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ARTICLE

Finite Element Analysis of Resonance Frequencies for Impellers of Centrifugal Compressors Made of Metal Matrix Composites

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ABSTRACT: Vibrations occurring in the impellers of centrifugal compressors are among the primary factors that influence the reliability and service life of main gas pumping station units. A promising approach to improving the dynamic performance of centrifugal compressors is the use of modern metal-matrix composite materials for impellers. This article presents a prediction of the dynamic behavior of an impeller under actual operating conditions with the aim of eliminating resonance vibrations. Detailed geometric and finite-element modeling enabled an investigation of the prestressed state caused by centrifugal forces on the natural frequency spectrum of the impeller. To investigate the influence of the material's physical properties, calculations were performed for boron-aluminum composites with varying boron carbide particle content. For boron-aluminum impellers, the effect of centrifugal forces on the natural frequencies did not exceed 2%–3%. However, increasing the boron carbide content from 1.5% to 7.5% results in an increase in the first 10 natural frequencies of up to 6.8%. Campbell diagrams were constructed for the fundamental and harmonic frequencies of the disturbing forces, taking into account the interaction of the blades of the compressor's stator and rotor components.

KEYWORDS: Boron-aluminum composite; finite element method; centrifugal compressor; natural frequency

1 Introduction

Centrifugal compressors are widely used as turbochargers in internal combustion engines, in chemical engineering, and in industrial gas compression systems [1]. They are critical components of gas pumping stations on main gas pipelines, which are of immense importance to the modern energy sector.

Vibrations in the impellers of centrifugal compressors are one of the key factors determining the reliability and durability of main gas pumping station units. When operating under cyclic loads, variable flow and pressure conditions, and the effects of gas-dynamic disturbances, impellers experience a complex stress-strain state. Elevated vibration levels can lead to the development of fatigue cracks, reduced compressor efficiency, and equipment failure, resulting in significant technical and economic losses.

Modern methods for analyzing the dynamics of rotating systems allow for a deeper investigation of the interaction between dynamic processes and centrifugal forces, which determine the spectrum and amplitudes of impeller vibrations. Nevertheless, for centrifugal compressors in pumping stations operating across a wide range of operating conditions, the issues of verifying computational models and identifying critical vibration modes remain relevant. The use of modern metal-matrix composite materials opens up new possibilities for improving the technical performance of compressors, which has attracted significant attention from researchers.

Vibration and resonance phenomena in turbomachinery have been extensively investigated, particularly focusing on the interaction between aerodynamic excitations and structural dynamics. Nordwall et al. [2] examined resonant stresses induced by inlet guide vane wakes, demonstrating the importance of unsteady flow effects. Similarly, Walton and Tan [3] analyzed the forced response generated by impeller–diffuser interactions, confirming that aerodynamic coupling is a major source of excitation. These findings are further supported by the work of Mayorca et al. [4], who developed reduced-order models for predicting aero elastic behavior, and Beirow et al. [5], who investigated vibration mitigation through intentional mistuning of bladed disks. Experimental and numerical investigations by Zhao et al. [6] highlighted that even small deviations from cyclic symmetry can significantly influence vibration amplitudes and resonance conditions.

Failure mechanisms of centrifugal impellers have also been widely studied. Radgolchin and Anbarsooz [7] identified fatigue as the primary failure mode in compressor impellers, often driven by resonance and cyclic stresses. Complementary studies by McKay and Anbalagan [8] demonstrated that material selection and operating conditions, such as erosive flow environments, play a critical role in structural degradation.

To address the limitations of conventional materials, recent research has increasingly focused on advanced materials, particularly metal matrix composites (MMCs). These materials offer superior specific stiffness, strength, and thermal resistance compared to traditional alloys [9,10]. Recent developments in high-entropy alloy-reinforced composites have further expanded their potential applications due to enhanced mechanical and tribological properties [11,12].

From a vibration analysis perspective, several studies have focused on composite and functionally graded materials. Oh and Yoo [13] investigated rotating pretwisted blades made of functionally graded materials, showing that material distribution significantly affects natural frequencies. Pan et al. [14] and Roy et al. [15] extended this research to hybrid and metal matrix composite blades, demonstrating enhanced dynamic stability. More recently, Zhang et al. [16] analyzed nonlinear vibrations of graphene-reinforced composite blades under thermal gradients, highlighting the importance of advanced material design in controlling vibration behavior.

Despite these advances, the application of metal matrix composites in centrifugal compressor impellers remains insufficiently explored, particularly in relation to resonance behavior under real operating conditions. Most existing studies focus either on aerodynamic performance or structural optimization, without fully integrating material effects into dynamic analyses. Moreover, the combined influence of centrifugal prestress, excitation frequencies, and material properties on resonance characteristics requires further investigation.

Most studies have employed the finite element method (FEM). This method is successfully used in mechanical engineering and civil engineering for the strength, dynamic, and thermal analysis of complex-shaped components. FEM allows for the prediction of stress states, vibrations, and the stability of structural elements during the design phase. In civil engineering, FEM is widely used to analyze buildings, bridges, and dams under various loads, including seismic effects. In the aerospace industry, the method helps optimize aircraft designs by reducing mass while maintaining strength characteristics. Thanks to the development of powerful software packages, FEM has become the dominant computational tool in all knowledge-intensive sectors of modern industry.

