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NONAXISYMMETRICAL VIBRATION MODES OF ROTOR SYSTEM SIMULATION USING EXPERIMENTALLY VERIFIED SUPERELEMENTS

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A numerical method of rotor system vibrations finite-element modeling using experimentally verified superelements is presented in the paper. This technique allows numerical simulation of rotor systems with elements which can produce nonaxisymmetrical vibration modes (fan, flexible disks and etc.) in operating frequency range. Also because of substructures verification procedure, it provides a better precision of a whole rotor model. Testing of proposed mathematical modeling method is produced using special rotor test rig. The procedure of superelements creation and verification is provided. Critical speeds of test rotor system were determined numerically and experimentally. Comparative analysis of results is provided.

Keywords: rotor dynamics, finite element model reducing, superelements, nonaxisymmetrical modes, experimental verification

1. Introduction

Numerical simulation of rotor systems dynamic behavior using finite element method is a well-known way to avoid the problem of rotors dangerous vibration. For a long time, one-dimensional (1D) beam models with lumped masses were used for most problems [1, 2]. Recently three-dimensional (3D) finite-element (FEM) models are becoming more prevalent (especially for linear and symmetrical 3D models [3]). And in terms of accuracy and breadth of the described phenomena, the first approach (in case of using advanced 1D models) and the second one can be comparable [4].

However, there are some difficulties with complex rotor systems (such as rotor system of aircraft gas-turbine engines) which mechanical properties are connected with the tendency of engines mass decreasing. This tendency causes decreasing of shafts and disks stiffness, anisotropy of stator parts stiffness appearing and shift of nonaxisymmetrical vibration modes toward operating frequency range [5,6].

All this is forcing to abandon classical 1D models and move to 3D finite element calculations without cyclic symmetry. This is a most popular method to above-mentioned problems but this leads to a significant increase of finite element models dimension and increases computational time.

Increasing of model dimensions is caused by necessity to consider in the model such parts of the rotor such as a fan wheel, flexible disks and etc. These elements have a complex spatial structure and therefore require a smaller grid to correctly calculate their vibrational properties.

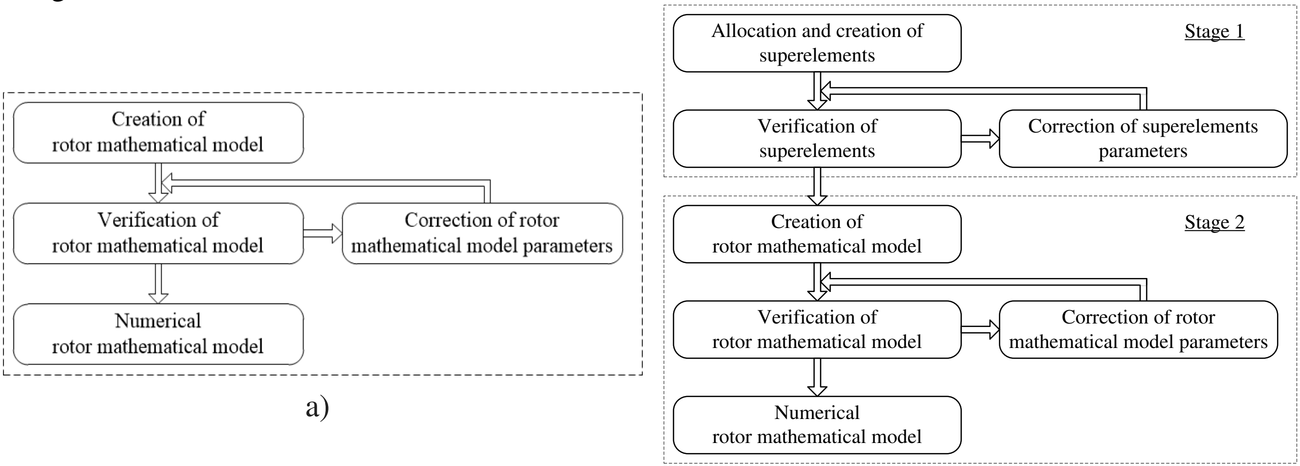
Another problem of using 3D models of whole rotor system is the problem of verification because the only way to verify critical speeds is to perform an experiment with real rotor system (see

Fig. 1a). It is always difficult because of the cost of such experiments and often useless because of obtained specter «decoding» (resonance frequencies and their sources matching) difficulty.

