INTERNATIONAL JOURNAL OF
COASTAL, OFFSHORE & ENVIRONMENTAL ENGINEERING

IJCOE Vol.10/No. 4/ Autumn 2025 (8-18)

Available online at: www. Ijcoe.org

Vibration Control of a Composite Panel Carrying Concentrated Masses Under Different Boundary Conditions

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ARTICLE INFO

Article History:
Received : 21 Mar 2025
Accepted : 25 Dec 2025

Keywords:
Composite panel
Vibration behavior
Concentrated mass effect
Updated FE model
Vibration control

ABSTRACT

Composite panels are commonly used structural forms in marine environments, subjected to complex dynamic loads and vibration forces due to harsh operational conditions. Understanding their vibrational behavior is crucial for ensuring structural reliability and performance. The presence of localized mass variations, such as concentrated masses from attached equipment or marine growth, significantly influences their dynamic response. This study investigates the impact of concentrated masses on the vibrational characteristics of a composite panel under fixed and pinned boundary conditions through comprehensive experimental and numerical analyses. A finite element (FE) model was developed and updated based on experimental modal analysis results. The findings reveal that the location and magnitude of concentrated masses play a critical role in altering mode shapes and reducing natural frequencies. The introduction of concentrated masses leads to a systematic frequency shift, with greater reductions observed for higher mass magnitudes. The validated FE model enables an extended parametric analysis, reducing the dependency on extensive experimental testing. These insights provide a foundation for optimizing vibration control strategies in composite marine structures, contributing to improved design methodologies for enhanced structural integrity and performance.

1. Introduction

Composite materials are engineered from two or more distinct constituents, each contributing unique properties to the final product. The resulting composite structure typically exhibits enhanced physical and mechanical characteristics compared to the individual components [1]. Composite materials have attracted significant interest in recent years and are now commonly considered a viable alternative to traditional materials like metals and wood. They have been vastly adopted in the marine industry over time [2,3]. The increasing utilization of composites can largely be attributed to the numerous advantages composites offer, such as high specific strength and stiffness, enhanced damage tolerance, as well as superior resistance to fatigue and corrosion. These advantages make them ideal for watercraft, submersibles, offshore structures, and other components used in marine construction [4,5].

One of the most common structural forms for composites is the panel type [6]. Composite panels serve as essential structural components across various engineering applications particularly in marine settings [7,8]. Depending on their intended use and taking into account their operation in harsh marine and offshore conditions, composite panels utilized in these applications might experience various dynamic and vibration forces [9]. Intense vibration can significantly impact the safety and reliability of these thin-walled structures. Therefore, evaluating the dynamic and vibrational properties of marine and offshore composite panels and assessing their behavior under vibration loads is very important and has become challenging for engineers [10].

In many engineering structures, particularly those of a panel-type design, local mass variations frequently appear as concentrated mass connections [11]. Concentrated masses may develop in marine and offshore structures because of the weight of detachable equipment utilized and, in certain situations, due to the accumulation of marine growth [12]. These concentrated masses can significantly impact the structural behavior and vibration response under different loading conditions [13]. Mass is a critical factor influencing the dynamic characteristics and vibration behavior of the structures [14]. Alterations in the local mass distribution at specific locations within a structure can result in significant shifts in its natural frequencies. Consequently, variations in mass can substantially alter the behavior of the structure under vibrational loads, causing changes in the modal parameters and the overall vibration response [15]. This could lead to unexpected deformations that differ from those predicted during the design phase [16]. Furthermore, in the context of composite structures, which are often engineered to achieve specific performance metrics, the effects of local mass variations can be even more pronounced [17]. These materials are typically designed to optimize strength-to-weight ratios, stiffness, and other mechanical properties. Therefore, understanding how local mass variations affect vibration behavior is critical for ensuring that their performance aligns with design expectations and that they meet safety and functional requirements [18]. Chaubey et al. [19] conducted a study on the free vibration analysis of various types of laminated composite shells with concentrated mass by creating a finite element (FE) formulation that utilizes third-order shear deformation theory (TSDT). Yang and Oyadiji [20] studied delamination in composite beams, caused by concentrated mass loading which affects the variations in modal frequency. Eken et al. [21] formulated the dynamic stability of thin-walled composite beams in rotation with rigid bodies. Hossain et al. [22] investigated the vibrations of composite plates with concentrated mass, examining how varying amounts of concentrated mass and their locations affected the eigenfrequencies of the plates. Zhong et al. [23] studied the stability of rectangular composite plates subjected to an arbitrary mass, specifically analyzing how concentrated mass and its spatial distribution influenced the plate's frequency response. Mandal and Haldar [24] assessed the free vibrations of composite plates and panels featuring concentrated mass at the center, considering various boundary conditions. Vatin et al. [25] investigated the vibrations of a composite plate with concentrated masses in a nonlinear geometric context, exploring the effects of concentrated mass values, their locations, property variations of the composite materials, and other parameters on the frequency response of vibrations.

