International Journal of Engineering

Journal Homepage: www.ije.ir

Multi-objective Optimization of Hybrid Carbon/Glass Fiber Reinforced Epoxy Composite Automotive Drive Shaft

A. Khalkhali*[a], E. Nikghalb[b], M. Norouzian[c]

[a] Department of Automotive Engineering, Iran University of Science and Technology, Tehran, Iran
[b] Department of Mechanical Engineering, Amirkabir University of Technology, Tehran, Iran
[c] Department of Mechanical & Materials Engineering, University of Nebraska-Lincoln, Nebraska, USA

PAPER INFO

Paper history:
Received 02 July 2014
Received in revised form 07 January 2015
Accepted 29 January 2015

Keywords:
Composite Drive Shaft
Multi-objective Optimization
Finite Element Method
Modified NSGA II Algorithm

ABSTRACT

Using composite materials in design and fabrication of drive shafts with high value of fundamental natural frequency which at represents high value of critical speed could provide longer shafts with lighter weight compared to typical metallic materials. In this paper, multi-objective optimization (MOP) of a composite drive shaft is performed considering three conflicting objectives: fundamental natural frequency, critical buckling torque and weight of the shaft. Fiber orientation angle, ply thickness and stacking sequence are also considered as the design variables in this MOP. Modified Nondominated Sorting Genetic Algorithm (modified NSGA II) is employed to solve this MOP. To calculate fundamental natural frequency and critical buckling torque, a finite element model of composite drive shaft of a truck is carried out using commercial software ABAQUS/Standard. Finally, optimum design points are obtained and from all non-dominated optimum design points, some trade-off points are picked using a multi-criteria decision making approach named TOPSIS and the points are discussed.

doi: *10.5829/idosi.ije.2015.28.04a.13*

1. INTRODUCTION

Power transmission drive shafts have many industrial applications, such as in vehicles, cooling towers, pumping sets, aerospace, structures and etc. Drive shaft length is limited by its critical speed. Since the fundamental bending natural frequency of a one-piece metal drive shaft is normally lower than 5700 rpm (95 Hz) when the length of the drive shaft is around 1.5 m [1], the steel drive shaft is usually manufactured in two pieces to increase the fundamental bending natural frequency since the bending natural frequency of a shaft is inversely proportional to the square of beam length and proportional to the square root of specific modulus. Substituting composite structures for conventional metallic structures has many advantages because of higher specific stiffness and strength of composite materials. Using composite cylindrical tubes would permit the use of longer shafts for a specified critical speed.

In 1993, Kim et al. [2] developed an analytical method for critical speed of a composite drive shaft using different shell theories. Bert et al. [3] proposed an analytical method for torsional buckling using different shell theories on a laminated composite drive shaft and also investigated the effect of bending moment on the buckling. Lee et al. [4] investigated the design and construction of a one-piece hybrid aluminum-composite automotive drive shaft. The outcome of the work was a 75 percent decrease in the mass and 160 percent increase in torque resistance in comparison with common two-piece shaft models. Shokrieh et al. [5] conducted researches on the torsional buckling of a composite drive shaft. In addition to proposing an analytical equation for buckling torque, they also used a finite element modeling (FEM) solution along with an experimental model to investigate different parameters

*Corresponding Author's Email: ab_khalkhali@iust.ac.ir (A. Khalkhali)

Please cite this article as: A. Khalkhali, E. Nikghalb, M. Norouzian,Multi-objective optimization of hybrid carbon/glass fiber reinforced epoxy composite automotive drive shaft, International Journal of Engineering (IJE), IJE TRANSACTIONS A: Basics Vol. 28, No. 4, (April 2015) 583-592

effects such as boundary conditions, fiber orientation angle and stacking sequence on the buckling torque. Mutasher [6] conducted researches to predict the torsional resistance of an aluminum-composite drive shaft and investigated the effects of number of layers, stacking sequence and fiber orientation angle and compared them with experimental results. Abu Talib et al. [7] developed a finite element model to analyze a composite drive shaft. In designing this composite drive shaft, they used a combination of carbon and glass fibers layers and studied the effects of fiber orientation angle and stacking sequence on the first natural bending frequency and bending torque. In 2011, Badie et al. [8] investigated a fully composite drive shaft. Using FEM solution, classic lamination theory and experimental tests, they analyzed the effects of stacking sequence and fiber orientation angle on the buckling torque, torsional stiffness, bending natural frequency and fatigue life.

Essentially, achieving low values for weight, high values for critical buckling torque and also high values for critical speed are desired outcomes in the optimal design of the drive shafts. Therefore, to consider such criteria simultaneously, a complex multi-objective optimization problem (MOP) must be solved. Many different methods have been proposed by previous researchers for solving MOPs [9-12]. Non-dominated Sorting Genetic Algorithm (NSGA-II) proposed by Srinivas and Deb [13], which is a Pareto based approach, is one of the efficient algorithms for solving MOPs. It generates a set of non-dominated solutions (Pareto solutions), where a non-dominated solution performs better on at least one criterion than the other solutions. To improve NSGA-II, Nariman-Zadeh proposed modified NSGA-II which uses ε-elimination algorithm instead of crowding factor [14]. This method is employed successfully in many recent studies [15-18]. After finding out the non-dominated points, it is desired to find some trade-off optimum points compromising objective functions. For this purpose, a multi-criteria decision making approach named Technique for Ordering Preferences by Similarity to Ideal Solution (TOPSIS) can be used. TOPSIS is based upon simultaneous minimization of distance from a positive ideal point (PIS) and maximization of distance from a negative ideal point (NIS).

