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FORCED VIBRATION ANALYSIS OF ROTATING BLADES SUBJECTED TO LARGE DEFORMATIONS: A NUMERICAL INVESTIGATION WITH VIBRANT

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ABSTRACT

Rotating blades in aircraft engines and wind turbines are critical components subjected to dynamic loads that can cause significant deformations. Accurate analysis of their forced vibration behaviour under large deformations is essential for ensuring performance and structural integrity. This study utilises the Vibration BehaviouR ANalysis Tool (VIBRANT) to investigate complex nonlinearities in blade systems, including large deformation scenarios. Through academic examples, VIBRANT demonstrates its reliability and capability to model nonlinear behaviours, including modal coupling and resonance phenomena. The analysis explores the influence of parameters such as blade geometry and material type, including metallic alloys (e.g., aluminium and titanium) and composites. Detailed frequency response analyses reveal the importance of accurately modelling interactions between material properties and geometric configurations. Comparative results highlight the impact of design choices on natural frequencies, mode shapes, and damping efficiency. VIBRANT's ability to handle large deformations, combined with its flexibility and efficiency, positions it as a powerful tool for engineers and researchers. Its user-friendly structure and robust performance make it suitable for designing and optimising rotating blade systems across various industries while maintaining computational efficiency.

Keywords: Nonlinear vibration, large displacements, nonlinearity, time marching, forced response, fan blades of aircraft engines, wind turbine blades

NOMENCLATURE

Roman letters

b	Width of the beam [m]
E	Young's modulus [Pa]
F_{ex}	Harmonic excitation force [N or kN]
G	Shear modulus [Pa]
h	Height of the beam [m]
$\mathbf{K}$	Stiffness matrix
L	Length of the beam [m]
$\mathbf{M}$	Mass matrix
u_i	Displacement components ($i = 1, 2, 3$)
mag	Magnitude of total displacement
V_f	Foam volume fraction
ρ	Density [kg/m^3]

Greek letters

α	Mass proportional damping coefficient
β	Stiffness proportional damping coefficient
γ	Shear viscoelastic constant
μ	Axial viscoelastic constant
ν	Poisson's ratio
ξ	Damping ratio
ω	Natural frequency [rad/s]

Superscripts and Subscripts

1, 2	Indices for first and second natural frequencies
ex	Excitation force
b	Beam properties
tip	At the free tip of the beam

Acronyms

HBM	Harmonic Balance Method
RHBM	Receptance Harmonic Balance Method
AFTM	Alternating Frequency/Time Method
NLvib	Nonlinear Vibration Tool
VIBRANT	VIbration BehaviouR ANalysis Tool

1. INTRODUCTION

Rotating blades, such as those used in aircraft engines and wind turbines, are critical components subjected to dynamic loads that can lead to significant deformations during operation. Accurate analysis of their forced vibration behaviour, particularly its nonlinear nature, is essential for understanding performance and ensuring structural integrity. However, analysing nonlinear mechanical systems presents a formidable challenge in engineering,

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especially when these systems experience large deflections, contact interactions, or nonlinear material behaviour [1–10]. Among these, the dynamic behaviour of aeroengine components, particularly bladed disc assemblies, represents a critical area due to stringent performance and reliability requirements [11–13]. These components often operate under extreme conditions, necessitating robust modelling tools for their design and modelling [14].

A range of modelling approaches has been developed to investigate the dynamic response of nonlinear mechanical systems. The Harmonic Balance Method (HBM) and time marching techniques are among the most established. HBM, a frequency-domain approach, approximates nonlinear dynamic equations using a series of harmonics, converting time-dependent problems into algebraic equations. This method is particularly efficient for periodic solutions in large-scale systems [15] and has been widely used in applications involving unsteady nonlinear behaviours [16], stochastic analyses under uncertainty [17], and model order reduction for complex geometries [18]. Furthermore, HBM has proven effective for systems involving friction, enabling accurate representation of nonlinearities in vibrating mechanical systems [16, 19–21].

Time marching methods, in contrast, operate in the time domain, offering a step-by-step solution of dynamic equations. Although computationally intensive for larger systems, they are indispensable for capturing intricate nonlinear dynamics, such as those seen in frictional brake systems, with excellent agreement between experimental and theoretical results [22–25]. Innovations in time marching, such as pseudotime stepping and multigrid acceleration, have significantly improved computational efficiency, especially for stiff nonlinear systems [16, 26, 27].

