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FINITE ELEMENT MODEL FOR ANALYSIS OF CHARACTERISTICS OF SHROUDED ROTOR BLADE VIBRATIONS

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Abstract. *The paper presents the approaches to FE modelling of blade airfoil, contact between the shrouds and operational damage. The regularities are established concerning the influence of the finite element type, finite element mesh and model of contact interaction on the spectrum of natural frequencies of blade assemblies. The use of the developed computational models is substantiated to determine the forced vibration characteristics of the selected objects of investigation. Based on the performed numerical experiments it was substantiated of finite element model selection for analysis of characteristics of shrouded rotor blade vibrations.*

Keywords: FE modeling, modal analysis, forced vibration analysis, shrouded blade, nonlinear vibrations

MODEL ELEMENTÓW SKOŃCZONYCH DO ANALIZY CHARAKTERYSTYK DRGAŃ ŁOPAT WIRNIKA OSŁONIĘTEGO

Streszczenie*: W artykule przedstawiono podejścia do modelowania elementów skończonych płata łopaty, styku osłon oraz uszkodzeń eksploatacyjnych. Ustalono prawidłowości dotyczące wpływu typu elementu skończonego, siatki elementów skończonych oraz modelu interakcji stykowej na widmo częstotliwości drgań własnych zespołów łopatek. Uzasadnione jest wykorzystanie opracowanych modeli obliczeniowych do wyznaczania charakterystyk drgań wymuszonych wybranych obiektów badań. Na podstawie przeprowadzonych eksperymentów numerycznych uzasadniono wybór modelu elementów skończonych do analizy charakterystyk drgań osłoniętych łopat wirnika.*

Słowa kluczowe: modelowanie MES, analiza modalna, analiza drgań wymuszonych, łopata osłonięta, drgania nieliniowe

Introduction

In the practice of modern aircraft engine design, determination of mechanisms of vibrations of compressor and turbine rotor blades is of current concern. To ensure high operational requirements for engines, their reliability, and functional serviceability, it is necessary to reduce the material consumption and improve technical and economic performance [3, 8].

Due to the significant increase in costs for the development of rotor blades, it is difficult to perform comprehensive laboratory and full-scale testing. Therefore, numerical experiment along with modern methods of computer modeling based on three-dimensional calculation models of blades and their assemblies is of great importance. It is one of the most common modes of discretization of the systems with the infinite number of degrees of freedom and developed procedures of their modeling. This method for the solution to the tasks of determination of the dynamic stress state of the specified objects is implemented in the majority of the modern software products.

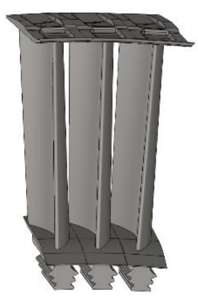
Rotor blades with flange shrouding are the most critical and, at the same time, most heavily-loaded structural elements of gas-turbine engines. Finite element modeling (FEM) of rotor blades should consider the features of the airfoil geometry, contact interaction between the shrouds of adjacent blades, as well as their possible local damages in the engine operation [3, 5, 7].

However, the review of the scientific and technical manuscripts demonstrates that the accuracy of the obtained results depends (considerably) on the type of finite element (FE) and its dimensions, as well as the approaches to modeling of various types of interaction between the structural elements of the object under investigation. The goal of this paper is in the generalization of the methods and approaches used by the authors for FEM of such mechanical systems as assemblies of turbine engines with flange shrouding of blades in the determination of characteristics of their vibrations [1, 7, 8].

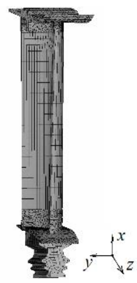
1. Object of investigation and general approaches to its finite element modeling

A turbine rotor wheel (Fig. 1a) with its blade assembly containing 136 (N = 136) shrouded blades made of heat-resistant alloy ZhS 26-VI is chosen to perform the numerical calculations. The physical and mechanical characteristics of the alloy are as follows: the elastic modulus $E = 1.9 \cdot 1011$ Pa; density $\rho = 8570$ kg/m³; Poisson's ratio $\mu = 0.3$.

