Design and analysis of Composite Drive Shaft for Rear-Wheel Drive Engine

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***Abstract*—In current market, drive shaft is the most important component to any power transmission application; automotive drive Shaft is one of this. A drive shaft, also known as a propeller shaft or Cardan shaft, it is a mechanical part that transmits the torque generated by a vehicle's engine into usable motive force to propel the vehicle. Physically, it is tubular in design, with an outside and inside diameter, which spins at a frequency governed by engine output. Drive shaft must operate in high and low power transmission of the fluctuating load. Due this fluctuating load it becomes fail and tends to stop power transmission. Thus it is important to make and design this shaft as per load requirement to avoid failure. Now a day's two pieces steel shaft are mostly used as a drive shaft. 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.**

However, in this project work an attempt is made to evaluate the suitability of composite material such as E-Glass/Epoxy for the purpose of automotive drive shaft application. A one-piece composite shaft is optimally analyzed using Finite Element Analysis Software for E-Glass/Epoxy composites with the objective of minimizing the weight of the shaft, which is subjected to the constraints such as torque transmission, critical buckling torque capacity and bending natural frequency.

Keywords—**Drive shaft, Torsional buckling, Bending natural frequency**

I. Introduction

A driveshaft is a rotating shaft that transmits power from the engine to the differential gear of a rear wheel drive vehicles [1]. Driveshaft must operate through constantly changing angles between the transmission and axle. High quality steel is a common material for construction. 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. The two piece steel drive shaft consists of three universal joints, a center supporting bearing and a bracket, which increase the total weight of a vehicle.

Power transmission can be improved through the reduction of inertial mass and light weight. Substituting composite structures for conventional metallic structures has many advantages because of higher specific stiffness and higher specific strength of composite materials. Composite materials can be tailored to efficiently meet the design requirements of strength, stiffness and composite drive shafts weight less than steel or aluminum

II. Design Constraints

A. Design Specifications

The primary load carried by the drive shaft is torsion. The shaft must be designed to have enough torsional strength to carry the torque without failure. In addition, the possibility of torsional buckling must be considered for a thin-walled tube. The third major design requirement is that the drive shaft has a bending natural frequency which is sufficiently high. An optimum design of the drive shaft is desirable, which is cheapest and lightest but meets all of the above load requirements. Based on some reliable collected data the above three load-carrying requirements are summarized in Table 1.

TABLE I.

LOAD REQUIREMENTS FOR DRIVE SHAFT DESIGN

Regular	Values	Safety Factor
Maximum torque	2020	3
Minimum buckling torque	>2020 N-m	
Minimum flxural frequency	93.3	

The physical dimensions of the test pieces shaft to be designed are assumed as given in Table 2.The type of material selected for the design is E- Glass/epoxy, the thickness of each lamina is considered as 1.5mm. The properties used for this lamina were taken from PROMAL [5].

TABLE II.

PHYSICAL DIMENSIONS FOR DRIVE SHAFT DESIGN

Test piece	d_o	L	d_i

I	75	1000	25
II	75	950	25
III	70	1000	25

d_o-outer diameter, d_i-inner diameter, L-length of the shaft

The mechanical properties of the E-Glass/Epoxy are given as in table 3.

TABLE III.

MECHANICAL PROPERTIES OF E-GLASS/EPOXY COMPOSITE

Properties	Value	Unit
E_x (Longitudinal elastic modulus)	50	GPa
E_y (Transverse elastic modulus)	12	GPa
G_{xy} (Modulus of rigidity)	5.6	GPa
N (Poisson's ratio)	0.3	-
σ_x (working stress in x-direction)	800	MPa
σ_y (working stress in y-direction)	40	MPa
τ_{xy} (shear stress)	72	MPa
ρ (density of the material)	2000	Kg/m^3
Ply thickness	1.5	mm

B. Assumptions

The following are the assumption that can be used for designing the drive shaft. It rotates at a constant speed about its longitudinal axis. The shaft has a uniform, circular cross section. The shaft is perfectly balanced, i.e., at every cross section, the mass center coincides with the geometric center. All damping and nonlinear effects are excluded. The stress-strain relationship for composite material is linear & elastic; hence, Hook's law is applicable for composite materials. Since lamina is thin, so it is considered as under the plane stress.

