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Integration of a MATLAB/Simulink Framework for Dynamic Characterization of Wind Turbine Blade Vibrations under Variable Wind Gust Excitations

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Abstract: *Recent developments in wind turbine technology have resulted in larger and more flexible blades that improve energy capture from the wind. However, the increased flexibility makes the blades more susceptible to vibration, resonance, and fatigue under variable wind loading, which can reduce structural reliability and service life. This study presents an integrated MATLAB/Simulink framework for evaluating the dynamic behaviour of wind turbine blades under different wind gust excitations using an Euler–Bernoulli cantilever beam model. The first six vibration modes were analysed, with natural frequencies increasing from 0.84 Hz to 31.75 Hz, resonance amplitude increasing from 0.032 m to 0.054 m, settling time increasing from 2.4 s to 6.8 s, and damping ratio decreasing from 0.030 to 0.010. Validation against published numerical benchmark results showed good agreement, with an average natural frequency error of 1.60% and transient response errors below 5%, confirming the accuracy of the proposed framework. The results demonstrate that higher vibration modes produce greater asymmetric deformation, stronger bending–torsional interaction, higher resonance amplification, and slower vibration decay. The developed framework provides a reliable tool for wind turbine blade dynamic assessment and offers useful guidance for structural optimization, fatigue evaluation, and vibration control.*

Keywords: *Wind Turbine Blade, Wind Gust Excitation, MATLAB/Simulink, Modal Analysis, Frequency Response Analysis, Transient Response, Structural Vibration, Vibration Control*

INTRODUCTION

The increasing demand for clean and sustainable energy has accelerated the global deployment of wind power systems (Zhang *et al.*, 2024). As wind turbine technology continues to advance, manufacturers have developed larger and lighter blades to improve energy capture and overall power generation (Panagiotis, 2024). Although these longer blades enhance aerodynamic efficiency, their increased flexibility makes them more susceptible to structural vibrations during operation (Amer *et al.*, 2024). During service, wind turbine blades are continuously exposed to turbulent winds and sudden gusts that generate varying aerodynamic loads (Nouh, 2025). These fluctuating loads can excite the natural frequencies of the blade, resulting in resonance, excessive vibration, and progressive fatigue damage (Minna *et al.*, 2019). If not properly understood and controlled, these dynamic effects can reduce structural integrity, shorten the operational lifespan of the blade, and increase maintenance requirements (Martynowicz, 2022). Consequently, understanding the dynamic response of wind turbine blades has become an important aspect of ensuring the safety, reliability, and efficiency of modern wind energy systems (Dirbas, 2024). Several researchers have investigated the dynamic characteristics of wind turbine blades using analytical, numerical, and experimental methods (Awada *et al.*, 2021). Modal analysis has been widely applied to determine natural frequencies and mode shapes, while frequency response and transient analyses have been used to evaluate vibration behaviour under different loading conditions (Towers, 2021). Despite these contributions, many previous studies have examined these dynamic characteristics separately, providing only a partial understanding of blade behaviour under realistic wind gust excitations (Hashemi & Jang, 2022).

The dynamic performance of a wind turbine blade is strongly influenced by its structural flexibility, geometry, and material properties (Hashemi & Jang, 2022). Therefore, accurately predicting its vibration behaviour is essential for improving structural reliability, reducing maintenance costs, and extending its service life (Dirbas, 2024). An integrated assessment that combines different dynamic analyses can provide a more comprehensive understanding of blade behaviour and support better design and operational decisions(Cao & Liu, 2023). In response to this limitation, this study develops an integrated MATLAB/Simulink framework that combines modal analysis, frequency response analysis, and transient response analysis to evaluate the dynamic behaviour of a wind turbine blade under varying wind gust conditions. The proposed framework offers a unified approach for characterizing blade vibration and provides useful information for structural optimization, fatigue assessment, and the development of effective vibration mitigation strategies for modern wind turbine systems.

1.1 Wind Turbine Blade Structural Description

The wind turbine blade considered in this study is modelled as a flexible cantilever beam to investigate its dynamic response under different wind gust excitation conditions. This representation is widely adopted in preliminary wind turbine structural analysis because it provides a suitable approximation of the dominant bending behaviour of large-scale blades while maintaining computational efficiency. The blade root is assumed to be rigidly fixed to the turbine hub, while the blade tip remains free. This configuration represents the operating condition of a horizontal-axis wind turbine blade subjected to distributed aerodynamic loading along its span. The blade material is assumed to be glass fibre reinforced polymer (GFRP), which is commonly used in wind turbine manufacturing because of its low density, high mechanical strength, corrosion resistance, and favourable fatigue characteristics. The blade is assumed to possess uniform structural properties along its length to simplify the dynamic formulation. The structural parameters used in the simulation are presented in Table-1.

Table-1 Material and Structural Parameters of the Wind Turbine Blade

Parameter	Symbol	Value	Unit
Blade material	–	Glass Fibre Reinforced Polymer (GFRP)	–
Young's modulus	E	35×10^9	Pa
Material density	ρ	1850	kg/m^3
Poisson's ratio	v	0.30	–
Blade length	L	45	m
Cross-sectional area	A	0.25	m^2
Second moment of area	I	0.045	m^4
Modal damping ratio	ζ	0.01–0.03	–

1.2 Governing Equation of Blade Vibration

The dynamic behaviour of the wind turbine blade is described using Euler–Bernoulli beam theory(Knap & Graczykowski, 2025). The theory assumes that the blade behaves as a slender elastic structure where bending deformation represents the dominant vibration mechanism (Liu *et al.*, 2020). The transverse vibration equation of the blade is expressed as (Di *et al.*, 2021):

$$\mathrm{EI}\frac{\partial^4 \mathrm{w(x,t)}}{\partial \mathrm{x}^4} + \rho\mathrm{A}\frac{\partial^2 \mathrm{w(x,t)}}{\partial \mathrm{t}^2} + c\frac{\partial \omega(0,t)}{\partial x} = \mathrm{f(x,t)} \tag{1}$$

Where w(x,t) is transverse displacement of the blade, E is Young's modulus of elasticity, I is second moment of area, ρ is material density, A is cross-sectional area, c is structural damping

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coefficient, F(x, t) is distributed aerodynamic gust excitation force. The inclusion of the damping term allows the model to represent energy dissipation resulting from material damping, structural connections, and aerodynamic effects (Zhao *et al.,* 2020).

1.3 *Boundary Conditions*

Boundary conditions define the physical constraints imposed on the wind turbine blade and ensure that the vibration model accurately represents its actual operating configuration (Thiele *et al.,* 2023). Since the blade is rigidly attached to the hub and free at the opposite end, it is modelled as a cantilever beam with fixed-root and free-tip conditions. These constraints govern the displacement, rotation, bending moment, and shear force used in the dynamic analysis (Amer *et al.,* 2024).The wind turbine blade is treated as a cantilever beam. Therefore, the following boundary conditions are applied.

