

Using Computer Modeling and Artificial Neural Networks for Ensuring the Vibration Reliability of Rotors

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Abstract. The paper is devoted to the development of the refined numerical and mathematical models of rotor dynamics for high-performance turbomachines. These models consider the nonlinear regression dependencies between the radial stiffness of bearing supports and rotor speed of the shaft. As a result of the comprehensive application of the proposed approach, the algorithms of identification of the stiffness parameters of the bearing supports were proposed based on the critical frequencies of the rotor system. Additionally, the methodology for estimating the system of initial imbalances by the related displacements of the rotor axis in the correction and calculation cross-sections was improved. The proposed approaches based on the comprehensive application of the CAE software and computational intelligence systems allow conducting the modal and harmonic analysis and realizing the virtual balancing with the significant decrease of the preparatory and machine times without the loss of relative accuracy. Additionally, the developed mathematical models of free and forced oscillations of rotor systems were implemented as a program code of the operating files “Critical frequencies of the rotor” and “Forces oscillations of the rotors” within the computer algebra system that allows improving the dynamic balancing procedure in order to estimate the system of initial imbalances. The high accuracy of the proposed approach is confirmed by the verification of dynamic deflections of the rotor axis by the system of residual imbalances in accordance with the norms of vibration reliability.

Keywords: Critical Frequencies, Forced Oscillations, Virtual Balancing, CAE Software, Computational Intelligence Systems.

1 Introduction

The permanent acceleration of the development in the field of energy technologies is due to the use of up-to-date highly-intensive equipment, particularly multistage centrifugal machines (i. e. pumps and turbocompressors). The consequent increase in of their technical characteristics leads to increasing complexities in terms of vibration

reliability. In this case, features in designing of the power equipment require a more detailed analysis of the dynamic state for pump and compressor units as the most energy-intensive ones. Due to this trend, the use of existing mathematical and numerical models is practically inappropriate, since the relatively high rotational speeds that caused significant nonlinear effects. Due to the abovementioned, there is a necessity to create the refined and reliable models of rotor dynamics, which will be allow carrying out the modal and harmonic analysis, as well as evaluating the impact of the operating parameters on the stiffness characteristics of the bearing supports in order to qualitatively realizing the rotor balancing procedure.

The abovementioned problem is solved by the determination of critical frequencies and related mode shapes of rotors, as well as by conducting the previous virtual balancing procedure. Thus, the paper is aimed at developing computational and analytical approaches both for ensuring vibrational reliability of rotary systems both for estimating the parameters of the dynamic state of turbomachines and for realizing the virtual balancing procedure.

To achieve the stated aim, the ANSYS software, as well as working files “Critical frequencies of the rotor” and “Forced oscillations of the rotor” are used. These two files were developed at the Department of General Mechanics and Machine Dynamics of Sumy State University and implemented into the MathCAD software.

2 Literature Review

The abovementioned statement of the problem can be successfully solved after the precise analysis of the recent researches in the fields of computer modeling of rotor dynamics. Particularly, the research paper [1] deals with the estimation of the reliability of the rotors’ automatic axial-balancing devices in multistage centrifugal pumps. As a result, the methodology for numerical calculations was proposed. The research work [2] presents the methods of static and dynamic analysis of the closing rotor balancing device of the multistage centrifugal pumps. However, the proposed approaches are based on the criterion of the permissible amplitudes of axial oscillations only.

The phenomenon of rotor’s instability for the case of tilting pad bearings is described in the paper [3]. As a result of numerical simulation of rotor dynamics considering stiffness of bearing supports, spectral analysis was carried out, and related phase paths of rotor’s motion were presented, as well as features of rotor dynamics in a range of unstable rotational speeds were analyzed. Consequently, ways for designing the radial sliding bearing supports with the hydrostatically suspended pads for ensuring the dynamic stability of rotors are proposed in the paper [4], as well as the technology of improving the quality of contact surfaces for optimizing the operating characteristics of bearing supports is presented in the research work [5]. However, the abovementioned approaches do not allow estimating parameters of bearing stiffness.

