ROTOR DYNAMICS OF COMPACT GAS TURBINE UNIT WITH GAS BEARINGS INVESTIGATION

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Abstract: The aim of work is to estimate sensitivity dynamics vibration of compact gas turbine rotor supported by gas bearings to the working characteristics: unbalance, damping and temperature. Rotor dynamic model is developed within the limits of research and comprises the shaft model with the details attached, multidisciplinary model of the support with gasdynamic bearing and gas turbine casing model. Gas fluid flow calculation in bearing is based on the solution of the two-dimensional nonlinear Reynolds equation for compressible fluid. Method of successive loadings is implemented with error correction. Shaft model implemented in the calculations is based on the beam theory. Solid rotor model is used for the beam rotor model verification. With the help of this model rotor dynamics investigation is carried out in the whole range of rotating frequencies up to the operating frequency. Orbits of the rotor stationary revolution are determined by the direct integration of equations of rotor motion for particular rotation frequency values up to the operational frequency value. Spectral analysis of rotor revolution orbits is represented. Characteristic regimes of rotor operation and rotor vibration frequencies are calculated.

INTRODUCTION

Compact gas turbine units found a wide application as the energy sources in the different fields of industry. Selection of high-speed rotor support type and its structural development are one of main aspects in compact gas turbine units design. That support must provide the desired lifetime and bearing load-carrying capacity considering the restrictions applied to its overall dimensions. Slide bearings with elastic foils operating in gasdynamic lubrication regime satisfy such criteria in most cases.

Development of rotor in gasdynamic bearings high-level model lets one to chose rotor dynamic system optimal parameters for ensuring stable operating regimes or rotor rotation. Application of high-level rotor in nonlinear supports model allow to investigate sensitivity of rotor rotation to main parameters variation and also obtain results for different bearing structures. One of the main questions in the process of the methodology development is a creation of an adequate rotor dynamic model and making calculation algorithms for the gas fluid support characteristics maximum precise determination. The complexity of this problem is connected with its multidisciplinary nature. Slide bearings characteristics determination is carried out with simultaneous solving of the problem of the gas film flow in the gap, and the problem of foil and bump foil bearing elements elastic deformations. General scheme of investigation are shown in fig.1.