This study investigates the influence of concentrated masses on the dynamic behavior and vibrational response of a composite panel under fixed and pinned boundary conditions. The primary objective is to examine how the location and magnitude of these masses affect mode shapes and natural frequencies, providing critical insights into vibration control strategies for composite structures. To achieve this, an integrated experimental and numerical approach was adopted. Experimental modal analysis was conducted to capture the vibrational characteristics of the panel under various mass configurations, while a FE model was developed and updated based on these results. The updated FE model was then employed to extend the parametric analysis, enabling a detailed assessment of frequency shifts and mode shape variations across different mass scenarios. Furthermore, the sensitivity of the panel's dynamic response to support conditions was analyzed, revealing distinct trends in frequency reduction and vibrational behavior alterations. The findings demonstrate that the positioning and magnitude of concentrated masses significantly impact the structural dynamics, leading to notable shifts in natural frequencies and mode deformation patterns. These insights contribute to the development of more effective vibration control methodologies, ultimately enhancing the structural performance and reliability of composite marine components.

2. Methodology

2.1. Modal Analysis and Experimental Tests

Modal analysis is a highly effective method for assessing and identifying the dynamic characteristics of a structure, such as natural frequencies, mode shapes, and damping ratios, and includes both theoretical and experimental techniques [26]. In recent years, with the trend towards designing structures with less weight and higher strength, modal analysis has gained a special place in vibration analysis and studying the dynamic behavior of various structures [27]. The main objective of modal analysis is to obtain more accurate dynamic characteristics of a structure using analytical, numerical, and experimental or combined methods. In this study, experimental and numerical modal analysis was used to investigate the effect of added concentrated masses on the vibration behavior and dynamic characteristics of a composite panel structure [28].

In order to perform experimental modal analysis, a physical model of the composite panel was fabricated using hand layup and vacuum bagging mold process. The materials selected and the method of fabrication were determined according to DNV guidelines and specifications for building composite structures intended for marine and offshore applications [29]. In the fabricated composite panel, woven glass fibers were used for the reinforcement phase and a mixture of polyester resin and hardener with a weight ratio of 70 and 30 for the matrix phase. To construct the model under study, 10 layers of glass fibers were used with the stacking sequence of $[0,90]_s$ to achieve a final thickness of 2 mm. The final dimensions of the composite panel after cutting were $750mm \times 800mm \times 2mm$.

There are several methods available for conducting modal testing, with the shaker test being the most widely used. In this approach, a shaker excites the structure using a specific signal, and the response of the structure is captured as time histories using sensors such as accelerometers [30]. The frequency response functions (FRFs) are then derived by applying the fast Fourier transform to the recorded time history responses, allowing for the calculation of the structure's natural frequencies [31]. Since the modal parameters of the structure are influenced by mechanical characteristics like mass and stiffness, any alterations to these characteristics will lead to corresponding changes in the modal parameters [32,33].

To perform the experimental modal analysis, the physical model of the composite panel was excited by a shaker under both fixed and pinned support conditions. The excitation force applied to the model was a sinusoidal signal with a zero mean. Acceleration time history data were recorded at several points of the model using accelerometers. These data were transferred to the Pulse software for processing. The experimental modal analysis was conducted under two different scenarios to comprehensively assess the effects of concentrated masses on structural behavior. The first scenario involved the analysis without any added concentrated masses, allowing for a baseline understanding of the system's modal characteristics. The second scenario included alterations to the local mass by adding concentrated masses to evaluate their influence on the dynamic response of the structure. Furthermore, the analysis incorporated both fixed and pinned support conditions, which enabled a thorough investigation of how different boundary conditions affect the modal response. The vibrational response was measured at multiple points throughout the composite panel to establish a detailed representation of its dynamic performance under various test conditions. The concentrated masses were applied in the center of the panel with two weights of 100g and 200g. Figure 1 shows the overview and method of performing the experimental modal analysis, as well as the location of the excitation point, accelerometers, and added concentrated masses.

