DESIGN AND ANALYSIS OF DRIVE SHAFT FOR OPTIMUM MATERIAL SELECTION USED IN AUTOMOBILE APPLICATION

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Abstract—Automotive drive shaft is typically fabricated in two pieces in order to increase the fundamental bending natural frequency because it is inversely proportional to the square of beam length and proportional to the square root of specific modulus. Many exploration work have been done to toward this path to replace two pieces drive shaft with single piece made of composites. This makes the single piece hollow shaft subjected to torsional load instability which is more critical in the design of composite shafts. In this specific situation, extensive ways to deal with examine the new composite drive shafts material is discovered to be essential. In this paper an attempt has been made to check the suitability of one piece composite drive shaft with various composite material combinations to satisfy the functional requirements. Firstly, a finite element model of composite drive shaft made of Steel, Glass/epoxy, Graphite/epoxy and Carbon/epoxy Composite is developed to analyze for static, modal & buckling analysis using ANSYS. From the results obtained, it is observed that composites are having better shear strength, bending natural frequency, and less weight compared to steel and Carbon/epoxy has good buckling strength capability as compared with other composites. These Finite element analysis results are compared with analytical values and MATLAB coding observed that the single piece composite drive shaft is better suitable for driveline applications.

Keywords— Composite Drive Shaft, Finite element analysis, Shear Strength, Buckling Load, natural frequency, MATLAB, ANSYS

I. INTRODUCTION

Drive shafts in cars are generally made of steel. An automobile manufacturer is seriously thinking of changing the material to a composite material. The reasons for changing the material to composite materials are that composites
1. Reduce the weight of the drive shaft and thus reduce energy consumption
2. Are fatigue resistant and thus have a long life
3. Are noncorrosive and thus reduce maintenance costs and increase life of the drive shaft
4. Allow single piece manufacturing and thus reduce manufacturing cost

II. DESCRIPTION OF THE PROBLEM

Practically all automobiles have transmission shafts. The weight decrease of the drive shaft can have a specific part in the overall weight decrease of the vehicle and is an exceptionally alluring
objective, in the event that it very well may be accomplished without increment in cost and diminishing in quality and reliability. It is conceivable to accomplish plan of composite drive shaft with less weight to increase the first natural frequency of the shaft and to decrease the bending stresses.

III. DESIGN OF SHAFT

3.1 Design Considerations for a Shaft
Based on the engine overload torque of 115 N-m, the drive shaft needs to withstand a torque of 3500 N-m. The shaft needs to withstand torsional buckling. The shaft has a minimum bending natural frequency of at least 80 Hz. Outside radius of drive shaft = 50 mm. Length of drive shaft is 1450 mm. Factor of safety = 3. Only \([0/90/0/45/90/-45]\), Stacking Sequence can be used.

3.1.1 Steel Shaft Design
For steel, use the following properties:
Young’s modulus \(E = 210\) GPa, Poisson’s ratio \(\nu = 0.3\), Density of steel \(\rho = 7800\) kg/m\(^3\)
Ultimate shear strength \(\tau_{ult} = 80\) MPa.

**Torsional strength:** The primary load in the drive shaft is torsion. The maximum shear stress, \(\tau_{Max}\) in the drive shaft is at the outer radius, \(r_o\), and is given as
\[
\tau_{Max} = \frac{T r_o}{J}
\]
where \(T = \) maximum torque applied in drive shaft (N-m) \(; r_o = \) outer radius of shaft (m); \(J = \) polar moment of area (m\(^4\))
\[
r_i = 0.03794 m
\]
Therefore, the thickness of the steel shaft is
\[
t = 12.06 \text{ mm}
\]

**Torsional buckling:** This requirement asks that the applied torsion be less than the critical torsional buckling moment. For a thin, hollow cylinder made of isotropic materials, the critical buckling torsion, \(T_b\) is given by
\[
T_b = \frac{2\pi r_m^2 t (0.272)(t/r_m)^{3/2}}{0.272}\]
where \(r_m = \) mean radius of the shaft (m) \(; t = \) wall thickness of the drive shaft (m) \(; E = \) Young’s modulus (Pa)
Using the thickness \(t = 3\) mm calculated in criterion (1) and the mean radius
\[
r_m = \frac{(r_0 + r_i)}{2} = 0.04397 m
\]
\[
T_b = (2\pi)(0.04397)^2(0.012)(0.272)(210 * 10^9)(0.012/0.04397)^{3/2}
\]
\[
= 1187133.10 \text{ N-m}
\]
The value of critical torsional buckling moment is larger than the applied torque of 3500 N-m.

