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Optimum Design And Analysis Of A Composite Drive Shaft For An Automobile By Using Genetic Algorithm And Ansys

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Abstract:

Substituting composite structures for conventional metallic structures has many advantages because of higher specific stiffness and strength of composite materials. This work deals with the replacement of conventional two-piece steel drive shafts with a single-piece e-glass/epoxy, high strength carbon/epoxy and high modulus carbon/epoxy composite drive shaft for an automotive application. The design parameters were optimized with the objective of minimizing the weight of composite drive shaft. The design optimization also showed significant potential improvement in the performance of drive shaft.

Keywords:-Torque transmission, Torsional buckling capacities, Fundamentallateral Natural frequency, Bernoulli Euler theory, Timoshenko beam theory, Static analysis, Modal analysis, Buckling analysis, Ansys.

1. INTRODUCTION

Advancedcomposite materials can be defined as combination of materials appropriately arranged using reinforcing fibers, carefully chosen matrixes, and some times auxiliary materials like adhesive core and other inserts. These combinations after proper manipulation and processing result in finished structure/item with synergistic properties i.e. properties achieved after fabrication cannot be obtained by individual components acting alone. The ACMs can be classified in different categories on the basis of micro structures, multiphases, reinforcements, manner of packing fibers layered compositions, method of composition, matrix system processing methods etc. Basic components of ACMs are (i) Reinforcement (fibers) (ii) Matrix (iii) Honey comb core/adhesives (for sand witched structures). The great variety of fibers materials in various forms, shapes and sizes have been recently developed for use in ACMs and in the construction industries. Steel, glass, carbon,

Aramid (kevlar), boron, silicon carbide, silicon nitrates, alumina fibers are some of the commonly used high performance reinforcement fibers in ACMs. The reinforcements may be called by different names according to sizes such as Whisker (< 0.025 mm), fiber (0.025 – 0.8 mm), Wire (0.8 – 6.4 mm), rod (6.4 – 50 mm) and bar (> 50 mm).

In general the continuous filamentary type reinforcement is important from structural application point of view.

It is the reinforcement which is primarily responsible for the mechanical properties of ACMs. Usually all the reinforcements (fibers) are stronger in tension than steel, but weak in shear (i.e. brittle) requiring the filler material (Matrix) relatively strong in shear which will protect reinforcement against abrasion or environmental corrosion. Matrix also helps in distributing the load from reinforcement, absorbing energy, reducing stress concentration and preventing cracks propagation. Thermosetting and thermo plastic types of organic polymers are used as Matrix (e.g. epoxide, phenolic, polyamide resins etc.).

Some of the important fibers used as reinforcement in ACMs along with their characteristic properties are discussed briefly.

Glass fiber properties.

Property	**E-glass**	**R-glass**	**D-glass**	**S-glass**
Density (g/cm^3)	2.60	2.55	2.16	2.49
Tensile strength (Mpa)	3400	4400	2500	4580
Tensile modulus (Gpa)	73	86	55	86.93
Elongation at break (%)	4.5	5.2	4.5	5.4
Filament diameter	3-14	3-14	3-14	–

Properties of Aramid fibers

property	**Polyester**	**Monex**	**Kevlar29**	**Kevlar49**	**Teflon**
Density 9/cm^3	1.38	1.38	1.44	1.45	2.15
Tensile Strength(MPa)	900	670	2700	3500	-
Tensile Modulus(GPa)	18	60	135	133	–
Elongation at break %	10-15	20-30	4	2.5	20-30
Filament diameter	10-12	–	–	–	20

2. Design of a Composite Drive Shaft

2.1 Specification of the Problem

The specifications of the composite drive shaft of an automotive transmission are same as that of the steel drive shaft for optimal design.

2.2 Assumptions

1. The shaft rotates at a constant speed about its longitudinal axis.
2. The shaft has a uniform, circular cross section.
3. The shaft is perfectly balanced, i.e., at every cross section, the mass center coincides with the geometric center.
4. All damping and nonlinear effects are excluded.
5. The stress-strain relationship for composite material is linear & elastic; hence, Hooke's law is applicable for composite materials.
6. Acoustical fluid interactions are neglected, i.e., the shaft is assumed to be acting in a vacuum.
7. Since lamina is thin and no out-of-plane loads are applied, it is considered as under the plane stress.

