DESIGN AND ANALYSIS OF COMPOSITE DRIVE SHAFT

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ABSTRACT

Almost all automobiles which correspond to design with Rear wheel drive and front engine installation have transmission shafts. In heavy duty vehicles driveshaft is one of the important components. The weight reduction of the drive shaft can have a certain role in the general weight reduction of the vehicle and is a highly desirable goal, it can be achieved without increase in cost and decrease in quality and reliability. The investigation of this work is to replace the conventional steel driveshaft of automobiles with an appropriate composite driveshaft. The conventional drive shafts are made in two pieces for reducing the bending natural frequency, whereas the composite shafts can be made as single-piece shafts, thus reducing the overall weight. The design parameters were optimized with the objective of minimizing the weight of composite drive shaft. The composite drive shaft made up of high modulus material is designed by using CAD software and tested in ANSYS for optimization of design or material check and providing a best material. The replacement of composite materials can results in considerable amount of weight reduction if compared to conventional steel shaft.

***Keyword :**- CAD, ANSYS*, Graphite, Carbon, Kevlar, Glass and FEA

1. INTRODUCTION

The advanced composite materials such as Graphite, Carbon, Kevlar and Glass with suitable resins are widely used because of their high specific strength (strength/density) and high specific modulus (modulus/density). Advanced composite materials seem ideally suited for long, power driver shaft (propeller 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. The drive shafts are used in automotive, aircraft and aerospace applications. The automotive industry is exploiting composite material technology for structural components construction in order to obtain the reduction of the weight without decrease in vehicle quality and reliability. It is known that energy conservation is one of the most important objectives in vehicle design and reduction of weight is one of the most effective measures to obtain this result. Actually, there is almost a direct proportionality between the weight of a vehicle and its fuel consumption, particularly in city driving.

1.1 Drive Shaft

The term ‘Drive shaft’ is used to refer to a shaft, which is used for the transfer of motion from one point to another. In automotive, driveshaft is the connection between the transmission and the rear axle.

1.2 Purpose of the Drive Shaft

The torque that is produced from the engine and transmission must be transferred to the rear wheels to push the vehicle forward and reverse. The drive shaft must provide a smooth, uninterrupted flow of power to the axles. The drive shaft and differential are used to transfer this torque.

1.3 Drive Shaft arrangement in Automobile

Drive shaft arrangement in rear wheel drive vehicle is shown in figure below;

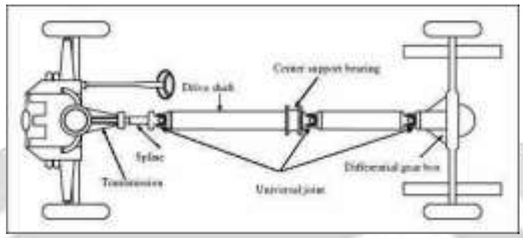


Fig-1: Two piece drive shaft arrangement in automobile

1.4 Different Types of Shafts

- Transmission shaft: used for transmit the power between power sources and machines.
- Machine shaft: used as an integral part in machine itself.
- Axle: are used for transmitting a bending moment only.
- Spindle: is used as a short shaft that imparts motion either to a cutting tool or to a work – piece.

1.5 Working Principle

Fig-2: Drive shaft in automobile

The torque that is produced from the engine and transmission must be transferred to the rear wheels to push the vehicle forward and reverse. The drive shaft must provide a smooth, uninterrupted flow of power to the axles. The drive shaft and differential are used to transfer this torque. First, it must transmit torque from the transmission to the differential gear box. During the operation, it is necessary to transmit maximum low-gear torque developed by the engine. The drive shafts must also be capable of rotating at the very fast speeds required by the vehicle. The drive shaft must also operate through constantly changing angles between the transmission, the differential and the axles. As the rear wheels roll over bumps in the road, the differential and axles move up and down. This movement changes the angle between the transmission and the differential. The length of the drive shaft must also be capable of changing while transmitting torque. Length changes are caused by axle movement due to torque reaction, road deflections, braking loads and so on. Alsip joint is used to compensate for this motion. The slip joint is usually made of an internal and external spline. It is located on the front end of the drive shaft and is connected to the transmission.

1.6 Analysis

1. Modeling of the High Strength Carbon/Epoxy composite drive shaft using SOLID WORKS.
2. Static, Modal and Buckling analysis are to be carried out on the High Strength Carbon/Epoxy composite drive shaft using SOLIDWORKS.
3. To calculate

a) Mass reduction when using the High Strength Carbon

b) The change in deformation, stress and strain values of composite drive shaft by varying thickness of shaft.

