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Optimization Of Automotive Composite Drive Shaft Using Cuckoo Search Algorithm

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Abstract:

Due to advances in automotive technology, greater interest has been developed in the design of lightweight and cost-effective vehicle components to meet the vehicle design requirements and to improve the cost and fuel efficiency. In this research, new nature-inspired metaheuristic algorithm, called the Cuckoo Search Algorithm (CS) algorithm is applied for the design optimization of automotive composite drive shaft and it is found that better results can be obtained. Thus, CS is a suitable optimization method for obtaining the best solution for the structural design optimization problems and light vehicle designs

Keywords: Cuckoo Search Algorithm (CS), design optimization, stacking sequence, Composite drive shaft.

1. Introduction

In the rear-wheel-drive automobile, power developed by the engine is transmitted to the final drive through a system consists of a gearbox, clutch, drive shaft and a differential [1]. More weight, critical speed and vibration problems are the major disadvantages of Conventional metallic drive shafts. Besides these conventional metallic drive shafts are manufactured in two pieces to meet the functional requirements. Two-piece drive shaft made of steel consists of three universal joints, a center supporting bearing and a bracket, which increases the total weight of an automotive vehicle and decreases fuel efficiency. Andrew Pollard [2] proposed the polymer Matrix composites in driveline applications. Because carbon fibre epoxy composites have more than four times specific stiffness (E/ρ) of steel or aluminium materials, hence it is possible to manufacture composite drive shafts in one-piece without whirling vibration over 9200 rpm. The composite drive shaft has many benefits such as reduced weight, better fuel efficiency, less noise and vibration. Bauchau et al. [3] have proposed a theoretical model of torsional buckling behaviour of circular cylindrical shells made of laminated composites based on general shell theory and to validate this model by comparing its predictions with experimental results. Lee et.al.

[4] designed and manufactured hybrid aluminium/composite shafts. They have noticed that the manufactured one-piece hybrid automotive drive shaft has 75% less weight and 160% increase in torque capability compared to the conventional two-piece steel drive shaft configuration. The torsional stability of a composite drive shaft under torsion was studied by Shokrieh et al. [5] studied the effect of fibre orientations and the stacking sequences on torsional strength. They concluded that the stacking sequence and number of layers of composite drive shaft strongly affect the buckling torque and increasing the applied torque decreases the natural torsional frequencies. Badie et al. [6] analysed the effects of fibre orientation angles and stacking sequence on the torsional stiffness, natural frequency, torsional buckling strength and failure modes of composite tubes using both FEA and experimental techniques. They found that carbon fibres have the major contribution over glass in increasing the torsional stiffness. The 45° fibre orientation was found to be the optimum in increasing the torsional stiffness while carbon fibre tubes experienced higher fracture strain than that of glass fibres.

The torsional behaviour of hybrid composite shafts was studied with numerical and experimental approach by Ercan Sevkat et al. [7]. The results obtained by FEA were validated using experimental test results and exhibited quite similar torsional behaviour. Rastogi [8] used a hybrid of carbon/epoxy and glass/epoxy to optimize the cost versus performance requirements. Shinde et al. [9] designed a glass-epoxy composite drive shaft for light motor vehicles to examine the torsional stability, natural frequency and critical speed. The results obtained were validated with a steel drive shaft. From the results it is found that, 73% weight reduction was achieved with composite drive shaft compared to metallic drive shaft. Nadeem et al. [10] conduct out a review on the design, optimization and perform evaluation of of the composite drive shafts made of different materials, such as carbon, glass, Kevlar, and boron with epoxy resin. T. Rangaswamy and S. Vijayarangan [11] have proposed the optimization procedure to design a composite drive shaft. The designed drive shafts are optimized using genetic algorithm (GA) for better stacking sequence, better torque transmission capacity, and bending vibration characteristics. They noticed that considerable amount of weight was achieved by using composite materials and optimization technique compared to conventional steel. K. Manjunath and T. Rangaswamy [12] have performed the stacking sequence optimization of composite driveshaft using particle swarm optimization algorithm (PSOA) and they noticed that PSOA yields better results than GA. Xin-She Yang and Suash Deb developed latest nature-inspired metaheuristic algorithm, using Cuckoo search (CS) in engineering area to solve engineering design optimization problems in 2009 [13]. CS is based on the brood parasitism of some cuckoo species. Further, this algorithm is greatly enhanced by the so-called Lévy flights, rather than by simple isotropic random walks. Recent studies show that CS is potentially far more efficient than PSOA and GA. In the current work an attempt is made to investigate the suitability of composite material for the purpose of automotive power transmission applications. A single -piece composite drive shaft for rear-wheel-drive automobile is optimally designed and analyzed using CS and ABAQUS software respectively for E-Glass/Epoxy and HM-Carbon/Epoxy composites with weight minimization as a objective function and subjected to the constraints such as torque transmission, torsional buckling strength capabilities and natural bending frequency.