Therefore, the present study aims to perform a comprehensive numerical analysis of the resonance behavior of centrifugal compressor impellers made of metal matrix composites. Using finite element modeling and modal analysis, the influence of material composition and centrifugal forces on the natural frequency spectrum is evaluated. Campbell diagrams are employed to identify critical resonance regions and define safe operating conditions. This approach contributes to improving the reliability, durability, and performance of turbomachinery systems through the integration of advanced materials and dynamic analysis techniques.

The novelty of this study lies in the integrated and application-oriented numerical framework developed for evaluating the dynamic behavior of centrifugal compressor impellers made from boron–aluminum metal matrix composites. Unlike conventional analyses that often neglect operational preload conditions, this work explicitly incorporates centrifugal prestressing into a fully three-dimensional finite element model, enabling a more realistic prediction of natural frequencies and mode shapes under actual working conditions. This approach allows for a more accurate identification of resonance risks in high-speed rotating systems.

A further original contribution is the systematic investigation of the influence of boron carbide (B_4C) volume fraction on dynamic performance. By correlating reinforcement content with changes in elastic properties and vibrational response, the study demonstrates a clear pathway for tailoring material composition to shift critical resonance zones beyond typical operating ranges. This material–dynamic coupling is rarely addressed in such depth for centrifugal compressor applications.

Additionally, the use of detailed Campbell diagram analysis to distinguish between different excitation sources—such as rotor–stator interactions and imbalance effects—provides new insights into the dominant mechanisms governing resonance. The finding that higher-order harmonic excitations are more critical than imbalance-induced vibrations add practical value for design optimization.

2 Mathematical Formulation of the Problem

2.1 Governing Equations of Motion

The study of dynamic properties of MMC disks is based on general equations of motion for an isotropic elastic medium. In the absence of volume forces, the complete system of equations of motion in a rectangular Cartesian coordinate system reduces to three solving equations for displacements [17]:

$$\begin{aligned}
\mu \nabla^2 u + (\lambda + \mu) \frac{\partial \theta}{\partial x} &= \rho \frac{\partial^2 u}{\partial t^2} \\
\mu \nabla^2 v + (\lambda + \mu) \frac{\partial \theta}{\partial x} &= \rho \frac{\partial^2 v}{\partial t^2} \\
\mu \nabla^2 w + (\lambda + \mu) \frac{\partial \theta}{\partial x} &= \rho \frac{\partial^2 w}{\partial t^2}
\end{aligned} \tag{1}$$

where: u, v, w-are the components of the displacement vector; λ and μ are the Lame elastic constants; ∇^2 is the Laplace operator; θ is the volumetric strain; and ρ is the material density.

For the analysis of the natural frequencies and modes of a rigid body, the displacements during the principal vibrations are represented as:

$$\begin{aligned}
u &= U(x, y, z) \cdot \sin(\omega t) \\
v &= V(x, y, z) \cdot \sin(\omega t) \\
w &= W(x, y, z) \cdot \sin(\omega t)
\end{aligned} \tag{2}$$

Substituting the representation of displacements Eq. (2) into the equations of motion Eq. (1) it is obtained a system of homogeneous equations with respect to the eigenfunction forms of vibrations.

$$\mu\nabla^2 U + (\lambda + \mu)\frac{\partial}{\partial x}\left(\frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} + \frac{\partial W}{\partial z}\right) + \rho\omega^2 U = 0$$
$$\mu\nabla^2 V + (\lambda + \mu)\frac{\partial}{\partial y}\left(\frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} + \frac{\partial W}{\partial z}\right) + \rho\omega^2 V = 0$$
$$\mu\nabla^2 W + (\lambda + \mu)\frac{\partial}{\partial z}\left(\frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} + \frac{\partial W}{\partial z}\right) + \rho\omega^2 W = 0 \qquad (3)$$

here: U, V, W-are shape functions that must satisfy the boundary conditions on the body's surface. The eigenfrequencies are determined from the existence conditions for non-trivial solutions of system Eq. (3) under the given boundary conditions.

To solve eigenvalue problems in mathematical physics, various formulations are employed, and many methods that have become classical have been developed. Formulations in the form of boundary value problems for partial differential equations, as well as variational formulations, are widely used. Recently, the methods utilizing the capabilities of neural networks have been actively developed. Examples of the use of machine learning in problems of deformable body mechanics can be found in [18,19]. Physically informed models [19] are particularly effective, including for nonlinear problems.

2.2 Finite Element Formulation of the Eigenvalue Problem

Solving eigenvalue problems in the three-dimensional formulation for bodies of complex shape is possible only using approximate methods. In this work, the finite element method is used to find the natural frequencies and vibration modes of centrifugal compressor discs. The equations of motion (1) are written in discrete form:

$$[M]\frac{d^2}{dt^2}\left\{\vec{U}\right\} + [K]\left\{\vec{U}\right\} = 0 \qquad (4)$$

where $\left\{\vec{U}\right\}$ is the displacement vector at the nodes of the finite element model, $[M]$ is the global mass matrix of the entire finite element model, and $[K]$ is the global stiffness matrix, which may account for the influence of the prestressed state.