C. Design Considerations

In order to design the drive shaft, three conditions can be considered; Torque conditions, Connection considerations and, Manufacturer's options. Since the primary load carried by the drive shaft is torsion. The shaft must be designed to have enough torsional strength to carry the torque without failure. Thus in this project work, the three test piece (specimen) should be able to withstand at least 2020Nm torque with a factor of safety 1.5. The torque can be obtained from engine and vehicle. Regarding second consideration, the torque test machine is specially used to test drive shaft; so it requires a flanged connection to fix the test specimen. Besides these the composite material manufacturer option provides a drive shaft having inner diameter of 25mm.

III. MATHEMATICAL FORMULAE

The shearing stress(τ_{xy}) is defined and given by Equation (1)

$$\tau_{xy} = \frac{TR}{J} = \frac{2TR}{\pi\left(R^4 - r^4\right)} \dots\dots\dots\dots\dots (1)$$

Where T is the applied torque and R is the outer radius and r is the inner radius of the shaft, J is polar moment of inertia of the shaft.

The shear stress and yield stress is related using equation (2)

$$\tau_{xy} = \frac{\sigma_y}{2 \times FOS} \dots\dots\dots\dots\dots\dots\dots\dots (2)$$

Where σ_y is yield stress, and FOS is factor of safety

The critical torsional buckling torque, T_b is given by Equation (3)

$$T_b = (2\pi r_m^2 t)(0.272)\left(E_x \times E_y^3\right)^{0.25} \left(\frac{t}{r_m}\right)^{1.5} \dots\dots (3)$$

Where t is the overall wall thickness, r_m is the mean radius, and E_x and E_y are the average in-plane elastic moduli in the axial and transverse directions respectively.

The drive shaft is idealized as a pinned-pinned beam. The lowest natural frequency is calculated using the Equation (4).

$$f_n = \frac{\pi}{2}\sqrt{\frac{gE_xI}{WL^4}} = \frac{\pi}{2}\sqrt{\frac{E_xI}{mL^4}} \dots\dots\dots\dots\dots\dots\dots . (4)$$

Where, f_n is the lowest natural frequency in hertz. $W/g = m$ is the mass per unit length, I, is the moment of inertia and L is the length of the drive shaft.

The critical speed of the shaft (N) and natural frequency (f_n) are related by using equation (5)

$$N = 60 f_n \dots\dots\dots\dots\dots\dots\dots\dots (5)$$

IV. DESCRIPTION OF THE ANALYSIS METHOD

Using equation (1) torque value is 2020Nm, and substituting the other dimensional parameters, the shear stress for each test specimen I, II, and III become 24.69 MPa, 24.69MPa, and 30.49 MPa respectively. While according to the maximum shear theory, maximum shear stress occurs at the outer surface. This maximum shear stress is defined as with respect to yield stress and factor of safety, and given by equation (2) above, thus the yield stress for each test specimen become 74.07MPa for specimen I and II, and for test specimen III it is about 91.47MPa.

Since the drive shaft is hollow and long, there is a possibility for it to buckle. To avoid this buckling, shaft could not be thin. This requirement asks that the applied torque should be less than the critical buckling torque. The critical buckling torque is given by equation (3) and the critical buckling torque for each test specimen can be calculated and the result become 12856 Nm for test specimen I and II, and 9148Nm for test specimen III. From this analysis we understand that, the value of critical torque buckling capacity of each test specimen is greater than the applied external torque (2020Nm), thus each of them do not buckle. In addition to torque and buckling analysis the design of drive shaft should include a critical frequency well above the car operating conditions. If the drive shaft were to turn at its natural frequency, it could vibrate severely and possibly

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disintegrate. To reduce the damage and severe vibration, the fundamental flexural frequency of the shaft should be determined. The drive shaft is designed to have a critical speed (60 times the frequency), that is high enough to exceed the rotational speed. If both are become coincident a large amplitude vibration (whirling) occurs. Thus the lowest natural frequency expression is given by Timoshenko beam theory as shown in equation (4)
To use the equation mass per unit length of the shaft as well as polar moment of inertia could be found, this is given in table IV.