An effective way to solve these problems is use of superelement for modeling of parts with complex geometry. Using superelement (substructures) in the rotor dynamics calculation provides full-sized complex rotor systems with regard to their design features and operating conditions. Superelemental approach reduces the complexity of the calculations, as well as it significantly improves the accuracy of the model by allowing experimental verification of individual substructures. It significantly simplifies the process of resonance frequencies detecting and matching with their sources.

Thus, the purpose of this work is to develop a method of mathematical modeling of the rotor system vibration with pre-verified superelement which allows considering nonaxisymmetrical vibration modes of the rotor system.

Proposed modeling method is presented in Fig. 1b and includes a two-step process of model creating.

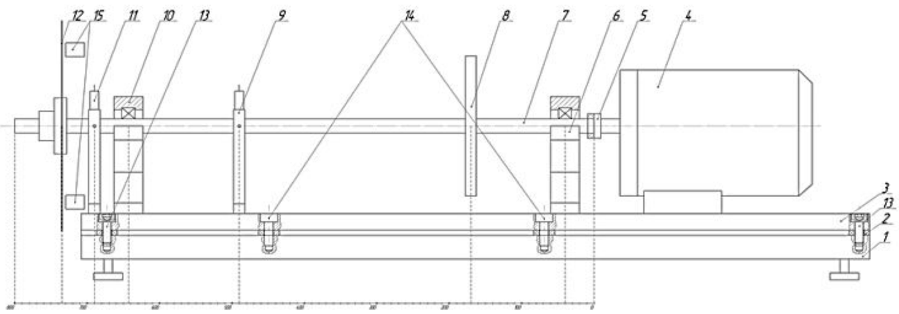


a) b)

Figure 1: Method of rotor system mathematical model creating. Common approach (a) and with pre-verified superelement (b)

2. Proposed method testing

Testing of the proposed modeling technique was carried out on a special rotor test rig simulating the low-pressure rotor of a gas turbine engine. (See. Fig. 2).



a)

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Figure 2: Diagram (a) and general view (b) of the test rig: 1 - base 2- bolted joint, 3 - base, 4 - the electric motor, 5 - flexible coupling, 6, 10 – housing of rotor supports, 7 – shaft, 8 - load rotor disk, 9,11- frame holders with displacement sensors, 12 – flexible disk, 13 - bolted joint , 14 – joints with eliminated bolts, 15 – proximity sensors for registration of flexible disk end displacements

2.1 Experimental set up

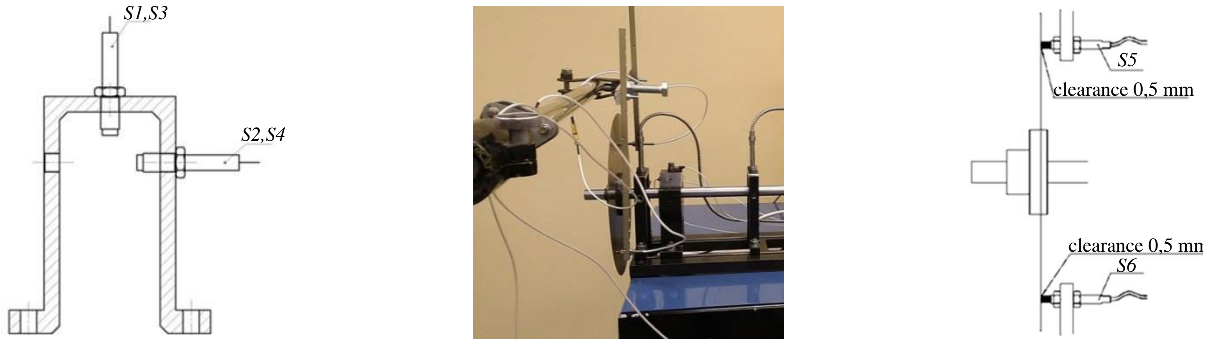
The unit is designed for practicing mathematical modelling of vibrating processes occurring in the rotor system of gas turbine engines, taking into account the structural and operational factors which are presented in them.

