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Comprehensive Approach for Identification of Nonlinear Stiffness Characteristics of Bearing Supports for the Oxidizer Turbopump of the Liquid Rocket Engine

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Abstract. This article dials with the refinement of the mathematical and computational models of the oxidizer turbopump rotor considering bearing gaps, axial preloading, compliance of the housing parts and the effect of rotation. The loading scheme consists of four substeps is proposed considering preliminary displacement of the outer cage, axial displacement as a result of the support deformation due to the axial preloading force, radial displacement due to the support deformation, as well as centrifugal forces of inertia caused by rotation of the rotor with an inner cage. Modelling of contacts interactions using ANSYS software is carried out according to the appropriate models of contact behaviour. The contact areas between the rolling elements, inner and outer cases are obtained. The contact angle is determined. Isosurfaces of axial and radial displacements for the bearing supports are built. Nonlinear stiffness of bearing supports is determined as the tangent of the angle of inclination for the curve "radial load – radial displacement". The proposed approach, which used for designing turbopump units for liquid rocket engines, will allow refining the reliable mathematical and computational models of rotor dynamics for turbopump units and providing appropriate computer simulation of forced oscillations of the rotor systems for given permissible residual imbalances considering nonlinear stiffness characteristics of bearing supports.

Keywords: compliance of housing, initial clearance, axial preloading, radial load, contact angle, radial stiffness, axial stiffness.

1 Introduction

The creation of reliable models of rotary systems is an urgent problem that allows designing high-power rotary machines including turbopump units of liquid rocket engines. The process of modelling should be based on the existing experience in designing of reliable equipment the requirements for which are permanently increased.

Based on the experience of the Faculty of Technical Systems and Energy Efficient Technologies (Sumy State University, Ukraine) in mathematical modelling of rotary systems, and the Faculty of Mechanical Engineering (University of West Bohemia, Czech Republic) in providing numerical simulations, as well as the Faculty of Manufacturing Technologies with a seat in Prešov (Technical University of Košice, Slovakia) in carrying out experimental research, this paper is devoted to refine the computational model of rotor dynamics for turbopump units and further computer simulation of forced oscillations of the rotor systems for given permissible residual imbalances considering the nonlinear stiffness of bearing supports.

The proposed clarifications are related to effect of rotation on deformation of moving parts of bearings, compliance of the housing, as well as gaps and axial preloading of the bearing supports.

The clarification of the stiffness parameters of the bearing supports is carried out by a combination of 2D and 3D finite element models using up-to-date computational means.

The proposed approach will allow refining the reliable mathematical and computational models of rotor dynamics for turbopump units and providing computer appropriate simulation of forced oscillations of the rotor systems for given permissible residual imbalances considering nonlinear stiffness characteristics of bearing supports.

2 Literature Review

The problem of identification of the nonlinear bearing stiffness is highlighted in recent research works. Particularly, the paper [1] dials with the investigation of nonlinear reactions in rotors' bearing supports of turbopump units for liquid rocket engines. However, this work does not consider the impact of initial bearing gap and axial preloading.

The problem of dynamics and diagnostics of vertical rotors with nonlinear supports stiffness is solved in the doctoral thesis [2].

Clarifications considering the impact of gap seals with floating rings are presented in monograph [3]. Additionally, it is shown that the presence of gaps in the bearing supports reduces the bearing stiffness. However, gap seals with floating ring, axial preloading and rotation of the rotor cause an increasing dependence of the bearing stiffness on the rotor speed. Consequently, the critical frequencies of the rotor increase.

A review of investigations of nonlinear dynamic on bearings with rolling element is presented in the research work [4].

The calculation of forced oscillations under the system of imbalances (direct synchronous precession) performed using the computer program [5] considers the dependences of the bearing stiffness on the rotor speed and impact of the gyroscopic moments of inertia of bushing parts.

Paper [6] is aimed at theoretical and experimental study of spindle ball bearing nonlinear stiffness.

General scientific and methodological approach for the identification of mathematical models of mechanical systems using artificial neural networks is delivered in paper [7].

Nonlinear dynamic response for the system "cylindrical roller bearing – rotor system" is presented in the paper [8]. The proposed mathematical model with 9 degrees of freedom allows considering combined localized defect at inner–outer races of bearings.

The need to consider clarified mathematical model of rotor dynamics for investigation of critical frequencies considering stiffness of bearings and seals is justified in the work [9] on the example of on examples of the centrifugal compressor's rotor.

The influence of bearing stiffness on the nonlinear dynamics of a shaft-final drive system is presented in the research paper [10].