A comprehensive methodology for identification of load categories in rotor systems based on vibration analysis is proposed in the paper [6]. It can be successfully

applied for the fault diagnosis of pumps and compressors. Additionally, ways of the faults detection in gas turbines' rotors using vibration analysis under varying conditions are completely described in the paper [7]. Moreover, enhancement of performance for steam turbines using artificial intelligent systems and electric circuit design is realized within the research [8]. As a result, a good agreement between the data of numerical simulation and a physical experiment was achieved that allowed predicting the operating parameters with a relatively small error.

The reliability analysis on resonance for low-pressure compressor rotor blade based on least squares support vector machine with leave-one-out cross-validation is presented in the paper [9]. Additionally, the approach for detection and modeling of vibration behavior of a gas turbine based on artificial neural networks is used in the research works [10, 11], as well as the methodology for the detection of propagating cracks in rotors using neural networks is used in the paper [12]. Moreover, ways of using artificial neural networks and genetic algorithms for bearing fault detection are presented in the paper [13]. However, the abovementioned approaches do not allow estimating parameters of bearing supports by means of the implemented computational intelligence systems.

Due to the abovementioned, a comprehensive approach of combined using the numerical simulation and computational intelligence systems should be developed both for identification of the related mathematical models and ensuring the vibration reliability of rotor systems.

3 Research Methodology

3.1 Eigenfrequencies and Critical Frequencies of the Rotor

The objects of the study are free and forced oscillations of the flexible rotor of the centrifugal compressor 225GC2-135/12-50M1245 (Fig. 1) at the 1643 MW Navoiy Thermal Power Plant (Yangi-Ariq, Uzbekistan) with the following technical parameters: flow rate 25 m³/s; inlet pressure 1.20 MPa; outlet pressure 4.95 MPa; pressure ratio 4.13; polytropic efficiency 70 %; operating frequency 934 rad/s; nominal power 7.33 MWt; inlet temperature 301 K; temperature difference 156 °C; compression ratio 0.98.



Fig. 1. A rotor of the multistage centrifugal compressor 225GC2-135/12-50M1245.

The calculation of eigenfrequencies provides the determination of potentially dangerous rotor speeds under its rotation at the operating frequency, which is equal to 934 rad/s. Therefore, the bearing stiffness for the common calculations was adopted constant and equal to $c_w = 2.94 \cdot 10^8$ N/m at the operation rotor speed.

In fact, the calculation of critical frequencies needs to consider that bearing supports have different stiffness at different rotor speed. The working file “Critical frequencies of the rotor” allows directly determining the nonlinear dependencies of the stiffness characteristics of the rotor’s bearing supports on the rotational speed as the following approximation [14]:

$$c(\omega) = c_0 + \alpha\omega + \beta\omega^2 \quad (1)$$

with the unknown coefficients c_0 , α , and β evaluated on the basis of the linear regression procedure by minimizing the following error function:

$$R(c_0, \alpha, \beta) = \sum_{k=1}^n (c_0 + \alpha\omega_k + \beta\omega_k^2 - c_k)^2 \rightarrow \min, \quad (2)$$

where c_k , ω_k – k -th experimental values of bearing stiffness and frequency, respectively; n – total number of experimental dataset.

In this case, the system of equations for determining the unknown parameters takes the form:

$$\begin{cases} \frac{\partial R}{\partial c_0} = 2 \sum_{k=1}^n (c_0 + \alpha\omega_k + \beta\omega_k^2 - c_k) = 0; \\ \frac{\partial R}{\partial \alpha} = 2 \sum_{k=1}^n (c_0 + \alpha\omega_k + \beta\omega_k^2 - c_k)\omega_k = 0; \\ \frac{\partial R}{\partial \beta} = 2 \sum_{k=1}^n (c_0 + \alpha\omega_k + \beta\omega_k^2 - c_k)\omega_k^2 = 0, \end{cases} \quad (3)$$

which can be rewritten as a matrix equation:

$$[\Omega]\{X\} = \{C\}, \quad (4)$$

where $[\Omega]$, $\{C\}$, and $\{X\}$ are the following matrix 3×3 and the column vectors 3×1 :

$$\Omega_{i,j} = \sum_{k=1}^n \omega_k^{i+j-2}; \quad C_i = \sum_{k=1}^n c_k \omega_k^{i-1}; \quad \{X\} = \{c_0, \alpha, \beta\}^T. \quad (5)$$

The solution of the matrix equation (4) is

$$\{X\} = [\Omega]^{-1}\{C\}. \quad (6)$$

In the case of the compressor 225GC2-135/12-50M1245, the evaluated values of the coefficients are $c_0 = 1.77 \cdot 10^8$ N/m, $\alpha = 1.30 \cdot 10^5$ N·s/m, and $\beta = -4.97$ N·s²/m.