2.3 Selection of Cross-Section

The drive shaft can be solid circular or hollow circular. Here hollow circular cross-section was chosen because:

- The hollow circular shafts are stronger in per kg weight than solid circular.
- The stress distribution in case of solid shaft is zero at the center and maximum at the outer surface while in hollow shaft stress variation is smaller. In solid shafts the material close to the center are not fully utilized.

2.4 Selection of Reinforcement Fiber

Fibers are available with widely differing properties. Review of the design and performance requirements usually dictate the fiber/fibers to be used.

- **Carbon/Graphite fibers:** Its advantages include high specific strength and modulus, low coefficient of thermal expansion, and high fatigue strength. Graphite, when used alone has low impact resistance. Its drawbacks include high cost, low impact resistance, and high electrical conductivity.
- **Glass fibers**: Its advantages include its low cost, high strength, high chemical resistance, and good insulating properties. The disadvantages are low elastic modulus, poor adhesion to polymers, low fatigue strength, and high density, which increase shaft size and weight. Also crack detection becomes difficult.
- **Kevlar fibers:** Its advantages are low density, high tensile strength, low cost, and higher impact1. resistance. The disadvantages are very low compressive strength, marginal shear strength, and high water absorption. Kevlar is not recommended for use in torque carrying application because of its low strength in compression and shear Here, both glass and carbon fibers are selected as potential materials for the design of shaft.

2.5 Selection of Resin System

The important considerations in selecting resin are cost, temperature capability, elongation to failure and resistance to impact (a function of modulus of elongation). The resins selected for most of the drive shafts are either epoxies or vinyl esters. Here, epoxy resin was selected due to its high strength, good wetting of fibers, lower curing shrinkage, and better dimensional stability.

S. No.	Property	Units	E-Glass / Epoxy	HS Carbon / Epoxy	HM Carbon / Epoxy
1	E11	GPa	50.00	134.0	190.0
2.	E22	GPa	12.0	7.0	7.7
3.	G12	GPa	5.6	5.8	4.2
4.	ν12	-	0.3	0.3	0.3
5.	$S^t_1=S^c_1$	MPa	800.00	880.00	870.0
6.	$S^t_2=S^c_2$	MPa	40.0	60.0	54.0
7.	S_{12}	MPa	72.0	97.0	30.0
8.	ρ	Kg/m^3	2000.00	1600.0	1600.0

2.6 Selection of Materials

Properties of E-Glass/Epoxy, HS Carbon/Epoxy and HM Carbon/Epoxy

Based on the advantages discussed earlier, the E-Glass/Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy materials are selected for composite drive shaft. The Table shows the properties of the E-Glass/Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy materials used for composite drive shafts.

3. Design optimization:

Optimization of an engineering design is an improvement of a proposed design that results in the best properties for minimum cost. Most of the methods used for design optimization assume that the design variables are continuous. In structural optimization, almost all design variables are discrete. A simple Genetic Algorithm (GA) is used to obtain the optimal number of layers, thickness of ply and fiber orientation of each layer. All the design variables are discrete in nature and easily handled by GA. With reference to the middle plane, symmetrical fiber orientations are adopted.

3.2 Optimization Techniques.

GA´s differs from traditional optimization algorithm in many ways. A few are listed here .

GA does not require a problem specific knowledge to carry out a search. GA uses only the values of the objective function. For instance, calculus based search algorithms use derivative information to carry out a search.

GA uses a population of points at a time in contrast to the single point approach by the traditional optimization methods. That means at the same time GAs process a number of designs.

In GA, the design variables are represented as strings of binary variables that correspond to the chromosomes in natural genetics. Thus the search method is naturally applicable for solving discrete and integer programming problems. For continuous variable, the string length can be varied to achieve any desired resolution.

GAs uses randomized operators in place of the usual deterministic ones. In every generation, a new set of strings is produced by using randomized parents selection and crossover from the old generation (old set of strings).