2. Problem Specification

The aim of the present work is to replace a two-piece steel drive shaft with a single piece fibre reinforced plastic (FRP) shaft for the rear-wheel-drive system of light motor vehicle. The Torsional strength of the drive shaft for passenger cars, small trucks, and vans should be larger than 3,500 Nm and fundamental natural bending frequency of the drive shaft should be larger than 6,500 rpm to avoid whirling vibration. The outer diameter of the drive shaft should not more than 90 mm due to space constraints.

3. Design Requirements of a Composite Hollow Drive Shaft

The single piece composite drive shaft and two-dimensional drawing of the shaft is shown in Figure 1 and Figure 2 respectively. The drive shaft of transmission system is designed for the specified design requirements as shown in Table 1. The mechanical properties of conventional steel shaft and composite materials considered for the CS of composite drive shafts are given in Table 2 and Table 3 respectively.

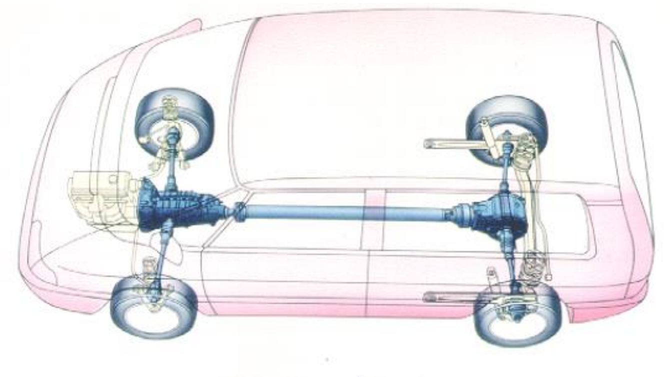

Figure 1 Single Piece Composite Drive shaft

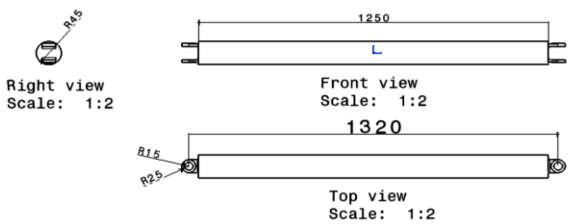

Figure 2 2D Drawing of Single piece composite hollow Drive Shaft

Table 1 Design Requirements of the automobile Drive Shaft

Parameters	Values
Outer diameter d_o (mm)	90
Length L (mm)	1250
Torque transmitted T (N-mm)	3500×10^3
Speed of Transmission N (rpm)	6500

Table 2 Mechanical Properties of Conventional Steel shaft (SM45C)

Mechanical properties	**Symbol**	**Steel (SM 45C)**
Young's Modulus (GPa)	E	207
Shear Modulus (GPa)	G	80
Poisson Ratio	γ	0.3
Density (Kg/m^3)	ρ	7600
Yield strength (MPa)	S_y	370
Shear Strength (MPa)	S_s	370

Table 3 Mechanical Properties of Each Lamina of the Laminates

Property	**Units**	**E-Glass/Epoxy**	**HM Carbon/ Epoxy**
E_{11}	GPa	50.0	190.0
E_{22}	GPa	12.0	7.7
G_{12}	GPa	5.6	4.2
ν_{12}		0.3	0.3
$\sigma^T{}_1=\sigma^C{}_1$	MPa	800.0	870.0
$\sigma^T{}_2=\sigma^C{}_2$	MPa	40.0	54.0
τ_{12}	MPa	72.0	30.0
ρ	Kg/m^3	2000.0	1600.0
V_f		0.6	0.6
t_k	mm	0.4	0.12

3.1 Torque Transmission Capability

The second analytical step determines the ultimate torsional strength. From classical laminated plate theory, the general relationship between in-plane stress resultant, moment, off-axis strain and curvature is found.