3 Research Methodology

The connection of shafts of the turbopump unit causes a weak dynamic interaction between them. In this case, the partial critical frequencies of separate rotors do not differ much from the corresponding frequencies, obtained as a result of calculations for the entire rotor. This fact is confirmed by the calculations presented in the work [2]. Therefore, the oxidizer turbopump and fuel pump rotor systems should also be considered as the separate dynamic systems. The design scheme of the oxidizer turbopump rotor is presented in Figure 1.

The ANSYS software with the modules "Static Structural" and "Transient Structural" is used for determining the bearing stiffness considering initial gaps, axial preloading, rotation effect and compliance of the housing. The basic design schemes for loading bearing supports are presented in Figure 2.

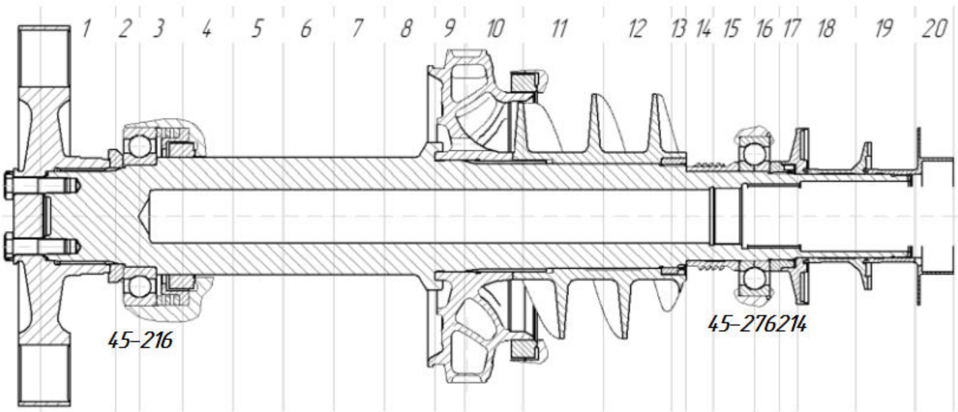

Figure 1 – Design scheme of the oxidizer turbopump rotor

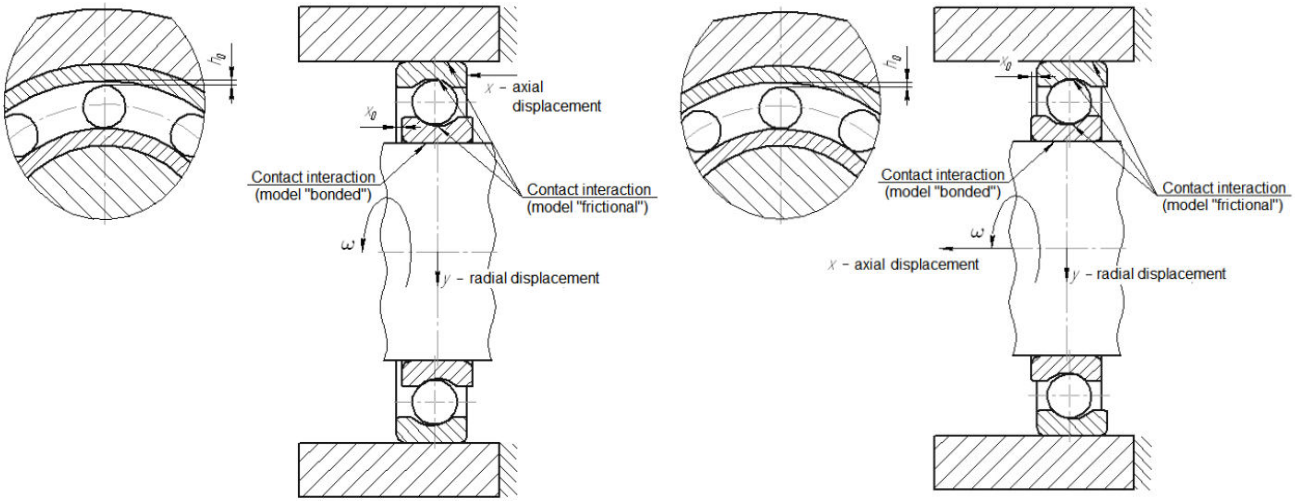

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Figure 2 – Design scheme of the bearing supports 45-216 (a) and 45-276214 (b)

The loading scheme consists of four substeps:

1. Preliminary displacement of the outer cage (for the bearing support 45-216) and the shaft (for the bearing support 45-276214) towards the application of axial pre-loading force. Axial displacement x_0 is determined due to the maximum radial gap h_0 = 0.095 mm [11].

2. Determination of the axial displacement x as a result of the support deformation due to the axial preloading force T = 4,5 kN [11]. Investigation of forced oscillations of the rotor on ball bearings.