**Natural frequency:** The lowest natural frequency for a rotating shaft is given by
\[
f_n = \frac{\pi}{2} \sqrt{\frac{E I}{m L^2}}
\]
Now the second moment of area, \(I\), is
\[
I = \frac{\pi}{4}(r_0^4 - r_i^4)
\]
\[
I = 3.28 * 10^{-6} \text{ m}^4
\]
The mass per unit length of the shaft is
\[
m = \pi (r_0^2 - r_i^2) \rho
\]
\[
m = 25.98 \text{ kg/m.}
\]
This value is greater than the minimum desired natural frequency of 80 Hz. Thus, the steel design of a hollow shaft of outer radius 50 mm and thickness $t = 3$ mm is an acceptable design.

### 3.1.2 Glass/epoxy, Graphite/epoxy, Carbon/epoxy Shaft Design

**Table 1** properties of composite materials

<table>
<thead>
<tr>
<th>Properties</th>
<th>Symbol</th>
<th>Glass/epoxy</th>
<th>Graphite/epoxy</th>
<th>Carbon/epoxy</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Longitudinal elastic modulus</td>
<td>$E_1$</td>
<td>38.6</td>
<td>181</td>
<td>172.7</td>
<td>GPa</td>
</tr>
<tr>
<td>Transverse elastic modulus</td>
<td>$E_2$</td>
<td>8.27</td>
<td>10.3</td>
<td>7.2</td>
<td>GPa</td>
</tr>
<tr>
<td>Major Poisson’s ratio</td>
<td>$\nu_{12}$</td>
<td>0.26</td>
<td>0.28</td>
<td>0.3</td>
<td></td>
</tr>
<tr>
<td>Shear modulus</td>
<td>$G_{12}$</td>
<td>4.14</td>
<td>7.17</td>
<td>3.76</td>
<td>GPa</td>
</tr>
<tr>
<td>Density</td>
<td>$\rho$</td>
<td>2000</td>
<td>2260</td>
<td>1600</td>
<td>kg/m³</td>
</tr>
</tbody>
</table>

**Torsional strength**: The maximum shear stress of the shaft will be calculated by,

$$\tau_{\text{max}} = \frac{T}{(2\pi r_m^2 t)}$$

**Torsional buckling**: An orthotropic thin hollow cylinder will buckle torsionally if the applied torque is greater than the critical torsional buckling load given by

$$T_b = \frac{(2\pi r_m^2 t)(0.272)(E_x*E_y)^{3/4}}{(t/r_m)^{3/2}}$$

where, $E_x =$ Young’s modulus in x direction ; $E_y =$ Young’s modulus in y direction

Here, we considered the composite driveshaft as orthotropic lamina. So, for orthotropic lamina, the longitudinal elastic modulus will be calculated by following formula

$$\frac{1}{E_x} = \cos^4 \theta \frac{1}{E_1} + (1/G_{12} - (2\theta_{12}/E)) \sin^2 \theta \frac{\cos^2 \theta}{E_2}$$

$$\frac{1}{E_y} = \sin^4 \theta \frac{1}{E_1} + (1/G_{12} - (2\theta_{12}/E)) \sin^2 \theta \frac{\cos^2 \theta}{E_2}$$

The values of $E_x$ and $E_y$ will be found out with the help of $E_1, E_2, G_{12}$ & $\theta_{12}$ values. The value of $\theta$ i.e. Stacking sequence angle will be taken so that the torsional buckling strength will be more than that of the maximum torque applied.

