International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 4 Issue 05, May-2015

Fatigue Analysis of Composite Drive Shaft

Prof. Mahesh. N. Pradhan
Mechanical engg dept. M.I.T, Kothrud ,Pune- 52.
Pune, India.

Hrishikesh .T. Gaikwad
Student Mech. Engg Dept. M.I.T, Kothrud ,Pune- 52.
Pune, India.

***Abstract* - The projects aims at replacing the conventional drive shaft with composite drive shaft which will provide us the better mechanical properties i.e (Torque transmitting capacity, and fatigue life of the shaft).The research paper will include the comparison of these properties of conventional steel shaft with the composite drive shaft .The drive shaft selected is applicable to TATA 407 pick up vehicle which is at present using the conventional steel drive shaft. Our main aim is to show that the fatigue life of composite drive shaft is much better than conventional steel drive shaft.**

Keywords- Composite shaft, conventional shaft, TATA 407 pick up, fatigue analysis, mass saving.

1.INTRODUCTON

Power transmitted from the engine to the final drive where useful work is applied through a system consists of a gearbox, clutch, universal joint, drive shaft and a differential in the rear-drive automotives. In conventional drive shafts there is problem of instant crack propagation in case of heavy loads unlike the crack arrest property of composite materials. As compare to conventional metallic drive shaft, the composite drive shafts have many parameters to be altered, namely the fiber orientation angles, stacking sequences, layers thicknesses and number of layers. These parameters, due to the tailorability of elastic constants, could provide a large number of possible designs, which must satisfy optimally the performance characteristics of the composite drive shaft (critical speed, fatigue life and load carrying capacity). It is well-known that the steel drive shaft is usually manufactured in two pieces. There are many design studies but the information about the design variables and their effect on the performance characteristics is not comprehensive.

Generally, all accessed design studies were not including the fatigue consideration, which may be needed to be explored in relation to composite shafts design. Therefore, the aim of this work is to investigate numerically the effect of stacking sequence and fiber orientation angle on the performance of drive shaft. The numerical results Will be validated by results obtained from analytical solutions. The specimens used will be filament wounded. Till now very less work is done on fatigue analysis of shaft made up of composite material.

1.1 Methodology

The composite shaft will be manufactured using the process known as 'filament winding'. The process involves winding filaments under tension over a male mandrel. The mandrel rotates while a wind eye on a carriage moves horizontally, laying down fibers in the desired pattern .Filament winding is well suited to automation, and there are many applications, such as pipe and small pressure vessel that are wound and cured without any human intervention.

Many sample specimens will be tested for fatigue failure and the optimum one would be selected. Detail stress analysis will be performed on ansys software .The matlab software will be used to verify the analytical values and to obtain quick results of the same.

2. DESIGN SPECIFICATION

a. Conventional drive shaft of TATA 407 pick up.

The drive shaft outer diameter should not exceed 100 mm due to space limitations. Here outer diameter of the shaft is taken as 75 mm.

Based on standards available specifications of drive shaft are –

i. The torque transmission capacity of the drive shaft(T) = 2058.75 N-m
ii. Speed of drive shaft = 2800 rpm
iii. Outside diameter of drive shaft = 75 mm
iv. Length of drive shaft = 1.3 m

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

Table 1

Properties of steel (SM45C)				
S.r No.	Mechanical properties	Symbol	Units	Steel
1	Young's Modulus	E	GPa	207
2	Shear modulus	G	GPa	80
3	Poisson's ratio	µ	-	0.3
4	Density	P	Kg/m3	7600
5	Yield Strength	S_y	MPa	370

Gear Ratio for Tata 407 pickup
No.of Gears are 5 forward and 1 reverse.
Table.2

Gear ratios for TATA 407 Pick up		
S.r No.	Gear Position	Gear ratio
1	1st Gear	6.01:1
2	2nd Gear	3.46:1
3	3rd gear	1.97:1
4	4th Gear	1.37:1
5	Reverse Gear	5.69:1

