

Article

Neural Approach to Study the Vibration Behavior of Damaged Composite Rotating Beams

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Abstract

In recent decades, Artificial Neural Networks (ANNs) have become a robust tool for addressing complex engineering challenges. This paper implements an ANN-based methodology to determine the natural frequencies of rotating sandwich composite beams with core defects. The study focuses on the influence of rotation speed and defect characteristics (size and location) on a beam made of carbon fiber face-sheets and a honeycomb core, selected for its high strength-to-weight ratio in next-generation designs. The primary novelty lies in providing a simplified model that, through an ANN-based surrogate, establishes an automated and high-speed process for frequency prediction. This approach bypasses the prohibitive computational costs of 3D-FEM simulations, enabling near-instantaneous results essential for real-time Structural Health Monitoring (SHM) applications.

Keywords: neural networks; numerical model; natural frequencies; composite material; damaged rotating beams

1. Introduction

Renewable energy sources offer significantly lower environmental impact than conventional power sources. Unlike fossil fuels, they do not emit greenhouse gases or pollutants, thereby mitigating their contribution to climate change. Wind energy generation has emerged as a leading alternative, currently representing the fastest-growing energy sector worldwide [1]. Consequently, optimizing maintenance processes is vital to improving system reliability and operational efficiency. As blades are among the most critical components of a wind turbine, any advancement in their design, maintenance, or structural analysis directly enhances the overall performance of the energy system. To facilitate these advances, researchers often employ simplified rotating beam models as a fundamental benchmark [2,3], allowing for a precise analysis of dynamic responses that would be computationally prohibitive in full-scale blade geometries.

Wind turbine blades are typically constructed using composite sandwich structures, valued for their low specific weight and high strength-to-weight ratio. These structures consist of two thin, high-strength face-sheets bonded to a lightweight core. While current commercial standards often rely on Glass Fiber (GFRP) and foam cores, Carbon Fiber (CFRP) is increasingly being investigated for next-generation large-scale blades due to its superior mechanical properties [4]. Regarding core materials, although solid foam is widely used, honeycomb cores—comprised of thin-walled hexagonal cells—minimize material



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volume while offering exceptional specific strength [5]. The use of a CFRP/honeycomb configuration within a simplified rotating beam framework allows for the evaluation of the dynamic behavior of these materials under rotational conditions. This approach provides a reference for characterizing the influence of defects on the dynamic signature of advanced composites, serving as a precursor to more complex, site-specific blade applications.

Many authors have studied the vibration of rotating beams [6–18]. Many of these studies have focused on the analysis of beams of traditional materials [6–11]. Giurgiutiu and Stafford [8] developed the equations of motion including shear and rotatory inertia for coupled linear vibrations of blade rotating at constant angular velocity in a fixed plane. Bhat [9] studied the natural frequencies and the mode shapes of a rotating uniform cantilever beam with a tip mass by using beam characteristic orthogonal polynomials in the Rayleigh-Ritz method. Other researchers have studied the vibratory behavior of rotating blades made with composite structures [12–18]. Patel and Ganapathi [13] investigated the vibrations of rotating beam made of anisotropic laminated composite beam using a new three-noded finite element. Lee et al. [14] examined the behavior of flapwise vibration of rotating multilayer composite beam using Timoshenko beam theory. Arvin and Bakhtiari-Nejad [16] investigated linear and nonlinear free vibrations of rotating composite Timoshenko beams. Ozdemir and Kaya [15] analyzed extension and flapwise bending vibrations of a rotating piezolaminated composite Timoshenko beam. Aksencer and Aydogdu [12] studied the vibration of rotating composite beams using different beam theories including Euler-Bernoulli, Timoshenko and Reddy theories. The same authors, in a publication posterior, studied vibration and buckling of rotating composite cantilever beam with clamped-off the rotation axis [19]. They investigated the effects of various parameters like rotation speed, critical speed, orthotropy ratio, length to thickness ratio and beam theories on the vibration of rotating clamped off the rotation axis composite beams. However, in the knowledge of the authors, no studies has been found in the literature focused on vibrations in rotating beams made with sandwich structures with CFRP face-sheets and honeycomb cores such as those used in this study.

Moreover, the blades of a wind turbine can be damaged during its service life. Due to manufacturing defects or working conditions, together with the aggressive environments in which they operate, defects may appear in the blades that seriously affect their structural integrity. It is quite evident that for damaged rotating beams the dynamic properties such as frequency and vibration modes change [20]. Therefore, in order to repair or replace the element before irreversible failure occurs, it is very important to know techniques that allow the detection of the defect [20–26]. Chang and Chen [2] presented a technique for blade damage detection based on spatial wavelet analysis. For this, they applied the Finite Element method for the analysis of the vibrational behavior of the cracked rotating beams. The effects of transverse shear deformation and rotatory inertia were taken into account. Lee et al. [22] studied the variations in the natural frequencies of vibration of a multi-cracked beam produced by different rotational speeds and crack locations. They used the Transfer Matrix and Frobenius methods and validated the results with a numerical model. Valverde-Marcos et al. [20] presented a methodology to accurately calculating the flapwise bending vibration natural frequencies of a rotating cracked steel beam. They developed a 3D dynamic numerical model and the natural vibration frequencies were calculated by means of a frequency domain analysis of accelerations of the free end of the rotating beam. However, the number of studies on the behavior of damaged composite rotating beams is small [27–29]. Chen and Shen [27] developed a finite element model to study the dynamic stability aspects of a composite cracked rotating beam of general orthotropy. They described the effects of rotation speed, local flexibility and the fiber orientations on the static buckling loads and dynamic instability regions of the orthotropic

beam. Kim and Kim [28] investigated the effect of a crack of a composite rotating beam by applying the Timoshenko beam theory. The Finite Element Method was used to model and numerically analyze the crack, respectively. They studied the effects of various parameters such as crack depths, crack positions, fiber angles, volume fractions, and rotating speeds of the beam. Shahedi and Mohammadimehr [29] studied the high-order free vibration analysis of rotating fully-bonded and delaminated sandwich beams.

Over the past two decades, numerous researchers have employed artificial neural networks (ANNs) to address a wide range of engineering challenges, including pattern recognition, optimization, and prediction [30–33]. An ANN is a computational model inspired by biological neural networks, consisting of interconnected neurons arranged in layers: an input layer, one or more hidden layers, and an output layer. Each neuron is connected by adjustable weights that determine the strength of signals between them.

