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STUDY OF THE INFLUENCE OF HEAT LOAD AND CAVITATIONS ON THE CARRY CAPACITY OF A TURBOCHARGER BEARING

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The aim of the study is to research the thermal and high-speed operating conditions for the tendency of the formation and distribution of fields of the volume fraction of vapor, pressure and temperature in the lubricating layer of a radial bearing. In this work, the numerical simulation of hydrodynamic processes in the lubricant layer of a real design of a radial bearing of a turbocharger. To solve the problem, authors used numerical methods of calculation: the method of finite differences and the method of finite dimensions. The novelty of the work lies in determining the parameters of the cavitation model for oil and carrying out calculations on a real design of a journal bearing, taking into account the experimental results of thermometry. As a result, studies were obtained of the dependence of the change in the bearing capacity and volumetric vapor in the lubricating layer depending on the thermal state and the rotor speed. An assessment of the influence of thermal and speed modes on the bearing capacity of a turbocharger rotor bearing is carried out. The results obtained indicate an increase in the volume fraction of vapor with the rotor speed. In this case, the thermal state of the bearing affects the shape of the vapor distribution and the value of the bearing capacity in the lubricating layer.

Key words: turbocharger, Journal bearings, Computational fluid dynamics, Cavitations. Thermal state.

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Научная статья
Статья в открытом доступе

ИССЛЕДОВАНИЕ ВЛИЯНИЯ ТЕПЛОВОЙ НАГРУЗКИ И КАВИТАЦИИ НА НЕСУЩУЮ СПОСОБНОСТЬ ПОДШИПНИКА ТУРБОКОМПРЕССОРА

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Аннотация

Целью исследования является изучение влияния теплового и скоростного режимов работы на тенденцию к формированию и распределению полей объемной доли пара, давления и температуры в смазочном слое радиального подшипника. В данной работе проводится численное моделирование гидродинамических процессов в смазочном слое реальной конструкции радиального подшипника турбокомпрессора. Для решения поставленной задачи использовались численные методы расчета: метод конечных разностей и метод конечных объемов. Новизна работы заключается в определении параметров кавитационной модели для моторного масла и проведению расчетов на реальной конструкции подшипника скольжения, с учетом экспериментальных результатов термометрирования. В результате исследования были получены зависимости изменения несущей способности подшипника и объемной доли пара в смазочном слое в зависимости от теплового состояния и скорости вращения ротора. Проведена оценка влияния тепловых и скоростных режимов на несущую способность подшипника ротора турбокомпрессора. Полученные результаты указывают на увеличение объемной доли пара с увеличением скорости вращения ротора. При этом тепловое состояние подшипника влияет на форму распределения паров и значение несущей способности в смазочном слое.

Ключевые слова: турбокомпрессор, подшипник скольжения, вычислительная гидродинамика, кавитация, тепловое состояние.

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Introduction

The reliability of various units and machines is determined by the durability of tribological conjugations, which should ensure operability in conditions of high temperatures and an extended range of rotational speeds of the rotor shaft.

Turbochargers are used to forcing the internal combustion engine. The use of turbocharging leads to an increase in the pressure and temperature of the gases in the cylinders. Accordingly, the amount of heat that is removed from the parts of the cylinder-piston group to the oil system and the cooling system increases. This leads to an increase in the total thermal load. Journal bearings are elements that receive radial or axial loads in a turbocharger. According to studies [1, 2], the most common cause of turbocharger failure is the failure of journal bearings (from 43.8 to 47.9% of cases).

The hydromechanical operational characteristics of journal bearings are affected by a wide range of parameters: the quality of the lubricant and its operational parameters; build quality of both the turbocharger and the bearing assembly; rotor balancing; temperature condition; speed and load conditions during operation [3], etc.

The most probable types of destruction of journal bearings are: abrasive, adhesive, fatigue and cavitation wear [4]. Temperature conditions have a direct impact on the probability of abrasive and adhesive wear. With increasing temperature, the viscosity of the lubricant decreases, and the bearing shell expands. As a result, the thickness of the lubricant layer is reduced. This can lead to boundary lubrication and damage to friction bearing surfaces from contact with abrasive particles. In the worst case, a local break of the oil film occurs. In this case, metal contact zones arise where, under the influence of molecular forces, micro-welding bridges are formed. These bridges cause deep gap of the material, i.e. adhesive wear.