For free harmonic vibrations, the displacement vector is expressed as:

$$\left\{\vec{U}\right\} = \left\{\vec{\Phi}_i\right\} \cdot \sin(\omega_i t) \qquad (5)$$

where: $\left\{\vec{\Phi}_i\right\}$ is the vector that determines the waveform at the i-th natural frequency. For harmonic oscillations, Eqs. (4) and (5) define the relationship between the frequencies and the waveforms:

$$\left([K] - \omega_i^2 [M]\right)\left\{\vec{\Phi}_i\right\} = 0 \qquad (6)$$

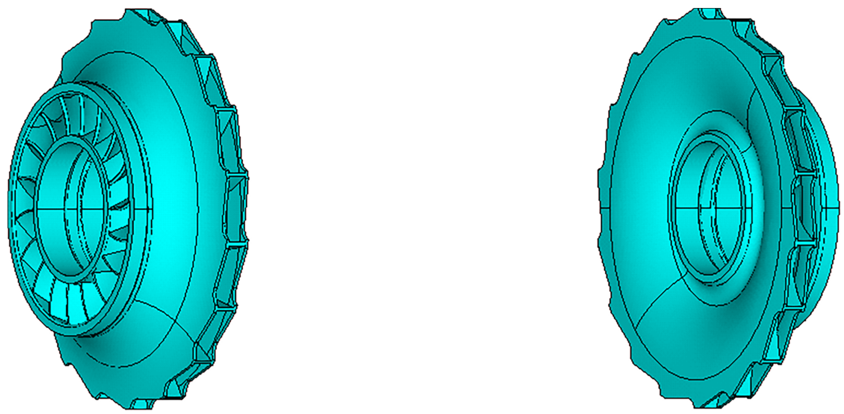
The condition for the existence of nonzero solutions to Eq. (6) leads to the problem of finding the eigenvalues and eigenvectors of the symmetric matrix:

$$\det\left([K] - \omega_i^2 [M]\right) = 0 \qquad (7)$$

To solve this problem, the Block Lanczos method, implemented in ANSYS software package, is used.

3 Modal Analysis Using the Finite Element Method

A numerical study of dynamic properties of impellers made of metal-matrix composites was conducted using as example a typical centrifugal compressor designed for main gas pipelines. The 3D geometric model of the impeller is shown in Fig. 1.



(a) (b)

Figure 1: 3D model of impeller: (**a**)–view from the cover disc side, (**b**)–view from the main disc side.

The impeller main dimensions are as follows: inner bore radius–0.15 m, outer radius–0.45 m, number of working blades–18.

To perform the modal analysis in ANSYS software, a finite element model was created using SOLID 98 finite element. This tetrahedral element is defined by ten nodes, each of which has up to six degrees of freedom. In modal analysis, three components of displacement vector at each node are used. Within the element, displacements are approximated by quadratic functions. The complete finite element model consists of 47,692 elements and includes 90,582 nodes. A general view of the finite-element model is shown in Fig. 2, as seen from the side of the cover disc and the main disc.

(a) (b)

Figure 2: FE model of the impeller: (**a**)–view from the cover disc side, (**b**)–view from the main disc side.

To verify the consistency of the calculation results, a control analysis was performed using a significantly more detailed mesh: 215,192 elements and 364,884 nodes. This refinement of the mesh resulted in frequency changes of no more than 1%.

The Jacobian Ratio, Aspect Ratio, and Maximum Angle metrics were used to verify the quality of the finite element mesh. All the 47,692 elements successfully passed the verification based on these metrics. The relative number of elements with warnings based on these metrics was as follows: Jacobian Ratio–0.42%, Aspect Ratio–0.02%, and Maximum Angle–0.09%. This is an excellent result for technical calculations of natural frequencies.

Under operating conditions, the impeller rotates at speeds of up to 5200 rpm. The stresses caused by centrifugal forces affect the natural frequencies and vibration modes of the impeller. To account for this effect, a static analysis of the prestressed state of the impeller was performed. A modal analysis was performed in two variants: with and without consideration of centrifugal forces. The calculations were performed by extracting the first 10 natural frequencies, with the mode shapes normalized to unity. For all analyses, a rigidly fixed boundary condition was assumed at the inner bore surface of the impeller.

4 Properties of Materials

Particle-reinforced metal matrix composites (MMCs) possess a number of excellent properties, such as high specific modulus of elasticity, increased strength, stable behavior at elevated temperatures, and greater wear resistance compared to traditional alloys. The most commonly used reinforcing particles are Silicon Carbide, Aluminum Oxide and Nitride, Zirconium Dioxide, and Boron Carbide. Aluminum and its various alloys are used as the matrix due to their low density and good processing properties. This study investigates the possibility of using boron-aluminum composites [20] for centrifugal compressor discs taking into account its dynamic characteristics. The matrix material used is LM6 aluminum alloy, which has a high elongation ratio (5%) and low density (2700 kg/m^3). The aluminum is reinforced with boron carbide (B_4C) particles, which have a density (2500 kg/m^3) lower than that of aluminum. The average particle size was 35–40 μm. The composite preparation process involves melting the aluminum at 700°C and adding boron carbide powder while stirring with a vortex mixer.

The authors of paper [20] conducted microscopic studies of the structure of the boron-aluminum composite. It was found that the alloy exhibited a uniform distribution of intermetallic particles and regions with fine dendrites. It was noted that boron carbide is uniformly distributed in the aluminum matrix and can be considered a homogeneous isotropic material. The percentage of boron carbide has the primary influence on the elastic properties. These results are utilized in the present study.

To analyze the dynamic properties of the discs, three variants of a boron-aluminum composite were examined—with filler contents of 1.5, 4.5 and 7.5% B_4C. The physical properties required for the calculations are listed in Table 1. The elastic modulus and shear modulus are based on experimental tests [20], while Poisson's ratio was calculated theoretically assuming the material is isotropic. The density of the composite was determined using the mixture rule for each filler contents.

Table 1: Properties of composites.