3.3 Objective Function

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

Weight of the shaft, $m = \rho AL$

$$m = \rho \frac{\pi}{4}\left(d_0^2 - d_i^2\right)L$$

3.4 Design Variables

The design variables of the problem are

- Number of plies
- Thickness of the ply
- Stacking Sequence

The limiting values of the design variables are given as follows.

1. $n \geq 0$
2. $-90 \geq \theta_k \leq 90$
3. $0.1 \leq t_k \leq 0.5$

Where k = 1, 2 …………….. n and n = 1, 2, 3, …………. 32

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 up to 32 numbers of plies are sufficient.

3.5 Design Constraints

1. Torque transmission capacity of the shaft $T \geq T_{\max}$
2. Bucking torque capacity of the shaft $T_{cr} \geq T_{\max}$
3. Lateral fundamental natural frequency $N \geq N_{crt}$

- The constraint equations 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

$$C = C_1 + C_2 + C_3$$

Using the method of Rajeev and Krishnamoorthy, the constrained optimization can be converted to unconstrained optimization by modifying the objective function is

$$\phi = m(1 + k_1 C)$$

For all practical purposes, k_1 is a penalty constant and is assumed to be 10.

3.6 Input GA Parameters

Input GA parameters of E-Glass / Epoxy, HS Carbon / Epoxy and HIM Carbon / Epoxy composite drive shafts are shown in the table

Input GA Parameters

Number of Parameters	n/2+2 if n is even
	(n+1)/2+2 if n is odd
Total string length	139
Population size	50
Maximum generations	150
Cross-over probability	1
Mutation probability	0.003
String length for number of plies	5
String length of fiber orientation	8
String length for thickness of ply	6

4. Design Analysis

Finite Element Analysis (FEA) is a computer-based numerical technique for calculating the strength and behavior of engineering structures. It can be used to calculate deflection, stress, vibration, buckling behavior and many other phenomena. It also can be used to analyze either small or large scale deflection under loading or applied displacement. It uses a numerical technique called the finite element method (FEM). In finite element method, the actual continuum is represented by the finite elements. These elements are considered to be joined at specified joints called nodes or nodal points. As the actual variation of the field variable (like displacement, temperature and pressure or velocity) inside the continuum is not known, the variation of the field variable inside a finite element is approximated by a simple function.

The approximating functions are also called as interpolation models and are defined in terms of field variable at the nodes. When the equilibrium equations for the whole continuum are known, the unknowns will be the nodal values of the field variable.

In this project finite element analysis was carried out using the FEA software ANSYS. The primary unknowns in this structural analysis are displacements and other quantities, such as strains, stresses, and reaction forces, are then derived from the nodal displacements.

4.1 Modeling Linear Layered Shells

SHELL99 may be used for layered applications of a structural shell model as shown in Figure SHELL99 allows up to 250 layers. The element has six degrees of freedom at each node: translations in the nodal x, y, and z directions and rotations about the nodal x, y, and z-axes.

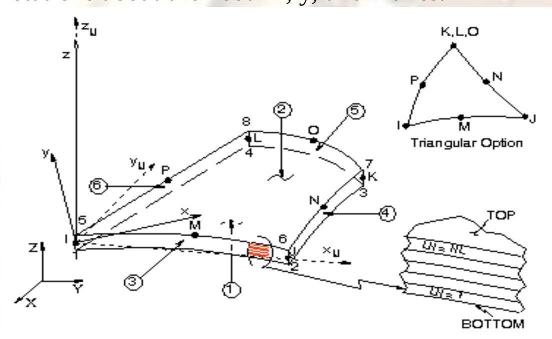


Figure : SHELL99 Linear Layered Structural Shell

4.2 Input Data

The element is defined by eight nodes, average or corner layer thicknesses, layer material direction angles, and orthotropic material properties. A triangular-shaped element may be formed by defining the same node number for nodes K, L and O. The input may be either in matrix form or layer form, depending upon KEYOPT (2). Briefly, the force-strain and moment-curvature relationships defining the matrices for a linear variation of strain through the thickness (KEYOPT (2) = 2) may be defined as:

