ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
A Peer Reviewed Open Access International Journal

Optimal Design and Analysis of A Composite Drive Shaft of an Automobile

Devathala Manoj Kumar
Department of Mechanical Engineering,
Avanthi Institute of Engineering & Technology,
Cherukupally Vizianagaram, Andhra Pradesh - 535006, India.

J.Jagadesh
Department of Mechanical Engineering,
Avanthi Institute of Engineering & Technology,
Cherukupally, Vizianagaram, Andhra Pradesh - 535006, India.

ABSTRACT:

High-technology structures often have stringent requirements for structural dynamics. Suppressing vibrations is crucial to their performance. Passive damping is used to suppress vibrations by reducing peak resonant response.

Viscoelastic damping materials add passive damping to structures by dissipating vibration strain energy in the form of heat energy. The incorporation of damping materials in advanced composite materials offers the possibility of highly damped, light-weight structural components that are vibration-resistant.

The effect of damping on the performance of isotropic (like Steel) and orthotropic (like Carbon Epoxy & E-Glass Epoxy) structures are analysed by using Finite Element Analysis. Damping factor is increased by 16.3%, 9.2% and 6.2% for Steel Shaft, Carbon Epoxy Shaft and E- Glass Epoxy Shaft respectively, while embedding the rubber into the structure. The fundamental natural frequency is increased by 12.76%, 7.2% and 1.61% for Steel Shaft, Carbon Epoxy Shaft and E- Glass Epoxy Shaft respectively, when rubber is added to the structure. The deflection value is decreased by 21.38%, 12.71% and 6.74% for Steel Shaft, Carbon Epoxy Shaft and E- Glass Epoxy Shaft respectively, when rubber is embedded into the structure. A micro mechanical model is established and closed form expressions for the effective storage and loss properties of the damping material are given. An optimal relation between design parameters, such as the length, diameter, spacing, and Young's modulus of fibers and the shear modulus of viscoelastic matrix, is derived for achieving maximum damping performance. It is found that for maximum damping performance, the characteristic value of the composite should be set to 0.75.

Introduction:

All engineering structures experience vibratory motion whether in reference to the world's tallest building or a printed circuit board in a flight control computer. The effect of operating environments and inherent dynamic behaviour cause the transmission of periodic waves throughout a structure. In turn, the structure undergoes mechanical vibrations. These unwanted vibrations result in fatigue and catastrophic failure of structures. Hence, the control of vibrations is a serious concern for engineering design. A structure subjected to oscillatory deformation consists of a combination of kinetic and potential energy. In the case of real structures consists of energy dissipative elements as some of the energy is lost in each cycle. The amount of energy dissipated is a measure of the structure's inherent damping. Damping can be viewed as the conversion of mechanical energy of a vibrating structure into thermal energy, which ultimately is lost to the structure's environment [1].

COMPOSITE MATERIALS

Composite materials are those containing more than one bonded material, each with different material properties.

Cite this article as: Devathala Manoj Kumar, J.Jagadesh, "Optimal Design and Analysis of A Composite Drive Shaft of an Automobile", International Journal & Magazine of Engineering, Technology, Management and Research, Volume 5 Issue 4, 2018, Page 26-34.

The major advantages of composite materials are that they have a high ratio of stiffness to weight and strength to weight. A principal advantage of composite materials lies in the ability of the designer to tailor the material properties to the application

VISCOELASTIC MATERIAL:

A Viscoelastic material sometimes is called material with memory. This implies that a Viscoelastic material's behavior depends not only on the current loading conditions, but also on the loading history. They are characterized by possessing both viscous and elastic behavior. Figure 1.2 shows how various types of materials behave in the time domain [2].

A purely elastic material is one in which all the energy stored in the sample during loading is returned when the load is removed. As a result, the stress and strain curves for elastic materials move completely in phase. For elastic materials, Hooke’s Law applies, where the stress is proportional to the strain, and the modulus is defined at the ratio of stress to strain.

Components of pressure vessel:

The components of pressure vessel are as follows
1) Shell
2) Head
3) Stiffeners

Literature review:

Viscoelastic damping materials add passive damping to structures by dissipating vibration strain energy and generate heat energy. The incorporation of damping materials in advanced composite materials offers the possibility of highly damped, light weight structural components that are vibration resistant [4].

