JOURNAL OF ENGINEERING SCIENCES
ЖУРНАЛ ІНЖЕНЕРНИХ НАУК
ЖУРНАЛ ИНЖЕНЕРНЫХ НАУК
Web site: http://jes.sumdu.edu.ua
DOI: 10.21272/jes.2018.5(1).d2 Volume 5, Issue 1 (2018)

UDC 534.13:[621.22+621.515.1]

Investigation of Non-linear Reactions in Rotors' Bearing Supports of Turbo-pump Units for Liquid Rocket Engines

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Article info:
Paper received: December 10, 2017
The final version of the paper received: March 11, 2017
Paper accepted online: March 17, 2018

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Abstract. This paper is aimed at refinement of the computational model of the turbopump rotor systems associated taking into consideration the effect of rotation of moving parts and compliance of bearing supports elements. The up-to-date approach for investigation of non-linear reactions in rotor's bearing supports is proposed for turbo-pump units for liquid rocket engines. Five models for modelling contact interaction are investigated, and comparative bearing stiffness characteristics are given. The geometry of the housing and corresponding design scheme are set for each support due to the assembly drawing of the turbopump unit. Rotation of the shaft is taking into account by applying corresponding inertial forces to the inner cage of the bearing. Experimental points of the dependence "load – displacement" as the diagram "F – v" are built by the calculated points as an array of numerical simulation data, obtained by the ANSYS software. As a result of numerical simulation, including loading of the bearing support on the scheme "remote force" in a wide range of rotor speeds, the corresponding displacements are determined. The brand-new approach for evaluation of bearing stiffness coefficients is proposed based on the linear regression procedure. As a result, the obtained values of coefficients are summarized and approximated by the quadratic polynomials.

Keywords: Ansys Workbench, axial preloading, centrifugal force, contact interaction, finite element analysis, numerical simulation, remote force, stiffness characteristic.

1 Introduction

Intensification of the development in the field of power engineering occurs by using the modern energy-intensive equipment, an essential role of which is performed by multistage rotor machines. Permanently raising theirs parameters leads to increasingly significant problems of vibration reliability. Furthermore, the problem of investigation of dynamics of flexible rotors is based on determination of the critical frequencies and corresponding mode shapes. This problem is currently actual due to the impossibility of absolutely accurate dynamic rotor balancing [1].

General approaches are used for investigation of the rotor dynamics that are closely intersected with the issues of strength of materials and the theory of elasticity, the theory of linear and nonlinear oscillation of mechanical systems, as well as the problems for the identification of mathematical models of dynamic systems. Most problems can be solved in combination of 2D and 3D formulation by using modern software.

The problem of identification of bearing stiffness characteristics is complicated in the case of new designs with the insufficient experimental data. At the same time, the process of creating reliable mathematical models of the rotor dynamics is usually carried out in a permanent comparison with experimental data by means of the identification of coefficients of mathematical models and structures of design schemes. This process takes place in researching the vibration reliability and rotor balancing for centrifugal pumps and turbochargers [2, 3].

2 Literature Review

Up-to-date approaches for refinement of mathematical models of oscillatory systems according to experimental data is presented in the work [4]. The monograph [5] is aimed at evaluation of coefficients of mathematical models for oscillatory systems, including rotary systems for multistage centrifugal machines. The paper [6] dials with the phenomena of stability loss of rotor rotation at tilting pad bearings.

Modern treatments in the feld of linear and non-linear rotor dynamics is stated in the work [7] with the related practical applications. Estimation of segment bearing stiffness with the balancing procedure for flexible rotors of turbocharge units in the accelerating-balancing stand are presented in the paper [8]. Modern approaches for determination of active magnetic bearings stiffness and damping identification from frequency characteristics of control systems are realized within the work [9].

Application of the finite element analysis for stiffness and critical speed calculation of a magnetic bearing-rotor system for electrical machines is proposed in the work [10]. The problem of stability and vibration analysis of non-linear comprehensive flexible rotor bearing systems is analyzed in the paper [11]. A phenomenon of subharmonic resonance of a symmetric ball bearing-rotor system is investigated in the paper [12]. Approaches for analytical research and numerical simulation for investigation of critical frequencies of a centrifugal compressor rotor taking into account non-linear stiffness characteristics of bearings and seals are proposed in the paper [13].

3 Research Methodology

The ANSYS Workbench software is used for determination of bearing stiffness. The related design scheme is presented in Figure 1.

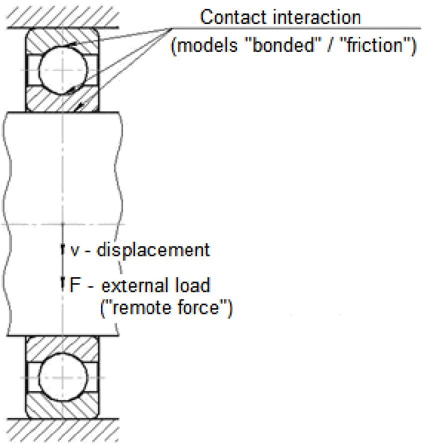


Figure 1 – Design scheme of bearing supports loading and determination of corresponding displacements

In the simulation of contacts by using ANSYS software, one of the most important problems is the selection of reliable model of interaction between elements of the contact pairs “target – contact”. There are five models of contact interaction, the comparative characteristics of which are given in Table 1.

Further calculations are provided for each of the selected contact type:

– “bonded” – from the group of linear contacts;
– “frictional” – from the group of nonlinear contacts.

These models allow determining the maximum possible range of variation for the stiffness of bearing supports.