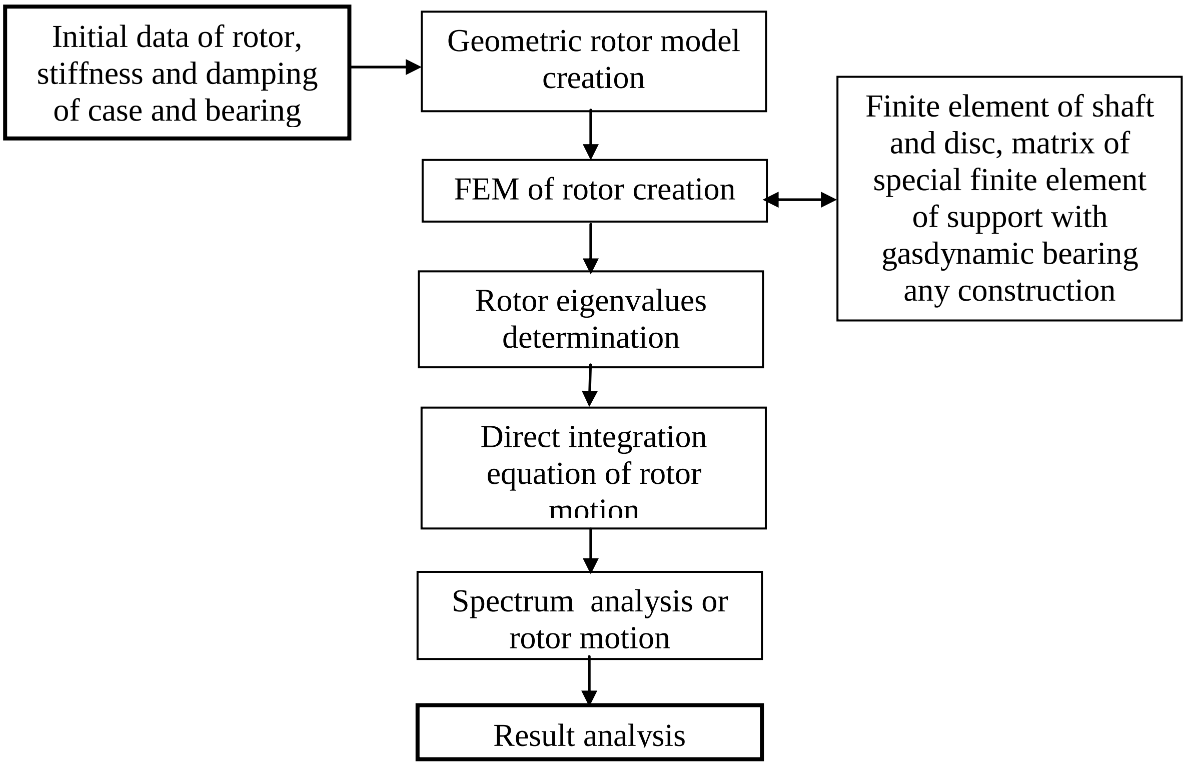


Figure.1. General scheme of investigation

ROTOR MATHEMATICAL MODEL

Rotor of compact gas turbine unit supported in gasdynamic bearings are shown on fig. 2.

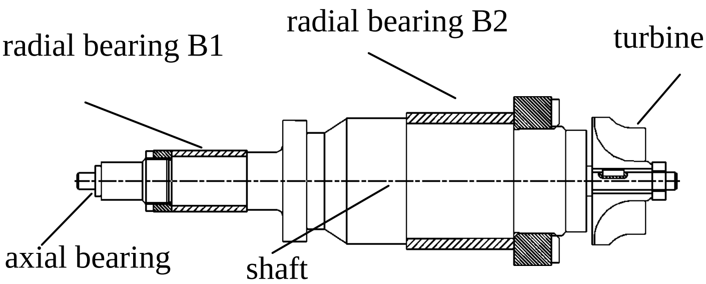


Figure. 2: Rotor in gasdynamic bearings

Multidisciplinary rotor dynamics model involving the shaft model, rotating disks model and support model is developed for the present research. Rotor finite element model is described in the stationary coordinate system by the FE matrix equations of the following form:

$$[M]\{\ddot{U}\}+\left[C\left(\{U\},\omega\right)\right]\{\dot{U}\}+\left([K_R]+\left[K_S\left(\{U\},\omega\right)\right]\right)\{U\}=\{F\} \quad (1)$$

where and - are rotor axial nodal displacement, velocity and acceleration vectors respectively; [KR] – shaft stiffness matrix, formed by the local shaft finite element stiffness matrices; [M] – mass matrix of the shaft with the details attached (discs, rotary parts of bearing and seals); [KS({U}, ω)] – stiffness matrix, considering contribution from supports and seals and formed by the local seal and support stiffness matrices; [C({U}, ω)]= [C1({U}, ω)] +□{[KR]+[KS({U}, ω)]} – damping matrix, considering matrix of gyroscopic moments [C1({U}, ω)] and structural damping of rotor casing and supports with coefficient of proportionality □; {F} –external forces vector; ω – angular speed.

Mass [M] and stiffness [KR] matrices are constant in the process of rotor revolution at the predefined frequency. In contrast matrices [KS] and [C] coefficients depend not only on the rotor speed but also the deformed shaft axis position with respect to the supports. Matrix [KS] coefficients are determined by the bearing and seal stiffness parameters, wheels aerodynamic forces and gas turbine unit casing stiffness. Gyroscopic moments, working medium (gas or air) fluid parameters in the seal devices, friction in journal bearings affect damping matrix [C] coefficient values. Thus [KS] и [C] are the matrices of the general form with the coefficients nonlinearly depending on the current shaft axis position, shaft speed and lubricant parameters.

Equation (1) is obtained for the arbitrary rotor FE model. Inertia characteristic of the details attached to the shaft are taken into account by the concentrated mass finite element in the beam model. The complex approach is implemented in the rotor solid model – rotary parts are modeled by the solid elements and in the case this way is inapplicable by the beam and concentrated mass finite elements.

Solid rotor modeling requires large PC resources. In this case the solution of every static problem takes considerably more time in comparison with analogous solution by the beam model. So the implementation of solid rotor models in the transient rotor dynamics is very resource-intensiveness, especially for the high-speed rotors, because in that case it is necessary to solve from 10 to 1000 static problems for the good quality orbits computation. In order to keep the realistic solving time the beam rotor model is implemented in the present work. Solid rotor model is necessary for modeling of rotor parts interaction in the process of its assembly, its speed-up up to the operational speeds and determination of natural frequencies. The results obtained by the solid models are used for the beam model update and verification, that lets one to use the highly precise “fast” beam model in further modeling.