Concurring viscoelastic damping materials in composites has shown to be successful in greatly increasing the damping of composite structures. The damping performance, however, is often not as high in composites as in secondarily bonded composites, where the damping material does not undergo the cure process [5].

Substituting composite structures for conventional metallic structures has many advantages because of higher specific stiffness and specific strength of composite materials [6].

The fiber enhanced viscoelastic damping polymer is intended to be applied to lightweight flexible structures as a surface treatment for passive vibration control. A desirable packing geometry for the composite material is proposed, which is expected to produce maximum shear strain in the viscoelastic damping matrix [7].

A general method for modeling material damping in dynamical systems is presented and it is primarily concerned with a dissipation model based on viscoelastic assumptions [3].

Different numerical approaches for modeling and analyzing the behavior of structures having constrained layer damping. Two numerical studies are presented that reveal the accuracy limits of the different finite element modeling approaches for additive and integrally damped plate type structures. Now through the use of improved computational – based approaches (i.e. finite element method) along with the availability of reliable damping materials with accurate thermal and dynamic property characterizations, it is possible to incorporate damping treatments as part of the initial structural design process, thereby virtually eliminating sharp resonant peaks [9].

Cylindrical shells with a constrained damping layer treatment are studied using three theories. Constrained layer damping in structures is a very popular method to control resonant amplitudes of vibration. Shells of revolution (e.g., cylindrical and conical) find wide application in the aerospace industry [1-5].

An efficient method is described for finite element modeling of three-layer laminates containing a viscoelastic layer. Modal damping ratios are estimated from undamped normal mode results by means of the Modal Strain Energy (MSE) method [1-3].

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International Journal & Magazine of Engineering, Technology, Management and Research
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Problem definition

Problem I:

The torque transmission capability of the propeller shaft for passenger cars, small trucks, and vans should be larger than 3,500 Nm and fundamental natural bending frequency of the propeller shaft should be higher than 6,500 rpm to avoid whirling vibration. The outer diameter of the propeller shaft should not exceed 100 mm due to space limitations. The propeller shaft of transmission system is designed for following specified design requirements as shown in Table 3. 1. Due to space limitations the outer diameter of the shaft is restricted to 90.24 mm. The one-piece hollow composite drive shaft for rear wheel drive automobile should satisfy three design specifications, such as static torque transmission capability, torsional buckling capacity and the fundamental natural bending frequency. For given specification, the damping factor for Steel, carbon Epoxy and E-Glass Epoxy [4] are to be calculated and compared with and without damping material (Rubber).

Table 3.1 Problem Specification

Sl. No.	Parameter	Notation	Units	Value
1.	Torque	T	N-m	3500
2.	Max Speed	N	RPM	6500
3.	Length	L	m	1.250

RESULTS

Static ANALYSIS

Static analysis calculates the effects of steady loading conditions on a structure, while ignoring inertia and damping effects, such as those caused by time-varying loads. A static analysis, however, includes steady inertia loads (such as gravity and rotational velocity), and time-varying loads that can be approximated as static equivalent loads (such as the static equivalent wind and seismic loads commonly defined in many building codes).

Loads in a Static Analysis

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. Steady loading and response conditions are assumed; that is, the loads and the structure's response are assumed to vary slowly with respect to time. The kinds of loading that can be applied in a static analysis include:

- Externally applied forces and pressures
- Steady-state inertial forces (such as gravity or rotational velocity)
- Imposed (non-zero) displacements
- Temperatures (for thermal strain)
- Fluences (for nuclear swelling)

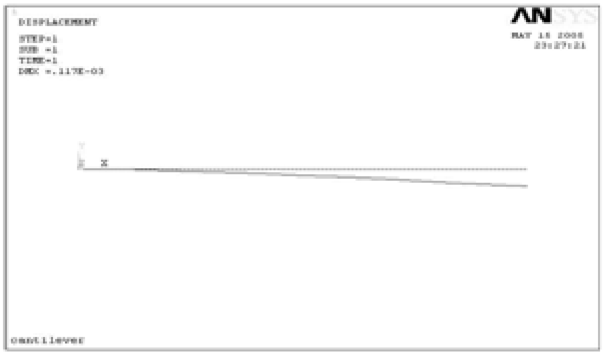
Static Analysis of Steel Shaft without viscoelastic Damping Material by Beam and Shell Element

In this part static deflection of the steel shaft is calculated and compared with ANSYS results. The specification for the shaft is given in the Table 4.1. For calculating the deflection, the cantilever boundary condition is taken by considering its self-weight.