1.7 Functions of the Drive Shaft

1. First, it must transmit torque from the transmission to the differential gear box.
2. During the operation, it is necessary to transmit maximum low-gear torque developed by the engine.
3. The drive shafts must also be capable of rotating at the very fast speeds required by the vehicle.
4. The drive shaft must also operate through constantly changing angles between the transmission, the differential and the axles. As the rear wheels roll over bumps in the road, the differential and axles move up and down. This movement changes the angle between the transmission and the differential.
5. The length of the drive shaft must also be capable of changing while transmitting torque. Length changes are caused by axle movement due to torque reaction, road deflections, braking loads and so on. A slip joint is used to compensate for this motion.

1.8 Demerits of a Conventional Drive Shaft

1. They have less specific modulus and strength.
2. Increased weight.
3. Conventional 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. Therefore the steel drive shaft is made in two sections connected by a support structure, bearings and U-joints and hence over all weight of assembly will be more.
4. Its corrosion resistance is less as compared with composite materials.
5. Steel drive shafts have less damping capacity.

1.9 Merits of Composite Drive Shaft

1. They have high specific modulus and strength.
2. Reduced weight.
3. The fundamental natural frequency of carbon fiber composite drive shaft can be twice as high as that of steel because the carbon fiber composite material has more than 4 times the specific stiffness of steel, which makes it possible to manufacture the drive shaft of cars in one piece. A one piece composite shaft can be manufactured so as to satisfy the vibration requirements. This eliminates all the assembly, connecting the two piece steel shafts and thus minimizes the overall weight, vibrations and the total cost.
4. Due to the weight reduction, fuel consumption will be reduced.
5. They have high damping capacity hence they produce less vibration and noise.
6. They have good corrosion resistance.
7. Greater torque capacity than steel shaft.
8. Longer fatigue life than steel shaft.
9. Lower rotating weight transmits more of available power.

1.10. Drive Shaft Vibration

Vibration is the most common drive shaft problem. Small cars and short vans and trucks (LMV) are able to use a single drive shaft with a slip joint at the front end without experiencing any undue vibration. However, with vehicles of longer wheel base, the longer drive shaft required would tend to sag and under certain operating conditions would tend to whirl and then setup resonant vibrations in the body of the vehicle, which will cause the body to vibrate as the shaft whirls. Vibration can be either transverse or torsional. Transverse vibration is the result of unbalanced condition acting on the shaft. This condition is usually by dirt or foreign material on the shaft, and it can cause a rather noticeable vibration in the vehicle. Torsional vibration occurs from the power impulses of the engine or from improper universal join angles. It causes a noticeable sound disturbance and can cause a mechanical shaking. Whirling of a rotating shaft happens when the center of gravity of the shaft mass is eccentric and so is acted upon by a centrifugal force which tends to bend or bow the shaft so that it orbits about the shaft longitudinal axis like a rotating skipping rope. As the speed rises, the eccentric deflection of the shaft increases, with the result that the centrifugal force also will increase. The effect is therefore cumulative and will continue until the whirling become critical, at which point the shaft will vibrate violently. From the theory of whirling, it has been found that critical whirling speed of the shaft is inversely proportional to the square of shaft length. If, therefore, a shaft having, for example, a critical whirling speed of 6000 rev/min is doubled in length, the critical whirling of the new shaft will be reduced to a quarter of this, i.e. the shaft will now begin to rotate at 1500 rev/min. The vibration problem could solve by increasing diameter of shaft, but this would increase its strength beyond its torque carrying requirements and at the same time increase its inertia, which would oppose the vehicle's acceleration and deceleration. Another alternative solution frequently adopted by large vehicle manufacturers is the use of two-piece drive shafts supported by intermediate or center bearings. But this will increase the cost considerably.

1.11. Description of the Problem

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 drive shaft can have a certain role in the general weight reduction of the vehicle and is a highly desirable goal, if it can be achieved without increase in cost and decrease in quality and reliability. It is possible to achieve design of composite drive shaft with less weight to increase the first natural frequency of the shaft.

2. DESIGN OF STEEL DRIVE SHAFT

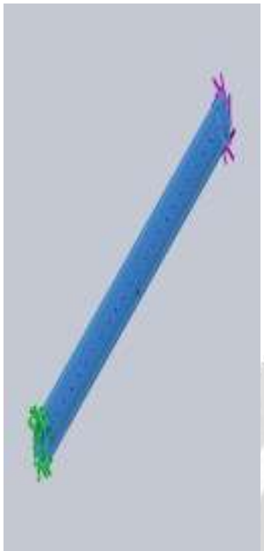
2.1. Specification of the Problem

The fundamental natural bending frequency for passenger cars, small trucks, and vans of the propeller shaft should be higher than 6,500 rpm to avoid whirling vibration and the torque transmission capability of the drive shaft should be larger than 3,500 Nm. The drive shaft outer diameter should not exceed 100 mm due to space limitations. Here outer diameter of the shaft is taken as 90 mm. The drive shaft of transmission system is to be designed optimally for following specified design requirements as shown in Table 3.1.