It should be borne in mind that increased temperature loads affect the performance of seals. Lubricant leaks increase; overheating of tribological conjugations leads to an increase in the amplitude of oscillations of the rotor. These factors increase the likelihood of contact between the blades and the turbocharger housing, which leads to their destruction and failure of the rotor shaft.

In turn, cavitation occurs due to a drop in oil pressure below the saturated vapor pressure of the liquid. Due to this, cavitation cavities (bubbles) appear, which are common in the hydrodynamic flow. With increasing pressure, the bubbles explode, which leads to cavitation wear. Many hydraulic devices are affected by this type of wear: turbine blades, pumps, journal bearings, etc. [5, 6]. Therefore, this phenomenon cannot be ignored in hydrodynamic calculations.

Other works [7, 8] confirm the significant influence of the static and dynamic characteristics of a sliding bearing on the efficiency and stability of turbomachines.

The relevance of the study is confirmed by the numerous works devoted to the study of physical processes occurring in the bearing. When studying the processes occurring in friction units, various techniques are used.

Currently, CFD modeling, which allows calculating the characteristics of stream processes using computational and physical-mathematical methods, has become widespread.

The study [9] is devoted to a three-dimensional CFD model for studying friction losses of bearings of a neck of a turbocharger. The study had an isothermal assumption neglecting viscous heat and viscosity changes. In his next work [10], the author got rid of this assumption.

Serrano research also focuses on heat transfer processes occurring in turbochargers. In study [11], the main problem is that during engine hot stops due to lack of oil flow within the captured oil in the turbocharger bearing burns, since the turbine housing exchanges heat with the central housing. The formation of coke can lead to a significant reduction in the endurance of the turbocharger, since the bearings become clogged and damage the shaft.

Article [12] is devoted to the study of internal convection in small turbochargers. The author focuses on ignoring heat loss in small turbochargers. But at low loads, this energy transfer can reach values even higher than the mechanical power of a turbocharger. Therefore, this paper presents a brief methodology for measuring and modeling these heat fluxes using a simplified lumped model, and also analyzes the main convective heat transfer coefficients.

In article [13], a study was made of the performance of journal bearings, taking into account thermal and cavitation effects. Moreover, the author compared the characteristics of journal bearings, which were operated with different lubricating fluids (Water, sea water, lubricating oil). According to the results of the study, the maximum temperature of the bearing with oil lubrication is achieved in the area with the minimum film thickness. But for bearings that are lubricated with plain and sea water, the maximum temperature value is formed in places with a minimum film thickness.

The three-dimensional calculation model presented by Gil [14] allows one to study in detail the heat transfer processes occurring in the central housing and lubrication channels, as well as to determine the maximum temperature at different points in the system of the bearing assembly of an vehicle turbocharger. According to the results of the work, the author concludes that the model can be used as a diagnostic tool for assessing the phenomena of oil coking, as one of the most common failure conditions in turbochargers. The disadvantage is the simplified design of the turbocharger. But it can be justified by the desire to apply a more accurate methodology for calculating the phenomena of heat transfer. Moreover, the study is conducted for a small number of speed regimes. And also there is no calculation of gas-dynamic processes of the impellers to create more detailed initial data of temperature fields.

The main purpose of research [15] was to study the effect of an oil whirl and cavitation on the static and dynamic characteristics of a journal bearing. Comparison of the models of single-phase and two-phase flows showed that the pressure of the oil film had similar distribution features in the zone of a converging wedge. But there was a difference in the diverging part of the gap. Moreover, the presence of cavitation reduced the maximum pressure in the liquid film.

In modern studies, dedicated to the study of hydro-mechanical characteristics of journal bearings, there are a number of serious assumptions. They influence the truth of knowledge about the operation of the bearing assembly. First of all, it is worth noting the simplified bearing geometry [10, 13, 15]. The geometry is often simplified to a sleeve with multiple grease inlets. Because of this, it is impossible to adequately recreate the picture of the ongoing physical processes in real bearing designs. The second significant disadvantage is the lack of consideration of the skew of the rotor depending on the speed and thermal operating conditions. And this affects the reliability of the results obtained from the calculation model.

From the foregoing, we can conclude that the study of hydrodynamic processes occurring in radial journal bearings is an urgent task. In this work, the simulation of physical processes in the lubricant layer of the real design of the turbocharger radial bearing is carried out. The purpose of this study is to study the influence of thermal and high-speed operating modes on the tendency for the formation and distribution of the gas volume fraction, pressure and temperature fields in the lubricating layer of a radial bearing. An assessment of the influence of thermal modes and speed modes on the bearing capacity of a turbocharger rotor bearing is carried out.

