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Investigation of Dynamic Behavior and Durability of Double Cardan Drive Shafts with Composite Materials for High-Speed Vehicles

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ABSTRACT

This research aims to design and analyze a Double Cardan Driveshaft for high-speed vehicles using advanced computational tools and composite materials. The study employs a comprehensive methodology that combines theoretical concepts with practical design and analysis techniques. The driveshaft is designed using SOLIDWORKS software, and numerical simulations are performed using ANSYS Workbench. Modal analysis is conducted to determine the natural frequencies and mode shapes of the driveshaft, while harmonic response analysis investigates the driveshaft's behavior under the influence of torque at high rotational speeds. The study focuses on integrating composite materials, such as carbon fibre-reinforced polymers (Vf50%) and epoxy resin-infused fabric (biaxial), to enhance the performance and durability of the driveshaft. The numerical simulations are carried out in two stages, analysing the driveshaft made of traditional steel and then incorporating composite materials. The results demonstrate that integrating composite materials significantly improves the performance of the Double Cardan Driveshaft. The optimized design with epoxy resin-infused fabric (biaxial) exhibits a 338% increase in the safety factor and a 99.6% reduction in the damage factor compared to the traditional steel design.

1. INTRODUCTION

Drive shafts are essential components in various vehicles, playing a crucial role in ensuring efficient power transmission and smooth operation. As a key mechanical link between the engine and the wheels, a drive shaft's main function is to transfer the power generated by the engine to the vehicle's wheels, allowing for forward and backward motion. Additionally, it maintains proper alignment between the engine and the wheels, ensuring that power is transferred efficiently without significant misalignment. This alignment is necessary for a vehicle's smooth and controlled operation. Furthermore, drive shafts are important for reducing and absorbing vibrations generated during vehicle operation, contributing to a more comfortable and stable ride for the occupants. In vehicles with independent suspension systems, drive shafts must also accommodate the vertical movement of the wheels while maintaining constant power transmission, allowing each wheel to move independently without compromising overall performance [1].

The versatility of automotive drive shafts extends beyond passenger vehicles, finding applications in various domains such as rear-wheel drive vehicles, four-wheel drive (4WD) and all-wheel drive (AWD) vehicles, trucks and commercial vehicles, performance and racing vehicles, off-road and utility vehicles, construction and agricultural equipment, industrial applications, and even marine and boating. In each of these applications, drive shafts are designed to handle specific power and torque requirements, often utilizing lightweight and durable materials to optimize performance. As a result, the study and optimization of drive shafts have become increasingly important in the automotive industry, with researchers and engineers continuously seeking ways to enhance their efficiency, durability, and overall performance to meet the ever-growing demands of modern vehicles [2].

Despite the advancements in drive shaft technology, conventional steel drive shafts face limitations in terms of their ability to cope with the demands of high-speed applications. The inherent characteristics of steel, such as its weight and sensitivity to vibrations, can hinder vehicle performance and fuel efficiency. Moreover, the increasing complexity of modern drive systems necessitates the development of advanced design and analysis techniques that can accurately predict and optimize drive shaft behavior under various operating conditions.

2. LITERATURE REVIEW

Previous research has investigated various aspects of drive shaft design and analysis, focusing on the use of advanced composite materials, structural performance, and the impact of manufacturing techniques on the mechanical behavior of these critical components. Samuel and Tayong [3] studied the structural performance of Carbon Fiber Reinforced Polymers (CFRP) in drive shafts, highlighting their mechanical advantages over traditional metallic shafts in the automotive sector. The researchers used COMSOL Multiphysics software to model drive shafts made from Steel AISI 4340, Aluminum, and CFRP, subjecting them to various mechanical excitations. The CFRP drive shaft, designed using the layered material function, showed the most favorable mechanical behavior with a [90°/0°/-45°/+45°] lay-up. Analytical and numerical calculations of natural frequencies revealed that CFRP had the highest fundamental natural frequency compared to metallic materials. Moreover, fatigue and critical buckling analyses demonstrated the superior fatigue usage factor and lighter weight of CFRP drive shafts compared to steel and aluminum tubes. Similarly, Kamboh et al. [4] focused on the design and analysis of drive shafts using advanced composite materials, investigating the causes of shaft failure. The researchers compared light alloys and composite materials to heavy steel using Finite Element Analysis (FEA) and analytical estimation, exploring the challenges that arise when designing shafts under different loading conditions and identifying areas for further research, particularly regarding discontinuities and combined loading. The study also considered replacing traditional steel drive shafts with lighter titanium alloys and Kevlar composites, assessing their exceptional, effective, and natural frequency bending properties. In another study, Kalaskar et al. [5] investigated the impact of cracks on the performance of a hybrid aluminum-composite drive shaft, conducting experimental modal analysis and finite element analysis (FEA) to evaluate the natural frequencies of the shaft with and without cracks. The findings emphasized the importance of early crack detection in maintaining the structural integrity of composite drive shafts and presented various nondestructive techniques for crack identification. Additionally, Kumar et al. [6] conducted a comprehensive analysis examining the structural and modal aspects of a single-piece drive shaft for heavy vehicles, focusing on material selection. The study compared conventional materials like Steel SM45 C and Stainless Steel with composite materials such as HS carbon epoxy and E Glass Polyester Resin Composite, indicating that single-piece composite material drive shafts offer advantages over conventional two-piece steel shafts, including higher specific strength, longer life, reduced weight, higher critical speed, and greater torque-carrying capacity. Finite Element Analysis (FEA) was employed for both structural and modal analysis under real-world conditions, and the findings were validated against experimental results from the literature, highlighting the superiority of composite materials in drive shaft applications for heavy vehicles. Furthermore, Gonsalves et al. [7] explored the dynamics of composite material shafts in high-speed rotor-bearing systems, aiming to enhance rotor dynamics and understand the effects of internal damping in these shafts. The researchers conducted a comprehensive analysis using practical rotor-bearing systems, considering factors such as mode shapes and frequency values, contributing to the understanding of composite materials' suitability for high-speed applications and offering implications for the design and optimization of rotor systems. Moreover, Kumar et al. [8] investigated the vibration characteristics of drive shafts made from conventional and composite materials using Finite Element Analysis, comparing the vibration responses of these materials in heavy vehicle drive shafts and finding that composite materials, particularly HS carbon epoxy, exhibited better natural frequency and critical speed, highlighting their suitability for heavy vehicle applications. This study provides valuable insights into the design and material selection of drive shafts for such vehicles. In addition, Shinde and Sawant [9] employed finite element analysis to investigate the performance and durability of glass-epoxy composite drive shafts for light motor vehicles, considering aspects such as strength and stability under various operational conditions. The findings from this study offer significant insights into the design and optimization of drive shafts for light vehicles, emphasizing the benefits of glass-epoxy composites. Another study conducted by Yang et al. [10] focused on the random fatigue life analysis of carbon fiber-reinforced plastic (CFRP) for automotive drive shafts, incorporating environmental temperature data to understand how temperature variations affect the fatigue life of CFRP in automotive applications, which is crucial for ensuring reliability and durability in varying environmental conditions. Finally, Sun et al. [11] investigated the vibration properties of CFRP drive shafts, focusing on the impact of filament winding technology on their vibration characteristics. The researchers combined finite element analysis (FEA) and experimental methods to explore the influence of various fibre ply angles and thicknesses on the natural frequency and damping of the CFRP shafts, providing insights crucial for their design and application in various industries.

The research gaps for this study focus on the design and analysis of high-performance drive shafts, specifically the underexplored aspect of harmonic response analysis. Previous studies have largely overlooked this crucial element, leaving a significant gap in the literature regarding the performance of drive shafts under high rotational speeds. By investigating this aspect in detail, this research provides new insights into the dynamic behavior of composite drive shafts, thereby addressing a critical gap in current driveshaft technology.

This research aims to utilize the potential of composite materials and advanced computational tools to enhance the design and analysis of high-performance drive shafts. The study integrates carbon fiber reinforced polymers (Vf50%) and epoxy resin-infused fabric (biaxial) to exploit their superior mechanical properties. Advanced computational tools, including SOLIDWORKS for 3D modeling and ANSYS Workbench for numerical simulations, are employed to conduct comprehensive analyses under various loading scenarios. Modal analysis and harmonic response analysis are performed to compare the performance of traditional steel and composite material designs, ultimately contributing to the development of safer, more efficient, and reliable high-speed vehicles.

3. METHODOLOGY

This section presents the design and analysis of a Double Cardan Driveshaft for high-speed vehicles. It covers the fundamental aspects of machine design, stress analysis, fatigue failure criteria, composite materials, and their failure criteria. The theory of modal analysis and harmonic response analysis is also discussed, essential for understanding the drive shaft's dynamics. The design and numerical analysis of the Double Cardan Driveshaft is the focus of this section. The design uses SOLIDWORKS software, and numerical analysis is performed using ANSYS Workbench in two stages. The first stage examines the steel shaft, while the second and third stages incorporate composite materials. The analysis includes meshing, boundary condition application, loading, and subjecting the model to various operating conditions to simulate real-world scenarios. Stress, vibration, and dynamic analyses are conducted to evaluate the drive shaft's performance and structural integrity. The results are presented along with the final driveshaft model, mesh results, boundary conditions, and applied loads. This section provides a comprehensive approach to developing a high-efficiency Double Cardan Driveshaft by combining theoretical concepts with practical design and analysis techniques.