Considering strict cyclic or rotational symmetry of the blade assembly its analysis is reduced to the consideration of certain cycle with the corresponding boundary conditions, which can involve one or several blades. The determining feature in FEM of the rotor wheel is the mesh of its blades, which should reflect (as possible) their geometry, properties of the material and acting loads. Noteworthy is that in practice it is impossible to consider these factors in full due to imperfections in the blade structure (and their materials) and different operation modes. Therefore, some idealization of the shape of blades, properties of the materials and conditions of external effects takes place in the development of FE models of rotor blades depending on the tasks assigned. Here the successful selection of the model's parameters defines the scope of calculations, validity, and accuracy of the obtained results.



a)



b)

Fig. 1. Fragment of the blade assembly under investigation (a) and FE model of isolated blade (b)

From the results of FEM calculations, it is apparent that regular mesh is the most preferred for any machine building structures. The only exceptions are those zones where irregular geometric shapes are observed, for instance, at the connections of the blades and the shrouds, as well as within areas containing different damages. Due to the specific geometric features, the model includes domains with different types of mesh [2, 6, 11].

To construct the FE models of the shrouded blades, three-dimensional 8-nodes and 20-nodes finite elements are used (Fig. 2). These types of elements were chosen in this investigation to solve the issue.

Figure 1b illustrates the FE model of the isolated blade using the approaches for it creating above.

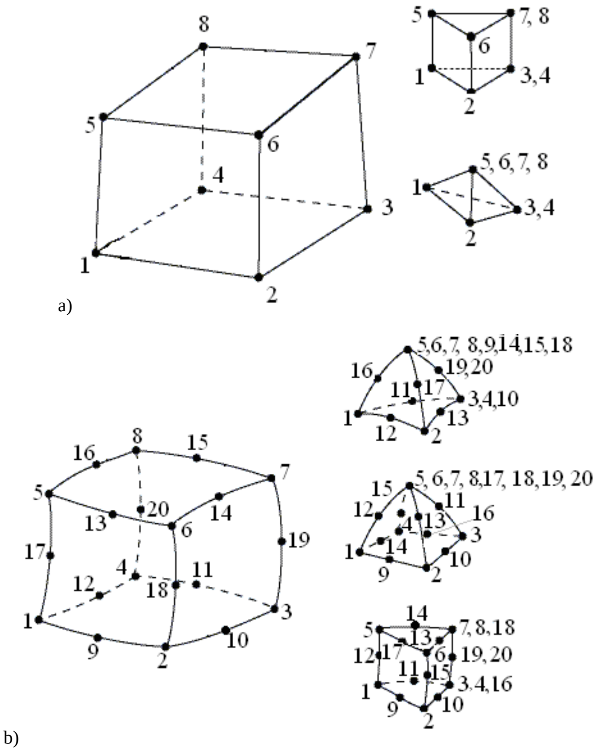


Fig. 2. Finite elements and their modifications: a – linear 8-nodes; b – quadratic 20-nodes

Let us consider the approaches to FEM of the shrouded blades and their assemblies in the solution to the issue aimed at the determination of characteristics of their vibrations [7, 9, 14].

2. Determination of the spectrum of natural vibration frequencies

Using the developed FE model of the rotor blade a set of numerical investigations was performed to determine the influence of the type and dimensions of FE, as well as contact interaction modeling, on the spectrum of the natural frequencies of vibrations of the blade assembly under investigation. The calculations were conducted employing various types of finite elements and the density of FE mesh, which is defined by the number of finite elements *k*, as well as procedures of linearization of the contact between the shrouds of the adjacent blades.

The structural cyclic symmetry of the blade assembly is assumed in this paper. As is known [4, 12], in this case, when performing the computational experiments, it is possible to confine ourselves to the consideration of the only one sector for a blade row, the number of which, in the presence of the annular shrouding, coincides with the number of blades. A mandatory requirement here is the fulfillment of the boundary conditions at the nodes of connecting the adjacent sectors along the contact surfaces of the shrouds and the disk.