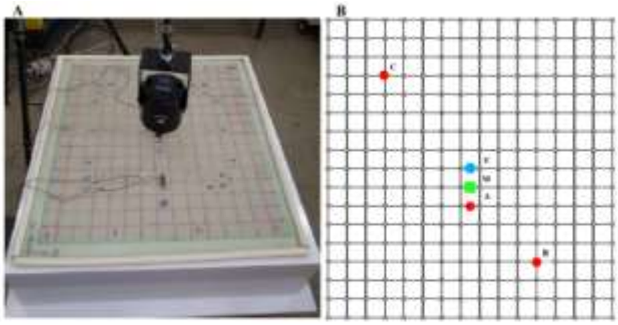


Figure 1. Experimental modal analysis: A) overview of the experimental test; B) location of excitation point, accelerometers, and added concentrated masses

The data obtained from the experimental modal analysis and pulse software processing for the composite panel were recorded as time history responses and FRFs. The required data were measured at three points A, B and C in the state of no added concentrated masses and with excitation at point E on the composite panel model. Then, by applying 100g and 200g weights at point M on the composite panel as concentrated masses, the response of the structure was also collected.

2.2. Finite Element Model of the Composite Panel

An FE model of the composite panel was developed using ABAQUS software, with similar geometry to the physical model, to ensure valid comparisons between simulated and experimental data (Figure 2). The model was designed with layered configurations to simulate the fiber orientations and stacking sequence. Pinned and fixed boundary conditions were applied to the model. A meshing strategy was employed that resulted in 6,375 quadrilateral elements, which were chosen to provide a balance between computational efficiency and the accuracy of the results.

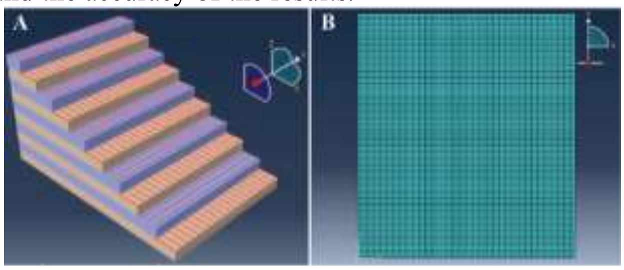


Figure 2. FE modeling of the composite panel via ABAQUS: A) Fibers orientation; B) Meshing

Material properties of the composite panel were obtained based on characteristics of the constituent material used in fabrication which are presented in Table 1. To calculate the material properties of the composite panel the rule of mixtures equations were used.

Table 1. Mechanical properties of the composite panel's constituent material

		Reinforcement phase		Matrix phase	
Material properties	**Unit**	**Symbol**	**Value**	**Symbol**	**value**
Elastic modulus	GPa	E_f	30.1	E_m	3.5
Poisson ratio	-	υ_f	0.2	υ_m	0.35
Shear modulus	GPa	G_f	10	G_m	2
Volume fraction	%	V_f	70	V_m	30

According to the rule of mixtures, the elastic modulus, shear modulus, and Poisson ratio can be calculated using these equations:

$$E_c = E_f V_f + E_m(1 - V_f) \tag{1}$$

$$G_c = \frac{G_f G_m}{G_m V_f + G_f V_m} \tag{2}$$

$$\upsilon_c = V_f \upsilon_f + V_m \upsilon_m \tag{3}$$

where E_c, G_c, and υ_c are the elastic modulus, shear modulus, and Poisson ratio of the composite, respectively. The equations used for predicting the mechanical properties of composites are well-documented and established, allowing for accurate estimation based on the properties of the constituent materials [34]. This predictive capability is valuable for the design and optimization of composite materials in practical applications [35]. By employing these equations, it would be possible to assess the performance of composites without the need for extensive experimental testing, saving time and resources in the development phase. The estimated mechanical properties of the composite panel are outlined in Table 2. This information was used in the modeling of the composite panel.

Table 2. Mechanical properties of the composite panel

Material properties	Unit	Symbol	Value
Density	Kg/m^3	ρ	1500
Longitudinal Elastic Modulus	GPa	E_1	22.4
Transverse Elastic Modulus	GPa	E_2	22.4
Poisson ratio	-	υ_{12}	0.23
Shear Modulus in the 1-2 Plane	GPa	G_{12}	4.7
Shear Modulus in the 2-3 Plane	GPa	G_{23}	2.6
Shear Modulus in the 1-3 Plane	GPa	G_{13}	3.6

2.3. Numerical Modal Analysis

The numerical modal analysis process involves developing and refining a comprehensive mathematical model that effectively represents the structural system. This is achieved through the application of FE analysis, which enables the examination of the system's equations of motion under different boundary conditions and loadings [36]. For the model under investigation, numerical modal analysis was performed using ABAQUS software.