The operating file “Critical frequencies of the rotor” is based on the calculation of the 2D-beam model of rotor dynamics (Fig. 1 a). Consequently, the input data is a description of each i -th finite element for the following characteristics: density ρ_i , kg/m; Young’s modulus E , Pa; length L_i , m; external and internal diameters D_i and d_i , correspondently, m; single mass m_i , kg; polar moment of inertia I_i , kg·m²; radial stiffness of the bearing support $c_i(\omega)$, N/m according to the dependence (1).

The ANSYS software allows simulating the rotor dynamics based on the finite element method both for 2D-beam and 3D finite elements (Fig. 2 b). The first one is not of interest because the obtained results will not be fundamentally different from the previously calculated using the working file “Critical frequencies of the rotor”.

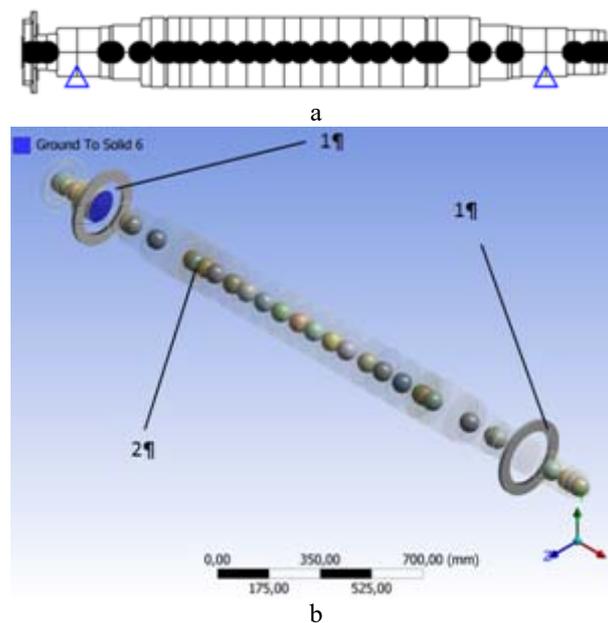


Fig. 2. The finite element models for simulating free oscillations using the working file “Critical frequencies of the rotor” (a) and ANSYS software (b): 1 – bearings; 2 – single masses.

It should be noted that capabilities of the ANSYS software do not allow calculating the critical frequencies directly, as it possible using the working file “Critical frequencies of the rotor”. Therefore, the research in ANSYS is based only on building and further analyzing the Campbell diagram.

3.2 Forced Oscillations of the Rotor

The working file “Forced oscillations of the rotor” allows calculating the forced oscillations of a rotor under the given rotation speed and the influence of a certain system of imbalances. As a result, amplitudes of the rotor’s deflections and mode shapes are determined an operating frequency.

The following data is required to solve the problem of forced oscillations: density ρ_i , kg/m; Young's modulus E , Pa; length L_i , m; external and internal diameters D_i and d_i , correspondently, m; single mass m_i , kg; polar moment of inertia I_i , kg·m²; operating frequency $\omega_w = 934$ rad/s; radial stiffness of the bearing support at the operating frequency $c_r(\omega_w)$, N/m, determined by the equation (1); the system of imbalances D_i , kg·m.

Theoretically, imbalances of a flexible rotor are characterized by a certain spatial curve as a hodograph of the continuous system of imbalance vectors perpendicular to the rotor axis. Since this approach is rather time-consuming, subsequent calculations can be limited by a finite number of local imbalances located in the cross-sections of impellers and other massive parts.

The harmonic analysis is based on the 3D finite element model and realized using the ANSYS software. In this case, there is a need to set parameters for imbalances distribution. Fig. 3 presents the calculation models based on the 2D-beam and 3D finite element models.