Table 4.1. Specification for steel shaft

Sl. No.	Parameters	Values
1	Outer Diameter	0.09024 m
2	Thickness	2.1 e^{-3}

Static Deflection for Beam 3



Using ANSYS 7.0 the deflection value is calculated. The value is 0.117e -3 m. The deformed shape of the shaft is shown in the Fig 4.1.

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Table 4.2 Deflection Values for Beam 3

Results	Theoretical	ANSYS 7.0	Deviation in %
Static Deflection	1.16667 e^{-4} m	0.117 e^{-3} m	0.285

Steel Shaft without Damping Material for Shell 99

In this case shell element is taken to calculate the deflection value for steel shaft. Here Shell element is taken due to specify the number of layers to include the damping polymer [5]. Here steel shaft without damping material is considered and specifications are tabulated in table 5.1.

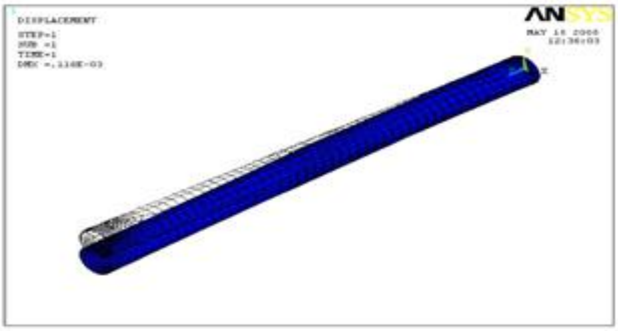

Fig 4. 2 Static Deflection for Steel Shaft

Using ANSYS 7.0 the deflection value is calculated. The value is 0.116*10-3 m. The deformed shape of the shaft is shown in the Fig 5.2.

Steel Shaft with Damping Material

In this type a damping material (i.e.) Rubber is inserted between the two layers of shaft and the deflection value is calculated using ANSYS. The specification of the shaft with damping material is shown in the Table 5.3

Table 4. 3 Specifications for Steel Shaft with Rubber

Sl. No.	Parameters	Values
1	Outer Diameter	0.09024 m
2	Thickness of each layer	1.05 e^{-3} m
3	Number of layers	3
4	Damping Material	Rubber
5	Element	Shell 99

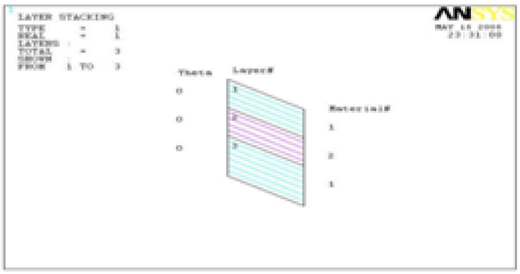

Fig 4.3 Stacking Sequence for Steel Shaft with Rubber

In Fig 4.3 stacking sequence of the steel shaft with damping material is shown.

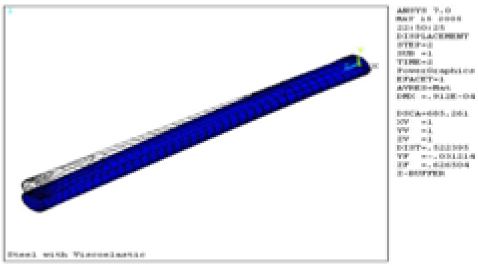

Fig 4.4 Deflection of Steel Shaft with Rubber

Using ANSYS 7.0 the deflection value is calculated. The value is 0.912 e -4 m. The deformed shape of the shaft is shown in the Fig 4.4.