The rotor system which represents a flexible shaft with two supports and two disks is considered. Rotor supports includes bearings. One of disks which is used as a load, locates between the supports (size of the disk is 300 mm x15 mm). The second one, which is flexible, has larger diameter (300 mm) and much smaller thickness (1 mm). The flexible disk is installed cantilevered on the shaft. The feature of that rotor system is presence of nonaxisymmetric vibration modes with Eigen frequencies near the critical frequency.

Test rig has an experiment control and parameter registration system on the base of the National Instruments and results data processing software based on the LabView is provided. It is used to obtain following signals:

- vibration displacements of shaft in two orthogonal directions in two sections (9,11 in Fig.2)
- vibration displacements of flexible disk end displacements
- rotating speed

Numbering and location of eddy current sensors shown in Fig.3



a) b) c)

Figure 3: Positioning and numbering of the displacement sensor at the shaft (a) (frame holder) and the disk (b, c)

2.2 Creation and Verification of the Superelements

The initial phase of rotor system mathematical modelling is based on allocation and creating of superelements. Two elements of test rotor system were modelled via superelements: flexible disk with mounting elements and load disk. Model creating and numerical modal analysis was performed in SAMCEF software.

Creating a three-dimensional super element involves the build of a geometric model, material characteristics assignment, retained nodes selection, which were used for shaft model and superelement connection, FEM meshing and reduced matrices generation. It should be noted that gyroscopic moment matrices were considered in superelement model. The method of condensation is component mode synthesis (CMS).

3D model of a flexible disk superelement is shown in Fig.4. Location of retained nodes is shown in Fig. 4a and Fig. 4b. Due to the complexity of mounting elements (contacts, threaded connections, Fig. 4d) it was decided to include them in the general model of the disk and then verify it in assembly using experimental modal analysis. Finite-element model mesh before condensation consisted of 32028 hexahedral elements and 49612 nodes and after condensation it has 1 element and 1156 nodes which are placed in disk inner cylindrical surface (see Fig. 4a and 4b). Following parameters of material were taken: elastic modulus – $1.55*10^{11}$ Pa, Poisson's ratio – 0.3, material density – 7415 kg/m^3.