TABLE IV.

CALCULATED MASS AND POLAR MOMENT OF INERTIA FOR EACH SHAFT TEST PIECE

Test piece	J (m^4)	m (kg/m)
I	3.068E-6	7.854
II	3.068E-6	7.854
III	2.319E-6	6.715

Therefore minimum bending natural frequency for test specimen I, II, and III become 204.29 Hz, 226.36 Hz, and 192.08 Hz.

Critical speed is the speed at which a spinning shaft will become unstable. This is one of the single largest factors in driveshaft selection. When the whirling frequency and the natural frequency coincide, any vibrations will be multiplied, So that the shaft may be destruct. In other words if a shaft naturally vibrates at 130 times a second, and one point on the shaft passes through 0 degrees 130 times a second (7800 RPM) then the shaft has hit a critical speed. There are several ways to raise the critical speed of a driveshaft. These ways are by making it lighter, stiffer, or increase diameter without increasing weight. This is the reason for composite fiber makes a good driveshaft; it is stiff and light and can be made to any diameter or wall thickness. Thus as per equation (5) the critical speed of the drive shaft I, II, and III becomes 12257 RPM, 13582 RPM, and 11520 RPM respectively. All are higher than cars operating speed which is equal to 5600 RPM (93.3 Hz). Therefore the result shows that all the three drive shafts will be safely operated under their maximum RPM.

V. MASS SAVING CALCULATION

The amount of mass that can be saved by utilizing composite materials as compared to steel drive shaft with the same dimensions can be given by:

$$\%mass\ saving = \frac{Steel\ mass - composite\ mass}{steel\ mass}$$

Thus using this formula the mass of the steel drive shaft and composite test specimen I, II, and III are calculated, and the results are 12.25kg, 3.142kg, 2.984kg and 2.686kg respectively. Therefore percentage of mass saving by using composite drive shaft is calculated as 74.35%, 75.65% and 78.08% for specimen I, II, and III respectively. By using specimen III large amount of mass can be saved.

VI. FINITE ELEMENT ANALYSIS (FEA)

In FEA the composite shaft tube can be analyzed using 2-D shell or 3-D solid elements available in standard FEA solvers. The FEA software available in CATIA was used to model the entire of the composite shaft, generally anisotropic material was used to model the composite shaft. The shaft is fixed at one end in all degree of freedoms and is subjected to torsion at the other end. After performing a static analysis of the shaft, the displacements are saved. The material properties were given to the FEA modeller in the form of a general anisotropic stiffness/compliance matrix.

A. Finite Element Analysis Results

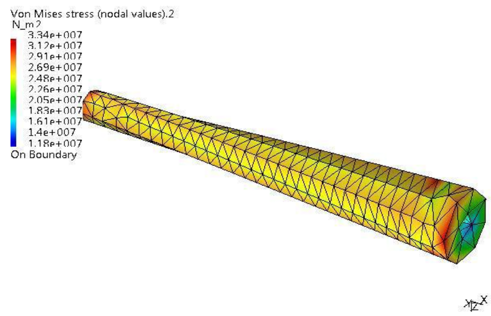

Figure 1.Von Mises stress Nodal value for Test Specimen I

This shows FEA for test specimen I, there is a maximum Von Mises stress of 33.4MPa at the outer surface which is marked as red, while at the inner surface there is less stress marked blue, it is about 11.8Mpa.