Carbon Epoxy Shaft without Damping Material

In this case Carbon Epoxy shaft is modeled with 13 layers by considering the shell element. The specifications are shown in the Table 5.4

Table 4.4 Specification for Carbon Epoxy Shaft

Sl. No.	Parameters	Values
1	Outer Diameter	0.09024 m
2	Thickness of each layer	1.5 e^{-4} m
3	Number of layers	13
4	Element	Shell 99

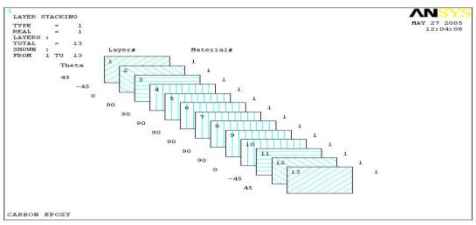

Fig 4.5 Stacking Sequence for Carbon Epoxy Shaft

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In fig 4.5 the stacking sequence of the Carbon Epoxy shaft without damping material is shown.

Using ANSYS 7.0 the deflection value is calculated. The value is 0.824 e -4 m. The deformed shape of the shaft is shown in the Fig 4.6.

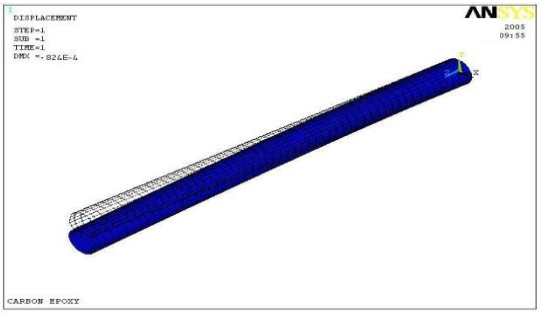

Fig 4.6 Static Deflection for Carbon Epoxy Shaft

Carbon Epoxy Shaft with Damping Material

In this case Carbon Epoxy shaft is modeled with damping material (Rubber) and it is incorporated in between the layers. The specification of the shaft is shown in the Table 4.5.

Sl. No.	Parameters	Values
1	Outer Diameter	.09024 m
2	Thickness of each layer	$1.5\ e^{-4}$ m
3	Number of layers	14
4	Damping Material	Rubber
5	Element	Shell 99

The stacking sequence of the Carbon Epoxy shaft with damping material (Rubber) is shown in the fig 4.7. Here the 7th layer is the rubber.

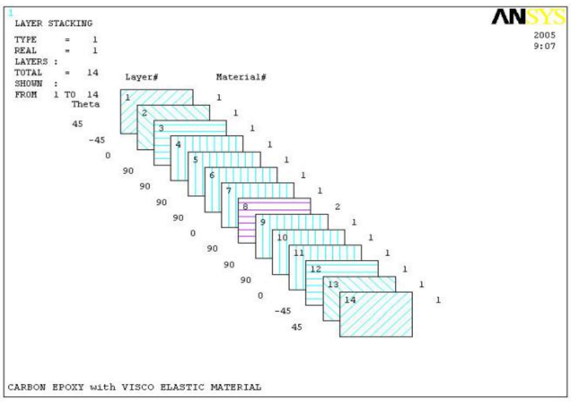

Fig 4.7 Stacking Sequence for Carbon Epoxy Shaft

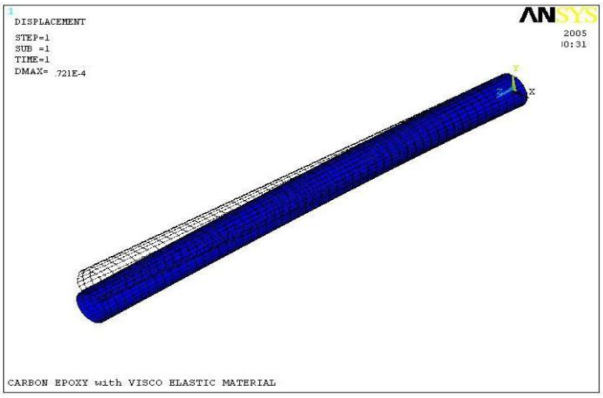

Fig 4.8 Static Deflection for Carbon Epoxy Shaft with Rubber

Using ANSYS 7.0 the deflection value is calculated. The value is 0.712 e -4 m. The deformed shape of the shaft is shown in the Fig 5.8.

