_x) is given as follows:

$$f_x = \frac{\pi}{2} \sqrt{\frac{g E_x I_x}{W L^4}}$$

Where W/g = mass per unit length of composite

I_x is the moment of inertia of a thinned walled tube, and is given as

$$I_x = \pi r^3 t$$

The natural frequency specification is likely to be a significant design driver and is likely to require that fibers be placed in the axial direction to achieve sufficient axial stiffness. Usually these axial fibers must be relatively stiff.

V. RESULTS & DISCUSSIONS

The applied torque, composite properties and stress values are expressed in the

Maximum torque applied	=2030 N-m
Max Twist For carbon/epoxy shaft	= 0.12° (under static condition)
For glass/epoxy shaft	= 0.25° (under static condition)
Maximum stress in x-dir	= 187MPa
Maximum stress in y-dir	= 120MPa

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Maximum stress in z-dir = 120MPa
Maximum shear stress in layers

Layer No.	Shear Stress - Carbon/Epoxy		Shear Stress - Glass/Epoxy	
	ANSYS	CADEC	ANSYS	CADEC
1 - Top	13.03	11.2	15.05	14
1 - Bot	12.6	10.2	14.65	13.3
2 - Top	0.131	0.172	-0.058	-0.681
2 - Bot	0.127	0.181	-0.232	-0.873
3 - Top	-0.139	-0.16	0.051	0.513
3 - Bot	-0.132	-0.172	0.13	0.681
4 - Top	6.91	6.52	7.79	7.38
4 - Bot	6.78	6.21	7.63	7
5 - Top	6.832	6.87	8.03	8.05
5 - Bot	6.64	6.26	7.91	7.95
6 - Top	7.01	7.47	8.184	8.15
6 - Bot	6.882	6.87	8.03	8.05
7 - Top	7.39	7.84	8.24	8.92
7 - Bot	7.317	7.64	8.178	8.68
8 - Top	-0.186	0.018	-0.033	-0.631
8 - Bot	-0.181	0.09	0.0056	-0.422
9 - Top	0.191	0.072	0.106	0.747
9 - Bot	0.186	0.081	0.035	0.631
10 - Top	15.3	18	17.39	18.9
10 - Bot	14.75	16.4	16.87	17.9

Table 1: Shear Stresses in each fiber (top – bottom)

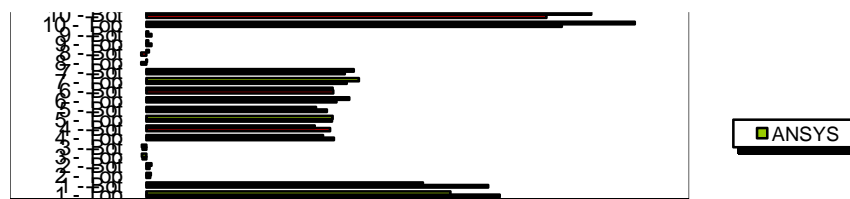


Fig. 3 Comparison of stresses (ANSYS- CADEC)

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FREE VIBRATION ANALYSIS

Free vibration analysis was done on the composite propeller shaft to find the fundamental and derived natural frequencies of the shaft and their respective mode shapes. No load is applied on the shaft except boundary conditions by assuming the shaft as pinned-pinned shaft. The fundamental and derived natural frequencies of the propeller shaft are shown in the Table 2.

