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Enhancement of Fatigue Life of Shafts

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ABSTRACT

This study aims at guiding against fatigue failure to enhance fatigue life. The question of crack initiation, propagation, and failure in parts undergoing stress reversals or repeated stresses is a problem in machine design. Stiffness is degraded, and strength is reduced. Fatigue-based analytical model has been developed. The design philosophy is set on Design for Finite/Infinite life in which the design stress is below some fatigue strength, the endurance limit. Finite/Infinite fatigue life design tool (and calculations using Marin equations) was utilized to obtain the shaft specimen endurance limit of 235.7 MPa. Below this stress level, fatigue life was prolonged. Attention is also paid to Fail safe design, where cracks initiate and propagate; then a structure (composite material) is designed (against stress concentrations) to impede the crack growth. The materials for the shaft (steel matrix and carbon laminates) are notable for certain properties such as strength, which is due to choice, treatment, or processing, and is an inherent property of the part. Finally, there is Evaluation of stiffnesses and strengths. Stiffness depends on both the modulus of elasticity and the geometry. Primary parameters were calculated (fatigue stress concentration factor of 1.5) to aid other parameters (endurance limit of 235.7). The fiber *laminates reinforced the matrix. Elimination of degradation in fatigue life of the elastic property produced enhancement. Strength is present whether there is load on the material. The study shows that by using composite materials and designing below the endurance limit, fatigue life of the shaft subjected to loading can be prolonged.*

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KEYWORDS: Fail-Safe, Finite/Infinite life, Steel/Carbon Composite, Stiffness, Strength.

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1. INTRODUCTION

Experimental efforts on fatigue problems have been conducted for some time. In the loading configurations, useful data were collected on crack initiation, propagation, and eventual specimen failure (Shigley & Mischke, 2001). However, there has not been enough study on fatigue life improvement scheme. In this work, material properties necessary to enhance fatigue life are obtained. In addition to that, stiffness and strengths are evaluated. Fatigue cracks usually initiate from the location of stress concentration. High cycle fatigue caused by vibration is the main failure mechanism. Fatigue failures are induced by the structure, material defect, processing techniques, etc. (Zhang *et al*., 2016).

The experimental data and analysis tools are used to predict behavior under service conditions. Fatigue crack growth curves are obtained by applying comprehensive tensile cyclic loading (Reifsnider, 2018). The crack growth curves are influenced by elevated temperature fatigue behavior. Fatigue crack growth rates are among the most sensitive of all fracture mechanics phenomena related to properties. This results in the use of generic properties for anything other than short-term crack growth predictions, or rough estimates. Increase in fillet radii, and change in the position of the support of the shaft decrease the stress concentration factor, and increase both the endurance limit, and fatigue factor of safety of the shaft (Marudachalam *et al*., 2011).

This study deals with fatigue resulting from reversed stresses with maximum value as the largest algebraic value, and minimum value as the lowest algebraic value. The mean stress is the average of the maximum and minimum stresses (Afolabi *et al*., 2019). In comprehensive-tensile cyclic loading, the fatigue crack growth curves give effective stress intensity factor range that more accurately described the crack growth property (Azuma & Yamazaki, 2018). Simulation of crack propagation of steel was done by finite element analysis (FEA) using adaptive re-meshing (Hjiej *et al*., 2018). This allows the prediction of fatigue crack growth life using results of tests on specimens as an input. There has been increased interest in prolonging the fatigue life of machine components such as shafts. This stimulated numerous research efforts directed towards the understanding the fatigue phenomenon better. Variational derivatives relate change in a fatigue crack stress function to change in a function on which the stress function depends (Westergaard, 1939).

Probabilistic assessment of residual life improves the understanding of the mechanism of fatigue and the impact of defects on structural integrity (Cerny *et al*., 2017). Compressive residual stresses improved fatigue life (Pramanik *et al*., 2017). This project aims to analyze and model a steel shaft and a steel/carbon composite shaft to show how fatigue life can be enhanced. Replacing the conventional drive shaft with a composite drive shaft provides better mechanical properties, torque transmitting capacity, and fatigue life of the shaft (Mahesh *et al*., 2015). The fatigue life of a composite drive shaft is much better than a conventional steel drive shaft. The elimination of vibration fatigue within the part enhances its life (Fisher *et al.*, 2015). Devising a solution to the problem, material properties for finite/infinite life design are used concurrently with endurance limit to enhance life. This way, a safe and economical machine shaft is achieved. Surface dents enhance fatigue life, and fatigue-like enhancement comes from the reduced local traction. Subjecting the shaft to remote bending and torque, the stresses are analyzed, the results calculated and plotted on the S-N diagram for fatigue analysis and to make predictions (Xuejun *et al*., 2018; Karaagac, 2013).