For the purposes of simulating the shrouding of the blades, the procedure described earlier [5, 13, 15] was proposed, whose main thesis are as follows:

1. The shroud is cut in such a way that its parts are connected along the common surface, and the nodes lying on this surface coincide.
2. The cut-off part of the shroud with the FE mesh is displaced by the angle $\beta = 360^{\circ}/N$ (shown by the arrow in Fig. 3a) in the cylindrical system of coordinates, which makes it possible to simulate the tension along the contact surfaces of the shrouds.

Considering the cyclic symmetry of the assembly the conditions of coupling on the cut surfaces C1 and C2 of its sector are specified as shown in Fig. 3b.

It is also necessary to take into consideration the conditions of the blade-to-disk interaction along the contact surfaces of the root since this can influence the contacting character of the shrouds. Since the assembly process assumes that the contact surfaces of the fir-tree blade root fit closely to the corresponding surfaces of the disk, and also to avoid its influence on the investigation results, the rigid attachment of the blade root at the disk groove was adopted as the boundary conditions.

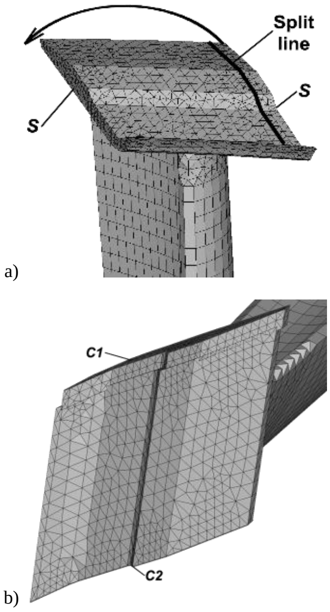


Fig. 3. Schemes of the three-dimensional cut of the shroud (a) and contact interaction modeling (b)

The procedure of substantiation of the developed FE model of the assembly was based on the solution to the issue in the absence of the shrouded coupling between the blades, i.e. individual or isolated blade. Figure 4 illustrates the diagrams of the first four natural frequencies of vibrations *pi* ($i = 1, \ldots, 4$) of the blade in the variation of its mesh density for the selected types of finite elements.

The analysis of the data indicates that the natural frequencies of vibrations of the considered blades are not affected by both the FE type and density of FE mesh. As is obvious does not exceed 2.22%. However, the calculation time is more efficient while using the models based on 8-node finite elements in comparison to 20-node elements.

Let us consider the determination of the influence of the FE type and FE mesh density on the spectrum of natural frequencies of the blade assembly under investigation providing its cyclic symmetry. Such numerical investigations imply exception of any nonlinear nature of deformation of the system under study. Thus, the modeling of contact interaction between the shrouds (it is generally of nonlinear nature) requires its linearization.