$$\begin{Bmatrix} N \\ M \end{Bmatrix} = \left[\begin{array}{c|c} A & B \\ \hline B & D \end{array}\right] \begin{Bmatrix} \varepsilon \\ \kappa \end{Bmatrix} - \begin{Bmatrix} MT \\ BT \end{Bmatrix}$$

$$[A] = \begin{bmatrix} A_1 & A_2 & A_3 & A_4 & A_5 & A_6 \\ A_2 & A_7 & A_8 & A_9 & A_{10} & A_{11} \\ A_3 & A_8 & A_{12} & A_{13} & A_{14} & A_{15} \\ A_4 & A_9 & A_{13} & A_{16} & A_{17} & A_{18} \\ A_5 & A_{10} & A_{14} & A_{17} & A_{19} & A_{20} \\ A_6 & A_{11} & A_{15} & A_{18} & A_{20} & A_{21} \end{bmatrix} \; or \; [A]_{3\times 3} = \begin{bmatrix} A_1 & A_2 & A_3 \\ A_2 & A_4 & A_5 \\ A_3 & A_5 & A_6 \end{bmatrix}$$

Sub matrices [B] and [D] are input similarly. Note that all sub matrices are symmetric. {MT} and {BT} are for thermal effects. The layer number (LN) can range from 1 to 250.

In this local right-handed system, the x'-axis is rotated an angle THETA (LN) (in degrees) from the element x-axis toward the element y-axis. The total number of layers must be specified (NL). The properties of all layers should be entered (LSYM = 0). If the properties of the layers are symmetrical about the mid-thickness of the element (LSYM = 1), only half of properties of the layers, up to and including the middle layer (if any), need to be entered. While all layers may be printed, two layers may be specifically selected to be output (LP1 and LP2, with LP1 usually less than LP2).

The results of GA forms input to the FEA. Here Finite Element Analysis is done on the HS Carbon/Epoxy drive shaft.

4.3 Static Analysis

Static analysis deals with the conditions of equilibrium of the bodies acted upon by forces. A static analysis can be either linear or non-linear. All types of non-linearities are allowed such as large deformations, plasticity, creep, stress stiffening, contact elements etc. this chapter focuses on static analysis. A static analysis calculates the effects of steady loading conditions on a structure, while ignoring inertia and damping effects, such as those carried by time varying loads. A static analysis is used to determine the displacements, stresses, strains and forces in structures or components caused by loads that do not induce significant inertia and damping effects. A static analysis can however include steady inertia loads such as gravity, spinning and time varying loads.

In static analysis loading and response conditions are assumed, that is the loads and the structure responses are assumed to vary slowly with respect to time. The kinds of loading that can be applied in static analysis includes,

Externally applied forces, moments and pressures

Steady state inertial forces such as gravity and spinning

Imposed non-zero displacements

A static analysis result of structural displacements, stresses and strains and forces in structures for components caused by loads will give a clear idea about whether the structure or components will withstand for the applied maximum forces. If the stress values obtained in this analysis crosses the allowable values it will result in the failure of the structure in the static condition itself. To avoid such a failure, this analysis is necessary.

4.4 Boundary Conditions

The finite element model of HS Carbon/Epoxy shaft is shown in Figure .One end is fixed and torque is applied at other end,

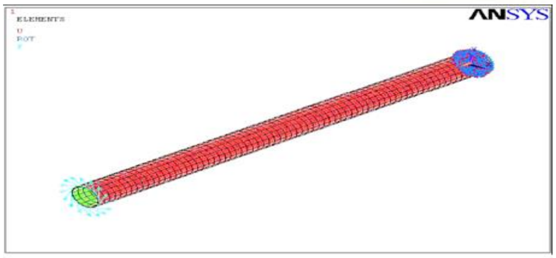


Figure : Finite element model of HS Carbon/Epoxy shaft

4.5 Modal Analysis

When an elastic system free from external forces is disturbed from its equilibrium position it vibrates under the influence of inherent forces and is said to be in the state of free vibration. It will vibrate at its natural frequency and its amplitude will gradually become smaller with time due to energy being dissipated by motion. The main parameters of interest in free vibration are natural frequency and the amplitude. The natural frequencies and the mode shapes are important parameters in the design of a structure for dynamic loading conditions.