Contents of B_4C	Young's Modulus (GPa)	Shear Modulus (GPa)	Poisson's Ratio	Density (kg/m^3)
1.5%	72.5	27.1	0.338	2697
4.5%	80.6	30.6	0.325	2691
7.5%	83.2	31.6	0.316	2685

When studying mechanical vibrations and waves in solid deformable bodies, the relationship between a material's density and elastic modulus is of significant importance. For boron-aluminum composites, increasing the boron carbide content leads to an increase in the elastic modulus while having almost no effect on the composite's density.

5 Results of the Modal Analysis

When configuring the options for performing modal analysis, the frequency range for calculating eigenvalues using the Lanczos block method was set from 10 to 10,000 Hz. The results indicated a relatively dense natural frequency spectrum, with the first 10 natural frequencies all remaining below 2500 Hz. Table 2 presents the natural frequency values for three types of boron-aluminum composites with different boron carbide contents.

Table 2: Natural frequencies of the impeller (Hz).

Frequency Number	Material 1, 1.5% B_4C		Material 2, 4.5% B_4C		Material 3, 7.5% B_4C	
	Neglecting Centrifugal Forces	Accounting for Centrifugal Forces	Neglecting Centrifugal Forces	Accounting for Centrifugal Forces	Neglecting Centrifugal Forces	Accounting for Centrifugal Forces
1	882.94	890.98	930.89	938.51	945.29	952.82
2	1003.1	1011.5	1055.5	1062.8	1074.9	1082.8
3	1064.4	1077.4	1118.7	1131.0	1140.9	1153.0
4	1170.7	1177.4	1233.0	1239.4	1253.7	1260.0
5	1382.9	1400.8	1451.6	1468.8	1483.0	1499.8
6	1734.6	1759.1	1820.3	1843.4	1860.4	1883.4
7	2001.1	2005.6	2104.4	2108.6	2144.4	2148.6
8	2014.5	2048.5	2115.2	2147.2	2160.0	2192.1
9	2165.5	2240.4	2280.3	2350.8	2319.5	2392.5
10	2182.7	2243.3	2303.2	2370.9	2337.9	2395.0

Analysis of the natural vibration modes revealed the presence of multiple eigenvalues. The proximity of the natural frequency values is attributed to the rotational symmetry of the centrifugal compressor impeller. Figs. 3–12 illustrate the first ten vibration modes of an impeller manufactured from an aluminum–boron carbide composite containing 1.5% B_4C. These mode shapes were obtained through modal analysis, considering the stress state induced by centrifugal forces during operation.

Analysis of the components of displacement vector for various modes of vibration reveals the qualitative characteristics of the vibrations at different natural frequencies. Thus, at the first natural frequency (Fig. 3), the impeller undergoes torsional vibrations with predominant displacements in the circumferential direction and no nodal lines. At the second natural frequency (Fig. 4) and the third natural frequency (Fig. 5), the impeller undergoes bending vibrations with one and two nodal diameters, respectively. The dominant displacements during bending vibrations are in the axial direction. The fourth natural frequency corresponds to an umbrella-shaped bending mode (Fig. 6) without nodal circles.

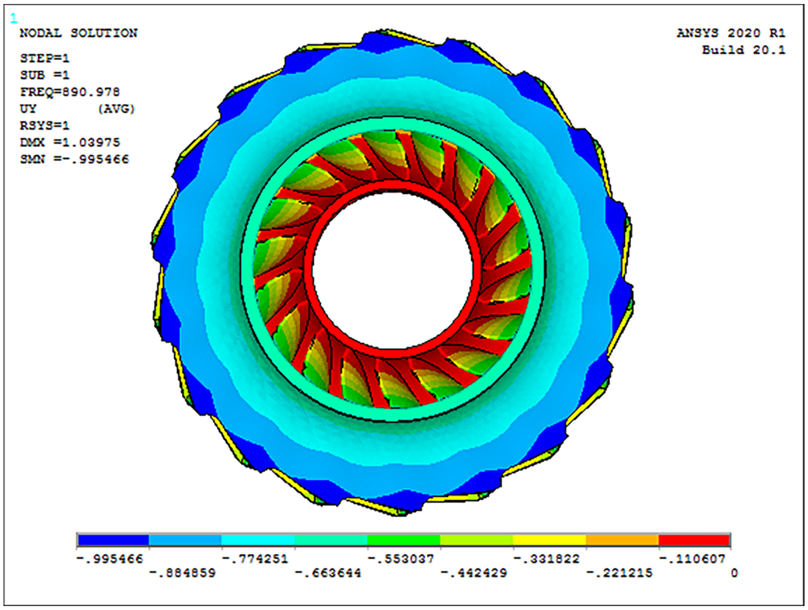


Figure 3: First mode of oscillation.

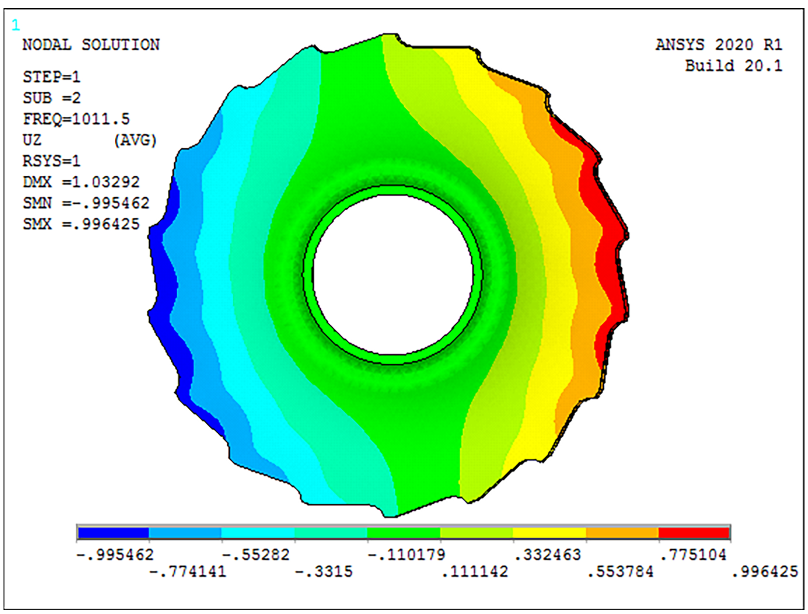


Figure 4: Second mode of oscillation.