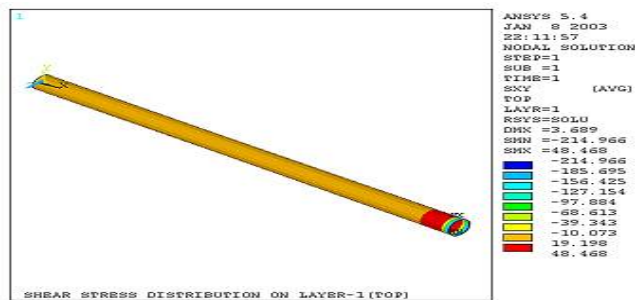


Fig. 4 Comparison of stresses

Mode number	Frequency (Hz)-Glass	Frequency (Hz)-Carbon
1	7.0421	11.342
2	7.0421	11.342
3	19.041	30.772
4	19.041	30.772
5	36.425	59.100
6	36.425	59.100
7	56.939	95.312
8	58.484	95.312
9	58.484	96.982
10	62.005	105.85

Table 2: Results of modal analysis for carbon & glass shafts

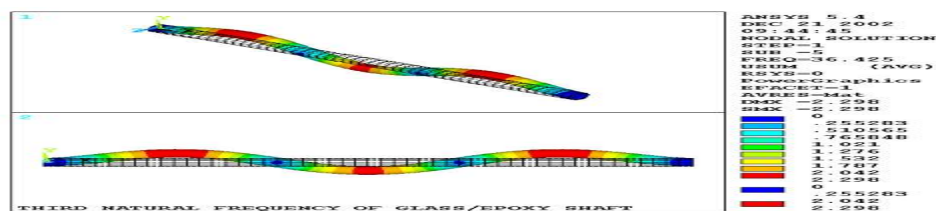


Fig 4: Third frequency mode shape

BUCKLING ANALYSIS

Advanced composite materials seem ideally suited for long, power driver 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. For thin walled shafts, the failure mode under an applied torque is torsional buckling rather than material failure. On the other hand the rotational speed is limited by lateral stability considerations. Most designs are subcritical, i.e. rotational speed must be lower than the first natural bending frequency of the shaft.

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Material	First	Second	Third
Carbon/Epoxy	12045	13870	17283
Glass/Epoxy	6729.5	7502.2	8871.2
Steel	10160	26640	30551

Table 3: Results of torsion buckling analysis for carbon, glass and steel shafts

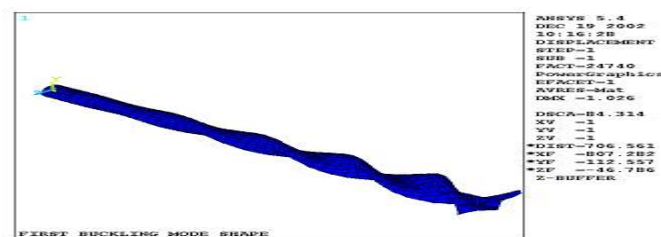


Fig 5: First buckling mode shape at the load of 12045 N-m

In the present work an attempt was made to develop composite propeller shaft, which can work as a viable alternative to the conventional steel propeller shaft that was used in automobile transmission systems. The performance of the composite prop shaft was found to be good and ride comfort with this propeller shaft is more when compared with the conventional driveshaft. The stresses developed in the shaft under static and dynamic conditions are found to be within limits. The results from FEA with different thickness of layered elements were compared with those of calculated values. The optimum fiber thickness for each layer was found to take torsional as well as bending loads also up to some extent. The total weight of the composite propeller shaft is 3.2064kg for carbon/epoxy shaft and 3.579kg for glass/epoxy shaft. The maximum shear stress developed under static conditions is 35.56MPa with a factor of safety 2.81 for carbon shaft and 31.673Mpa with a factor of safety 2.36.

VI. CONCLUSION

- The static deflection of the carbon propeller shaft is 0.12° , for the glass shaft is 0.25° , and for hybrid shaft 0.2° which are less than conventional steel propeller shaft.
- The fundamental natural frequency of the carbon-epoxy propeller shaft is lesser than glass shaft and steel shafts.
- The stresses and displacement amplitudes of propeller shafts are reaching maximum values at the frequencies, which are far away from the operating frequency range.
- The torsional buckling load is higher (nearly 5 times) than the ultimate torque transmission by the shaft, so it is safer.

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