So, the torque is maximum at 1st gear, when the speed of vehicle is low. Therefore the maximum torque will be,

$$T_{max} = 225 \times 6.01 \times 1.5$$
$$T_{max} = 2058.75 \times 10^3 \text{ N - mm}$$
$$M_{t_{max}} = 2058.75 \times 10^3 \text{ N - m}$$

2.1 Torsional Strength:

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

$$\tau = \frac{16M_t}{\Pi d_o^3(1 - C^4)} = \frac{16(2058.75 \times 10^3)}{\Pi(75)^3(1 - C^4)}$$

$$C = 0.9223$$

We know,

$$C = d_i / d_o$$

$$0.9223 = \frac{d_i}{0.075}$$

$$d_i = 69.182\text{mm}$$

Mass of steel drive shaft

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

$$m = 6.575\text{kg}$$

2.2 Design of shaft against fatigue loading

Bending moment in shaft

We assume that only the force due to self weight of the shaft is acting on the shaft .It is acting at the centered the shaft.

$$P = w \times g$$
$$p = 6.575 \times 9.81$$
$$p = 64.5N$$

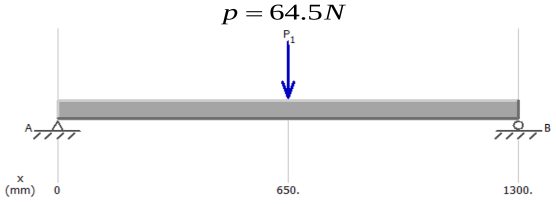

Fig.1 loading diagram

$$M_a = 64.5 \times 650 - (1300 \times B_y) = 0$$
$$B_y = 32.25\text{N}$$

similarly

$$A_y = 32.25\text{N}$$
$$M = A_y \times 650$$
$$M = 32.25 \times 650$$
$$M = 20962.5\text{N.mm}$$

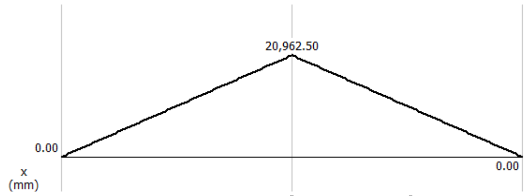

Fig.2 Bending moment diagram

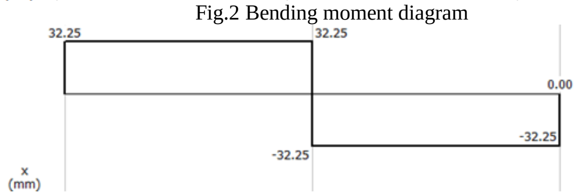

Fig.3 Shear diagram

2.3 Fatigue life prediction

$(Mb)Max=20.963 \times 10^3$ N-mm.
(Mb)Min= 0 N-mm
Then,
a. To find Mean & Amplitude Stresses.

(Mb)m= 1/2(Mb)max + (Mb)min)
(Mb)m= 10.48 X 10^3 N-mm.

(Mb) a = 1/2(Mb)max - (Mb)min)
(Mb) a = 10.481 x 10^3 n-mm

(Mt) max = 225 x 6.01 x 1.5 = 2058.75 x 10^3 N-mm (given)
(Mt) min =225 x 1.37 x 1.5 = 4.49 X 106 N-mm (given)