In this paper, we have used an ANN to calculate the natural frequencies of rotating composite beams made with CFRPs face-sheets and honeycomb cores that have a transverse discontinuity in the core. For this purpose, we have developed a 3D finite-element model in Abaqus/Implicit code. The model has been validated against solutions from the literature and used to study the vibratory characteristics of the beam as a function of the discontinuity size, location, and rotation speed. The acceleration at the beam's tip was chosen as the output signal to obtain the natural frequencies. Based on this numerical data, an ANN was derived to establish an automated, high-speed diagnostic framework. For the first time, in the knowledge of the authors, an ANN to determine the natural frequencies of damaged composite beams formed by sandwich structures with carbon fiber reinforced polymer face-sheets and honeycomb core in terms of the rotation speed and the characteristics of the defect (size and location) has been presented. The proposed methodology offers a robust and computationally efficient option to analyze the vibrations of damaged rotating sandwich structures, bypassing the costs of traditional 3D-FEM for real-time applications.

2. Rotating Damaged Beam Model

Problem Statement

A rotating damaged composite sandwich beam with a rectangular section has been considered. The dimensions of the beam are shown in Figure 1. The beam rotates counterclockwise around the Y axis with constant angular velocity Ω and has a transverse discontinuity in the core at a distance L_d from the clamped end of the beam. The geometrical configuration of the discontinuity has been defined with the following parameters:

- The relative discontinuity size $\alpha = \frac{d_1}{L}$. Different relative sizes of the discontinuity in the honeycomb core have been considered: $\alpha = 0, 0.01, 0.02$ and 0.03 , with the value $\alpha = 0$ corresponding to an intact beam. Figure 2 shows the geometry of the transverse discontinuity.
- The relative discontinuity location $\xi = \frac{L_d}{L}$. We have considered the following locations: $\xi = 0.14, 0.355, 0.57$ and 0.785 (see Figure 1).

The composite sandwich beams has been based on face-sheets of Carbon Fiber Reinforced Polymer (CFRP) and Nomex[®] honeycomb core. The carbon-fiber composite face-sheets, IM7/MTM-45-1, have been made of 16 sheets and present a stacking sequence equal to $[0/ + 45/90/ - 45]_{25}$. The main material properties are shown in Table 1. The core is a regular hexagonal honeycomb cell structure and has been made of Nomex[®] by Toray Advanced Composites and with the designation ANA-3.2-48, which means a cell size of 3.2 mm and a nominal density of 48 kg/m^3 [34]. The properties of Nomex[®] are those shown in Table 2.

In addition, eight rotational speeds ranging from $\Omega = 30 \text{ rad/s}$ to $\Omega = 100 \text{ rad/s}$ with increments of 10 rad/s have been considered. It should be noted that the blades of a wind

turbine rotate between 1.36 and 2 rad/s. To scale the problem, the speeds equivalent to the size of the considered beam would vary between 56.7 and 87.2 rad/s. However, the present study intends to cover a large number of cases, so angular velocities outside the range of speeds at which wind turbine blades normally rotate have been considered. The beam has an encastre at the end $X = 0$, which coincides with the axis of rotation, and it is free at the other end $X = L$.

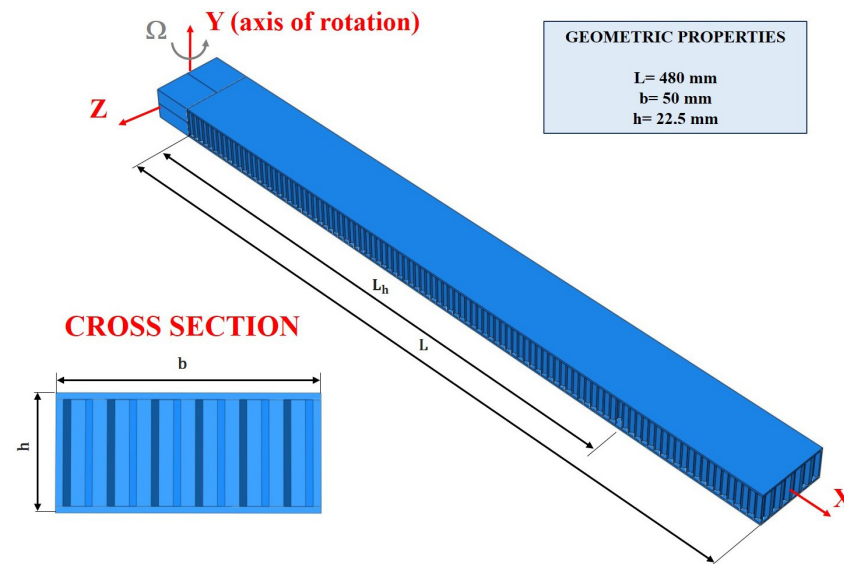


Figure 1. Geometric model.

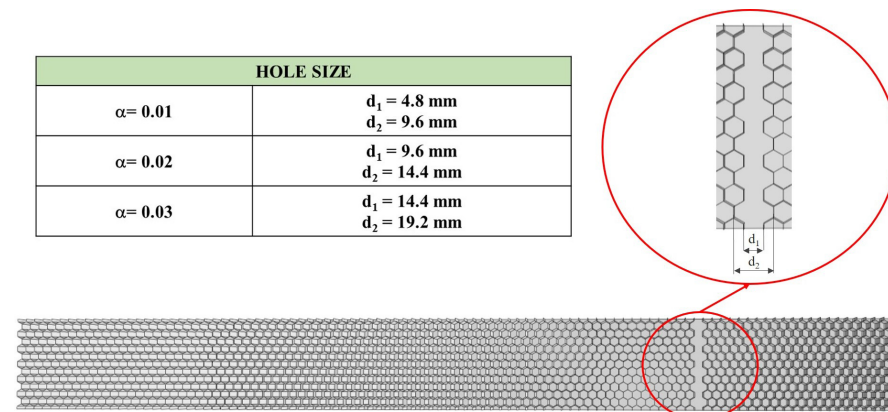


Figure 2. Geometric configuration of the transverse discontinuity.

Table 1. Material properties of IM7/MTM-45-1 face-sheets.

Density	1600 Kg/m ³
Young's Modulus E_1	173 GPa
Young's Modulus $E_2 = E_3$	73.6 GPa
Poisson ratio ν_{12}	0.32
Poisson ratio $\nu_{13} = \nu_{23}$	0.5
In plane shear modulus G_{12}	3.89 GPa
Interlaminar shear modulus G_{13}	3.89 GPa
Interlaminar shear modulus G_{23}	2.94 GPa

Table 2. Honeycomb Core Nomex[®] Properties.