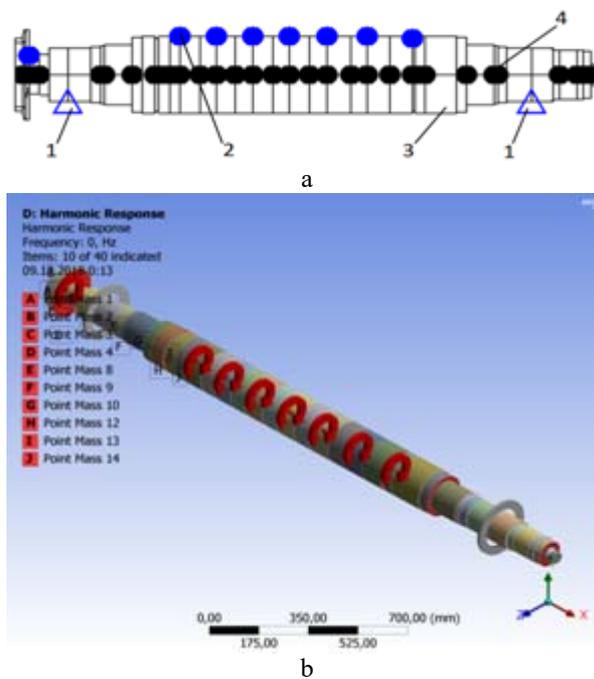


Fig. 3. The finite element models for simulating forced oscillations using the working file “Forced oscillations of the rotor” (a) and ANSYS software (b): 1 – bearings; 2 – single masses; 3 – beam finite elements; 4 – local imbalances.

3.3 Virtual Balancing of the Rotor

For realizing the virtual balancing procedure, initially, the amplitudes of the rotor's oscillations are measured based on the numerical simulation in 8 correction planes (7 impellers and 1 drive unit), in which balancing masses will be installed.

Traditionally, the balancing procedure based on the calculation model assumes a linear relationship between the imbalances and related movements. In this case, the experimental results allow calculating the coefficients of influence, as well as estimating the imbalances, which should be located in the related cross-sections and cause complex displacements $W_{a,i}$. For this purpose, it is necessary to conduct 9 measurements of the amplitude in the correction planes. The first measurement is in its original state, and the other one – with installing a trial imbalance $\Delta D = 0.01 \text{ kg}\cdot\text{m}$ in each of the 8 correction planes. The dependence of the amplitude for the rotor displacements on the system of imbalances is as follows [15]:

$$\{Y\}_0 = [W]\{D\}, \quad (7)$$

where $\{Y_0\}$ – column vector of amplitudes, m; $[W]$ – a matrix of the influence coefficients, kg; $\{D\}$ – column vector of the estimated imbalances, kg·m.

After introducing the i -th vector of trial imbalances as a set of imbalances located in the i -th correction plane, the following equation can be written for the i -th start ($i = 1, 2, \dots, 8$) of the rotor system:

$$\{Y\}_i = [W]\{\{D\} + \{\Delta D\}\}, \quad (8)$$

where $\{\Delta D\}$ – column vector of trial imbalances.

Subtracting (7) from (8) allows determining the matrix of influence coefficients, elements of which are equal to

$$W_{a,i} = \frac{Y_{a,i} - Y_{0a}}{\Delta D}, \quad (9)$$

where $a = \{1, 2, \dots, k\}$ – row number of the matrix $[W]$; $k = 8$ – total number of the correction planes.

The determined matrix of the influence coefficients $[W]$ and the column vector of amplitudes $\{Y\}_0$ allow directly estimating the imbalances in the correction planes. Thus, from the formula (7), it is can be obtained:

$$\{D\} = [W]^{-1}\{Y\}_0. \quad (10)$$

In the general case of rectangular matrix $[W]$, this equation should be rewritten in the following form:

$$\{D\} = ([W]^T [W])^{-1} [W]^T \{Y\}_0. \quad (11)$$

4 Results and Discussion

4.1 Results of the Modal and Harmonic Analysis

For the bearing stiffness calculated by the formula (1) in a range of rotation speeds from 0 to 3000 rad/s with a step 250 rad/s including the operating rotor speed 934 rad/s, the eigenfrequencies were determined. As a result, the Campbell diagram was built (Fig. 4) with the X-axis as an operating frequency, and Y-axis as an eigenfrequency.

