Experimental Investigation of an Automotive Hybrid Aluminum/Composite Drive Shaft

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***Abstract*- By adding composites, materials have many advantages over the conventional metallic materials such as higher specific stiffness and higher specific strength. In this paper an experimental investigation of an automotive hybrid aluminum/composite drive shaft is done.**

Ply orientation 20^0, $\pm 45^0$ and $90^0$is considereddue to their higher advantages such as, ply orientation$\pm 45^0$is increases the torsional strength/stiffness, 90^0 ply increase the critical torsional buckling load, and the 20^0 plies increase the natural frequency of the drive shaft.

It is found that by using hybridmaterial increases natural frequency of the shaft and reduced the weight of the shaft over the metallic material.

***Keywords* - Hybrid Shaft, Composites, Torsional strength, Critical Speed.**

I. INTRODUCTION

The drive shaft is used in an automobile to transmit power from the engine to the differential gear of a wheel drive vehicle. The advanced composite material such as Graphite, Graphite, Kevlar and Glass with suitable resins is widely used because of theirhigh specificstrength and modulus. The drive shafts are used in an aircraft, aerospace and the automobile application. The weight reduction of the drive shaft is an important role in the weight saving of the vehicle and is a highly desired goal, if it can be achieved without increase in price and decrease in quality and reliability. The torque competence of the drive shaft for passenger cars should be higher than 3500 N-m and the fundamental bending natural frequency should be greater than 9200 rpm to avoid spinning vibration. The bending natural frequency of a one piece drive shafts made of aluminum or steel is normally reduced than 5700 rpm when the length of the drive shaft is around 1.5 m, then the steel drive shaft is generally manufactured in two pieces to maximize the necessary bending natural frequency

Dai Gil Lee et al. (2003) [1]studied by replacing composite structures for conventional metallic structures has many advantages because of higher stiffness and greater specific strength of composite materials. From experimental results, it was obtained that the developed one-piece automotive hybrid aluminium/composite drive shaft had 75% mass decrease, 160% maximize in torque capability compared with a two piece steel drive shaft.

M. A. Badie et al. (2010) [2]examinedthat the effect of fibre orientation angles and stacking order on the buckling strength, natural frequency, fatigue life,torsional stiffness failure modes and torsional stiffness of composite tubes. Finite element analysis has been used to forecast the fatigue life of composite drive shaft by using linear dynamic analysis for different stacking order. A finite element analysis result shows that the natural frequency maximize with minimizingfibre orientation angles. The reduction in a composite drive shaftis equal to 54.2%.

Y. A. Khalid et al. (2005) [4] studied that, the hybrid shafts is manufactured by using filament winding technique. Glass fibre with a matrix of epoxy resin and hardener were used to make the external composite layers needed. Throughout this experimental study, a bending fatigue analysis conducted for hybrid aluminum/composite drive shafts. Four cases were learnt using aluminum tube warped by different layers of the composite materials and different stacking sequence or fibre orientation angles. Results obtained from this study shows that increasing the number of layers would progress the fatigue strength of aluminum tube up to 40%, for $[\pm 45]_{3s}$.

Durk Hyun Cho et al. (1997)[5]explainedthe natural bending frequency of a torque transmission capability of the shaft can be increased by without decreasingthe torque transmission capability, if the shaft is made using both carbon fibre composite and aluminium, the former increases the natural bending frequency and the latter sustains the applied torque. From the dynamic test, it was found that first natural bending frequency and the least static torque transmission capability of the hybrid drive shaft were 9000 rpm and 3500 Nm, respectively, and the shaft did not fail until 10^7 cycles under a dynamic load.

O. Montagnieret et al. (2012) [6]study deals with the optimization of hybrid composite drive shafts operating at subcritical speeds, using a genetic algorithm. A formulation for the flexural vibrations of a composite drive shaft mounted on viscoelastic supports including the shear effects is developed. The solutions obtained using the method presented here made it possible to greatly reduced the number of shafts and the weight of the driveline under subcritical conditions, and even more under high critical conditions.

