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Design of Automobile Driveshaft using Carbon/Epoxy and Kevlar/Epoxy Composites

R. Srinivasa Moorthy, Yonas Mitiku & K. Sridhar
School of Mechanical and Industrial Engineering, IoT, Bahir Dar University, Bahir Dar, Ethiopia.

Abstract: - Use of advanced composites has resulted in remarkable achievements in many fields including aviation, marine and automobile engineering, medicine, prosthetics and sports, in terms of improved fatigue and corrosion resistances, high specific strength and specific modulus and reduction in energy requirements owing to reduction in weight. The aim of this work is to replace the conventional steel driveshaft of automobiles with an appropriate composite driveshaft. The conventional driveshafts 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. Carbon/Epoxy and Kevlar/Epoxy composites were designed and analysed for their appropriateness in terms of torsional strength, bending natural frequency and torsional buckling by comparing them with the conventional steel driveshaft under the same grounds of design constraints and the best-suited composite was recommended. Light has been thrown upon the aspects like mass saving, number of plies and ply distribution.

Keywords: - *Carbon/Epoxy, Composite driveshaft, Kevlar/Epoxy, ply distribution.*

I. INTRODUCTION

A composite is a structural material consisting of two or more combined constituents that are combined at a macroscopic level, not soluble in each other. One constituent called *reinforcing* phase in the form of fibres, flakes or particles, is embedded in a continuous *matrix* phase. The inability of monolithic metals and their alloys to meet the complex functional requirements of advanced technologies lead to the use of composites more and more [1]. Generally composite materials have very high specific strength and specific modulus. The strength of graphite epoxy may be the same, but its specific strength is thrice as that of steel. This translates into reduced material and energy costs. Though the material cost is 10-15 times that of steel, manufacturing techniques such as SMC (Sheet Moulding Compound) and SRIM (Structural Reinforcement Injection Moulding) are substantially lowering the cost and production time in manufacturing automobile parts. Unlike metals, composite materials are not isotropic - their properties are not the same in all directions, thus necessitating more material parameters. Nine stiffness and strength constants are needed to conduct mechanical analysis for a single layer of a composite as against four stiffness and strength constants in the case of monolithic materials like steel. Such complexities render structural analysis computationally and experimentally more complicated and highly intensive.

An automotive driveshaft is a rotating shaft that transmits power from the engine to the differential gear of rear wheel drive (RWD) vehicles. Conventional steel driveshafts 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 the span length. But the two-piece steel driveshaft involves three universal joints, an intermediary thrust bearing and a supporting bracket in its assemblage, which increases the total weight of the vehicle [2].

Since one-piece composite driveshaft will suffice in the place of a two-piece steel driveshaft, it substantially reduces the inertial mass. Moreover, a composite driveshaft can be perfectly designed to effectively meet the strength and stiffness requirements. Since composite materials generally have a lower elasticity modulus, during torque peaks in the driveline, the driveshaft can act as a shock absorber. Moreover, the breakage of composite a driveshaft (particularly in SUV's) is less-risky, since it results in splitting up of the fine fibres as compared to the scattering of broken steel parts in various directions [3].

II. DESIGN SPECIFICATIONS

The following specifications were assumed suitably, based on the literature and available standards of automobile driveshafts:

1. The torque transmission capacity of the driveshaft (**T**) = 2000 N-m.
2. The shaft needs to withstand torsional buckling (**T_b**) such that $T_b > T$.
3. The minimum bending natural frequency of the shaft ($f_{nb\,(min\,)}$) = 80 Hz.
4. Outside radius of the driveshaft (**r_o**)= 60 mm.
5. Length of the driveshaft = 1.8 m.

III. DESIGN OF CONVENTIONAL STEEL DRIVESHAFT

First, the conventional steel shaft was designed to facilitate comparison in terms of mass savings. Be it the conventional driveshaft or the composite one, the design should be based on the following criteria:

- Torsional strength
- Torsional buckling and
- Bending natural frequency.

SM45C steel was selected, since it is widely being used for the design of conventional steel shaft. The properties of SM45C steel are:

Young's modulus (E)	=	207 GPa
Shear modulus (G)	=	80 GPa
Poisson's ratio (ν)	=	0.3
Density of steel (ρ)	=	7600 kg/m^3
Yield strength (σ_y)	=	370 MPa.