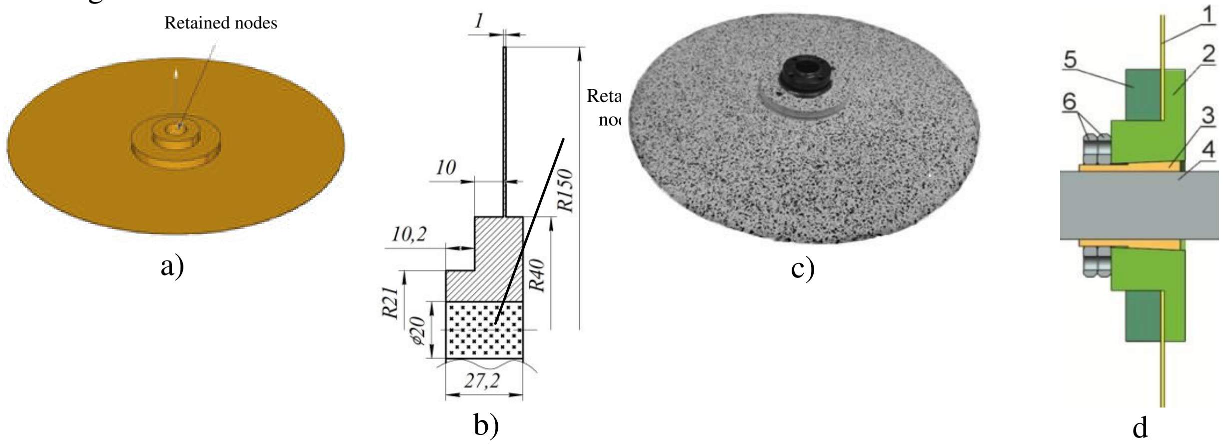
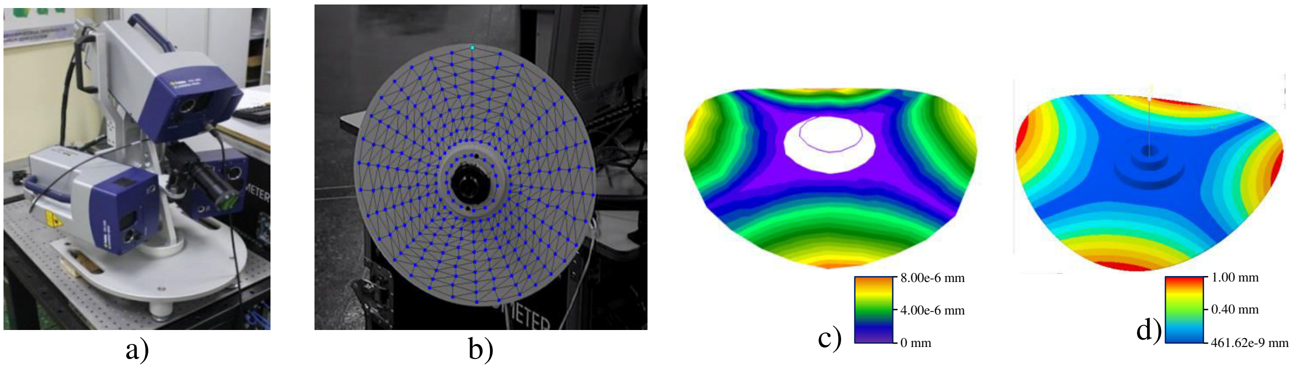


Figure 4: 3D model (a, b) of flexible disk (c) and its mounting elements (d) (1 – flexible disk, 2 – sleeve, 3 – conical sleeve, 4 – shaft, 5 – disk locking nut, 6 – locking nuts).

Verification procedure is assumed as a comparison of natural frequencies of a disc which were obtained experimentally (modal analysis) with the calculated values of natural frequencies at the same boundary conditions to the same modes of vibrations.

Experimental modal analysis was performed via three-component scanning laser vibrometry (see Fig 5a). Method of experimental determination of the natural frequencies and modes of oscillations of units using three-component scanning laser vibrometry described in [7].

To eliminate the influence of fixation stiffness as boundary conditions were considered free boundary conditions which have been realized in the experiment via elastic suspension (see Fig 5b). Scanning grid had 288 nodes. First five disk natural frequencies were determined (up to 250 Hz) and compared with numerically obtained ones according to their vibration modes (see an example of modes with two nodal diameters obtained experimentally and numerically in Fig 5c and Fig. 5d).



a) b) c) d)

Figure 5: Experimental modal analysis of a flexible disk via three-component scanning vibrometry (a - laser vibrometer heads, b – hanging disk with scan grid c - an example of modes with two nodal diameters obtained by the three-component scanning laser vibrometer and calculated mode - d)