3.2 Stress-Strain Relations for Unidirectional Lamina

If the lamina is thin and if no out-of-plane loads are applied, it is treated as the plane stress problem. Hence, it is possible to reduce the 3-D problem into 2-D problem. For unidirectional 2-D lamina, the stress-strain relationship in terms of principal material directions is given by

$$\begin{pmatrix}\sigma_1\\ \sigma_2\\ \tau_{12}\end{pmatrix}=\begin{pmatrix}Q_{11} & Q_{12} & 0\\ Q_{12} & Q_{22} & 0\\ 0 & 0 & Q_{66}\end{pmatrix}\begin{pmatrix}\varepsilon_1\\ \varepsilon_2\\ \gamma_{12}\end{pmatrix} \qquad (1)$$

Where σ, τ, γ and ε represent stresses and strains in material directions. The matrix Q is known as reduced stiffness matrix for the layer and its terms are given by

$$Q_{11} = \frac{E_{11}}{1 - \nu_{12}\nu_{12}}$$

$$Q_{12} = \frac{\nu_{12}E_{22}}{1 - \nu_{12}\nu_{12}} \qquad (2)$$

$$Q_{22} = \frac{E_{22}}{1 - \nu_{12}\nu_{12}}$$

3.3 Stress-Strain Relation in Arbitrary Orientation

For an angle-ply lamina, where fibers are oriented at an angle with the positive X-axis (longitudinal axis of shaft), and the stress strain relationship is given by

$$\begin{pmatrix} \sigma_1 \\ \sigma_2 \\ \tau_{12} \end{pmatrix} = \begin{pmatrix} \overline{Q}_{11} & \overline{Q}_{12} & \overline{Q}_{16} \\ \overline{Q}_{12} & \overline{Q}_{22} & \overline{Q}_{26} \\ \overline{Q}_{16} & \overline{Q}_{26} & \overline{Q}_{66} \end{pmatrix} \begin{pmatrix} \varepsilon_x \\ \varepsilon_y \\ \gamma_{xy} \end{pmatrix} \qquad (3)$$

Where σ and ε represent normal stresses and strains in X, Y and XY directions respectively and bar over Q_{ij} matrix denotes transformed reduced stiffnesses. Its terms are individually given by

$$\begin{aligned}
\overline{Q}_{11} &= Q_{11}C^4 + Q_{22}S^4 + 2(Q_{12} + 2Q_{66})S^2C^2 \\
\overline{Q}_{12} &= (Q_{11} + Q_{22} - 4Q_{66})S^2C^2 + Q_{12}(C^4 + S^4) \\
\overline{Q}_{16} &= (Q_{11} - Q_{22} - 2Q_{66})SC^3 - (Q_{22} - Q_{12} - 2Q_{66})CS^3 \\
\overline{Q}_{22} &= Q_{11}S^4 + Q_{22}S^4 + 2(Q_{11} + 2Q_{66})S^2C^2 \\
\overline{Q}_{26} &= (Q_{11} - Q_{12} - 2Q_{66})CS^3 - (Q_{22} - Q_{12} - 2Q_{66})C^3S \\
\overline{Q}_{66} &= (Q_{11} + Q_{22} - 2Q_{12} - 2Q_{66})S^2C^2 + Q_{66}(S^4 + C^4)
\end{aligned} \qquad (4)$$

With C=Cosɵ and S=Sinɵ

Figures 3 & 4 shows fibre orientation angle with respect to the driveshaft axis for single lamina filament wound composite shaft.

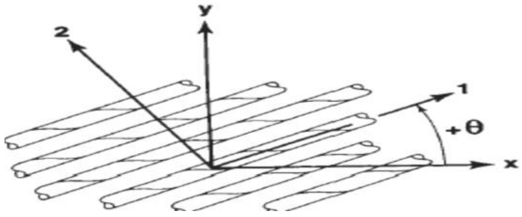

Figure 3 Shows relation between material coordinate system and XY coordinate system

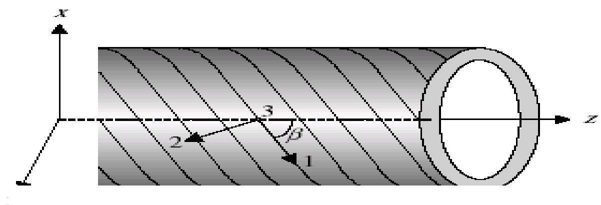

Figure 4 Single lamina of a filament-wound shaft

3.4 Force and Moment Resultants

For a symmetric laminate, the B matrix vanishes and the in plane and bending stiffnesses are uncoupled.