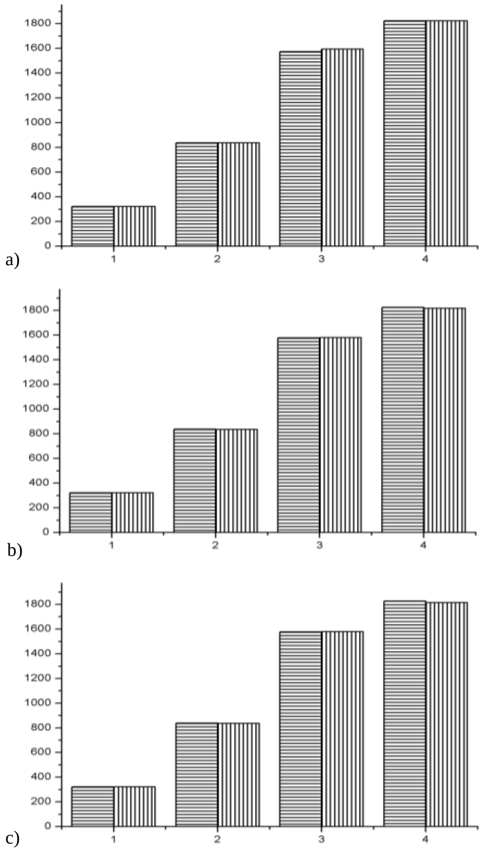


Fig. 4. *Diagrams of the first four natural frequencies of the blade vibrations at k = 5000 (a), 7000 (b) and 12000 (c) of 8-nodes (horizontal lines) and 20-nodes (vertical lines) finite elements*

There are different variants of modeling of the contact interaction between the shrouds, namely [2]:

1. The conditions of consistency of displacements are imposed on all the nodes of finite elements of the contact surfaces.
2. The conditions of consistency of displacements are imposed on all the nodes of finite elements of the contact surfaces with the non-zero values of contact pressure. Here the static contact issue was previously solved. The results of the solution are used to choose the nodes for the imposition of the corresponding conditions of consistency of displacements.
3. The elastic elements by two degrees of freedom (Ux and Uy) are added between the nodes of finite elements with the non-zero values of contact pressures. Moreover, the value of contact pressure (observed within the corresponding nodes) is specified for each such element. The conditions of consistency of displacements in the radial direction (along the blade airfoil) are imposed on the specified nodes within the region of the maximum values of contact pressure along the conjugated surfaces of the shrouds.

The values of contact pressure and its distribution over the planes of contact between the shrouds required to perform the numerical experiments are taken from [5, 10, 15]. Based on these calculation results, the natural frequencies of vibrations of the blade assembly under investigation were determined for the selected variants of the contact interaction between the shrouds. The obtained values of natural frequency were applied to plot the frequency functions (Fig. 5) as the dependencies of the vibration frequency on the number of nodal diameters m, where n is the number of nodal circles. The solution was made for the modes of the assembly vibrations $n = 0$.

It is seen that variant 1 is characterized by higher values of the natural frequencies of vibrations of the blade assembly as compared with variants 2 and 3. It is explained by the fact that variant 1 involves the blade assembly as a system with elastic coupling, which results in the considerable overestimation of the results of calculation of its natural vibration frequencies. Variant 2 describes the conditions of contact between the shrouds that are of identical character for all the nodes within its area, which also causes the stiffness increase for the system under consideration. Variant 3shows the calculation results that comply to the actual values of natural frequencies of the blade vibrations determined in full-scale testing. Also, the maximum difference ε in the values of natural frequencies of vibrations determined using the considered FE models of the blades is about 7% with the average density of the blade mesh.