Fig. 5 shows the following calculation results of the rotor: critical frequencies and their mode shapes. The corresponding numerical values are summarized in Table 1. The results of calculating forced oscillations as rotor displacements are shown in Fig. 6.

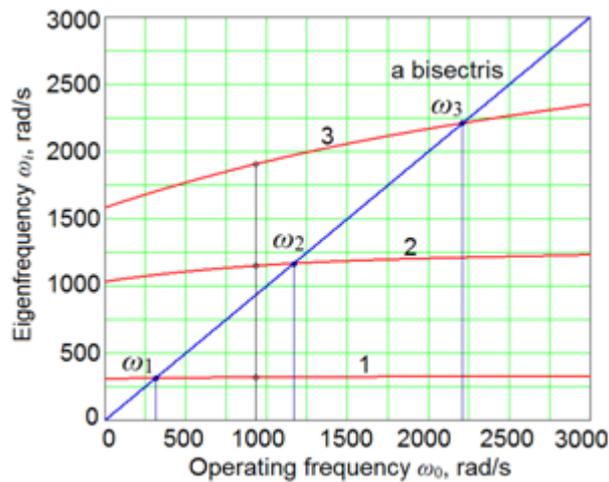


Fig. 4. The Campbell diagram.

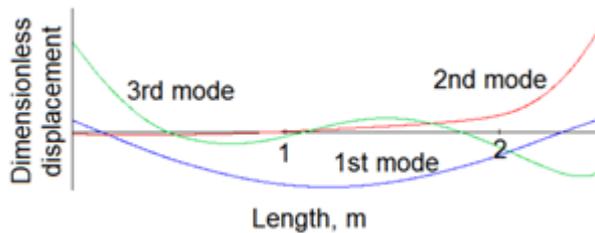


Fig. 5. The mode shapes of the rotor.

Table 1. Eigenfrequencies and critical frequencies, rad/s.

Mode shape number	Eigenfrequency			Critical frequency		
	MathCAD	ANSYS	Relative error, %	MathCAD	ANSYS	Relative error, %
1	318	319	0.3	312	313	0.3
2	1148	1125	2.0	1164	1140	2.1
3	1906	1867	2.0	2210	2150	2.7

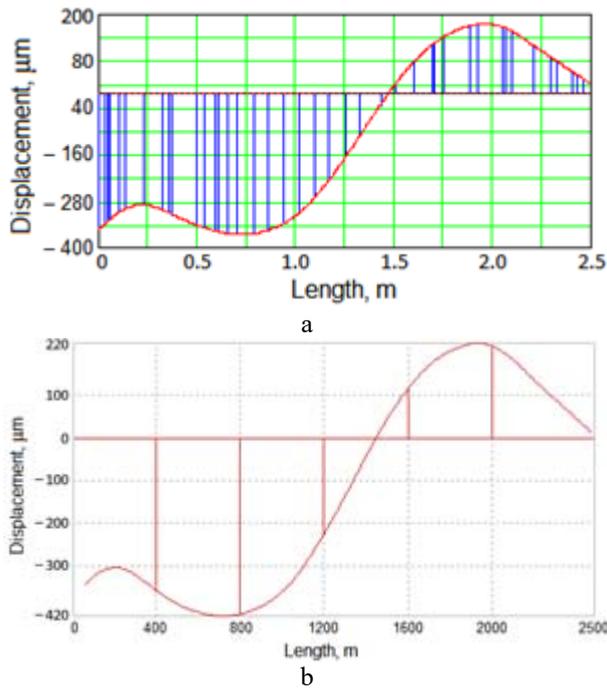


Fig. 6. The mode shapes of the unbalanced rotor at the operating frequency using the working file “Forced oscillations of the rotor” (a) and ANSYS software (b).

4.2 Evaluation of the System of Imbalances

The results of the virtual balancing procedure are given in Table 2. The related shapes of the rotor under the residual imbalances after the balancing procedure is presented in Fig. 7.

Thus, as a result of the virtual balancing procedure, the deflections of the rotor’s axis under the system of residual imbalanced meet the established standards in vibration reliability [16].

Table 2. The results of the rotor balancing procedure.