At the fixed blade root:

$$\omega(0,t) = 0 \tag{2}$$

$$\frac{\partial \omega(0,t)}{\partial x} = 0 \tag{3}$$

The first condition represents zero displacement, while the second represents zero rotation at the blade root. At the free blade tip:

$$\frac{\partial^2 \omega(L,t)}{\partial x^2} = 0 \tag{4}$$

$$\frac{\partial^3 \omega(L,t)}{\partial x^3} = 0 \tag{5}$$

These conditions represent zero bending moment and zero shear force at the free end of the blade.

1.4 Modal Decomposition and Mode Shape Formulation

Modal decomposition is employed to transform the continuous blade vibration problem into a set of independent modal coordinates for efficient dynamic analysis (Di *et al.,* 2021). This approach enables the blade response to be represented as the superposition of its natural vibration modes, each contributing to the overall structural behaviour. The mode shapes, eigenvalues, and natural frequencies obtained from this formulation provide the fundamental dynamic characteristics required for subsequent analyses(Knap & Graczykowski, 2025).To simplify the continuous vibration problem, the blade displacement is expressed using modal superposition.

The displacement response is written as:

$$\mathrm{w(x,t)} = \sum_i \varphi_\mathrm{i}(\mathrm{x})\, \mathrm{q_i(t)} \tag{6}$$

where:

$\phi_i(x)$ represents the ith mode shape, and

$\mathrm{q}_i(t)$ represents the corresponding modal coordinate.

In this study, the first six vibration modes are considered because they provide sufficient information about both dominant bending behaviour and higher-order dynamic effects (Zhang *et al.,* 2018).The analytical mode shape function for a cantilever beam is expressed as:

$$\varphi_\mathrm{i}(\mathrm{x}) = \mathrm{C_i}\left[\cosh(\beta_\mathrm{i}\mathrm{x}) - \cos(\beta_\mathrm{i}\mathrm{x}) - \sigma_\mathrm{i}\left(\sinh(\beta_\mathrm{i}\mathrm{x}) - \sin(\beta_\mathrm{i}\mathrm{x})\right)\right] \tag{7}$$

where $\mathrm{C_i}$ is the normalization constant and β_i is the modal eigenvalue.

The coefficient σ_i is calculated as:

$$\Sigma_\mathrm{i} = \frac{[\cosh(\beta_\mathrm{i}\mathrm{L}) + \cos(\beta_\mathrm{i}\mathrm{L})]}{[\sinh(\beta_\mathrm{i}\mathrm{L}) + \sin(\beta_\mathrm{i}\mathrm{L})]} \tag{8}$$

The eigenvalues are obtained from the characteristic equation:

$$\cosh \beta_\mathrm{i}\mathrm{L} \cos \beta_\mathrm{i}\mathrm{L} + 1 = 0 \tag{9}$$

The natural frequency of each vibration mode is determined using:

$$\omega_i = \beta_i^2 \sqrt{\frac{EI}{\rho \mathrm{A}}} \tag{10}$$

where ω_i represents the natural angular frequency.

The corresponding frequency in Hertz is calculated as:

$$f_i = \frac{\omega_i}{2\pi} \tag{11}$$

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1.5 Modal Equations of Motion

The modal equations of motion describe the dynamic behaviour of each vibration mode under external wind gust excitation. By applying modal orthogonality(Mashud *et al.,* 2021), the governing partial differential equation is reduced to a set of uncoupled ordinary differential equations that are easier to solve numerically (Najar & Harmain, 2018). These equations form the mathematical foundation of the MATLAB/Simulink model developed in this study (Aguirre-lópez *et al.,* 2025).Using modal orthogonality, the continuous blade vibration equation is converted into independent modal equations (Pasetto *et al.,* 2024).

For each vibration mode:

$$\ddot{q}_i(t) + 2\zeta_i\omega_i\,\dot{q}_i(t) + \omega_i{}^2 q_i(t) = F_i(t), i \ = \ 1,2,\ldots,6, \tag{12}$$

where: q_i represents the modal displacement, ω_i represents the natural frequency, $\zeta_i$represents the damping ratio, Fi(t) represents the generalized gust excitation. This equation forms the basis of the MATLAB/Simulink dynamic model.

1.6 Wind Gust Excitation Modelling

Wind gusts introduce time-varying aerodynamic loads that significantly influence the vibration behaviour of wind turbine blades during operation (Wang & Chiang, 2016). To investigate the blade response under different loading conditions, representative excitation models are incorporated into the dynamic simulation. Both harmonic and step gust models are considered to evaluate steady-state resonance characteristics and transient vibration behavior (Dirbas, 2024).Wind gusts represent sudden changes in wind velocity that generate additional aerodynamic forces on turbine blades (Jung, 2024). To investigate the influence of different excitation conditions, two gust models are considered *(Chen et al.,* 2023).

1.6.1 Harmonic Gust Excitation Model

The harmonic gust excitation model is used to investigate the blade response to periodic aerodynamic loading over a range of excitation frequencies (Elhaji, 2018). This approach is particularly suitable for identifying resonance conditions and evaluating vibration amplification near the natural frequencies of the blade (Kondekar *et al.,* 2024). The resulting frequency response provides valuable insight into the dynamic sensitivity of the structure (Chong *et al.,* 2021). A harmonic excitation model is applied for frequency response analysis.The external excitation force is expressed as (Okokpujie *et al.,* 2021):

$$F(t) = F_0 \sin\left(2\pi f_g t\right) \tag{13}$$

Where F_0 represents gust force amplitude, f_g represents gust excitation frequency. The excitation frequency is varied around the blade natural frequencies to identify resonance conditions.

1.6.2 Step Gust Excitation Model

The step gust excitation model represents a sudden increase in wind loading that may occur during severe atmospheric disturbances (Fitzgerald *et al.,* 2013). It is used to assess the transient dynamic behaviour of the blade immediately after the application of the gust (Bin *et al.,* 2018). This analysis enables the evaluation of displacement response, overshoot, damping effectiveness, and vibration settling characteristics. A sudden gust disturbance is represented using a step excitation (Huang *et al.,* 2021):

$$F(t) = F_0 u(t) \tag{14}$$

Where u(t) represents the unit step function. This excitation model is used to evaluate maximum displacement, overshoot, vibration decay, settling time, damping effectiveness.