The primary goal of this analysis was to validate the FE model by calculating numerical frequencies and subsequently comparing these values with corresponding experimental results obtained through physical testing. This comparison is essential as it serves to highlight differences between the experimental and the predicted results. To enhance the accuracy of the FE model, modifications and updates were made based on the findings from this comparison, ensuring that the numerical model reflects a more realistic representation of the structural behavior. Given that conducting experimental modal analysis is costly and very time-consuming, particularly when dealing with different locations and magnitudes of concentrated masses applied to the model, the updated FE model was utilized to overcome these challenges. This approach not only simplified the analysis process but also enabled a comprehensive evaluation of the composite panel's behavior across a range of conditions, without the need for extensive physical testing.

3. Results and Discussion

Following the modal analysis, the impact of sensor positioning and support conditions on the quality and accuracy of the measured data was assessed using coherence functions. Figure 3 presents the coherence functions for the acceleration time history responses of the composite panel, evaluated under fixed and pinned support conditions at three locations: A, B, and C.

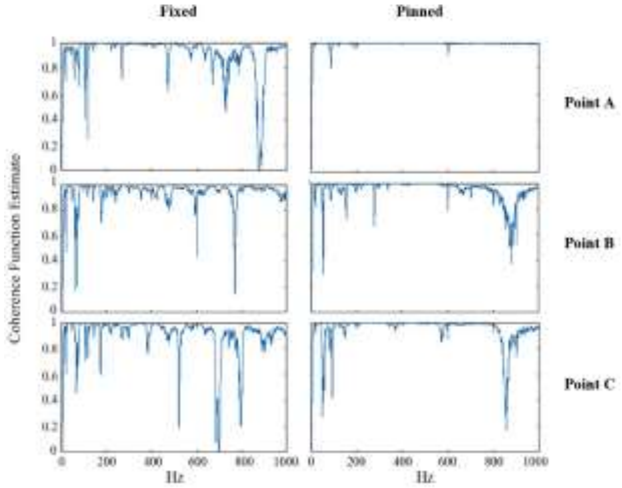


Figure 3. Coherence functions of acceleration time history responses of the composite panel

As illustrated in Figure 3, the coherence function tends to decay more significantly as the sensors are positioned farther from the excitation point and closer to the supports. This trend is attributed to the reduction in vibration amplitude at locations near the supports. Since the coherence function indicates the relationship between the response signal and the input excitation force signal, a decrease in amplitude and excitation strength results in the coherence value shifting from one to zero [26]. This phenomenon is distinctly evident in the presented coherence functions. Moreover, the decay in coherence is more pronounced under fixed support conditions, as the fixed support imposes greater restrictions on the amplitude of the excitation signal compared to the pinned support.

Based on the coherence functions, it can be concluded that positioning the sensor near the excitation point yields a more accurate and higher-quality response. Therefore, in the modal analysis of the composite panel with added concentrated masses, to achieve a more precise evaluation, the accelerometers were placed at point B, closer to the excitation point, to ensure the collection of higher-quality data. Figure 4 shows the coherence functions of the acceleration time history responses for the composite panel carrying concentrated masses at the center of the panel.

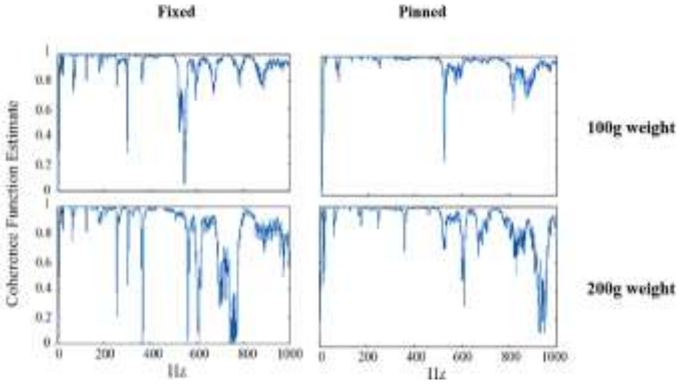


Figure 4. Coherence functions of acceleration time history responses of the composite panel carrying concentrated masses

According to Figure 4, it is evident that added masses result in coherence functions trending closer to zero. This decline indicates a reduction in the vibrational coherence of the system, suggesting that the added masses disrupt the energy distribution across the composite panel. Moreover, as the magnitude of these concentrated masses increases, the decay of the coherence functions becomes more pronounced, highlighting a more significant impact on vibrational behavior. To assess the impact of concentrated masses on the vibration behavior of the composite panel, the natural frequencies both with and without concentrated masses were calculated. Additionally, the analysis considered two different support conditions (fixed and pinned) to provide a more thorough understanding of how the support conditions affect the panel's vibration behavior. The results are summarized in Table 3.