Correction plane number i	Initial displacement Y_{i_j} , m	Balancing imbalance D_{i_j} , kg·m
1	$-3.36 \cdot 10^{-4}$	0.038
2	$-3.63 \cdot 10^{-4}$	0.010
3	$-3.63 \cdot 10^{-4}$	0.038
4	$-3.37 \cdot 10^{-4}$	0.007
5	$-2.69 \cdot 10^{-4}$	0.069
6	$-1.64 \cdot 10^{-4}$	0.018
7	$-3.15 \cdot 10^{-5}$	0.009
8	$7.90 \cdot 10^{-5}$	0.080

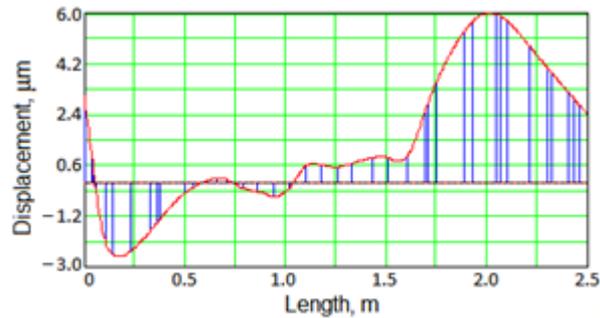


Fig. 7. Displacements of the rotor's axis after the balancing procedure.

4.3 General Approach for the Application of Artificial Neural Networks

The linearity of the traditional mathematical models is the main disadvantage of the abovementioned procedures for determining the stiffness of bearing supports, critical frequencies of the rotor, and dynamic deflections of the axis. Additionally, the general approach does not regulate the general rule for the numerical value of a trial imbalance. Moreover, conducting a series of calculations in an amount not less than $(k + 1)$ is obligatorily to determine the system of imbalances. However, considering the nonlinear dependencies between the stiffness parameters of the bearing supports and critical frequencies of the rotor, as well as between the axis deflections and a system of residual imbalances is a highly-complicated problem, solving of which will allow offering a refined virtual balancing procedure.

In the case of significant nonlinearities of mathematical models, using the computational intelligence systems were realized by the row of researchers. Particularly, in the paper [17], a method of wavelet packet combining neural network fusion is implemented In order to improve the accuracy of fault diagnosis. A neural network monitoring system is used in the research [18] for the frequency vibration prediction in a gas turbine. Additionally, the paper [19] deals with the early detection of damages in rolling bearings of an asynchronous machine. As a result, vibration

simulations were proved that neural networks can be effective agents in the diagnosis of motor bearing fault through vibration signature. Finally, in the paper [20], the motor bearing fault diagnosis was realized using deep convolution neural networks with the related analysis of the vibration signal.

Therefore, a new approach is proposed based on the use of artificial neural networks for the direct identification of the system of imbalances, avoiding the procedure of determining the matrix of the influence coefficients. The scientific and methodological approaches for the identification of mathematical models of mechanical systems, in general, are presented in the research [21].

The fundamental approach of using an artificial neural network for identification of bearing stiffness characteristics in rotor dynamics analysis was previously used for the case of solving the direct and inverse problems of rotor dynamics [22].

The design scheme of the comprehensive application of computer modeling systems and artificial neural networks for the dynamic analysis of rotor systems is presented in Fig. 8. Non-variable input parameters are colored by black, and variable input parameters – by red, as well as output parameters – by blue.