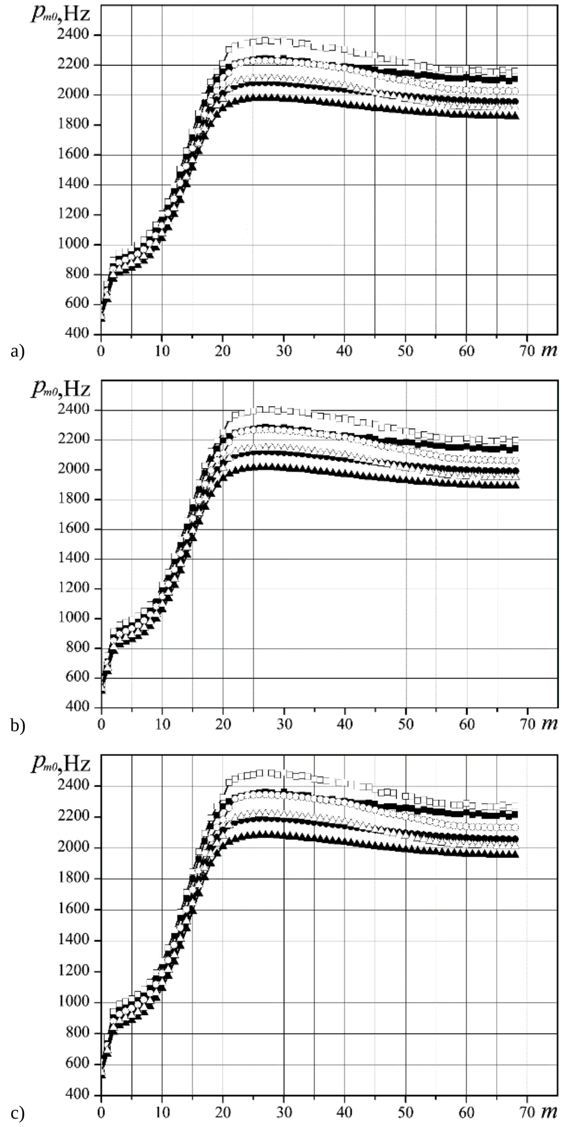


Fig. 5. *Frequency functions of the blade assembly for variants 1 (■, □), 2 (●, ○) and 3 (▲, Δ) of the contact interaction k = 5000 (a), 7000 (b) and 12000 (c) 8-nodes (■, ●, ▲) and 20-nodes (□, ○, Δ) finite elements*

3. Determination of characteristics of forced vibrations

The calculations on the determination of the influence of contact interaction modeling of the shrouds were made employing the assembly consisting of two blades, which is the simplest regular system as shown in [5, 15]. Here it is possible to fully investigate the influence of different factors on the vibrations of the studied structures.

Figure 6 illustrates the model of the selected assembly without the consideration of the root. Node A, where the characteristics of its forced vibrations were determined, is also shown here. There are two approaches to model the contact interaction between the shrouds. The first approach involves the solution of the contact issue using the Newton-Raphson procedure, while the second one – linearization of the contact interaction in compliance with variant 2 modeling considered in the analysis of the spectrum of natural vibration frequencies of the blade assembly. Later the equations of the forced vibrations were solved using the Newmark scheme.

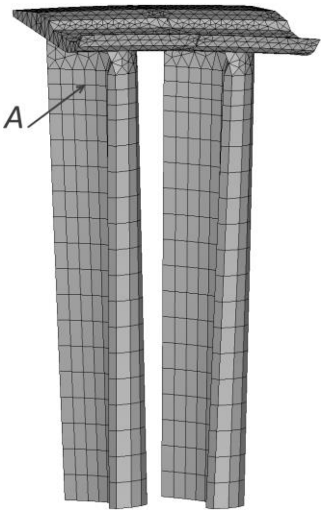


Fig. 6. FE model of the assembly consisting of two blades

The distribution of exciting forces Q acting on the blades due to the working body flow is of a complex nature. All the calculations were performed for two cases of harmonic excitation of vibrations of the blade assembly: in-phase $Q_1 = Q_2 = Q_0 \sin vt$ and anti-phase $Q_1 = -Q_2 = Q_0 \sin vt$. The frequency of the exciting force v varied in the range of the spectrum of natural frequencies of the assembly vibrations.

From the results of the calculations, the amplitude-frequency characteristics (AFC) of the assembly were determined. They were used to build the dependencies of the amplitude of displacements within the chosen node A on the relations between the vibration frequencies v/p_0, where p_0 is the resonance frequency of the assembly vibrations.

Figure 7 illustrates the examples of the AFC. Its detailed consideration and analysis allow one to draw the following conclusions:

1. The character of the AFC is almost identical during in-phase excitation using both nonlinear and linearized calculation models [5].
2. A significant nonlinearity of the system is evident only at anti-phase excitation of the blade assembly vibrations.
3. Depending on the calculation model used, the maximum stresses in the assembly will be observed at various frequencies of exciting force, their difference does not exceed 5%. When employing the linear model, the maximum stresses are observed at the lower frequency of exciting force in case of in-phase shape. For anti-phase shape, the maximum stresses are detected at the larger frequency of exciting force as compared with the results of calculations based on the nonlinear model. This is explained by the variation of the system stiffness.