Description of construction

The object of research was the radial journal bearing of the turbocharger, manufactured by Scientific and Production Association "Turbocomplekt". Operational speed is 90,000 rpm. The diameter of the axle of the rotor is 8 mm, and the diametrical clearance is 27 μm. The length of the bearing surface from the turbine side is 6 mm, and from the compressor side it is 4.8 mm. A section of a simplified design of a turbocharger is shown in Fig. 1. A more detailed creation of a computational grid model taking into account the inner prismatic layers is described in [16].

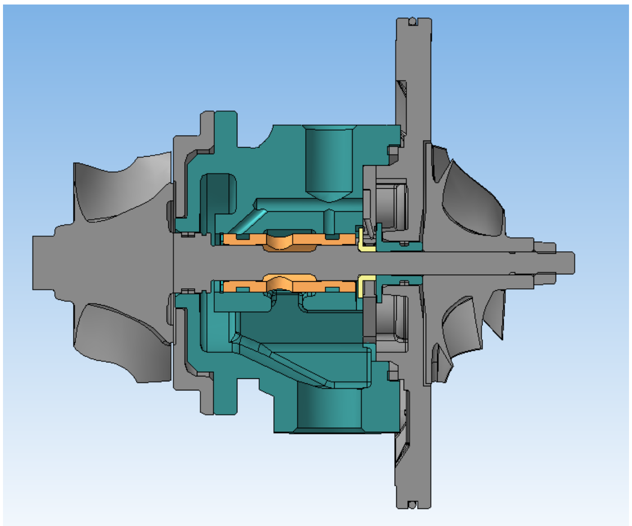

Fig. 1. The simplified turbocharger model 50.09.16
Рис. 1. Упрощенная модель турбокомпрессора 50.09.16

Conclusion mathematical model

The hydrodynamic forces in the lubricating layer of a liquid enclosed between two arbitrarily moving surfaces were determined based on the determination of the pressures in this lubricating layer. The pressure distribution was determined by solving the Reynolds equation The Reynolds equation was solved using the multigrid method. The convergence parameter was set at 10-4. In doing so, the non-Newtonian properties of the lubricant were taken into account [16]. The Swift-Stieber boundary conditions were used.

The theory of thermal processes in Ansys Fluent is based on the energy equation, which has the following form:

$$\frac{\partial}{\partial t}(\rho E) + \nabla(\vec{v}(\rho E + p)) = \nabla\left(k_{eff}\nabla T - \sum_j h_j\vec{J}_j + (\tau_{eff}\vec{v})\right) + S_h. \quad (1)$$

Here k_{eff} is effective conductivity, $k_{eff}=k+k_t$, where k_t is turbulent thermal conductivity, determined in accordance with the turbulence model used, and $\vec{J}_j$ is the diffusion particle flux j.

On the right side of equation (1), the first three terms represent energy transfer due to conductivity, particle diffusion, and viscous dissipation. At the same time, S_h includes volumetric heat sources, which are determined by the user, but not heat sources, which are formed during volumetric or surface reactions at a finite rate. Because enthalpy of formation is taken into account when calculating the total enthalpy.

To simulate cavitation, it is necessary to correctly take into account the process of growth and destruction of bubbles. Therefore, cavitation models are based on the generalized Rayleigh-Pless equation, which describes the growth of one vapor bubble in a liquid [17]:

$$R_b\frac{d^2R_b}{dt^2} + \frac{3}{2}\left(\frac{dR_b}{dt}\right)^2 = \left(\frac{P_b - P}{P_l}\right) - \frac{4\upsilon_l}{R_b}R_b - \frac{2\sigma}{\rho_l R_b}. \quad (2)$$

Where R_b is bubble radius, σ is liquid surface tension coefficient, ρ_l is liquid density, ν_l is liquid kinematic viscosity, P_b is bubble surface pressure, P is local far-field pressure.

Equation (2) is quite difficult to apply when modeling a multiphase flow. Therefore, neglecting the second-order terms and the surface tension force, the equation takes the following form [18]

$$\frac{dR_b}{dt} = \sqrt{\frac{2}{3}\frac{P_b - P}{\rho_l}}. \quad (3)$$

Based on equation (3), three mass transfer models were created: the Singhal model, the Zwart-Gerbert-Belamri model, the Schnerr-Sauer model [19, 20, 21].