Table 3. Natural frequencies resulted from experimental modal analysis (Hz)

Support condition	Mode	No mass	100g mass	200g mass
Fixed	1	35.5	33.42	28.66
	2	65	61.95	57.38
	3	73	70.46	66.94
	4	104	100.1	95.42
Pinned	1	21.3	18.23	16.87
	2	48.47	46.68	46.04
	3	55.56	54.36	53.68
	4	82.45	81.89	79.94

The natural frequencies of the numerical model were also calculated using the initial FE model without applying concentrated masses. The results of the numerical modal analysis were compared with the experimental findings to validate and assess the accuracy. The results of numerical modal analysis are presented in Table 4.

Table 4. Natural frequencies resulted from numerical modal analysis using initial FE model (Hz)

Support condition	Mode	No mass	100g mass	200g mass
Fixed	1	20.65	17.89	15.89
	2	38.74	38.51	38.33
	3	45.93	45.54	45.20
	4	59.72	59.65	57.39
Pinned	1	10.31	9.37	8.63
	2	24.89	24.12	23.94
	3	29.82	29.41	28.56
	4	41.24	41.23	41.14

As observed in Tables 3 and 4, there are notable differences between the experimental findings and the results obtained from numerical simulations. These differences arise primarily from the simplifications and assumptions made during the numerical modeling process, which may not fully capture the complexities of the actual behavior of the composite panel. To address these differences and enhance the accuracy of the analyses, the FE model of the composite panel was updated. This updated model incorporates more realistic parameters and refined assumptions to better align the simulation results with the experimental data. The calculated natural frequencies, as derived from this updated model, are summarized in Table 5, providing a clearer understanding of the dynamic characteristics of the composite panel. This updated model allows for improved predictive capabilities and a more reliable representation of the panel's behavior under various conditions.

Table 5. Natural frequencies resulted from numerical modal analysis using updated FE model (Hz)

Support condition	Mode	No mass	100g mass	200g mass
Fixed	1	35.35	28.74	24.77
	2	66.38	65.08	64.72
	3	78.98	76.34	75.64
	4	110	100.1	93.31
Pinned	1	20.13	17.17	15.15
	2	47.77	47.51	47.31
	3	56.53	56.10	55.73
	4	80.53	80.06	73.21

A thorough comparison of the frequencies obtained from the experimental modal analysis with those calculated using the updated numerical model demonstrates only a minimal discrepancy. This finding highlights the accuracy and reliability of the updated model. The enhanced model not only confirms the validity of the numerical results but also allows for a comprehensive evaluation of the composite panel's performance under a range of conditions. Importantly, this capability reduces the need for conducting numerous physical tests, enabling efficient analysis and insights into the behavior of the composite material across different scenarios and conditions.

In order to thoroughly evaluate the behavior of the composite panel carrying a concentrated mass and to achieve an insight into the vibration control of this type of structure, its frequency responses were calculated under different conditions. The updated FE model was implemented for this purpose. In this regard, concentrated masses with different magnitudes in various locations on the composite panel were applied. The modal analysis was performed under fixed and pinned support conditions to obtain the natural frequencies and extract the mode shapes at each scenario. The positioning of the applied concentrated masses is illustrated in Figure 5. The arrangement of mass locations was selected in such a way as to enable the assessment of their influence both in proximity to the center of the plate and near the supports.