II. DESIGN SPECIFICATION

The following specifications are consideredwhich are based on the literature and available standards for the automobile drive shaftis

1) Torque transmission capacity of the steel driveshaft= 3676 N-m
2) Speed of drive shaft = 2400 rpm

3) Outside diameter of drive shaft = 0.068 m
4) Length of drive shaft = 1.3 m

III. DESIGN OF CONVENTIONAL STEEL DRIVE SHAFT

The conventional steel drive shaft should satisfy three design specifications such as buckling torque capability,torque transmission capability and bending natural frequency. Steel (SM45C) used for automotive drive shaft applications. The mechanical material properties of the steel (SM45C) are given in table-1

Table-1: Mechanical Properties of the Steel

Mechanical Properties	Symbol	Steel (SM45C)	Unit
Young's Modulus	E	207	GPa
Shear Modulus	G	80	GPa
Poisson's Ratio	μ	0.3	-
Density	ρ	7800	Kg/m^3
Yield Strength	S_y	370	MPa

A. Design of Steel Drive Shaft Based on Torsional Strength

The primary load in the steel drive shaft is torsion. The maximum shear stress (τ_{max}), in the drive shaft is at the outer radius and is given as,

$$\tau_{max} = \frac{T_{max}\, r_0}{J} \quad (1)$$

We know, polar moment of inertia of the hollow circular shaft in cross section is given by,

$$J = \frac{\pi}{32}(d_0^4 - d_i^4)$$

Therefore,

$$\frac{0.5\, S_y}{FOS} = \frac{16\, T_{max}}{\pi\, d_0^3 (1 - C^4)}$$

$$C = \frac{d_i}{d_0}$$

$$d_i = 0.05254 \text{ m}$$

B. Mass of Steel Drive Shaft

$$m = \rho\, A\, L \quad (2)$$

Where,
m = mass of steel drive shaft (kg)
ρ = material density (kg/m^3)
L = length of the shaft (m)

$$A = \frac{\pi}{4}(d_0^2 - d_i^2)$$

Therefore,

$$m = \rho \times \frac{\pi}{4}(d_0^2 - d_i^2) \times L$$

$$m = 14.84 \text{ kg}$$

C. Torsional Buckling Capacity of the Steel Drive Shaft

The torsional buckling capacity of the steel drive shaft given by

$$T_b = (2\,\pi\, r_m^2\, t)(0.272)\,(E)\left(\frac{t}{r_m}\right)^{\frac{3}{2}} \quad (3)$$

Where,
T_b= buckling torque (N-m)
r_m = mean radius of shaft (m)
E = young's modulus (GPa)
t = thickness of steel drive shaft (m)

$$T_b = 322.60 \times 10^3 \text{ N-m}$$

The value of critical torsional buckling moment is larger than the applied torque of 3676×10^3 N-m. Thus the shaft need to withstand torsional buckling (T_b) capacity such that T_b >T. Hence the condition is satisfied.

D. Natural Frequency of Steel Drive shaft

Natural frequencies can be found by using two theories

i. Timoshenko Beam Theory.
ii. Bernoulli-Euler Beam Theory.

Out of these two theories, as per assumptions Bernoulli-Euler Beam Theory is more suitable in this case.
The first natural frequency is given by

$$f_{nt_1} = \frac{\pi\, P^2}{2L^2}\sqrt{\frac{E\, I_x}{m_1}} \quad (4)$$

Where,
f_{nt_1} =first natural frequency (Hz)
m_1 =mass per unit length (kg/m^3)

$$f_{nt_1} = 103.61 \text{ Hz}$$

This value is greater than the minimum desired natural frequency of 40 Hz. Thus, the steel design of hollow dive shaft of outer diameter 0.068m and thickness 0.00773 m is an acceptable design.

E. Critical speed

The first critical speed is given by

$$N_{crt_1} = 60\, f_{nt_1} \quad (5)$$

Where,
N_{crt_1} =first critical speed (rpm)

$$N_{crt_1} = 6168.62 \text{ rpm}$$

The values of various parameters of steel drive shaft are shown in table 2.