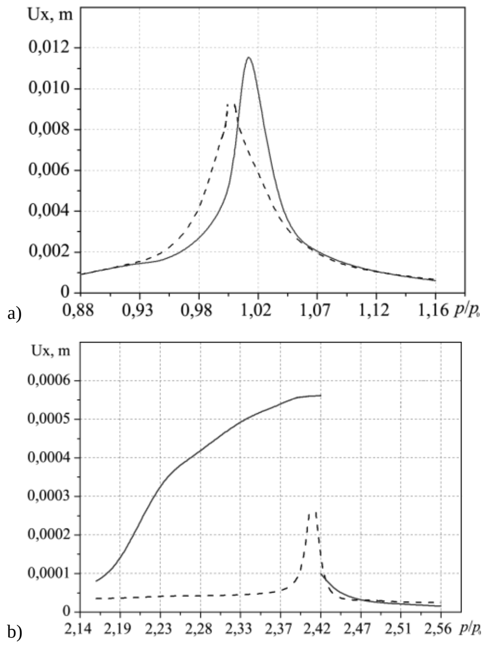


Fig. 7. Amplitude-frequency characteristics of displacements along axis 0x for nonlinear (solid lines) and linear (dashed lines) models during in-phase (a) and anti-phase (b) harmonic excitation of vibrations

4. Investigation of blade vibrations considering damages

Damage modeling in the investigation of its influence on the vibration characteristics of the structural elements is important. In the case of dents, erosion and other damages, it is possible to model the defect presented as a rectangular or wedge-shaped notch with fine FE mesh in its zone [15]. The mass of the system is maintained in the presence of a fatigue crack but its stiffness varies in the process of its cyclic deformation.

Based on the previous experience [3], a fatigue crack was modeled as the mathematical cut. In case of open crack, there is a mutual non-penetration of its faces. This is ensured by the introduction of the surface contact elements and solution of the contact issue along with modeling of contact interaction between the shrouds of rotor blades.

Considering that the calculation of forced vibrations for such complicated structural elements requires large computer resources, in the investigation of the blade airfoil the decision was made to consider only its less twisted part of the length $L = 0.086$ m (Fig. 8). The calculations on the influence of the crack presence on the vibration behavior of the chosen investigation objects were performed for the case where the crack is at the distance $x_T = 0.1L$ from the edge cross-section of the blade airfoil on the side of the blade root and rigid fixation of the beam as a cantilever.

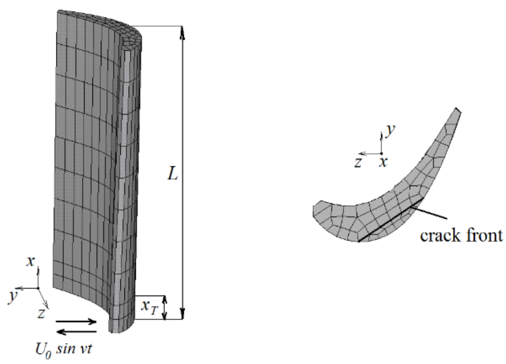


Fig. 8. FE model of the blade airfoil with damage and cross-section of the airfoil with a crack

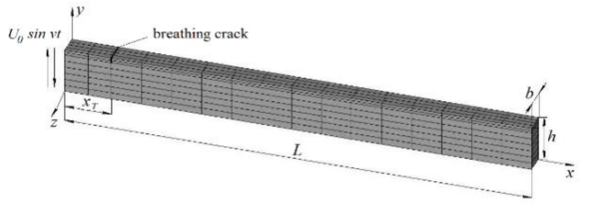


Fig. 9. FE model of the beam of the rectangular cross-section with breathing crack

To verify the accuracy of the proposed crack model, the results of calculation were compared with similar ones obtained for the beam of a rectangular cross-section of $b \times h = 0.004 \times 0.009$ m and $L = 0.086$ m (Fig. 9) [7] with identical dimensions in terms of area and crack location over the beam length.