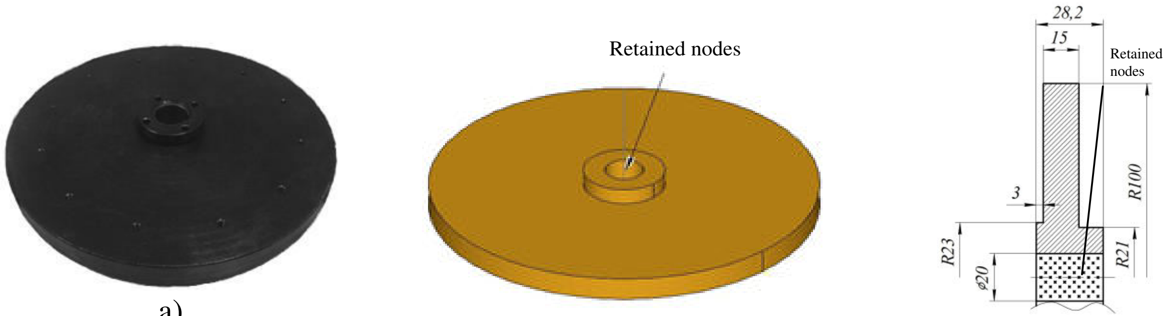
The result of the first five modes comparison is shown in Table 1.

Table 1. Natural frequencies of a flexible disk obtained numerically and experimentally

Mode #	m – nodal round quantity, n – nodal diameter quantity	Frequency, Hz		Discrepancy, %
		numerical	experimental	
1	m=0; n=0	71.14	60	15.66
2	m=0; n=2	71.06	76	6.95
3	m=1; n=1	147.45	122	17.26
4	m=0; n=3	125.42	133	6.04
5	m=0; n=4	212.45	217	2.14

The biggest difference in the frequencies obtained by calculation and experimental methods are determined for umbrella forms (modes #1 and #3, 15.66% and 17.26% respectively). On other modes (nonaxisymmetric) discrepancy was not more than 4.75%. These results can be considered suitable for following rotor system vibration characteristics determination and there is no superelement correction procedure needed.

Superelement of a load disk is created by the method described above either. Finite-element model mesh before condensation consisted of 34276 hexahedral elements and 42636 nodes and after condensation it has 1 element and 771 nodes. It was determined that the lowest natural frequency of load disk is 423 Hz and this frequency is out of test rig working speed range (0-100 Hz). So there should be no influence of load disk on vibration characteristics of whole rotor system and because of that no verification procedure is needed. Superelement and its view are shown in Fig. 6.



a) b) c)

Figure 6: Three-dimensional model of a load disk (a) created as a superelement (b, c)

2.3 Rotor mathematical model assembling

Rotor shaft was modeled as a 3D FEM model (cylinder with length 1000 mm and 20 mm diameter) made from 16,821 hexahedral elements with 17152 nodes.

The numerical model of the rotor system is shown in Fig. 7. It contains two disks simulated as superelements, rigidly attached to the shaft. Also model takes into account the anisotropy of stiffness supports and elastic connection to electric motor.

Figure 7: Numerical rotor system model: 1,5 - rotor supports, 2 - rotor shaft, 3 - load disk (superelement), 4 - flexible disk (superelement)

2.4 Numerical Determination of Rotor System Critical Speeds

Obtained finite element model was used to calculate critical speeds and vibration modes of rotor system. It was produced in SAMCEF FIELD 17.1.

Campbell diagram for the speed range from 0 to 6000 rpm is shown in Fig. 8. Eight rotor critical speeds were founded in the range. Vibration modes of the researching rotor system for the corresponding critical speeds are shown in Fig. 9.

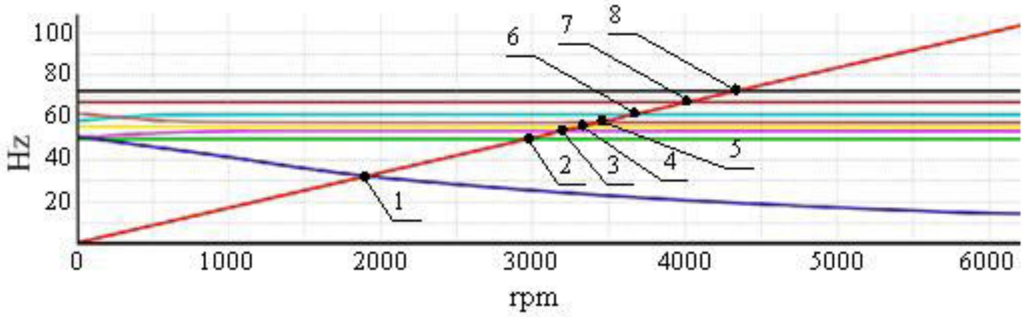


Figure 8: Campbell diagram obtained numerically. Critical regimes are marked by the dots