$$\begin{Bmatrix} N_x \\ N_y \\ N_{xy} \end{Bmatrix} = \begin{bmatrix} A_{11} & A_{12} & A_{16} \\ A_{12} & A_{22} & A_{26} \\ A_{16} & A_{26} & A_{66} \end{bmatrix} \begin{Bmatrix} \varepsilon_x^o \\ \varepsilon_y^o \\ \gamma_{xy}^o \end{Bmatrix} \quad (5)$$

$$\begin{Bmatrix} M_x \\ M_y \\ M_{xy} \end{Bmatrix} = \begin{bmatrix} D_{11} & D_{12} & D_{16} \\ D_{12} & D_{22} & D_{26} \\ D_{16} & D_{26} & D_{66} \end{bmatrix} \begin{Bmatrix} K_x^o \\ K_y^o \\ K_{xy}^o \end{Bmatrix} \quad (6)$$

Where $N_{x,}$ $N_{Y,}$ N_{xy} and $M_{x,}$ $M_{y,}$ M_{xy} in (5.5) and (5.6) are forces and moments per unit width

$$A_{ij} = \sum_{k=1}^{n} \left(\overline{Q}_{ij}\right)_k (h_k - h_{k-1}) \quad (7)$$

$$B_{ij} = \frac{1}{2} \sum_{k=1}^{n} \left(\overline{Q}_{ij}\right)_k (h_k^2 - h_{k-1}^2) \quad (8)$$

$$D_{ij} = \frac{1}{3} \sum_{k=1}^{n} \left(\overline{Q}_{ij}\right)_k (h_k^3 - h_{k-1}^3) \quad (9)$$

Where represent $A_{ij,}$ B_{ij} and D_{ij} are extensional, coupling and bending stiffnesses having i, j = 1, 2...6 respectively, h_k is the distance between the neutral fiber to the top of the K^{th} layer.

Strains in the reference surface are given by:

$$\begin{Bmatrix} \varepsilon_x^o \\ \varepsilon_y^o \\ \gamma_{xy}^o \end{Bmatrix} = \begin{bmatrix} a_{11} & a_{12} & a_{16} \\ a_{12} & a_{22} & a_{26} \\ a_{16} & a_{26} & a_{66} \end{bmatrix} \begin{Bmatrix} N_x \\ N_y \\ N_{xy} \end{Bmatrix} \quad (10)$$

Where

$$\begin{bmatrix} a_{11} & a_{12} & a_{16} \\ a_{12} & a_{22} & a_{26} \\ a_{16} & a_{26} & a_{66} \end{bmatrix} = \begin{bmatrix} A_{11} & A_{12} & A_{16} \\ A_{12} & A_{22} & A_{26} \\ A_{16} & A_{26} & A_{66} \end{bmatrix}^{-1} \quad (11)$$

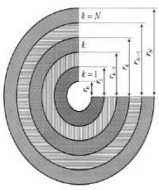

Figure 5 N-layered laminate [16,17]

3.5 Elastic Constants of the Composite Drive Shaft Tubes

$$E_x = \frac{1}{t}\left[A_{11} - \frac{A_{12}^2}{A_{22}}\right] \quad \text{or} \quad E_x = \frac{1}{a_{11}}\frac{1}{t} \quad = \text{Young's Modulus of the shaft in axial direction}$$

$$E_y = \frac{1}{t}\left[A_{22} - \frac{A_{12}^2}{A_{11}}\right] \quad \text{or} \quad E_y = \frac{1}{a_{11}}\frac{1}{t} \quad = \text{Young's Modulus of the shaft in hoop direction} \quad (12)$$

$$G_{xy} = \frac{A_{66}}{t} \quad \text{or} \quad G_{xy} = \frac{A_{66}}{t}\frac{1}{t} = \text{Rigidity Modulus of the shaft in xy plane}$$

When the composite hollow drive shaft is subjected to a torque T, then the resulting forces acting on the laminated hollow shaft by considering the effect of centrifugal forces due to rotation are given by

$$N_x = 0$$

$$N_{xy} = \frac{T}{2\pi r^2} \quad \text{Effect of centrifugal force} \quad (13)$$

Where ρ is the density, t is the thickness, r mean radius and ω is the angular velocity of the composite shaft.

Maximum stress theory is used to calculate torque transmission capability in this study. By knowing the stresses in each ply, the failure of the laminate is determined by using the first ply failure criteria. That is, the laminate is assumed to fail when the stress induced is the first ply reaches its allowable failure stress.