3.1 Machine design

Machine design is a complex process that combines creativity and analytical skills to develop functional machines. It involves planning and designing individual components and integrating them into a system that performs a specific function. Designing a machine requires understanding mechanical principles and considering practical constraints such as material properties, manufacturing techniques, sustainability, and cost. The process starts with conceptualization, followed by detailed engineering and analysis to optimize components for performance and reliability. It often involves simulation and prototyping to validate design choices and refine the machine before final production. Effective machine design is crucial for innovation in technology and industry, enabling the development and improvement of machinery [12].

3.1.1 Axial stress

Axial stress is the stress that happens when normal forces are applied along the length of a structure, such as in columns or struts. This stress could be either (tensile or compressive) nature. It can be calculated using the equation below [13]:

$$\sigma_{axial} = \frac{F}{A} \tag{1}$$

In this equation, σ_{axial} represents the axial stress, F is the axial force (tensile or compressive), and A is the cross-sectional area of the object. This is essential for determining whether a material will withstand loading conditions without failing under tension or compression [13].

3.1.2 Shear stress

Shear stress is caused by shear forces that make the material particles of an object slip past each other, such stress is most often observed in beams and shafts. It can be calculated by the equation below [13]:

$$\tau_{shear} = \frac{V}{A} \tag{2}$$

Here, τ_{shear} is the shear stress, V is the shear force, and A is the area over which the shear force is acting. Calculating shear stress is critical for ensuring components like beams, bolts, and pins can handle such forces without shearing [13].

3.1.3 Flexural stress (from bending moment)

A material is under flexural stress, or bending stress when it is subjected to a moment of flexure. It is widely analyzed in beams. It can be calculated by the equation below [13]:

$$\tau_{\text{shear}} = \frac{M.y}{I} \tag{3}$$

σ bending is the bending stress, M is the bending moment, y is the distance from the neutral axis to the outermost fiber of the beam (the height of the cross-section), and I is the moment of inertia of the cross-section. This equation helps designers ensure that beams are capable of sustaining loads without bending excessively or failing [13].

3.1.4 Flexural stress (from bending moment)

Torsional stress occurs when torque is applied to a cylindrical object, for example, a shaft, and causes it to twist. It can be calculated by the equation below [13]:

$$\tau_{\text{torsional}} = \frac{T.r}{J} \tag{4}$$

The given formula expresses τ torsional as torsional stress, T as the torque, r as the radius of the shaft, and J as the polar moment of inertia of the cross-section. Shafts and other rotating elements should be designed so as not to fail because of twisting under load.

3.2 Composite materials

Composite materials are made up of two or more separate materials combined to produce enhanced mechanical or physical properties. These materials are typically produced from a blend of metals, polymers, ceramics, and reinforcement materials like fibers. As technology has advanced, the scale of these materials has been improved to include nano-sized components such as nanotubes and nanofibers, which exhibit excellent properties. Composite materials have become increasingly important, particularly when conventional materials cannot offer the required performance. They are preferred in industries such as aerospace, automotive, and civil infrastructure, where strength-to-weight ratio, endurance, and resistance to environmental factors are crucial. Composites can display properties that are unattainable with individual constituent materials alone. The introduction of nanocomposites, which are reinforced at the nanoscale, has further advanced the field of composite materials, making them irreplaceable in engineering applications with lightweight structures in aircraft and high-performance uses in sports equipment [14].

3.2.1 Composite material failure criteria

Maximum strain criterion

The maximum strain criterion is a failure theory primarily employed in the case of orthotropic materials like composite laminates. It fails when the strain in any major direction of the material in question violates its corresponding allowable strain limit. This parameter generalizes the hypothesis of maximum normal strain, which is usually employed for isotropic materials, to handle the anisotropic behaviour of composites by determining strains in the directions of principal material axes. It can be calculated using the equations below [14]:

$$-\epsilon_L^- < \epsilon_1 < \epsilon_L^+ \tag{5}$$

$$-\epsilon_T^- < \epsilon_2 < \epsilon_T^+ \tag{6}$$

$$|\gamma_{12}| < \gamma_{LT} \tag{7}$$

Maximum stress criterion

In a basic failure theory designed for orthotropic materials such as composites, the maximum stress criterion predicts failure when the stress in any principal material axis equals and exceeds the corresponding permissible stress limit. This criterion applies the maximum normal stress theory used for isotropic materials to the peculiarities of composites by analysis of stresses in main material directions [14]. It can be calculated using the equations below [14]:

$$-s_L^- < \sigma_1 < s_L^+ \tag{8}$$

$$-s_T^- < \sigma_2 < s_T^+ \tag{9}$$

$$|\tau_{12}| < \tau_{LT} \tag{10}$$

where, σ_1, σ_2: Stress components along the longitudinal and transverse directions of the lamina, respectively, s_L^-+, s_L^+: Ultimate tensile and compressive strengths in the longitudinal direction, s_T^-, s_T^+: Ultimate tensile and compressive strengths in the transverse direction. τ_{12}: Shear stress in the plane of the lamina. τ_{LT}: Ultimate shear strength allowable before failure.

Quadratic interaction criteria

The quadratic interaction criteria are complex failure theories of composite materials with consideration of the interaction between different stress components. Contrary to the more straightforward failure theories which consider stress or strain to be applied independently, these criteria define failure by the combined effect of stresses through a quadratic formula. This method is more all-encompassing because it accounts for the intricate relationship's characteristics in anisotropic materials such as composites [14]. It can be calculated using the equations below [14].

$$F_i\sigma_i + F_{ij}\sigma_i\sigma_j = 1 \tag{11}$$

where, σ_i: Stress components, *i*: represents different directions (e.g., longitudinal, transverse, shear). F_i: Coefficients representing the linear influence of the stress components. F_{ij}: Coefficients representing the interaction between different stress components.

3.3 Modal analysis

Modal analysis is a vital engineering concept, which deals with identifying the natural vibrational properties of systems. It deals with the determination of the natural frequencies, mode shapes and damping factors of a structure or mechanical system. The natural modes determine the vibrations the system will produce when left to its own devices free from any external forces. It is critical to know that these characteristics duly affect the dynamic response of the system to external forces. Modal analysis helps engineers to forecast the responses of structures to various types of loads and vibrations, which is very important for engineering and safety evaluations. The process utilizes theoretical and experimental techniques whereas the theoretical analysis uses numerical methods such as the finite element method for simulations. In contrast, the experimental modal analysis relies on physical testing to validate the numerical models [15].

3.3.1 Natural frequency equation

$$\omega_n = \sqrt{\frac{k}{m}} \tag{12}$$

The above equation is the fundamental equation for calculating the undamped natural frequency (ω_n) of a single-degree-of-freedom (SDOF) system, where k is the stiffness of the system, and *m* is the mass.

3.3.2 Damped natural frequency

$$\omega_d = \omega_n\sqrt{1-\zeta^2} \tag{13}$$

In systems where damping is present, the damped natural frequency (ω_d) is used. Here, ζ is the damping ratio, and (ω_n) is the undamped natural frequency.

3.3.3 Basic equation of motion in modal space

$$[M][\psi]\ddot{y} + [K][\psi]y = 0 \tag{14}$$

where, $[M]$ is the mass matrix, $[K]$ is the stiffness matrix, $[\psi]$ represents the mode shapes, and y denotes the modal coordinates.

3.4 Harmonic response analysis

Harmonic response analysis, also called frequency response analysis, is a dynamic linear analysis method employed to determine a system's response to excitation at certain frequencies. It uses the output of a previously performed modal analysis to get the required input frequencies. Modal superposition is used in harmonic response analysis to determine the system response to steady-state sinusoidal loads at specified frequencies that can be in and out of phase with each other. This method is often used to evaluate stresses in machines, vehicles, or process equipment under continuous harmonic loading, for instance, rotor imbalance or cyclic loading in combustion engines [16].

3.4.1 Equation of motion

$$M\ddot{u} + C\dot{u} + Ku = F \tag{15}$$

This equation is the generalized dynamic equilibrium equation, where *M* is the mass matrix, *C* is the damping matrix, *K* is the stiffness matrix, and *F* is the forcing function.

3.4.2 Forcing function and displacement

$$F = F_{max}e^{i\psi}e^{i\Omega t} \tag{16}$$

$$u = u_{max}e^{i\phi}e^{i\Omega t} \tag{17}$$

Here, F_{max} and u_{max} are the amplitude of the force and displacement respectively, ψ and ϕ are their phase shifts, and Ω is the imposed circular frequency.

3.4.3 Harmonic equations of motion

$$-\Omega^2 M + i\Omega C + K[u_1 + iu_2] = [F_1 + iF_2] \tag{18}$$

This equation converts the complex notation into a form that includes both the real and imaginary parts of displacement and force vectors.

3.4.4 Modal superposition method

$$y_{jc} = \frac{f_{jc}}{(w_j^2 - \Omega^2) + i(2w_j\Omega\zeta_j)} \tag{19}$$

This equation is employed in the modal superposition method, with y_{jc} representing complex modal coordinates, f_{jc} denoting complex modal forces, w_j representing natural circular frequencies, Ω denoting the excitation frequency, and ζ_j is the damping ratio [17].

3.4.5 Modal complex displacements calculation

$$u_c = \sum_{j=1}^{n} \phi_j y_{jc} \tag{20}$$

In this formulation, u_c represents the complex displacement vector, ϕ_j are the mode shapes, and y_{jc} is the complex modal coordinates computed previously [17].