$E_x = 23.435$ GPa ; $E_y = 23.435$ GPa

$$T_b = 2\pi(0.048625)(0.00275)(0.272)\left[(23.435*10^9)(23.435*10^9)\right]^{1/4}(0.00275/0.048625)^{3/2}$$

$$= 3502.47 \text{ N-m}$$

The value of critical torsional buckling moment is larger than the applied torque of 3500 N-m.

**Natural frequency**: The bending natural frequency of the shaft is given by.

$$f_n = \frac{\pi \sqrt{\frac{Edx}{ml^4}}}{2}$$

The mass per unit length of the shaft is

$$m = \pi (ro^2 - ri^2) \rho$$

Therefore,

$$f_n = 88.77 \text{ Hz}$$

This value is greater than the minimum desired natural frequency of 80 Hz. Thus, the Glass epoxy design of a hollow shaft of outer radius 50 mm and thickness $t = 2.75$ mm is an acceptable design.

Similarly calculate for carbon and graphite epoxy

### 3.1.5 Mass Savings
The percentage of mass saving by replacing steel shaft with composite material is shown in table 3.1

<table>
<thead>
<tr>
<th>S.No</th>
<th>Material</th>
<th>Mass Per Unit Length(Kg/m)</th>
<th>Mass (Kg)</th>
<th>Percentage Savings</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Steel</td>
<td>25.98</td>
<td>37.67</td>
<td>-</td>
</tr>
<tr>
<td>2</td>
<td>Glass epoxy</td>
<td>1.65</td>
<td>2.4</td>
<td>93.62</td>
</tr>
<tr>
<td>3</td>
<td>Graphite epoxy</td>
<td>1.90</td>
<td>2.75</td>
<td>92.70</td>
</tr>
<tr>
<td>4</td>
<td>Carbon epoxy</td>
<td>1.344</td>
<td>1.95</td>
<td>94.82</td>
</tr>
</tbody>
</table>

Table 2 Percentage of mass savings

3.2 Theoretical Results

<table>
<thead>
<tr>
<th>S.No</th>
<th>Material</th>
<th>Outer Diameter (mm)</th>
<th>Inner Diameter (mm)</th>
<th>Thickness (mm)</th>
<th>Length (mm)</th>
<th>Torsional Buckling (N-m)</th>
<th>Natural Frequency (Hz)</th>
<th>Mass (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Steel</td>
<td>50</td>
<td>37.94</td>
<td>3</td>
<td>1450</td>
<td>1187133.1</td>
<td>121.657</td>
<td>37.67</td>
</tr>
<tr>
<td>2</td>
<td>Glass epoxy</td>
<td>50</td>
<td>47.25</td>
<td>2.75</td>
<td>1450</td>
<td>3502.47</td>
<td>88.77</td>
<td>2.4</td>
</tr>
<tr>
<td>3</td>
<td>Graphite epoxy</td>
<td>50</td>
<td>47.25</td>
<td>2.75</td>
<td>1450</td>
<td>14295.3</td>
<td>167.12</td>
<td>2.75</td>
</tr>
<tr>
<td>4</td>
<td>Carbon epoxy</td>
<td>50</td>
<td>47.25</td>
<td>2.75</td>
<td>1450</td>
<td>13443.47</td>
<td>192.6</td>
<td>1.95</td>
</tr>
</tbody>
</table>

Table 3 Theoretical results

IV. DESIGN OF SHAFT USING MATLAB

4.1 Design of Shaft Using MATLAB

The design process is very time consuming and made some calculation errors also in this a generalized MAT Lab code is written to design shaft.