In engineering applications, objective functions are not usually obtainable using simple analytical relations. Nowadays, plenty of objective functions which are considered by designers are calculated using software modeling. In these cases, objective functions are mostly calculated in a runtime coupling process between a programming software product (like MATLAB) and an engineering analysis software product (like ABAQUS) during MOP process. In a general case, coupling of engineering software products makes the synergetic use of those software products possible.

The present study aims at maximizing the fundamental natural frequency and critical buckling load and minimizing the weight of composite drive shaft as the three conflicting objective functions. First, FEM using commercial software ABAQUS/Standard is employed to determine the effects of fiber orientation angle, ply thickness and stacking sequence on fundamental natural frequency and critical buckling torque. The model is then directly implemented in modified NSGA-II to find the best possible combination of the three objective functions. Then, the outputs of the MOP are discussed. Finally, in order to find trade-off among these conflicting objectives, TOPSIS, Mapping and Nearest to Ideal Point methods are employed to select some compromising optimum design points.

2. DESIGN ASPECTS

The critical speed of drive shaft is determined by its first lateral natural frequency. The critical rotational speed is 60 times larger than the fundamental natural frequency of the drive shaft [8]. If the shaft reaches this rotational speed,whirling vibration occurs and it would go under large amplitude vibration and therefore, cannot operate properly. Using a simple assumption, a shaft could be considered as a pined-pined beam and with this model; the first natural frequency can be written in the following form:

$$f_n = \frac{\pi}{2}\sqrt{\frac{gE_x I}{WL^4}} \tag{1}$$

In this expression, f_n is the shaft first natural frequency, g is gravitational acceleration, E_x is the average modulus in axial direction, W is the shaft mass per unit length, L is the shaft length and I is the second moment of inertia for a thin-walled tube which is given as:

$$I = \frac{\pi}{4}({r_o}^4 - {r_i}^4) \approx \pi r^3 t \tag{2}$$

where, t and r are total thickness and mean radius of thin-walled tube, respectively. In a composite tube, if the fiber orientation angle decreases (fibers oriented in the axial direction have zero angle and radially oriented fibers have 90 degrees), the axial modulus of elasticity will increase and consequently, so will the fundamental natural frequency. Therefore, carbon fibers are usually utilized in the axial direction of the tube due to their high modulus of elasticity. Stacking sequence would not have a significant effect on the first natural frequency of the drive shaft [8].

Buckling can be defined as: loosing stability of an equilibrium configuration without fracture or separation to its constituents or at least prior to it [19]. Since drive shaft is a hollow long tube, there is a high possibility of buckling. Equation (3) represents the critical buckling torque for an orthotropic thin-walled tube:

$$T_{cr} = \left(2\pi r^2\right)\left(0.272\right)\left[E_x E_h{}^3\right]^{1/4}\left(\frac{t}{r}\right)^{3/2} \tag{3}$$

In this expression, T_{cr} is the critical buckling torque, E_x is the stiffness modulus in the axial direction, E_h is stiffness modulus in hoop direction, t is the tube thickness, and r is radius. It is obvious that stiffness modulus in hoop direction plays an effective role in increasing the critical torque.

3. FINITE ELEMENT ANALYSIS

Finite element simulation of the composite drive shaft is performed using ABAQUS/Standard. The shaft is modeled as a cylindrical tube of length equal to 1.73m and average diameter equal to 50.3mm. The shaft is assumed to be homogeneous elastic and the stress-strain relationship is linear. The composite used is carbon/epoxy and E-glass/epoxy which are modeled as orthotropic materials. The properties of these two composites are presented in Table 1.

From ABAQUS element library, S4R element is selected for finite element analysis of the shaft. This element has four nodes which each node has 6 degrees of freedom (three translations and three rotations). The element is used in thick and thin shells analysis [20]. Figure 1 demonstrates the composite tube modeled in ABAQUS, layup structure and S4R element.

Modal analysis is implemented to find natural frequencies and undamped mode shapes of a system and would solve an eigenvalue problem. The boundary condition of the drive shaft is considered to be roller-roller which makes the axial direction movement possible, therefore, two degrees of freedom are constrained in each support. According to this boundary condition, the system has rigid vibration mode in axial direction which is neglected in computing the first natural frequency.

TABLE 1. Material properties of E-glass/epoxy and carbon/epoxy composites used in drive shaft

	E-glass/epoxy	Carbon/epoxy
Volume fraction (%)	60	60
E_{11} (GPa)	40.3	126.9
E_{22} (GPa)	6.21	11.0
G_{12} (GPa)	3.07	6.6
Poisson ratio (υ_{12})	0.2	0.28
Ultimate strength (MPa)	827	1170
Weight density (kg/m^3)	1910	1610

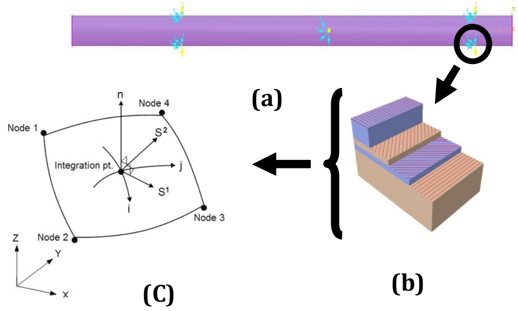

Figure 1. (a) Composite tube model in ABAQUS, with fiber orientation depicted on it **(b)** composite laminate configuration [90, +45, -45, 0] **(c)** S4R element geometry.

Moreover, because the goal is to find the natural frequency of the system, there is no need to exert force on the object.