$$\sigma_{xm} = \frac{32(M_b)_m}{\pi d^3}$$

σxm = 0.253 N/mm2

$$\sigma_{xa} = \frac{32(M_b)_a}{\pi d^3}$$

σxa = 0.253 N/mm2

$$(M_t)_m = \frac{1}{2}[(M_t)_{\max} + (M_t)_{\min}]$$

$$(M_t)_m = \frac{1}{2}[2058.75 + 462.38] \times 10^3$$

$$(M_t)_m = 1260.565 \times 10^3 N - mm$$

$$(M_t)_a = \frac{1}{2}[(M_t)_{\max} - (M_t)_{\min}]$$

$$(M_t)_a = \frac{1}{2}[2058.75 - 462.38] \times 10^3$$

$$(M_t)_m = 798.185 \times 10^3 N - mm$$

$$\tau_{xym} = \frac{16(M_t)_m}{\pi d^3}$$

$$\tau_{xym} = 15.2177 N / mm^2$$

$$\tau_{xya} = \frac{16(M_t)_a}{\pi d^3}$$

$$\tau_{xya} = 9.6358 N / mm^2$$

$$\sigma_m = \sqrt{(\sigma_{xm} \times K_b)^2 + 3(\tau_{xym} \times K_t)^2}$$

$$\sigma_m = \sqrt{(0.253 \times 2)^2 + 3(15.2177 \times 1.5)^2}$$

$$\sigma_m = 39.54 \ x \ 10^6 Pa$$

$$\log(\sigma_m) = 7.597 Pa$$

$$\sigma_a = \sqrt{(\sigma_{xa} \times K_b)^2 + 3(\tau_{xa} \times K_t)^2}$$

$$\sigma_a = \sqrt{(0.253 \times 2)^2 + 3(15.2177 \times 1.5)^2}$$

$$\sigma_a = 25.0375 \ x \ 10^6 Pa$$

$$\log(\sigma_a) = 7.39 Pa$$

Since ,

$$\sigma_a = \sigma_{eq} = 39.54 N / mm^2$$

So, von mises stresses are 39.54 N / mm^2

$$\tan\theta = \frac{\sigma_a}{\sigma_m} \ \& \ \tan\theta = \frac{s_a}{s_m}$$

$$\tan\theta = 0.633$$

$$\therefore \theta = 32.33 \cong 33$$

b. To find endurance strength S_e

$$S_e = K_{load} \times K_{size} \times K_{surf} \times K_{temp} \times K_{reliability} \times K_d \times K_{stressconcentration} \times S'_e$$

$$K_{load} = 1 \qquad K_{size} = 0.75$$

$$K_{surf} = A.S_{ut}^{\ 0.265} = 0.82 \qquad K_{temp} = 1$$

for $T \leq 450^\circ$

$$K_{reliability} = 90\% = 0.897 \qquad K_d = \frac{1}{2} = 0.5$$

$$S_e = 1 \times 0.75 \times 1 \times 0.82 \times 0.897 \times 0.5 \times 315$$

$$S_e = 86.88 N / mm^2$$

Acoording to modified goodman method

$$\frac{S_a}{S_e} + \frac{S_m}{S_{yt}} = 1$$

Put above values ,

$$\frac{0.632 S_a}{86.88} + \frac{S_m}{370} = 1$$

$$S_m = 100.125 N / mm^2$$

$$S_a = 63.36 N / mm^2$$

$$FOS = \frac{S_a}{\sigma_a} = \frac{63.36}{25.03}$$

Fatigue factor of safety = **2.53**

c. To find number of cycles

$$S_f = \frac{\sigma_a \times S_{ut}}{S_{ut} - \sigma_m}$$

$$S_f = \frac{25.03 \times 630}{630 - 39.54}$$

$$S_f = 26.706 Mpa$$

0.9 S_{ut} =0.9(630) = 567 Mpa

log10(0.9 S_{ut}) = 2.5670

log10(S_e) = 1.6474

log10(S_f) = 1.3090

$$Log_{10}(N) = \frac{(6-3)(2.75358 - 1.4266)}{(2.75358 - 1.9389)}$$

$$Log_{10}(N) = 4.889 + 3 = 7.889$$

N = 77.446 x 10^6 cycles

3. DESIGN OF COMPOSITE DRIVE SHAFT

3.1 Selection of Material

Carbon Fiber(Panex 35):

Panex® 35 continuous carbon fiber is manufactured from polyacrylonitrile (PAN) precursor. The consistency in yield and mechanical properties that are provided by large filament count strands gives the user the ability to design and manufacture composite materials with greater confidence and allows for efficient and fast buildup of carbon fiber reinforced composite structures.Material properties of Carbo fiber(panex 35) is as follows.