<table>
<thead>
<tr>
<th>S.No</th>
<th>Material</th>
<th>Mass Per Unit Length(Kg/m)</th>
<th>Mass (Kg)</th>
<th>Percentage Savings</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Steel</td>
<td>25.9867</td>
<td>37.6807</td>
<td>-</td>
</tr>
<tr>
<td>2</td>
<td>Glass epoxy</td>
<td>1.6804</td>
<td>2.4366</td>
<td>93.5336</td>
</tr>
<tr>
<td>3</td>
<td>Graphite epoxy</td>
<td>1.8988</td>
<td>2.7533</td>
<td>92.6932</td>
</tr>
<tr>
<td>4</td>
<td>Carbon epoxy</td>
<td>1.3443</td>
<td>1.9492</td>
<td>94.8270</td>
</tr>
</tbody>
</table>

Table 4 Percentage of mass savings using MATLAB Code

4.2 MATLAB Results

<table>
<thead>
<tr>
<th>S.No</th>
<th>Material</th>
<th>Outer Diameter (mm)</th>
<th>Inner Diameter (mm)</th>
<th>Thickness (mm)</th>
<th>Torsional Buckling (N-m)</th>
<th>Natural Frequency (Hz)</th>
<th>Mass (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Steel</td>
<td>50</td>
<td>37.90</td>
<td>12.1</td>
<td>1.2018e+06</td>
<td>121.657</td>
<td>37.680</td>
</tr>
<tr>
<td>2</td>
<td>Glass epoxy</td>
<td>50</td>
<td>47.30</td>
<td>2.75</td>
<td>3.5025e+03</td>
<td>87.9670</td>
<td>2.4366</td>
</tr>
<tr>
<td>3</td>
<td>Graphite epoxy</td>
<td>50</td>
<td>47.30</td>
<td>2.75</td>
<td>1.4295e+04</td>
<td>167.1823</td>
<td>2.7533</td>
</tr>
<tr>
<td>4</td>
<td>Carbon</td>
<td>50</td>
<td>47.30</td>
<td>2.75</td>
<td>1.3443e+04</td>
<td>192.6827</td>
<td>1.9492</td>
</tr>
</tbody>
</table>
Table 5 MATLAB Results

V. ANALYSIS OF SHAFT

5.1 Static Structural Analysis of Steel Shaft
Apply boundary conditions by considering the shaft is simply supported and apply a twisting moment of 3500N-m. The Total displacement, Max stress and max shear stress is find out the results are shown below

(i) Steel shaft (i) Equivalent Stress, (ii) Total Deformation, (iii) Max Shear stress

5.1.1 Static Structural Analysis of Glass epoxy Shaft
Modeling groups with stacking sequence [0/90/0/45/90/-45]. Total 11 ply obtained with thickness 2.75mm. Now import this modal to Static Structural analysis apply Boundary Conditions Find out Total Displacement, Equivalent Stress And Maximum Shear stress

(i) Glass epoxy shaft (i) Equivalent Stress, (ii) Total Deformation, (iii) Max Shear stress

5.1.2 Static Structural Analysis of Graphite epoxy Shaft

(i) Graphite epoxy shaft (i) Equivalent Stress, (ii) Total Deformation, (iii) Max Shear stress
5.1.3 Static Structural Analysis of Carbon epoxy Shaft

(i) Carbon epoxy shaft (ii) Equivalent Stress, (iii) Total Deformation, (iv) Max Shear stress

5.3 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). The total deformation find out by buckling analysis is shown in following figures

(i) Steel (ii) Glass epoxy (iii) Graphite epoxy (iv) Carbon epoxy

Total deformation by buckling analysis (i) Steel (ii) Glass epoxy (iii) Graphite epoxy (iv) Carbon epoxy

5.4 Modal Analysis The main parameters of interest in free vibration are natural frequency and the amplitude.
Total deformation by modal analysis (i) Steel (ii) Glass epoxy (iii) Graphite epoxy (iv) Carbon epoxy

VI. RESULTS AND DISCUSSION

6.1 Static Structural analysis of shaft

The results that is equivalent stress, displacement and maximum shear stress obtained in the static structural analysis is shown in following table 6.1