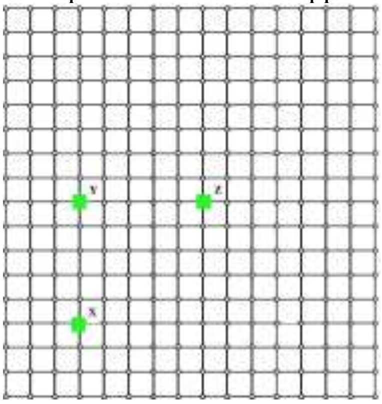
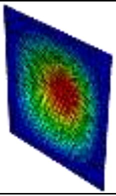
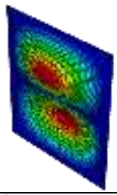
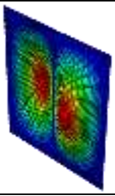
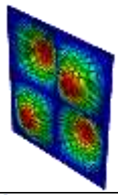
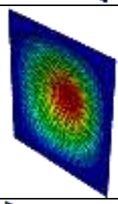
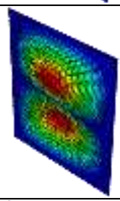
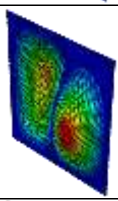
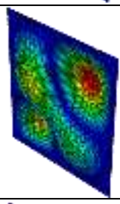
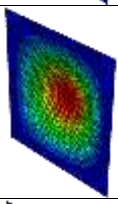
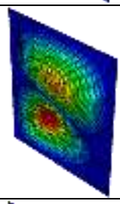
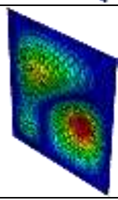
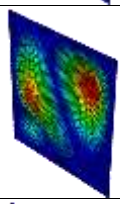
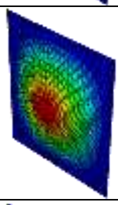
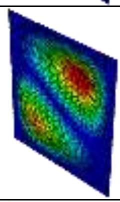
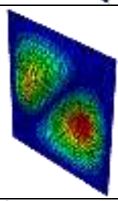
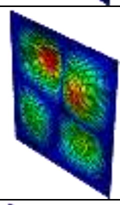
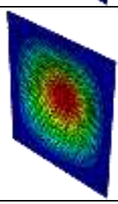
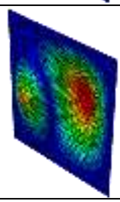
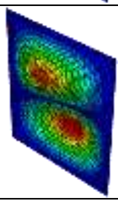
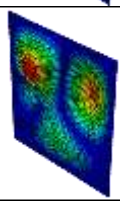


Figure 5. Schematic illustration of the positioning of concentrated masses in the updated FE model

Concentrated masses with different magnitudes were applied to the numerical model using the inertia tool in ABAQUS by defining point mass on the specified locations of the composite panel model. Subsequent to the selection of the required set, the ABAQUS solver was used to perform the numerical modal analysis with various conditions of concentrated masses locations and magnitudes. The frequency responses and mode shapes were derived for each condition. Mode shapes of the composite panel carrying concentrated masses are presented in Table 6 and Table 7 under fixed and pinned support conditions, respectively.

Table 6. Mode shapes and natural frequencies of the model carrying concentrated mass under fixed support condition

Concentrated mass condition	**Mode 1**		**Mode 2**		**Mode 3**		**Mode 4**	
	Mode shape	Frequency (Hz)	Mode shape	Frequency (Hz)	Mode shape	Frequency (Hz)	Mode shape	Frequency (Hz)
No concentrated mass		34.93		65.6		77.22		102.52
100g concentrated mass at point A		34.77		63.48		74.8		89.68
200g concentrated mass at point A		34.56		58.18		71.14		82.68
100g concentrated mass at point B		33.52		64.74		67.06		101.26
200g concentrated mass at point B		31.91		57.73		65.86		98.54