Table .3

Properties of panex 35			
Sr.no	Parameter	SI	US
1	Tensile Strength	4137 MPa	600 ksi
2	Tensile Modulus	242 GPa	35 msi
3	Electrical conductivity	0.00155 ohm-cm	0.00061 ohm-in
4	Density	1.81 g/cc	0.065 lb/in3
5	Fiber Diameter	7.2 microns	0.283 mils
6	Carbon content	95%	95%
7	Yield	270 m/kg	400 ft/lb
8	Spool weight	5.5 kg, 11 kg	12 lb, 24 lb
9	Spool Length	1,500 m , 3,000 m	1,640 yd, 3,280 yd

3.2 Epoxy resin

Epoxy resins are polyether resins containing more than one epoxy group capable of being converted into the thermoset form.

MECHANICAL PROPERTIES EPOXY RESIN (HEXCEL HEXPLY EH04 EPOXY RESIN)

Table.4

MECHANICAL PROPERTIES EPOXY RESIN (HEXCEL HEXPLY EH04 EPOXY RESIN)			
Sr. No	Mechanical Property	Value	Units
1	Tensile Strength	81	MPa
2	Tensile Modulus	3.45	GPa
3	Specific Gravity	1.34	g/cc
4	Poisson's ratio	0.3	-
5	Shear Strength	1.34	MPa
6	Shear Modulus	1.3269	GPa

Mechanical Analysis of Lamina

a. Volume fraction of fiber – 0.7 (70%)
b. Volume fraction of matrix – 0.3 (30%)
c. Volume of composites – 1 (100%)

we know ρ_f = 1810 kg/m^3, ρ_m =1340 kg/m^3

$$\rho_c = \rho_f V_f + \rho_m V_m$$
$$\rho_c = 1810 \times 0.6 + 1340 \times 0.4$$
$$\rho_c = 1622 kg / m^3$$

Table . 5

Sr. No	Composite Properties	Symbol	Value	Unit
1	Young's Modulus(Longitudinal direction)	E_1	146.58	GPa
2	Young's Modulus(Transverse direction direction)	E_2	8.44	GPa
3	Major Poison's ratio	μ_{12}	0.3	-
4	Minor Poisson's ratio	μ_{21}	0.3	-
5	Shear Modulus	G_{12}	3.23	GPa
6	Ultimate longitudinal strength	$(\sigma_1^T)_{ult}$	2500	MPa
7	Ultimate transverse strength	$(\sigma_2^T)_{ult}$	30.97	MPa
8	Ultimate longitudinal compressive strength	$(\sigma_1^C)_{ult}$	70.4	MPa
9	Ultimate transverse compressive strength	$(\sigma_2^C)_{ult}$	33.88	MPa
10	Minimum fiber volume fraction	$(V_f)_{min}$	0.529	%
11	Critical fiber volume fraction	$(V_f)_{cr}$	0.540	%
12	Shear strength	τ	11.46	MPa

5. Mass Saving

1. Mass of steel drive shaft = 6.575 kg
2. Mass of Composite drive shaft

$$m = \Pi(r_0^2 - r_i^2) L \rho$$
$$m = \Pi(0.0375^2 - 0.0360^2)(1.3)(1622)$$
$$m = 0.7303 Kg$$

3. Percentage of mass saving over steel is

$$= \frac{6.575 - 0.7303}{6.575} \times 100$$
$$= 88.89\%$$

Bending moment in shaft

We assume that only the force due to self weight of the shaft is acting on the shaft .It is acting at the centered the shaft.