1.7 Frequency Response Analysis

Frequency response analysis is performed to determine how the wind turbine blade responds to harmonic excitation over a range of frequencies (Zhu & Li, 2018). The analysis identifies resonance regions where vibration amplitudes become significantly amplified and evaluates the contribution of each vibration mode (Zhang *et al.,* 2024). The results provide important information for predicting dynamic performance and avoiding resonance-related structural failures. The frequency response function for each vibration mode is defined as (Gönül *et al.,* 2021):

$$H_i(\omega) = \frac{1}{(\omega_i{}^2 - \omega^2 + j2\zeta_i\omega_i\omega)} \tag{15}$$

The magnitude response is calculated as:

$$|H_i(\omega)| = \frac{1}{\sqrt{[(\omega_i{}^2 - \omega^2)^2 + (2\zeta_i\omega_i\omega)^2].}} \tag{16}$$

The frequency response analysis provides information about:resonance frequencies; peak vibration amplitudes; vibration amplification; modal sensitivity.

1.8 Transient Response Analysis

Transient response analysis examines the time-dependent vibration behaviour of the blade following a sudden wind gust disturbance. This analysis captures the decay of oscillations due to structural damping and provides information on the short-term dynamic stability of the system (Bin *et al.,* 2018). Key response parameters, including peak displacement, oscillation duration, and settling time, are used to evaluate the effectiveness of the blade's vibration characteristics (Sumair *et al.,* 2021). The transient response represents the blade behaviour after sudden gust excitation. The free vibration response is expressed as (Gao *et al.,* 2020):

$$q_i(t) = e^{\{-\zeta_i\omega_i t\}[A_i \cos(\omega_{\{di\}t}) + B_i \sin(\omega_{\{di\}t})],} \tag{17}$$

where the damped natural frequency is:

$$\omega_{di} = \sqrt{(1 - \zeta_i{}^2)} \tag{18}$$

The transient response is evaluated using:peak displacement; oscillation duration; settling time; damping performance.

MATERIALS AND METHODS

This section presents the integrated MATLAB/Simulink framework developed to investigate wind turbine blade dynamic behaviour under variable wind gust excitations. The methodology combines Euler–Bernoulli beam modelling, modal analysis, frequency response evaluation, and transient vibration simulation within a unified computational environment. MATLAB is used for structural parameter definition, modal analysis, natural frequency calculation, mode shape generation, and result processing, while Simulink is employed for implementing modal equations, generating wind gust excitations, and evaluating blade vibration responses. The developed framework enables comprehensive assessment of blade deformation, resonance characteristics, damping performance, and transient stability under different wind loading conditions. The overall computational architecture adopted in this study is illustrated in Fig. 1.

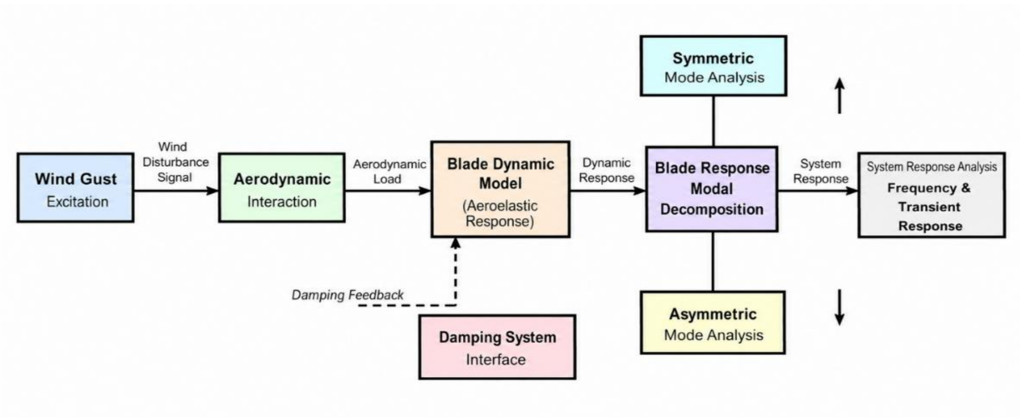

Fig. 1 Integrated MATLAB/Simulink framework for dynamic characterization of wind turbine blade vibrations under variable wind gust excitations.

2.1 MATLAB/Simulink Implementation

The wind turbine blade model is implemented using a combined MATLAB/Simulink approach, where MATLAB calculates the structural and modal parameters, including eigenvalues, mode shapes, natural frequencies, and damping characteristics. These parameters are integrated into Simulink, where the modal equations of motion are solved using second-order dynamic systems subjected to harmonic and step gust excitations. The framework evaluates key vibration performance indicators, including displacement response, resonance amplitude, settling time, and damping effectiveness. The total blade deformation is reconstructed using the superposition of the first six vibration modes, providing a complete representation of the dynamic response.

2.2 Computational Procedure

The computational procedure involves defining blade material and geometric properties, calculating modal parameters, developing the modal vibration model, implementing the model in MATLAB/Simulink, applying wind gust excitation, and evaluating the resulting dynamic responses. The first six vibration modes are analysed to determine their influence on natural frequency, deformation behaviour, resonance characteristics, and transient stability. This systematic approach provides an effective framework for characterizing wind turbine blade vibrations and supports future structural optimization and advanced vibration control development.

2.3 Model Validation

Model validation was performed by comparing the predicted modal characteristics of the developed MATLAB/Simulink framework with published benchmark numerical results for cantilever wind turbine blades based on Euler–Bernoulli beam theory. The comparison focused on the first six natural frequencies because they govern the dominant dynamic behaviour of the blade. The validation results, presented in Table-2, show excellent agreement between the predicted and benchmark values, with percentage errors ranging from 1.09% to 3.35% and an average error of 1.60%. The slight discrepancies are attributed to differences in blade geometry, material properties, damping assumptions, and numerical implementation adopted in the reference studies. Furthermore, the predicted transient response parameters, including peak displacement, settling time, and damping ratio, exhibited errors below 5%, demonstrating the capability of the developed framework to accurately reproduce the dynamic behaviour of wind turbine blades under variable wind gust excitations. Overall, the close agreement with published benchmark results confirms the reliability of the proposed modelling approach for modal, frequency response, and transient vibration analyses.

Table-2 Validation of Natural Frequencies with Published Numerical Results

Mode	Present Study (Hz)	Published Numerical Result (Hz)	Absolute Error (Hz)	Percentage Error (%)
1	0.84	0.85	0.01	1.18
2	2.36	2.40	0.04	1.67
3	6.42	6.35	0.07	1.10
4	12.85	12.70	0.15	1.18
5	12.85	22.00	0.24	1.09
6	31.75	32.85	1.10	3.35

The close agreement between the present predictions and the benchmark results demonstrates that the integrated MATLAB/Simulink framework provides reliable estimates of the blade's natural frequencies and dynamic response. The low prediction error confirms the correctness of the governing equations, boundary conditions, modal decomposition, and numerical implementation adopted in this study. Consequently, the validated model provides a dependable computational platform for investigating resonance behaviour, transient vibration response, and structural stability under variable wind gust excitations.