Density	1500 Kg/m ³
Young's Modulus E_1	3.95 GPa
Young's Modulus E_2	5.05 GPa
Poisson ratio ν_{12}	0.2
Interlaminar shear modulus G_{12}	1.6 GPa
Interlaminar shear modulus G_{13}	1.6 GPa
Interlaminar shear modulus G_{23}	1.6 GPa

3. Numerical Model

In the present work, a dynamic implicit analysis has been performed by implementing a 3D model of the damaged composite rotating beam using the Commercial Code ABAQUS. The model consists of different solids of different sizes, the first one being deformable, corresponding to the sandwich beam, and the second one being rigid, simulating a physical axis of rotation. To join the sandwich beam and the rigid part, a “Tie” constraint (according to ABAQUS nomenclature) has been established to prevent relative displacement between the two parts (see Figure 3). Moreover, the honeycomb core and the face-sheets have been modelled independently, and also “Tie” constraints have been used between the core and face-sheet interfaces.

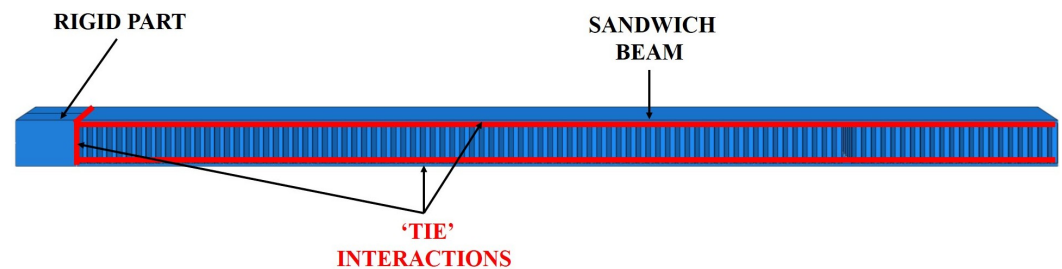


Figure 3. Different solids of the numerical model.

The complete mesh including all the solids has consisted of 80,000 elements and 160,000 nodes, approximately. For the rigid part, 3-node linear triangular rigid elements for three-dimensional analysis (R3D3 according to ABAQUS nomenclature) have been used; while for the deformable part, two types of elements have been used: the facesheets have been meshed using 8-node hexahedral elements with reduced integration and hourglass control (SC8R in ABAQUS nomenclature) and for the core, 3-node linear triangular shell elements (S3 according to ABAQUS nomenclature) have been used. A detail of the beam mesh can be seen in Figure 4.

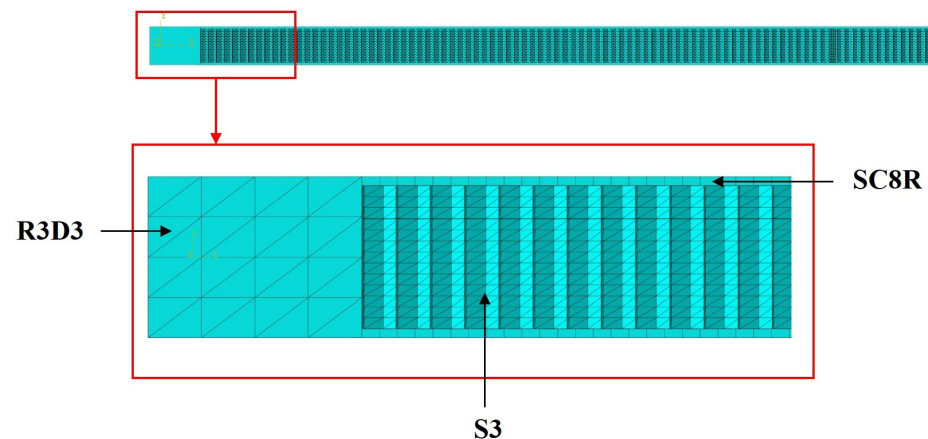


Figure 4. Detail of beam mesh.

The rotation speed has been simulated using a predefined constant angular velocity field over the entire beam following the procedure of Valverde-Marcos et al. [20]. For each rotation speed, to reach convergence of the results, it has been necessary to complete 7 rotations of the beam. The sampling frequency is 10,000 Hz.

4. Obtaining Frequencies from Numerical Model

A frequency domain analysis of accelerations in Y-direction of the free end of the rotating beam has been used to obtain the natural vibration frequencies. The acceleration

data in Y-direction of the free end have been extracted from numerical model and the signal has been processed using the Fast Fourier Transform (FFT). Figure 5 shows an example of the results of the obtained acceleration and signal processing for an undamaged beam with a rotational speed $\Omega = 100$ rad/s to obtain the first four natural frequencies in the XY plane. As can be seen, in the first instants of the simulation, there is a transient state of higher amplitude in the acceleration history. A Kaiser window function [35] has been used, with a form factor $\beta = 10$, which reduces the leakage effect and attenuates the signal laterally.