Force P is given as ,

$$P = w \times g$$
$$p = 1.772 \times 9.81$$
$$p = 17.384 N$$

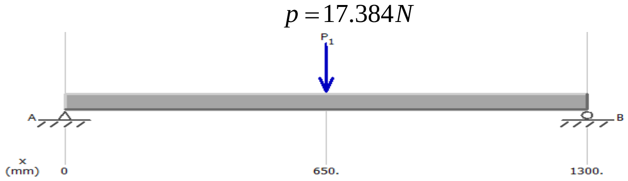

Fig.4 Loading diagram

$$M_a = 17.384 \times 650 - (1300 \times B_y) = 0$$
$$B_y = 8.69 N$$
$$similarly, A_y = 8.69 N$$

Maximum bending moment is given as ,

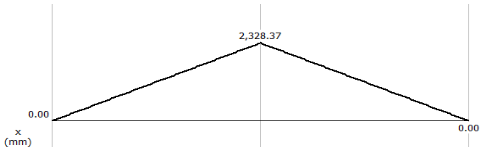

Fig.5 Bending Moment diagram

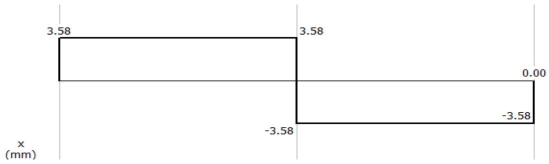

Fig.6 Shear dagram

Maximum bending moment is given as ,

$$M = A_y \times 650$$
$$M = 8.69 \times 650$$
$$M = 5648.5 N.mm$$
$$M = 5.6483 N.m$$

6. Fatigue life prediction

(Mb)Max=5.648 x 10^3 N-mm.
(Mb)Min= 0 N-mm
Then,

6.1 To find Mean & Amplitude Stresses.

(Mb)m= 1/2(Mb)max + (Mb)min)
(Mb)m= 2.834X 10^3 N-mm.

(Mb) a = 1/2(Mb)max - (Mb)min)
(Mb) a = 2.824 x 10^3 n-mm

(Mt) max = 225 x 6.01 x 1.5 = 2058.75 x 10^3 N-mm (given)
(Mt) min =225 x 1.37 x 1.5 = 4.49 X 106 N-mm (given)

$$\sigma_{xm} = \frac{32(M_b)_m}{\pi d^3}$$
$$\sigma_{xm} = 0.068224 \; N/mm^2$$

$$\sigma_{xa} = \frac{32(M_b)_m}{\pi d^3}$$
$$\sigma_{xa} = 0.068224 \; N/mm^2$$

$$(M_t)_m = \frac{1}{2}[(M_t)_{\max} + (M_t)_{\min}]$$
$$(M_t)_m = \frac{1}{2}[2058.75 + 462.38] \times 10^3$$
$$(M_t)_m = 1260.565 \times 10^3 \; N - mm$$

$$(M_t)_a = \frac{1}{2}[(M_t)_{\max} - (M_t)_{\min}]$$
$$(M_t)_a = \frac{1}{2}[2058.75 - 462.38] \times 10^3$$
$$(M_t)_a = 798.185 \times 10^3 \; N - mm$$

$$\tau_{xym} = \frac{16(M_t)_m}{\pi d^3} = \frac{16 \times 1260.565 \times 10^3}{\pi (75)^3}$$

$$\tau_{xym} = 15.2177 \frac{N}{mm^2}$$

$$\tau_{xya} = \frac{16(M_t)_a}{\pi d^3} = \frac{16 \times 798.185 \times 10^3}{\pi (75)^3}$$

$$\tau_{xya} == 9.6358 \; \frac{N}{mm^2}$$

$$\sigma_m = \sqrt{(\sigma_{xm} \times K_b)^2 + 3(\tau_{xym} \times K_t)^2}$$
$$\sigma_m = \sqrt{(0.02704 \times 2)^2 + 3(15.2177 \times 1.5)^2}$$
$$\sigma_m = 39.53 \; x \; 10^6 \, Pa$$