Stress concentration factors are calculated as primary parameters that aid in computing the bending stresses at critical location with the fail-safe design technique, which, in turn, aid in deciding finite/infinite life conditions. Fatigue crack growth rates and the fitting parameters are among the most sensitive of all properties. Stiffnesses and strengths require analysis to predict life enhancement (Sanford, 2013). The analytical tool, in this study, is the Finite Element Analysis (FEA), MATLAB code. The design and finite element analysis of composite drive shafts result in better materials used in the fabrication of the shafts (Shoaib *et al*., 2016). Composites facilitate the fabrication techniques and are of good crack extension resistance (Leake *et al.*, 2018). Finite element-based computer program simulate crack propagation and estimate crack-tip stress intensity factors (Lecki & Ballarini, 2018). In this study, a fatigue-based analytical model was developed for fatigue life enhancement of shafts. Mathematical Methods in Engineering approach was used to carry out the stress analysis. Solid works software and MATLAB code were also utilized to evaluate the state of strength and stiffness.

2 MATERIALS AND METHODS

2.1 Materials

The shaft in this study was subjected to a static bending load of 7 kN. When the load was removed, reloading in the opposite direction produced a similar effect. The reversed stresses constituted fatigue loading, causing crack to initiate and propagate to failure of the shaft. The procedure for developing material selection criteria included identifying the primary function of the component (bending rotation), and the weight which is the most important part. The steel here is the American Iron and Steel Institute (AISI) 1020 Steel, Cold-Rolled. It has a carbon content of about 0.20 percent. A carbon-fiber-reinforced laminated composite, (two perfectly bonded thin layers made of different orthotropic materials) was selected for this study. Its advantages include high specific strength and modulus, low coefficient of thermal expansion, and high fatigue strength which is an ideal fiber for torque transmitting shaft applications. The selection was based on their attributes such as fatigue strength, and ductility. The modulus of elasticity is the relevant modulus and played an important role in the decision to use the steel and the composite.

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2.2 Methods

2.2.1 Fail-Safe Design

By fail-safe design, the shaft can continue to perform its basic function even after sustaining a specific level of damage. The calculation for stress concentration factors is necessary as primary parameters to aid others such as bending stress. In Fig.1, the shaft is loaded by a non-rotating force F, of 7.0 kN. Failure is expected to occur at B or at C, instead of at the point of maximum moment. This is so since points B and C have smaller cross sections, a higher bending moment, and a higher stress concentration factor than the location of maximum moment. The location of maximum moment has a larger size and no stress concentration. The strength will be different elsewhere than at point B or C. This strength is estimated first and then compared with the stress at the same point.

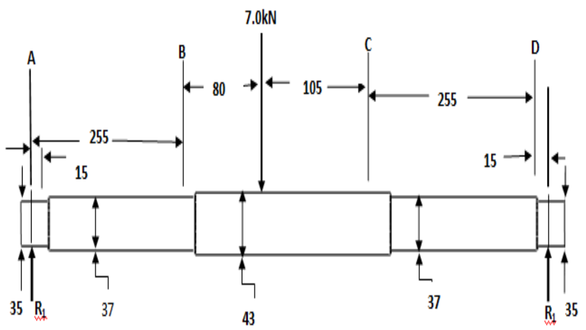

Fig. 1: Shaft model (all dimensions in mm and fillets, 3 mm radius)

The theoretical Stress Concentration, K_t aids in life enhancement. The shaft shoulder is filleted with the fillet radius of 3 mm. The stress concentration factor, K_t, related to maximum principal stress is given by the general expression (Shigley & Mischke, 2001):

$$K_t = 0.632 + 0.377\left(\frac{D}{d}\right)^{-4.4} + \left(\frac{r}{d}\right)^{-0.5}$$

$$\sqrt{\left[\frac{-0.14-0.363\left(\frac{D}{d}\right)^2+0.503\left(\frac{D}{d}\right)^4}{1-2.39\left(\frac{D}{d}\right)^2+3.368\left(\frac{D}{d}\right)^4}\right]} \quad (1)$$

where D and d are the larger and smaller diameter portions of the shaft, respectively. Since, D/d = 1.162 and r/d = 0.0811, K_t gives 1.88