Table 1 – Comparative table of the main characteristics of models for the contact interaction between the surfaces of mating parts

Contact model	Contact type	Number of iterations	Normal behavior	Tangent behavior
Bonded	Linear	One	Not allowed	Not allowed
No separation				Allowed
Rough	Nonlinear	Several	Allowed	Not allowed
Frictionless				Allowed
Frictional				Allowed

“Bonded” is the contact model, in which the target and contact surfaces of the matched bodies are connected to each other, and the contact area does not change under the action of the applied loads. The sliding between faces and edges, as well as their separation is not allowed.

“Frictional” is the contact model that takes into account the sliding of the surfaces “target” and “contact” relative to each other. In this case, the contact area changes, if the module of the tangential force takes the limiting value.

4 Results

4.1 Basic approach

As a result of numerical simulation (loading of the bearing support according to the scheme “remote force”) for discrete values of the force F in a range from zero to the maximum load capacity, the corresponding displacements are determined (Figures 2, 3).

The calculated points allow determining an array of data by means of a numerical simulation, on which the points of the “load – displacement” diagram “$F - v$” are built (Table 2, Figure 4).

The obtained data are interpolated by the corresponding curves $F = F(v)$. In this case, the stiffness of the bearing supports for linear models is determined as the tangent of the initial slope angle α of the diagram “$F - v$”:

$$c = tg\,\alpha = \left(\frac{\partial F}{\partial v}\right)_0 . \qquad (1)$$

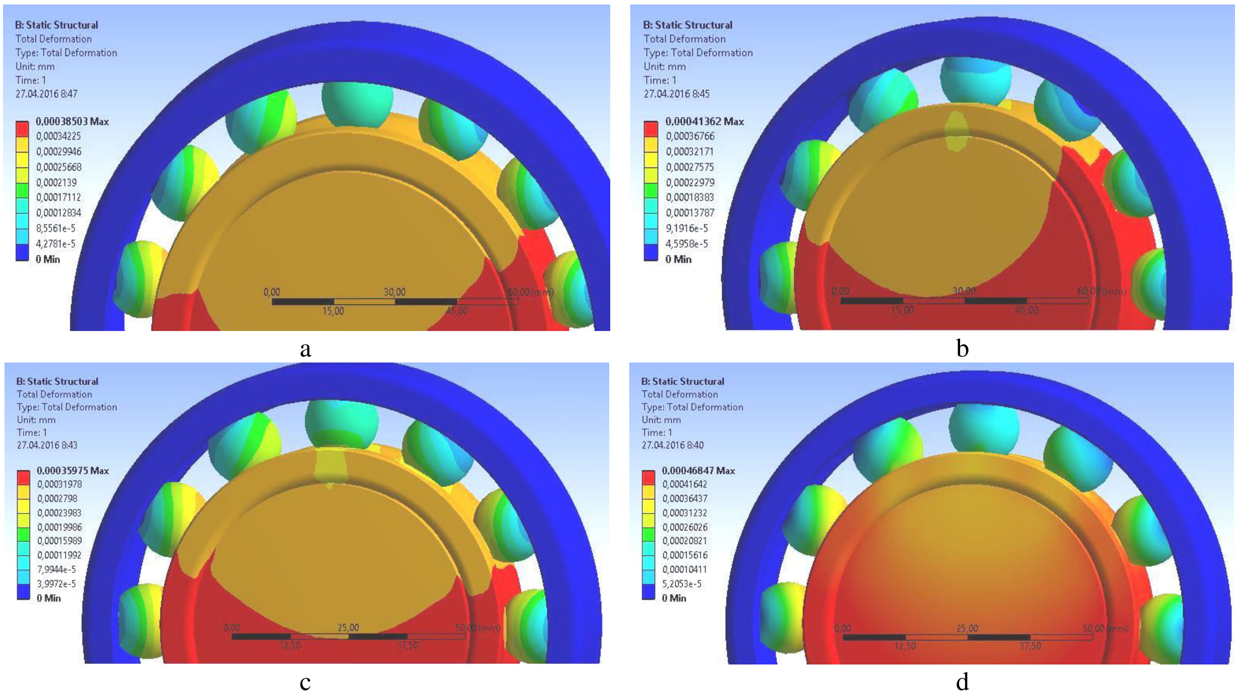


Figure 2 – Determination of the bearing stiffness for the model “bonded” of the contact interaction between the rolling elements with cages of bearings 45-216 (a), 45-276214 (b), 46-276212 (c) and 36-211 (d)