In a structure composed of four layers stacked as [$\pm 45_{glass}/0_{carbon}/\theta_{glass}$], the system fundamental natural frequency decreases by changing the θ angle from 0 to 90 degrees. At zero angle, the glass fibers are oriented in the axial direction and therefore, produce the maximum axial stiffness modulus (E_x) according to Equation (1). Increasing the glass fibers orientation angle from 0 to 90 degrees, the fibers would arrange in hoop direction and therefore, hoop modulus increases (E_h). However, the fundamental natural frequency decreases. Figure 2 depicts the frequency with respect to θ angle. As it can be observed from the Figure 2, with changing from 0 to 90 degrees in the θ angle, the fundamental natural frequency decreases about 10.28%. It is worth noting that the variation of natural frequency is not always monotonic by changing the θ angle. Variation in the thickness of each layer changes the shaft total modulus of stiffness in the same direction of the layer and also increases the moment of inertia in shaft cross section. These two factors would respectively increase and decrease the natural frequency. At small thicknesses (less than 1.5 mm), the effects of thickness variations could be observed on frequency. However, at bigger thicknesses, the thickness variation has no significant effect on natural frequency. In Figure 3, fundamental natural frequency variation is plotted with respect to θ for a composite structure for three different thicknesses. Figure 3 exhibits that as the angle increases to θ = 40°, increasing the thickness would cause the natural frequency to increase while for angles more than 40°, thickness variation does not have a significant effect on the natural frequency. Besides, stacking sequence has no effect on the natural frequency, because in frequency analysis without damping, the system frequencies are computed using stiffness matrices and mass matrix which none of these matrices are dependent to the stacking sequence. Figure

4 demonstrates the first and second mode shapes of natural frequency and their corresponding frequency values of composite tube stacked as [$\pm 45_{glass}/90_{glass}/0_{carbon}$].

Buckling eigenvalue analysis is similar to eigenvalue modal analysis except that in this method, the mass matrix is replaced by stiffness matrix. In eigenvalue technique, in order to find the maximum torque which can be supported prior to losing stability, it is assumed that the linear stiffness matrix remains unchanged before the buckling. After analysis is finished, an eigenvalue is obtained for each buckling mode which multiplying it by the initial loading, the buckling torque would be computed. In this analysis, the boundary condition is fully constrained in both ends of the cylindrical tube, the edges are constrained to the central point of cylinder and a torque equal to 2000 N.m is exerted on the central point of the tube. In this work, only the first buckling mode is considered.

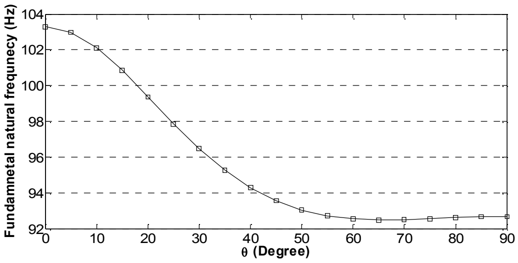

Figure 2. Effect of fiber orientation angle (θ) on fundamental natural frequency of stacking [$\pm 45_{glass}/0_{carbon}/\theta_{glass}$]

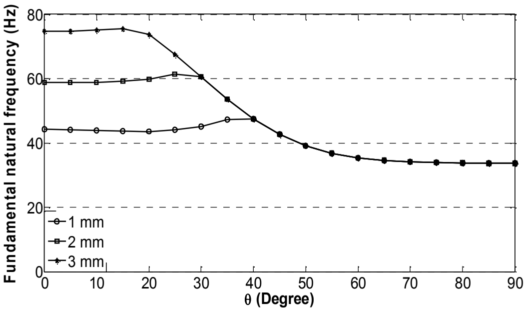

Figure 3. Effect of ply thickness on fundamental natural frequency. θ is changing between 0 to 90 degree in [$\pm\theta_{glass}$]$_2$ stacking. Legend shows the total thickness of composite tube

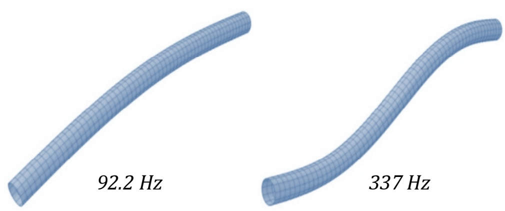

Figure 4. The first (left) and second (right) mode shapes of composite tube and their corresponding frequency values.

In order to compute the critical buckling torque more accurately and rapidly using ABAQUS solver, the initial torque should be considered close to critical buckling torque.

As it is mentioned, the fibers orientation angle in axial and hoop direction would respectively increase the axial (E_x) and hoop stiffness modulus (E_h). According to Equation (3), the critical buckling torque is directly related to $E_x^{1/4}$ and $E_h^{3/4}$. Therefore, increasing the hoop stiffness modulus has more influence on increasing the buckling torque. It can be seen from Figure 5 that in a structure composed of four layers stacked as [$\pm 45_{glass}/0_{carbon}/\theta_{glass}$], changing the θ angle from 0 to 90° would increase the critical buckling torque about 44.85%. In Figure 6, the critical buckling torque is plotted with respect to θ angle for a composite structure [$\pm\theta_{glass}$]$_2$ for three thicknesses. As it is expected, the critical buckling torque increases as the thickness increases. The variation in the stacking sequence has a significant effect on the critical buckling torque. The effects of stacking sequence variations on the critical buckling torque are given in Table 2 for four specimens. Figure 7 demonstrates the first torsional buckling mode and its corresponding eigenvalue and buckling torque of composite tube stacked as [$\pm 45_{glass}/0_{carbon}/90_{glass}$].