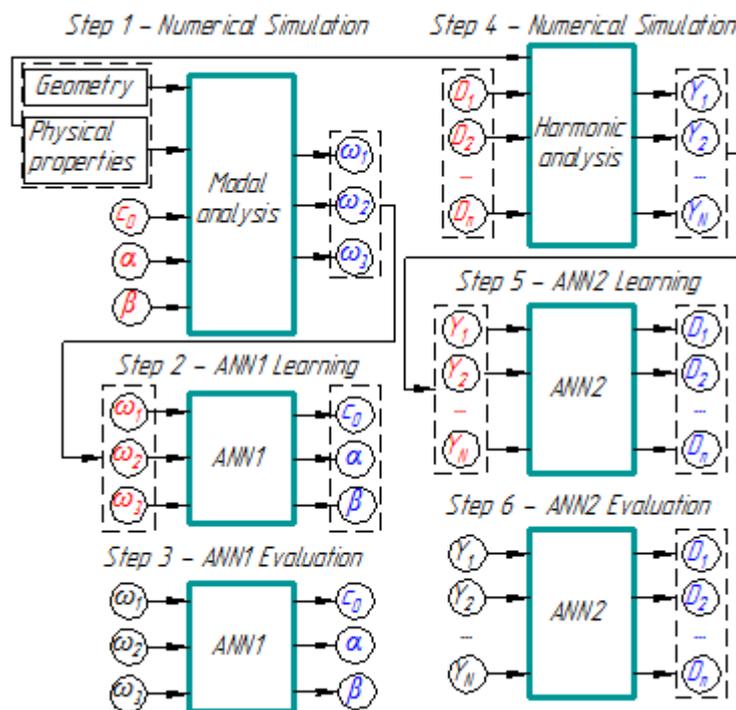


Fig. 8. The design scheme of the comprehensive implementation of numerical simulation and artificial neural networks for the dynamic analysis of the rotor.

At the first stage, modal analysis is carried out using either ANSYS software or working file "Critical frequencies of the rotor". Variation of the stiffness parameters

c_0 , a , and β for the bearing supports in a given range with the corresponding step allows obtaining an array of 3 critical frequencies ω_1 , ω_2 , and ω_3 of the rotor. This data is used at the second stage as the input data for training the artificial neural network ANN1 created, particularly, using the Visual Gene Developer software. The process of estimating the stiffness coefficients is realized at the third stage and based on the evaluation of the output parameters of ANN1 by the experimental values of critical frequencies measured on the balancing machine “Schenck” at PJSC “Sumy Machine-Building Science-and-Production Association” (Sumy, Ukraine).

At the fourth stage, harmonic analysis is conducted using either ANSYS software or working file “Forced oscillations of the rotor”. Variation of imbalances D_1, D_2, \dots, D_n (n – total number of the correction planes) in a given range with the corresponding step allows obtaining an array of dynamic deflections Y_1, Y_2, \dots, Y_N (N – total number of the planes of measurement). This data is used at the fifth stage as the source data for training the artificial neural network ANN2. The process of estimating unknown imbalances is realized at the sixth stage. It is based on the calculation of the output parameters of ANN2 by the experimental values of dynamic deflections in the planes of measurement.

5 Conclusions

In the research paper, the refined mathematical model of rotor dynamics for the multistage centrifugal compressor is developed using the 2D-beam finite element models. This model considers the dependence of the radial stiffness of bearing supports on the rotational speed. It was justified on the example of the compressor 225GC2-135/12-50M1245. The reliability of the mathematical model is confirmed by the permissible difference between the obtained results in a range of 0.3–2.7 %, as well as by observing the theorem on the mutual arrangement in the spectrum of eigenfrequencies and critical frequencies.

The obtained results indicate that the use of the working file “Critical frequencies of the rotor” implemented into the system of the computer algebra is more expedient to solve the applied problems in the field of rotor dynamics since it allows arbitrary creating the dependencies of the stiffness characteristics of bearing supports on the rotation speed. Moreover, unlike the modal analysis by the ANSYS software, the calculation using the proposed approach avoids building the Campbell diagram, as well as leads to significant increase the preparatory and machine times without loss of the calculation accuracy.

A model of forced oscillations of the rotor under the influence of the system of local imbalances is also developed. The corresponding algorithm for calculating the vibration state of the rotor is implemented using the working file “Forced oscillations of the rotor” with further improvement through the virtual balancing procedure. The accuracy of this procedure is confirmed by the verification of dynamic deflections of the rotor’s axis under the system of residual imbalances in terms of vibration reliability.

Additionally, the paper proposes the use of a general approach, which allows comprehensively applying the computational means and artificial intelligence systems. Particularly, on the example of combining the ANSYS and Visual Gene Developer software, the algorithm of identification of the stiffness parameters for bearing supports by three critical frequencies is proposed. This approach also allows evaluating the system of imbalances by the corresponding displacements of the rotor.

The obtained results will be useful for researchers in the field of computational means for numerical modeling of rotor dynamics, as well as for specialists in the implementation of the calculation intelligence systems for solving applied highly-complicated nonlinear problems in the field of rotor dynamics.

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