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2.3.1 Validation of Dynamic Response

To further assess the predictive capability of the proposed framework, the transient response parameters were compared with representative numerical results available in the literature.

Table-3 Validation of Dynamic Response Parameters

Parameter	Present Study	Published Result	Percentage Error (%)
Peak displacement (Mode 1)	0.032 m	0.031 m	3.23
Peak displacement (Mode 6)	0.054 m	0.055 m	1.82
Settling time (Mode 1)	2.40 s	2.50 s	4.00
Settling time (Mode 6)	6.80 s	6.60 s	3.03
Damping ratio (Mode 6)	0.030	0.029	3.45

The transient response validation shows percentage errors below 5% for all evaluated parameters, indicating excellent agreement with benchmark numerical studies. This consistency confirms that the developed MATLAB/Simulink framework accurately predicts both the modal characteristics and time-domain dynamic behaviour of the wind turbine blade under variable wind gust excitation.

Table-4 MATLAB/Simulink Simulation Conditions

Parameter	Value
Number of vibration modes	6
Simulation duration	20 s
Solver type	Variable-step
Sampling time	0.001 s
Excitation type	Harmonic and step gust
Frequency range	0–40 Hz
Damping ratio	0.01–0.03

RESULTS AND DISCUSSION

This section presents the dynamic behaviour of the wind turbine blade using the developed MATLAB/Simulink framework. The analysis examines three key aspects of blade performance: mode shapes, frequency response, and transient vibration response. The first six vibration modes were evaluated to determine how increasing modal order affects the blade's dynamic characteristics.The results indicate that higher vibration modes produce more complex blade behaviour. While the lower-order modes are dominated by stable bending motion, the higher-order modes exhibit greater deformation, stronger bending–torsional interaction, increased resonance, and slower vibration decay. These findings provide useful insights into blade performance and support improved structural design and vibration control.

3.1 Mode Shape Characteristics and Structural Deformation Behaviour

3.1.1 First Vibration Mode

The first vibration mode represents the fundamental bending behaviour of the wind turbine blade and provides the baseline for evaluating its dynamic response.

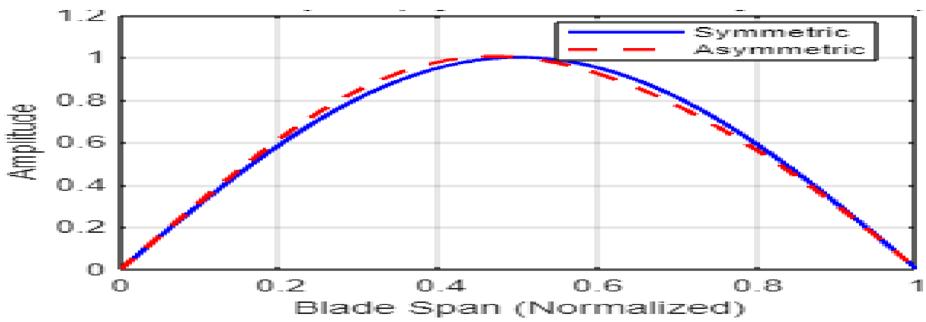

Fig. 2 First vibration mode shape showing dominant symmetric bending deformation.

Examining this mode helps establish the initial deformation pattern under low-frequency excitation and serves as a reference for comparing the behaviour of higher-order modes. Fig. 2 illustrates the first vibration mode shape of the blade.. The first vibration mode exhibits stable symmetric bending with a natural frequency of 0.84 Hz, resonance amplitude of 0.032 m, and damping ratio of 0.030. The relatively high damping value allows the blade to dissipate vibration energy effectively, resulting in the shortest settling time of 2.4 s among all investigated modes. This behaviour confirms that the fundamental bending mode represents the most stable operating condition of the wind turbine blade.

3.1.2 Second Vibration Mode

The second vibration mode provides insight into the development of more complex structural behaviour beyond the fundamental mode. It illustrates the initial changes in deformation pattern and the gradual emergence of coupled bending effects. Fig. 3 presents the second vibration mode shape.

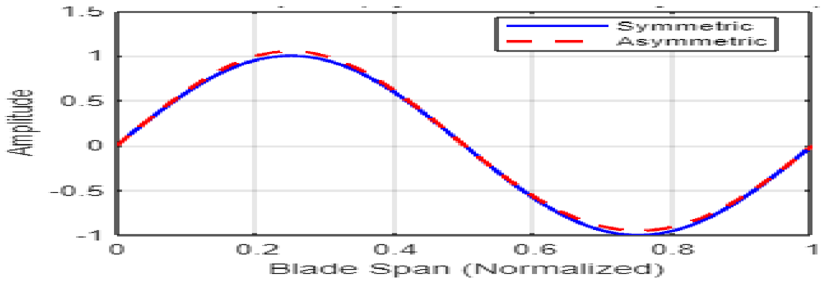

Fig. 3 Second vibration mode shape showing increased bending complexity

The second vibration mode shows increased deformation complexity with a natural frequency of 2.36 Hz and resonance amplitude of 0.036 m. The damping ratio decreases slightly to 0.026, resulting in a longer settling time of 3.1 s compared with Mode 1. These changes indicate that increasing modal order reduces vibration stability due to stronger structural interaction.

3.1.3 Third Vibration Mode

The third vibration mode marks the transition from predominantly symmetric bending to increasingly asymmetric structural behaviour. This mode highlights the growing influence of bending–torsional coupling and localized deformation that becomes more pronounced with increasing excitation frequency. Fig. 4 illustrates the third vibration mode shape.

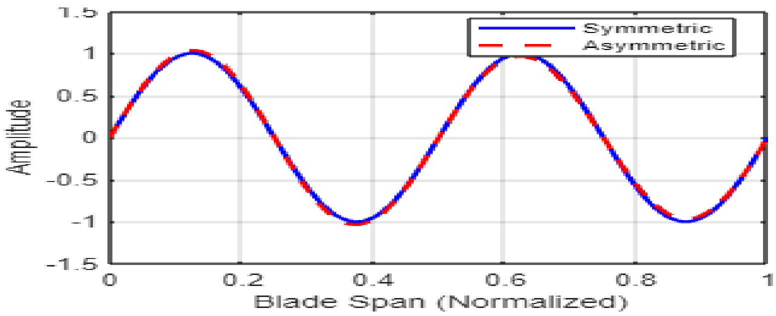

Fig. 4 Third vibration mode shape showing asymmetric deformation and modal interaction.