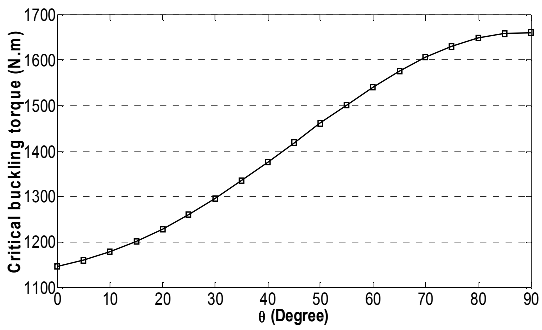

Figure 5. Effect of fiber orientation angle (θ) on critical buckling torque of stacking [$\pm 45_{glass}/0_{carbon}/\theta_{glass}$]

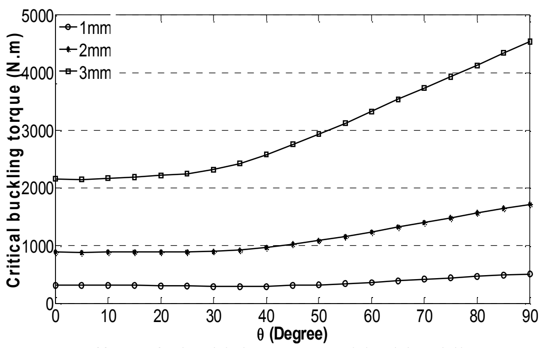

Figure 6. Effect of ply thickness on critical buckling torque. θ is changing between 0 to 90 degree in [$\pm\theta_{glass}$]$_2$ stacking. Legend shows the total thickness of composite tube

TABLE 2. Stacking sequence effects on critical buckling torque and fundamental natural frequency

	Fiber orientation angle (Degree)	Ply thickness (mm)	Critical buckling torque (N.m)	Fundamental natural frequency (Hz)
Specimen 1	$[90_{glass}/\pm45_{glass}/0_{glass}]$	[0.6/0.6/0.6/0.6]	2196	57.61
	$[\pm45_{glass}/90_{glass}/0_{glass}]$		1309.8	57.55
Specimen 2	$[90_{glass}/\pm45_{glass}/0_{carbon}]$		3085.4	86.85
	$[0_{carbon}/\pm45_{glass}/90_{glass}]$		2408.7	84.59

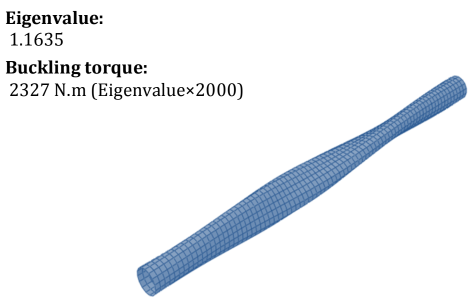

Figure 7. First torsional buckling mode and its corresponding eigenvalue and buckling torque

The results of the numerical simulation of this paper have been compared by the results obtained by Badie et al. [8] and the comparison of results shows an acceptable agreement. This verified finite element method is employed for multi-objective optimization of the drive shaft in the next section.

4. MULTI-OBJECTIVE OPTIMIZATION

Multi-objective optimization, which is also called multi criteria optimization or vector optimization, has been defined as finding a vector of decision variables satisfying constraints to give acceptable values to all objective functions [21, 22]. In these problems, there are several objective or cost functions (a vector of objectives) to be optimized (minimized or maximized) simultaneously. These objectives are often in conflict with each other so that improving one of them will deteriorate another. Therefore, there is no single optimal solution as the best with respect to all the objective functions. Instead, there is a set of optimal solutions, known as Pareto optimal solutions or Pareto front [21, 23-26] for multi-objective optimization problems. The concept of Pareto front or set of optimal solutions in the space of objective functions in MOPs stands for a set of solutions that are non-dominated to each other but are superior to the rest of solutions in the search space. This means that it is not possible to find a single solution to be superior to all other solutions with respect to all objectives so that changing the vector of design variables in such a Pareto front consisting of these non-dominated solutions could not lead to the improvement of all objectives simultaneously. Consequently, such a change will lead to deterioration of at least one objective. Thus, each solution of the Pareto set includes at least one objective inferior to that of another solution in that Pareto set, although both are superior to others in the rest of search space.

Evolutionary algorithms have been widely used for MOP because of their natural properties suited for these types of problems. This is mostly because of their parallel or population-based search approach. Therefore, most of the difficulties and deficiencies within the classical methods in solving MOP problems are eliminated. For example, there is no need for either several runs to find the Pareto front or quantification of the importance of each objective using numerical weights. In this way, the original non-dominated sorting procedure given by Goldberg [27] was the catalyst for several different versions of multi-objective optimization algorithms [23, 24]. However, it is very important that the genetic diversity within the population be preserved sufficiently. This main issue in MOPs has been addressed by many related research works [28]. In this work Modified NSGA II is implemented as MOP process. The detailed explanation of Modified NSGA II can be found in [13, 14].

In order to investigate the optimal design of composite drive shaft in different conditions of design variables (fiber orientation angle, stacking sequence and ply thickness), a 3-objective optimization problem have been solved considering three conflicting objectives: fundamental natural frequency (F), critical buckling torque (T) and weight of structure (W).

The finite element model obtained in previous section and modified NSGA-II algorithm [14] are now deployed simultaneously for multi-objective optimization. In this way, two software products, MATLAB and ABAQUS, are coupled together during the run time. Modified NSGA-II code has been written in MATLAB and ABAQUS input file for calculation of F and T has also been developed. In each generation, design vectors are produced by modified NSGA-II code and are sent to ABAQUS. Then, after calculating objective functions for each design vector, the obtained values are returned to MATLAB and the optimization process is continued. Finally, the non-dominated optimum values of objective functions and the corresponding design vectors can be obtained.