Progress in HBM has also led to specialised techniques tailored for specific applications. The Receptance Harmonic Balance Method (RHBM), for instance, is particularly suited for aeroengine models with nonlinear bearings, achieving enhanced efficiency when combined with time marching solvers [28]. Similarly, the Global Residue Harmonic Balance Method refines solution accuracy by accounting for residual errors in previous iterations, making it highly effective for nonlinear beams [29]. The Alternating Frequency/Time Method further extends HBM's capabilities by addressing complex nonlinearities, such as those in fractional exponential models [30–34].

For comprehensive modelling of large-scale nonlinear systems, the integration of frequency and time domain methods is often required. This combined approach has been highlighted as critical for capturing the full spectrum of dynamic behaviour, as evidenced by studies on turbine blade-disc systems with under-platform dampers [35]. Effective modelling in these cases relies on accurately predicting contact pressure distributions and incorporating zero-harmonic terms in multiharmonic expansions [36]. Techniques such as quasi-linearisation further simplify the analysis by transforming nonlinear differential equations into algebraic counterparts [37].

In recent years, advanced computational tools have been developed to address the unique challenges posed by nonlinear dynamics. Software such as NLvib excels in simulating nonlinearities in structures, particularly in applications relevant to aeroengine components [16]. MANLAB and Mousai, two notable tools, provide powerful capabilities for exploring nonlinear periodic solutions, with the former leveraging continuation techniques and the latter offering a general-purpose harmonic balance solver [38–41]. Proprietary codes like FORSE, developed at Imperial College London, and PERMAS, a commercial tool from Intes, further enhance the ability to model forced responses in nonlinear systems [42–49].

This paper conducts a detailed numerical study using the Vibration BehaviouR ANalysis Tool (VIBRANT), a robust computational platform designed for advanced vibration analysis of mechanical systems. The study showcases VIBRANT's capability to model complex geometric nonlinearities through academic examples that validate its reliability as a modelling tool and an application where its capabilities are demonstrated. A comprehensive frequency response analysis is performed across a range of excitation frequencies and amplitudes, capturing nonlinear behaviours due to large deformations. The results underline the significance of accurately modelling of geometric configurations to predict performance under dynamic loads. Comparative assessments demonstrate the computational performance and efficiency of VIBRANT against a well-established nonlinear vibration modelling tool called NLvib, including peak frequencies in the forced responses and damping characteristics. VIBRANT's ability to handle large deformations and complex vibrational responses, combined with its computational efficiency and flexibility, positions it as a robust tool for engineers and researchers. This study demonstrates how VIBRANT facilitates detailed analysis of rotating blade systems and a wide range of industrial applications.

2. STRUCTURE OF VIBRANT

VIBRANT is a computational platform developed to streamline time marching frequency sweep analyses for generic mechanical systems. By integrating the advanced simulation capabilities of Abaqus with the scripting versatility of Python, VIBRANT automates complex workflows to deliver efficient and precise results. Its primary focus is on solving dynamic problems and accommodating nonlinear phenomena such as contact interactions, localised plasticity, large deformations and energy dissipation due to various sources of damping. The software also supports the incorporation of multiphysical effects. While these features can result in significant computational demands, VIBRANT utilises parallelisation techniques to enhance performance, making it a valuable tool for engineers in aerospace and mechanical disciplines.

2.1 Software Framework

VIBRANT relies on a combination of robust software tools to manage model setup, simulation execution, and data visualisation. Abaqus serves as the core for finite element analysis, with Python scripts organising/steering tasks like automation and data handling. **NumPy** provides numerical computation support, and **Matplotlib** enables effective visual representation of results. Additionally, native Python libraries such as `os`, `math`, and `time` are employed for system-level operations.

To deploy VIBRANT, users must ensure Python and Abaqus are correctly installed and configured. The script should either reside in the directory containing Abaqus files or have its path specified for seamless operation.

2.2 Operational Workflow

The operational workflow of VIBRANT is shown in Figure 1. Each step is represented as a distinct block in a flowchart to enhance visual understanding.