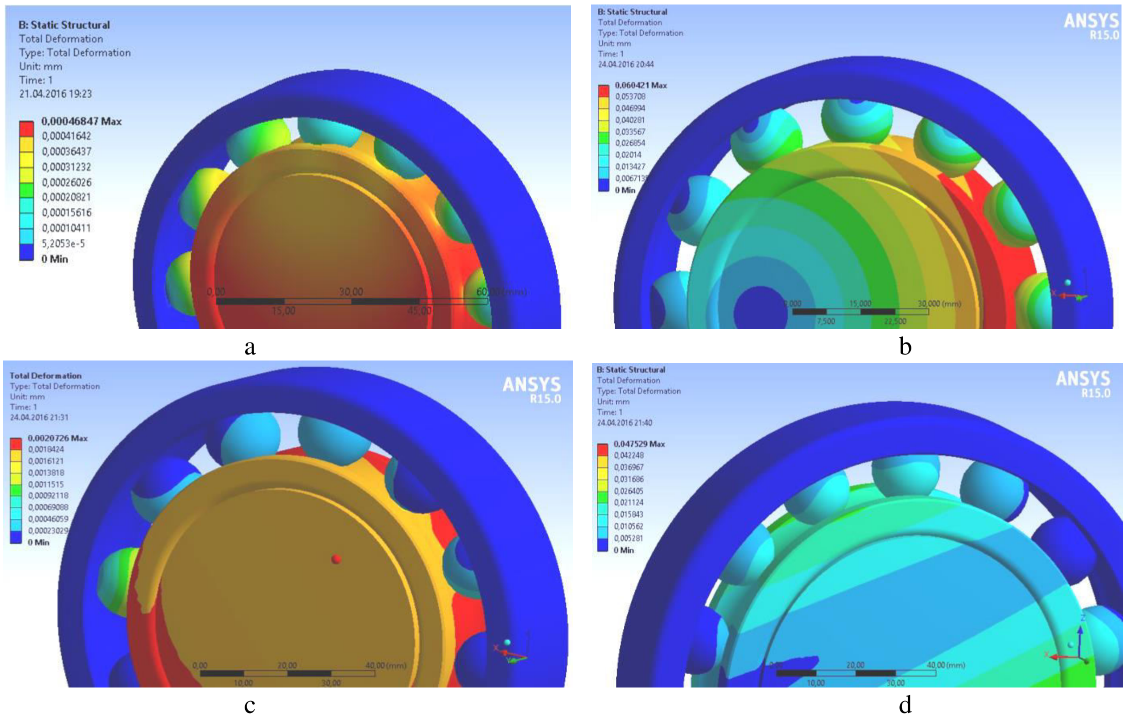


Figure 3 – Determination of the bearing stiffness for the model “frictional” of the contact interaction between the rolling elements with cages of bearings 45-216 (a), 45-276214 (b), 46-276212 (c) and 36-211 (d)

Table 2 – Results of numerical simulation for determining diagram “$F - v$”

Bearing	Load, N	Displacement, m	
		“frictional”	“bonded”
45-216	$1 \cdot 10^3$	$3.03 \cdot 10^{-6}$	$3.43 \cdot 10^{-7}$
	$1 \cdot 10^4$	$2.12 \cdot 10^{-5}$	$3.43 \cdot 10^{-6}$
	$7 \cdot 10^4$	$1.12 \cdot 10^{-4}$	$2.40 \cdot 10^{-5}$
45-276214	$1 \cdot 10^3$	$4.76 \cdot 10^{-6}$	$3.62 \cdot 10^{-7}$
	$1 \cdot 10^4$	$1.96 \cdot 10^{-5}$	$3.62 \cdot 10^{-6}$
	$6 \cdot 10^4$	$1.0 \cdot 10^{-4}$	$2.17 \cdot 10^{-5}$
46-276212	$1 \cdot 10^3$	$7.69 \cdot 10^{-6}$	$3.16 \cdot 10^{-7}$
	$1 \cdot 10^4$	$2.43 \cdot 10^{-5}$	$3.16 \cdot 10^{-6}$
	$5 \cdot 10^4$	$8.82 \cdot 10^{-5}$	$1.56 \cdot 10^{-5}$
36-211	$1 \cdot 10^3$	$3.85 \cdot 10^{-6}$	$4.18 \cdot 10^{-7}$
	$1 \cdot 10^4$	$3.90 \cdot 10^{-5}$	$4.18 \cdot 10^{-6}$
	$4{,}5 \cdot 10^4$	$1.26 \cdot 10^{-4}$	$1.88 \cdot 10^{-5}$

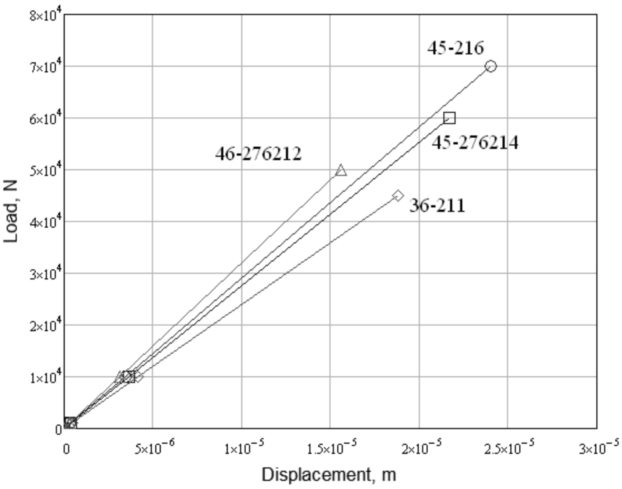


Figure 4 – Diagram “ $F - v$” for the model “bonded”

The stiffness coefficients for bearings calculated by the abovementioned procedure, are summarized in Table 3.

Table 3 – Stiffness coefficients for bearing supports

Bearing	Stiffness coefficient, 10^8 N/m	
	“frictional”	“bonded”
45-216	3.3	29.2
45-276214	2.1	27.6
46-276212	1.3	31.7
36-211	2.6	23.9

4.2 Refinement of the numerical model

This part is aimed at refinement of the computational model of the turbopump rotor systems associated taking into consideration the effect of rotation of moving parts and compliance of bearing supports elements. The first factor causes an increasing quadratic dependence of the bearing stiffness on the rotor speed, and consequently, shift of the spectrum of critical frequencies to the right. This circumstance increases the detuning from the resosanse mode. The second factor decreases the bearing stiffness and critical frequencies.

The clarification of the stiffness parameters of the supporting units is carried out by combination of two computational means. Firstly, the loading patterns of supporting units using ANSYS software (three-dimensional finite element models) are considered due to a significant computational time.

ANSYS software is used for determination the bearing stiffness with considering rotation of the rotor and compliance of housing elements. The related design scheme is presented on Figure 5.