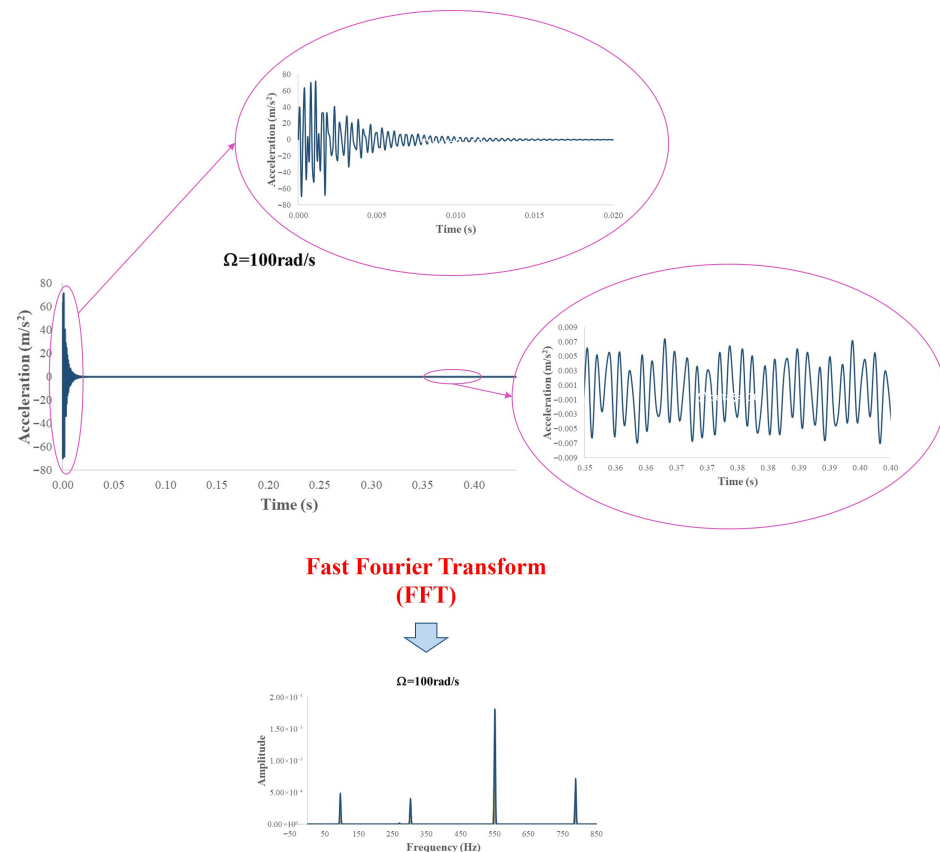


Figure 5. Acceleration and signal processing for an undamaged beam with a rotational speed $\Omega = 100$ rad/s.

5. Verification of the Numerical Model

The verification of the proposed model has been carried out by comparing the results obtained with others available in the literature.

5.1. Comparison with Literature Results of Rotating Beams

First, in order to verify the procedure of obtaining the frequencies, the results have been compared with the results obtained by the analytical method used by Muñoz-Abella et al. [25]. In [25] closed-form solutions for determining the first two natural frequencies of the flapwise bending vibration of a cracked beam at different rotational speed were obtained. Since no results have been found in the literature for the rotating sandwich beams under study, the numerical results obtained with the proposed model manufactured with a conventional material such as steel have been compared with results from [25], obtaining excellent results. Specifically, the values of the first and second natural frequencies for different rotational speeds have been compared. Table 3 shows the relative errors in absolute value, according to in Equation (1) obtained in the comparison. It can

be seen how the results are very similar with a maximum relative error of 2.15% for the lower speed.

$$Error(\%) = 100 \cdot \left| \frac{\omega_{FEM} - \omega_M}{\omega_M} \right| \quad (1)$$

where ω_{FEM} is the frequency obtained by the developed numerical model and ω_L is the frequency obtained by Muñoz-Abella et al. [25].

Table 3. Relative errors in absolute value.

Ω (rad/s)	Error $_{\omega_1}$ (%)	Error $_{\omega_2}$ (%)
30	0.03	2.15
40	0.14	2.05
50	0.35	1.96
60	0.60	2.00
70	0.58	1.79
80	0.23	1.84
90	1.31	1.67
100	0.88	1.73

5.2. Comparison with Literature Results for Non-Rotating Beams

Second, the natural frequencies of the developed sandwich beam have been compared with those of Pourriahi et al. [36] obtained with different boundary conditions. In [36] the first three frequencies of a free sandwich beam with free-free boundary conditions formed by composite face-sheets and honeycomb core, like the beam object of this work, were obtained. For the comparison, apart from the change in boundary conditions, it has been necessary to adapt the beam dimensions and material because at [36] the face-sheets were made of 7075 aluminum and the honeycomb core was made of 5052 aluminum. Table 4 shows the results of the comparison of the natural frequencies for three different modes along with the relative errors. It can be seen how the results practically coincide with relative errors of less than 0.8%. With the comparison of the first three frequencies, the vibration behavior of the beam is considered to be verified.

Table 4. Comparison with the frequencies obtained by Pourriahi et al. [36].

Mode	ω_{Model} (Hz)	$\omega_{Pourriahi}$ (Hz)	Error (%)
Mode 1	1244.5	1235.2	0.75
Mode 2	3062.5	3041.5	0.69
Mode 3	5336.1	5377.1	0.77

6. Numerical Results

6.1. Influence of the Rotation Speed

From the validated numerical model, the natural frequencies have been obtained for all the cases considered. Firstly, the variation of natural frequencies as a function of the rotation speed has been analyzed. Finite element models from $\Omega = 30$ rad/s to $\Omega = 100$ rad/s with increments of 10 rad/s have been developed for the undamaged beam and for different transverse discontinuity sizes and locations. Therefore, a total of 104 cases have been developed. For each case, the first four natural frequencies have been obtained. Figures 6–8 show an example of the results obtained for an intact beam, for a beam with a discontinuity with relative location $\xi = 0.785$ and relative size $\alpha = 0.01$ and for a beam with a transverse discontinuity with relative location $\xi = 0.355$ and relative size $\alpha = 0.03$, respectively. Each graph represents the Fast Fourier Transform (FFT) spectra of the structural response for the first four natural frequencies (ω_1 – ω_4) across various rotational speeds (Ω)(the scales of the graphs are different to facilitate its visualization). The following can be seen:

- The value of the natural frequency increases as the angular velocity increases. This makes sense, since increasing the rotation speed increases the stiffness of the beam [20].
- The increase in frequency value with speed is less for the first natural frequency.
- Likewise, for all cases considered, the amplitudes of the signal also increase as the rotational speed increases.

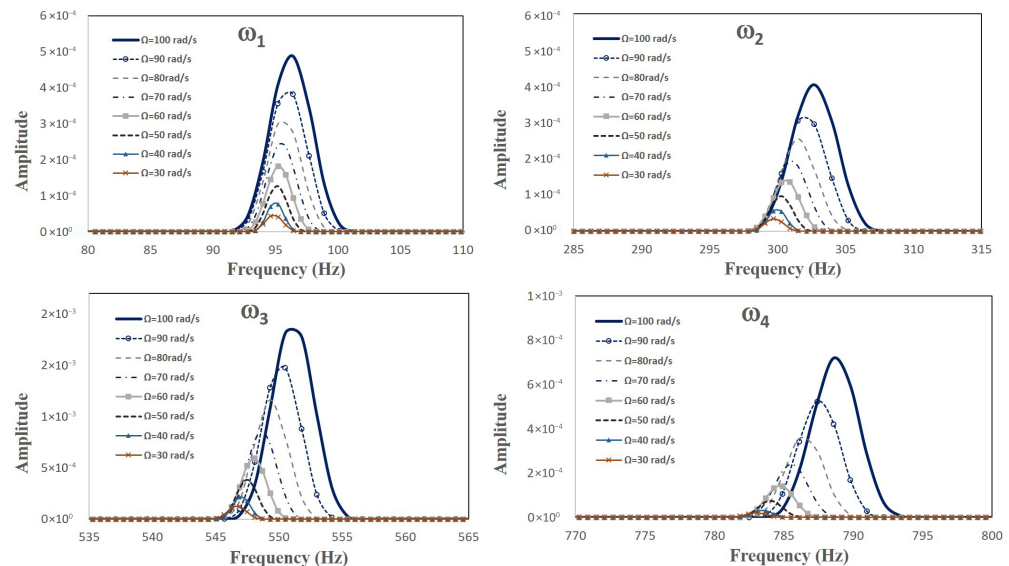


Figure 6. Frequency spectrum (FFT) for an intact beam, showing the first four natural frequencies (ω_1 – ω_4) at the different rotational speeds.