3.6 Torsional Buckling Strength

For a long drive shaft, the effect of torsional buckling can be very significant as they have large slenderness ratio. An empirical relation [16] for approximating the buckling torque is given by

$$T_{cr} = (2\pi r^2 t) \times (0.272) \times (E_x E_y^3)^{0.25} \times (t/r)^{1.5} \quad (14)$$

Where E_x and E_y are longitudinal and transverse stiffness of the composite shaft, t is the thickness and r is the average radius. From Eqn. 5.14, it is found that the torsional buckling capability of a composite shaft is strongly dependent on the thickness of composite shaft and the average modulus in the hoop direction.

3.7 Lateral Vibration or Bending Vibration

The shaft is considered as a simply supported beam undergoing transverse vibration or can be idealized as a pinned-pinned beam. Bending natural frequency can be found using Timoshenko beam theory. It considers both transverse shear deformation as well as rotary inertia effects. Natural frequency based on the Timoshenko beam theory is given by [16].

$$f_{nt} = k_s \frac{30\pi p^2}{L^2}\sqrt{\frac{Er^2}{2\rho}} \quad (15)$$

(16)

$$\frac{1}{K_s^2} = 1 + \frac{p^2 \pi^2 r^2}{2L^2}\left[1 + \frac{f_s E}{G}\right]$$

Where f_{nt} is fundamental bending natural frequency and p = 1 for first mode. E, G and ρ are material properties; r mean radius of steel shaft and f_s is shape factor (equals to 2) for hollow circular cross-sections. (17)

Critical speed can be calculated as $N_{crt} = 60 f_{nt}$

4. Cuckoo search optimization algorithm (CS)

In this paper, the CS is used to solve structural design optimization problems. The Cuckoo Search optimization algorithm (CS) is inspired by some species of a bird family called Cuckoo because of their special life style and aggressive reproduction strategy. These species lay their eggs in the nests of other host birds (almost other species) with amazing abilities such as selecting the recently spawned nests and removing existing eggs that increase hatching probability of their eggs. On the other hand, some of host birds are able to combat this parasite behavior of Cuckoos and throw out the discovered alien eggs or build their new nests in new locations. This algorithm contains a population of nests or eggs. For simplicity, the following representations are used, where each egg in a nest represents a solution and a Cuckoo egg represents a new one. If the Cuckoo egg is very similar to the hosts, then this Cuckoo egg is less likely to be discovered; thus, the fitness should be related to the difference in solutions. The aim is to employ the new and potentially better solutions (Cuckoos) to replace a not so- good solution in the nests [13, 15]. For simplicity in describing the CS, the following three idealized rules are utilized

a) Each Cuckoo lays one egg at a time and dumps it in a randomly chosen nest
b) The best nests with high quality of eggs are carried over to the next generations
c) The number of available host nests is constant, and the egg, which is laid by a Cuckoo, is discovered by the host bird with a probability of pa in the range of [0, 1].

The later assumption can be approximated by the fraction p_a of the n nests are replaced by new ones (with new random solutions). With these three rules, the basic steps of the CS can be summarized as the pseudo code shown in Figure 6. In the first step according to the pseudo code, one of the randomly selected nests (except the best one) is replaced by a new solution, which is produced by random walk with a Lévy flight around the so far best nest, considering the quality. But in the new version, all of the nests except the best one are replaced in one step by new solutions.

```
Objective function f (x), x = (x1, . . ., xd)T
Generate initial population of n host nests xi (i = 1, b. . . n)
While (t <MaxGeneration) or (stop criterion)
   Get a cuckoo randomly
   Generate a solution by Lévy flights [e.g.Eqn. (9.2)]
   Evaluate its solution quality or objective value fi
   Choose a nest among n (say, j) randomly
   if (fi < fj ), then
       Replace j by the new solution i
   end if
   A fraction (Pa) of worse nests are abandoned and
   New nests/solutions are built / generated by Eqn. (9.1)
   Keep the best solutions (or nests with quality solutions);
   Rank the solutions and find the current best
   Update t ← t + 1
end while
Postprocess results and visualization
```

Figure 6 Pseudo code of the cuckoo search for a minimization problem.

To generate new solutions $x_i^{(t+1)}$ for the ith Cuckoo, a Lévy flight is performed using the equation: $x_i^{(t+1)} = x_i(t) + \alpha \cdot S$ (17)

where $\alpha > 0$ is the step size parameter and should be chosen considering the scale of the problem, is set to unity in the CS and decreases function as the number of generations increases in the modified CS[14]. It should be noted that in this new version, the current positions of the solutions are used instead of the best solution so far as the origin of the Levy flight is concerned. The step size is considered as 0.1 in this work, because it results in an efficient performance of algorithm in our example. The parameter S is the length of a random walk with Levy flights according to Mantegna's algorithm as described in Equation (18). In the second step, the p_a fraction of the worst nests is discovered and replaced by new ones. However, in the new version, the parameter pa is considered as the probability of a solution's component to be discovered. Therefore, a probability matrix is produced as

$$P_{ij} = \begin{cases} 1 \text{ if } rand < pa \\ 0 \text{ if } rand \geq pa \end{cases} \quad (18)$$

where rand is a random number in [0, 1] interval and $P_{i,j}$ is the discovering probability for the j^{th} variable of the i^{th} nest. Then, all the nests are replaced by new ones produced by random walks (point-wise multiplication of random step sizes with probability matrix) from their current positions according to quality.