The system of equations of motion (1) is solved by means of Newmark integration scheme [1], the iterative refinement of the stiffness and damping matrix coefficients is performed at each time step.

SUPPORT MODEL

The hypothesis of consecutive insertion of elastic elements in unified calculation model is used for the support model development. These elements simulate elastic properties of gas layer, foil bearing and support casing in gas turbine power unit. Thus, a special finite element stiffness matrix of support included in matrix:

$$[K_{\sup}] = \left([K_b]^{-1} + [K_c]^{-1}\right)^{-1}$$

where [Kb] is a stiffness matrix of gas layer that considers deformations of foils in bearing; [Kc] is a support case stiffness matrix. In order to define matrix [Kb] coefficients the multidisciplinary elastohydrodynamic contact problem that includes the problem of foils deformation in bearing and the problem of the gas flow in the gap must be solved. Calculations of the bearing working surfaces deformation are performed for the complex bearing structures with different foil shapes having different stiffness. Matrix [Kc] coefficients are determined in calculation of support case structure.

FOIL GASDYNAMIC BEARING MODEL

Foil gasdynamic bearing structure, shown in fig. 3 comprises five lobes formed by top and bump foils each. Pressure distribution in the gap between the top foil, bearing sleeve and shaft journal and foils elastic deformations, corresponding to it, determine main support characteristics such as the relationship between carrying force and shaft journal position in bearing, stiffness and damping parameters for the gas film – elastic foils system.

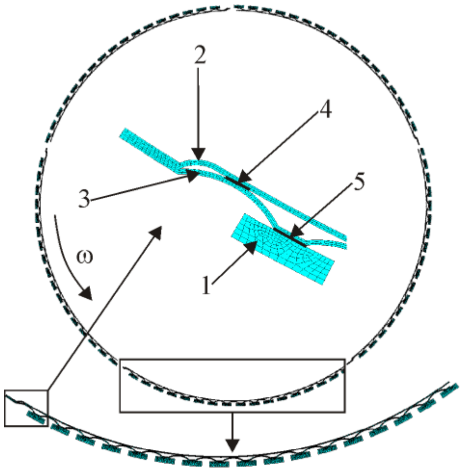


Figure 3: Bearing FE model, 1 – bearing race; 2 – top foil, 3 – bump foil, 4 – contact zones 1, 5 – contact zones 2

Therefore determination of the gasdynamic bearing characteristics requires the coupled solution of the gas film flow problem in the gap, provided by shaft journal and top foils, and the determination of top and bump foils radial deformation contribution into the gap change under the action of the gas film pressure.

Gas flow between journal bearing surfaces is described by the nonlinear Reynolds equation for the compressible fluid which is solved numerically using finite-element discretization procedure. General schematics of the foil gasdynamic bearing characteristics determination problem with foil deformations taken into account and model results verification by comparison with models and experimental data from other authors [2, 3] are prescribed in [1]. Models from [1] are used in the present work for determination of bearing parameters with consideration of elastohydrodynamic contact interaction between gas film and foils, between foils and bearing sleeve, with the application of the self-correcting algorithm [4]. Contact interaction between foils and bearing sleeve is modeled by the special contact finite elements in bearing finite element model shown in fig.3. Calculations are carried out for the relative eccentricity values χ = 0.1–0.8 and the various directions of the shaft journal displacements described by the angle. Iterative process of the coupled solution of the gas flow and the elastic element deformation problems is realized for each eccentricity and direction value.

BEARING CALCULATION RESULTS

Foil bearing film thickness and pressure distributions received on the base of 2D Reynolds equation model are shown in fig. 4. As it can be seen from the foil bearing gap calculation results with the bump foils taken into account, the gap in the pressurization area increases due to the top foil deflections between the bumps. This effect is greater at higher shaft journal eccentricities and is considerable for 0.8 eccentricity ratio. Cumulative bearing characteristics (dimensionless carrying force [Q] and the angle φ between the directions of the carrying force and shaft journal displacement) are shown in fig. 5 versus eccentricity ratios for different shaft journal displacement directions.