Table 2: Analytical Results of the Steel Drive Shaft

Sr. No	Parameters	Symbol	Steel Drive Shaft
1	Applied Torque	T	3676 N-m
2	Length	L	1.3 m
3	Outer Diameter	d_0	0.0068 m
4	Inner Diameter	d_i	0.05254 m
5	Thickness	t	0.00773 m
6	Mass	m	14.84 kg
7	Torsional Buckling Capacity	T_b	322.60×10^3 N-m
8	Natural Frequency	f_{nb}	103.6 Hz
9	Critical Speed	N_{crt}	6168.62 rpm

IV. DESIGN OF COMPOSITE DRIVE SHAFT

Classical lamination theory is used for design of composite drive shaft.

A. Selection of Material

1) Carbon Fiber

2) Epoxy Resin
3) Aluminum Material

In this project work the design of drive shaft by a combination of aluminum and composite materials carbon fibers/epoxy resin. The major role of the aluminum tube is to sustain a applied torque while the role of the carbon fiber epoxy composite is to increase natural bending frequency.

B. Macro mechanical Analysis of Laminate

Macro mechanical analysis will be developed for a laminate. A real structure however will not consist of a one lamina but a laminate consisting of more than one lamina bonded together through their thickness. Based on in-plane loads of extension, shear, bending and torsion, stresses and strains will be found in the local and global axes of each ply. Stiffness's of whole laminates will also be calculated. Intuitively, one can see that the strength, stiffness properties of laminate will depend on

1) Elastic moduli.
2) Stacking sequence.
3) Thickness.
4) Angle of orientation of each lamina.

C. Composite Ply Orientation

Only 20^0, $\pm 45^0$ and 90^0 were consider for the ply orientations, due to their greater advantages, such as $\pm 45^0$ plies increases the torsional strength/stiffness, and 90^0 plies increase the critical torsional buckling load, and the 20^0 plies increase the natural frequency of the drive shaft.

D. The Stiffness Matrices [A], [B] and [D] for Eight-ply

$[90^0/45^0/-45^0/20^0]$ *Carbon/ Epoxy Laminate*

The extensional stiffness matrix [A] is given by,

$$A_{ij} = \sum_{k=1}^{8} \left[\overline{Q}_{ij}\right]_k (h_k - h_{k-1}) \qquad (6)$$

Where,

A_{ij} = extensional stiffness matrix

h_{k-1} = top surface

h_k = bottom surface

Longitudinal modulus

$$E_x = 47.16 \text{ GPa}$$

Transverse modulus

$$E_y = 57.10 \text{ GPa}$$

The Bending stiffness matrix [D] is given by,

$$D_{ij} = \frac{1}{3}\sum_{k=1}^{8} \left[\overline{Q}_{ij}\right]_k \left({h_k}^3 - {h_{k-1}}^3 \right) \qquad (7)$$

$$E_x = 21.58 \text{ GPa}$$

$$E_y = 106 \text{ GPa}$$

E. Torsional Strength of Shaft

Assuming that the drive shaft is a thin hollow cylinder, an element in the cylinder can be assumed to be flat laminate. The applied torque then is

$$T = \left(\tau_{xy}\right)_{average} \pi \left(r_0^2 - r_i^2\right) r_m \qquad (8)$$

The average shear stress is given by,

$$N_{xy} = \left(\tau_{xy}\right)_{average} \times t \qquad (9)$$

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

$$\left(\tau_{xy}\right)_{average} = 100.23 \text{ MPa}$$

Therefore,

$$T = 3895.06 \text{ N-m}$$

F. Torsional Buckling Capacity of Composite Shaft

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

$$T_c = (2\,\pi\, r_m^2\, t)(0.272)(E_x\, E_x^3)^{\frac{1}{4}} \left(\frac{t}{r_m}\right)^{\frac{3}{2}} \qquad (10)$$

T_c= 69293.63 N-m

The value is greater than the applied torque of 3676 N-m, thus the composite drive shaft is safe in buckling.