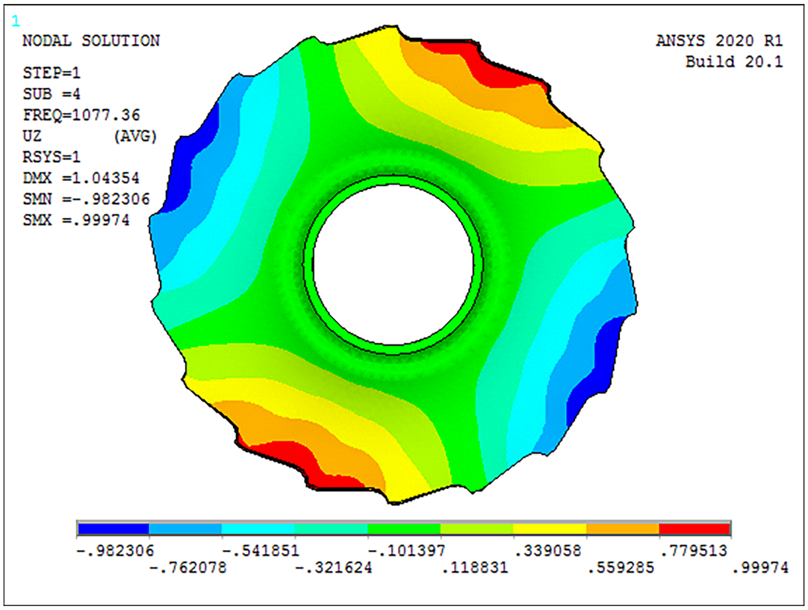


Figure 5: Third mode of oscillation.

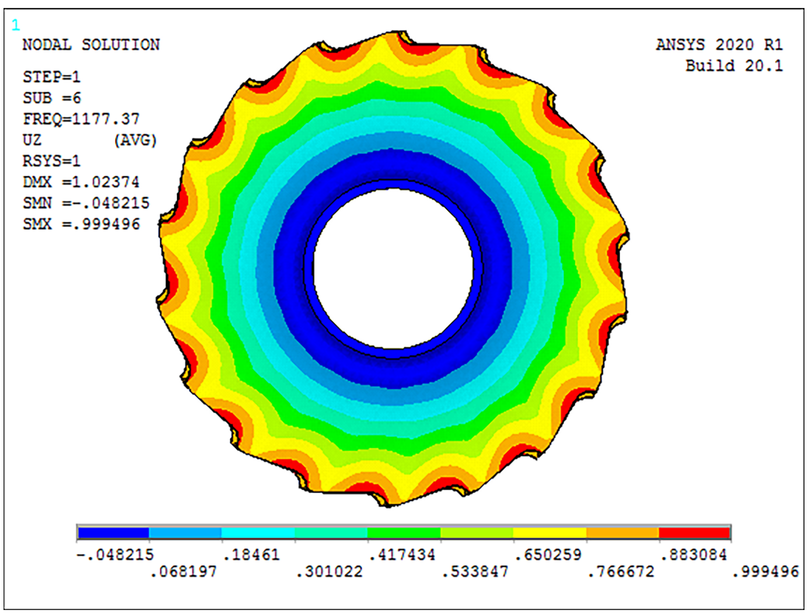


Figure 6: Fourth mode of oscillation.

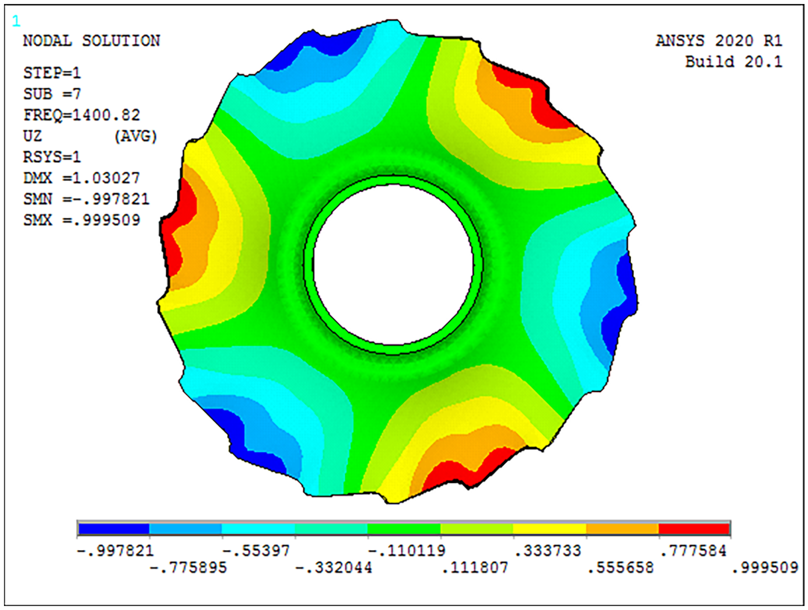


Figure 7: Fifth mode of oscillation.