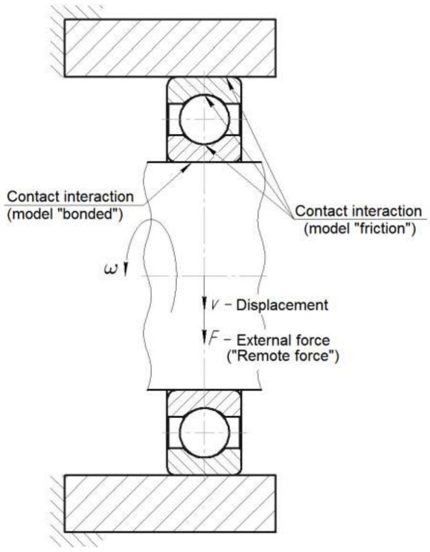


Figure 5 – Refined design scheme of bearing supports loading

The geometry of the housing and corresponding design scheme are set for each support due to the assembly drawing of the turbopump unit. The rotation is taking into account by applying corresponding inertial forces to the rotating (inner) cage of the bearing.

Modelling of contacts by using ANSYS software is performed according to Table 4.

Table 4 – Models of contact interaction between surfaces

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

As a result of numerical simulation (loading of the bearing support on the scheme “remote force” $F = 1 \cdot 10^3$ N for the following values of operating rotor speed: 0, 10 500, 18 750, and 21 150 rpm), the corresponding displacements are determined (Figures 6–9).

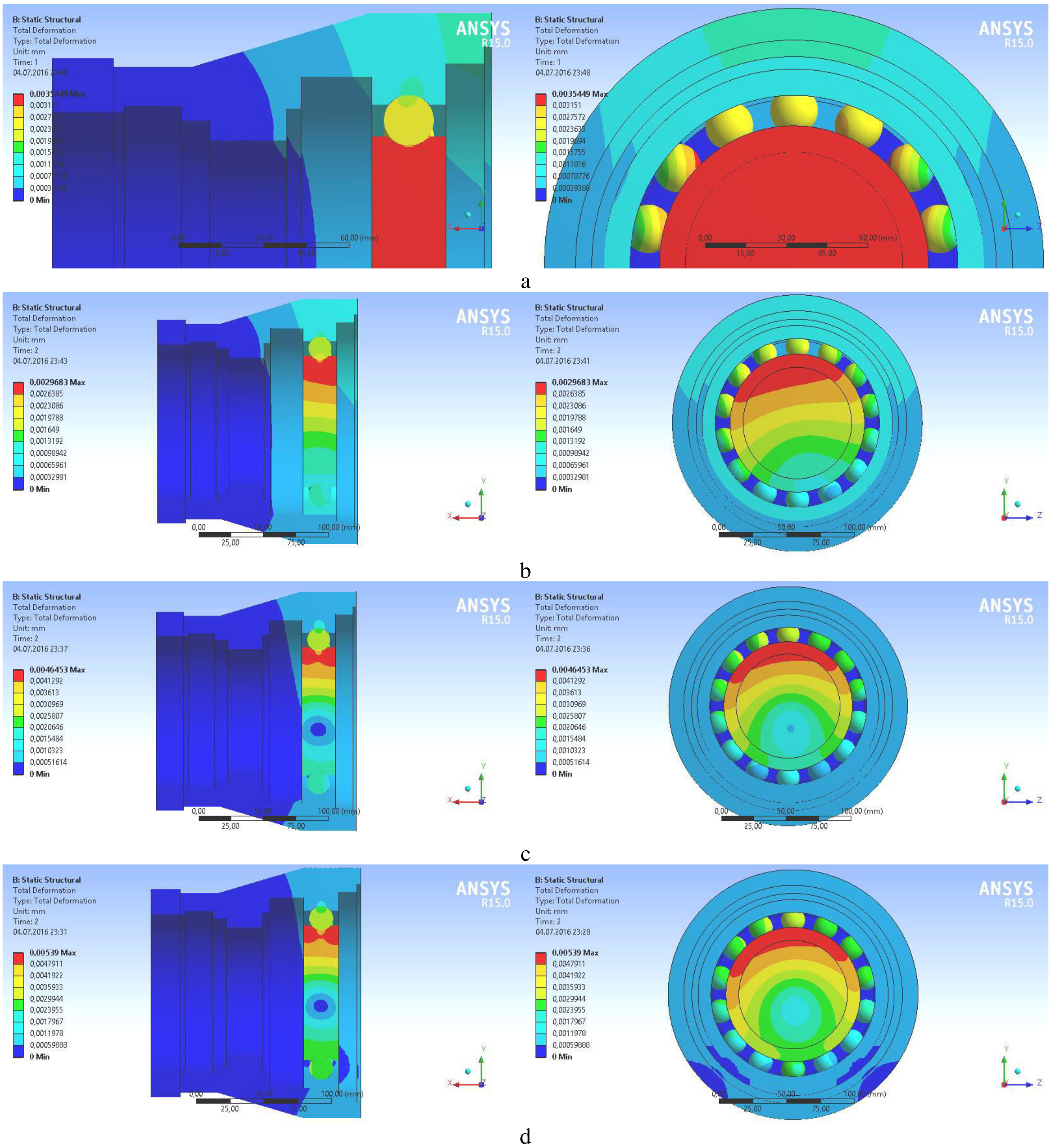


Figure 6 – Bearing stiffness for the support 45-216: 0 rpm (a), 10500 rpm (b), 18750 rpm (c), 21150 rpm (d)

In this case, determined bearing stiffness coefficients are summarized in Table 5.