3.5 Drive shaft design and numerical analysis

This section focuses on the design and numerical simulation of a Double Cardan Driveshaft for high-speed vehicles to develop a new design that improves performance, durability, and reliability. The section is divided into two main parts. The first part deals with the design of the driveshaft using SOLIDWORKS software, where a 3D model is created with unique modifications. The second part involves numerical simulations using ANSYS Workbench, conducted at high rotational speeds to replicate the demanding operating conditions of high-speed vehicles. The numerical simulations are carried out in two stages. The first stage simulates the driveshaft using steel, without composite materials, to provide a baseline for the driveshaft's behavior. The second stage includes composite materials in the driveshaft design, with a thickness of 5mm, consisting of 10 layers, each 0.5mm thick, to explore their potential benefits in performance enhancement. During the simulations, various operating conditions, such as loads, speeds, and vibrations, are applied to the model to closely resemble real-world scenarios. Stress analysis, vibration analysis, and dynamic analysis are performed to predict the driveshaft's performance under these conditions. The results are analyzed to assess the design's structural integrity, load-bearing capacity, and dynamic stability at high rotational speeds. By using SOLIDWORKS for design and ANSYS Workbench for numerical simulations, this section aims to develop an optimized Double Cardan Driveshaft design that improves performance and reliability in high-speed applications. The findings from this approach contribute to the advancement of driveline systems and provide insights for future design improvements in the automotive industry.

3.5.1 Double Cardan Driveshaft design

This component was designed using SOLIDWORKS software, as clarified before due to its accuracy and ease of use for such applications [18]. The figures below illustrate the final model, along with the dimensions of the component.

The 3D shape of the Driveshaft that used in this study is shown in Figure 1.

All dimensions of the driveshaft are shown in Figure 2.

3.5.2 Double Cardan Driveshaft numerical analysis

Numerical analysis without composite material

In this study, two modules in the ANSYS software will be used: (Modal analysis) and (harmonic response analysis). Modal analysis will be performed to determine the natural frequencies of the drive shaft. These natural frequencies will then be linked to the harmonic response analysis to investigate the stresses and deformations on the shaft at different frequencies while applying the external load.

After completion of the design with SOLIDWORKS, the next phase is numerical simulation and analysis employing ANSYS Workbench. Meshing is the first stage in numerical analysis to achieve correct solutions. In this work, the body sizing with a 10mm element size was used to get an accurate mesh. Figure 3 illustrates the meshing results.

The process of meshing includes splitting the driveshaft into smaller segments with nodes in between. The mesh quality impacts the accuracy of the numerical solutions. After the finishing of the mesh, the remaining analysis steps can be performed like imposing boundary conditions, specifying the material properties, and configuring the required analysis types. The next sections will cover the detailed analyses conducted to determine the performance and durability of the Double Cardan Driveshaft under different loading conditions.

Regarding the boundary conditions, a fixed support will be applied to one end of the drive shaft, as shown in Figure 4 below.

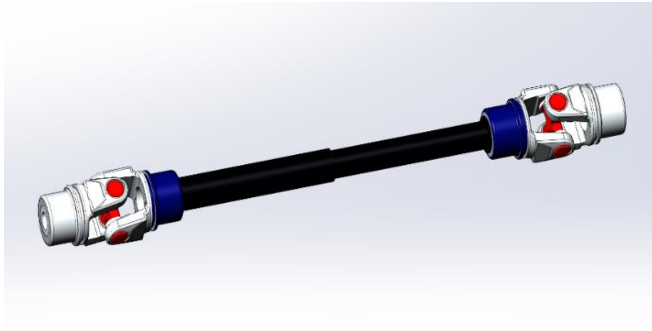


Figure 1. Final 3D model of Double Cardan Driveshaft

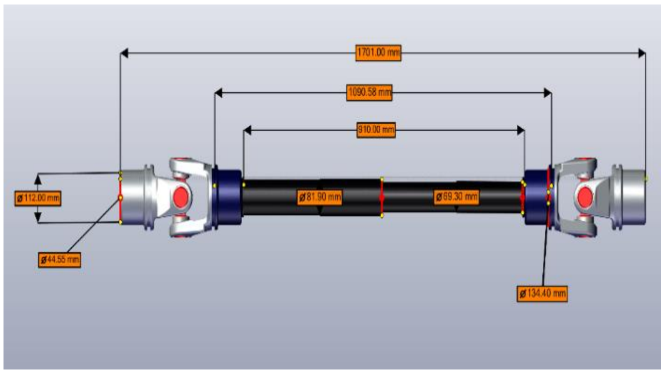


Figure 2. Double Cardan Driveshaft dimensions

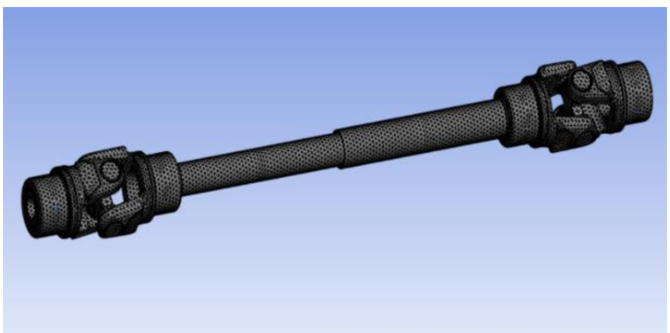


Figure 3. Double Cardan Driveshaft mesh results

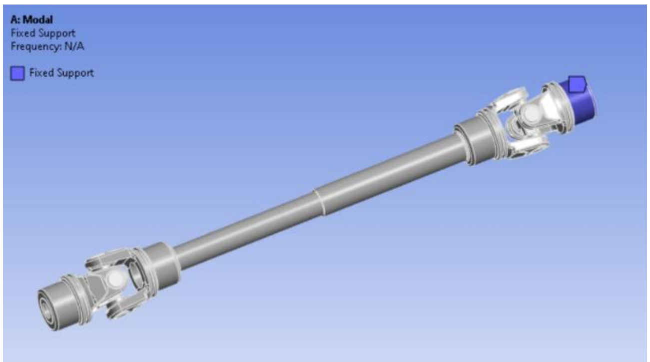


Figure 4. Fixed support on Double Cardan Driveshaft

For the applied load, torque will be applied to the other end of the drive shaft. This torque will be determined at high speeds using the relationship between torque, horsepower, and RPM, as shown in the equation below [19].

$$\text{Horsepower (HP)} = \frac{\text{Torque (T in lb-ft)} \times \text{RPM}}{5252} \qquad (21)$$

Since the study will be conducted at high speeds, a car with 600 horsepower and 6000RPM will be considered. Based on the equation above, the resulting load (torque) applied to the drive shaft will be:

Torque(lb.ft)=(5252×Horsepower)/RPM=(5252×600)/6000=525.2lb.ft.

Figure 5 shows the calculated moment on the driveshaft.

The project schematic of the first case study is shown in Figure 6.

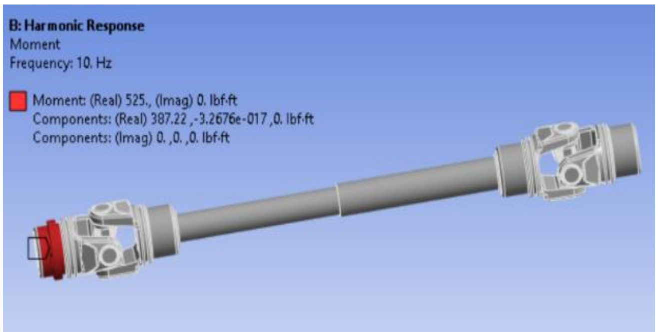


Figure 5. Torque applied on Double Cardan Driveshaft

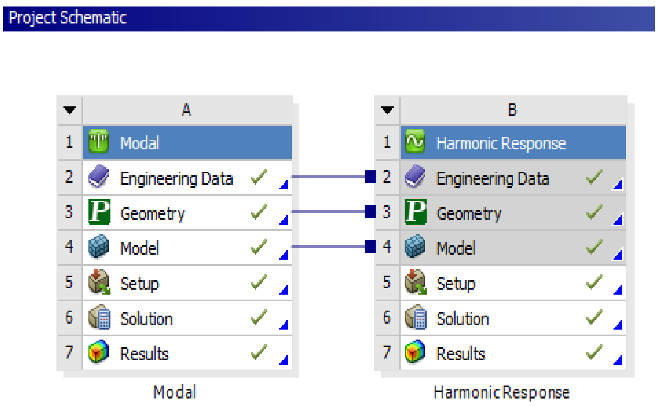


Figure 6. Project schematic for the first case study

Numerical analysis with composite material

Composite materials will be added to the second and third case studies. In the second case study, the composite material will be (Carbon matrix, carbon fibre reinforced (Vf50%)), and in the third case study, it will be (Composite, Epoxy CF, resin-infused fabric, biaxial). Modal Analysis and Harmonic Response Analysis will be used to investigate the frequencies, and the ACP (ANSYS Composite PrepPost) module will be utilized to add the composite materials to the middle section of the Double Cardan Drive Shaft. Ten layers of composite materials will be added, each with a thickness of 0.5mm, resulting in a total thickness of 5 mm. Each layer will have a specific orientation, with the stacking sequence of [0, 45, -45, 0, 90]2s, representing the fiber orientation in each layer of the composite material.

Table 1 shows the composite material layers and directions.

The properties of the two composite materials used in this research are shown in Tables 2 and 3.

The area where the composite material was added is shown in Figure 7.