5. Design Optimization

5.1 Objective Function

The main objective for the design optimization of the composite drive shaft is the minimization of weight, so the objective function of the problem is given as

$$\text{Weight of the shaft}, \quad m = \rho AL \quad \text{or} \quad m = \rho \frac{\pi}{4}\left(d_o^2 - d_i^2\right)L \quad (19)$$

5.2 Design Variables

In optimization techniques, the variables, which are very sensitive in altering the value of objective function, are known as design variables. In the present problem of composite drive shaft optimization, the design variables considered are number of plies[n], stacking sequence [θ_k] and thickness of the ply [t_k]. Table 5.4 presents the design variables with their limiting values.

Table.4 Design Variables and Their Limiting Values

Design variables	Limiting values of the design variables
Number of plies[n]	$n > 0$; $n = 1, 2, 3 \ldots 32$
Stacking Sequence[θ_k]	$-90 \le \theta_k \le 90$; $k = 1, 2 \ldots n$
Thickness of the ply[t_k]	$0.1 \le t_k \le 0.5$

The number of plies required depends on the design constraints, allowable material properties, thickness of plies and stacking sequence. Based on the investigations it was found that 32 plies are sufficient.

5.3 Design Parameters

The parameters, that are sensitive in changing the objective function value but required to be kept as constants (for example material properties) are known as design parameters. Table 5. Shows the design parameters considered in current research work [11, 12].

Table 5 Design Parameters of Steel and Composite Drive Shafts Considered

Parameter	Units	Steel (SM45C)	E-Glass /Epoxy	HM Carbon /Epoxy
Do	mm	90	90	90
L	mm	1250	1250	1250
T_{max}	Nm	3500	3500	3500
N_{max}	rpm	6500	6500	6500
t_k	mm	3.318	0.4	0.12

5.4 Design Constraints

The constraint equations for the problem of optimum design of composite drive shafts are given below.

1. Torque transmission capacity of the shaft : $T \ge T_{max}$
2. Bucking torque capacity of the shaft : $T_{cr} \ge T_{max}$ (20)
3. Lateral fundamental natural frequency : $N_{crt} \ge N_{max}$

The constraint equations C_1, C_2 and C_3 may be written as:

1. $C_1 = \left(1 - \frac{T}{T_{max}}\right)$ If $T < T_{max}$

   $=$ 0 otherwise

2. $C_2 = \left(1 - \frac{T_{cr}}{T_{max}}\right)$ If $T_{cr} < T_{max}$

= 0 otherwise

3. $C_3 = \left(1 - \frac{N_{crt}}{N_{max}}\right)$ If $N_{crt} < N_{max}$

= 0 otherwise (21)