3. Numerical calculation of the radial displacement y of the shaft axis as a result of the support deformation due to the radial force R = 10 kN.

4. Considering centrifugal forces of inertia caused by rotation of the rotor with an inner cage of the bearing support.

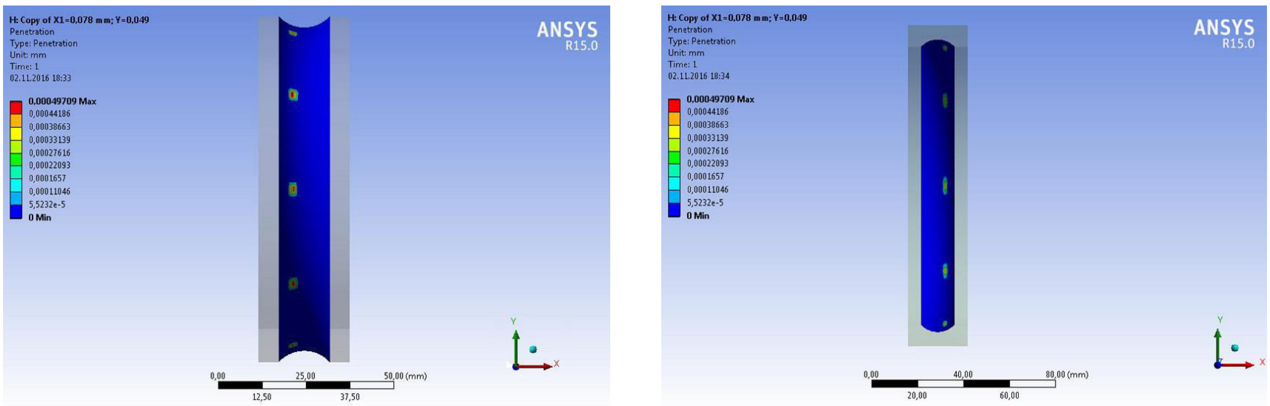
Modelling of contacts by using ANSYS software is carried out according to Table 1.

Table 1 – Models of contact interaction between the mating surfaces

Mating surfaces		Contact model
Shaft	Inner cage	“bonded”
Inner cage	Rolling elements	“frictional”
Rolling elements	Outer cage	
Outer cage	Housing	

The contact areas between the rolling elements and cases are shown in Figure 3. The loading schemes accog-ding to substeps 2–4 of bearing loading are presented on Figures 4–6.

Isosurfaces of axial and radial displacements for the bearing supports are presented on Figures 7–8.



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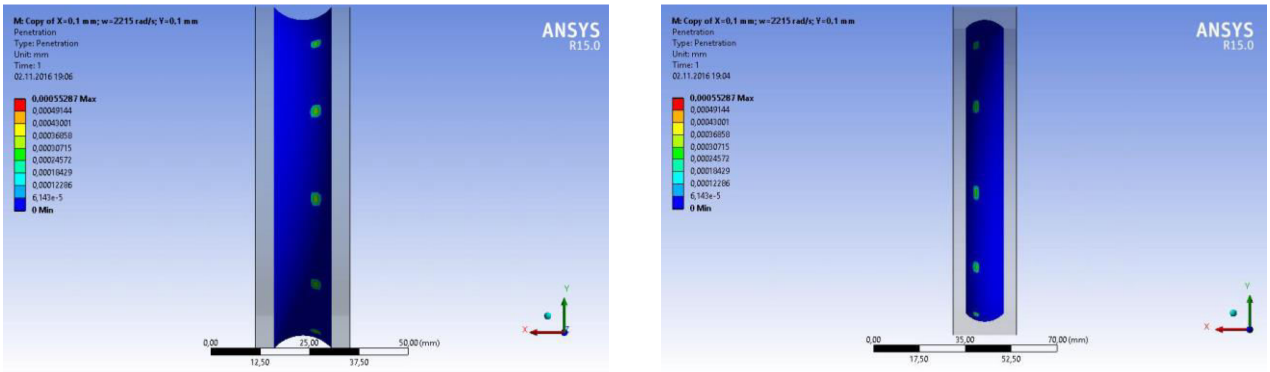

c d

Figure 3 – Contact areas between the rolling bodies, inner (a, c) and outer (b, d) cages
for the bearin supports 45-216 (a, b) and 45-276214 (c, d)