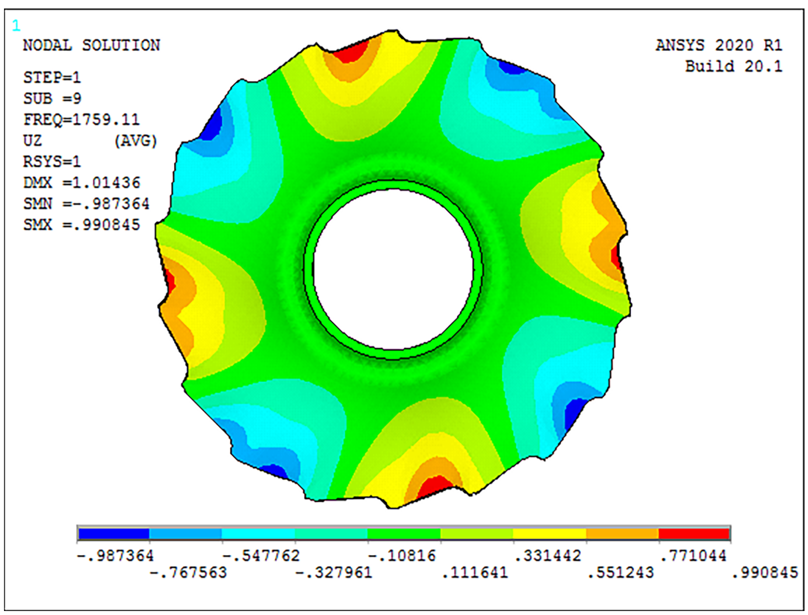


Figure 8: Sixth mode of oscillation.

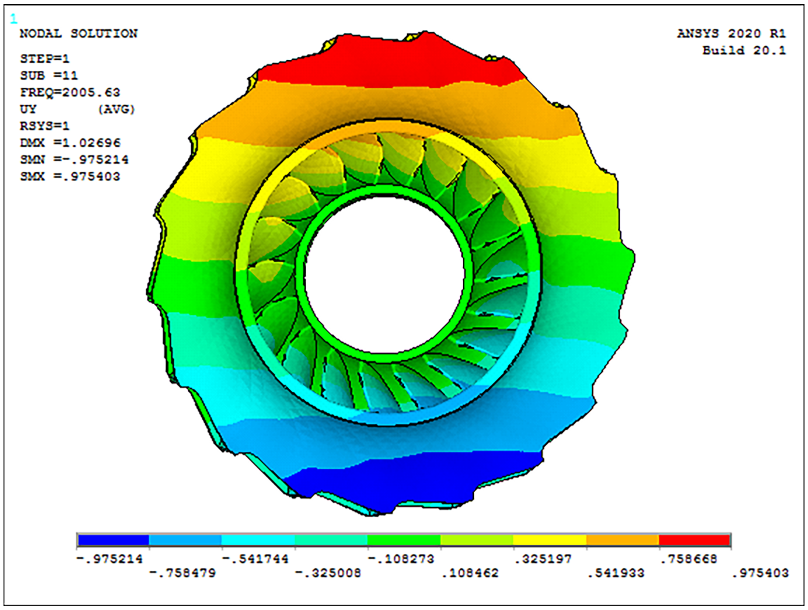


Figure 9: Seventh mode of oscillation.

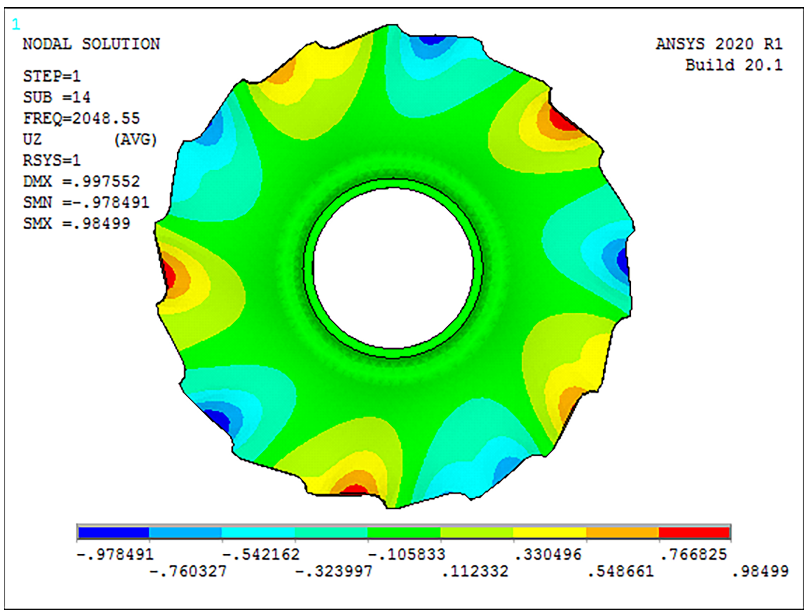


Figure 10: Eighth mode of oscillation.