<table>
<thead>
<tr>
<th>S.No</th>
<th>Material</th>
<th>Equivalent Stress (Mpa)</th>
<th>Total Displacment(mm)</th>
<th>Maximum Shear Stress(Mpa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Steel</td>
<td>77.682</td>
<td>0.16279</td>
<td>44.85</td>
</tr>
<tr>
<td>2</td>
<td>Glass epoxy</td>
<td>107.36</td>
<td>2.412</td>
<td>57.414</td>
</tr>
<tr>
<td>3</td>
<td>Graphite epoxy</td>
<td>134.49</td>
<td>0.87991</td>
<td>67.658</td>
</tr>
<tr>
<td>4</td>
<td>Carbon epoxy</td>
<td>126.15</td>
<td>1.2121</td>
<td>64.152</td>
</tr>
</tbody>
</table>

Table 6 Static Structural analysis results

6.2 Buckling analysis of shaft

The total displacement obtained by buckling analysis is shown in table 6.2

<table>
<thead>
<tr>
<th>S.No</th>
<th>Material</th>
<th>Maximum Deformation (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Steel</td>
<td>1.105</td>
</tr>
<tr>
<td>2</td>
<td>Glass epoxy</td>
<td>1.0242</td>
</tr>
<tr>
<td>3</td>
<td>Graphite epoxy</td>
<td>1.007</td>
</tr>
<tr>
<td>4</td>
<td>Carbon epoxy</td>
<td>1.0042</td>
</tr>
</tbody>
</table>

Table 7 Buckling analysis results

6.3 Modal analysis of shaft
Natural frequency obtained by modal analysis is shown in table 6.3

<table>
<thead>
<tr>
<th>S.No</th>
<th>Material</th>
<th>Natural frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Steel</td>
<td>153.04</td>
</tr>
<tr>
<td>2</td>
<td>Glass epoxy</td>
<td>274.31</td>
</tr>
<tr>
<td>3</td>
<td>Graphite epoxy</td>
<td>309.1</td>
</tr>
<tr>
<td>4</td>
<td>Carbon epoxy</td>
<td>345.74</td>
</tr>
</tbody>
</table>

Table 8 Modal analysis results

6.4 Comparison of mass per unit length obtained from Theoretical, MATLAB and ANSYS

The mass for different material obtained from theoretical analysis, MATLAB and ANSYS compared in the table 6.4 the results are almost equal

<table>
<thead>
<tr>
<th>S.No</th>
<th>Material</th>
<th>Theoretical</th>
<th>MATLAB</th>
<th>ANSYS</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Steel</td>
<td>25.98</td>
<td>25.9867</td>
<td>21.497</td>
</tr>
<tr>
<td>2</td>
<td>Glass epoxy</td>
<td>1.65</td>
<td>1.6804</td>
<td>2.5054</td>
</tr>
<tr>
<td>3</td>
<td>Graphite epoxy</td>
<td>1.9</td>
<td>1.8988</td>
<td>2.8311</td>
</tr>
<tr>
<td>4</td>
<td>Carbon epoxy</td>
<td>1.344</td>
<td>1.3443</td>
<td>2.0043</td>
</tr>
</tbody>
</table>

Table 9 Comparison of mass

6.5 Comparison of Static structural, buckling and modal analysis

The graph 6.2 shows the comparison of static straactural, buckling and moal analysis results in that static strength ,buckling deformaion and natural frequency were compared

Graph 6.2 comparison of static strength, buckling deformation and Natural frequency

VII. CONCLUSION

Brief study about automobile drive shaft with different materials is done in this project to replace convectional steel shaft with composite shaft, by considering maruthi Suzuki engine specifications theoretical design of shaft with steel, glass epoxy, graphite epoxy and carbon epoxy was done. The results validated with convectional MATLAB coding. Then perform static structural analysis to find structural strength of the shaft and buckling analysis to find buckling strength and modal analysis to find natural frequency of the shaft. From Graph 6.2 Graphite epoxy has more strength compared to other
materials, carbon epoxy has more natural frequency compared to other materials and carbon epoxy has less buckling deformation compared to other materials. From Table9 carbon epoxy shaft has less mass compared to other materials. So it was concluded that if only strength is constrain Graphite epoxy is the best replacement for steel, if Static strength, buckling strength and natural frequency and mass is considered carbon epoxy is best replacement for steel. so by all these constrains carbon epoxy is the best replacement for steel to the drive shaft application.

REFERENCES