G. Torsional Buckling Capacity of Aluminum Tube

The torsional buckling capacity of aluminium tube is given by,

$$T_{(Buckling)al} = \frac{\pi\sqrt{2}\, E_{al}}{3\left(1 - \mu_{avg}^2\right)^{0.75}} \sqrt{r_{avg} t_{al}^5} \qquad (11)$$

Where,

E_{al} = young's modulus of aluminum tube (GPa)

t_{al} = thickness of aluminum tube (m)

r_{avg}= average radius of aluminum tube (m)

$$T_{(Buckling)al} = 3857.31 \text{ N-m}$$

H. Torque Transmitted by the Hybrid Drive Shaft

The torque transmitted by the hybrid drive shaft is given by,

$$T = T_{al} + T_c \qquad (12)$$

Where,

T_{al}= torque transmitted by aluminum tube (N-m)

T_c= torque transmitted by aluminum tube (N-m)

Therefore,

$$T = 73150.94 \text{ N-m}$$

This values is greater than the applied torque of 1923.2 N-m, thus the composite drive shaft is safe in buckling.

I. Fundamental Bending Natural Frequencies of the Hybrid Drive Shafts

The bending natural frequencies of the hybrid drive Shaft is given by,

$$f_n = \frac{9.869}{L^2}\sqrt{\frac{E_{al} I_{al} + E_c I_c}{m_{al} + m_c}} \qquad (13)$$

Where,

E_{al} = young's modulus of aluminum tube (GPa)

E_c= young's modulus of composite tube (GPa)

I_{al}= moment of inertia of aluminum tube (m^4)

I_c= moment of inertia of composite tube (m^4)

m_{al}= mass per unit length of aluminum tube (kg/m)

m_c= mass per unit length of composite tube (kg/m)

Therefore,

$$f_n = 691.79 \text{ Hz}$$

Thus the calculated frequency is more than the lowest natural frequency i.e. 80 Hz. Thusthe requirement is also meet by the hybrid drive shaft.

J. Mass Saving

The mass saving of the carbon/epoxy drive shaft is

$$m = \frac{14.84 - 3.95}{14.84} \times 100$$

$$m = 73.38\ \%$$

V. FINITE ELEMENT ANALYSIS

The finite element analysis is a computer based analysis technique for calculating the strength and performance of structures. In the finite element method the structure is represented as finite elements. The finite element analysis is used to calculate the strains temperature, deflection, stresses, buckling behavior of the component. In our project finite element analysis (FEA) is carried out by using the ANSYS 15.

A. Modal analysis

When an elastic system free from external forces is distributed from its equilibrium position it vibrates under the influence of originate forces and is explained to be in the state of free vibration. It will be vibrate at its natural frequency and its amplitude will slowly become lesser with time due to energy being dissipated by motion.The modal analysis is used to find out mode shapes andthe natural frequencies of a structure.

The figure1 shows the first natural frequency of hybrid drive shaft and the deformation value with the help of colour bar. Colour bar is used to determine the value ranges on object. First natural frequency of hybrid drive shaft drive shaft is 571.36 Hz which are within permissible limit.

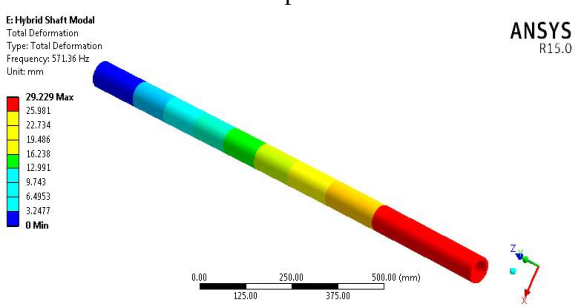

Figure1 First Natural frequency of hybrid drive shaft

Thefigure 2 shows the second natural frequency of hybrid drive shaft and thedeformation value with the help of colour bar. Colour bar is used to determine the value ranges on object. Second natural frequency of hybrid drive shaft drive shaft is 953.82 Hz which are within permissible limit.