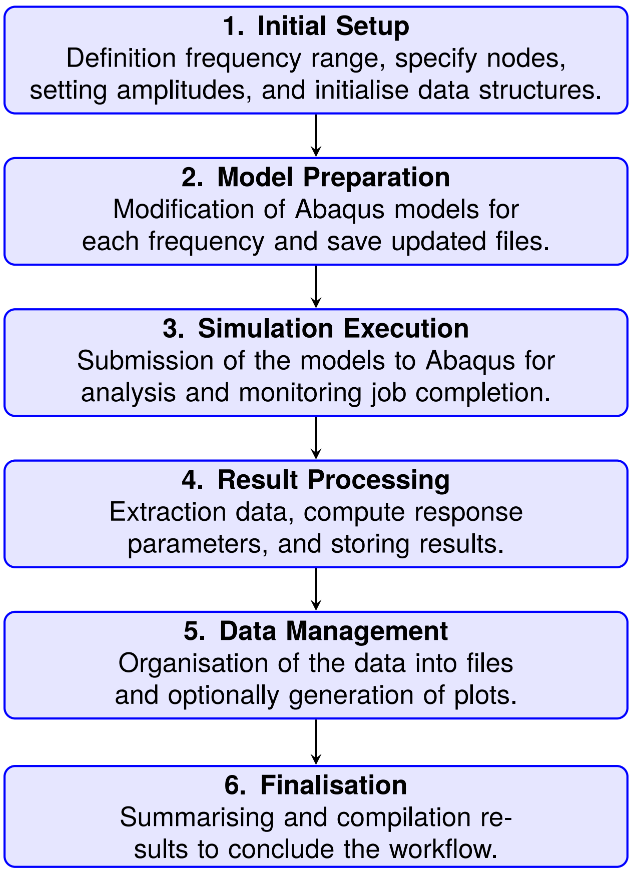

FIGURE 1: FLOWCHART OF VIBRANT'S OPERATIONAL WORKFLOW.

3. CASE STUDIES

3.1 Test Case for Validation with NLvib: Timoshenko Beam Representing DTU Reference Wind Turbine Blade

The first case study examines a Timoshenko beam undergoing large displacements, a scenario chosen to validate VIBRANT's ability to model geometric nonlinearities, which are distributed nonlinearities by nature. To ensure accuracy, simulations are performed using both VIBRANT and NLvib, with results compared to benchmark the capabilities of the two tools in capturing the dynamic response of the beam under significant forcing.

The beam used in this study, inspired by the DTU reference wind turbine blade (86.3 m), is modelled as a Timoshenko beam using averaged properties extracted from prior research [50–54].

The system incorporates Rayleigh damping, where the damping matrix ($\mathbf{C}$) is expressed as a linear combination of the mass matrix ($\mathbf{M}$) and stiffness matrix ($\mathbf{K}$):

$$\mathbf{C} = \alpha\mathbf{M} + \beta\mathbf{K}, \tag{1}$$

where:

- α is the mass proportional damping coefficient,
- β is the stiffness proportional damping coefficient.

The coefficients α and β are determined based on the damping ratio (ξ) at two natural frequencies (ω_1 and ω_2), using the following relationships:

$$\alpha = 2\xi\frac{\omega_1\omega_2}{\omega_1^2 - \omega_2^2}, \tag{2}$$

$$\beta = 2\xi\frac{1}{\omega_1 - \omega_2}. \tag{3}$$

Here, ω_1 and ω_2 represent the primary natural frequencies of interest in the system. This approach ensures that the damping ratio ξ is accurately applied at these frequencies, effectively capturing the dynamic behaviour of the structure. Rayleigh damping is suitable for systems where damping effects are dependent on both mass and stiffness properties, making it versatile for a wide range of engineering applications.

The beam, as illustrated in Figure 2, is discretised into 20 elements with a total of 21 nodes, and it is subjected to harmonic excitation applied at the free end. The simulation explores its nonlinear dynamic response over a range of excitation amplitudes (F_{ex} = 100.0 kN, 250.0 kN, 500.0 kN, and 750.0 kN), focusing on frequency response and displacement patterns. This case study validates VIBRANT by comparing its results against those obtained using NLvib with 10 harmonics in the HBM formulation.

3.2 Case for Demonstration: A Representative Fan Blade Structure of an Aeroengine

This study demonstrates the methodology using a fan blade representative of a high-bypass-ratio aeroengine. The blade geometry, shown in Figure 3, features a slender, pre-twisted and tapered profile designed for optimal aerodynamic performance while ensuring structural integrity under operating conditions. The blade's trailing edge exhibits sharp curvature, influencing the distribution of stresses and displacements. The model is replicated from previously published studies [11, 12]. Figure 4 shows the pressure and suction sides with the pressure load and measurement reading points annotated.

The fan blade rotates at $104 rad/s$, subjecting it to substantial centrifugal body forces along the axis of rotation alongside aero pressures that have a static and dynamic component. These forces generate tensile, bending, and torsional loads throughout the structure, with the highest stress concentrations occurring at the root due to the geometric constraints and the blade's attachment to the hub. In addition to the rotational forces, the blade experiences static and dynamic aerodynamic pressures, which play a critical role in its vibration behaviour. The static pressure represents the steady aerodynamic loading during operation, while the dynamic pressure distribution arises from harmonic oscillations in the airflow [11, 55].