Log(σ_m)=7.596 Pa

$$\sigma_a = \sqrt{(\sigma_{xa} \times K_b)^2 + 3(\tau_{xa} \times K_t)^2}$$
$$\sigma_m = \sqrt{(0.253 \times 2)^2 + 3(15.2177 \times 1.5)^2}$$

σ_a = 25.032 x 10^6 Pa

Log(σ_m)=7.4 Pa

Since , $\sigma_a = \sigma_{eq} = 39.54 N/mm^2$

So, von mises stresses are 39.54 N/mm^2

$$\tan\theta = \frac{\sigma_a}{\sigma_m} \; \& \; \tan\theta = \frac{S_a}{S_m}$$
$$\tan\theta = 0.633$$
$$\therefore \theta = 32.33 \cong 33$$

6.2 To fing endurance strength S_e

$$S_e' = 0.5 \times S_e$$
$$S_e' = 0.5 \times 1169.58$$
$$S_e' = 584.5 Mpa$$

$$S_e = K_{load} \times K_{size} \times K_{surf} \times K_{temp} \times K_{reliability} \times K_d \times K_{stressconcentration} \times S'_e$$

$$K_{load} = 1 \; K_{size} = 0.75$$

$K_{surf} = A.S_{ut}^{\;0.265} = 0.82 \; K_{temp} = 1$ for

$$T \leq 450^{\circ}$$

$K_{reliability} = 90\% = 0.897$ $K_d = \frac{1}{2} = 0.5$

$S_e = 1 \times 0.75 \times 1 \times 0.82 \times 0.897 \times 0.5 \times 584.5$

$S_e = 161.46 \frac{N}{mm^2}$

Acoording to modified goodman method

$$\frac{S_a}{S_e} + \frac{S_m}{S_{yt}} = 1$$

Put above values ,

$$\frac{0.624 S_m}{174.46} + \frac{S_m}{1169} = 1$$

$$S_m = 208.91 \frac{N}{mm^2}$$

$$S_a = 132.29 \frac{N}{mm^2}$$

$$FOS = \frac{S_a}{\sigma_a} = \frac{132.14}{25.03}$$

Fatigue factor of safety = 5.28

6.3 *To find number of cycles*

$$S_f = \frac{\sigma_a \times S_{ut}}{S_{ut} - \sigma_m}$$

$$S_f = \frac{25.032 \times 1169}{1169 - 39.53}$$

$$S_f = 25.92 Mpa$$

$0.9 S_{ut}$ =0.9(1265) = 1138.5Mpa

log10(0.9 S_{ut}) = 3.0263

log10(S_e) = 2.24

log10(S_f) = 1.413

$$Log_{10}(N) = \frac{(6-3)(3.0263-1.413)}{(3.0563-2.207)}$$

$$Log_{10}(N) = 6.043 + 3 = 9.043$$

N = 834 x 10^6 cycles

7. RESULTS AND DISCUSSION

Table .6

COMPARISON BETWEEN STEEL AND COMPOSITE DRIVE SHAFT.

Sr.No	Parameter	Steel shaft	Composite shaft
1	Applied Torque (T)	2058.75 N-m	2058.75 N-m
2	Fatigue factor of safety	2.53	5.28
3	Number of cycles (N)	77.446 x10^6	834.02 x 10^6
4	Mass (m)	6.575 Kg	0.7303 kg
5	Percentage of mass saving	-	88.89%

8.CONCLUSION

a. From the above results we Come to the conclusion that the fatigue factor of safety of composite drive shaft is much higher than steel drive shaft.

b.The number of cycles sustained by the composite drive shaft is considerably high .

c.We have also achieved a high percentage of mass saving.

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