Table 5 – Bearing stiffness of the supports

Bearing	Stiffness coefficient, 10^8 N/m, for the operating frequency, rad/s			
	0	1100	1963	2215
45-216	2.9	5.3	7.3	8.3
45-276214	2.4	2.9	4.4	4.5
46-276212	2.2	2.4	3.4	4.4
36-211	1.1	1.1	1.2	1.3

The analytical dependence for creating the mathematical models of free and forced oscillations of the turbopump rotor is proposed taking into account the rotation:

$$c = c_0 + \alpha \omega^2, \qquad (2)$$

where c – stiffness coefficient of the bearing support; ω – rotor speed, rad/s; c_0 – stiffness coefficient in case of $\omega = 0$; α – additional coefficient, $N{\cdot}s^2/m$.

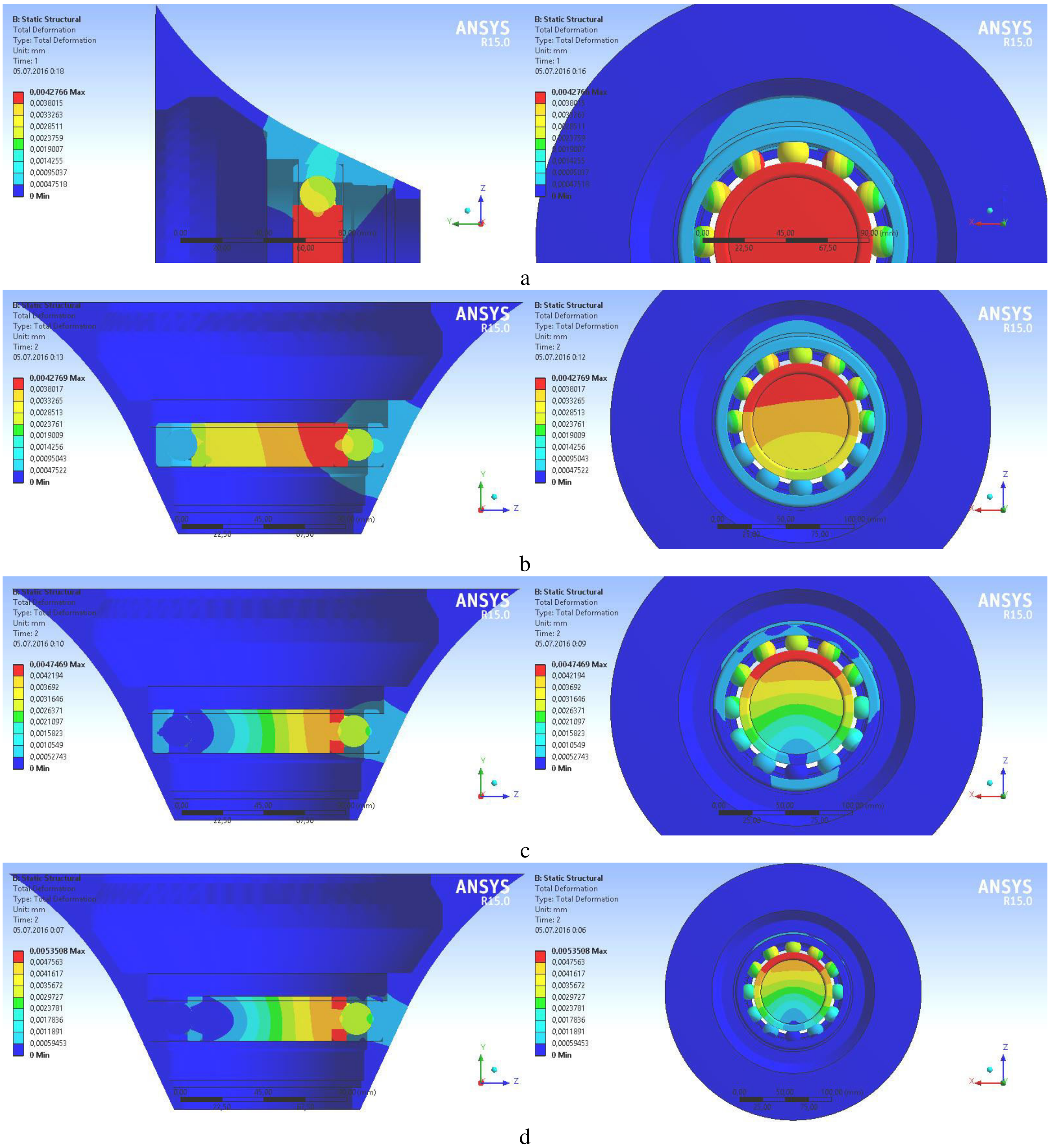


Figure 7 – Bearing stiffness for the support 45-276214: 0 rpm (a), 10500 rpm (b), 18750 rpm (c), 21150 rpm (d)

The evaluation of the coefficient α of the formula (2) is carried out by the linear regression procedure according to the following formula:

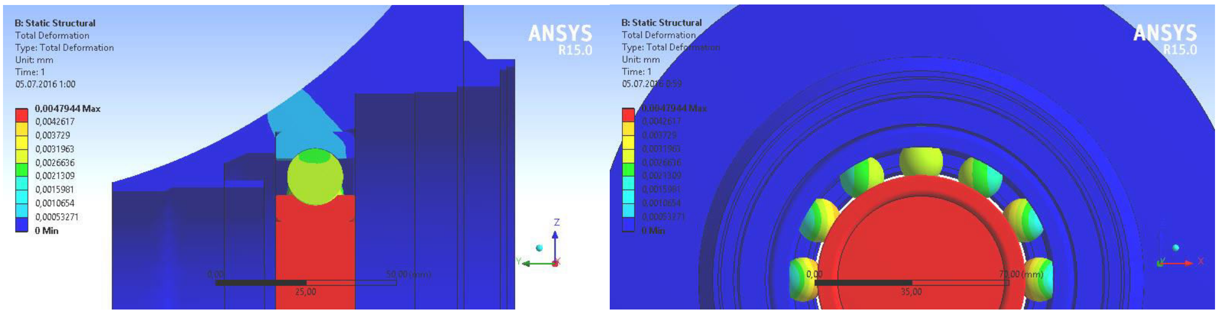
$$\alpha = \frac{\sum_{k=1}^{3} (c_k - c_0)\omega_k^2}{\sum_{k=1}^{3} \omega_k^4}, \qquad (3)$$

where c_k – bearing stiffness, determined as a result of the numerical simulation for the rotor speed ω_k (Table 5); k – number of the experimental point.