The static pressure loading reflects the steady forces acting on the blade during cruising conditions, simulating the continuous

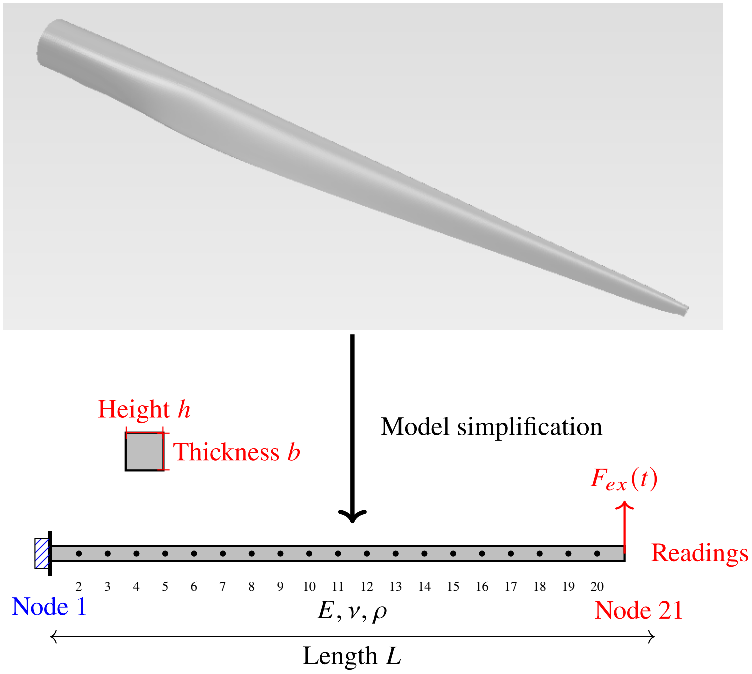

FIGURE 2: THE GEOMETRY OF THE DTU REFERENCE WIND TURBINE BLADE AND THE SCHEMATIC OF THE TIMOSHENKO BEAM WITH EXCITATION AT THE FREE TIP WHERE THE READINGS ARE ALSO TAKEN FROM (NODE 21).

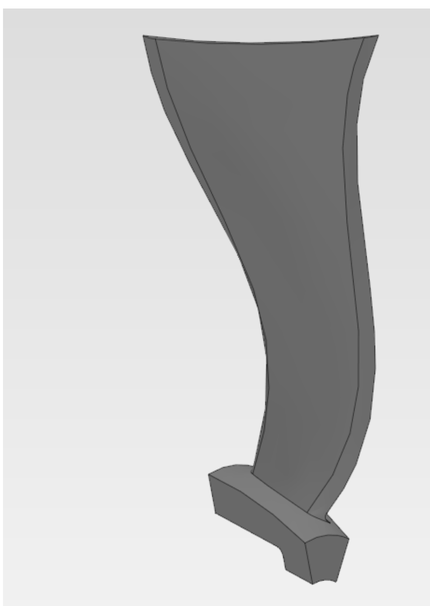

FIGURE 3: THE GEOMETRY OF THE FAN BLADE USED.

aerodynamic load distribution that balances thrust generation and structural demands. On the other hand, the dynamic pressure distribution introduces periodic forcing, which interacts with the blade's natural frequencies and mode shapes. These dynamic pressures can excite resonant vibrations, amplifying displacement amplitudes and potentially accelerating fatigue damage in the structure. This study assumes that the dynamic aero loads are fractions of the static aero loads [11, 55].

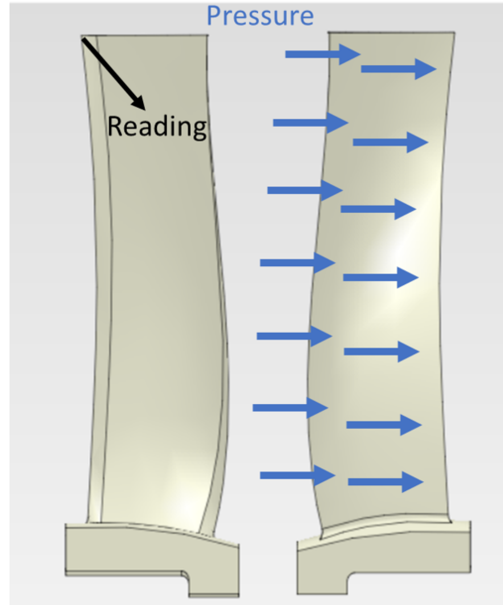

FIGURE 4: SUCTION AND PRESSURE SIDES OF THE FAN BLADE WITH LOADS AND MEASUREMENT POINTS SHOWN.