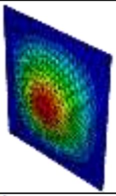
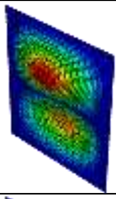
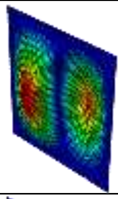
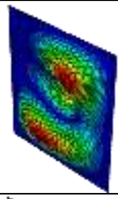
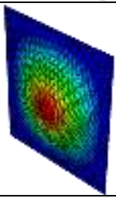
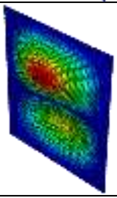
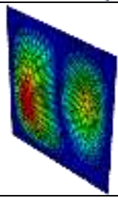
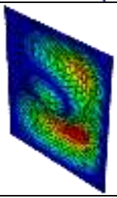
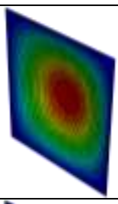
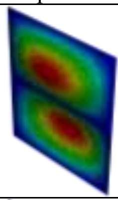
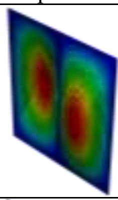
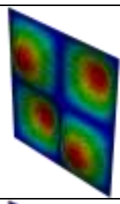
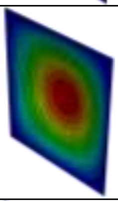
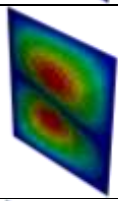
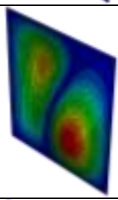
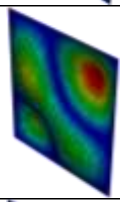
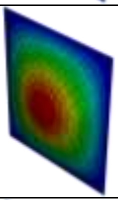
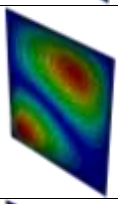
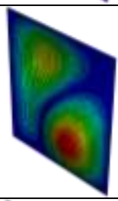
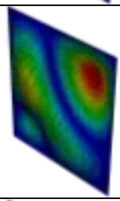
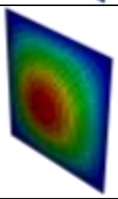
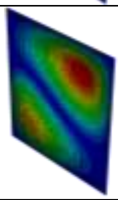
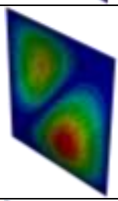
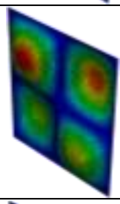
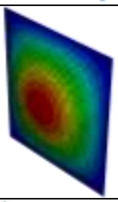
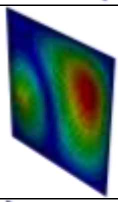
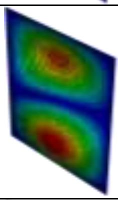
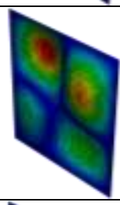
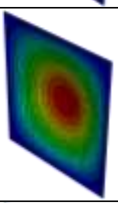
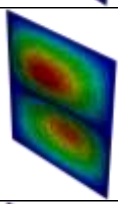
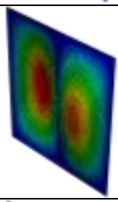
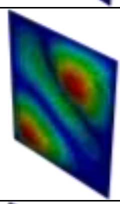
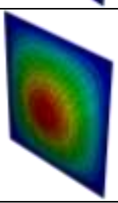
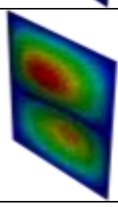
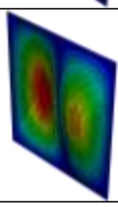
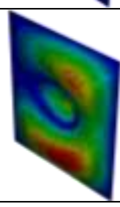
100g concentrated mass at point C		28.74		65.08		76.34		100.1
200g concentrated mass at point C		24.77		64.72		75.64		90.3

Table 7. Mode shapes and natural frequencies of the model carrying concentrated mass under pinned support condition

Concentrated mass condition	**Mode 1**		**Mode 2**		**Mode 3**		**Mode 4**	
	Mode shape	Frequency (Hz)	Mode shape	Frequency (Hz)	Mode shape	Frequency (Hz)	Mode shape	Frequency (Hz)
No concentrated mass		20.13		47.77		56.53		80.53
100g mass at A		19.75		43.99		53.96		69.92
200g mass at A		19.32		39.19		52.61		65.79
100g mass at B		18.88		47.1		48.83		79.94
200g mass at B		17.7		42.71		47.99		79.06
100g mass at C		17.17		47.51		56.1		80.06
200g mass at C		15.15		47.31		55.73		73.21

The mode shapes and natural frequencies of the composite panel without any added concentrated masses were derived. These data were used to establish a baseline for subsequent comparative analysis. These mode shapes, along with their corresponding natural frequencies, are inherent to the material properties of the composite panel and its boundary conditions. To evaluate the sensitivity of these modal parameters to added concentrated masses at various locations with different magnitudes on the composite panel, the resultant frequency variations were calculated. The frequency variations of the composite panel induced by the applied concentrated masses are listed in Table 8 and Table 9 for fixed and pinned support conditions, respectively.

Table 8. Frequency variations under fixed support condition

Concentrated mass condition	Mode 1	Mode 2	Mode 3	Mode 4
100g mass at A	0.16	2.12	2.42	12.84
200g mass at A	0.36	7.43	6.10	19.83
100g mass at B	1.40	4.86	10.16	2.26
200g mass at B	3.02	7.87	11.36	3.98
100g mass at C	6.18	0.52	0.87	2.42
200g mass at C	10.15	0.88	1.58	9.22

Table 9. Frequency variations under pinned support condition

Concentrated mass condition	Mode 1	Mode 2	Mode 3	Mode 4
100g mass at A	0.38	3.78	2.57	10.61
200g mass at A	0.81	8.58	3.92	14.74
100g mass at B	1.25	0.67	7.7	0.59
200g mass at B	2.43	5.06	8.54	1.47
100g mass at C	2.96	0.26	0.43	0.47
200g mass at C	4.98	0.46	0.8	7.32

The mode shapes and frequency variations of the composite plate were analyzed under both fixed and pinned support conditions, revealing significant impacts of concentrated masses on the panel's vibration characteristics. Adding 100 and 200 grams of weight at various points (X, Y, and Z) alters these mode shapes, with noticeable decreases in natural frequencies.