Table-1: Design requirements and specifications

S.No	Name	Notation	Unit	Value
1.	Ultimate Torque	T_{max}	Nm	3500
2.	Max. Speed of shaft	N_{max}	rpm	6500
3.	Length of shaft	L	mm	1250
4.	Outer Diameter	do	mm	90
5.	Thickness	t	mm	5

Table-2: Material Properties of structured steel

MODEL REFERENCE	PROP		COMPONENTS
	**Name:**	**Structural Steel**	SolidBody1(Extrude Thin1) (Part1), Shell1(SolidBody1 (ExtrudeThin1)) (Part1)
	Model type:	Linear Elastic Isotropic	
	Default failure	Max von Mises	
	criterion:	Stress	
	Yield strength:	6.20422e+00N/m^2	
	Tensile strength:	7.23826e+008N/m^2	
	Elastic modulus:	2.1e+011 N/m^2	
	Poisson's ratio:	0.28	
	Mass density:	7600 kg/m^3	
	Shear modulus:	7.9e+010 N/m^2	
	Thermal: expansion coefficient	1.3e-005 /Kelvin	

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

a) Mass of the steel drive shaft

m = ρAL

= ρ × Π/4 × (do2- di2) ×L … (1)

= 7600 x 3.14/4 x (90^2 – 80^2) x 1250

= 12.68 Kg

b) Torque transmission capacity of steel drive shaft

T = Ss× Π /16 × [(do4- di4) / do] … (2)

= 16.67 ×10^3 N-m

=24

c) Fundamental Natural frequency

The natural frequency can be found by using the two theories:

1) Timoshenko Beam theory

2) Bernoulli Euler Theory

Timoshenko Beam Theory –Ncrt

fnt= Ks (30 Π p^2) / L^2 X √ (Er^2 / 2ρ)…..(3)

Ncrt = 60 fnt"….. (4)

fnt= natural frequency base on Timoshenko beam theory, HZ

Ks = Shear coefficient of lateral natural frequency

p = 1, first natural frequency

r = mean radius of shaft

Fs = Shape factor, 2 for hollow circular cross section

n = no of ply thickness, 1 for steel shafts

1 / Ks^2 = 1 + (n^2*Π^2* r^2) / 2*L^2 * [1 + fs E / G] …. (5)

Ks = 0.986

Fnt = 0.986 (30 x Π x 1) / 1250^2 x √ (210x10^3 x 85/2x 7600)

Fnt = 317.3 Hz

Ncrt = 60*fnt

= 19038 rpm

d) Design of Composite Drive Shaft

The specifications for the composite drive shaft are same as that of steel drive.

No. of layers = 5

Thickness of each layer = 1mm

Stacking sequence = 0-45-90-45-0

e) Mass of the Composite drive shaft

m = ρAL

= ρ × Π / 4 × (do2- di2) × L … (1)

= 1600 x 3.14/4 x (90^2 – 80^2) x 1250

= 2.669 Kg

f) Assumptions

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 center coincides with the geometric center.

4. All damping and nonlinear effects are excluded.

5. The stress-strain relationship for composite material is linear & elastic; hence, 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 as under the plane stress.

g) Selection of Cross-Section

The drive 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.

2. 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.

h) Torsional buckling capacity

The long thin hollow shafts are vulnerable to torsional buckling; so the possibility of the torsional buckling of the composite shaft was calculated by the expression for the torsional buckling load T cr of a thin walled orthotropic tube:

$$T_{cr} = (2\pi r^2 t)(0.272)(E_x E_y^3)^{0.25}(t/r)^{1.5} \quad \text{…… (6)}$$

Where E x and E y are the Young's modulus of the shaft in axial and hoop direction, r and t are the mean radius and thicknesses of composite shaft.

i) Lateral Vibrations

Natural frequency of composite shaft is based on Timoshenko's beam theory,

$$f_{nt} = K_s*(30*\Pi*p^2/L^2)*(E_x r^2/2\rho)^{0.5} \quad \text{…….(7)}$$

$$1/K_s^2 = 1 + (p^2*\Pi^2*r^2/2L^2)(1 + f_s E_x/G_{xy}) \quad \text{…… (8)}$$

Where fnt and p are the natural and first natural frequency.

Ks is the shear coefficient of the natural frequency (< 1),

fs is a shape factor (equals to 2) for hollow circular cross-sections.

Critical speed:

$$N_{crt} = 60\, f_{nt}.$$

3. DESIGN 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 behavior of the member. In our project FEA is carried out by using the SOLISWORKS 12.0. Initially we don't know the displacement and other quantities like strains, stresses which are then calculated from nodal displacement.

3.1 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. 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.