6.2. Influence of Defect Size and Location

In order to analyze the influence of the discontinuity size and location, the frequency as a function of the discontinuity location has been plotted for the different relative discontinuity sizes. Figure 9 shows an example of the results for the four natural frequencies and for the rotational speed $\Omega = 100$ rad/s. In view of the results shown in the figure, which are analogous to those obtained for the other rotational speeds, the following can be concluded:

- For all cases considered, the presence of the transverse discontinuity in the honeycomb core reduces the value of the natural frequencies due to an increase in local flexibility [37].
- In the case of the first natural frequency it is observed, as expected, that the value of the frequency decreases as the defect size increases and the effect of the defect is smaller as the relative position increases. This coincides with the results obtained by [26].
- Regarding the second natural frequency, the same behavior can be seen with respect to size, the frequency value decreases with size. As for the influence of the location, for $\xi = 0.14$ the minimum in the value of the frequency is found for all sizes, then the frequency increases to $\xi = 0.355$, where it remains nearly constant regardless of defect severity. This point corresponds to a vibration antinode of the second mode; at this location, the modal curvature (second derivative of the mode shape) is minimal, rendering the defect's impact nearly negligible for this specific frequency. Beyond this point, the frequency decreases again, reaching a second local minimum at $\xi = 0.785$.
- The third natural frequency exhibits a similar downward trend as the damage severity (α) increases. Regarding its location, the lowest frequency values across all sizes are found at $\xi = 0.355$ and $\xi = 0.785$, as these points are located in the immediate vicinity of vibration nodes where modal curvature is maximal. Conversely, at $\xi = 0.57$, the frequency exhibits lower sensitivity to the defect severity due to the proximity to a vibration antinode.

Although it does not coincide exactly with the theoretical point of zero curvature, its proximity to the antinode results in a marked local insensitivity to damage.

- Finally, the fourth natural frequency also exhibits a downward trend as defect severity α increases. Notably, this mode proves to be highly sensitive to the size of the transverse discontinuities, showing more pronounced frequency shifts compared to lower-order modes. Regarding the influence of location, the minimum frequency is observed at $\zeta = 0.57$, identifying this region as being in the vicinity of a vibration node for the fourth mode shape. Conversely, at $\zeta = 0.14$ and $\zeta = 0.785$, the frequency remains nearly invariant regardless of the defect size. This behavior indicates that these two locations are situated near vibration antinodes for this specific mode, where the minimal modal curvature minimizes the impact of core discontinuities on the global dynamic response. It should be noted that the current results for ω_4 suggest that a greater number of relative damage locations (ζ) would be necessary to fully capture its complex behavior. This does not significantly detract from the overall robustness of the model, as ω_4 is rarely included in such analyses.

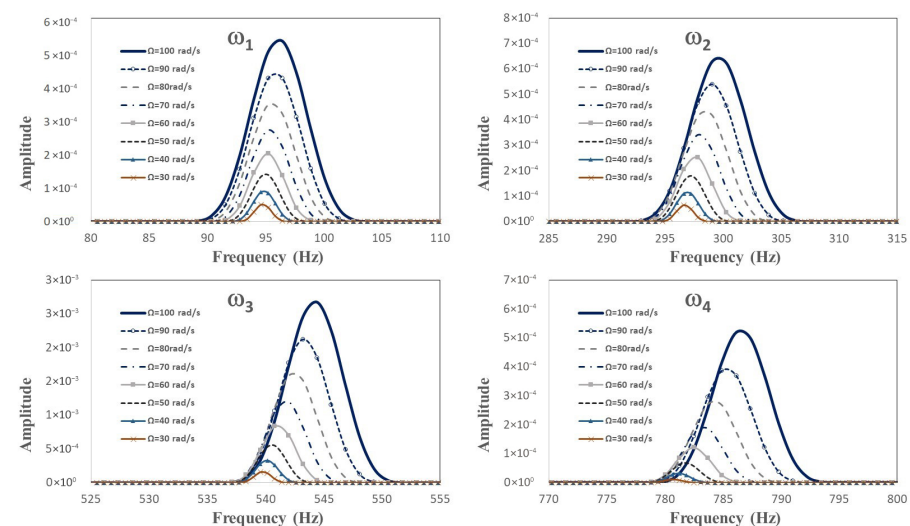


Figure 7. Frequency spectrum (FFT) for a beam with a discontinuity with relative location $\zeta = 0.785$ and relative size $\alpha = 0.01$, showing the first four natural frequencies (ω_1 – ω_4) at the different rotational speeds.

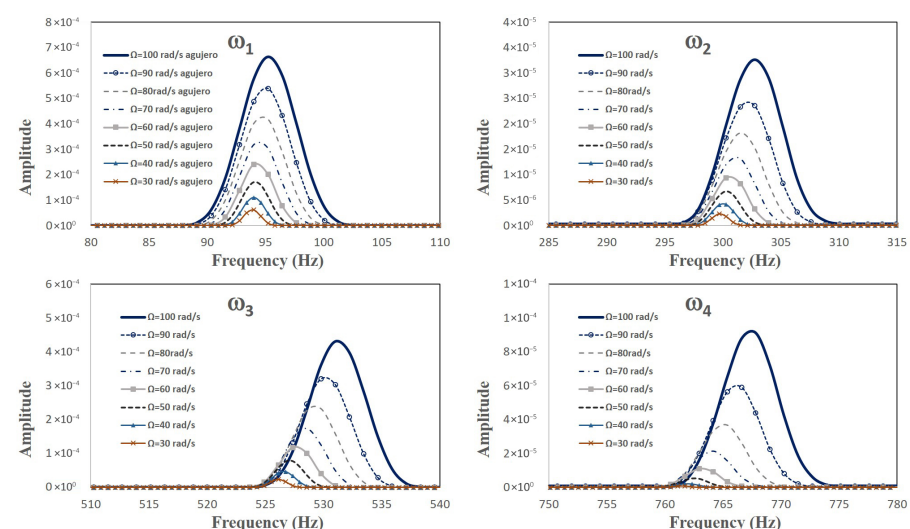


Figure 8. Frequency spectrum (FFT) for a beam with a transverse discontinuity with relative location $\zeta = 0.355$ and relative size $\alpha = 0.03$, showing the first four natural frequencies (ω_1 – ω_4) at the different rotational speeds.