Table 1. Carbon matrix, carbon fiber reinforced (Vf: 50%) properties [20]

Layer	Material	Thickness (mm)	Angle (°)
10	Composite, Carbon fiber reinforced carbon matrix	0.5	90
9	Composite, Carbon fiber reinforced carbon matrix	0.5	0
8	Composite, Carbon fiber reinforced carbon matrix	0.5	-45
7	Composite, Carbon fiber reinforced carbon matrix	0.5	45
6	Composite, Carbon fiber reinforced carbon matrix	0.5	0
5	Composite, Carbon fiber reinforced carbon matrix	0.5	90
4	Composite, Carbon fiber reinforced carbon matrix	0.5	0
3	Composite, Carbon fiber reinforced carbon matrix	0.5	-45
2	Composite, Carbon fiber reinforced carbon matrix	0.5	45
1	Composite, Carbon fiber reinforced carbon matrix	0.5	0

Table 2. Carbon matrix, carbon fiber reinforced (Vf: 50%) properties [20]

Properties	Value
Density (kg/m^3)	1700
Modulus of Elasticity (Gpa)	94.87
Poisson's Ratio	0.3198
Tensile Strength, Yield (Mpa)	503.4
Tensile Strength, Ultimate (Mpa)	503.4

Table 3. Composite, epoxy CF, resin-infused fabric, biaxial [20]

Properties	Value
Density (kg/m^3)	1515
Modulus of Elasticity (Gpa)	60.93
Poisson's Ratio	0.076
Tensile Strength, Yield (Mpa)	642.1
Tensile Strength, Ultimate (Mpa)	642.1

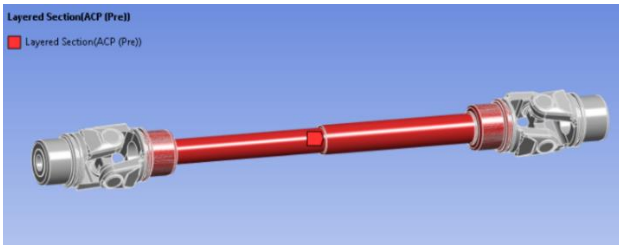


Figure 7. Double Cardan Driveshaft after composite material layers are added

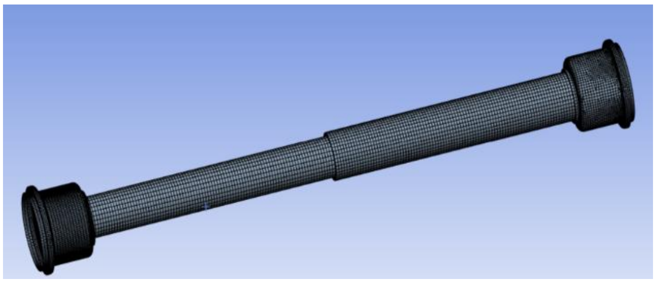


Figure 8. Mesh results for part of the driveshaft that the composite material added on it

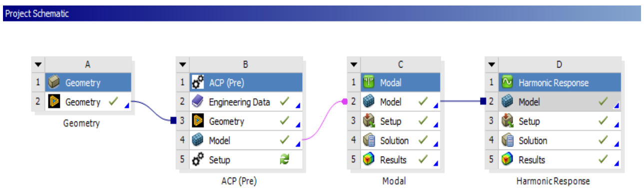


Figure 9. Project schematic for the second and third case studies

The meshing process was performed for the part to which the material composite material was added to concentrate the solution more on it, and the meshing results are shown in Figure 8.

The project schematic of the second and third case study is shown in Figure 9.

For clarification, the boundary conditions remain the same as in the previous section and are not discussed in this section. The results of all three case studies, including the analysis with composite materials, will be presented in the next chapter.

4. RESULTS AND DISCUSSION

This section presents a detailed review of the analysis results across three cases. In the first case, called the traditional scenario, a drive shaft made from steel is used. In the second and third cases, composite materials are added. These composite materials have 10 layers, each 0.5mm thick. The total thickness of the composite is 5mm. The results are then thoroughly reviewed and compared. This is done to find the best design and the strongest and most suitable material for this analysis.

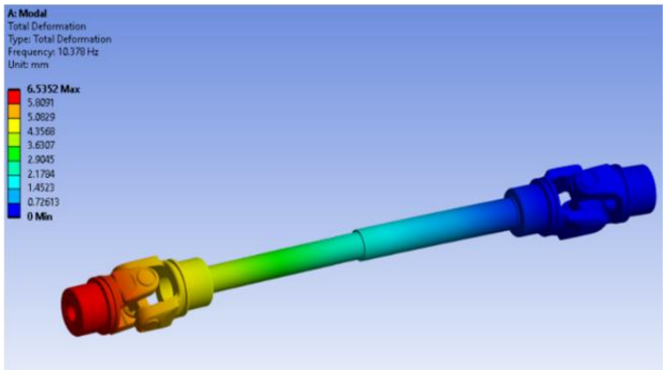
4.1 First case study results

In the first case, structural steel was used, which is a traditional base material for manufacturing drive shafts to ensure its safety and to compare this case with subsequent cases after adding composite materials. Below the figures and tables are the results of the natural frequency (modal) and harmonic response analyses.

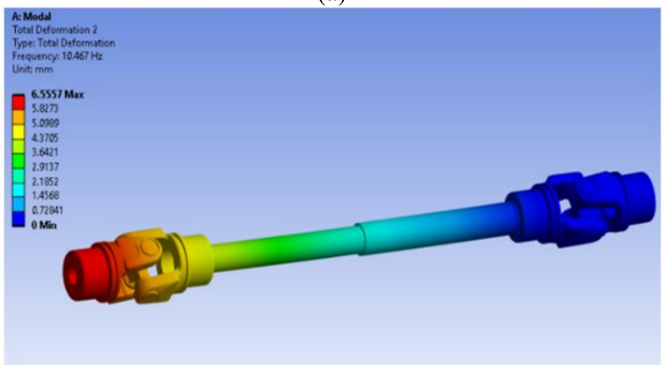
4.1.1 Modal analysis results

Figures 10 shows the modal analysis results for the first 4 modes for the first case study.

Figure 11 shows the modal analysis results for the second 4 modes for the first case study.



(a)



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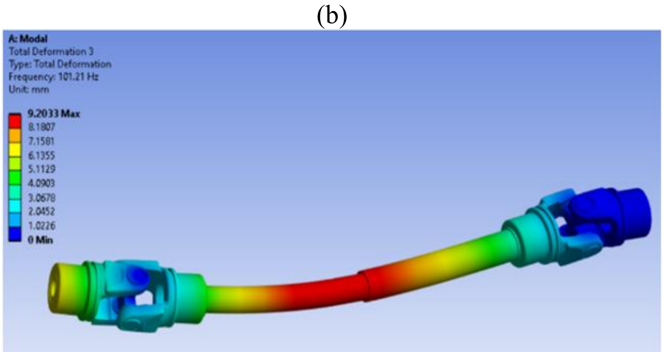
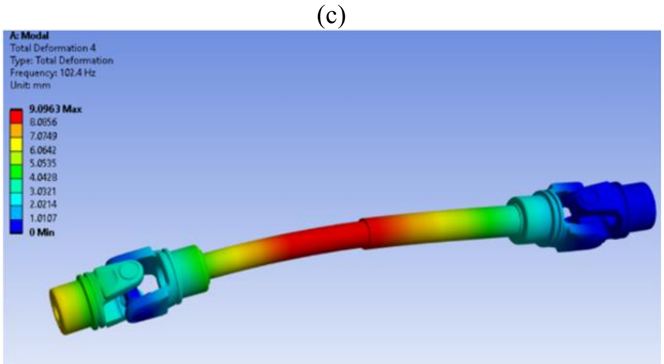
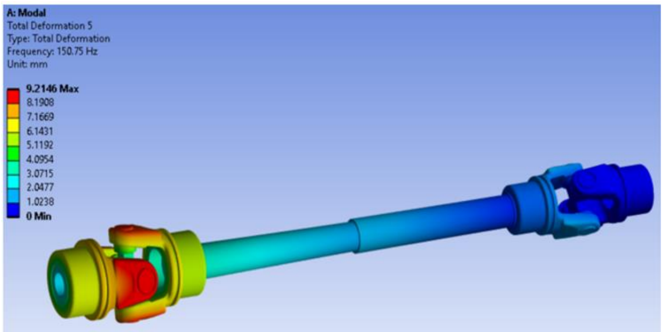


(c)

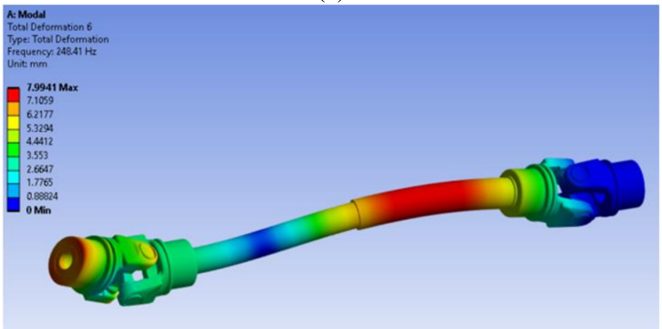


(d)

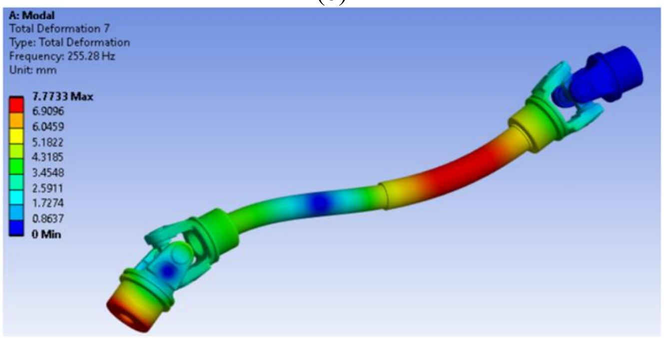
Figure 10. Natural frequency and total deformation results for first case study: (a) at 1st Mode; (b) at 2nd Mode; (c) at 3rd Mode; (d) at 4th Mode