3.2 Boundary conditions

The finite element model of HS Carbon / Epoxy shaft is shown in Figure .One end is fixed and torque is applied at other end.

Fig-4: Shows boundary condition of static analysis. Here right end is fixed and a torque of 3500 N is applied at left end

3.3 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. Modal analysis is used to determine the natural frequencies and mode shapes of a structure or a machine component. The rotational speed is limited by lateral stability considerations. 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.

3.4 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

4. RESULTS

4.1 Static analysis results

Fig-5: Steel Shaft-Static Study 1-Stress-Stress1

Fig.5 shows the stress values across the length of the shaft (dimensions given on table 3) for two different materials. Upper one is for steel shaft and below one is for carbon/epoxy shaft. Min. and max values of stresses are marked on the figures itself which are obtained at two different ends. Comparison shows that steel shaft have less stresses than carbon/epoxy shaft. From figure view steel shows higher stresses but this is due to the high deformation scale used in comparison to that of carbon/ epoxy steel shaft.

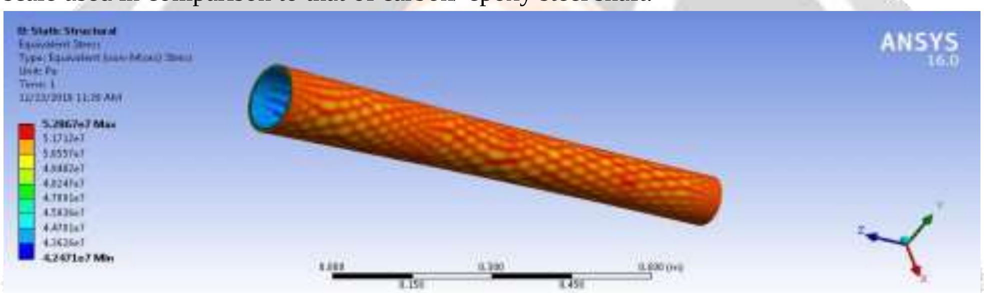


Fig-6: Equivent stress analysis

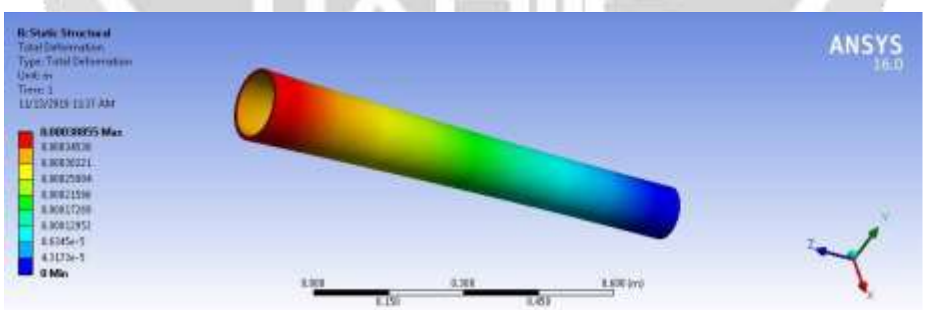


Fig-7: Steel Shaft-Static Study 1-Displacement-Displacement1

4.2 Buckling study results

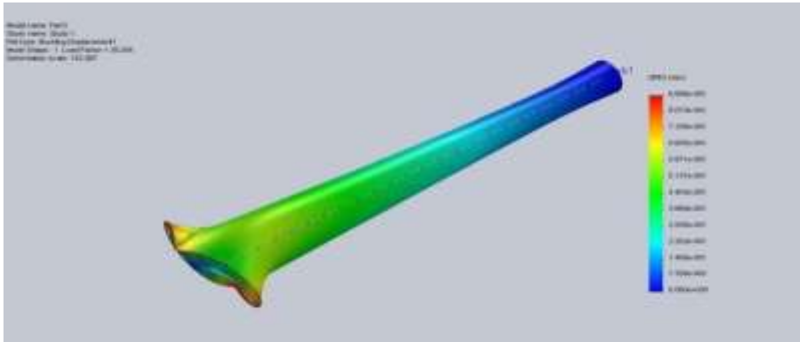


Fig-8: Steel Shaft-Buckling Study 1-Displacement-Displacement1

4.3 Frequency analysis results

Table-3: frequency values for steel shaft

Frequency Number	Rad/sec	Hertz	Seconds
1	1993.7	317.3	0.0031516
2	2207.2	351.28	0.0028467
3	5130.1	816.48	0.0012248

From table-3 we find out that first natural frequency occurring for carbon epoxy shaft is greater than that of steel shaft. And we also know that critical speed of shaft running in rpm is 60*first natural frequency in hz. So from the above data we find that carbon/epoxy shaft is capable of running at higher speeds than steel shaft. From above data

Ncrt = 60*345.88 (for carbon/epoxy shaft)

Which is much greater than the practical working speed i.e.3500rpm. So we may increase the length of the shaft to make it single piece shaft.