3.1 Torsional strength

Since the primary load on a driveshaft is torsion, the maximum shear stress **(τ_{max})** at the outer radius **(r_o)** of the shaft is given by:

$$\frac{\tau_{max}}{F.S.} = \frac{Tr_o}{J} \qquad (1)$$

Substituting for **J**:

$$\frac{\tau_{max}}{F.S.} = \frac{32Tr_o}{\pi\left[d_0{}^4 - d_i{}^4\right]} \qquad (2)$$

where,

T is the maximum torque applied in N-m

J is the polar area moment of inertia in m^4 and

d_0 and d_i are outer and inner diameters of the shaft in m.

Assuming $\tau_{max} = 80$ MPa and a factor of safety (**F. S.**) of 3,

$$d_i = 0.112735\, m.$$

Hence, the inner radius is, $r_i = 0.056368\, m.$

Thus the wall thickness of the hollow steel shaft:

$$\mathbf{t = r_0 - r_i} \qquad (3)$$
$$= 3.6325 \text{ x } 10^{-3} \text{ m.}$$

3.2 Torsional buckling

A shaft is considered as a long shaft, if [4]:

$$\left(\frac{1}{\sqrt{(1-\vartheta^2)}}\right)\frac{L^2t}{(2r)^3} > 5.5 \qquad (4)$$

where, **r** is the mean radius, such that:

$$\mathbf{r} = \left(\frac{r_i + r_o}{2}\right) \qquad (5)$$
$$= 0.058184\, m.$$

Substituting,

$$\left(\frac{1}{\sqrt{(1-0.3^2)}}\right)\frac{(1.8)^2(0.0036325\,)}{(2\times 0.058184\,)^3} = 7.8294\ (> 5.5)$$

For a long shaft, the torsional buckling capacity:

$$\mathbf{T_b = \tau_{cr}(2\pi r^2 t)} \qquad (6)$$

where, the critical stress **(τ_{cr})** is given by,

$$\tau_{cr} = \left[\frac{E}{3\sqrt{2}(1-\vartheta^2)^{3/4}}\right]\left({}^{t}/_{r}\right)^{3/2} \qquad (7)$$

Substituting,

$$\tau_{cr} = 81.6875 \times 10^7 \mathrm{N/m^2} \text{ and}$$
$$T_b = 63.11735 \times 10^3 \mathrm{N-m.}$$

Thus,

$$T_b > T.$$

3.3 Bending Natural Frequency

According to Bernoulli-Euler beam theory, by neglecting shear deformation and rotational inertia effects, the bending natural frequency of a rotating shaft is given by:

$$\mathbf{f_{nb}} = \frac{\pi p^2}{2L^2}\sqrt{\frac{EI_x}{m'}} \qquad \textbf{(8)}$$

where,

m′ is mass per unit length in kg/m

$\mathbf{I_x}$ is area moment of inertia in x-direction (longitudinal) in m^4.

$$\mathbf{I_x} = \frac{\pi}{64}\left(\mathbf{d_0^4} - \mathbf{d_i^4}\right) \qquad \textbf{(9)}$$
$$= 2.25 \text{ x } 10^{-6} \text{ m}^4.$$
$$\mathbf{m'} = \rho\left(\pi/4\right)\left[\mathbf{d_0^2} - \mathbf{d_i^2}\right] \qquad \textbf{(10)}$$
$$= 10.0925\ kg/m.$$

Substituting these values,

$$f_{nb} = 104.148 \text{ Hz.}$$

Thus,

$$f_{nb} > f_{nb\,(min)}.$$

Thus the designed SMC45 steel driveshaft meets all the requirements.

The total mass of the shaft is:

$$\mathbf{m} = \mathbf{m'L} \qquad \textbf{(11)}$$

Thus,

$$m = 18.1665 \text{ kg.}$$

IV. DESIGN OF COMPOSITE DRIVESHAFTS

Only $0°, \pm 45°\ and\ 90°$ were considered for the composite ply orientations, owing to their specific advantages.

4.1 Design of Carbon/Epoxy Driveshaft

60% fibre volume fraction Carbon/Epoxy shaft ($V_f = 60\%$) with standard ply thickness of 0.13 mm was selected.

4.1.1 Torsional strength

$$\frac{\tau_{max}}{F.S.} = \frac{T}{2\pi r^2 t} \qquad \textbf{(12)}$$

where,

r is the mean radius of the shaft.