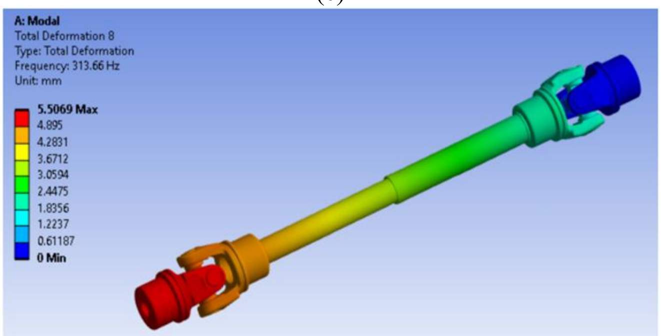
(a)



(b)



(c)



(d)

Figure 11. Natural frequency and total deformation results for first case study: (a) at 5th Mode; (b) at 6th Mode; (c) at 7th Mode; (d) at 8th Mode

Table 4. Natural frequency analysis vs. total deformation for the first case study

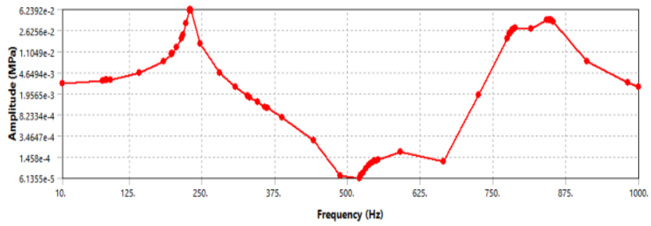
Mode	Frequency [Hz]	Total Deformation (mm)
1	10.378	6.53
2	10.467	6.55
3	101.21	9.20
4	102.40	9.09
5	150.75	9.21
6	248.41	7.99
7	255.28	7.77
8	313.66	5.50

Table 4 shows the modal analysis results for the first case study.

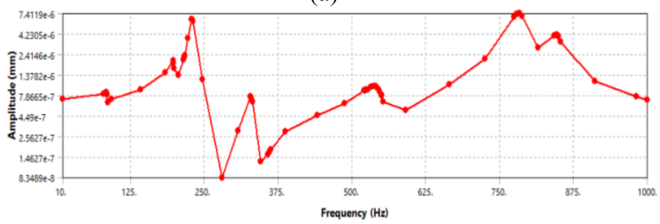
4.1.2 Harmonic response analysis results

The harmonic response results for the first case study without composite materials will be presented here. Figure 12 shows the harmonic response results at different frequencies.

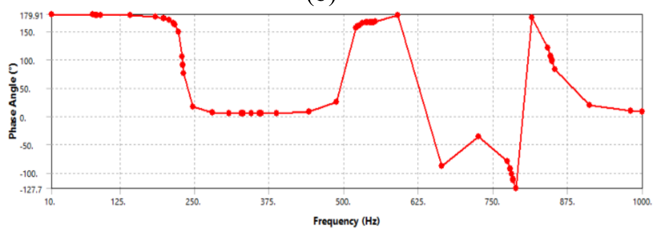
Based on the graphs above and the results from the ANSYS software, the highest stress and deformation values were observed at 229.73Hz. Therefore, the calculations of stress values, deformations, safety factors, and damage factors at this frequency will be conducted as shown in Figure 13.



(a)

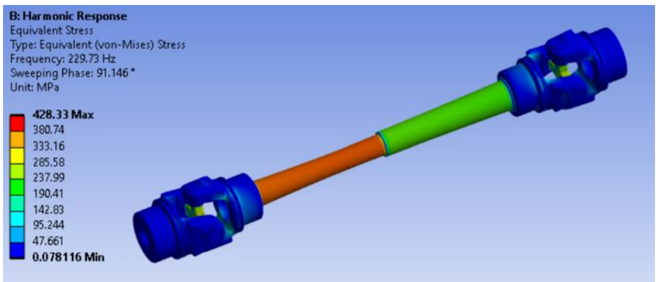


(b)

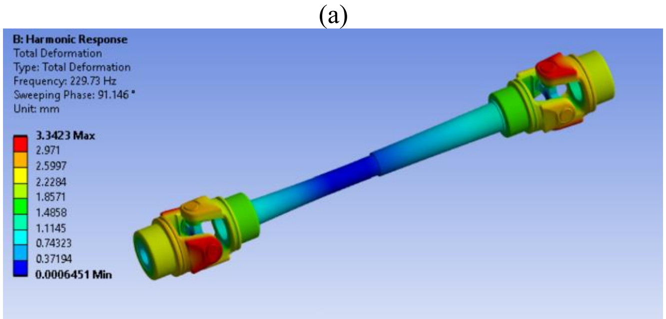


(c)

Figure 12. Harmonic response values for the first case study along different Frequencies: (a) Von mises stress results; (b) Total deformation results; (c) Phase angle



(a)



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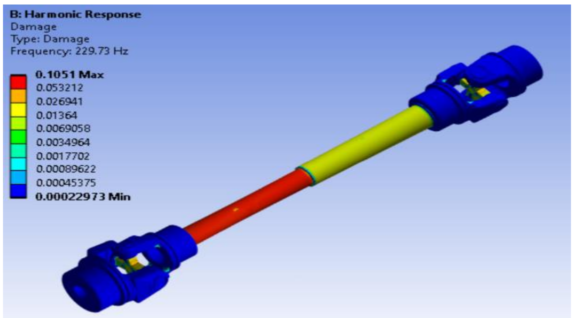

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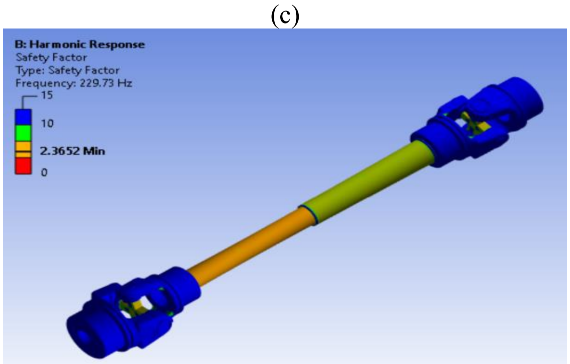


(d)

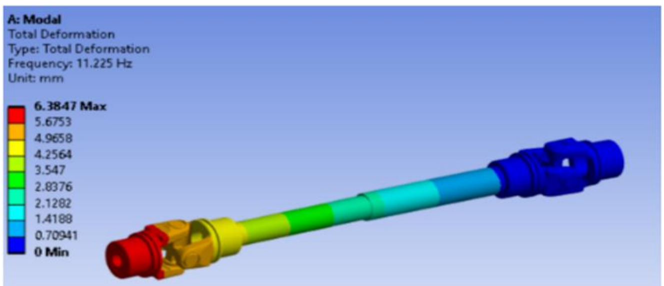
Figure 13. Harmonic response values for the first case study at 229.73Hz: (a) Von mises; (b) Total deformation results; (c) Damage factor; (d) Safety factor

Table 5. Harmonic analysis results at 229.73Hz for first case study

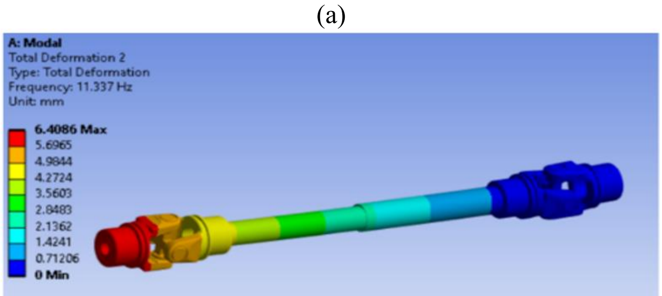
Parameter	Value
Maximum Von-mises stress	428.33 (MPa)
Total deformation	3.34 (mm)
Factor of safety	2.36
Damage factor	0.1051

Table 5 shows the harmonic analysis results at 229.73Hz for the first case study

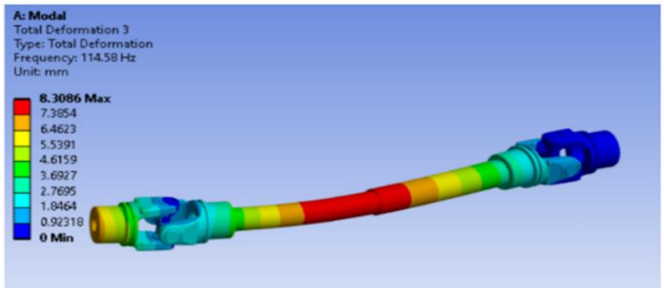
From the results shown above for the first case study, the drive shaft is safe. This is because the safety factor is greater than 1 and the damage factor is less than 1 when the maximum torque is applied to the shaft.