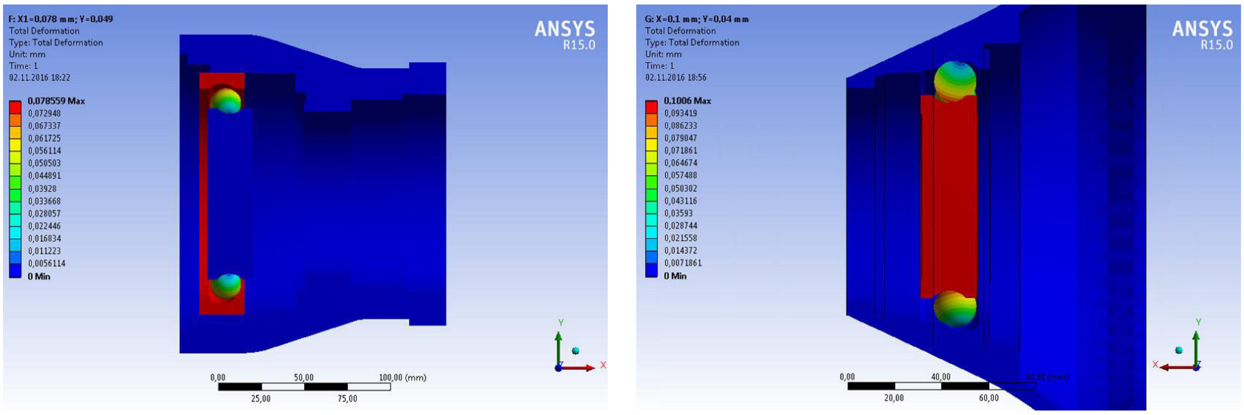

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Figure 4 – Deformations of the bearing supports 45-216 (a) and 45-276214 (b) as a result of axial preloading

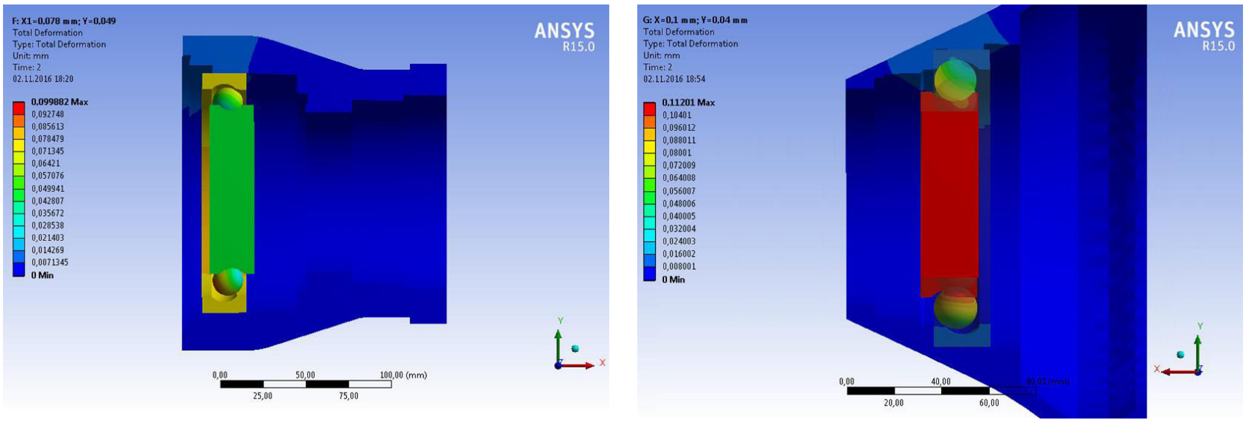

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Figure 5 – Deformations of the bearing supports 45-216 (a) and 45-276214 (b) as a result of axial preloading and radial force

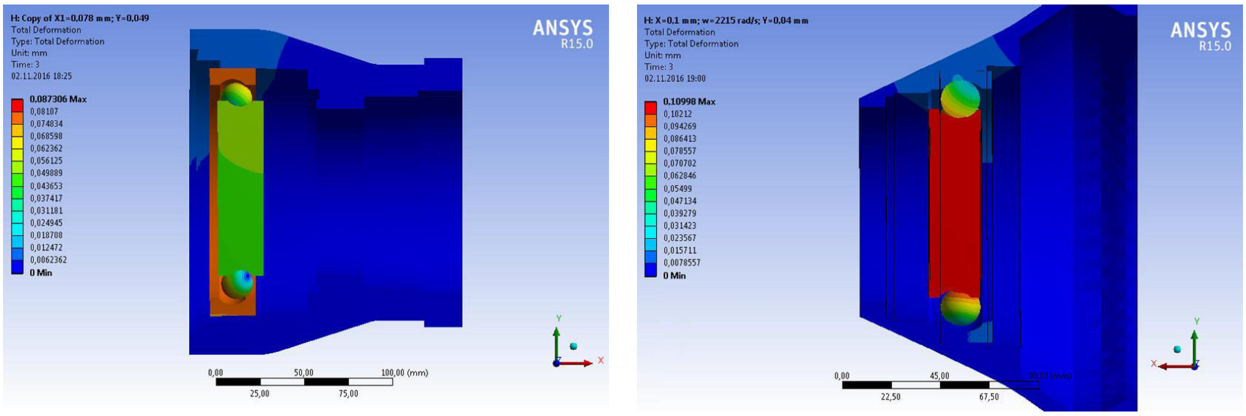

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Figure 6 – Deformations of the bearing supports 45-216 (a) and 45-276214 (b) as a result of axial preloading, radial force and rotation of the rotor