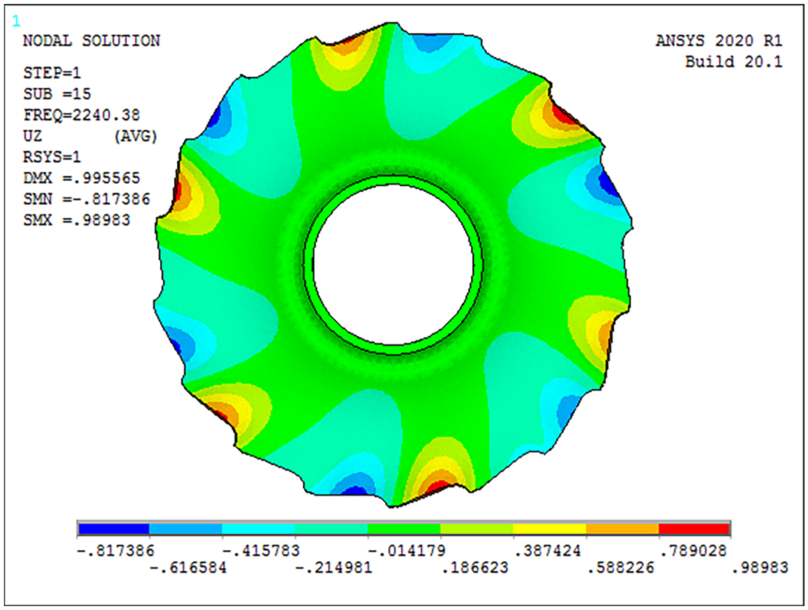


Figure 11: Ninth mode of oscillation.

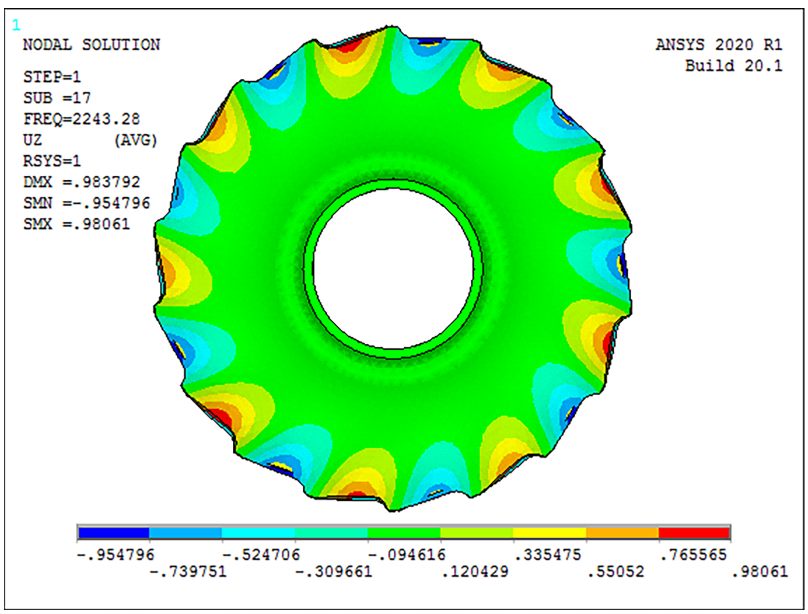


Figure 12: Tenth mode of oscillation.

Figs. 7 and 8 show the fifth and sixth natural modes of the impeller. The distribution of axial displacements on the surface of the main disk clearly demonstrates the presence of three and four nodal diameters in these modes. At the eighth natural frequency, the impeller undergoes predominantly torsional vibrations (Fig. 9). Unlike the first mode of torsional vibration (Fig. 3), the distribution of circumferential displacements shows the appearance of a single diametrical nodal line along which these displacements are absent. The last three modes of vibration (Figs. 10–12) are bending modes. The distribution patterns of axial displacements on the surface of the main disc show the presence of families of 10, 12, and 18 nodal lines. At the corresponding frequencies, the largest vibration amplitudes occur at the periphery of the main and cover discs of the impeller.

6 Analysis of Possible Resonant Vibrations

When analyzing the dynamics of compressor impellers, the most important task is to identify resonance phenomena. Resonance occurs when the natural frequencies of the structure coincide with the frequencies of the exciting forces. The main excitation frequency is caused by static and dynamic imbalances in the impeller. The presence of impeller blades causes periodic disturbances with a frequency that is a multiple of the number (k = 18) of blades. The discrete arrangement of blades (k = 16) in the inlet blade assembly leads to the appearance of additional harmonic components of the disturbances. High-frequency pressure pulsations occur at the periphery of the impeller, with a frequency that is a multiple of the number (k = 36) of diffuser blades. The presence of a dense spectrum of natural frequencies of the impeller complicates the analysis of resonance phenomena. Campbell diagrams are a useful tool during the impeller design stage. These diagrams allow one to identify dangerous rotational frequency ranges, assess the influence of various excitation sources, and justify acceptable compressor operating modes.

Fig. 13 shows the Campbell diagram for an impeller made of a boron-aluminum composite containing 1.5% boron carbide. The set of horizontal lines f1–f10 corresponds to the natural frequencies of the impeller. The sloping line k = 18 shows the dependence of the excitation frequencies caused by the impeller blades on the impeller rotational speed. The sloping line k = 16 corresponds to excitations from the inlet casing blades, while line k = 36 demonstrates the highest harmonic associated with the influence of the diffuser blades. Static and dynamic imbalances of the impeller do not cause resonant vibrations (line k = 1). A similar Campbell diagram for an impeller made of a boron-aluminum composite containing 7.5% B_4C is shown in Fig. 14.

An analysis of the Campbell diagrams shows that resonance modes are possible within certain ranges of the impeller's angular velocity. The widest range is associated with the high-frequency harmonic (k = 36) of disturbances caused by the interaction between the impeller periphery and the vane diffuser. For the impeller manufactured from the composite containing 1.5% B_4C, the resonance speed range was found to be between 1500 and 3800 rpm. In contrast, for the impeller made from the composite containing 7.5% B_4C, resonant vibrations may occur within the speed range of 1600 to 4000 rpm.