Carbon/epoxy shaft analysis at various thicknesses

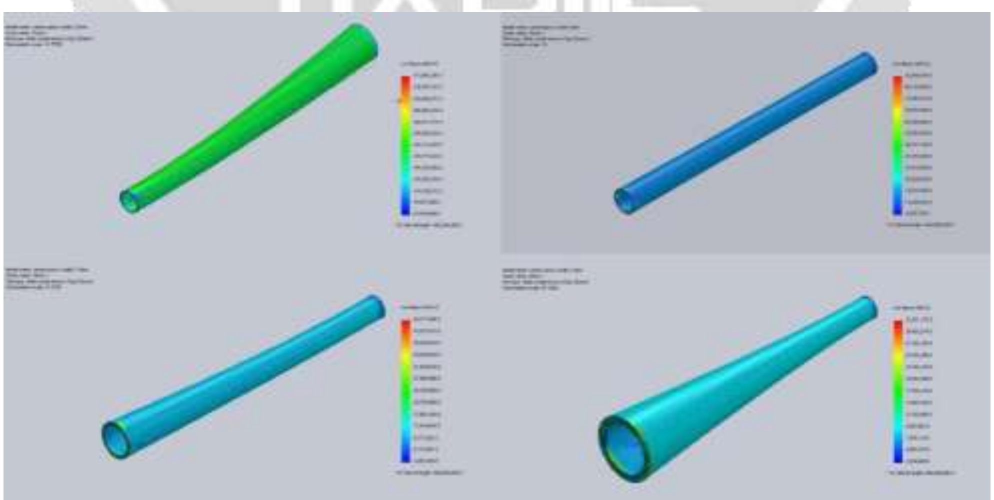


Fig-9: Stress analysis of different thickness shaft

Fig.9 shows the stress distribution along length of carbon/epoxy drive shaft which are of different thickness. a)

2.5mm thick shaft b) 5mm thick shaft c) 7.5mm thick shaft d) 10mm thick shaft. From all of above four figures we see that stress values goes on decreasing when we increase the thickness. Also for 2.5mm thick shaft stress value at fixed end i.e. 4.74*10^8N/mm2 crosses the yield strength (4.0*10^8). So 2.5 mm thick shaft fails with given dimensions so on increasing the thickness by not changing the other dimensions we find out that 4mm thick shaft is satisfying the design criterion. Also on increasing greater thickness we find we have a high stress tolerable shaft.