Finally, the obtained values of coefficients α are summarized in Table 6, and approximating curves (2) are also shown on Figure 10.

Table 6 – Bearing stiffness parameters

Bearing	Coefficients	
	c_0, N/m	α, N·s^2/m
45-216	2.9	116.3
45-276214	2.4	46.6
46-276212	2.2	38.3
36-211	1.1	4.1



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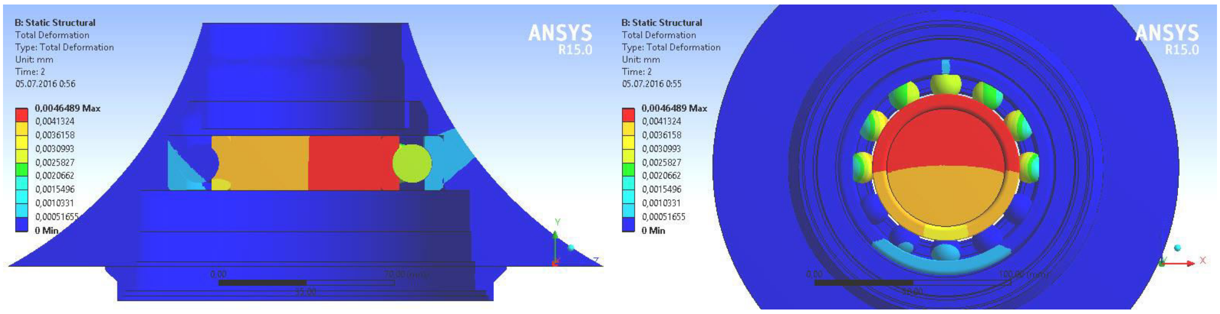

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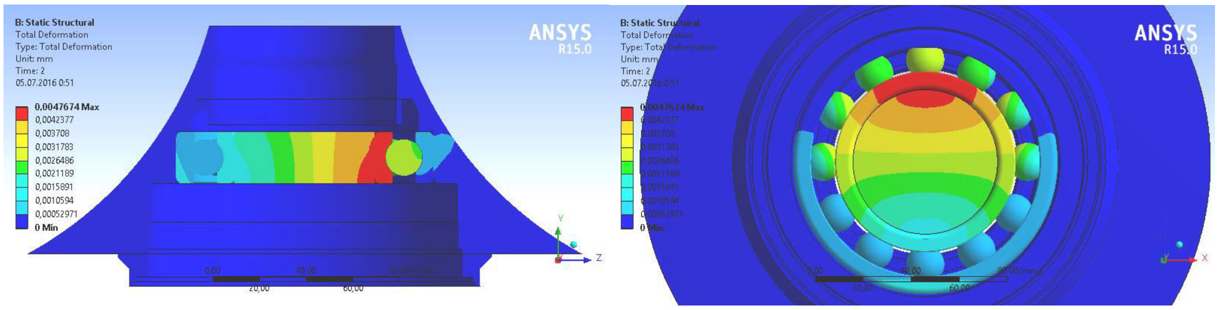