For fixed support, concentrated masses at point X (near the corner) introduce localized stiffness, slightly reducing lower modes' frequencies and effectively damping higher-frequency vibrations. By increasing the weight of the concentrated mass at this point, the effect on the mode shapes was more pronounced. The additional mass further reduces the natural frequencies, particularly for modes with deformation near the corner. At this point, mode four frequency reduction and mode shape variation were the most. At point Y, added concentrated masses influence the global stiffness and inertia of the plate. This causes a more uniform shift in the mode shapes, affecting both lower and higher modes. Higher frequency reductions are observed for Mode three. Weights at point Z (center) cause reductions in natural frequencies across all modes. The central mass primarily affects the fundamental mode, leading to a noticeable decrease in the natural frequency. Since the concentrated mass was located at the center of the plate, mode one and mode four experienced the most reduction in frequency. By increasing the weight, mode shape variations were more pronounced, particularly at mode four.

Pinned support conditions, allowing rotational freedom, show a similar trend but with generally less pronounced effects. Frequency variations confirm these observations: added masses decrease natural frequencies, with point X significantly affecting higher modes, while points Y and Z influence both lower and higher modes with more impact on lower modes.

The magnitude and location of concentrated masses have a substantial impact on the composite plate's natural frequencies and mode shapes. For instance, the closer the mass is to the center, the greater the reduction in the first mode's frequency, indicating a critical sensitivity to central loading. Higher mass values further decrease frequencies, emphasizing the cumulative effect of mass and location on the plate's vibration characteristics.

Adding weights at different points on the panel demonstrates that localized mass can effectively alter the vibration characteristics. Points closer to the corner (X) show localized damping effects, reducing higher-frequency vibrations more effectively. Weights added near the center (Y and Z) influence the global stiffness and inertia of the plate, leading to a more uniform reduction in natural frequencies and effective damping across all modes. For effective vibration control, the distribution of added masses can be optimized based on the desired damping effects. Central masses provide more uniform damping, while localized masses can target specific higher-frequency modes. Different modes react differently to mass placement and support conditions. For instance, Mode four is highly sensitive to corner placements (Point X) under fixed support, while lower modes are more affected by central placements (Point Z). This allows for targeted vibration control strategies based on mode sensitivity. The support condition (fixed vs. pinned) alters the frequency response characteristics. Fixed support conditions generally show higher frequency values compared to pinned, implying that the stiffness of the support condition plays a crucial role in determining the vibrational behavior.

4. Conclusion

This study provides a comprehensive investigation into the effects of concentrated masses on the dynamic response and vibration control of composite panels under varying boundary conditions. The findings underscore the critical role of sensor placement and support conditions in ensuring the accuracy and reliability of modal analysis data. Coherence function analysis reveals that the decay in coherence is more pronounced when sensors are positioned further from the excitation point and closer to the supports, with this effect being particularly significant under fixed support conditions. These observations highlight the necessity of optimal sensor positioning to enhance measurement precision in experimental modal analysis. Furthermore, the introduction of concentrated masses leads to a systematic decay in coherence functions, with higher mass magnitudes exacerbating this effect, further influencing the panel's vibrational response.

A comparative assessment of experimental and numerical modal analysis results identifies discrepancies arising from the inherent simplifications in numerical modeling. To address these inconsistencies, the finite element (FE) model was updated, incorporating more realistic parameters that better align with experimental observations. The updated FE model demonstrates strong agreement with experimental results, validating its accuracy and reliability. This enhancement not only improves predictive capabilities but also minimizes the dependency on extensive physical testing, offering a cost-effective and efficient approach for evaluating the vibrational characteristics of composite structures.

The study also highlights the significant influence of concentrated mass location and magnitude on the modal characteristics of the composite panel. Masses positioned near the center induce substantial reductions in natural frequencies, particularly affecting lower-order modes, while those placed near the edges exhibit a localized stiffening effect, altering higher-mode vibrations. Additionally, fixed support conditions yield higher natural frequency values compared to pinned supports, reinforcing the crucial role of boundary constraints in governing vibrational behavior.

These findings have important implications for vibration control strategies in composite structures. The ability to strategically manipulate mass distribution provides an effective means of tuning vibrational characteristics and optimizing damping performance. The study offers valuable insights into the interplay between mass placement, boundary conditions, and modal behavior, contributing to the development of advanced vibration control methodologies for composite marine and offshore structures.

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