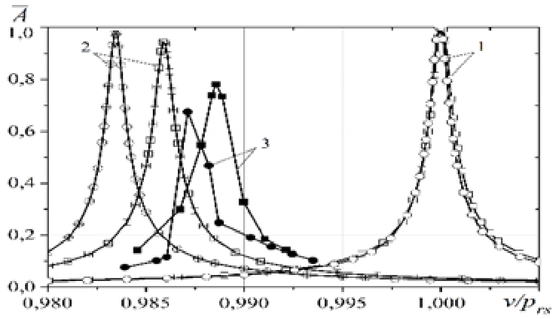


Fig. 10. Amplitude-frequency characteristics of the blade airfoil (○,⊕,●) and beam of the rectangular cross-section (□, ⊞ ,■) without damage (1) and with open (2) and breathing

The forced vibrations were induced by means of harmonic displacement U_0 *sin vt* of edge sections of the chosen objects along the axis 0*y*. Figure 10 displays the AFC of the selected objects of investigation without damage, as well as with open and breathing cracks, where p_{rs} is the resonance frequency of vibrations of the undamaged airfoil (beam), is the relation of the amplitude of forced vibrations and the maximum value of the amplitude of the undamaged airfoil (beam) at main resonance.

As it is seen AFC are practically identical for the undamaged blade airfoil and beam. AFC are different in the presence of open and breathing cracks of the airfoil and beam. This is due to the complex geometry of the blade airfoil and variation of the mode of its vibrations with damage.

Noteworthy is that the model of open crack (fatigue damage) exhibits the reduced value of the resonance frequency of vibrations and, as a rule, does not allow to evaluate the actual damage parameters.

During vibrations of the objects with a breathing crack on a frequency of the main resonance the vibration diagnostic parameters of its presence were determined, namely: variation of the resonance frequency of vibrations $\Delta\bar{\omega}_0$, as well as the relation between the amplitudes of the dominant harmonics of displacements $\bar{A}_2$ and accelerations $\bar{A}_2^a$ due to the stiffness system variation (table 1). They are in a good correlation between each other and can be used in the damage diagnostics.

Table 1. Vibration diagnostic parameters of the presence of breathing crack

Object of investigation	$\Delta\bar{\omega}_0$	$\bar{A}_2$	$\bar{A}_2^a$
Beam	0.011	0.0042	0.0168
Blade airfoil	0.0132	0.0055	0.022

Thus, the procedure of modeling of open and breathing cracks in turbine machine blades was proposed for further investigation of forced vibrations.

5. Conclusions

Based on the performed numerical experiments it was substantiated of finite element model selection for analysis of characteristics of shrouded rotor blade vibrations. Also, it was established the following:

- FE type and density of FE mesh do not practically affect the results of calculations of natural frequencies of the isolated blade vibrations. The results of determination of the spectrum of natural frequencies of the blade assembly vibrations depend significantly on the specified characteristics of FE mesh. For instance, the difference ε in the values of natural frequencies of vibrations (with the average density of FE meshes) obtained using the models based on 8 and 20-nodes finite elements is approximately 7%;
- modeling of the interaction between rotor blades using the imposition of the consistency of displacements between all the nodes of contact surfaces or those with non-zero contact pressures result in the increased values of natural frequencies of vibrations of the blade assembly as compared with those obtained using spring elements. This is explained by the maximum stiffness of the joint in comparison with the latter variant of modeling;
- for the in-phase mode of excitation of vibrations of the blades, the AFC's character obtained using non-linear and linear calculation models is practically identical. The considerable nonlinearity is observed only at the anti-phase mode of excitation. Here the difference between the frequencies, whereby the maximum stresses are observed, does not exceed 5% for the investigated models;
- breathing crack modeling with the contact elements allows one to determine the frequency and nonlinear characteristics of vibrations more accurately as compared with the open crack modeling, which enhances the efficiency of its use in the investigation of fatigue damages of rotor blades.

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