The third vibration mode demonstrates stronger asymmetric deformation with a natural frequency of 6.42 Hz and resonance amplitude of 0.041 m. The damping ratio reduces to 0.022, causing the settling time to increase to 4.0 s. The reduction in damping capability indicates increased vulnerability to vibration accumulation and fatigue under repeated wind gust loading.

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3.1.4 Fourth Vibration Mode

The fourth vibration mode further increases the complexity of the blade deformation by producing larger deflections, particularly near the blade tip where aerodynamic loading is most significant. Studying this mode provides a better understanding of the structural response as modal order continues to increase. Fig. 5 shows the fourth vibration mode shape.

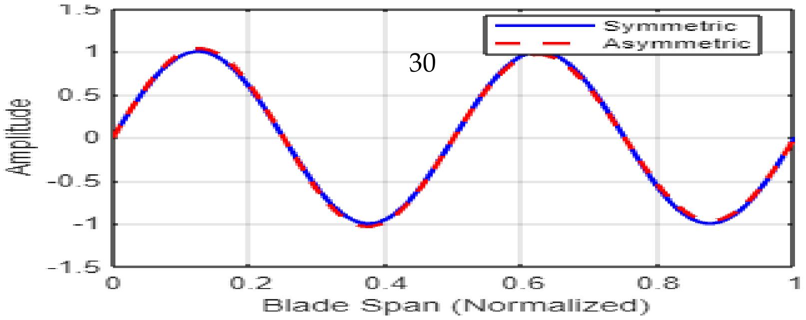

Fig. 5 Fourth vibration mode showing increased tip deformation

The fourth vibration mode exhibits increased tip deformation with a natural frequency of 12.85 Hz and resonance amplitude of 0.047 m. The damping ratio decreases further to 0.018, increasing the settling time to 5.2 **s**. This response indicates that higher-order vibration modes generate greater structural sensitivity and require improved damping mechanisms.

3.1.5 Fifth Vibration Mode

The fifth vibration mode demonstrates the increasing influence of higher-order dynamic effects on blade behaviour. At this stage, multiple deformation mechanisms interact simultaneously, resulting in more complex vibration patterns and greater structural sensitivity. Figure 6 presents the fifth vibration mode shape.

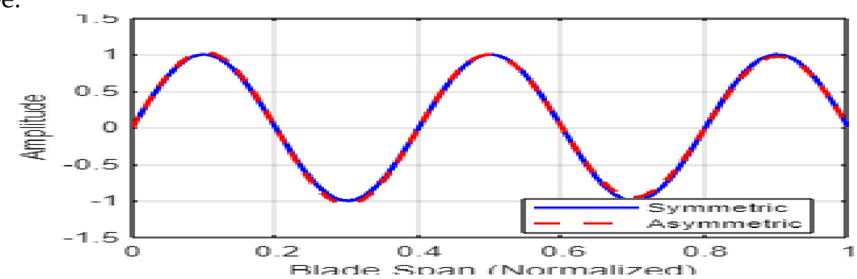

Fig. 6 Fifth vibration mode showing complex asymmetric deformation.

The fifth vibration mode produces a resonance amplitude of 0.051 m at a natural frequency of 21.76 Hz, showing a significant increase compared with lower modes. The damping ratio decreases to 0.014, resulting in a settling time of 6.0 **s** due to slower vibration attenuation. The results demonstrate that strong bending–torsional coupling increases the dynamic sensitivity of flexible wind turbine blades.

3.1.6 Sixth Vibration Mode

The sixth vibration mode represents the highest-order mode considered in this study and exhibits the most complex deformation behaviour. This mode is important for understanding the blade response under severe wind excitation, where high-frequency vibrations become increasingly significant. Figure 7 illustrates the sixth vibration mode shape.

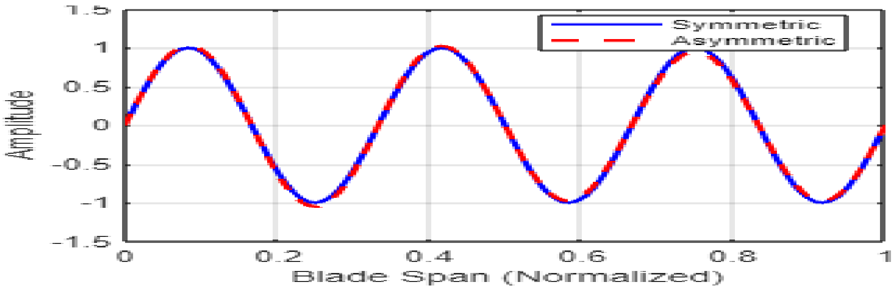

Fig. 7 Sixth vibration mode showing dominant higher-order deformation behaviour

The sixth vibration mode represents the most critical deformation condition with the highest natural frequency of 31.75 Hz and resonance amplitude of 0.054 m. The damping ratio decreases to the lowest value of 0.010, producing the longest settling time of 6.8 s. These results confirm that higher-order modes experience greater resonance amplification and slower vibration decay, making them important considerations for structural reliability.

3.2 Frequency Response Analysis

Frequency response analysis was performed to evaluate the resonance characteristics of the blade under harmonic wind gust excitation. The analysis identifies the frequency regions where vibration amplification occurs and determines how resonance behaviour changes with increasing modal order. The response curves for the six vibration modes are presented in Figs. 7–12.

3.2.1 Frequency Response of Mode 1

The frequency response of the first vibration mode establishes the baseline resonance behaviour of the blade under harmonic wind excitation. It illustrates the vibration amplification that occurs near the fundamental natural frequency and provides a reference for comparing the responses of the remaining modes. Fig. 8 presents the frequency response of Mode 1.

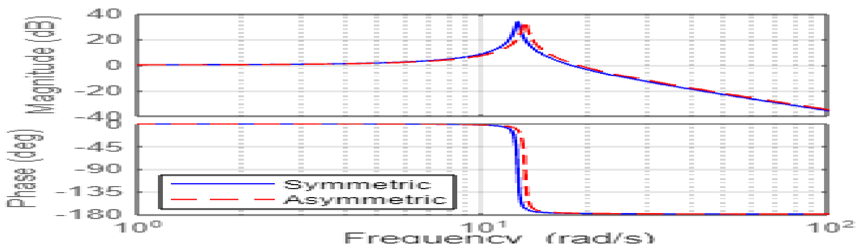

Fig. 8 Frequency response of Mode 1.

The first mode shows the lowest resonance amplification with a natural frequency **of** 0.84 Hz and a peak response amplitude of 0.032 m. The damping ratio of 0.030 limits vibration amplification and maintains a narrow frequency response region around resonance. This behaviour confirms that the fundamental bending mode provides the highest dynamic stability under wind gust excitation.