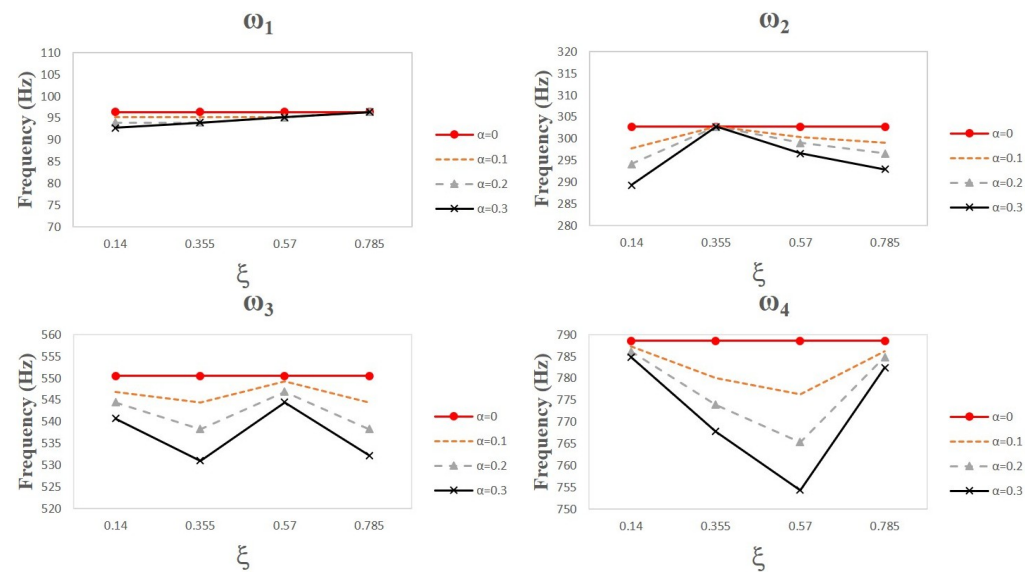


Figure 9. Natural frequencies versus relative discontinuity location (ω_1 – ω_4) for the different relative discontinuity sizes and for the rotational speed $\Omega = 100$ rad/s.

To quantify the measurable effect of structural damage, a numerical summary of the maximum relative frequency changes is provided:

- **First frequency:** The maximum change in frequency occurs approximately at $\xi = 0.14$.
- **Second frequency:** The maximum change in frequency occurs approximately at $\xi = 0.14$.
- **Third frequency:** The maximum change in frequency occurs approximately at $\xi = 0.355$ and $\xi = 0.785$.
- **Fourth frequency:** The maximum change in frequency occurs approximately at $\xi = 0.57$.

7. Artificial Neural Network Approach

Based on the data from the numerical model, an ANN that allows to obtain the value of the first four frequencies as a function of the characteristics of the defect (size and position) and the rotational speed of the beam has been implemented. An ANN is a flexible mathematical tool frequently employed in solving various engineering challenges, owing to its robustness and its capability to discern complex nonlinear relationships between input and output data. It is composed of units called neurons or nodes, which mimic biological neurons and, like them, are interconnected with each other. Each of the neurons receives several inputs, I_i , coming from the outside or from the outputs of other neurons. The output, O , is obtained from an activation function, f , which is applied to the sum of all inputs multiplied by their corresponding weights, w_i , plus a threshold value called bias, b , which limits the value of the output. Therefore, the relationship between the inputs I_i and the output O can be written as:

$$O = f(\sum w_i \cdot I_i + b) \quad (2)$$

Figure 10 shows the schematic of an isolated neuron.

The neurons are arranged in layers. The typical structure of an ANN includes: an input layer, one or more hidden layers, and an output layer. Each layer contains one or more neurons. The neurons in the input layer are linked to the external environment or raw data. The hidden layers process the input layer's outputs and transmit the results to the output layer to achieve the desired results. The type of activation function determines the configuration of the ANN. Various types of activation functions exist, each suited to different problems, including step, ramp, sigmoid, log-sigmoid, and Gaussian functions.

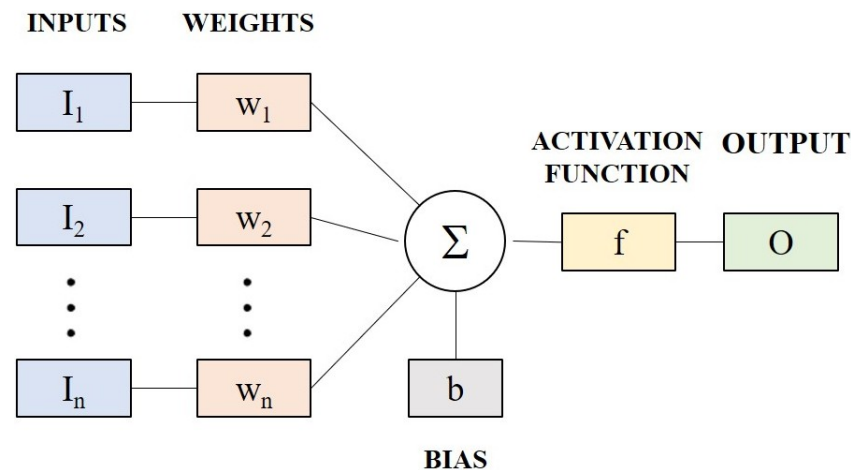


Figure 10. Scheme of a single neuron.

In this work, the multilayer perceptron feedforward neural network has been used. In this type of network, calculations flow in a single direction (feedforward), from the input data to the outputs. Each neuron in a layer is connected to every neuron in the subsequent layer through weighted connections, and these weights are adjusted during the training process to minimize the error in predictions. The available data have been randomly divided into 3 groups, used for training, validation and network testing, consisting of 70%, 15% and 15% of the data, respectively. First, the network training is performed, which consists of determining through an iterative process the optimal weights and thresholds to find the relationship between the input and output data. In this case, the Levenberg-Marquardt backpropagation forward supervised learning algorithm has been used, which consists of processing the inputs, obtaining the corresponding outputs and comparing them with the desired outputs until the error between the desired outputs and those obtained by the network is less than a predetermined value. Simultaneously to the training process, for each calculation step, a validation is performed, which consists of using the network calculated up to that moment to estimate data outputs different from those used during training and, finally, once the optimal weights and thresholds have been determined, a network test process is performed, in which a new set of input and output data, different from those used previously, is used.