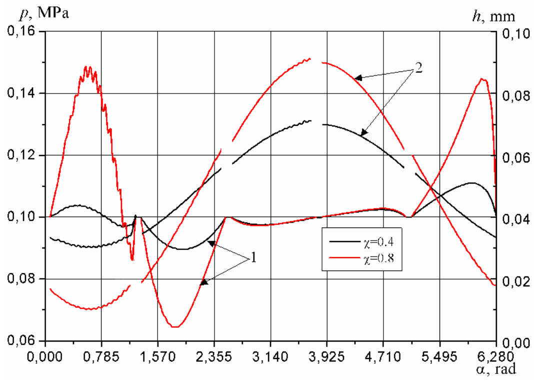


Figure 4: Film thickness and pressure at the bearing midplane for different shaft journal displacement directions: 1 – pressure, 2 - clearance

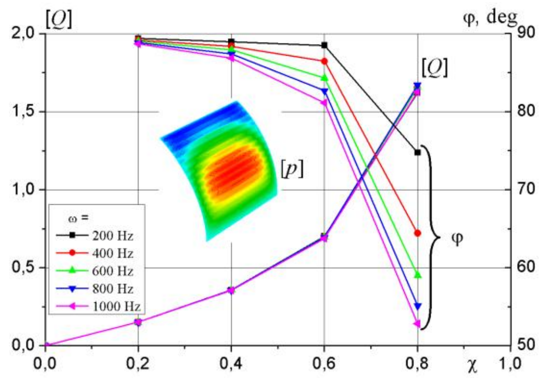


Figure 5: Carrying force and bearing attitude angle versus different eccentricity ratios and eccentricity directions

Bearing foils flexibility leads to the nonuniform stress distribution in bump and top foils. The top foils is pressed to the bump foil in the pressurized area (fig. 6b and 6c) and maximum bending stresses produced in top foil are about 50 MPa (fig. 6c).

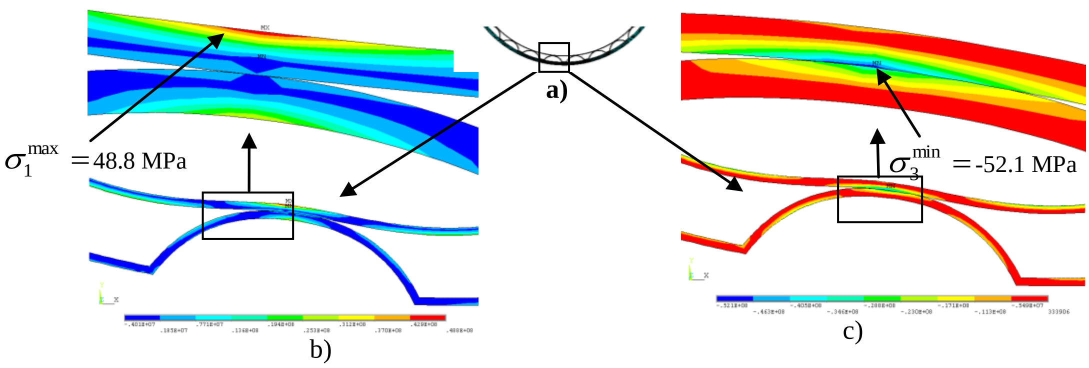


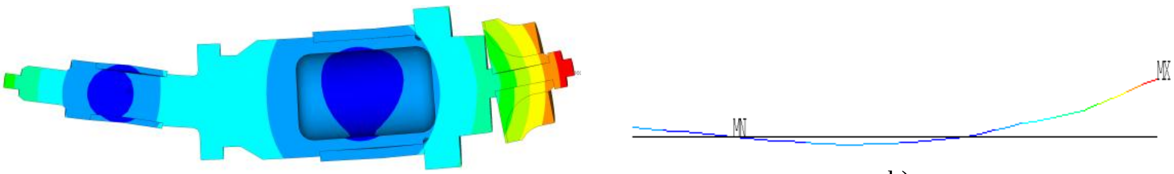
Figure 6: Foil gasdynamic bearing: a) model; b) the first principal stress in the pressurized area; c) the third principal stress in the pressurized area

ROTOR DYNAMICS SIMULATION RESULTS

Investigation of the foil gasdynamic bearing supported gas turbine unit rotor (fig. 2) is a complex problem that comprises, as it is considered in present paper, the problem of the high speed rotor assembly stress-strain state calculation under the action of centrifugal loads, its natural frequencies and modes determination and nonlinear rotor dynamics numerical simulation depending on structural damping, imbalance of the rotating parts and support temperature state.