$$C = \sum_{i=1}^{3} C_i \tag{22}$$

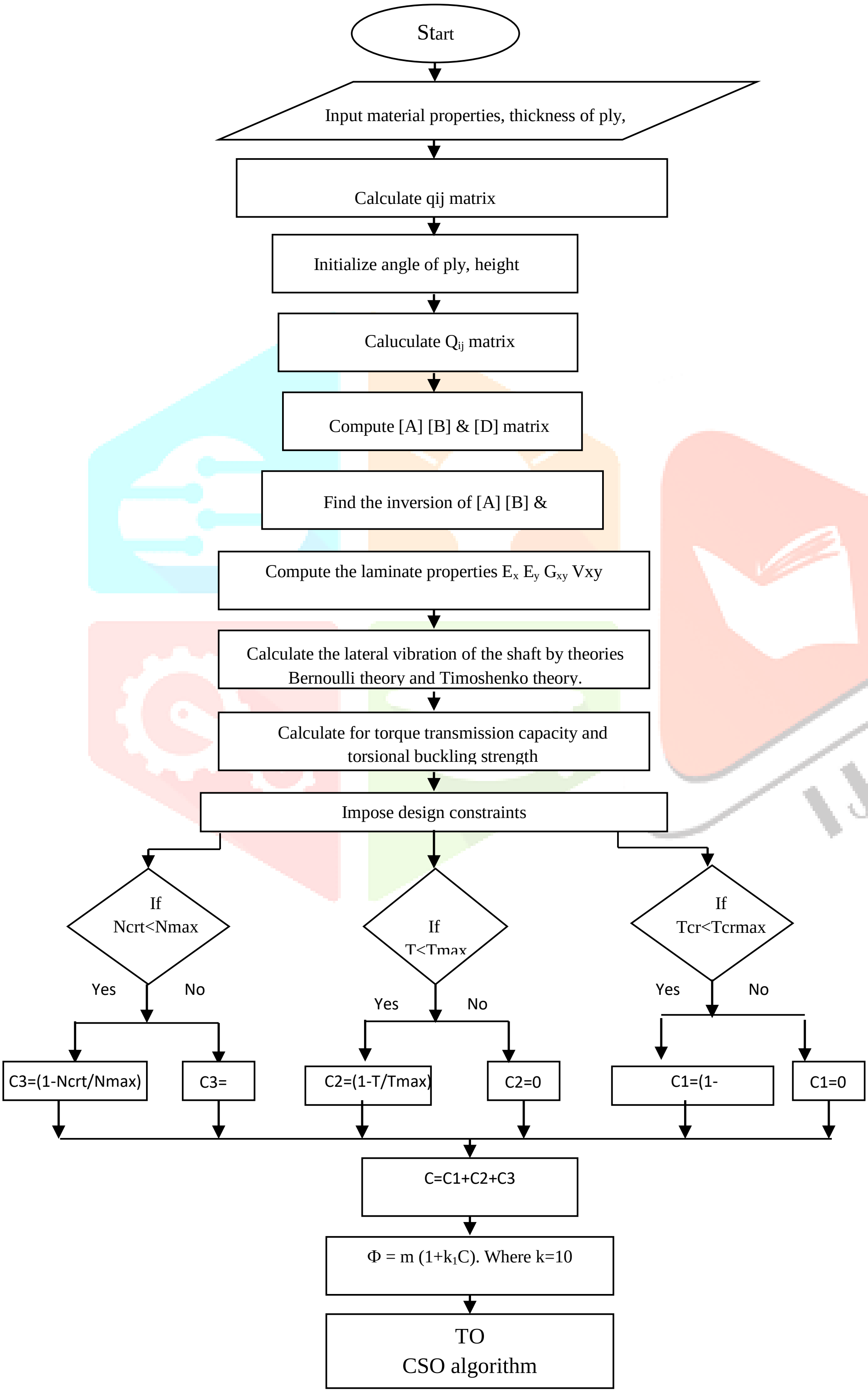

Figure **7** Design Algorithm of Composite Drive Shaft

The constrained optimization would be converted to unconstrained optimization by modifying the objective function as $\Phi = m\ (1+k_1C)$. For all practical purposes, k_1 is a penalty constant and is assumed to be 10. The flow chart describing the design algorithm of composite drive shaft and step by step procedure of optimizing the composite drive shaft using CS algorithm is shown in Figure 7 and program is written in MATLAB.

6. Results and Discussion

A program for CS was developed and optimized using the steps of CS algorithm described above and run by specifying the lower and upper bounds for three design variables such as number of layers, fiber orientation and thickness of each layer, with the objective of minimizing the weight of the composite shaft, which is subjected to the constraints such as torque transmission capacity, torsional buckling strength and fundamental lateral natural frequency. A program is developed and run using MATLAB V8 to perform the optimization process and to obtain the best optimal design values. Optimum design parameters of automotive composite drive shaft using Cuckoo search optimization algorithm for E-Glass/Epoxy, HM-Carbon/Epoxy are obtained and summarized in the Table 6.

Figure.8 shows the Variation of weight with the Number of nests for E-Glass/Epoxy, HM-Carbon/Epoxy, composite drive shafts. It was found that, with increasing number of nests [chosen as 1 to start the optimization process] there was quite a bit fluctuations in the weight and after a certain higher number of nests of cuckoo, the fluctuation of weight value becomes so minimal that the values remain steady for remaining observations. The steps were repeated for all different materials and the process in noting the optimized value for the algorithm was consistent.

The same method for selecting optimal values of number of layers was also adopted through CS. Figure.9 shows the variation of No. of Layers with Number of Nests for E-Glass/Epoxy, HM Carbon/Epoxy composite drive shafts.