Two material configurations are considered: a blade made of bulk Ti-6Al-4V (referred to as "solid") and a blade made of Ti-6Al-4V foam with a void volume fraction of 10% ($V_f = 0.1$). The blade made of solid/bulk materials consists of an isotropic material with a density (ρ) of 4600 kg/m^3 and Young's modulus (E) of 1.10×10^{11} Pa, representing high stiffness and strength. For the foam configuration, the density decreases to 4140 kg/m^3, and Young's modulus is reduced to 9.00×10^{10} Pa. The blade made of foam provides weight reduction while maintaining structural rigidity, significantly affecting the blade's dynamic response. This configuration modifies stiffness and damping characteristics, helping to mitigate resonant vibrations and redistribute stresses. The combination of foam and solid material enables an optimised design that balances weight, strength, and dynamic performance [11]. The material properties are summarised in Table 1. Finally, a low level of material damping is introduced using Rayleigh damping as done in the previous case.

TABLE 1: MATERIAL PROPERTIES FOR SOLID AND FOAM CONFIGURATIONS (V_f = 0.1) [11]

Property	**Solid** ($V_f = 0$)	**Foam** ($V_f = 0.1$)
Density (ρ) [kg/m^3]	4600	4140
Young's Modulus (E) [Pa]	1.10×10^{11}	9.00×10^{10}
Poisson's Ratio (ν)	0.31	0.30

The analysis focuses on the blade's displacement amplitudes as well as material damping behaviour under various excitation levels. Thus, the blade is characterised to demonstrate the vibration response of the blade under loads that are judged to be representative of the operational loads. Understanding how the blade responds to dynamic pressures and rotational forces is critical for predicting the blade's operational performance as well as failure mechanisms like fatigue and resonance-induced damage.

The presented results underscore the importance of material selection and aerodynamic loading in ensuring the structural and dynamic integrity of aeroengine components.

4. RESULTS AND DISCUSSION

4.1 Validation with NLvib: Timoshenko Beam Representing DTU Reference Wind Turbine Blade

This section explores the vibration behaviour of a cantilever beam representing the DTU reference wind turbine blade subjected to harmonic excitation. The frequency axis is normalised by dividing by the blade's fundamental natural frequency, while the displacement amplitudes are normalised by the static deflection under the excitation load $F_{\text{ex}} = 100.0\,\text{kN}$. This normalisation scheme permits clearer interpretation and comparison of the results.

Figure 5 shows the normalised displacement amplitudes versus the normalised excitation frequency for varying excitation forces: $F_{\text{ex}} = 100.0\,\text{kN}, 250.0\,\text{kN}, 500.0\,\text{kN}$, and $750.0\,\text{kN}$. The results reveal a consistent nonlinear softening behaviour, evidenced by a slight leftward shift in the resonance frequency as the excitation force increases. This shift, characteristic of nonlinear dynamic systems, arises from geometric nonlinearity due to the cantilever configuration and large displacement effects. The displacement amplitudes also rise at higher excitation forces. However, the displacement amplitudes climb slightly less than the forces are increased, reflecting increased energy dissipation and damping in the system.

The agreement between VIBRANT and NLvib predictions is within $2 - 3\%$ across all cases, demonstrating the validity of the VIBRANT methodology. While minor discrepancies are observed at higher excitation amplitudes, these are attributed to numerical differences in implementation and do not affect the overall trends or main dynamic behaviours. Both methods effectively capture the nonlinear softening, resonance peaks, and amplitude responses.

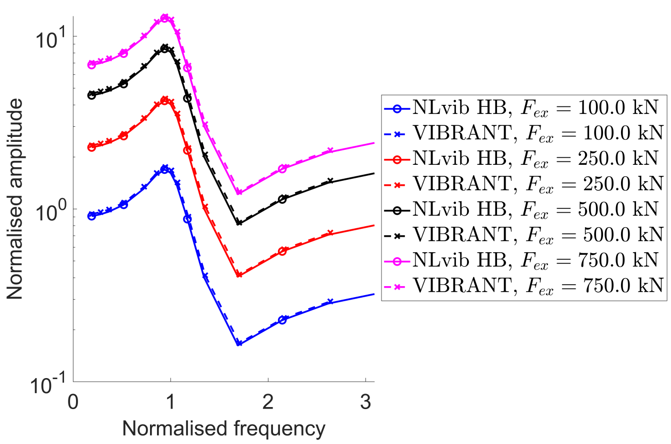

FIGURE 5: NORMALISED DISPLACEMENT AMPLITUDE VERSUS NORMALISED EXCITATION FREQUENCY FOR VARIOUS EXCITATION FORCES. RESULTS FROM VIBRANT AND NLVIB ARE COMPARED.