E-Glass Epoxy Shaft without Damping Material

In this case E-Glass Epoxy shaft is modeled with 23 layers by using the shell 99 element [6]. The specifications are shown in the Table 4.6

Sl. No.	Parameters	Values
1	Outer Diameter	.09024 m
2	Thickness of each layer	$1.5\ e^{-4}$ m
3	Number of layers	23
4	Element	Shell 99

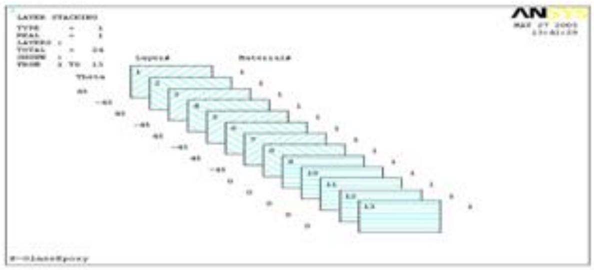

fig 4.9 Stacking Sequence Layers from 1 to 12

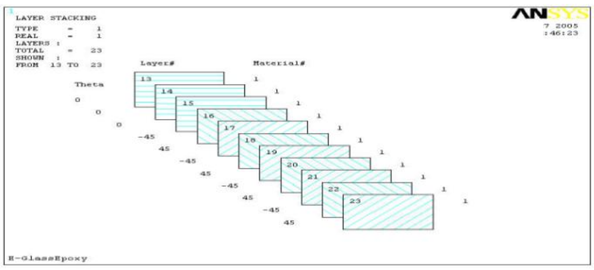

Fig 4.10 Stacking Sequence Layers from 13 to 23

The stacking sequence of the Carbon Epoxy shaft without damping material is shown in fig 4.9 and 4.10.

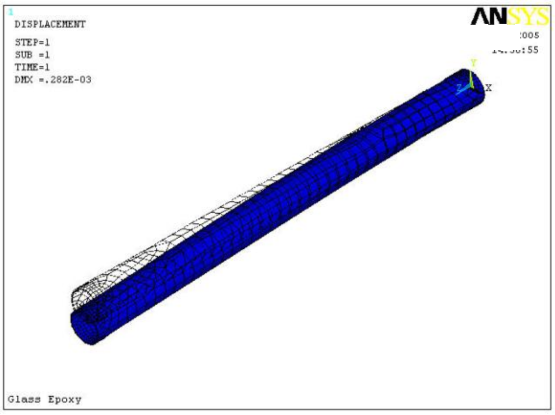

Fig 4.11 Static Deflection for E-Glass Epoxy Shaft

Using ANSYS 7.0 the deflection value is calculated. The value is 0.282 e -3 m. The deformed shape of the shaft is shown in the Fig 4.11.

E-Glass Epoxy Shaft with Damping Material

In this case E-Glass epoxy shaft is modeled with damping material (Rubber) and it is incorporated in between the layers. The specification of the shaft is shown in the Table 5.7.

Table 4.7 Specification for E- Glass Epoxy Shaft with Rubber

Sl. No.	Parameters	Values
1	Outer Diameter	.09024 m
2	Thickness of each layer	1.5 e^{-4} m
3	Number of layers	14
4	Damping Material	Rubber
5	Element	Shell 99

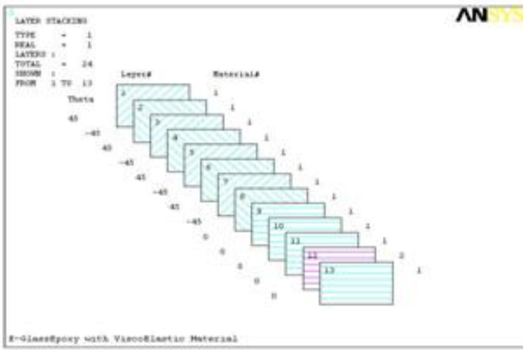

Fig 4.12 Stacking Sequence Layers from 1 to 13

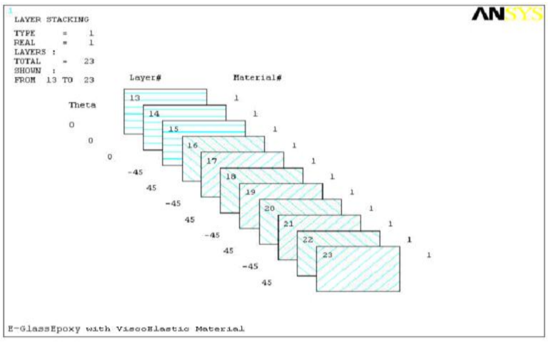

Fig 4.13 Stacking Sequence Layers from 14 to 24

The stacking sequence of the E-Glass Epoxy shaft with damping material (Rubber) is shown in fig 4.9 and 4.10. Here the 12th layer is the rubber.