The 3-objective optimization problem can be formulated in the following form:

$$\begin{cases} Maximize: f_1 = F(\theta, t, S) \\ Maximize: f_2 = T(\theta, t, S) \\ Minimize: f_3 = W(t, S) \end{cases} \quad (4)$$

Subject to inequalities costraints:

$$\begin{cases} T > 2030 N.m \\ F > 90.5 Hz \end{cases} \quad (5)$$

F, *T* and *W*, directly come from finite element analysis of ABAQUS/standard solver via a coupling process between MATLAB and ABAQUS. $\theta$and *t* denote set of fibers orientation angle and set of layers thickness as θ=[θ_1, θ_2, θ_3, θ_4] and t=[t_1, t_2, t_3, t_4], respectively Variable S denotes an index with the values of 1, 2, 3, 4 and 5 to determine carbon ply location in composite laminate configuration. The composite structure of the drive shaft is composed of four layers which each layer can be made from a type of carbon/epoxy or E-glass/epoxy. Utilizing carbon/epoxy material in the structure of a four-layer drive shaft is limited to one layer of carbon/epoxy material due to the high cost of carbon/epoxy materials. The variation limits of each design Variable are presented in Table 3. The evolutionary process of Pareto multi-objective optimization is accomplished by using the modified NSGAII where a population size of 40 used in 400 generations with crossover probability P_c and mutation probability P_m of 0.7 and 0.07, respectively.

TABLE 3. Variation limit for each variable in MOP

Parameter	Description	Variation limit
θ=[θ_1, θ_2, θ_3, θ_4]	θ_i demonstrates fiber orientation angle in *i*-th ply	-90°< θ_i<90°
t=[t_1, t_2, t_3, t_4]	t_i demonstrates *i*-th ply thickness in composite laminate configuration	0.1mm< t_i<1.1mm
S	S is an index that demonstrates carbon ply location in composite laminate configuration	S=1: [glass, glass, glass, glass] S=2: [carbon, glass, glass, glass] S=3: [glass, carbon, glass, glass] S=4: [glass, glass, carbon, glass] S=5: [glass, glass, glass, carbon]

5. RESULT

Figure 8 depicts the non-dominated individual design points of 3-objective optimization in the plane of (F- T). Such non-dominated individual design points have been shown in the plane of (F- W) and (T- W) in Figure 9 and Figure 10, respectively. It should be noted that there is a single set of individuals as a result of the 3-objective optimization of *F*, *T* and *W* which are shown in different planes.

Values of design variables and objective functions for all of the 25 obtained individual design points are shown in Table 4. These individuals are all non-dominated when considering all three objectives simultaneously.

It is now be desired to find trade-off optimum design points out of all non-dominated 3-objective optimization process compromising all three objective functions together. This can be achieved by three different methods employed in this paper, namely, TOPSIS method, the nearest to ideal point method and mapping method. TOPSIS is a technique to evaluate the performance of alternatives through the similarity with the ideal solution developed by Hwang and Yoon [29]. In the nearest to ideal point method, an ideal point with the best value of each objective function is considered initially. Afterwards, the distances of all non-dominated points to the ideal point is calculated. In this method, the desired point represents minimum distance to the ideal point. In the mapping method, the values of objective functions of all non-dominated point are mapped into interval 0 and 1. Using the sum of these values for each non-dominated point, the desired point simply represents the minimum of the sum of those values. Consequently, trade-off optimum design points '*a*', '*b*' and '*c*' are the points which have been obtained from the TOPSIS method, nearest to ideal point method and mapping method, respectively. Moreover, another trade-off design point '*d*' which is introduced later can be simply recognized from Figures 6-8. Obtained trade-off optimum design points are bolded in Table 4.

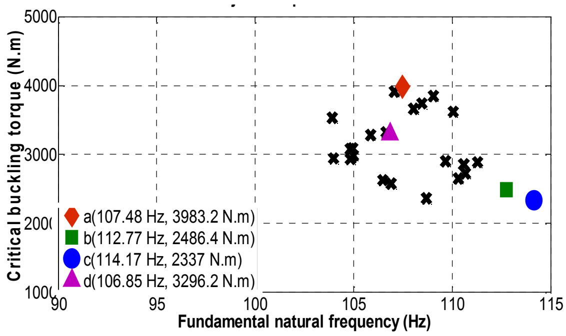

Figure 8. Critical buckling torque versus fundamental natural frequency in 3-objective optimization

TABLE 4. Non-dominated individual design points. Four chosen trade-off points are bolded