4.4 Dynamic analysis results

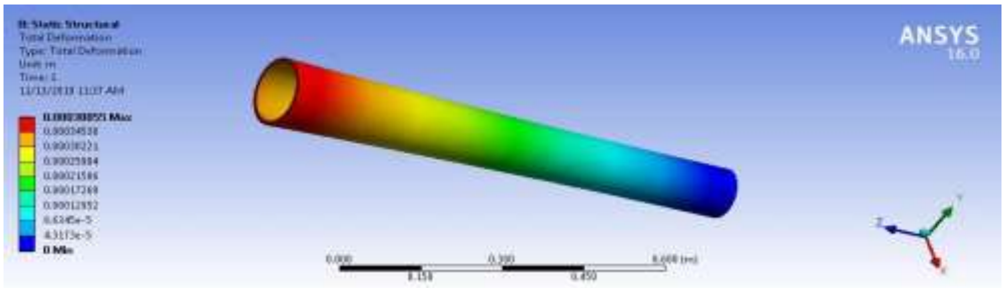


Fig-10: Steel Shaft- Static Study 1-Displacement-Displacement1

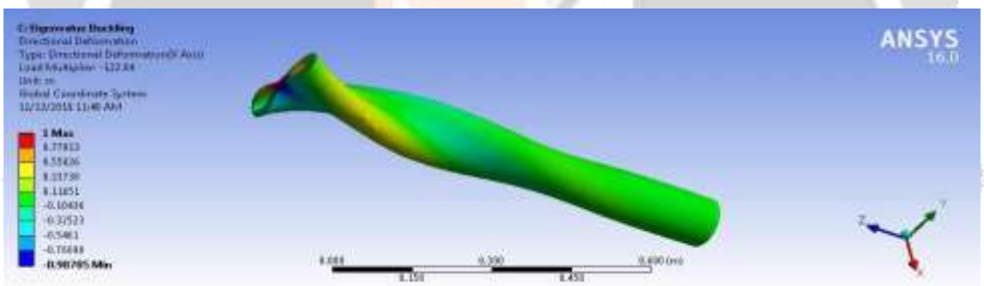


Fig-11: Steel Shaft- Buckling Study 1-Displacement-Displacement1

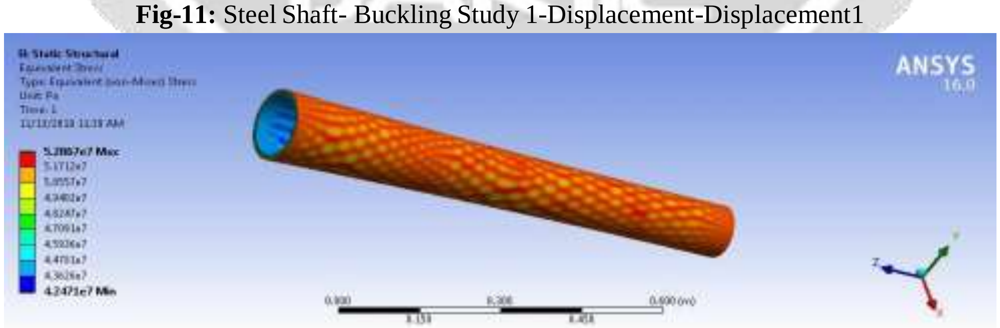


Fig-12: Steel Shaft-Static Study 1-Stress-Stress1

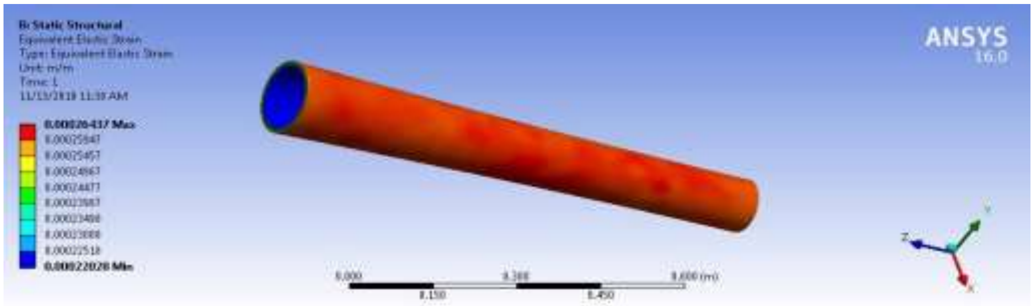


Fig-13: Steel Shaft-Static Study 1-Strain-Strain1

Table-4: Comparison of stress, strain and displacement values of carbon epoxy drive shaft with different thickness

Thickness / Properties	2.5mm	5.0mm	7.5mm	10mm
Max.stress(N/m^2)	4.74*10^8 (fails)	9.24*10^7	4.62*10^7	3.23*10^7
Min.stress(N/m^2)	5.44*10^7	4.95*10^6	2.48*10^6	2.30*10^6
Max. strain	0.00253	0.00134	0.000940	0.000734
Min. strain	0	0	0	0
Max. displacement (mm	12.80	2.52804	1.92540	1.43469
Min. displacement(mm)	0	0	0	0
Weight (kg)	1.374	2.669	3.887	5.026
% Wt. Reduction	89.16	78.95	69.34	60.36
No. of layers	5	5	5	5

5. FINITE ELEMENT ANALYSIS

5.1 Introduction

Finite Element Analysis (FEA) is a computer-based numerical technique for calculating the strength and behaviour of engineering structures. It can be used to calculate deflection, stress, vibration, buckling behaviour and many other phenomena. It also can be used to analyze either small or large scale deflection under loading or applied displacement. It uses a numerical technique called the finite element method (FEM). In finite element method, the actual continuum is represented by the finite elements. These elements are considered to be joined at specified joints called nodes or nodal points. As the actual variation of the field variable (like displacement, temperature and pressure or velocity) inside the continuum is not known, the variation of the field variable inside a finite element is approximated by a simple function. The approximating functions are also called as interpolation models and are defined in terms of field variable at the nodes. When the equilibrium equations for the whole continuum are known, the unknowns will be the nodal values of the field variable.

In this report finite element analysis was carried out using the FEA software ANSYS. The primary unknowns in this structural analysis are displacements and other quantities, such as strains, Stresses, and reaction forces, are then derived from the nodal displacements.

5.2 Modelling Linear Layered Shells

SHELL99 may be used for layered applications of a structural shell model as shown in Fig 15 SHELL99 allows up to 250 layers. The element has six degrees of freedom at each node: translations in the nodal x, y, and z directions and rotations about the nodal x, y, and z-axes.