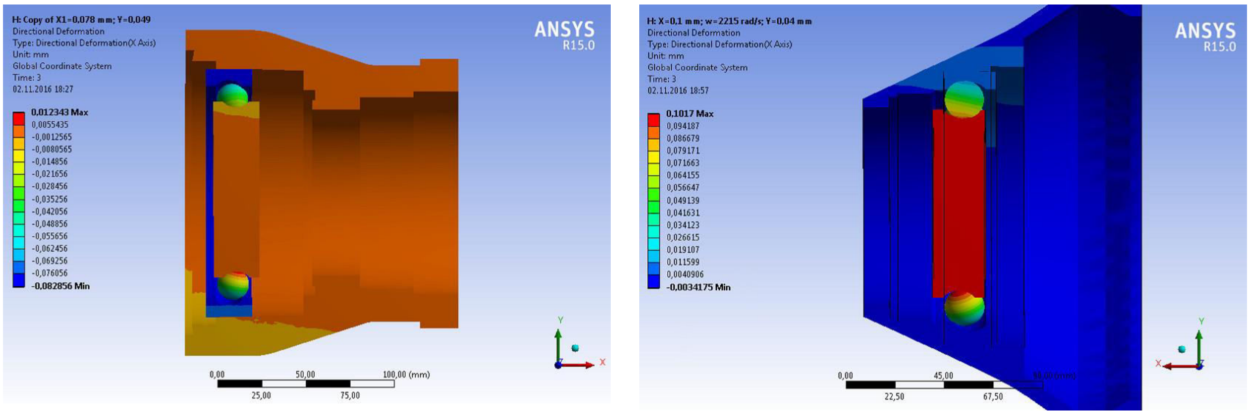

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Figure 7 – Isosurfaces of axial displacements for the bearing supports 45-216 (a) and 45-276214 (b)

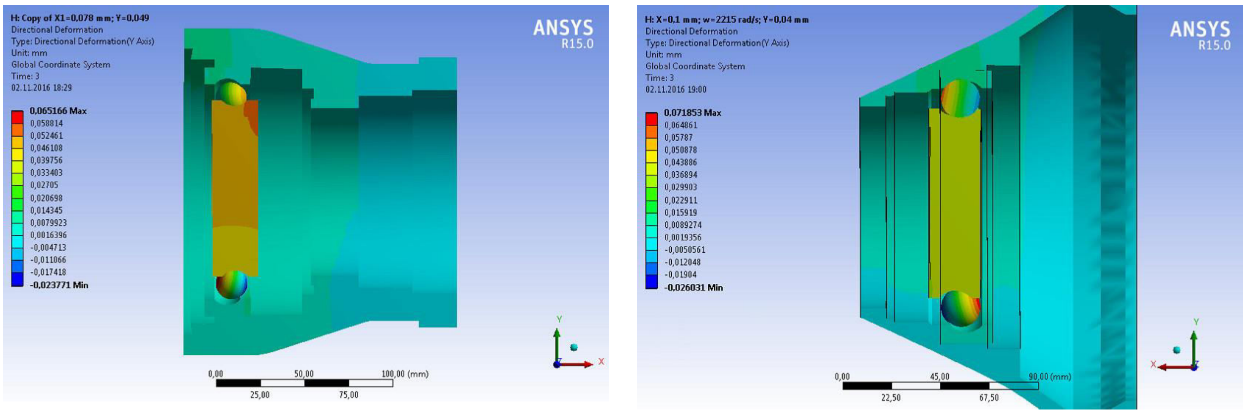

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Figure 8 – Isosurfaces of radial displacements for the bearing supports 45-216 (a) and 45-276214 (b)

4 Results

The results of numerical simulation are summarized in Table 2.

The stiffness c of the bearing support is determined due to the following formula:

$$c = \partial R / \partial y, \quad (1)$$

where R – radial force; y – radial displacement.

As a result of numerical simulation (the determination of radial displacements of the rotor axis under the discrete values of the radial force), the approximated curves “radial load – radial displacement” are defined (Figure 9).