At the Critical Location, the stress concentration factor, K_t (related to maximum von Mises stress) gives $K_t = 1.65$. The Fatigue Stress Concentration is determined using the modified Neuber equation (Sanford, 2013). The fatigue stress concentration factor, K_f (related to maximum principal stress) is:

$$K_f = \frac{LN\,(1, C_{kf})\,K_t}{1+\frac{2}{\sqrt{r}}\frac{K_t-1}{K_t}\sqrt{a}} \quad (2)$$

where K_f is lognormal (LN) distributed, r is the fillet radius, and $\sqrt{a}$ is a function of the mean ultimate tensile strength, $\bar{S}_u$; value of $\sqrt{a}$ for steels of the geometry of the shaft in this project is $139/\bar{S}_{ut} = 0.201$. So, $K_f = 1.71$. This stress concentration factor applies to 10^6 cycles or more. The fatigue stress concentration factor, K_f (related to maximum von Mises stress) is (Sanford, 2013):

$$K_f = \frac{LN\,(1, C_{kf})\,K_t}{1+\frac{2}{\sqrt{r}}\frac{K_t-1}{K_t}\sqrt{a}} = (K_f)_{10^6} = 1.5 \quad (3)$$

These stress concentration factors are used to compute the bending stress and to size the shaft. The bending stress computation aids in deciding if the shaft has finite or infinite life. So, the bending moment of the reaction force R_1 at end A of the shaft about the critical location, B is as follows:

$$\sum M_D = R_1(695) - 7.0\,(360) = 0 \quad (4)$$

from which R_1 is $\frac{7.0\,(360)}{695}$. So, moment of R_1 about point B is

$$\frac{7.0\,(360)}{695}\,255 = 925 \text{ MPa} \quad (5)$$

Applying the fatigue stress concentration factor, the bending stress is

$$\sigma = K_f\,\frac{M_B}{I/c} \quad (6)$$

where the section modulus I/C is $\pi d^3/32$

$$= 4.97 \text{ cm}^3$$

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Shaft geometry will satisfy the material strength requirements and shaft-supported element requirements. Based on von Mises stress, and fatigue stress concentration factor, K_f the bending stress,

$$\sigma = K_f \frac{M_B}{I/c} = \frac{1.5(695)}{3.22} = 323.8 \text{ MPa} \quad (7)$$

The bending stress will aid in deciding finite/infinite life status later. von Mises stress under combined bending and torsion loads (Shigley & Mischke, 2001):

$$\tau_{\max} = \sqrt{[(\frac{32M_b k_f}{\pi d^3})^2 + 1.5(\frac{16T}{\pi d^3})^2]} \quad (8)$$

from which the shaft diameter,

$$d = (\frac{32Mn}{\pi S_e})^{1/2} = 43 \text{ mm.} \quad (9)$$

After thousands of cycles of loading, cracks initiate as one of the results of damage due to fatigue. The shaft design is based on the stress at which the failure theory predicts that the weakest portion will fail. The physical situation corresponding to weakest portion failure is imagined to be cracking. The threshold of initiation of cracks can be well anticipated by laminate analysis. The crack problem, in Mode I crack loading, external forces open up crack faces. The resulting redistribution of stress components can be expressed using Westergaard's method. Tensile waves propagating across the body reload the crack tip. The crack is created by the classical analytical approach. An infinitesimal sized crack ds is considered linearized.
The slope is dy/dx, and the length, s is, using the quadratic formula:
$(ds)^2 = (dx)^2 + (dy)^2$. This implies that

$ds = \sqrt{[(dx)^2 + (dy)^2]}$, so, integrating both sides,

$$s = \int \sqrt{[1 + (\frac{dy}{dx})^2]}dx = 1\text{mm} \quad (10)$$

where the terms under the radical sign are each divided by dx, dy is y-increments.
Crack loading Mode II is in-plane shear with stress components as well. Factors influencing the crack growth include minimum/maximum stress ratio, microstructures, surface discontinuities, etc. The shaft was designed with the composite material as one package that meets the strength, geometry, and materials requirements. With the state of weakness, strength is reduced, as well as the stiffness, indicating evidences of damage. This resulted in stress redistribution.