The findings reinforce the significance of nonlinear effects in the vibration analysis of wind turbine blades. Neglecting these effects could lead to inaccurate predictions of resonance behaviour, resulting in potential structural inefficiencies or failures. By incorporating both geometric nonlinearities, the current approach provides a reliable framework for assessing dynamic performance under realistic aerodynamic and structural loading conditions. This is particularly valuable in designing wind turbine blades for durability, efficiency, and operational safety.

From a computational perspective, despite VIBRANT's higher computational intensity, its parallelisation capabilities significantly reduce computation time. On a system with the following specifications:

- **Processor:** Intel(R) Core(TM) i9-8950HK CPU @ 2.90GHz (12 cores),
- **Memory:** 32 GB RAM,
- **Operating System:** Windows 11 Pro 64-bit,
- **Graphics Card:** NVIDIA GeForce GTX 1080,

VIBRANT completes the simulation in approximately 3500 seconds using a six-core parallel Abaqus configuration, compared to 2500 seconds for NLvib on the same machine. This demonstrated that even though the methodology that VIBRANT employs to solve the vibration problems is computationally intensive, VIBRANT is still relatively efficient in managing computational time while maintaining accuracy, making it suitable for large-scale, complex simulations.

The outcomes highlight the capability of VIBRANT to simulate complex dynamic behaviours encountered in applications such as wind turbine blades. This comparison underscores VIBRANT's computational robustness and its potential for addressing real-world mechanical and aerospace engineering challenges.

4.2 Case for Demonstration: A Representative Fan Blade Structure of an Aeroengine

The dynamic response of a representative fan blade structure subjected to harmonic aerodynamic excitation is investigated. The results focus on the normalised displacement amplitudes (u_1, u_2, u_3, and the overall magnitude mag) and the dissipated power (material damping), highlighting the effects of excitation amplitude, material properties, and nonlinear behaviours on the system's vibrational response.

4.2.1 Displacement Responses.

Figures 6, 7, 8, and 9 illustrate the normalised displacement amplitudes for the components u_1, u_2, u_3, and the magnitude (mag), respectively. These plots reveal distinct trends in the response of foam and blade made of solid/bulk materials materials under varying excitation levels.

For the blade made of solid/bulk materials, resonance peaks are sharp and more obvious, with higher amplitudes compared to the blade made of foam. The blade made of foam exhibits broader peaks with reduced amplitudes, indicative of its superior damping efficiency. A significant observation is that the resonance peaks lean leftwards as the excitation amplitude increases, particularly for the blade made of solid/bulk materials. This leftward frequency shift suggests the presence of softening nonlinearity, where the system's natural frequency decreases as the amplitude of vibration increases. Such nonlinear behaviour becomes more pronounced at higher excitation levels, as evident from the increasing deviation of the resonance peaks.

The excitation amplitude also influences the rise of peak amplitudes. At lower excitation levels ($Exc0.1$), both blades made of foam and solid/bulk materials show relatively small amplitudes. As the excitation increases to $Exc0.3$ and $Exc0.5$, the peak amplitudes scale differently for foam and solid materials. The blade made of solid/bulk materials demonstrates a proportional rise in amplitude, indicating a largely linear response at resonance. In contrast, the blade made of foam exhibits a disproportional rise, which hints at nonlinear effects and enhanced damping contributions that suppress the amplitude growth.

The trends observed across the displacement components u_1, u_2, and u_3 are consistent, with the blade made of solid/bulk materials achieving higher amplitudes at resonance compared to the blade made of foam. The magnitude response (mag) further consolidates these observations, showing an aggregate view of the vibrational behaviour. These findings highlight the interplay between excitation amplitude, material properties, and nonlinear effects, with solid materials exhibiting stiffer, more linear responses and foam materials demonstrating broader, nonlinear damping characteristics.

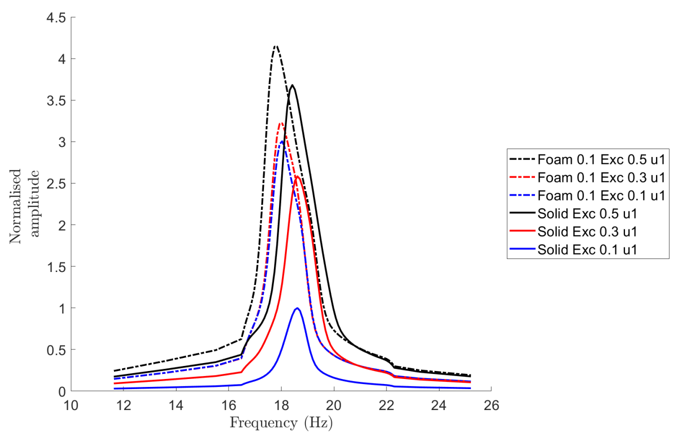

FIGURE 6: NORMALISED AMPLITUDE OF u_1 VS FREQUENCY.