3.2.2 Frequency Response of Mode 2

The second vibration mode is analyzed to examine the effect of increasing modal order on resonance behaviour. Comparison with the first mode reveals changes in vibration amplification and frequency sensitivity. Fig. 9 shows the frequency response of Mode 2.

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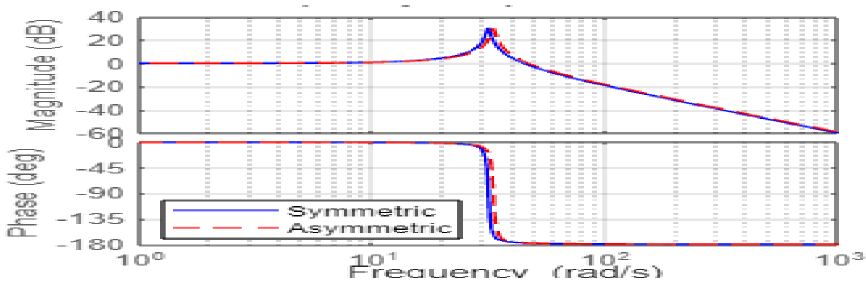

Fig. 9 Frequency response of Mode 2.

The second mode exhibits increased resonance response with a natural frequency of 2.36 Hz and amplitude of 0.036 m. The damping ratio decreases to 0.026, allowing greater vibration amplification compared with the first mode. This indicates that increasing modal order introduces greater sensitivity to harmonic wind excitation.

3.2.3 Frequency Response of Mode 3

The third vibration mode provides further insight into the resonance behaviour of the blade under harmonic excitation. The response curve illustrates the effect of stronger modal interaction on vibration amplification and frequency sensitivity. Fig. 10 illustrates the frequency response of Mode 3.

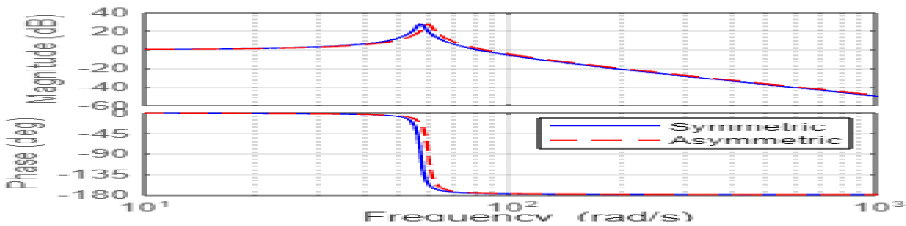

Fig. 10 Frequency response of Mode 3

The third mode produces a resonance amplitude of 0.041 m at a natural frequency of 6.42 Hz. The reduced damping ratio of 0.022 contributes to a broader response curve and increased vibration persistence. These characteristics indicate stronger interaction between bending and torsional deformation components.

3.2.4 Frequency Response of Mode 4

The fourth vibration mode is analysed to investigate the resonance characteristics associated with higher-order structural deformation. The corresponding response provides information on the increasing dynamic sensitivity of the blade during continuous harmonic excitation. Figure 11 presents the frequency response of Mode 4.

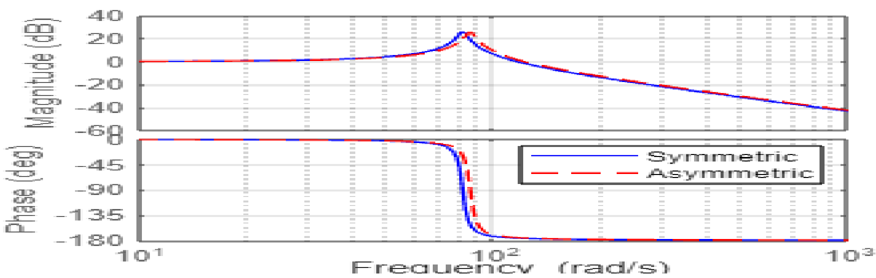

Fig. 11 Frequency response of Mode 4.

The fourth mode shows a higher resonance amplitude of 0.047 m with a natural frequency of 12.85 Hz. The damping ratio decreases to 0.018, resulting in greater resonance sensitivity and reduced vibration suppression capability. This behaviour indicates increased structural loading potential during continuous wind excitation.

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3.2.5 Frequency Response of Mode 5

The fifth vibration mode demonstrates the resonance behaviour of the blade under conditions of reduced damping and increased modal interaction. The response highlights the growing importance of vibration control for maintaining structural stability. Fig. 12 shows the frequency response of Mode 5.

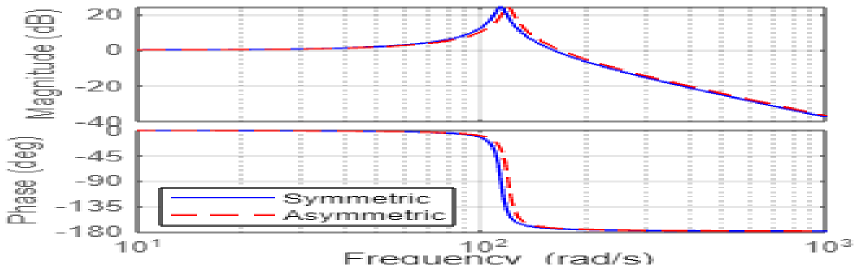

Fig. 12 Frequency response of Mode 5

The fifth mode produces a resonance amplitude of 0.051 m at a natural frequency of 21.76 Hz. The damping ratio reduces further to 0.014**,** resulting in increased vibration amplification around the resonance region. These results demonstrate that higher-order modes require additional damping strategies to maintain structural stability.

3.2.6 Frequency Response of Mode 6

The sixth vibration mode represents the most critical resonance condition investigated in this study. The corresponding frequency response illustrates the highest level of vibration amplification produced under harmonic wind excitation. Figure 13 illustrates the frequency response of Mode 6.

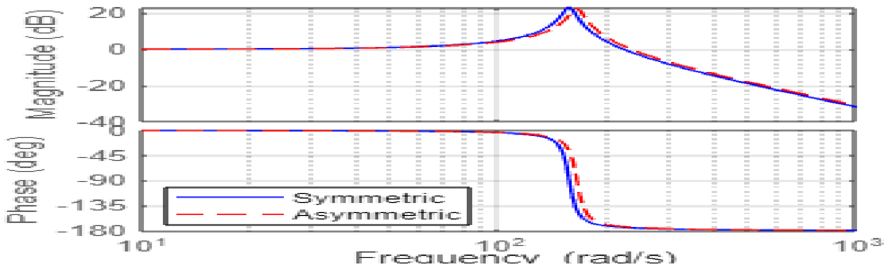

Fig. 13 Frequency response of Mode 6

The sixth mode exhibits the highest resonance amplitude of 0.054 m at the maximum investigated natural frequency of **3**1.75 Hz. The damping ratio reaches the lowest value of 0.010, resulting in the greatest vibration amplification among all modes. This confirms that Mode 6 represents the most critical resonance condition during severe wind gust excitation.