In chronological order, the very first interphase transfer model is the Singhal model. It takes into account all first-order effects (phase change, bubble dynamics, turbulent pressure fluctuations, and non-condensable gases). In particular, the model takes into account multiphase flows or flows with multiphase particle transport, the influence of sliding velocities between the liquid and gaseous phases, as well as thermal effects and compressibility of the two phases. The advantages of the model include: the ability to take into account non-condensed gas, a high degree of agreement between the results of numerical simulation of flows in centrifugal pumps and experimental data. But due to the selection of k, the model has poor convergence [22].

In order to eliminate the disadvantages of the Singhal model, the Zwart-Gerber-Belamri model was created. A series of simplifications was introduced into the model: the density of the liquid and the density of the liquid with bubbles are identical, there is no non-condensed gas in the liquid. The final mass transfer equation of the Zwart-Gerber-Belamri model is as follows:

$$\text{If } P \le P_v, \qquad R_e = F_{vap}\frac{3\alpha_{nuc}(1-\alpha_v)\rho_v}{R_b}\sqrt{\frac{2}{3}\frac{P_b - P}{\rho_l}}. \quad (4)$$

$$\text{If } P > P_v, \qquad R_e = F_{cond}\frac{3\alpha_v\rho_v}{R_b}\sqrt{\frac{2}{3}\frac{P_b - P}{\rho_l}}. \quad (5)$$

Where F_{vap} is evaporation coefficient, F_{cond} is condensation coefficient, α_{nuc} is nucleation site volume fraction, ρ_l is liquid density, ρ_v is vapor density, P_v is saturation vapor pressure, α_v is vapor volume fraction.

The advantage of this model is good convergence and selection of coefficients for certain tasks. However, the model is less accurate than the Singhal model [22]. In the presented work, the Zwart-Gerber-Belamri model is used, since it is more "flexible" due to the selection of coefficients for solving the required problem. From the analysis of [23, 24], the necessary parameters were selected for modeling cavitation in motor lubricating fluid, presented in table 1.

Table 1

Cavitation model parameters

Parameters	Value
Bubble diameter, μm	2
Nucleation site volume fraction	0,004
Evaporation coefficient	0,02
Condensation coefficient	0,01
Vaporization pressure, Pa	5400

Heat transfer processes are radically different in laminar and turbulent flows. In the extreme case, with low viscosity and thermal conductivity in the laminar flow, heat transfer is absent in principle, and the temperature of the liquid at each place in space does not change. In the extreme case of turbulent flow, heat transfer occurs and leads to a rapid temperature equalization in different parts of the flow. Since intensive heat exchange occurs inside the turbulent region due to strong mixing of the liquid, which is characteristic of any turbulent motion [25].

To determine the flow regime, the Reynolds criterion is used. The mathematical equation defining the Reynolds criterion for a journal bearing is as follows [26, 27]

$$R_e = \frac{\pi d n \rho S}{2\mu_e} \leq 41{,}3\sqrt{\frac{d}{S}} \quad (6)$$

Where d is shaft diameter, m; n is rotational speed of turbocharger shaft, s^{-1}; ρ is density of oil, kg/m^3; μ_e – dynamic viscosity of engine oil, Pa·s; S – diametrical clearance, m.

In most works [10, 13, 15], the flow of fluid in the bearing is assumed to be laminar, not taking into account turbulence at the inlet of the oil supply (Fig. 2). Therefore, in the simulation of cavitation in the journal bearing assembly must be oriented on the turbulent fluid flow regime.

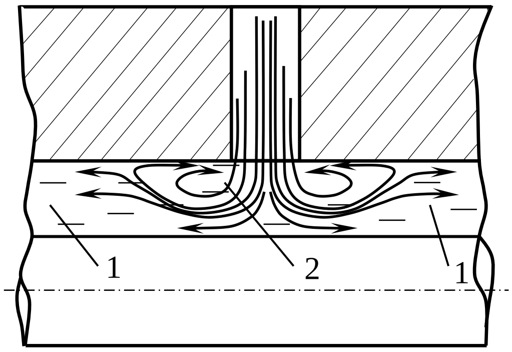

Fig. 2. The Flow regimes in different parts of the bearing:
1 – laminar; 2 – turbulent
Рис. 2. Режимы течения в различных частях подшипника:
1 - ламинарный; 2 - турбулентный

The effect of turbulence has a significant effect on cavitation. The turbulence model should predict the point of separation of the flow from the wall, where a critical vacuum occurs. When analyzing works [23, 24] devoted to cavitation in various mechanisms where engine oil is the working fluid, a tendency was observed to use the k-ε RNG model (the RNG modification increases the accuracy of prediction of separated flows). Since there is no possibility of checking with experimental data to select the most suitable turbulence model, it was decided to use the k-ε RNG model.