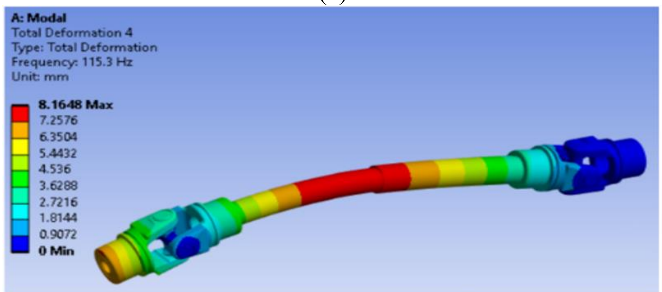
(a)



(b)



(c)



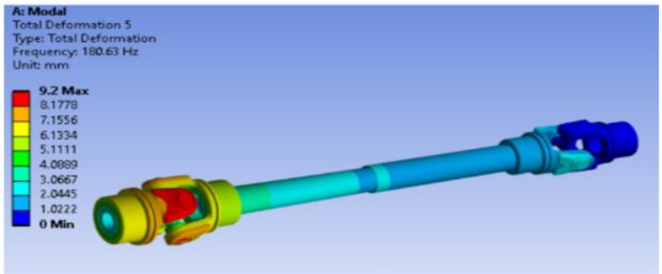
(d)

Figure 14. Natural frequency and total deformation results for second case study: (a) at 1st Mode; (b) at 2nd Mode; (c) at 3rd Mode; (d) at 4th Mode

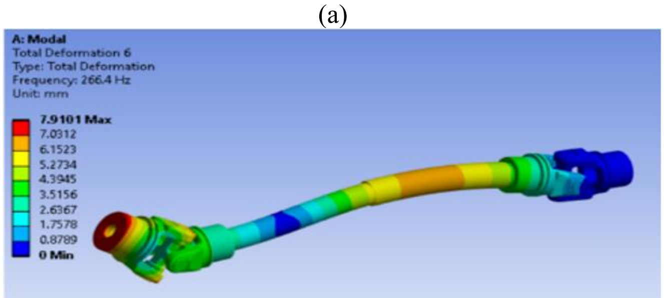
4.2 Second case study results

4.2.1 Modal analysis results

The material used in the second case study is a composite material: Carbon matrix, carbon fibre reinforced (Vf50%), composite. The results of the modal and harmonic response analysis are provided below in the figures and tables.



(a)



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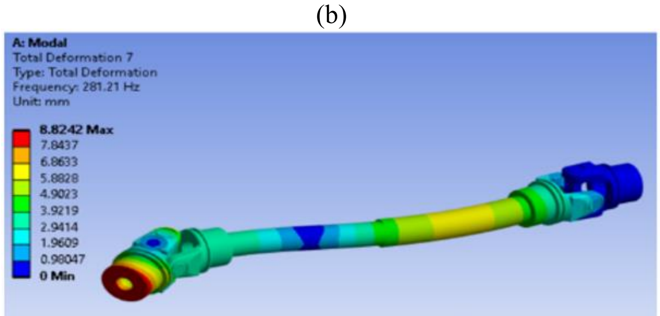
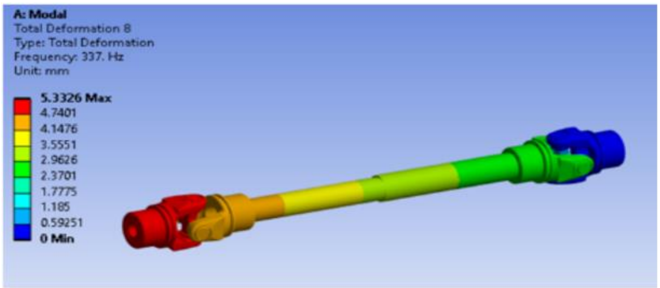


(c)



(d)

Figure 15. Natural frequency and total deformation results for second case study: (a) at 5th Mode; (b) at 6th Mode; (c) at 7th Mode; (d) at 8th Mode

Table 6. Natural frequency analysis vs. total deformation for the second case study

Mode	Frequency [Hz]	Total Deformation (mm)
1	11.22	6.38
2	11.33	6.40
3	114.58	8.30
4	115.3	8.16
5	180.63	9.2
6	266.4	7.91
7	281.21	8.82
8	337	5.33

Figure 14 shows the modal analysis results for the first 4 modes of the second case study.

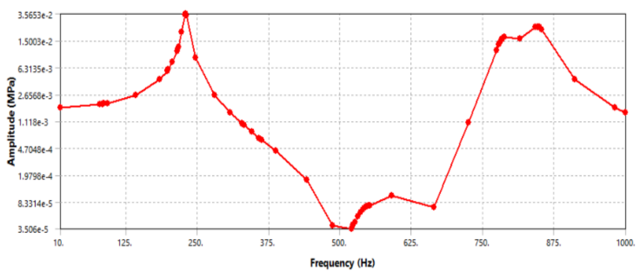
Figure 15 shows the modal analysis results for the second 4 modes for the case study.

Table 6 shows the harmonic analysis results at 229.73Hz for the second case study.

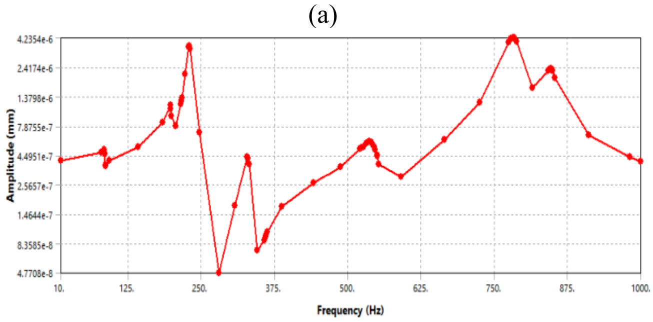
4.2.2 Harmonic response analysis results

The harmonic response results for the second case study with carbon fibre-reinforced (Vf50%) composite materials will be presented here. Figure 16 shows the harmonic response results at different frequencies.

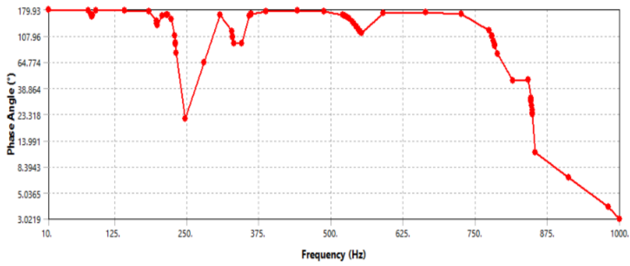
Figure 17 below shows the stress values, deformations, safety factors, and damage factors at this frequency will be conducted at second case study.



(a)

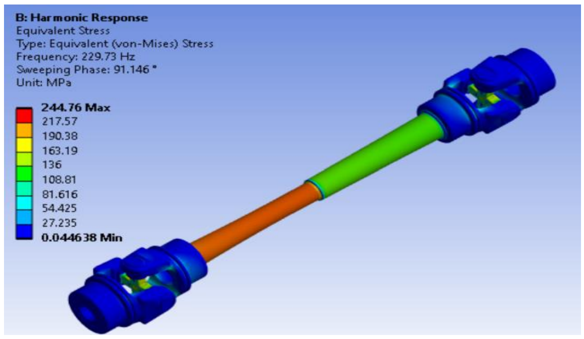


(b)

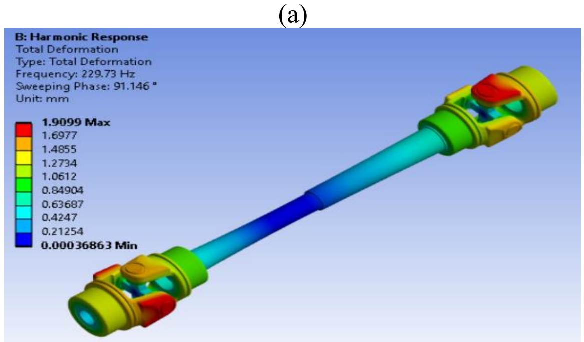


(c)

Figure 16. Harmonic response values for the second case study along different Frequencies: (a) Von mises stress results; (b) Total deformation results; (c) Phase angle



(a)



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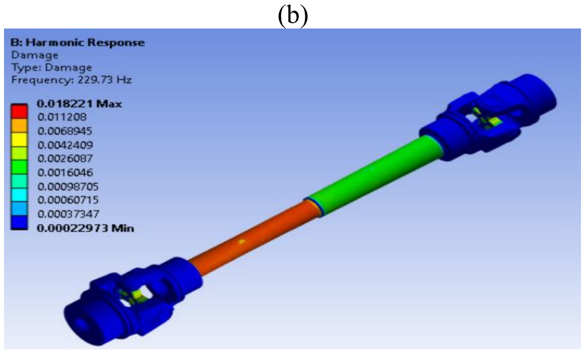

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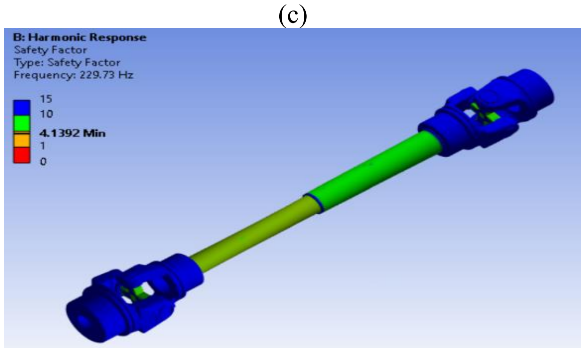


(d)

Figure 17. Harmonic response values for the second case study at 229.73Hz: (a) Von mises; (b) Total deformation Results; (c) Damage factor; (d) Safety factor

Table 7. Harmonic analysis results at 229.73Hz for second case study

Parameter	Value
Maximum Von-mises stress	244.76 (MPa)
Total deformation	1.90 (mm)
Factor of safety	4.13
Damage Factor	0.018

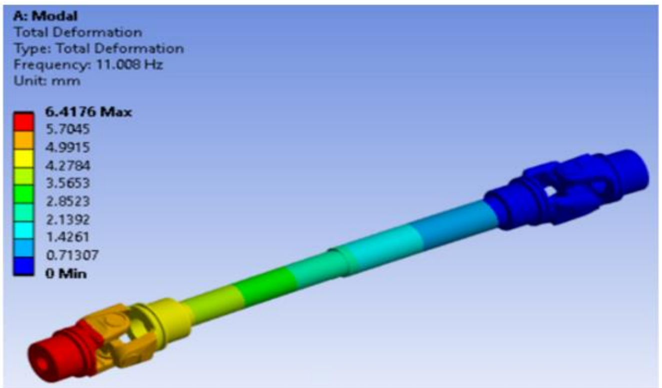
Table 7 shows the harmonic analysis results at 229.73Hz for the second case study.