2.2.2 Design for Finite/Infinite Life

Stress testing is preferred to strain testing for endurance limits. Many ideas inform the quantitatively estimated endurance limits and endurance strengths (the term used for endurance limits in variable loadings other than reversed loading). The Marin's equations for endurance limit, σ_e, or endurance strength at the critical location of a steel machine part is (Shigley & Mischke, 2001):

$$\sigma_e = k_a k_b k_c k_d k_e k_f \sigma'_e \quad (11)$$

where k_a is the surface condition modification factor (depending on quality of surface finish and on tensile strength). $k_a = a\sigma^{-b}_{ut} LN(1,c)$ MPa, a,b,c are from tables, LN(1,c) is unit variate lognormal distribution, the mean is 1, standard deviation is c. The mean of k_a is $\overline{K_a}$ = a σ^{b}_{ut} and standard deviation of σ_{ka} is c_{ka}. k_b is the size modification factor. The larger the size, the smaller the endurance strength. k_c, the reliability goal modification factor predicts the distortion energy theory $K_{C_{torsion}}$ to be 0.577. The temperature modification factor is k_d, while k_e is the stress concentration modification factor (not always available) and k_f is the miscellaneous effect modification factor. σ'_e = endurance limit of beam specimen.

In the geometry and condition of use, these facilitate work on fatigue problems. σ_e is the stress for which failure does not occur even for infinitely large number of loadings. Assuming processing to be cold drawn, then from Shigley and Mischke (2001): The ultimate tensile strength, σ_{ut} is 690 MPa, Yield stress, 580 MPa (to be used to decide the yielding or no yielding situation), material fatigue strength, σ_m is $0.8\sigma_{ut} =$ 552 MPa. σ'_e is then estimated by using the Marin's equation for bending:

$$\sigma'_e \text{ being } \emptyset_{0.30}\, \bar{\sigma}_{ut} = 0.506 \text{ LN } (1, 0.138) \leq 1460 \text{ MPa} \quad (12)$$

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where $\emptyset_{0.30}$ is fatigue ratio, LN is lognormal distribution of mean 1 and standard deviation 0.138. Therefore, the specimen endurance limit is 0.506 (690) = 349.1 MPa. Surface factor for machined (or cold drawn) part,

k_a is 4.45 $\bar{\sigma}_{ut}^{-0.265}$ LN (1, 0.058) (13)

Surface modification factor depends on the quality of the surface finish and on the tensile strength.

Hence $k_a = 4.45\,(690)^{-0.265} = 0.787$

$k_b = (d/7.62)^{-0.107} = 1.24d^{-0.107}$

(for $2.79 \leq d \leq 51$mm)

giving $0.859 = 32/7.62)^{-0.107} = 0.858$ and $k_c = K_d = K_e = 1$

$\sigma_e = 0.787(0.858)\ 349.1 = 235.7$MPa (14)

The material strength (at $N < 10^3$),

$\sigma_m = 0.9\sigma_{ut} = 0.9(690) = 621$ MPa (15)

The endurance limit will also aid in deciding finite/infinite life of the part.

2.2.3 Evaluation of the Stiffnesses and strengths

Investigation was carried out through the FEA (MATLAB Code) to learn how the direction of load application and the material orientation angle (for the unenhanced steel) influenced the strength and stiffness. With focus on how the lamina orientation angle influenced the strength in this study, the steel is the matrix. By itself, it has low mechanical properties compared to those of fiber which is the carbon laminates. The matrix influences many mechanical properties of the composite. These properties include transverse modulus with tensile strength, compressive strength, and fatigue strength. The focus was on the elasticity for the composite to reveal the stiffness status, and on the plasticity to understand the state of the strength.

3 RESULTS AND DISCUSSION

3.1 Analytical Fail-Safe Results

In this work, Maximum Principal Stress Vs von Misses Stress: the bending stress, σ based on maximum principal stress is,

$$\sigma = K_f \frac{M_B}{I/c} = 369 \text{ MPa.} \quad (16)$$

This is greater than the endurance limit (235.7 MPa). So, the shaft has finite life. It is also less than the yield stress σ_y (580 MPa); so, there is no yielding. Based on von Mises stress,

$$\sigma = K_f \frac{M_B}{I/c} = 323.8 \text{ MPa} \quad (17)$$

This is again greater than the endurance limit; so, it has finite life, and it is less than the yield stress so, there is no yielding.