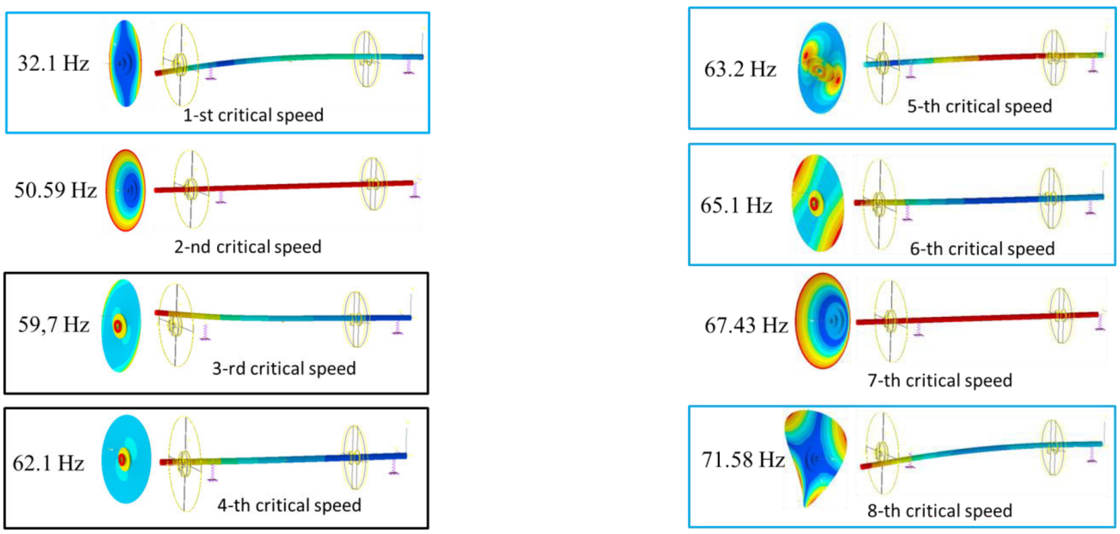


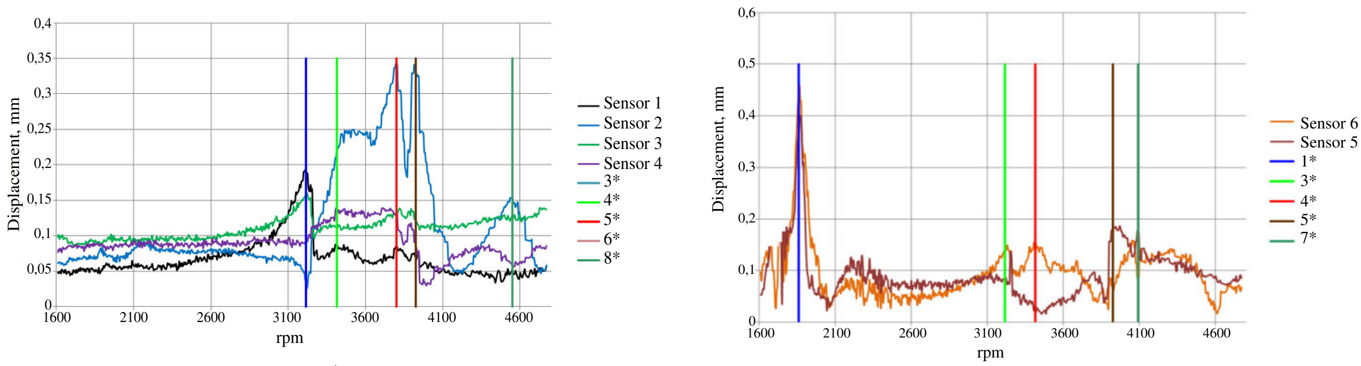
Figure 9: Critical speeds of test rotor system and corresponding vibration modes, determined numerically