Analytical expressions describing the dependence “radial load – radial displacement” determined as a result of approximation of the experimental data are summarized in Table 3.

Table 2 – The results of numerical simulation

Bearing	Displacement, μm						
	h_0	x_0	x	Rotor speed, 10^3 rad/s			
				0	1.1	2.0	2.2
45-216	95	620	80	51	50	48	47
45-276214	95	620	100	40	40	39	39

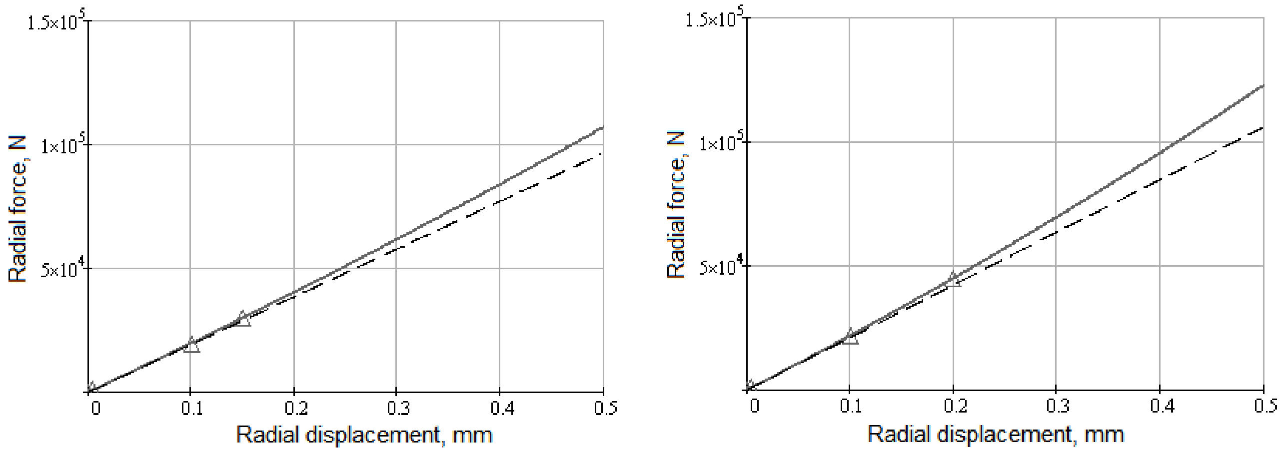

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Figure 9 – Dependence “radial force – radial displacement” for the bearing supports 45-216 (a) and 45-276214 (b)

Table 3 – Analytical dependences “radial load – radial displacement”

Bearing support	45-216			45-276214		
Radial displacement y, μm	5	100	150	5	100	200
Radial force R, kN	0.9	1.95	30	1	22	45
Approximated curve (R, N; y, m)	$R(y) = 1.94\cdot10^8\cdot y + 4.0\cdot10^{10}\cdot y^2$			$R(y) = 2.12\cdot10^8\cdot y + 6.7\cdot10^{10}\cdot y^2$		
Nonlinear radial stiffness (c, N/m; y, m; R, N)	$c(y) = 1.94\cdot10^8 + 8.0\cdot10^{10}\cdot y$			$c(y) = 2.12\cdot10^8 + 1.34\cdot10^{11}\cdot y$		
	$c(R) = 1.94\cdot10^8\cdot(1 + 4.2\cdot10^{-6}R)^{0.5}$			$c(R) = 2.12\cdot10^8\cdot(1 + 6.0\cdot10^{-6}R)^{0.5}$		

The linear radial stiffness of the support is defined as the tangent of the initial angle of inclination of the curve “radial load – radial displacement” (Figures 9, dash line):

$$c_0 = \left(\frac{\partial R}{\partial y}\right)_0 . \tag{2}$$

Taking into account the expressions given in Table 2, the values of the bearing stiffness c_0 of the supports 45-216 and 45-276214 are equal $1.94\cdot10^8$ N/m and $2.12\cdot10^8$ N/m respectively. Exceeding the bearing stiffness of the support 45-276214 in comparison with the bearing support 45-216 is explained by the relatively large number of rolling bodies.

Similar values of the bearing stiffness, determined without the rotor speed for the supports 45-216 and 45-276214 are equal $1.88\cdot10^8$ N/m and $2.10\cdot10^8$ N/m respectively.

For further designing the mathematical models of free and forced oscillations of the rotor systems for turbopump unit considering the impact of the rotor speed on bearing stiffness, the following analytical dependence is proposed:

$$c(\omega) = c_0 + \alpha\,\omega^2 . \tag{3}$$

In this case, the estimation of the coefficient α is carried out by the linear regression formula [1]:

$$\alpha = \frac{\sum_{k=1}^{3}(c_k - c_0)\omega_k^2}{\sum_{k=1}^{3}\omega_k^4}, \tag{4}$$

where c_k – bearing stiffness, determined as a result of the numerical simulation for the rotor speed ω_k (Table 2); k – number of the experimental point.