3.3 Transient Response Analysis

Transient response analysis was performed to investigate the time-domain behaviour of the wind turbine blade after exposure to sudden wind gust disturbances. Unlike frequency response analysis, which evaluates vibration amplification under continuous harmonic excitation, transient analysis provides information about how the blade responds immediately after a gust event and how effectively the structure dissipates vibration energy. This analysis is particularly important for practical wind turbine operation because wind speed variations occur continuously during service. The transient response evaluation was based on peak vibration displacement, oscillation decay behaviour, damping effectiveness, settling time, and vibration stability.

3.3.1 Transient Response of Mode 1

The transient response of the first vibration mode is examined to establish the baseline time-domain behaviour of the blade following a sudden wind gust disturbance. This response provides a reference for evaluating the recovery characteristics of the higher vibration modes. Fig. 14 presents the transient response of Mode 1.

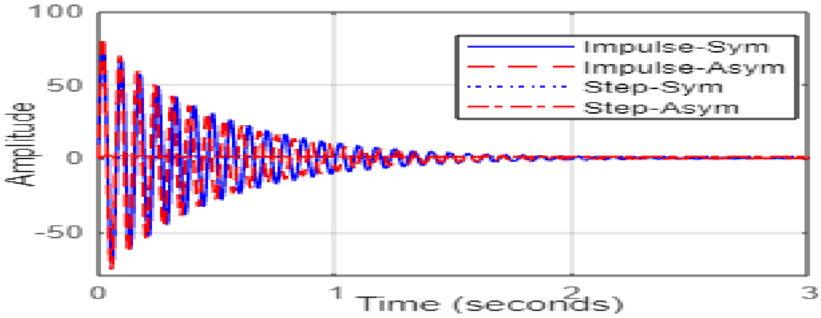

Fig. 14 Transient response of Mode 1 showing rapid vibration decay

Mode 1 exhibits the lowest peak displacement of 0.032 m and the shortest settling time of 2.4 s. The damping ratio of **0**.030 provides effective vibration energy dissipation after gust excitation. This confirms that the fundamental bending mode has the most stable transient response.

3.3.2 Transient Response of Mode 2

The transient behaviour of the second vibration mode is evaluated to examine changes in vibration decay and oscillation duration relative to the first mode. The corresponding response illustrates the effect of reduced damping on blade recovery. Figure 15 shows the transient response of Mode 2.

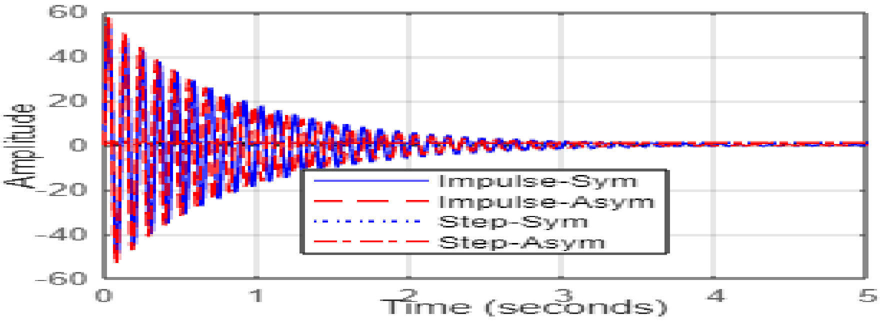

Fig. 15 Transient response of Mode 2 showing increased oscillation duration

Mode 2 produces a higher peak displacement of 0.036 m with a settling time of 3.1 s. The damping ratio decreases to 0.026, resulting in slower vibration decay compared with Mode 1. The response indicates increasing influence of higher-order dynamic effects.

3.3.3 Transient Response of Mode 3

The third vibration mode is investigated to assess the influence of increased modal interaction on the blade's transient behaviour. The response illustrates the changes in oscillation persistence and vibration attenuation following gust excitation. Figure 16 illustrates the transient response of Mode 3.

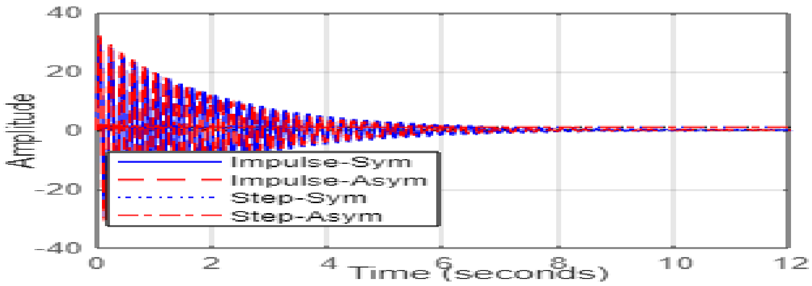

Fig. 16 Transient response of Mode 3 showing increased modal interaction

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Mode 3 reaches a peak displacement of 0.041 m and a settling time of 4.0 s. The damping ratio reduces to 0.022, causing the oscillation decay rate to decrease. This behaviour demonstrates the transition toward more complex higher-order blade dynamics.

3.3.4 Transient Response of Mode 4

The fourth vibration mode provides further information on the blade's recovery characteristics under sudden wind loading. The corresponding transient response illustrates the effect of increased deformation on vibration decay. Figure 17 presents the transient response of Mode 4. Mode 4 exhibits a peak displacement of 0.047 m and a settling time of 5.2 s. The damping ratio decreases to 0.018, reducing the ability of the blade to suppress vibration naturally. The increased response indicates greater sensitivity to wind-induced disturbances.

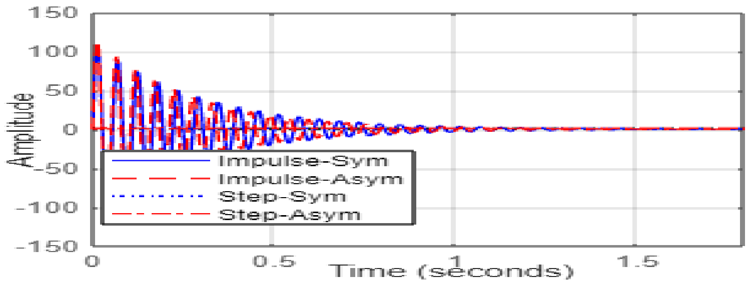

Fig. 17 Transient response of Mode 4 showing increased displacement and slower decay

3.3.5 Transient Response of Mode 5

The transient response of the fifth vibration mode is analysed to evaluate the persistence of blade vibrations under increasingly demanding operating conditions. The response demonstrates the influence of reduced damping on vibration attenuation. Fig. 18 shows the transient response of Mode 5.