	Design variables			Objective functions		
	θ (degree)	***t (millimeter)***	***S***	***F (Hz)***	***T (N.m)***	***W (Kg)***
1(c)	[89/-13/0/-84]	[0.37/0.30/0.94/0.26]	4	114	2337	1.80
2	[-5/2/22/89]	[0.99/0.12/0.50/0.61]	2	108	2361	2.17
3(b)	[89/-13/0/ -84]	[0.37/0.3/0.94/0.30]	4	112	2486	1.85
4	[86/7/67/-38]	[0.30/1.07/0.55/0.19]	3	106	2576	2.05
5	[86/7/67/-38]	[0.30/1.07/0.57/0.19]	3	106	2622	2.07
6	[-71/-5/12/79]	[0.27/0.97/0.50/0.36]	3	110	2652	2.05
7	[86/7/-1/-38]	[0.30/1.07/0.55/0.40]	3	110	2724	2.27
8	[86/7/-25/-38]	[0.30/1.07/0.55/0.40]	3	110	2854	2.27
9	[89/-13/1/-28]	[0.44/0.60/1.03/0.30]	4	111	2881	2.33
10	[-79/-5/21/79]	[0.27/0.97/0.53/0.35]	3	109	2900	2.08
11	[90/-9/6/-89]	[0.62/0.30/0.94/0.24]	4	104	2929	2.05
12	[90/-9/6/-78]	[0.62/0.30/0.91/0.25]	4	103	2940	2.04
13	[90/-9/6/-78]	[0.62/0.30/0.94/0.24]	4	105	2980	2.05
14	[86/7/67/-38]	[0.30/1.07/0.55/0.40]	3	105	3075	2.27
15	[89/-12/6/-78]	[0.62/0.30/0.94/0.26]	4	105	3086	2.07
16	[-71/-5/12/79]	[0.26/0.99/0.53/0.56]	3	106	3282	2.30
17(d)	[89/-13/2/-84]	[0.68/0.30/0.94/0.24]	4	107	3296	2.12
18	[90/-9/6/-84]	[0.44/0.6/0.94/0.24]	4	107	3319	2.18
19	[89/-13/2-84]	[0.71/0.30/0.90/0.30]	4	104	3528	2.18
20	[89/-13/0/-84]	[0.44/0.60/1.02/0.25]	4	110	3614	2.27
21	[90/-9/6/-84]	[0.44/0.60/1.02/0.25]	4	108	3656	2.27
22	[89/-13/6-84]	[0.44/0.6/1.02/0.25]	4	108	3739	2.27
23	[89/-13/0/-84]	[0.44/0.6/1.03/0.30]	4	109	3844	2.33
24	[90/-9/6/-84]	[0.44/0.6/1.03/0.30]	4	107	3904	2.33
25(a)	[89/-13/6/-84]	[0.44/0.60/1.03/0.30]	4	107	3983	2.33

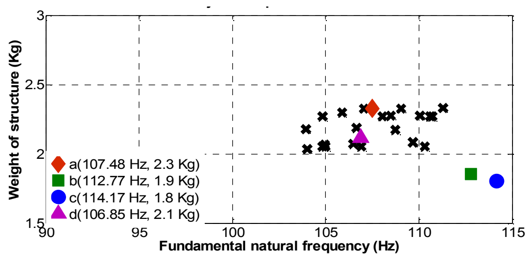

Figure 9. Total weight of structure versus fundamental natural frequency in 3-objective optimization

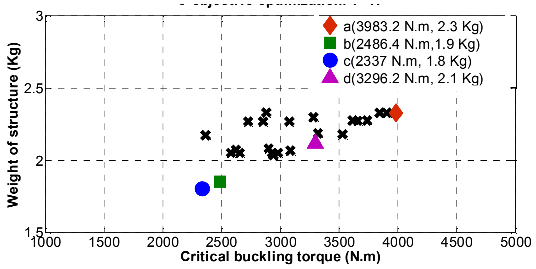

Figure 10. Total weight of structure versus critical buckling torque in 3-objective optimization

6. DISCUSSION

From the (F- T) plane in Figure 8, it could be observed that points '*b*' and '*c*' are near together. The points have fundamental natural frequency around 113 Hz (6800 RPM) and critical buckling torque around 2400 N.m which are almost 5.6% further and 35.5% less than design point '*a*', respectively. However, they have lower weights than design point '*a*', meaning weaker materials are also used to fabricate drive shaft with lower costs. The point '*a*' has the best critical buckling torque (3983 N.m) and the worst weight (2.3 kg) among non-dominated design points, but points '*b*' and '*c*' are dominate the point in fundamental natural frequency. Point '*b*' is the best in both fundamental natural frequency (114 Hz) and weight (1.8 kg), but it is also the worst in critical buckling torque among non-dominated design points. Point '*d*' has nearly the same frequency of point '*a*', but it has almost 17% less critical buckling torque and 9% less weight across point '*a*'. In comparison to point '*c*', design point '*d*' has almost 17.5% further weight and 6.5% less fundamental natural frequency, but it has 41% better critical buckling torque. It seems that point '*d*' has a moderate situation in compromising all 3-objective functions among the rest of the non-dominated design points.

Some informative design principles could also be extracted from non-dominated design points. According to the non-dominated design variables, it can be concluded that in a specific stacking sequence, the layer with carbon fibers would orient in the axial direction (0 degree) which is due to increasing the axial stiffness modulus and consequently, increasing the fundamental natural frequency. Also, at least one layer with E-glass fibers would orient in hoop direction (90 degrees) which is because of increasing hoop stiffness modulus and consequently, increasing critical buckling torque. The two remaining layers with E-glass fibers would arrange between 0 to 90 degrees which would lead to increasing the resistance to the torques exerting on those directions. In addition, it seems that it is not necessary to have symmetry in the composite laminate configuration.

In order to compare the performance of composite drive shaft with metallic drive shaft, the fundamental frequency, critical buckling load and total weight of two models of steel drive shaft is studied. The length and average radius of both models are considered to be equal to those of the composite models studied in this paper. The tube thickness of the first and second models is considered so that their buckling torques are 3000 N.m and 3700 N.m respectively. In this analysis, Young's modulus, Poisson's ratio and density of the steel are considered 200 GPa, 0.33 and 7800 Kg/m^3 respectively which are appropriate estimations for most of industrial steel alloys. In Table 5, the fundamental natural frequencies and weights of these two models are given and compared to those of the four composite drive shafts, 'a', 'b', 'c' and 'd'.