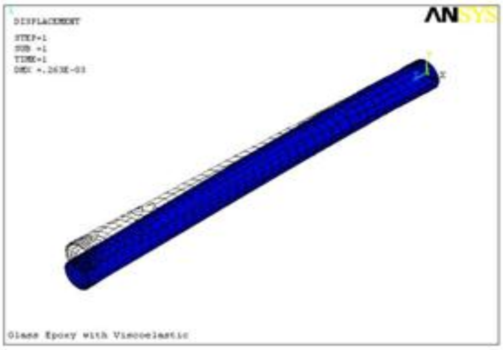

Fig 4.14 Static Deflection for E-Glass Epoxy Shaft with Rubber

Using ANSYS 7.0 the deflection value is calculated. The value is 0.263 e -3 m. The deformed shape of the shaft is shown in the Fig 5.14.

MODAL ANALYSIS

Any physical system can vibrate. The frequencies at which vibration naturally occurs, and the modal shapes which the vibrating system assumes are properties of the system, and can be determined analytically using Modal Analysis.

Modal analysis is the procedure of determining a structure's dynamic characteristics; namely, resonant frequencies, damping values, and the associated pattern of structural deformation called mode shapes. It also can be a starting point for another, more detailed, dynamic analysis, such as a transient dynamic analysis, a harmonic response analysis, or a spectrum analysis.

ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
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Modal analysis in the ANSYS family of products is a linear analysis. Any nonlinearities, such as plasticity and contact (gap) elements, are ignored even if they are defined. Modal analysis can be done through several mode extraction methods: subspace, Block Lanczos, Power Dynamics, Reduced, Unsymmetric and Damped. The damped method allows you to include damping in the structure [7].

USES OF MODAL ANALYSIS

Modal analysis is used to determine the natural frequencies and mode shapes of a structure. The natural frequencies and mode shapes are important parameters in the design of a structure for dynamic loading conditions. They are also required to do a spectrum analysis or a mode superposition harmonic or transient analysis. Another useful feature is modal cyclic symmetry, which allows reviewing the mode shapes of a cyclically symmetric structure by modeling just a sector of it.

MODAL ANALYSIS OF STEEL SHAFT WITHOUT DAMPING MATERIAL BY BEAM ELEMENT AND SHELL ELEMENT

In this part Modal analysis of the shaft is calculated using the same specifications given in the static analysis. Modal analysis is needed because the output of this analysis is used in the transient analysis to calculate the time step.

Modal Analysis of Steel Shaft using Beam Element without Rubber

Lateral vibration of beams

Consider the free-body diagram of an element of a beam shown in figure. Where M(r, t) is the bending moment, V(r, t) is the shear force and f (r, t) is the external force per unit length of the beam.

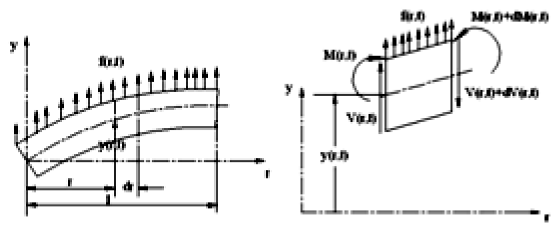

Fig 4.15. Beam in Bending

Euler -Bernoulli beam equations with external force, external moment is given by following equations.

$$EI\frac{\partial^4 y(r,t)}{\partial r^4}+\rho A\frac{\partial^2 y(r,t)}{\partial t^2}=f(r,t)$$
………… (1)

Using Assumed modes approach, by solving the above equation the Eigen values and mode shapes can be calculated. Using the variable separable method y(r, t) may be expressed by following equations.