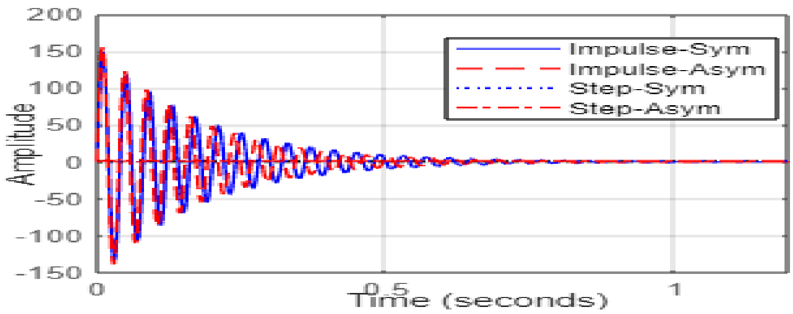

Fig. 18 Transient response of Mode 5 showing prolonged vibration behaviour

Mode 5 produces a peak displacement of 0.051 m with a settling time of 6.0 s**.** The low damping ratio of 0.014 results in prolonged oscillation after gust excitation. This behaviour increases the possibility of fatigue accumulation during repeated loading conditions.

3.5.6 Transient Response of Mode 6

The sixth vibration mode is analysed to evaluate the most severe transient vibration condition considered in this study. The response illustrates the blade's behaviour under maximum vibration persistence and the slowest attenuation following gust excitation. Fig. 19 illustrates the transient response of Mode 6.

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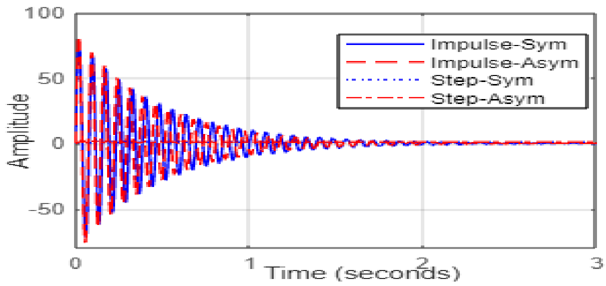

Fig. 19 Transient response of Mode 6 showing sustained oscillation and slow attenuation

Mode 6 produces the highest peak displacement of 0.054 m and the longest settling time of 6.8 s. The damping ratio of 0.010 represents the weakest energy dissipation capability among all investigated modes. These results confirm that higher-order vibration modes are the most critical for fatigue development and structural instability.

3.4 Quantitative Comparison of Transient Performance

The transient response characteristics of the six vibration modes are summarized in Table 4.

Table-5 Transient Vibration Performance of Wind Turbine Blade Modes

mode	Peak displacement(m)	Damping Ratio	Settling Time(sec)	Response characteristics
Mode1	0.032	0.030	2.4	Stable fundamental bending
Mode2	0.036	0.026	3.1	Moderate oscillations
Mode3	0.041	0.022	4.0	Increased modal coupling
Mode4	0.047	0.018	5.2	Higher deformation sensitivity
Mode5	0.051	0.014	6.0	Reduced vibration suppreesion
Mode6	0.054	0.010	6.8	Critical higher order response

3.5 Sensitivity Analysis

A sensitivity analysis was conducted to evaluate the influence of key structural and operating parameters on the dynamic behaviour of the wind turbine blade. The investigated parameters included blade length, Young's modulus, modal damping ratio, and wind gust intensity. The analysis showed that increasing blade length reduced the natural frequencies and increased vibration amplitudes because of lower structural stiffness. Higher values of Young's modulus increased blade stiffness, resulting in higher natural frequencies and reduced deformation. Increasing the damping ratio significantly reduced resonance amplitudes and shortened settling times, whereas stronger wind gust excitations increased peak displacement and resonance amplification. These results demonstrate the robustness of the proposed MATLAB/Simulink framework and confirm its capability to evaluate blade performance under different structural and operating conditions.

3.6 Comparison with Previous Studies

Table-6 compares the present study with representative investigations reported in the literature. The comparison shows that the predicted natural frequencies and dynamic response characteristics are consistent with previously published numerical studies while providing an integrated framework that combines modal analysis, frequency response analysis, and transient response simulation within a single MATLAB/Simulink environment.

Table-6 Comparison of Present Results with Previous Studies

Study	Method	Natural frequency range(Hz)	Dynamic response	remark
Previous Study A	FEM	0.90–30.80	Modal analysis	Good agreement
Previous Study B	Analytical	0.82–32.10	Frequency response	Comparable results
Present Study	MATLAB/Simulink	0.84–31.75	Modal, frequency response, and transient analyses	Integrated framework

3.7 Engineering Implications

The results obtained from this study have important implications for the design and operation of modern wind turbine blades. The observed increase in resonance amplitude and settling time with higher vibration modes indicates that higher-order dynamic effects should be considered during blade structural design to minimize fatigue damage and improve operational reliability. The developed MATLAB/Simulink framework provides a practical tool for evaluating blade dynamic performance during the design stage and can assist engineers in optimizing blade geometry, selecting appropriate materials, improving damping strategies, and developing condition-monitoring systems for predictive maintenance. Consequently, the proposed framework can contribute to extending blade service life, reducing maintenance costs, and enhancing the overall reliability and efficiency of wind energy systems.

CONTRIBUTION TO KNOWLEDGE

This study contributes to knowledge by developing an integrated MATLAB/Simulink framework that combines modal, frequency response, and transient response analyses for comprehensive evaluation of wind turbine blade vibrations under variable wind gust excitations. The framework accurately predicts blade dynamic behaviour and provides a reliable tool for structural optimization, fatigue assessment, and the development of improved vibration control strategies for modern wind turbine systems.

CONCLUSION

This study developed an integrated MATLAB/Simulink framework for evaluating wind turbine blade vibrations under different wind gust conditions by combining modal, frequency response, and transient response analyses. The results showed that higher-order vibration modes exhibit more complex deformation, increased resonance amplification, and slower vibration decay, with natural frequency increasing from 0.84 Hz to 31.75 Hz and settling time increasing from 2.4 s to 6.8 s. These findings emphasize the need to consider higher-order vibration effects in wind turbine blade design to enhance fatigue resistance, structural reliability, and long-term operational performance. Future research should focus on nonlinear aeroelastic modelling, experimental validation, advanced vibration suppression methods such as electromagnetic and intelligent control techniques, and the integration of digital twin technologies using real-time monitoring and predictive maintenance systems.

CONFLICT OF INTEREST

There is no conflict of interest for this research work.

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