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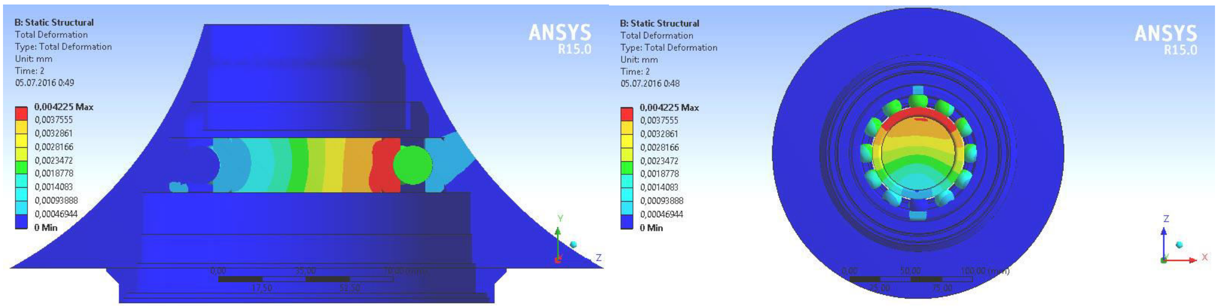
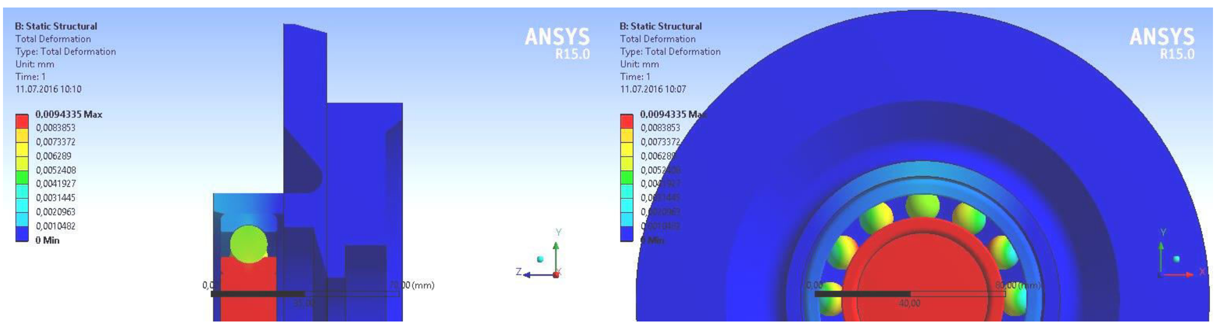
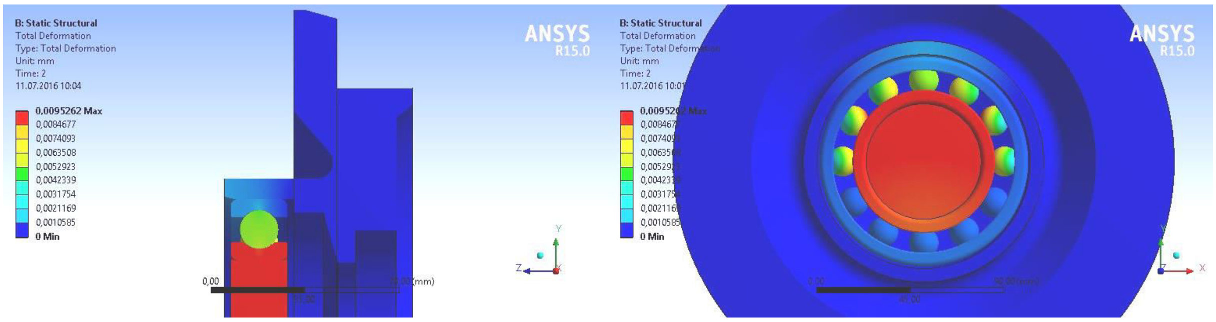


d

Figure 8 – Bearing stiffness for the support 46-276212: 0 rpm (a), 10500 rpm (b), 18750 rpm (c), 21150 rpm (d)



a



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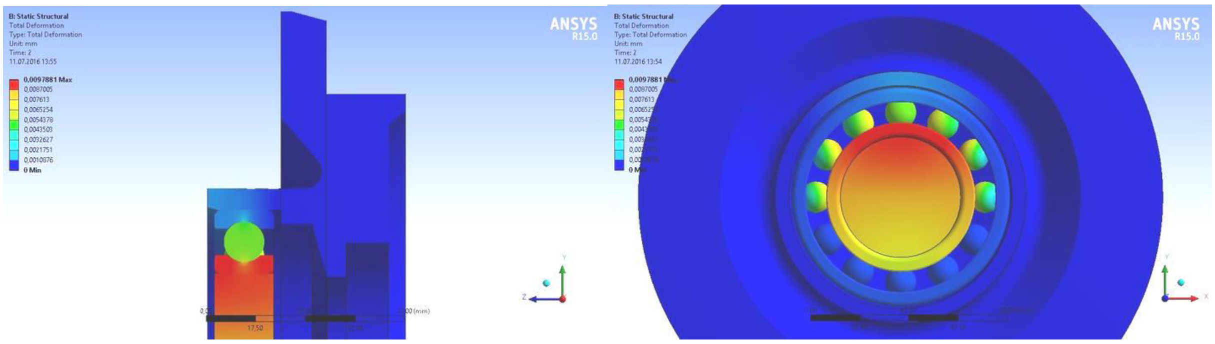

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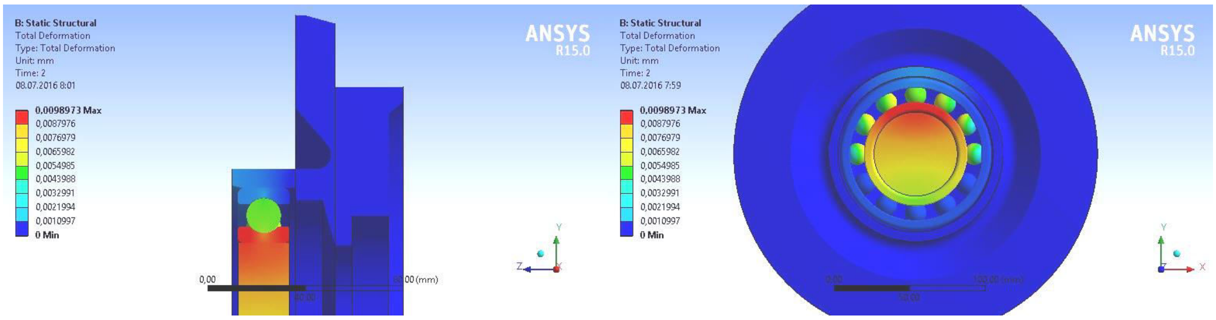

d

Figure 9 – Bearing stiffness for the support 36-211: 0 rpm (a), 10500 rpm (b), 18750 rpm (c), 21150 rpm (d)

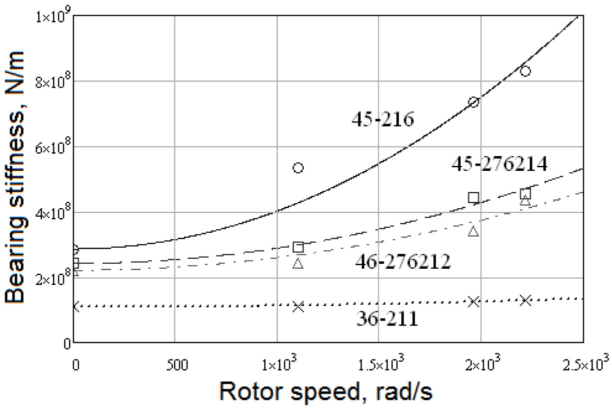

Figure 10 – Dependence of the bearing stiffness on the rotor speed

5 Conclusions

In this paper the methodology of determination of the bearing stiffness is proposed based on using different models of contact interaction between the mating surfaces of bearing parts. An appropriate methodology for refinement of the computational model is proposed taking into account the effect of rotation of moving parts and compliance of bearing supports elements.