4.2.2 Material Damping Responses. The material damping responses, represented by the normalised dissipated power, are presented in Figure 10. The blade made of solid/bulk materials exhibits sharp and pronounced peaks in dissipated power, particularly at higher excitation levels ($Exc0.3$ and $Exc0.5$). The peaks shift leftwards with increasing excitation, consistent with the observed softening nonlinearity in the displacement responses.

The blade made of foam demonstrates broader dissipated power peaks with lower values compared to the blade made of solid/bulk materials, reflecting its superior damping efficiency. Similar to the displacement response, the dissipated power for the blade made of solid/bulk materials scales proportionally with increasing excitation amplitude, suggesting a linear relationship between input excitation and energy dissipation. In contrast, the blade made of foam displays a disproportional rise, hinting at nonlinear damping mechanisms that influence the energy dissipation

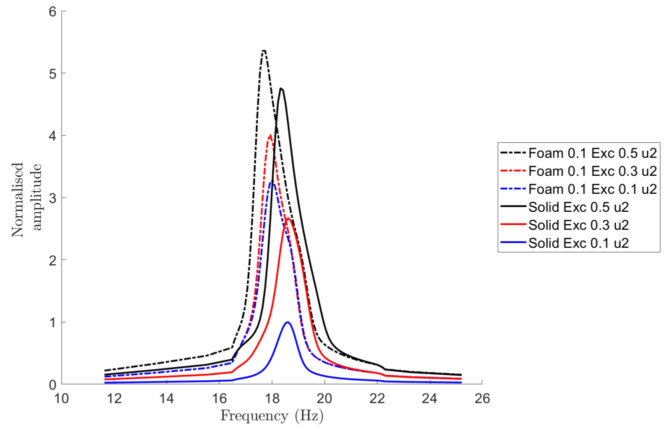

FIGURE 7: NORMALISED AMPLITUDE OF u_2 VS FREQUENCY.

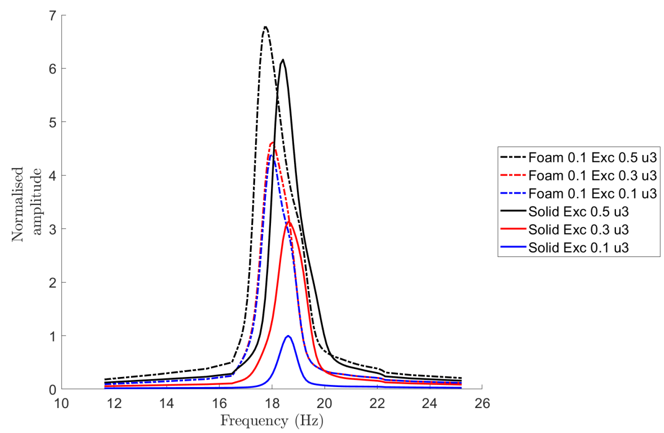

FIGURE 8: NORMALISED AMPLITUDE OF u_3 VS FREQUENCY.

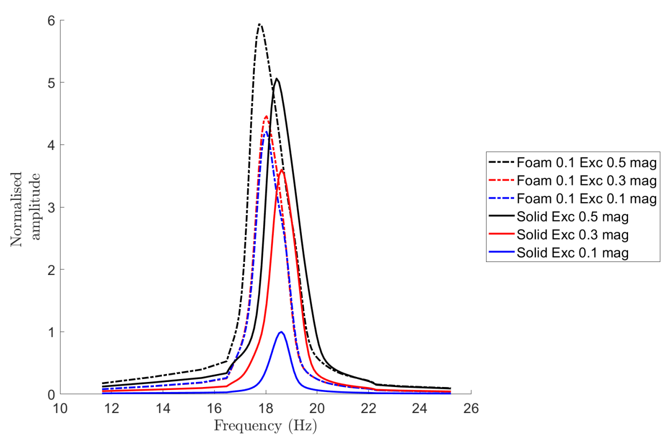

FIGURE 9: NORMALISED MAGNITUDE AMPLITUDE VS FREQUENCY.

process.

The excitation amplitude governs the overall trend in power dissipation, with both materials showing higher dissipation at elevated excitation levels. However, the material properties remain the dominant factor dictating the response shape. Solid materials localise vibrational energy and dissipate it in sharp peaks, whereas foam materials distribute the energy dissipation over a broader frequency range.