Since the nature of loading is pure shear, 70% of the plies can be set at $\pm 45°$ and the remaining 30% at 0º and 90º orientations.

From Fig. 1,

$$\tau_{max} = 293 \text{ MPa}$$

For a factor of safety **(F. S.)** of 6,

$$r^2 t = 6.5183 \times 10^{-6} m^3.$$

Thus,

$$t \geq 1.8106 \times 10^{-3} m.$$

Since the thickness of each ply is 0.13 mm,

$$n = 1.8106 \times 10^{-3}/0.13 \times 10^{-3}$$
$$= 13.93 \cong 14.$$

Hence the corrected values are:

$$t = 1.82 \text{ x } 10^{-3} \text{ m}$$
$$r_i = 0.05818 \text{ m and}$$
$$r = 0.05909 \text{ m.}$$

4.1.2 Torsional buckling

Considering the hollow composite shaft as an isotropic cylindrical shell, the buckling torque is given by:

$$\mathbf{T_b} = \mathbf{2\pi r^2 t(0.272)}\left(\mathbf{E_x E_y^3}\right)^{1/4}\left(\frac{\mathbf{t}}{\mathbf{r}}\right)^{3/2} \qquad \textbf{(13)}$$

where,

$\mathbf{E_x}$ and $\mathbf{E_y}$ are the Young's modulii in 'x' and 'y' directions respectively.
From Fig. 2,

$$E_x = 38709.5 \text{ MPa}.$$

By permuting (interchanging the percentages of 0° and 90° plies),

$$E_y = 38709.5 \text{ MPa}.$$

Upon substitution,

$$T_b = 2272.49 \text{ N} - \text{m}. \ (> T)$$

4.1.3 Bending natural frequency

$$\mathbf{f_{nb}} = \frac{\pi}{2L^2}\sqrt{\frac{E_x I_x}{m'}} \tag{14}$$

From TABLE I, the density of Carbon/Epoxy laminate (ρ) = 1530 kg/m^3.
Hence,

$$I_x = 1.179957 \times 10^{-6} \text{ m}^4 \text{ and}$$
$$m' = 1.03385 \ kg/m.$$

Upon substitution,

$$f_{nb} = 101.903 \text{ Hz } (> 80 \text{ Hz}).$$

The total mass of Carbon/Epoxy composite shaft is,

$$m = 1.86093 \text{ kg}.$$

The ply distribution for the Carbon/Epoxy driveshaft is shown in Fig. 3.
Accordingly, the ply orientation is,

$$[0°/\pm45^{\circ}_2/90°/\pm45°/90°/\pm45^{\circ}_2/0°].$$

4.2 Design of Kevlar/Epoxy Driveshaft

Setting 70% of the plies in $\pm45°$ and the remaining 30% in 0º and 90º, similar to the previous approach, from the respective figures [5],

$$\tau_{max} = 95 \text{ MPa}$$
$$E_x = 23900 \text{ MPa and}$$
$$E_y = 23900 \text{ MPa}.$$

Using a factor of safety (F.S.) of 6, for $V_f = 60\%$ and ply thickness = 0.13 mm,

$$t \geq 5.5844 \times 10^{-3} \text{m}.$$
$$n = 42.96 \cong 44.$$

The corrected values are:

$$t = 0.00572 \text{ m}$$
$$r_i = 0.05428 \text{ m and}$$
$$r = 0.05714 \text{ m}.$$

The calculated values of buckling torque, bending natural frequency and the total mass are:

$$T_b = 24161 \text{ N} - \text{m}$$
$$f_{nb} = 101.903 \text{ Hz and}$$
$$m = 4.99 \text{ kg}.$$

The ply distribution for the Kevlar/Epoxy driveshaft is shown in Fig. 4.
Accordingly, the ply orientation with mid-plane symmetry is,

$$[0^{\circ}_2/90^{\circ}_2/\pm45^{\circ}_6/90^{\circ}_2/0^{\circ}_2/-45^{\circ}/+45^{\circ}]_s.$$

V. RESULTS AND DISCUSSION

From the detailed analysis, the key results of wall thickness, torsional buckling capacity, bending natural frequency, number of plies and total mass for SM45C steel (as applicable), Carbon/Epoxy and Kevlar/Epoxy driveshafts were extracted and summarized in TABLE II.