3.2 Finite/Infinite Life

Table 1 shows some results obtained analytically to be now used to construct the stress-number of cycle (S-N) curve. These results are plotted in Fig. 2. For steel, from the curve, it is seen that at a certain stress range the curve becomes straight. This means that the shaft is having infinite life below that stress, and finite life above it. The steel fatigue life is highly influenced by the number of micro-mechanical defects.

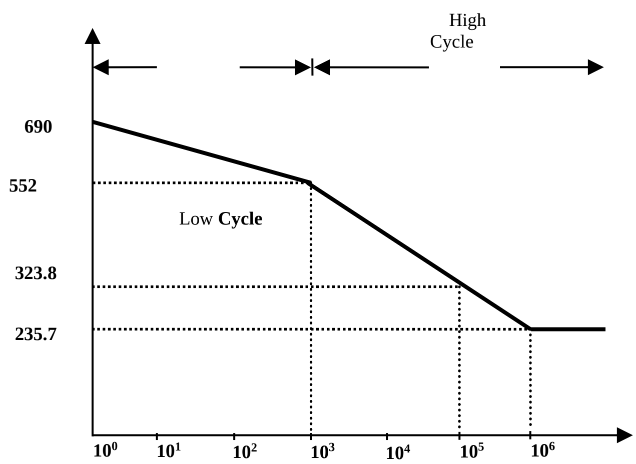

Fig. 2 S-N diagram for the AISI 1020 steel (ferrous material)

In Fig.3, the S-N curve for the fibre-laminate material, shows no well-defined endurance limit. It has no knee and so, has no endurance limit (typical of non-ferrous materials such as the carbon-fiber or typical aluminum alloy). Combining the results of strengths for the ferrous (Fig. 4) and the non-ferrous (Fig. 5) materials, it is seen that the composite shaft will have a stress value below the endurance limit and so, the life is

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enhanced. The two S-N curves in Figs. 4 and 5 obtained in this work compare with the standard curves for typical steel and for non-ferrous metals, respectively.

Table 1: Comparison of Analytical Results on Enhancement of Fatigue Factors with Standard

Symbol	Definition	Value (MPa) This Study	Value (MPa) Standard
σ_e	Endurance Limit	235.7	250
σ_{ut}	Ultimate Tensile Strength	690	
σ_y	Yield Strength	580	
σ_m	Fatigue Strength	552	
k_a	Surface Factor	0.787	
k_b	Size Factor	0.858	0.848
k_t	Stress Concentration Factor		
	(Based on maximum principal stress)	1.88	1.90
	(Based on maximum von Mises Stress)	1.65	
k_f	Fatigue Stress Concentration Factor		
	(Based on maximum principal stress)	1.71	1.5
	(Based on maximum von Mises stress)	1.5	2.0
σ_b	Bending Stress		
	(Based on maximum principal stress)	369	325.9
	(Based on maximum von Mises Stress)	323.8	

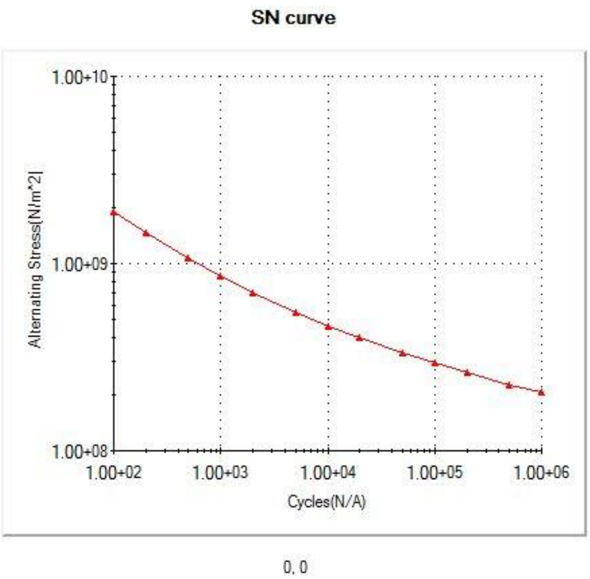

Fig. 3 S-N diagram of the carbon-fiber (non-ferrous material) from FEA

3.3 FEA (MATLAB Code) for Stiffness and Strength Results

In the unenhanced steel shaft, Fig. 4, a maximum of 200 GPa of strength was obtained.