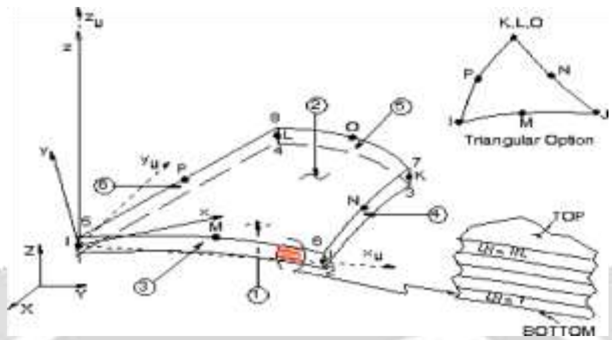


Fig-14: SHELL99 Linear Layered Structural Shell

5.3 Input Data

The element is defined by eight nodes, average or corner layer thicknesses, layer material direction angles, and orthotropic material properties. A triangular-shaped element may be formed by defining the same node number for nodes K, L and O. The input may be either in matrix form or layer form, depending upon KEYOPT (2). Briefly, the force-strain and moment- curvature relationships defining the matrices for a linear variation of strain through the thickness (KEYOPT (2) = 2) may be defined as:

$$\begin{Bmatrix} N \\ M \end{Bmatrix} = \begin{bmatrix} A & B \\ B & D \end{bmatrix} \begin{Bmatrix} \varepsilon \\ \kappa \end{Bmatrix} - \begin{Bmatrix} MT \\ BT \end{Bmatrix}$$

$$\underset{6\times6}{[A]} = \begin{bmatrix} A_1 & A_2 & A_3 & A_4 & A_5 & A_6 \\ A_2 & A_7 & A_8 & A_9 & A_{10} & A_{11} \\ A_3 & A_8 & A_{12} & A_{13} & A_{14} & A_{15} \\ A_4 & A_9 & A_{13} & A_{16} & A_{17} & A_{18} \\ A_5 & A_{10} & A_{14} & A_{17} & A_{19} & A_{20} \\ A_6 & A_{11} & A_{15} & A_{18} & A_{20} & A_{21} \end{bmatrix} \quad \text{or} \quad \underset{3\times3}{[A]} = \begin{bmatrix} A_1 & A_2 & A_3 \\ A_2 & A_4 & A_5 \\ A_3 & A_5 & A_6 \end{bmatrix}$$

Sub matrices [B] and [D] are input similarly. Note that all sub matrices are symmetric. {MT} and {BT} are for thermal effects. The layer number (LN) can range from 1 to 250. In this local right-handed system, the x'-axis is rotated an angle THETA (LN) (in degrees) from the element x-axis toward the element y-axis. The total number of layers must be specified (NL). The properties of all layers should be entered (LSYM = 0). If the properties of the layers are symmetrical about the mid-thickness of the element (LSYM = 1), only half of properties of the layers, up to and Including the middle layer (if any), need to be entered. While all layers may be printed, two layers may be specifically selected to be output (LP1 and LP2, with LP1 usually less than LP2).

The results of GA forms input to the FEA. Here Finite Element Analysis is done on the HS Carbon/Epoxy drive shaft.

5.3 Static Analysis

Static analysis deals with the conditions of equilibrium of the bodies acted upon by forces. A static analysis can be either linear or non-linear. All types of non-line arties are allowed such as large deformations, plasticity, creep, stress stiffening, contact elements etc. this chapter focuses on static analysis. A static analysis calculates the effects of steady loading conditions on a structure, while ignoring inertia and damping effects, such as those carried by time varying loads. 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

Static analysis result of structural displacements, stresses and strains and forces in structures for components caused by loads will give a clear idea about whether the structure or components will withstand for the applied maximum forces. 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.

5.4 Boundary Conditions

The finite element model of HS Carbon/Epoxy shaft is shown in Figure 15 One end is fixed and torque is applied at other end,

Fig-15: Finite element model of HS Carbon/Epoxy shaft

5.5 Modal Analysis

When an elastic system free from external forces is disturbed from its equilibrium position it 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. It can also be a starting point for another more detailed analysis such as a transient dynamic analysis, a harmonic response analysis or a spectrum analysis. Modal analysis is used to determine the natural frequencies and mode shapes of a structure or a machine component.

The rotational speed is limited by lateral stability considerations. 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. Boundary conditions for the modal analysis are shown in Fig 15

5.6 Buckling Analysis

Buckling analysis is a technique used to determine buckling loads (critical loads) at which a structure becomes unstable, and buckled mode shapes (The characteristic shape associated with a structure's buckled response).

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. The dominant failure mode, torsional buckling, is strongly dependent on fibre orientation angles and ply stacking sequence.

5.7 Types of Buckling Analysis

Two techniques are available in ANSYS for predicting the buckling load and buckling mode shape of a structure. They are,

- Nonlinear buckling analysis and
- Eigenvalue (or linear) buckling analysis.

5.7.1 Nonlinear Buckling Analysis

Nonlinear buckling analysis is usually the more accurate approach and is therefore recommended for design or evaluation of actual structures. This technique employs a nonlinear static analysis with gradually increasing loads to seek the load level at which your structure becomes unstable. Using the nonlinear technique, model will include features such as initial imperfections, plastic behaviour, gaps, and large- deflection response.