TABLE II reveals that use of Carbon/Epoxy results in a mass saving of 89.756% when compared to the conventional SM45C steel driveshaft, whereas Kevlar/Epoxy results in 72.53%. Obviously, the number of plies needed for Carbon/Epoxy is 14 with 1.82 mm wall thickness as compared to 44 plies with 5.72 mm wall thickness in the case of Kevlar/Epoxy. Moreover, the torsional buckling capacity and bending natural frequency are adequate enough to meet the design requirements in the case of Carbon/Epoxy driveshaft.

VI. CONCLUSION

Precisely, for the specifications chosen, using Carbon/Epoxy driveshaft in the place of conventional driveshaft will lead to an appreciable mass saving of 89.756% with barely half of the wall thickness of conventional steel shaft. Though the mass saving is substantial in both the composites considered, making either of the composites a better choice for the conventional high quality SM45C steel, using Carbon/Epoxy for making automotive driveshaft has multiple advantages as mentioned above.

This work relies purely upon analytical calculations and use of ply distribution tables/graphs pertaining to 60% volume fraction and 0.13 mm ply thickness. The approach can be extended to other widely used composites like Glass/Epoxy and Boron/Epoxy to check their suitability. This approach throws light upon ply distribution in standard orientations of 0º, 90º, +45º and -45º for the composite considered. The effect of varying ply staking sequence on the performance of composites can be found by using computational softwares. Moreover, considering the material and manufacturing cost will give better grounds to compare the overall efficacy, thus resulting in an appropriate selection of the best fibre/matrix combination for making automotive driveshafts.

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FIGURES AND TABLES

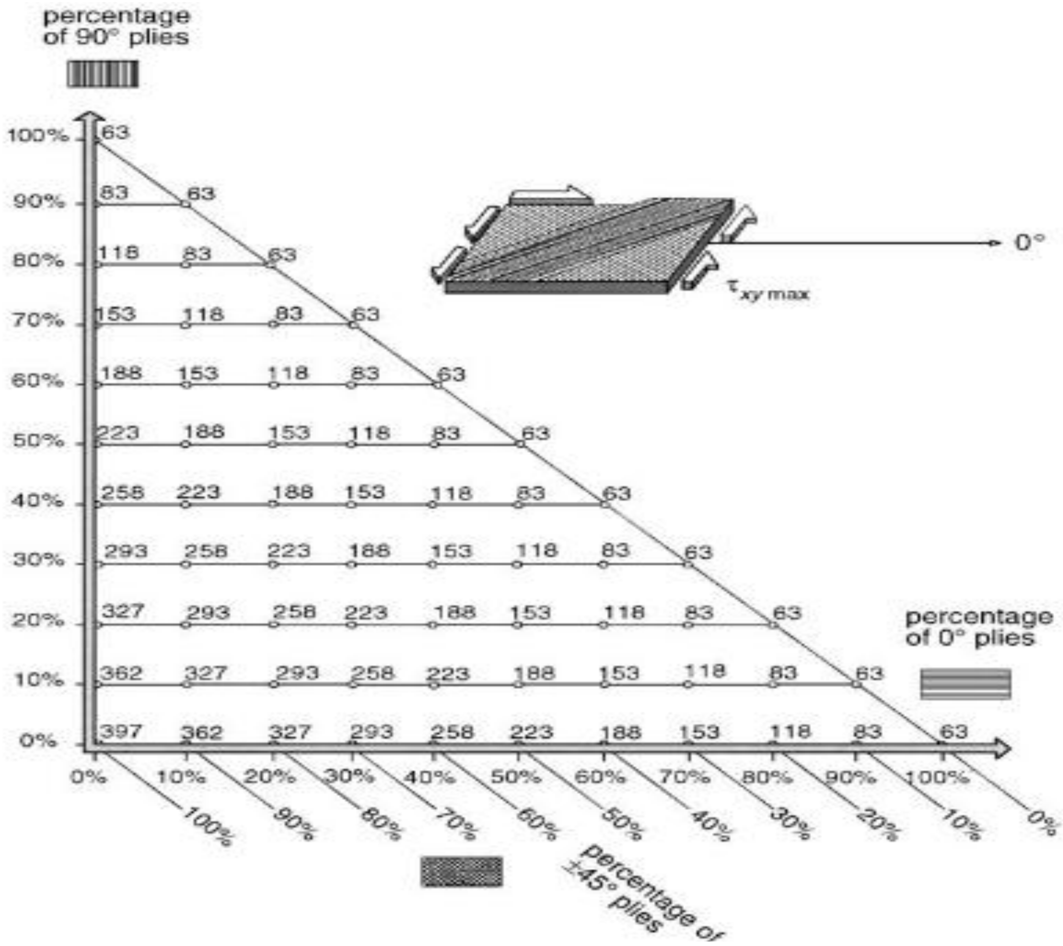

Figure 1: Maximum shear stress (τ_{max}) as a function of ply percentages for Carbon/Epoxy Laminate (V_f = 60%; Ply thickness = 0.13 mm) [5]