In the second case, when a composite material (Carbon matrix, carbon fibre reinforced (Vf50%), composite) was added, the safety factor was higher than 1 and the damage factor was less than 1. This indicates that the material is safe and shows an improvement over the first case.

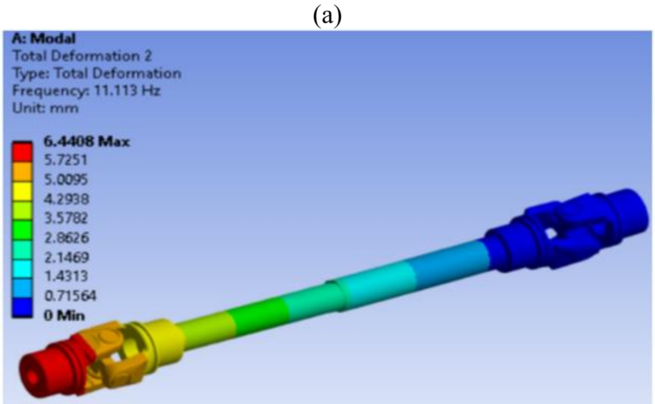
4.3 Third case study results

4.3.1 Modal analysis results

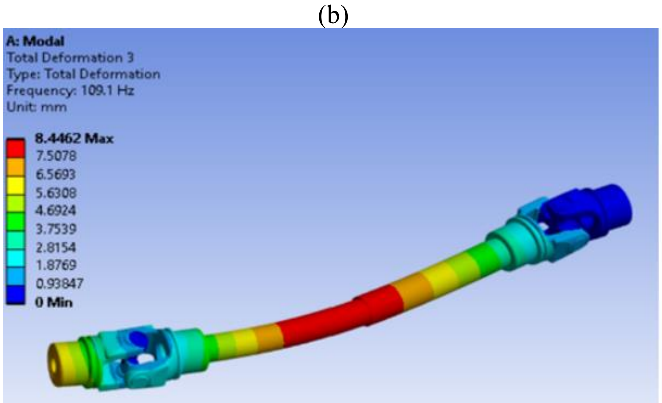
As clarified, the material used in the third case study is: Composite, Epoxy CF, resin-infused fabric, biaxial. The results of its analyses are provided below.



(a)



(b)



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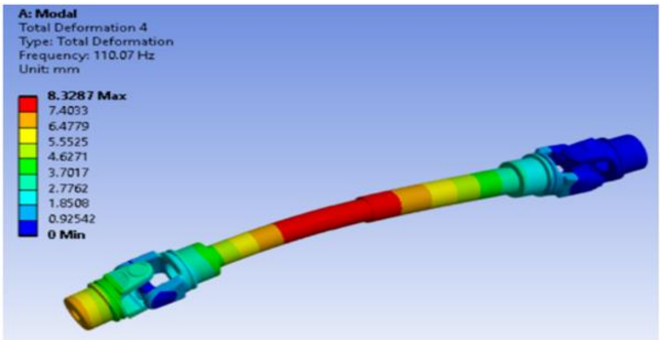


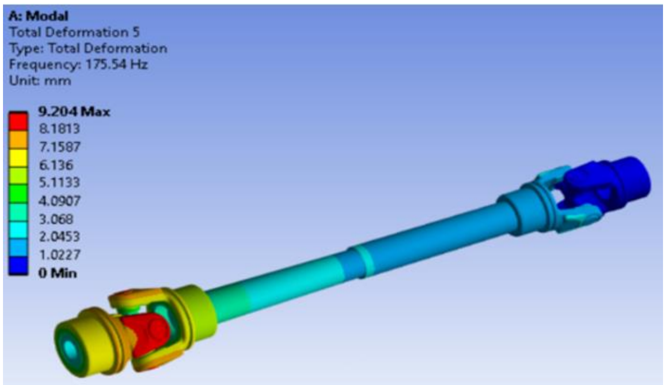
(d)

Figure 18. Natural frequency and total deformation results for third case study: (a) at 1st Mode; (b) at 2nd Mode; (c) at 3rd Mode; (d) at 4th Mode

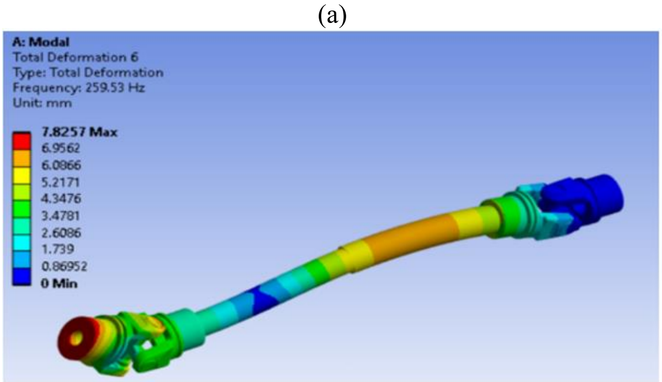
Figure 18 shows the modal analysis results for the first 4 modes of the third case study.

Figure 19 shows the modal analysis results for the second 4 modes of the third case study.

Table 8 shows the third case study's harmonic analysis results at 229.73Hz.



(a)



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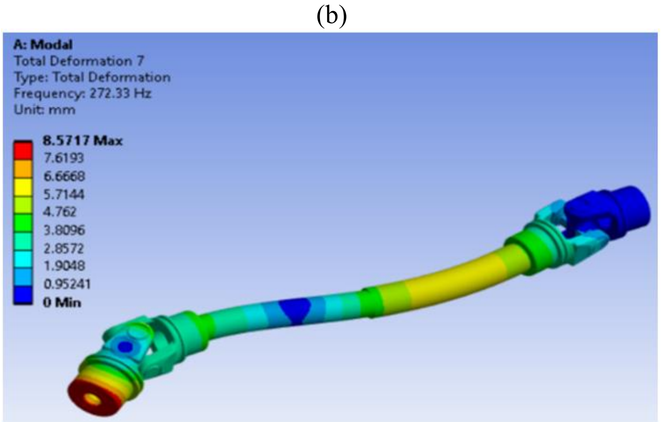
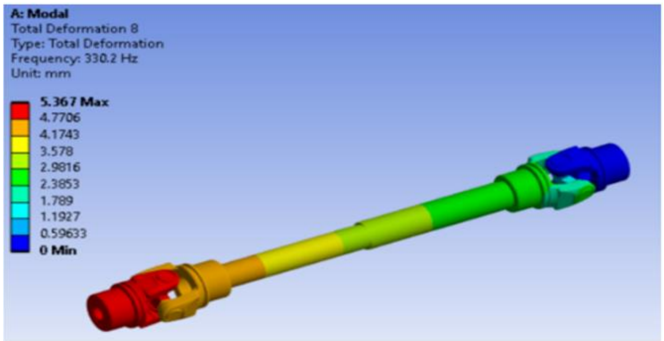


(c)

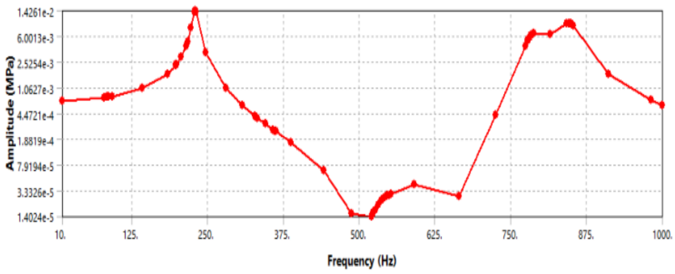


(d)

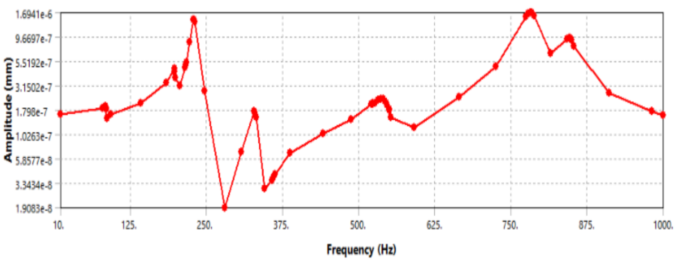
Figure 19. Natural frequency and total deformation results for third case study: (a) at 5th Mode; (b) at 6th Mode; (c) at 7th Mode; (d) at 8th Mode

Table 8. Natural frequency analysis vs. total deformation for the third case study

Mode	Frequency [Hz]	Total Deformation (mm)
1	11.008	6.41
2	11.11	6.44
3	109.1	8.44
4	110.07	8.32
5	175.54	9.20
6	259.53	7.82
7	272.33	8.57
8	330.2	5.36



(a)