Table.6 Optimal design values using Cuckoo search optimization (CSO)

Material	**d_o (mm)**	**L (mm)**	**t_k (mm)**	**N (Optimal**	**T (mm)**	**Optimum Stacking Sequence**	**T (Nm)**	**T_{cr} (Nm)**	**N_{crt} (rpm)**	**Wt (kg)**	***Wt Savin**
Steel (SM45 C)	90	1250	3.318	1	3.318	--------------	3512	42298	9365	8.6	-----
E - Glass /Epoxy	90	1250	0.4	18	7.2	[-74/26/-84/-51/-57/-73/13/82/-25]$_s$	3540	34178	6547	4.37	49.18
HM-Carbon	90	1250	0.12	18	2.16	[-33/-54/67/35/69/42/-	3694	4799	9784	1.18	86.27

/Epoxy						60/20/46]$_s$					

* Taking steel shaft weight as datum

Elastic Constants of the Composite Drive Shafts

The computed elastic constants of E-glass/epoxy, HM Carbon/Epoxy, shafts are as shown in Table 7

Table.7 Computed Elastic Constants of Composite Drive Shaft Tube

Material	E_x (GPa)	E_y (GPa)	G_{xy} (GPa)	ν_{12}
E-Glass/ Epoxy	28.16	24.12	8.311	0.248
HM Carbon/ Epoxy	66.32	58.51	26.75	0.362

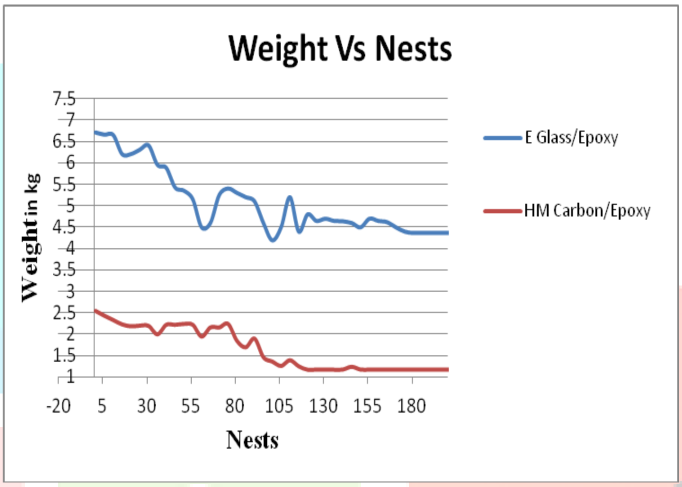

Figure 8 Variation of weight of Composite Drive shafts with number of nests

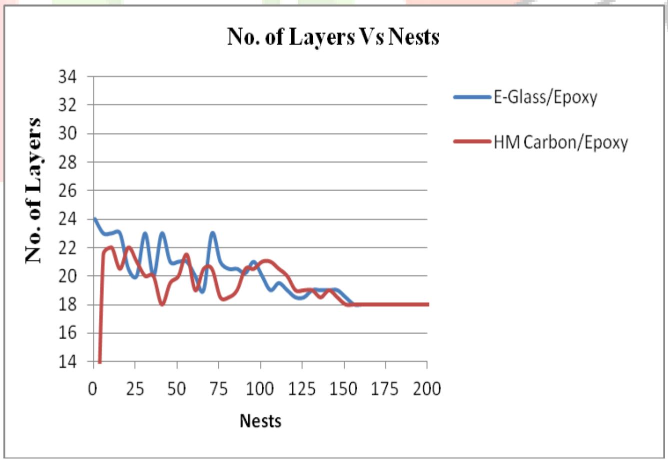

Figure 9 Variation of number of layers Composite Drive shafts with number of nests

7. Conclusions

- From this research, it can be found that there is a greater need to incorporate structural optimization techniques to support the innovative design, thereby reduce design, development time and cost.
- In the current study, an optimization method based on the Cuckoo search algorithm for structural design optimization of multilayered composite drive shaft is described.
- The laminate parameters such as ply thickness, number of plies and stacking sequences were optimized for E-Glass/Epoxy and HM- Carbon/Epoxy drive shafts with the objective of weight minimization of the composite shaft which is subjected to constraints such as torque transmission, torsional buckling load and fundamental natural frequency using Cuckoo search algorithm
- By using Design optimization technique and use of composite materials, weight saving in the range of 49.18% and 86.27% when compared with conventional steel drive shaft was achieved.
- It is found that better results can be obtained from the CS. Therefore, the CS is a most suitable optimization technique for the structural design optimization of vehicle components.

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