As a result, values of the coefficient α are obtained (Table 4), as well as the approximated curves are built (Figure 10).

The obtained data allows determining the dependence of the bearing stiffness on the radial force (Figure 11).

Table 5 and Figures 12–13 contain the data presenting the axial stiffness of the bearing supports.

Table 4 – Parameters of the nonlinear bearing stiffness

Bearing support	Parameter	
	c_0, N/m	α, N·s^2/m
45-216	1.88	1.223
45-276214	2.10	0.408

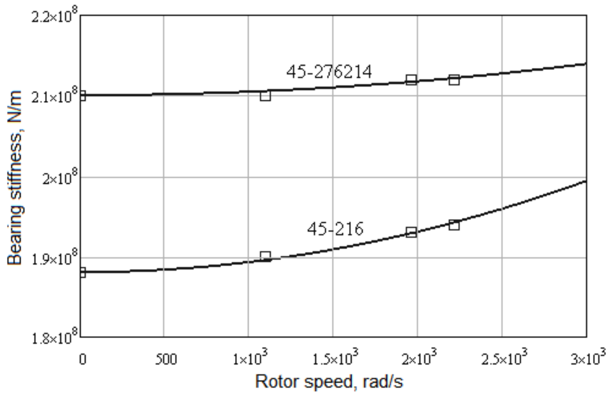


Figure 10 – Dependence of the bearing stiffness on the rotor speed

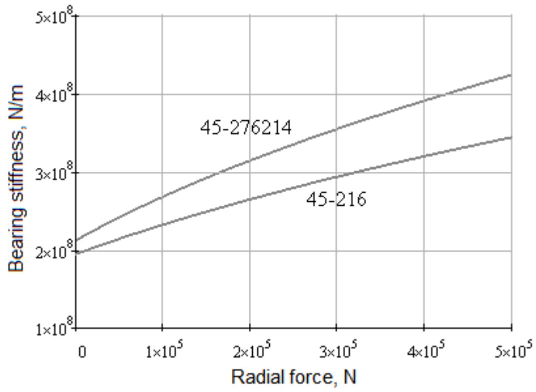
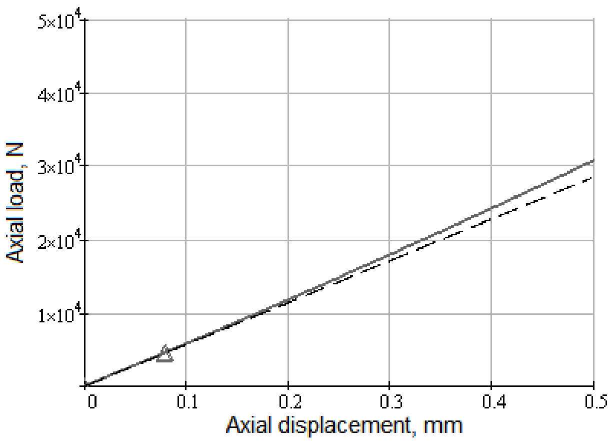


Figure 11 – Dependence of the bearing stiffness on the radial force



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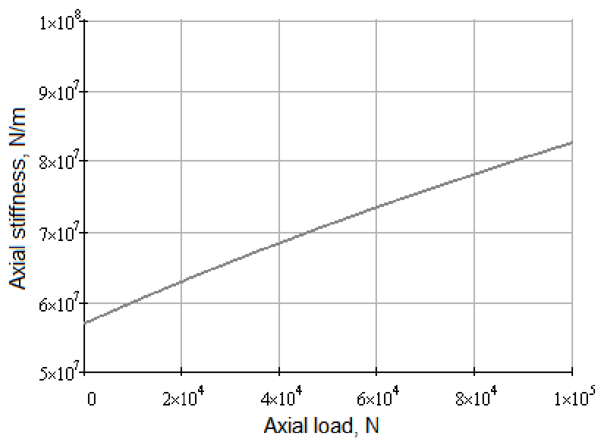