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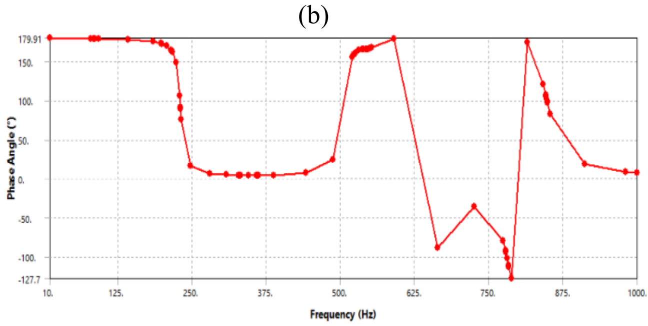
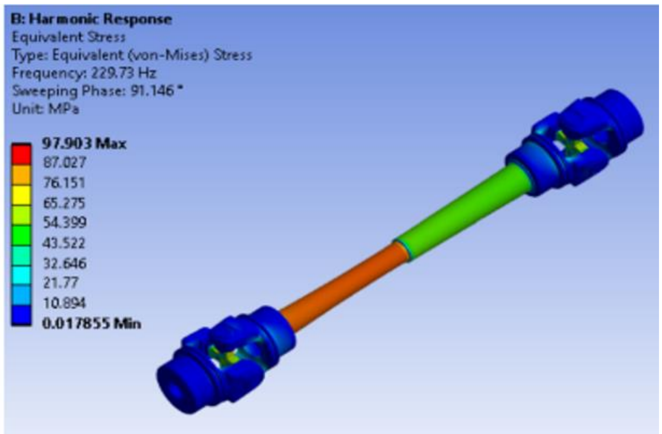
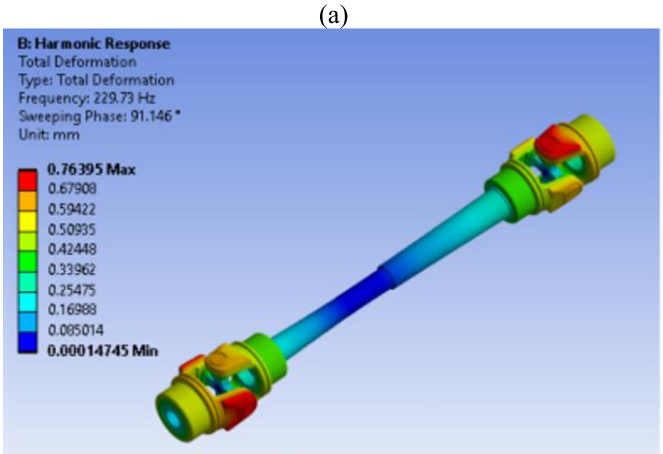


(c)

Figure 20. Harmonic response values for the third case study along different Frequencies: (a) Von mises stress results; (b) Total deformation results; (c) Phase angle



(a)



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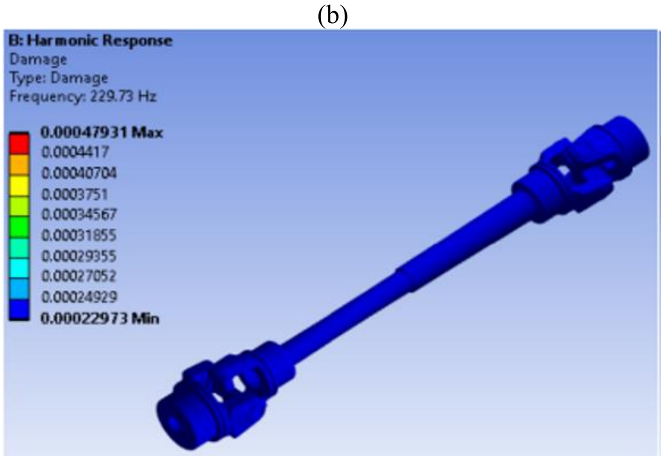

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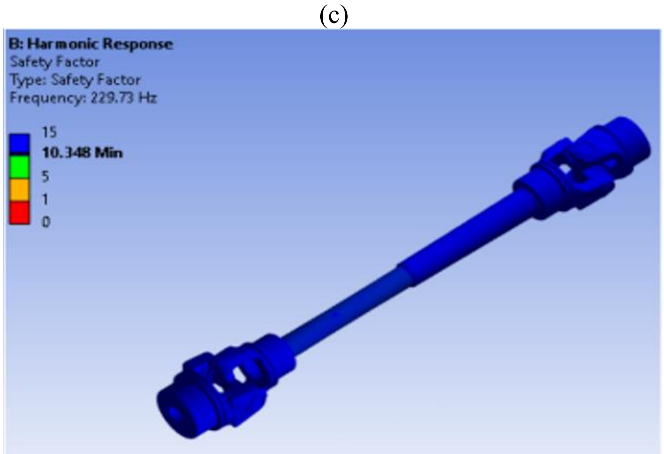

(d)

Figure 21. Harmonic response values for the third case study at 229.73Hz: (a) Von Mises; (b) Total deformation results; (c) Damage factor; (d) Safety factor

4.3.2 Harmonic response analysis results

The harmonic response results for the third case study with Composite, Epoxy CF, resin-infused fabric, biaxial composite materials will be presented here. Figure 20 shows the harmonic response results at different frequencies.

Figure 21 shows the stress values, deformations, safety factors, and damage factors at this frequency, which will be conducted in the third case study.

Table 9 below shows the harmonic analysis results at 229.73 Hz for the third case study.

Table 9. Harmonic analysis results at 229.73Hz for third case study

Parameter	Value
Maximum Von-mises stress	97.9 (MPa)
Total Deformation	0.76 (mm)
Factor of safety	10.348
Damage Factor	0.00047

For the third case, where the composite material (Composite, EpoxyCF, resin infused fabric, biaxial) was used, the drive shaft was also safe. The results, based on the safety factor and the damage factor, were better than in the previous two cases.

To find the enhancement in safety factor the equation below is used.

$$\% \text{ Increase} = \left(\frac{\text{Highest Safety Factor-Initial Safecty Factor}}{\text{Initial Safety Factor}}\right) \times 100 \tag{22}$$

$$\% \text{ Increase} = \left(\frac{10.348\text{-}2.3652}{2.3652}\right) \times 100 \approx \mathbf{338}\% \tag{23}$$

To find the enhancement in damage factor the equation below is used.

$$\% \text{ decrease } \left(\frac{\text{Initial Damage Factor-lowest Damage Factor}}{\text{Initial Safety Factor}}\right) \times 100 \tag{24}$$

$$\% \text{ decrease} = \left(\frac{0.1051\text{-}0.00047931}{0.1051}\right) \times 100 \approx \mathbf{99.6}\% \tag{25}$$

The first case, when the steel served as the base material for the drive shaft, showed that the design encountered a safety factor of 2.36 and a destruction factor of 0.1051. The second and third cases were designed with steel reinforced with composites to enhance the characteristics of the drive shaft. The second case was modifying the steel with composite material (Carbon matrix, carbon fiber reinforced (Vf50%), composite) and the results were far better. The safety factor was 4.13 and the damage factor was 0.018. Results of the third case, when the composite material (Composite, EpoxyCF, resin infused fabric, biaxial) was added introduced to the steel, was the best, with the safety factor increasing to 10.348 and the damage factor being the lowest (0.00047). Modal analysis revealed that steel natural frequencies increased slightly when composite materials were added, which meant that the drive shaft stiffness was improved. The harmonic analysis results turned out to be a significant decrease in the von Mises stress and total deformation values due to the addition of composite materials, the third case mostly. From these findings, it is apparent that the use of the composite materials (Composite, EpoxyCF, resin infused fabric, biaxial) on the steel led to a considerable gain in the state and performance of the drive shaft. The performance of the steel can be improved because of the great mechanical properties of composite materials such as high strength-to-weight ratio and high fatigue resistance, which makes the steel to be more applicable for drive shaft applications in high-speed vehicles. Thus, by amalgamating composites with steel in the construction of the drive shaft, one can achieve better performance and life with the preserved high strength of the steel.

When the findings are compared to previous studies, several key gaps in the field of high-speed drive shaft design are addressed by this research. Harmonic response analysis, which was often overlooked in earlier studies, has been integrated into this work. This analysis provides crucial insights into the dynamic behavior of drive shafts under high-speed conditions. A more comprehensive understanding of how these components perform in real-world scenarios is offered by this approach, moving beyond the limitations of static or purely modal analyses that were common in previous research. The advancement of high-speed drive shaft design is significantly contributed to by demonstrating the synergistic benefits of combining traditional materials with strategically selected composites. Improved stiffness, reduced stress, and enhanced damage resistance were observed in this study, paving the way for more efficient and durable drive shaft designs in high-performance vehicles. Furthermore, a robust framework for future research and development in this field is provided by the methodology of integrating modal and harmonic analyses. This approach could potentially influence industry standards for evaluating and optimizing drive shaft performance in high-speed applications.

5. CONCLUSION

In conclusion, this study demonstrates the significant benefits of integrating composite materials with traditional steel in Double Cardan Driveshafts for high-speed vehicles. Utilizing SOLIDWORKS and ANSYS Workbench, the research reveals that the Composite, EpoxyCF, resin-infused fabric, biaxial material substantially enhances driveshaft performance. Key findings include a 338% increase in safety factor and a 99.6% reduction in damage factor compared to traditional steel designs, along with improved stiffness and reduced stress under high-speed conditions. The novel incorporation of Harmonic Response Analysis provides crucial insights into dynamic loading behavior, enhancing result accuracy and reliability. These improvements underscore the potential of composite materials in creating stronger, more durable driveshafts for high-performance vehicles. Future research should focus on experimental validation, composite layup optimization, vehicle integration, and manufacturing optimization to further advance the design and efficiency of driveline systems for high-speed applications.

FUNDING

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v38p253

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NOMENCLATURE

V	shear force, N
F	axial force, N
T	torque, N.m
M	moment, N.m
I	moment of inertia, kJ
A	area, m^2
u	amplitude

Greek symbols

σ	axial stress, MPa
τ	shear stress, MPa
ϵ	strain, mm/mm