Rotor assembly stress-strain analysis at its speed-up up to the maximum operating speeds is carried out on the base of the axisymmetric model with rotor assembly elements contact interaction taken into account. Centrifugal load is applied step-by-step assuming the rotor is absolutely balanced and has a zero imbalance value.

Rotor beam model (fig. 7b) is corrected with respect to the solid model natural frequencies and modes (fig. 7a). Fictional finite elements, modeling the bolts, contact zone are implemented. Also the equivalent masses of the structural members are selected. Taking all this aspects into account in the beam rotor model leads to the good agreement of both models for the first three natural frequency values. That lets one to use the beam model that is equivalent to solid model.



a) b)

Figure 7: The first rotor vibration mode for different FE models: a) solid model, f = 1482 Hz; b) beam model, f = 1450 Hz

Orbits of rotor rotation in supports for rotational frequency range from 100 Hz up to 1000 Hz are determined for varied rotor imbalance values MR, coefficient of structural damping β and gas lubrication viscosity η in supports. Rotor system parameters for different computational cases presented in table, and results in fig. 8 and fig. 9.

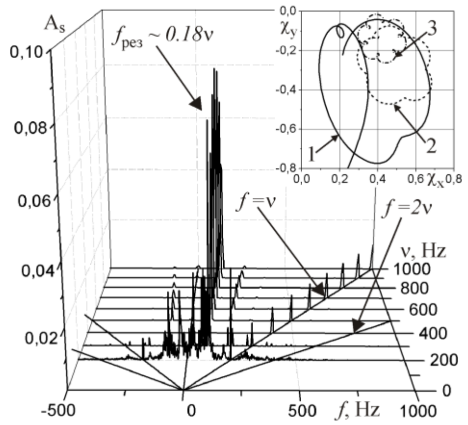
5-lobe foil bearing			
№	Initial unbalance value MR, kg·cm	Coefficient of structural damping β	Lubrication viscosity η, kg/(m·s)
1	0.0005	0,004	$1.85 \cdot 10^{-5}$
2	0.0015	0.004	$1.85 \cdot 10^{-5}$
3	0.002	0.004	$1.85 \cdot 10^{-5}$
4	0.002	0.003	$1.85 \cdot 10^{-5}$
5	0.002	0.004	$3.62 \cdot 10^{-5}$

Table 1: Rotor in foil bearings calculation parameters

Rotor orbits and full spectrum results for structural damping β = 0.004 for initial imbalance value in range from 0.0005 kg•cm to 0.002 kg•cm are presented in fig. 8. For initial imbalance value 0.0005 kg•cm rotor orbits in the support B1 sufficiently differ from circular shape, and the presence of subharmonics with large amplitudes is indicated in the full spectrum. Vibration amplitudes reach their maximum values at the rotational frequency values close to the support natural frequency and decreasing with the rotational frequency increase. For initial imbalance value 0.0015 kg•cm sufficient rotor vibrations exist only for rotor frequency values close to the support natural frequency and then decreasing with the rotational frequency increase. Rotor orbits in the support B1 for rotational frequency values above the support natural frequency have circular shape and the rotor vibrates at the rotational frequency. Further increase of the initial imbalance value up to 0.002 kg•cm provides higher vibrations in the field of the support natural frequency. Rotational frequency increase provides subharmonics dissipation and rotor vibrations at the rotational frequency. Rotor orbits have circular shape for the rotational frequency values above 400 Hz. Initial imbalance increase in the specified range leads to the elimination of the subharmonic vibrations in rotor orbits full spectrum characteristic.

Consideration of the simulation case with the structural damping being decreased down to $\beta = 0.003$ for the initial imbalance value 0.002 kg•cm (fig. 9a) leads to the indication of the great subharmonic vibrations at the support natural frequency in the wide range, as it has been received in the similar manner for the imbalance value of 0.0005 kg•cm (fig. 8a) but with greater amplitudes. Support temperature increase and corresponding gas dynamic viscosity increase provide the presence of the subharmonic vibrations zone at the support natural frequency for the rotational frequency range from 400 Hz up to 800 Hz. In the remaining range subharmonic vibrations do not exist and rotor orbits at low and high rotational frequencies case have a circular shape in that case.