Figure 11 shows the network schematic in which its inputs and outputs can be seen. There are four inputs (size, position, speed and frequency order) and one output (dimensionless natural frequency). The dimensionless frequencies, μ , have been obtained by dividing by the natural frequencies of an intact non-rotating beam. Regarding the activation functions, f , the log-sigmoid function has been used in all layers except in the output layer, where a linear function has been used.

The variables used to test the accuracy of the network have been the mean square error (MSE), calculated according to expression (3) and the correlation coefficient R^2 .

$$MSE = \frac{1}{n} \sum_{i=1}^n (\mu_i^{Estimated} - \mu_i)^2 \quad (3)$$

where μ_i is the value of the dimensionless frequency obtained with the numerical model and $\mu_i^{Estimated}$ is the estimated value with the network for each case. The best results have been obtained with a network with one hidden layer, with 35 neurons with a mean square error $MSE = 2.02 \cdot 10^{-6}$ and a correlation coefficient $R^2 = 0.9925$.

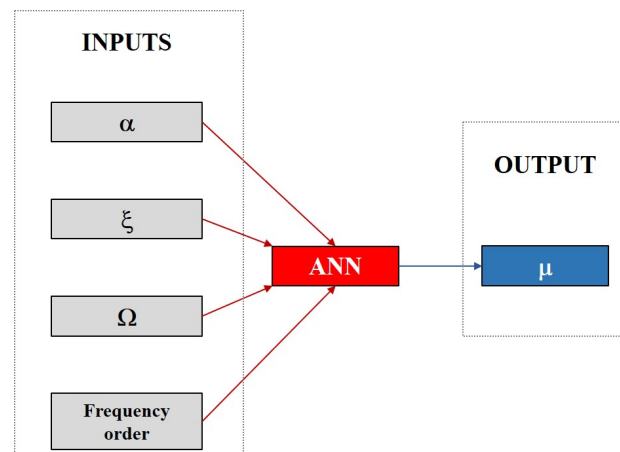


Figure 11. Scheme of the artificial neural network.

The training process of the ANN was monitored through the Mean Squared Error (MSE) convergence curves (see Figure 12). As shown in the results, the training, validation, and testing errors decrease consistently, reaching a stable state around the 30th epoch. The fact that the validation and test curves remain closely aligned with the training curve, without showing an upward trend (divergence), provides categorical evidence that the network is not overfitted.

7.1. Comparison with Values Used in Network Training

In order to verify the proposed ANN, a comparison of the numerical values used in the network development and those estimated by the network has been carried out. Figure 13 shows some examples of this comparison for the four natural frequencies and for the different positions of the defect. The values of the frequency as a function of the defect size have been plotted for the rotational speeds $\Omega = 30$ and 90 rad/s. It has been found that they are in good agreement. The cases corresponding to other frequencies and other defect positions are similar to the ones that have been shown. In addition, in Table 5 it can be seen the relative mean error according to expression (4) for every discontinuity size and location, taking into account all the frequency orders and all the rotational speeds.

$$Error(\%) = \frac{1}{n} \sum_{i=1}^n 100 \cdot \left| \frac{\mu_i - \mu_i^{Estimated}}{\mu_i} \right| \quad (4)$$

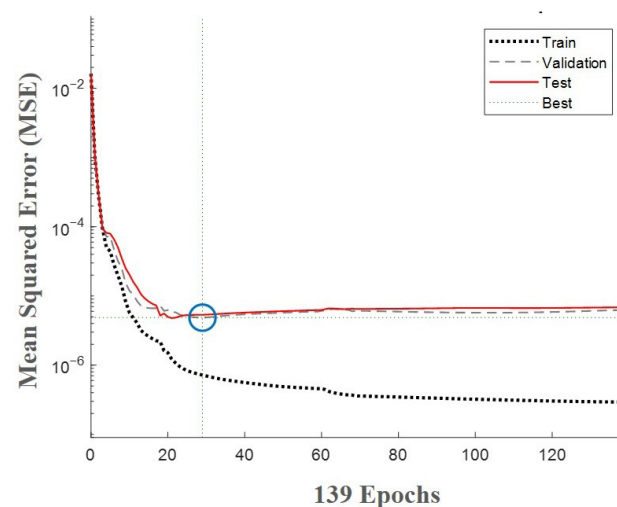


Figure 12. Convergence curves of the ANN training process.

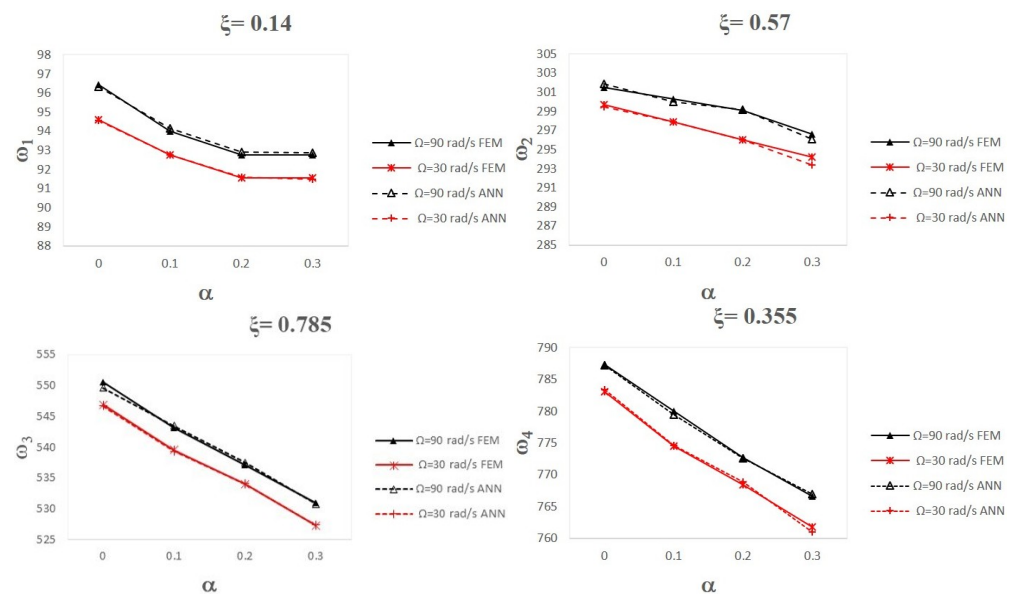


Figure 13. Comparison of the numerical values of the frequency and those estimated by the network.

Table 5. Mean relative error (%).