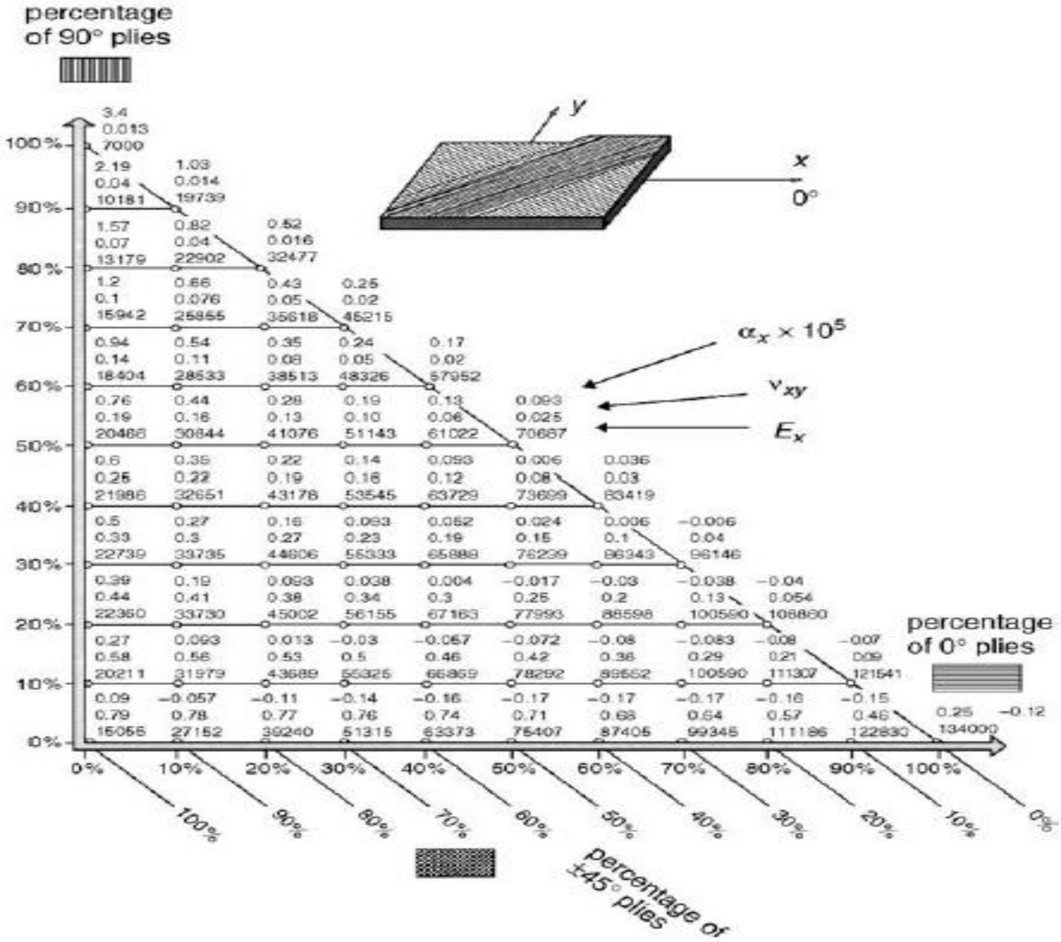

Figure 2: Young's Modulus (E_x in MPa), Poisson's ratio (υ_{xy}) and Co-efficient of thermal expansion (α) as functions of ply percentages for Carbon / Epoxy Laminate (V_f = 60%; Ply thickness = 0.13 mm) [5]