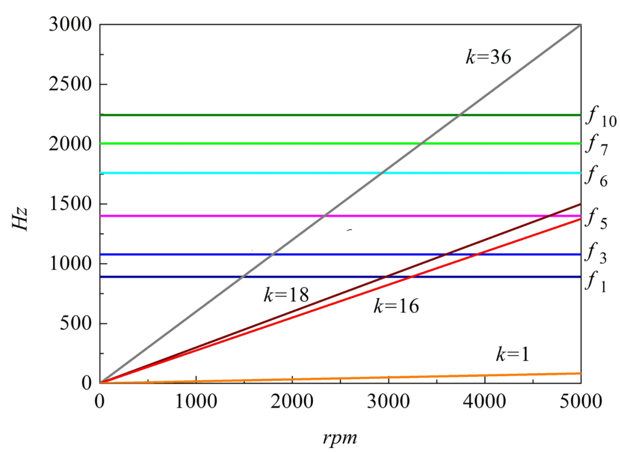


Figure 13: Campbell diagram for an impeller made of 1.5% B_4C composite.

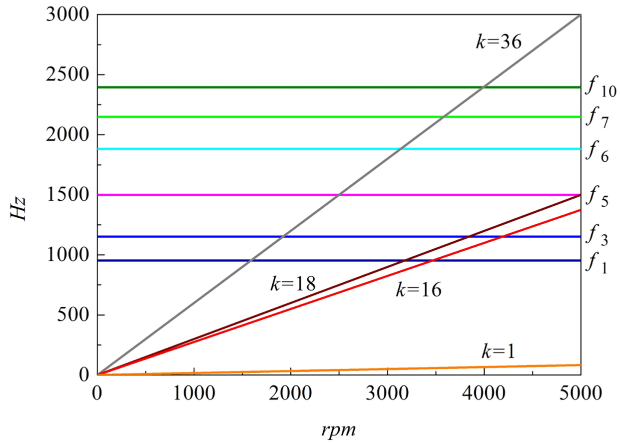


Figure 14: Campbell diagram for an impeller made of 7.5% B_4C composite.

Harmonic disturbances associated with the working blades (k = 18) and the inlet guide vanes (k = 16) define a narrower range of possible resonances. For an impeller made of a composite containing 1.5% B_4C, resonances occur at angular velocities ranging from 3000 to 4700 rpm. If the impeller is made of a composite with 7.5% B_4C, the resonance mode shifts to the range of higher speeds: from 3200 to 5000 rpm.

7 Conclusions

This study presents a comprehensive numerical investigation of the dynamic behavior of centrifugal compressor impellers manufactured from boron–aluminum metal matrix composites. Using detailed three-dimensional finite element modeling, the natural frequency spectrum and mode shapes of the impeller were determined while explicitly accounting for the prestressed state induced by centrifugal forces under realistic operating conditions. The obtained results confirm that centrifugal stiffening leads to a systematic increase in natural frequencies, an effect that must be considered when evaluating resonance risks in high-speed rotating machinery.

The results of the analysis of the prestressed state caused by centrifugal forces showed that the equivalent stresses (von Mises) in the impeller do not exceed 111 MPa. According to the data in [20], the yield strength for the composites under consideration ranges from 119.8 to 137.4 MPa, depending on the percentage of boron carbide. Consequently, for all the composites considered, centrifugal forces do not cause plastic deformation.

A comparative analysis of composites with different boron carbide contents showed that, for the three volume fractions considered—1.5, 4.5 and 7.5% B_4C—the elastic modulus increases significantly as the proportion of reinforcing phases increases, while the density decreases only slightly. As a result, impellers made of the composite with higher reinforcement content exhibit noticeably higher natural frequencies and a corresponding shift of critical resonance regions toward higher rotational speeds. This effect can be effectively exploited to improve the vibration reliability of centrifugal compressors without substantial increases in rotating mass.

In actual compressors, the impeller is connected to the shaft by a press fit. In this analysis, a rigid boundary condition is assumed, which ensures that there is no displacement on the inner surface of the impeller. For comparison, one variant with perfectly smooth contact was considered. The impeller was calculated using a material containing 1.5% B_4C, without accounting for centrifugal forces. The first natural frequency with rigid coupling to the shaft was 882.94 Hz, and for perfectly smooth contact, it was 882.86 Hz.

The analysis of vibration modes revealed a dense spectrum of eigenfrequencies associated with the rotational symmetry of the impeller, including torsional and bending modes with varying numbers of nodal diameters. Campbell diagram construction allowed identification of potentially dangerous resonance regions caused by harmonic excitations arising from rotor–stator blade interactions and diffuser vanes. It was shown that the most critical resonance zones are associated with higher-order harmonics generated at the impeller periphery, while excitations due to rotor imbalance do not lead to resonance within the examined speed range.

Overall, the results demonstrate that boron–aluminum metal matrix composites are a promising material choice for centrifugal compressor impellers from a dynamic performance standpoint. The developed computational approach provides a reliable tool for predicting resonance conditions at the design stage and can be used to optimize material selection and operating speed ranges. Future work should focus on experimental validation of the numerical predictions and on extending the model to include damping effects and fluid–structure interaction under unsteady flow conditions.

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Availability of Data and Materials: The authors confirm that the data supporting the findings of this study are available within the article.

Ethics Approval: Not applicable.

Conflicts of Interest: The authors declare no conflicts of interest.

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