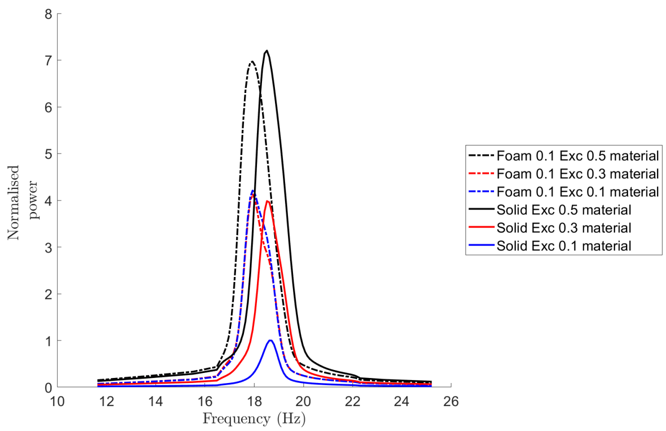

FIGURE 10: NORMALISED MATERIAL DAMPING (DISSIPATED POWER) VS FREQUENCY.

4.2.3 Discussion. The results highlight key differences between solid and blade made of foam materials, particularly in their response to increasing excitation amplitude and nonlinear effects. The leftward shift of resonance peaks across all excitation levels indicates the presence of softening nonlinearity, where the system's stiffness reduces as the vibration amplitude increases. This behaviour is more pronounced for the blade made of solid/bulk materials due to its lower damping capacity.

The proportional rise in displacement amplitudes and power dissipation for the blade made of solid/bulk materials suggests a predominantly linear response at resonance. In contrast, the blade made of foam exhibits disproportional rises, which can be attributed to nonlinear damping mechanisms and energy dissipation over a broader frequency range.

Overall, solid materials, with their sharp resonance peaks and concentrated dissipated power, are suitable for applications requiring precise vibrational control and minimal energy loss. Foam materials, on the other hand, provide enhanced damping performance, characterised by broader resonance peaks, lower amplitudes, and distributed power dissipation, making them ideal for vibration suppression and energy absorption. The findings further illustrate how excitation amplitude and material properties collectively influence the linear and nonlinear dynamic behaviour of fan blade structures.

5. CONCLUSIONS

This study demonstrates the effectiveness and versatility of the VIBRANT software in analysing and predicting the dynamic/vibrational behaviour of complex structures. Validation of VIBRANT against the NLvib framework, using a benchmark cantilever beam case (DTU Reference Wind turbine Blade), confirms its accuracy in modelling nonlinear vibration responses under harmonic excitation. The excellent agreement between the two methods, across a range of excitation levels and system parameters, underscores VIBRANT's robustness and reliability as an advanced computational tool for dynamic analysis.

The application of VIBRANT to a representative fan blade structure showcases its capabilities in handling realistic aero-engine components subjected to operational forces. By incorporating centrifugal body forces due to rotation at $104 rad/s$, as well as static and dynamic aerodynamic pressure distributions, the software provides a comprehensive assessment of the blade's dynamic behaviour. The study highlights the importance of capturing the interaction between aerodynamic loading and structural dynamics, particularly in the presence of lightweight materials such as foam configurations. Results demonstrate that VIBRANT accurately predicts displacement amplitudes and identifies the potential for resonance-induced damage, contributing to a better understanding of failure mechanisms and structural optimisation strategies.

Future work will expand VIBRANT's capabilities to address structural health monitoring, crack propagation, and machine learning-based analysis. When a crack initiates, the dynamic response of the structure undergoes significant changes due to localised stiffness reductions, frictional contacts on the crack surfaces, and altered damping characteristics. VIBRANT will be enhanced to model these effects, including the evolution of crack propagation, surface contact interactions, and associated changes in dynamic stiffness and energy dissipation. Such modelling will provide insights into the mechanisms governing fatigue failure and the structural integrity of rotating components.

Machine learning techniques will be integrated to identify uncertainties and predict crack initiation and propagation trends from large vibration datasets. By correlating dynamic response characteristics with crack growth and fatigue behaviour, machine learning algorithms can assist in identifying early warning signs of structural degradation and optimising maintenance schedules. This integration will also enable uncertainty quantification, improving confidence in the predictions for complex nonlinear systems under varying operational conditions.

The initiative to combine VIBRANT with structural health monitoring, crack propagation modelling, and machine learning opens new avenues for predictive maintenance and failure analysis. These advancements will make VIBRANT a comprehensive platform capable of analysing, monitoring, and predicting the vibrational behaviour of critical structures such as aeroengine components and wind turbine blades, ensuring reliable and optimised performance throughout their operational life.

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