From Tables 5, it could be observed that change in the thickness of steel does not play an important role in its fundamental natural frequency. Comparing the two models of steel drive shaft with four points resulted from trade-off, one could conclude that the fundamental natural frequency of composite drive shaft has a higher value than in steel drive shaft. Also, the weights of steel drive shafts are averagely twice the weight of composite drive shafts. For example, in application of medium duty truck, the drive shaft requires to carry a maximum torque of about 2030 N.m [8]. For this purpose, utilizing composite drive shaft 'a' instead of the first or second model of steel drive shaft results in a reduction of about 45 percent in the total mass and an increase in the maximum working RPM of about 780 RPM (13 Hz). Furthermore, using composite drive shaft 'c' instead of the steel models would result in about a 60 percent total mass reduction and an increase of about 1200 RPM (20 Hz) in the maximum working RPM. From this comparison, one concludes that for long distance power transmission as in the case of trucks, composite drive shafts perform so much better than steel drive shafts.

7. CONCLUSION

In this paper, a numerical model was generated for a composite drive shaft incorporating carbon and E-glass fibers within an epoxy matrix using ABAQUS/standard software and the effects of design variables variations were studied in this model. The main conclusions extracted from this model are as following:

TABLE 5. Comparison between composite and steel drive shafts

		Buckling Torque (N.m)	Fundamental natural frequency (Hz)	Total weight (Kg)
Steel drive shaft	Model 1	3000	94.33	4.3
	Model 2	3700	94.33	4.7
Composite drive shaft	Point a	3983	107.48 (14% better than Model 1,2)	2.3
	Point b	2486	112.77 (20% better than Model 1,2)	1.9
	Point c	2337	114.17 (21% better than Model 1,2)	1.8
	Point d	3296	106.85 (13% better than Model 1,2)	2.1

- v An increase in the fundamental natural frequency due to a decrease in the fibers orientation angle which is because of an increase in the axial stiffness modulus.
- v An increase in the critical buckling torque due to an increase in the fibers orientation angle which is due to an increase in the hoop stiffness modulus.
- v Non-monotonic changes of fundamental natural frequency and critical buckling torque by changes in the fiber orientation angle.
- v Different behaviours of fundamental natural frequency and critical buckling torque in different thicknesses

Afterwards, this model was deployed in a 3-objective Pareto-based optimization using modified NSGA II algorithm. The three objective functions were fundamental natural frequency, critical buckling torque and the total weight of drive shaft structure. The final non-dominated design points were demonstrated on the Pareto fronts as optimum points. In designing point of view, these points have interesting features and all of them can be chosen as an optimum point. Finally, four design points were chosen and they were discussed on the Pareto fronts and compared to each other.

The comparison between these four composite drive shafts and steel drive shafts demonstrated that in the same load carrying capacity, the composite models have higher fundamental natural frequencies and lower masses compared to the steel drive shaft models. Therefore, utilizing composite drive shafts in trucks are much more appropriate than its steel counterparts.

Consequently, such 3-objective optimization of fundamental natural frequency (F), critical buckling torque (T) and weight of structure (W) provides more optional choices of design variables based on non-dominated points which can be selected from a trade-off point of view or other design aspects such as fatigue life, load carrying capacity or torsional frequency.

8. REFERENCES

1. Schmelz, F., Seherr-Thoss, C.H.-C. and Aucktor, E., *Designing joints and driveshafts*, in *Universal joints and driveshafts*. (1992), 79-169.
2. Kim, C.-D. and Bert, C.W., "Critical speed analysis of laminated composite, hollow drive shafts", ***Composites Engineering***, Vol. 3, No. 7, (1993), 633-643.
3. Bert, C.W. and Kim, C.-D., "Analysis of buckling of hollow laminated composite drive shafts", ***Composites Science and Technology***, Vol. 53, No. 3, (1995), 343-351.
4. Kim, H.S., Kim, J.W. and Kim, J.K., "Design and manufacture of an automotive hybrid aluminum/composite drive shaft", ***Composite Structures***, Vol. 63, No. 1, (2004), 87-99.
5. Shokrieh, M.M., Hasani, A. and Lessard, L.B., "Shear buckling of a composite drive shaft under torsion", ***Composite Structures***, Vol. 64, No. 1, (2004), 63-69.

6. Mutasher, S., "Prediction of the torsional strength of the hybrid aluminum/composite drive shaft", ***Materials & Design***, Vol. 30, No. 2, (2009), 215-220.

7. Talib, A.A., Ali, A., Badie, M.A., Lah, N.A.C. and Golestaneh, A., "Developing a hybrid, carbon/glass fiber-reinforced, epoxy composite automotive drive shaft", ***Materials & Design***, Vol. 31, No. 1, (2010), 514-521.

8. Badie, M., Mahdi, E. and Hamouda, A., "An investigation into hybrid carbon/glass fiber reinforced epoxy composite automotive drive shaft", ***Materials & Design***, Vol. 32, No. 3, (2011), 1485-1500.

9. Lee, D., Gonzalez, L.F., Periaux, J., Srinivas, K. and Onate, E., "Hybrid-game strategies for multi-objective design optimization in engineering", ***Computers & Fluids***, Vol. 47, No. 1, (2011), 189-204.

10. Guillen-Gosalbez, G., "A novel milp-based objective reduction method for multi-objective optimization: Application to environmental problems", ***Computers & Chemical Engineering***, Vol. 35, No. 8, (2011), 1469-1477.

11. Bekker, J. and Aldrich, C., "The cross-entropy method in multi-objective optimisation: An assessment", ***European Journal of Operational Research***, Vol. 211, No. 1, (2011), 112-121.

12. Azizmohammadi, R., Amiri, M., Tavakkoli-Moghaddam, R. and Mohammadi, M., "Solving a redundancy allocation problem by a hybrid multi-objective imperialist competitive algorithm", ***International Journal of Engineering-Transactions C: Aspects***, Vol. 26, No. 9, (2013), 1031-1042.

13. Deb, K., Pratap, A., Agarwal, S. and Meyarivan, T., "A fast and elitist multiobjective genetic algorithm: Nsga-ii", ***Evolutionary Computation, IEEE Transactions on***, Vol. 6, No. 2, (2002), 182-197.