For calculation, the Ansys Fluent 19.2 software package is used. The general calculation algorithm has not changed and corresponds to the algorithm from the previous work [16]. Among the innovations, one can single out the introduction of the multiphase flow model, namely, the Multiphase model. This model is suitable for flows loaded with bubbles and particles, which is consistent with the task of modeling cavitation. In the calculation, the Coupled algorithm was used with the Pseudo Transient and High Order Term Relaxation options connected.

When modeling, we used the parameters of motor oil intended for use in diesel forced engines operating in difficult conditions mainly in the summer. All parameters, except the thermal conductivity of the oil, changed depending on the temperature. Table 2 presents the parameters of engine oil and oil vapor.

Table 2

Parameters of oil and oil vapor

Phase	T, K	Density, kg/m3	Specific heat, J/(kg*K)	Dynamic viscosity coefficient, Pa*s	Thermal conductivity, W/ m*K
Liquid	343	869,56	2070,5	0,0368	0,12

	363	857,3	2152,1	0,0194	
	383	845,04	2233,7	0,0116	
	393	838,91	2274,5	0,0092	
	413	826,65	2356,1	0,0062	
	433	814,39	2437,7	0,0044	
Vapor	–	0,0245	–	1,34e-05	0,0261

For the calculation, two thermal operating regimes were chosen: regime I, close to normal operating conditions, and regime II with excess temperature, simulating the operation of a turbocharger when it goes beyond the range of normal operating conditions. The model has 8 inlets and 4 outlets. The boundary conditions of the turbocharger thermal regimes are presented in Table 3.

Table 3

The boundary conditions of the turbocharger thermal regimes

Thermal state option	Field	Pressure, kPa	Temperature, K		
			Turbine side	Middle Outlets	Compressor side
I	Inlet	400	363	–	343
	Outlet	101	383	373	363
II	Inlet	400	413	–	393
	Outlet	101	433	423	413

Using the developed software for calculating the dynamics of a flexible rotor of a turbocharger on two hydrodynamic journal bearings, the relative eccentricity χ was calculated at different speed and thermal regimes of operation of the turbocharger. After that, the geometric model was changed and the calculation process was carried out in Ansys Fluent 19.2. The dependence of the relative eccentricity χ on temperature and rotational speed is shown in Fig. 3.

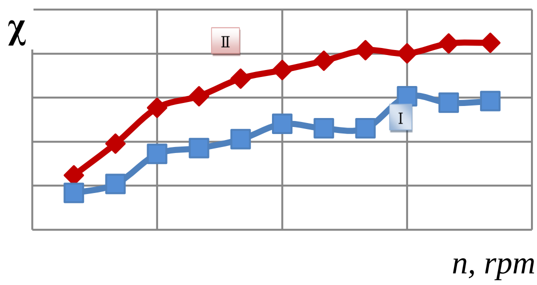

Fig. 3. The dependence of the relative eccentricity χ on temperature regime and rotational speed

Рис. 3. Зависимость относительного эксцентриситета χ от теплового режима и скорости вращения

Results

As a result, the values of various parameters were obtained. Figure 4 compares the fields of the gas volume fraction for different modes of heat loading.

The results show that cavitation prevails in the first (**I**) thermal regime. In this operating mode, steam occupies a larger part of the liquid layer than in thermal regime **II**. The maximum value of the volume fraction of gas is also observed in mode **I**, regardless of the rotation speed. A significant difference (44%) in the gas volume fraction is observed at 20,000 rpm.

Figure 5 compares the distribution of pressures at different modes of heat loading. The results showed that the pressure distribution depends on the relative eccentricity, which depends on the thermal operating conditions of the bearing and the rotor speed. At 10000-30000 rpm, the maximum pressure value is observed in the **I** operating mode from the compressor side, because the dynamic viscosity coefficient is higher due to lower temperatures compared to the turbine side. At 90,000 rpm, the maximum pressure is recorded in mode **II**. This is due to the high value of eccentricity in comparison with the first mode.