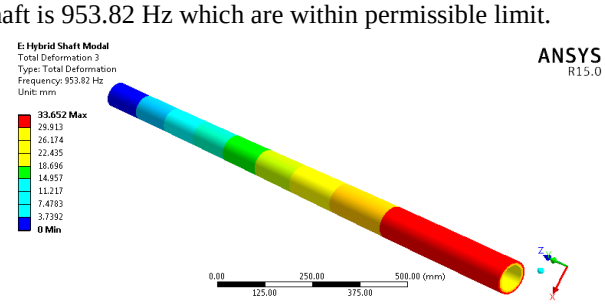

Figure2 Second Natural frequency of hybrid drive shaft

The figure 3 shows the third natural frequency of hybrid drive shaft and the deformation value with the help of colour bar. Colour bar is used to determine the value ranges on object. Third natural frequency of hybrid drive shaft drive shaft is 1801 Hz which are within permissible limit.

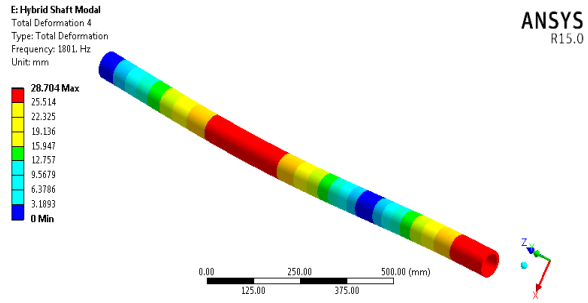

Figure 3Third Natural frequency of hybrid drive shaft

Comparison results of hybrid and steel drive shaft by using FEA are shown in table 3.

Table 3: FEA Results

Analysis Type	Hybrid drive shaft	Steel drive shaft
	Modal Analysis	
1st Frequency	571.36 Hz	103.17 Hz
2nd Frequency	971.36 Hz	409.68 Hz
3rd Frequency	1801.0 Hz	921.53 Hz

Modal analysis of Hybrid drive shaft is done by using Ansys 15.0 software. By using modal analysis mode shapes of the steel and hybrid drive shaft are calculated

VI. EXPERIMENTATION

A. Torsion Test Setup

Generally, torsion occurs when the torque is applied or twisting momentto a member according to figure4. The torque is the product of the tangential force is multiplied by the radial distance from the twisting axis and the tangent, measured in a unit of N-m. The torsion testing, the relationship between degree of rotation and torque is presented by the graph and parameters such as ultimate torsional shearing strength, shear strength at proportional limit and shear modulus are generally investigated. Moreover, fracture faces of specimens tested under torsion can be used to decide the characteristics of the materials whether it would fail in a brittle or a ductile manner.

Figure4Torsion testing machine

B. Results

The data that was collected during the torsion testing lab was Torque (kg-m), and angle of twist.

1) Plot a graph of applied torque (T) against angle of twist (θ)as a base for the elastic region
2) Plot a graph of applied torque(T) against angle of twist (θ) of the specimen as a base, for the complete test to destruction.

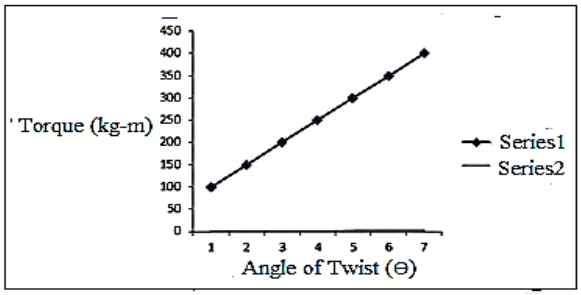

Figure5 Graph of Torque Vs Angle of Twist

The above graph shows that linear increase in deflection with increasing torque. In this torsion testing we apply torque up to 4415 N-m (450 kg-m) on the hybrid drive shaft. Here we want to transmit the torque up to 3676 N-m and the shaft is slightly twisted at 3950 N-m. This is an expected output and it checks with the theoretical behavior of a shaft subjected to torsional loading. The complete linear correlation between the applied torque and the angular deflection as shown in graphs.

VII. CONCLUSION

The hybrid drive shaft is designed to replace the steel drive shaft used for an automobile.The weight saving of the carbon/Epoxy drive shaft is 73.38 % as compared to steel drive shaft. The layers stacking order has no effect on the natural frequency then there is no load applied.Carbon fibers have the large contribution over glass in increases the torsional stiffness. From the graph of torque vs. angle of twist, it has been observed that the shaft can sustain the torque upto 3676 N-m.

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