14. Jamali, A., Nariman-Zadeh, N., Darvizeh, A., Masoumi, A. and Hamrang, S., "Multi-objective evolutionary optimization of polynomial neural networks for modelling and prediction of explosive cutting process", ***Engineering Applications of Artificial Intelligence***, Vol. 22, No. 4, (2009), 676-687.

15. Khakhali, A., Nariman-Zadeh, N., Darvizeh, A., Masoumi, A. and Notghi, B., "Reliability-based robust multi-objective crashworthiness optimisation of s-shaped box beams with parametric uncertainties", ***International Journal of Crashworthiness***, Vol. 15, No. 4, (2010), 443-456.

16. Khalkhali, A. and Safikhani, H., "Pareto based multi-objective optimization of a cyclone vortex finder using cfd, gmdh type neural networks and genetic algorithms", ***Engineering Optimization***, Vol. 44, No. 1, (2012), 105-118.

17. Khalkhali, A., Narimanzadeh, N., Khakshournia, S. and Amiri, S., "Optimal design of sandwich panels using multi-objective genetic algorithm and finite element method", ***International Journal of Engineering-Transactions C: Aspects***, Vol. 27, No. 3, (2013), 395-402.

18. Gorji, M., Kazemi, A. and Ganji, D., "Optimal thermodynamic design of turbofan engines using multi-objective genetic algorithm", ***International Journal of Engineering-Transactions C: Aspects***, Vol. 27, No. 6, (2013), 961-970.

19. Cook, R.D., "Concepts and applications of finite element analysis, John Wiley & Sons, (2007).

20. Hibbitt, H., Karlsson, B. and Sorensen, P., "Abaqus analysis user's manual version 6.10", ***Dassault Systèmes Simulia Corp.: Providence, RI, USA***, (2011).

21. Coello, C. and Christiansen, A.D., "Multiobjective optimization of trusses using genetic algorithms", ***Computers & Structures***, Vol. 75, No. 6, (2000), 647-660.

22. Osyczka, A., "Multicriteria optimization for engineering design", ***Design Optimization***, Vol. 1, (1985), 193-227.

23. Srinivas, N. and Deb, K., "Muiltiobjective optimization using nondominated sorting in genetic algorithms", ***Evolutionary Computation***, Vol. 2, No. 3, (1994), 221-248.

24. Fonseca, C.M. and Fleming, P.J., "Genetic algorithms for multiobjective optimization: Formulationdiscussion and generalization", in ICGA. Vol. 93, (1993), 416-423.

25. Coello, C.A.C., Van Veldhuizen, D.A. and Lamont, G.B., "Evolutionary algorithms for solving multi-objective problems, Springer, Vol. 242, (2002).

26. Pareto, V., "Cours d'economie politique, Librairie Droz, (1964).

27. Goldberg, D., "Genetic algorithms in search, optimization, and machine learning, addison-wesley, reading, ma, 1989", ***NN Schraudolph and J..*** Vol. 3, No. 1.

28. Toffolo, A. and Benini, E., "Genetic diversity as an objective in multi-objective evolutionary algorithms", ***Evolutionary Computation***, Vol. 11, No. 2, (2003), 151-167.

29. Gwo-Hshiung, T., "Multiple attribute decision making: Methods and applications", ***Multiple Attribute Decision Making: Methods and Applications***, (2010).

Multi-objective Optimization of Hybrid Carbon/Glass Fiber Reinforced Epoxy Composite Automotive Drive Shaft

A. Khalkhali[a], E. Nikghalb[b], M. Norouzian[c]

[a] *Department of Automotive Engineering, Iran University of Science and Technology, Tehran, Iran*
[b] *Department of Mechanical Engineering, Amirkabir University of Technology, Tehran, Iran*
[c] *Department of Mechanical & Materials Engineering, University of Nebraska-Lincoln, Nebraska, USA*

PAPER INFO

Paper history:
Received 02 July 2014
Received in revised form 07 January 2015
Accepted 29 January 2015

Keywords:
Composite Drive Shaft
Multi-objective Optimization
Finite Element Method
Modified NSGA II Algorithm

چکید

استفاده از مواد کامپوزیتی در طراحی و ساخت محورهای محرک سبب افزایش فرکانس طبیعی اولیه محور محرک و در نتیجه افزایش سرعت بحرانی آن شده و در مقایسه با مدل‌های فلزی، محورهایی با طول بزرگتر و وزن کمتر ارائه می‌کند. در این مقاله، بهینه‌سازی چند هدفی یک محور محرک کامپوزیتی با در نظر گرفتن سه هدف ناسازگار مورد توجه قرار گرفت: فرکانس طبیعی اولیه، گشتاور کمانش بحرانی و وزن محور. زاویه جهت‌گیری فیبرها، ضخامت هر لایه و ترتیب لایه‌چینی، به عنوان متغیرهای طراحی در مسئله بهینه‌سازی در نظر گرفته شدند. روش NSGA-II در فرآیند بهینه‌سازی چند هدفی به کار گرفته شد. برای محاسبه فرکانس طبیعی اولیه و گشتاور کمانش بحرانی، مدل المان محدود محور محرک یک خودروی باری در نرم‌افزار ABAQUS/Standard مورد استفاده قرار گرفته است. در نهایت نقاط طراحی بهینه استخراج شدند و از میان این نقاط طراحی غیر برتر، بعضی نقاط مصالحه شده به کمک روش تصمیم‌گیری چند معیاری TOPSIS انتخاب شده و مورد بحث قرار گرفته‌اند.

doi*: 10.5829/idosi.ije.2015.28.04a.13*