There are six bending vibration modes and two axial vibration modes. The first bending mode with frequency of 31.76 Hz corresponds to flexible disk vibration mode with one nodal diameter (nonaxisymmetrical vibration mode). Next bending modes with frequencies 59.7 Hz and 62.1 Hz are "real" critical speeds (differences in frequencies are caused by anisotropy of stiffness) and they correspond to the first bending mode of shaft. #5 and #6 bending modes correspond to vibrations of cantilever part of shaft with flexible disk. Bending mode #8 corresponds to flexible disk vibration mode with two nodal diameters (nonaxisymmetrical vibration mode). The first axial mode caused by elasticity of shaft and AC motor coupling and second one corresponds to umbrella vibration mode of flexible disk.

2.5 Experimental Determination of Rotor System Critical Speeds

Rotor system critical modes were determined experimentally to verify mathematical model. For this rotor system was sped up from 1500 rpm to 4800 rpm with angular acceleration of 30 1/s2.

Amplitude-frequency characteristics of displacement sensors were averaged by three experiments. Displacement sensors (S1, S2, S3, S4) measured shaft radial direction displacement and displacement sensors (S5, S6) measured disk end displacement. Results are shown in Fig.10.



a) b)

Figure 10: Averaged over three experiments amplitude frequency characteristics obtained from displacement sensors measuring the oscillations of the rotor shaft in radial direction (a) and flexible disk end (b). *Critical regimes determined experimentally marked by lines and numbered in accordance with Fig. 8

Seven critical regimes were detected. Cantilever vibrations of shaft part with the flexible disk and anisotropy of support stiffness caused different level of vibrations for radial displacement sensors placed at the same planes (e.g. S2 and S4). Regime #1 which corresponds to vibration mode with one nodal diameter were detected by sensors S5 and S6 (Annex B, b) and didn't affect radial vibrations of the rotor. Also, sensors S5 and S6 shows that with speed increasing level of disk end vibrations decreases because of hardening due to centrifugal force.

This shows complicated vibration behavior of rotor system caused by the presence of a flexible disk.

2.6 Rotor System Model Verification

Comparison of the results of numerical and experimental determination of rotor system critical regimes is shown in Table 2.

Table 2 – Comparing the results of numerical and experimental determination of rotor system critical regimes

#	Experimental, Hz	Numerical, Hz	Discrepancy, %
1	31	32.1	3.55
2	-	50.59	
3	53.6	59.7	11.38
4	56.9	62.1	9.14
5	60	63.2	5.33
6	63.7	65.1	2.20
7	68.2	67.43	1.13
8	74	71.58	3.27

Quantitative comparison of critical speeds determined experimentally and numerically was produced. Seven experimental critical regimes of eight numerical ones were obtained in the researching range of speeds. Second critical mode wasn't obtained because of a shaft axial direction excitation absence. Discrepancy between the speeds does not exceed 6%, except mode 3 and mode 4. For those modes differences are 11.38% and 9.14% respectively.

3. Conclusion

Rotor system vibration mathematical modeling method using pre-verified superelements was developed. It allows providing the rotor system vibration characteristics researching with considering nonaxisymmetrical vibration modes of rotor system in 3D. Proposed method was tested using experimental test rig. Numerical and experimental modal analysis of researching test rotor system was provided. Natural frequencies and vibration modes were obtained.

Developed approach to finite element modeling of rotor system vibration, consisting of finite element model reduction through of using experimentally verified superelements, can be used to improve finite element methods of gas-turbine engine vibration simulating.

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