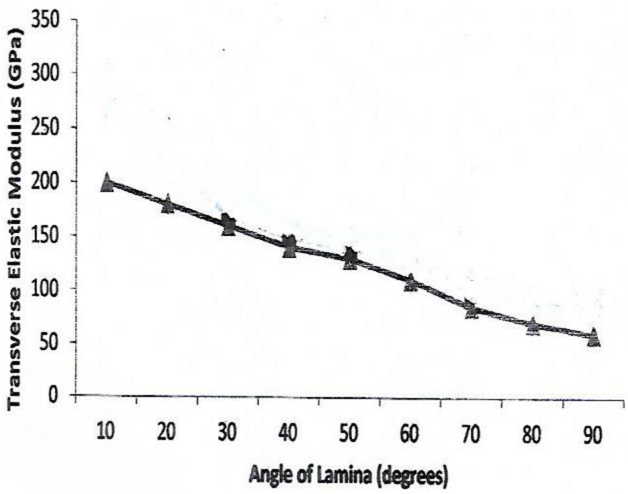

Fig.4: Transverse elastic modulus of the unenhanced steel as functions of lamina angle.

In this work, the enhanced shaft is made of the composite materials (steel being the matrix and carbon, the fiber laminate). The matrix functions include binding the fibers together, protecting fibers from the environment, shielding from

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damage due to handling, and distributing the load to fibers. Enhancement of the material in this work can be seen from the axis of load application and lamina orientation angle in Fig. 5. When a stress is applied to it, the elastic modulus is the slope of its stress–strain curve in the elastic deformation region, a measure of its strength. The figure also shows the relationship between lamina angle and transverse elastic modulus. The elasticity decreases as lamina angle increases (for steel, carbon/steel, and graphite/steel). This indicates higher stiffness. As shown, if the lamina angle decreased to 10 degrees, the composite achieves about 300 MPa of strength, which is an enhancement. If the fibers are oriented in more than one direction, there will be high stiffness (resistance to elastic deformation) and strength (ability to withstand an applied load without failure) in the directions of the fiber orientations.

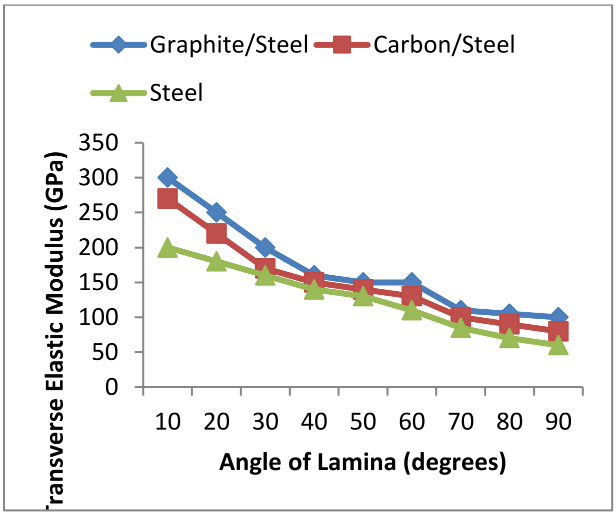

Fig. 5: Transverse elastic modulus of the composites, and steel, as functions of angle of lamina for the composites.

The fatigue life enhancement, obtained by Sanford in the literature, was due to differing conditions of free surface i.e., smooth (polished), machined, and pre-cracked, etc. However, in this work, fatigue life enhancement was achieved by replacing steel with composite.

4 CONCLUSIONS

This paper modeled a shaft subjected to fatigue loads, and presented the procedure for prolonging its life as follows:

(i) Material properties provided primary parameters that aided others.

(ii) The shaft is operated for finite life, and so, high stress amplitude influencing the stiffness. The analysis of results show that the endurance limit is 235.7 MPa which compares with the universal mean endurance limit for typical steel of about 250 MPa. The unique composite material, carbon-fiber composite, shows how fatigue life is improved using FEA tools. The fiber-reinforced composite transmission shaft has good resistance to crack propagation.

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