Orbits and their spectral expansion are represented in terms of the dimensionless eccentricity ratio χ varying in the range form -1 to 1.



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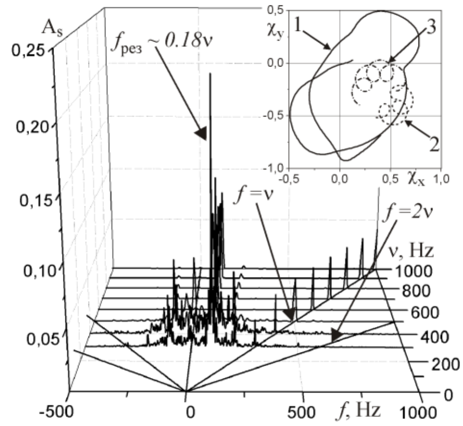

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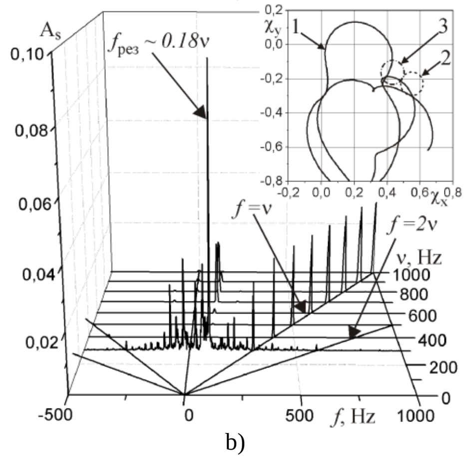

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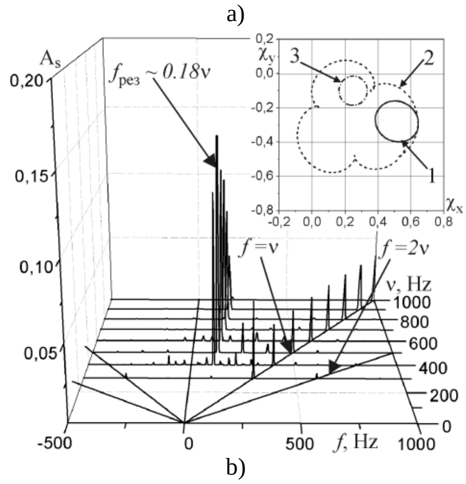

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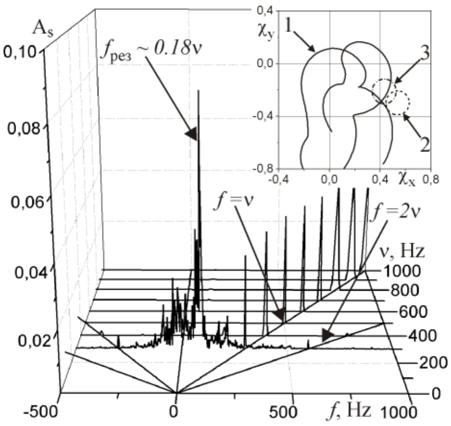

c)

Figure 8: **S**haft orbit and full spectrum:
1 – 200 Hz, 2 – 600 Hz, 3 – 1000 Hz:
a) case 1; b) case 2; c) case 3

Figure 9: **S**haft orbit and full spectrum: 1
1 – 300 Hz, 2 – 600 Hz, 3 – 1000 Hz:
a) case 4; b) case 5

REFERENCES

[1] Temis J.M., Temis M.J., Mescheryakov A.B. (2007): Elastohydrodynamic Contact Theory in Foil Gas Bearing. Proc. 4rd Int. Symp. on Stability Control of Rotating Machinery. Calgary, Canada, pp. 228-238.

[2] . Peng Z.-C., Khonsari M.M. (2004): Hydrodynamic Analysis of Compliant Foil Bearings with Compressible Air Flow. J. of Tribology, Trans. ASME, Vol. 126, pp. 542-546.

[3] Heshmat H., Walowit J., Pinkas O. (1983): Analysis of Gas Lubricated Journal Bearing. J. Lubr. Tech. Trans. ASME, Vol. 105, № 4, pp. 647-655.

[4] Temis J.M. (1980): Selfcorrection Step by Step Method for Solution of Nonlinear Elasticity and Plasticity Problems. CIAM Proceedigs, № 918, 24 p. (in Russian).