5.7.1 Eigenvalue Buckling Analysis

Eigenvalue buckling analysis predicts the theoretical buckling strength (the bifurcation point) of an ideal linear elastic structure. This method corresponds to the textbook approach to elastic buckling analysis: for instance, an Eigenvalue buckling analysis of a column will match the classical Euler solution. However, imperfections and nonlinearities prevent most real- world structures from achieving their theoretical elastic buckling strength. Thus, Eigenvalue-buckling analysis often yields conservative results, and should generally not be used in actual day-to-day engineering analyses.

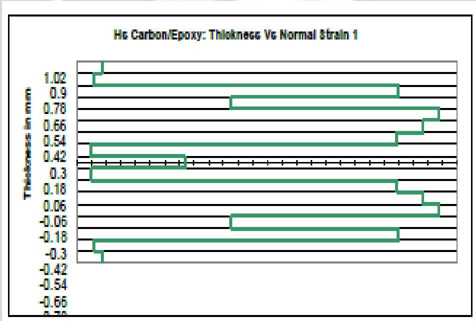


Fig-16: Variation of £**1** through thickness of HS Carbon/Epoxy Drive Shaft

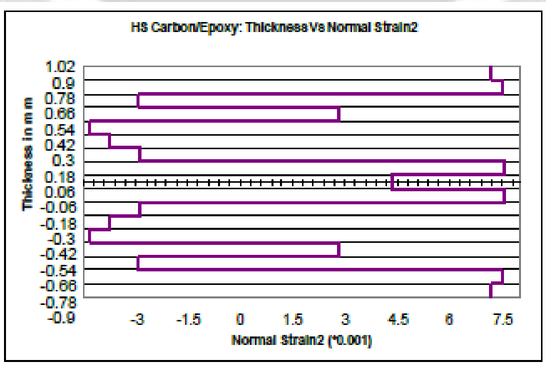


Fig-17: Variation of £**2** through thickness of HS Carbon/Epoxy Drive Shaft

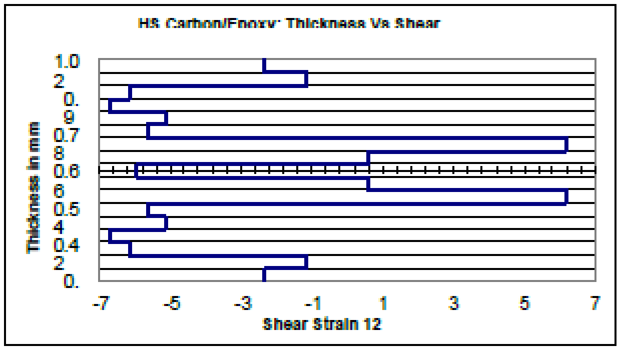


Fig-18: Variation of γ**12** through thickness of HS Carbon/Epoxy Drive Shaft

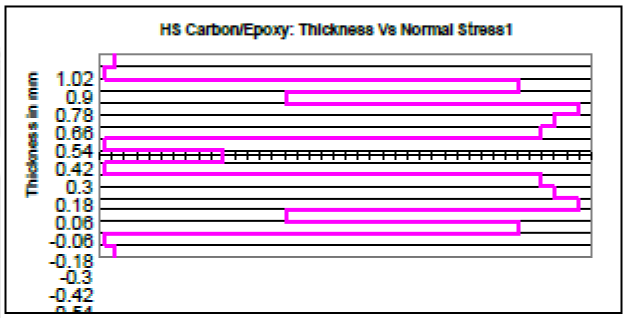


Fig-19: Variation of σ**1** through thickness of HS Carbon/Epoxy Drive Shaft

6. CONCLUSION

The replacement of conventional drive shaft results in reduction in weight of automobile. The finite element analysis is used in this work to predict the deformation of shaft. The deflection of steel, Glass Epoxy / HS Carbon and HM Carbon / Epoxy shafts was 298.296, 311.945 and 397.189 mm respectively. Natural frequency using Bernoulli – Euler and Timoshenko beam theories was compared. The frequency calculated by Bernoulli – Euler theory is high because it neglects the effect of rotary inertia & transverse shear. The FEA analysis is done to validate the analytical calculations of the work. We analysis composite material will continue on phase two. As the report is to reduce the weight and find strength of the drive shaft. The major sources used for this purpose are composite materials. By using three different kind of composite materials steel, carbon epoxy, E-glass epoxy the result has been carried out. Shaft is analyzed using layer stacking method in Abacus software which utilizes finite element method technologies. These layer stacking techniques are employed for shafts with and without binder material. Static analysis is done for observing the steady loading conditions. The results have shown that the shaft made of composite material has high strength when compared steel material. On the basis of stress calculation, report says that composite material shaft is advisable to replace conventional material shaft.

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