Figure 6 compares the temperature fields at different thermal conditions. The rotor speed is 90,000 rpm. In thermal mode **II**, the flow from the inlets is distributed less efficiently than in the first one. This affects the temperature distribution in the lubricating layer. Most likely, this is due to the high value of the relative eccentricity. There are no other significant differences in temperature distribution.

The dependence of the bearing capacity on the thermal regime and rotation speed is shown at Figure 7. The maximum bearing capacity is achieved at thermal regime I, which indicates the need to operate the turbocharger under normal temperature conditions. Thus, the bearing assembly will show the best hydromechanical characteristics.

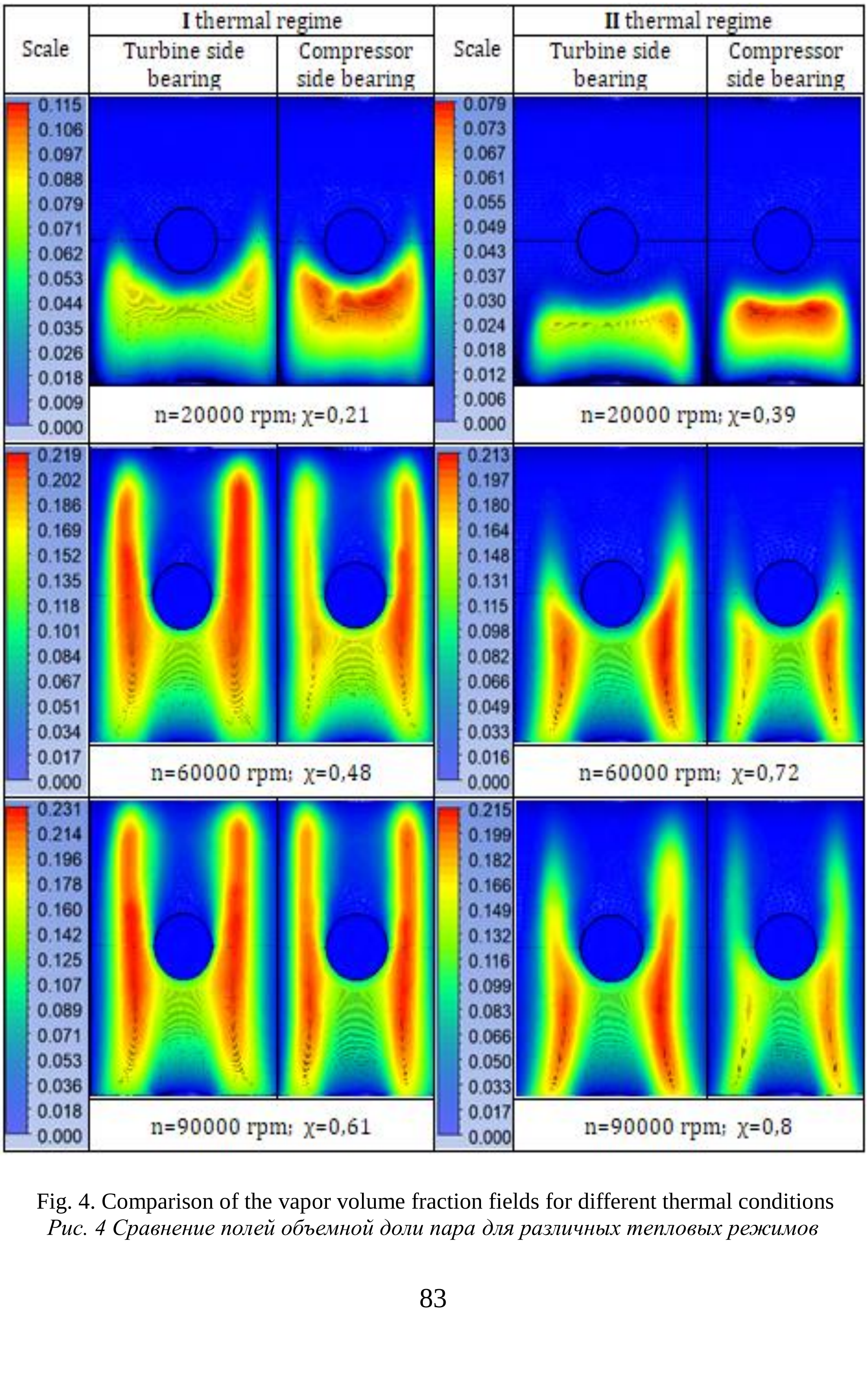

Fig. 4. Comparison of the vapor volume fraction fields for different thermal conditions
Рис. 4 Сравнение полей объемной доли пара для различных тепловых режимов