Modal analysis is used to determine the vibration characteristics such as natural frequencies and mode shapes of a structure or a machine component while it is being designed. It can also be a starting point for another more detailed analysis such as a transient dynamic analysis, a harmonic response analysis or a spectrum analysis. Modal analysis is used to determine the natural frequencies and mode shapes of a structure or a machine component.

The rotational speed is limited by lateral stability considerations. Most designs are sub critical, i.e. rotational speed must be lower than the first natural bending frequency of the shaft. The natural frequency depends on the diameter of the shaft, thickness of the hollow shaft, specific stiffness and the length. Boundary conditions for the modal analysis are shown in Figure.

4.6 Buckling Analysis

Buckling analysis is a technique used to determine buckling loads (critical loads) at which a structure becomes unstable, and buckled mode shapes (The characteristic shape associated with a structure's buckled

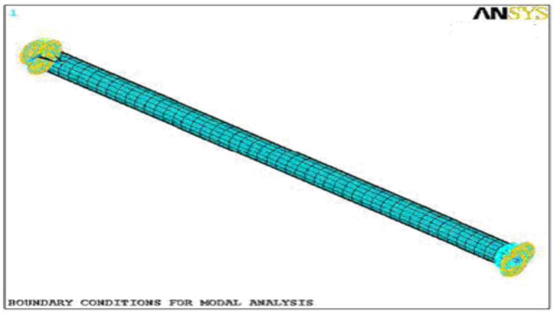


Figure: Boundary Conditions for the Modal Analysis

For thin walled shafts, the failure mode under an applied torque is torsional buckling rather than material failure. For a realistic driveshaft system, improved lateral stability characteristics must be achieved together with improved torque carrying capabilities. The dominant failure mode, torsional buckling, is strongly dependent on fiber orientation angles and ply stacking sequence.

5. Results

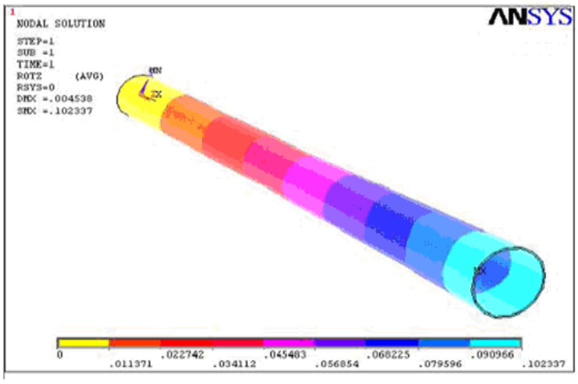
5.1GA Results

A composite drive shaft for rear wheel drive automobile was designed optimally by using genetic Algorithm for E-Glass/ Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy composites with the objective of minimization of weight of the shaft which is subjected to the constraints such as torque transmission, torsional buckling capacities and natural bending frequency.

5.2 Summarization of GA Results

The GA results are shown in Table

Parameter	**Steel**	**E-Glass / Epoxy**	**HS Carbon/Epoxy**	**HM Carbon/Epoxy**
do (mm)	90	90	90	90
L (mm)	1250	1250	1250	1250
tk (mm)	3.318	0.4	0.12	0.12
Optimum no. of layers	1	17	17	17
t (mm)	3.318	6.8	2.04	2.04
Optimum stacking sequence	-	[46/-64/-15/-13/ 39/-84/-28/20/-27]s	[-56/-51/74/-82/67/70/13/- 44/-75]S	[-65/25/68/- 63/ 36/- 40/-39/74/- 39]S
Weight (kg)	8.604	4.443	1.1273	1.1274
Weight saving (%)	-	48.36	86.90	86.90



5.3 Static Analysis of HS Carbon/Epoxy Drive Shaft

The twist about the axis of the shaft and 1st principal stress along the fiber direction are shown in Figures