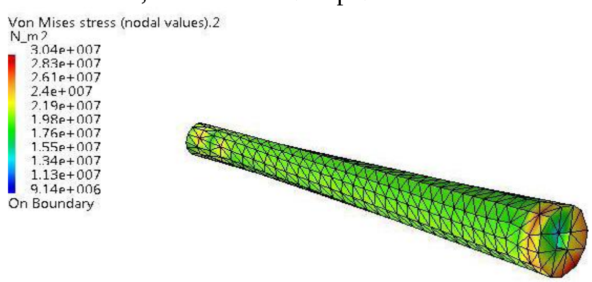

Figure 2.Von Mises stress Nodal value for Test Specimen II

This FEA result shows there is a maximum Von Mises stress of 30MPa at the outer surface which is marked as red, while at the inner surface there is less stress marked blue, it is about 9.34Mpa.

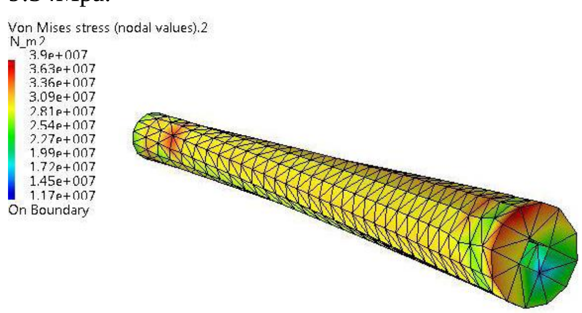

Figure3. Von Mises stress Nodal value for Test Specimen III
This shows FEA results for test specimen III, there is a maximum Von Mises stress of 39MPa at the outer surface which is marked as red, while at the inner surface there is less stress marked blue, it is about 11.8Mpa.

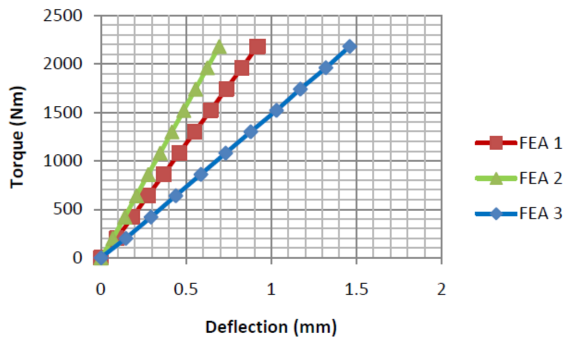

Figure 4.Torque versus deflection for Test specimen I, II, and III

All the above results are showing a linear increase in deflection with increasing torque. 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.

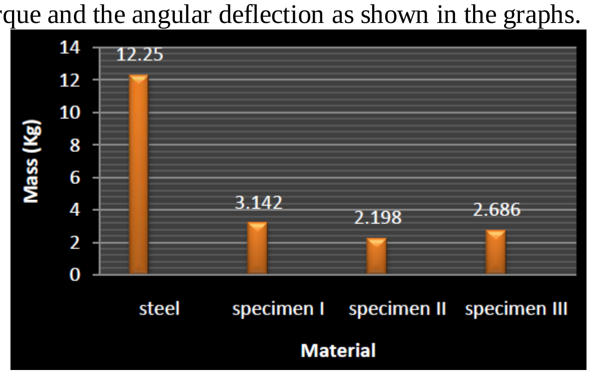

Figure 4.shows the amount of mass that can be saved by using Composite material.

VII. CONCLUSIONS

Composite shaft made from E-Glass/ Epoxy is investigated theoretically and numerically by taking torsion load. To analyse the shaft using FEA the composite shaft is fixed from one end and the torque is applied to the other end. For numerical study, the test specimens are modeled using CATIA software and analysed with FEA. After applying boundary conditions and torque, the torsional deflections are obtained for each torque value. List of the results are graphed. The result shows there is a linear relationship between torque and deflection, between torque and stress, also between torque and strain. The least deflection is observed in Test Specimen II (0.694mm), this is due to smallest length with biggest diameter, where Test Specimen I become stiffer than Test Specimen III (1.46mm), this is because of larger diameter while both having the same length. In conclusion, when the Finite Element Analysis results compared with the theoretical results, the observations and study carried on the topic is successful and yield very small variations from the expected results.

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