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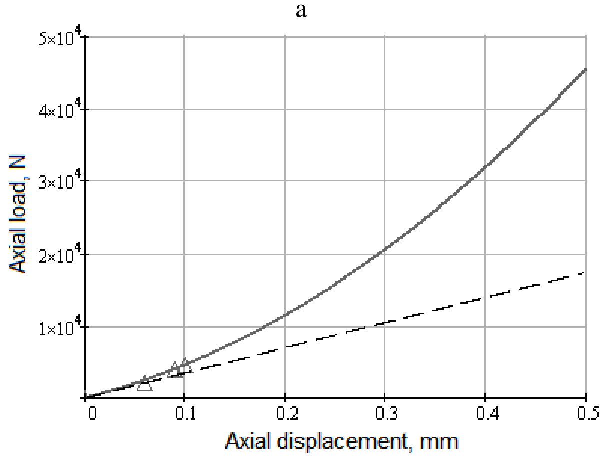

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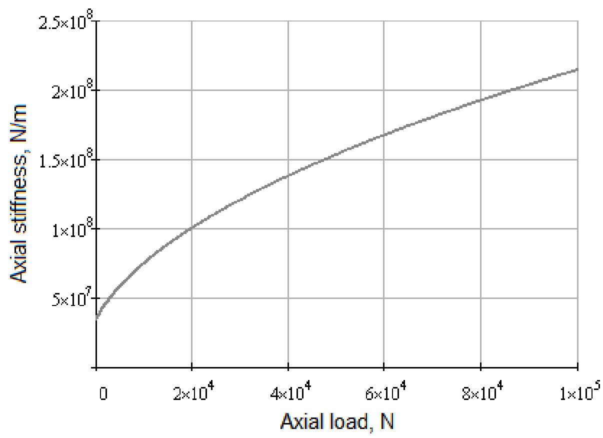

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Figure 12 – Dependence “radial load – radial displacement” for the bearing supports 45-216 (a) and 45-276214 (b)

Figure 13 – Dependence “axial load – axial displacement” for the bearing supports 45-216 (a) and 45-276214 (b)

Thus, the proposed comprehensive approach is approved on the example of the oxidizer turbopump's rotor for the liquid rocket engine that will allow refining the reliable mathematical and computational models of rotor dynamics for turbopump units and providing appropriate computer simulation of forced oscillations of the rotor systems for given permissible residual imbalances considering nonlinear stiffness characteristics of bearing supports.

Further research will be aimed at obtaining spectrums of critical frequencies and related mode shapes for the rotor systems in abovementioned bearing supports, as well as at the detailed analysis of forced oscillations of the turbopump rotor for the nonlinear stiffness of the bearing supports considering radial gaps, initial clearance, axial preloading, maximum rotor speed and compliance of the housing for the system of residual imbalances, which led to the maximum values of centrifugal forces.

5 Conclusions

As a result of numerical simulation (the determination of radial displacements of the rotor axis under the discrete values of the radial force), the approximated curves "radial load – radial displacement" are defined. Analytical expressions describing the mentioned dependence are determined as a result of approximation of the experimental data.

The results of more precisive calculations of rotor dynamics for the turbopump considering bearing gaps, axial preloading, rotor speed and compliance of the housing parts will allow clarifying the detuning from the resonance.

6 Acknowledgements

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Комплексний підхід до ідентифікації нелінійних жорсткісних характеристик підшипникових опор турбонасоса окислювача рідинного ракетного двигуна

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Анотація. Стаття присвячена уточненню математичних та обчислювальних моделей ротора турбонасоса окислювача з урахуванням зазорів у підшипниках, попереднього осьового навантаження, податливості корпусних елементів та впливу обертання вала. Запропонована схема навантаження складається з чотирьох шагів з урахуванням попереднього зміщення зовнішньої обойми, осьового зміщення у результаті переміщення опори внаслідок попереднього осьового навантаження, радіального переміщення внаслідок деформації підшипникової опори, а також відцентрових сил інерції, викликаних обертанням ротора разом із внутрішньою обоймою. Моделювання контактної взаємодії з використанням програмного забезпечення ANSYS здійснюється відповідно до достовірних моделей. Встановлені зони контакту між тілами кочення і внутрішньою та зовнішньою обоймами, а також визначений кут контакту. Побудовано ізоповерхні осьових та радіальних переміщень підшипникових опор. Нелінійна жорсткість опор визначається як тангенс кута дотичної до кривої, що описує залежність «радіальне навантаження – радіальне зміщення». Запропонований підхід, який використовується для проектування турбонасосних агрегатів рідинних ракетних двигунів, дозволить уточнити достовірні математичні та обчислювальні моделі динаміки ротора і забезпечити якісне моделювання вимушених коливань роторних систем для заданої системи допустимих залишкових дисбалансів з урахуванням нелінійних характеристик жорсткості підшипникових опор.

Ключові слова: податливість корпуса, початковий зазор, попереднє осьове навантаження, радіальне навантаження, кут контакту, радіальна жорсткість, осьова жорсткість.