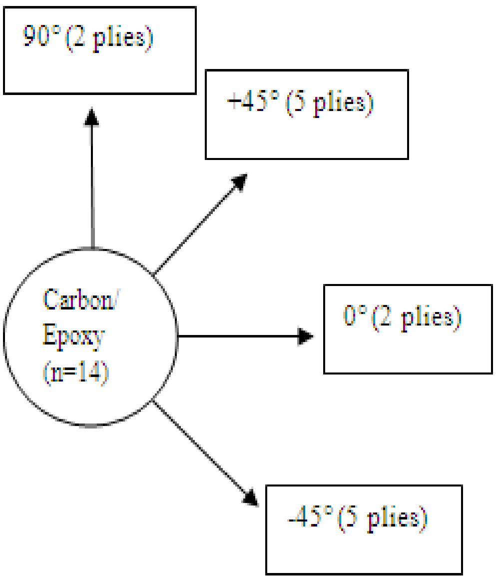

Figure 3: Ply distribution for Carbon/Epoxy driveshaft

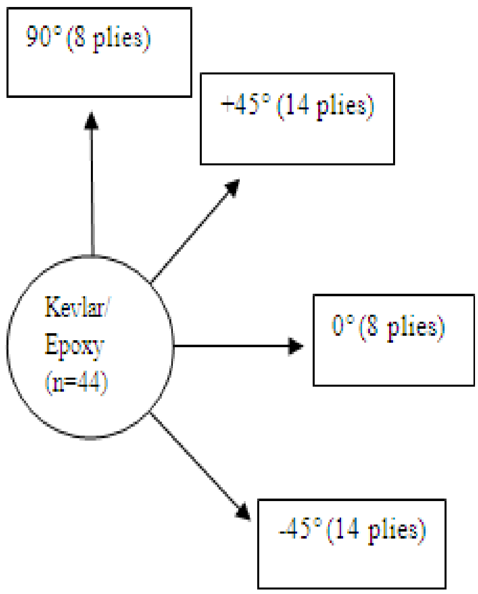

Figure 4: Ply distribution for Kevlar/Epoxy driveshaft

TABLE I: Properties of Fibre/Epoxy Laminates (V_f = 60%)

Property	***Carbon***	***Kevlar***
Specific mass (kg/m^3)	1530	1350
Longitudinal tensile fracture strength (MPa)	1270	1410
Longitudinal compressive fracture strength (MPa)	1130	280
Transverse tensile fracture strength (MPa)	42	28
Transverse compressive fracture strength (MPa)	141	141
Poisson's ratio	0.25	0.34

TABLE II: Comparison of SM45C steel, Carbon/Epoxy and Kevlar/Epoxy driveshafts

S. No.	***Material***	***t (mm)***	***T_b (N-m)***	***f_{nb} (Hz)***	***n***	***m (kg)***
1	SM45C steel	3.63	63117.35	104.148	-	18.1665
2	Carbon/Epoxy	1.82	2272.49	101.903	14	1.86093
3	Kevlar/Epoxy	5.72	24161	267.370	44	4.99

AUTHORS' BIBLIOGRAPHY

R. Srinivasa Moorthy obtained his Masters degree in Mechanical Engineering Design from Kongu Engineering College, Perundurai, Tamilnadu, India; has 17 + years of teaching experience; worked in Erode Sengunthar Engineering College, Erode, Tamilnadu, India, for 10 + years; worked in Eritrea Institute of Technology, Eritrea, North-East Africa, for 4 years; currently working as a Lecturer in School of Mechanical and Industrial Engineering, Institute of Technology, Bahir Dar University, Bahir Dar, Ethiopia.

Yonas Mitiku Degu received his B.Sc. in Mechanical Engineering from Bahir Dar University, Bahir Dar, Ethiopia in 2005; pursued M.Sc. in Applied Mechanics (Mechanical Design) in Addis Ababa University, Addis Ababa, Ethiopia in 2008; currently working as Assistant Professor and Director of School of Mechanical and Industrial Engineering, Institute of Technology, Bahir Dar University, Bahir Dar, Ethiopia.

K. Sridhar received his B.E. degree in Mechanical Engineering from Anna University, Chennai, India and M.E. from Anna University, Coimbatore, India; worked as a Lecturer/Senior Lecturer in S.S.M. College of Engineering, Komarapalayam, Tamilnadu, India for 4 years; currently working as a Lecturer in School of Mechanical and Industrial Engineering, Institute of Technology, Bahir Dar University, Bahir Dar, Ethiopia, from October 2011.