The clarification of the stiffness parameters of the supporting units is carried out by combination of several computational means.

Further research should be aimed at obtaining spectrums of critical frequencies and related mode shapes for the rotor systems in abovementioned bearing supports.

6 Acknowledgements

The main part of the achieved results were obtained within the research project "Rotordynamic Research for the Turbopumps of the Liquid Rocket Engines" (Reg. no. 51.24-01.15.SP).

References

1. Pavlenko, I., Simonovskiy, V., Pitel', J., & Demianenko, M. (2018). *Dynamic analysis of centrifugal machines rotors with combined using 3D and 2D finite element models*. Lüdenscheid, RAM-VERLAG.
2. Vance, J., Zeidan, F., & Murphy, B. (2010). *Machinery vibration and rotordynamics*. New York, John Wiley & Sons Inc.
3. Simonovskiy, V. I. (2006). *Rotor dynamics for centrifugal machines*. Sumy, Sumy State University.
4. Simonovskiy, V. I. (2010). *Refinement of mathematical models of oscillatory systems according to experimental data*. Sumy, Sumy State University [in Ukrainian].
5. Simonovskiy, V. I. (2015). *Evaluation of coefficients of mathematical models for oscillatory systems*. Saarbrücken, LAP LAMBERT Academic Publishing [in Russian].
6. Gadyaka, V. G., Leikych, D. V., & Simonovskiy, V. I. (2011). Phenomena of stability loss of rotor rotation at tilting pad bearings. *13th International Scientific and Engineering Conference "Hermetic, Vibration Reliability and Ecological Safety of Pump and Compressor Machinery (HERVICON-2011)"*, Vol. 39, 244–253.
7. Ishida, Y., & Yamamoto, T. (2012). *Linear and nonlinear rotordynamnics. A modern treatment with applications*. Verlag, Wiley-VCH.
8. Gadyaka, V. G., & Simonovskiy, V. I. (2005). Estimation of segment bearing stiffness while balancing flexible rotors for turbocharge units in the accelerating-balancing stand. *Bulletin of Sumy National Agrarian University, Series "Mechanization and Automation of Industrial Processes*, No. 11, 145–150.
9. Jin, C., Xu, Y., Zhou, J., et al. (2016). *Active magnetic bearings stiffness and damping identification from frequency characteristics of control systems*. Cairo, Hindawi Publishing Corporation.
10. Wang, T., Wang, F., Bai H., et al. (2008). *Stiffness and critical speed calculation of magnetic bearing-rotor system based on FEA*. Electrical machines and systems, IEEE Xplore.
11. Villa, C., Sinou, J., & Thouverez, F. (2008). Stability and vibration analysis of a complex flexible rotor bearing system. *Communications in Nonlinear Science and Numerical Simulation*, Vol. 13(4), 804–821.
12. Bai, C., Zhang, H., & Xu, Q. (2013). Subharmonic resonance of a symmetric ball bearing-rotor system. *International Journal of Non-Linear Mechanics*, Vol. 50, 1–10.
13. Pavlenko, I. V., Simonovskiy, V. I., Pitel, J., et al. (2017). Investigation of critical frequencies of the centrifugal compressor rotor with taking into account stiffness of bearings and seals. *Journal of Engineering Sciences*, Vol. 4, Issue 1, pp. C1–C6.

Дослідження нелінійних реакцій підшипникових опор роторів турбонасосних агрегатів рідинних ракетних двигунів

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Анотація. Стаття присвячена уточненню числової моделі дослідження роторних систем турбонасосних агрегатів рідинних ракетних двигунів, що базується на урахуванні обертання валопроводу та податливості елементів підшипникових опор. Запропоновано сучасний підхід до дослідження нелінійних реакцій у підшипникових опорах роторів турбонасосних агрегатів рідинних ракетних двигунів. Досліджено п'ять моделей контактної взаємодії та представлено відповідні порівняльні характеристики жорсткостей підшипникових опор. Врахована геометрія корпуса та відповідна складена відповідна конструкційна схема для кожної опори, що базується на складальному кресленні турбонасосного агрегату. Урахування обертання вала здійснено шляхом прикладання сил інерції до внутрішньої обойми підшипника. Побудовано експериментальні точки залежності «навантаження – переміщення» діаграми "F – v" за допомогою розрахованих даних як масиву результатів моделювання, отриманих із застосуванням програмного комплексу ANSYS. У результаті числового моделювання, у тому числі навантаження опорного підшипника за схемою «віддалена сила» в широкому діапазоні частот обертання ротора, визначені відповідні радіальні переміщення. Запропоновано новий підхід до оцінки коефіцієнтів жорсткості підшипників на основі процедури лінійної регресії. У результаті отримані значення коефіцієнтів, що апроксимуються поліномами другої степені.

Ключові слова: Ansys Workbench, попереднє осьове навантаження, відцентрова сила, контактна взаємодія, скінченноелементний аналіз, числове моделювання, віддалена сила, характеристики жорсткості.