$$y(r,t)=\sum_{i=1}^{\infty}\Phi_i(r)\,q_i(t)$$
………… (2)

$$EI\sum_{i=1}^{\infty}\Phi_i''''(r)q_i(t)+\rho A\sum_{i=1}^{\infty}\Phi_i(r)q_i''(t)=C_a\frac{\partial^2 V_a(r,t)}{\partial r^2}$$
………… (3)

$$EI\int_0^L\sum_{i=1}^{\infty}\Phi_i''''(r)q_i(t)dr+\rho A\int_0^L\sum_{i=1}^{\infty}\Phi_i(r)q_i''(t)dr=C_a\int_0^L\frac{\partial^2 V_a(r,t)}{\partial r^2}dr$$
………… (4)

By multiplying Φi (r) on both sides then apply the Orthogonality principle.

$$EI\int_0^L\sum_{i=1}^{\infty}\Phi_i''''(r)q_i(t)\Phi(r)dr+\rho A\int_0^L\sum_{i=1}^{\infty}\Phi_i(r)q_i''(t)\Phi(r)dr=C_a\int_0^L\frac{\partial^2 V_a(r,t)}{\partial r^2}\Phi(r)dr$$
…. (5)

$$\Phi_i''''(r)-\left(\frac{EI\omega^2}{\rho A}\right)\Phi_i(r)=0$$
………… (6)

$$\Phi_i^4(r)-\left(\lambda^4\right)\Phi_i(r)=0$$
………… (7)

The complementary solution of the above equation is given by

$$\Phi_i(r)=A_i\sin(\lambda_i r)+B_i\cos(\lambda_i r)+C_i\sinh(\lambda_i r)+D_i\cosh(\lambda_i r)$$
………… (8)

Boundary conditions of cantilever beam,

At the fixed end: $y(0,t)=0,\ EI\frac{\partial y(r,t)}{\partial r}=0$
………… (9)

At the free end: $EI\frac{\partial y^2(L,t)}{\partial r^2}=0,\ EI\frac{\partial y^3(L,t)}{\partial r^3}=0$
………… (10)

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Applying these boundary conditions the solution of differential equation transformed to

$$\Phi(r)=L\left[\cosh\lambda_i r-\cos\lambda_i r-\left(\frac{\cosh\lambda_i L+\cos\lambda_i L}{\sinh\lambda_i L+\sin\lambda_i L}\right)\times(\sinh\lambda_i r-\sin\lambda_i r)\right]$$

… (11)

Shear force at free end is zero, by applying this boundary condition in the above equation the above equation converted to following equation.

$$1+\cos\lambda_i L\,\cosh\lambda_i L=0 \qquad \ldots\ldots\ldots\ldots (12)$$

From the solution of the equation the λi value can be calculate. The natural frequency of the system is given by substituting λi can be calculated.

$$\omega_i=\sqrt{\frac{EI}{\rho A}}\lambda_i^2 \qquad \ldots\ldots\ldots\ldots (13)$$

Table 4.8 Modal Frequencies for Steel Shaft using BEAM 3

Mode Shapes	Theoretical Value (HZ)	Analytical Value Using ANSYS 7.0 (HZ)
1	58.29	57.289
2	359.68	355.98
3	1007.04	983.61
4	1972.94	1892
5	3260	3112.1

In Table 4.8 the theoretical values and analytical values for steel shaft using Beam 3 are tabulated.

DISCUSSIONS ON PROBLEM I

In this case all the results of Static, Modal and Transient Dynamic Analysis of Steel Shaft, Carbon Epoxy Shaft and E-Glass Epoxy shaft with and without damping polymer are tabulated and compared.

Comparison of Steel Shaft with and without Damping Material

Table 4.9 Comparison of Results for Steel Shaft

Type of the Shaft Results	Steel without Viscoelastic Material	Steel with Viscoelastic Material	Deviation (%)
Static Deflection (in m)	0.116 e-3	0.912 e-4	21.38
Fundamental Natural Frequency (in Hz)	57.587	64.937	12.76
Damping Factor	0.016766	0.0195	16.3

From the above results shown in the Table 4.9 it is found that

- The damping value and the fundamental natural frequency of the steel shaft is increased to 16.3% and 12.76 % respectively, when the shaft is embedded with viscoelastic polymer.
- The static deflection value of the steel shaft is decreased by 21.38% when the damping material is embedded.