	$\zeta = 0.14$	$\zeta = 0.355$	$\zeta = 0.57$	$\zeta = 0.785$
$\alpha = 0$	0.92	0.92	0.92	0.92
$\alpha = 0.1$	2.13	1.55	1.36	1.43
$\alpha = 0.2$	3.08	2.08	2.04	1.96
$\alpha = 0.3$	3.07	1.54	2.72	2.53

It can be observed that calculated errors are small, the largest not exceeding 3.1%. To complete the study of the comparison between the original values and those estimated by the network, the analysis of the residuals has been performed. Figure 14 shows the residuals ($\mu_i^{Estimated} - \mu_i$) versus the values predicted by the network. It can be seen that the residuals are randomly distributed and show no pattern.

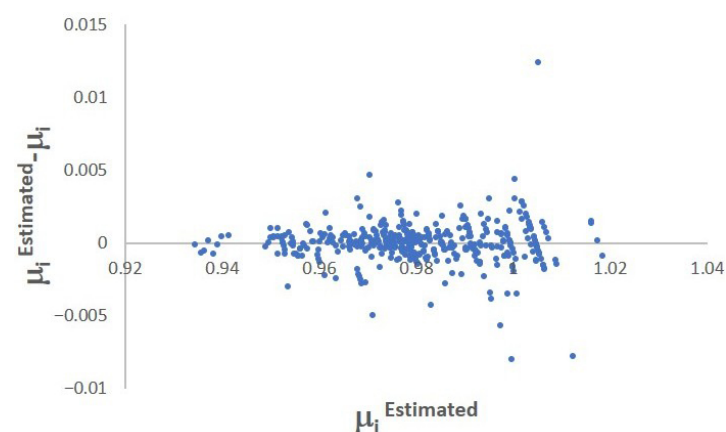


Figure 14. Residuals versus the values predicted by the network.

7.2. Comparison with Values Not Used in the Training of the Network

Finally, in order to test the robustness and accuracy of the proposed ANN, 6 cases of beams with different characteristics from those used for its training were randomly selected and the four frequencies have been calculated for each case. Table 6 shows the results obtained.

Table 6. Comparison with values not used in ANN training.

Ω (rad/s)	ζ	α	Frequency Order	ω	$\omega_{Estimated}$	Error (%)
96	0.635	0.1	1	95.21	95.57	0.38
			2	299.07	299.66	0.19
			3	550.54	542.84	1.39
			4	776.37	784.26	1.02
62	0.635	0.1	1	95.21	95.68	0.49
			2	297.85	298.67	0.27
			3	548.09	540.54	1.38
			4	772.05	780.09	1.04
45	0.635	0.1	1	96.60	95.01	0.42
			2	297.241	297.9	0.22
			3	547.48	540.26	1.32
			4	772.09	778.44	0.81
99	0.405	0.3	1	93.99	93.38	0.65
			2	303.95	303.49	0.15
			3	523.68	537.65	2.67
			4	783.69	750.85	4.19
67	0.405	0.3	1	92.77	93.22	0.49
			2	301.51	301.19	0.11
			3	521.24	534.79	2.60
			4	780.03	746.44	4.30
38	0.405	0.3	1	92.16	92.50	0.37
			2	300.90	300.64	0.09
			3	519.41	533.21	2.65
			4	778.20	743.75	4.42

As can be seen, the estimation provides good results, in none of the cases the error is greater than 4.5%. It is important to emphasize that the maximum error (4.5%) is confined to the fourth natural frequency (ω_4), whereas the error for the first and second frequencies—the most critical for practical applications—is much smaller. The methodology relies on the simultaneous analysis of all four frequencies, using the lower frequencies for stable detection and the higher-order modes as auxiliary data to resolve spatial ambiguities in damage location.

8. Conclusions

In the present paper, a methodology based on Artificial Neural Networks (ANN) has been developed to estimate the first four natural frequencies of damaged composite sandwich beams as a function of rotational speed, defect size, and location. Based on the results and the validation process, the following conclusions can be drawn:

- The dataset required to train the Artificial Neural Network (ANN) was generated using a 3D dynamic numerical model of a rotating sandwich beam. This model incorporates the complex architecture of CFRP face-sheets and a honeycomb core, a configuration widely utilized in high-performance applications due to its exceptional strength-to-weight ratio and superior mechanical properties. By employing a 3D approach, the model captures the intricate structural response of these materials under varying rotational speeds and damage conditions.
- The natural frequencies of the sandwich beam were extracted by applying the Fast Fourier Transform (FFT) to the acceleration signals generated by the dynamic model.
- The analysis of the FFT spectra reveals that the sensitivity of the natural frequencies is highly dependent on the damage location (ζ). The results indicate that frequency shifts are maximized when the discontinuity coincides with antinodes, while the system remains nearly insensitive to damage located near vibration nodes.
- The proposed ANN architecture offers excellent predictive accuracy. When compared with the formulation dataset, the relative error remains below 3.1%. When tested against independent data not used during training, the maximum error is 4.5%. This error of 4.5% is strictly confined to the fourth natural frequency (ω_4), whereas the

first and second modes—which are the most critical for practical structural monitoring—exhibit significantly higher accuracy.

- To the best of the authors' knowledge, this study presents, for the first time, an ANN capable of predicting the natural frequencies of damaged rotating sandwich beams formed by sandwich structures with carbon fiber reinforced polymer face-sheets and honeycomb core a function of rotation speed and defect characteristics (size and location). The proposed methodology offers a robust and computationally efficient framework for analyzing the vibrations of damaged sandwich structures, bypassing the high costs of traditional 3D-FEM for real-time applications.

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Abbreviations

Ω	Angular velocity
L	Length of the beam
L_h	Distance from the defect to the clamped end of the beam
h	Beam height
b	Beam width
α	Relative discontinuity size
ξ	Relative discontinuity location
d_1	Distance 1 from the transverse discontinuity
d_2	Distance 2 from the transverse discontinuity
ν	Poisson ratio
E	Young's Modulus
G	Interlaminar shear Modulus
β	Form factor
ω	Natural frequency
ω_{FEM}	Numerical value of the natural frequency
ω_M	Value of the natural frequency obtained with the analytical method used by [23]
$\omega_{Pourriahi}$	Value of the natural frequency obtained by [35]
I_i	neuron inputs
w_i	neuron weights
b	neuron bias
O	neuron output
f	activation function
MSE	mean squared error
μ_i	Numerical value of the dimensionless frequency for each case
$\mu_i^{Estimated}$	Estimated value of the dimensionless frequency with the network for each case

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