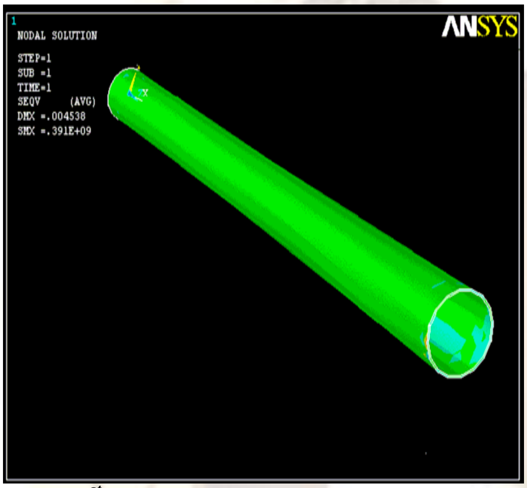


Fig: 1st principal stresses along longitudinal direction for HS Carbon / Epoxy shaft

5.4 Modal Analysis of HS Carbon/Epoxy Drive Shaft

Fig : 1st Vibration Mode shape of HS Carbon/Epoxy shaft

5.5 Buckling Analysis of HS Carbon/Epoxy Drive Shaft

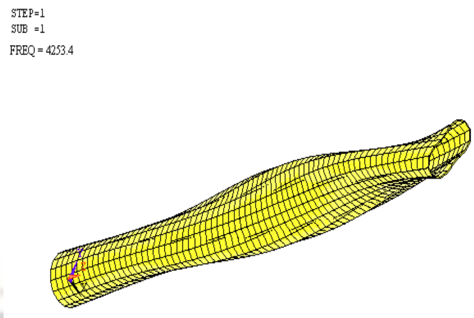


Figure : 1st Buckling Mode shape of HS Carbon/Epoxy shaft

6. Conclusion

The following conclusions are drawn from the present work.

The E – Glass / Epoxy, High Strength Carbon/ Epoxy and High Modulus Carbon/Epoxy composite drive shifts have been designed to replace the steel drive shaft of an automobile.

A composite drive shaft for wheel drive automobile has been designed optimally by using Genetic Algorithm for E – Glass/ Epoxy, High Strength Carbon/ Epoxy and High Modulus Carbon/ Epoxy composites with the objective of minimization of weight of the shaft which was subjected to the constraints such as torque transmission, torsional bucking capacities and natural bending frequency.

The weight savings of the E – Glass/ Epoxy, High Strength Carbon/ Epoxy and High Modulus Carbon/ Epoxy shafts were equal to 48.36%, 86.90%, and 86.90% of the weight of steel shaft respectively.

The optimum stacking sequence of E – Glass/ Epoxy, High Strength Carbon/ Epoxy and High Modulus Carbon / Epoxy shafts are shown in Table.

Table: Optimum Stacking Sequence

Material	Stacking Sequence
E – Glass / Epoxy	$[46/-64/-15/-13/39/-84/-28/20/-27]_s$
High Strength Carbon/ Epoxy	$[-56/-51/74/-8267/70/13/-44/-75]_s$
High Modulus Carbon/Epoxy	$[-65/25/68/-6363/-40/-39/74/-39]_s$

By using CLT, the variations of the stresses and strains along thickness of the E – Glass/ Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy composite drive shafts were plotted CLT. It has been observed that all the stresses were within the allowable limit.

The deflection of Steel, E – Glass/ Epoxy, High Strength Carbon/Epoxy and High Modulus

Carbon/Epoxy shafts were equal to 0.012407, 0.025262, 0.019288 and 0.012919 mm respectively.

The fundamental natural frequency of Steel, E – Glass/ Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/ Epoxy shafts were 9319.98, 6514.56, 7495.42 and 9270.28 rpm respectively.

The torsional buckling capacity of Steel, E – Glass/Epoxy, High Strength Carbon/ Epoxy and High Modulus Carbon/Epoxy shafts were 43857.96, 29856.45, 3772.11 and 3765.75 N-m respectively.

The torque transmission capacity of the composite drive shafts has been calculated by neglecting and considering the effect of centrifugal forces and it has been observed that centrifugal forces will be reduce the torque transmission capacity of the shaft.

Natural frequency using Bernoulli – Euler and Timoshenko beam theories was compared. The frequency calculated by using the Bernoulli Euler beam theory is high, because it neglects the effect of rotary inertia & transverse shear

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