Comparison of Carbon Epoxy Shaft with and without Damping Material

Table 4.10 Comparison of Results for Carbon Epoxy Shaft

Type of the Shaft Results	Carbon Epoxy without Damping Material	Carbon Epoxy with Damping Material	Deviation (%)
Static Deflection (in m)	0.824 e-4	0.721 e-4	12.71
Modal Frequency (in Hz)	67.601	72.443	7.2
Damping Factor	0.02657	0.029005	9.2

From the above results shown in the Table 4.10 it is found that

- The damping value and the fundamental natural frequency of the Carbon Epoxy shaft is increased to 9.2 % and 7.2 % respectively, when the shaft is embedded with viscoelastic polymer.
- The static deflection value of the Carbon Epoxy shaft is decreased by 12.71 % when the damping material is embedded.

Type of the Shaft Results	Glass Epoxy without Damping Material	Glass Epoxy with Damping Material	Deviation (%)
Static Deflection (in m)	0.282 e-3	0.263 e-3	6.74
Modal Frequency (in Hz)	34.817	35.379	1.61
Damping Factor	0.009273	0.00985	6.2

From the above results shown in the Table 4.11 it is found that

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- The damping value and the fundamental natural frequency of the E- Glass Epoxy Shaft is increased to 6.2 % and 1.61 % respectively, when the shaft is embedded with viscoelastic polymer.
- The static deflection value of the E- Glass Epoxy Shaft is decreased by 6.74 % when the damping material is embedded.

Conclusion

- The damping factor has been found out for Steel Shaft, Carbon Epoxy Shaft and E-Glass Epoxy Shaft with and without Viscoelastic polymer (Rubber).
- The Static, Modal and Transient Dynamic Analyses have been carried out using Finite Element Analysis.
- The following observations were made by embedding the Viscoelastic polymer (Rubber) into the structure.
- Damping factor increased by 16.3%, 9.2% and 6.2% for Steel, Carbon Epoxy and E- Glass Epoxy Shafts respectively.
- The fundamental natural frequency increased by 12.76%, 7.2% and 1.61% for Steel, Carbon Epoxy and E- Glass Epoxy Shafts respectively.
- The deflection value decreased by 21.38%, 12.71% and 6.74% for Steel , Carbon Epoxy and E- Glass Epoxy Shafts respectively.
- The increase in damping factor results in further suppression of vibrations and hence results in increased structural life.

REFERENCES

[1] Autar K. Kaw, "Mechanics of Composite Materials", CRC press, 2014

[2] Ahid D. Nashif, David I. G. Jones and John P. Henderson, "Vibration Damping", John Wiley & Sons Publication, 2015, Newyork.

[3] C. T. Sun and Y. P. Lu, "Vibration Damping of Structural Elements", Prentince Hall PTR, New Jeresy, 2012

[4] K. L. Napolitano, W. Grippo, J. B. Kosmatka and C. D. Johnson, "A comparison of two cocured damped composite torsion shafts", Composite Structures, Vol. 43, 2013, pp. 115-125.

[5] J. M. Biggerstaff and J. B. Kosmatka, "Damping Performance of Cocured Composite Laminates with Embedded Viscoelastic Layers", Journal of Composite Materials, Vol. 32, No.21/ 2013.

[6] Jin Kook Kim, Dai Gil Lee, and Durk Hyun Cho, 2001, "Investigation of Adhesively BondedJoints for Composite Propeller shafts", Journal of Composite Materials, Vol.35, No.11, pp. 999-1021.

[7] T. E. Alberts and Houchun Xia, "Design and Analysis of Fiber Enhanced Viscoelastic Damping Polymers", Journal of Vibration and Acoustics, Vol. 117, October 2010, pp. 398-404.

Author Details

Devathala Manoj Kumar
Department of Mechanical Engineering,
Avanthi Institute of Engineering& Technology,
Cherukupally Vizianagaram, Andhra Pradesh - 535006, India.

J.Jagadesh
Department of Mechanical Engineering,
Avanthi Institute of Engineering& Technology, Cherukupally, Vizianagaram, Andhra Pradesh 535006, India.