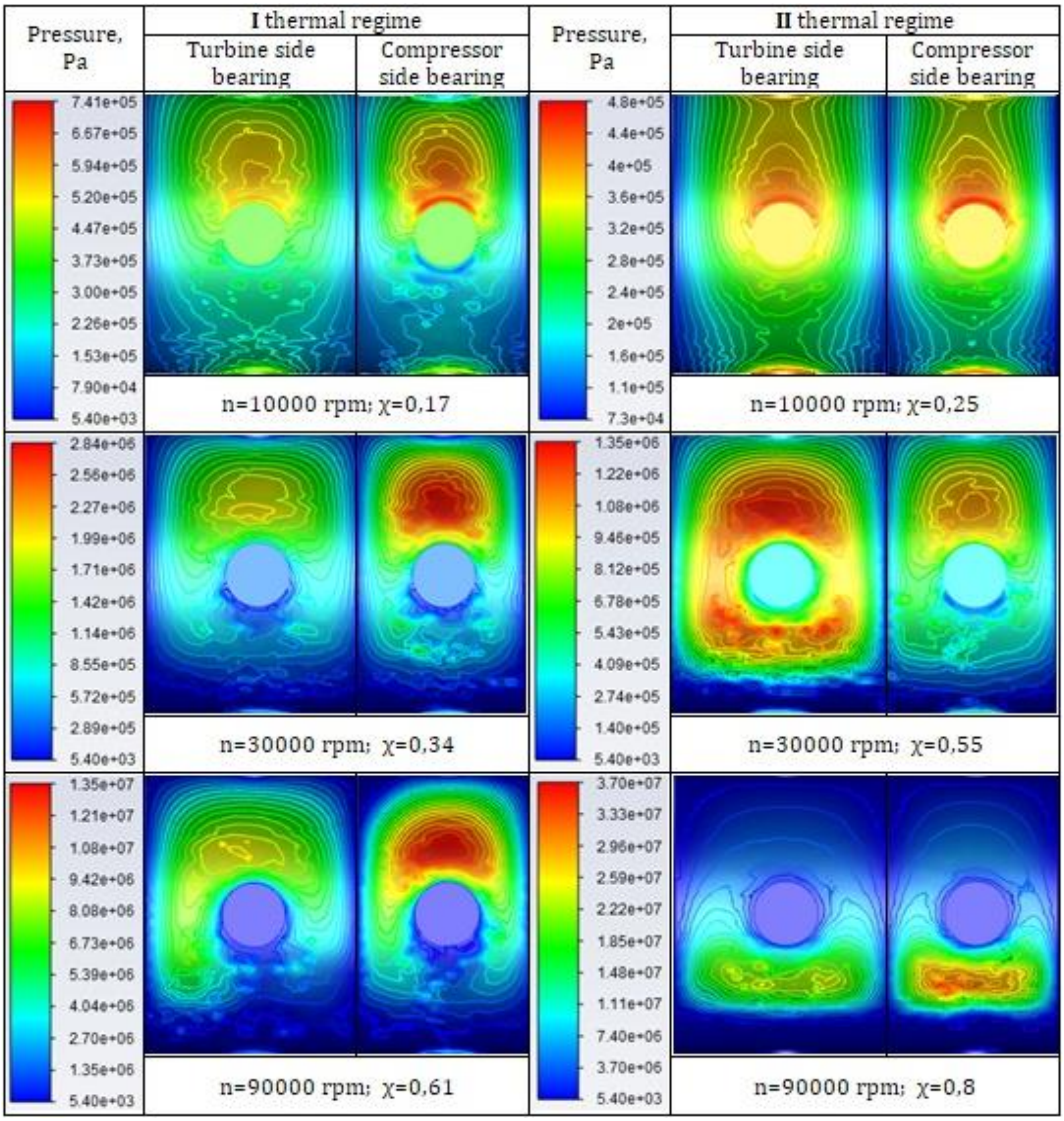

Fig. 5. Comparison of pressure fields for different thermal conditions

Рис. 5 Сравнение полей давления для различных тепловых условий

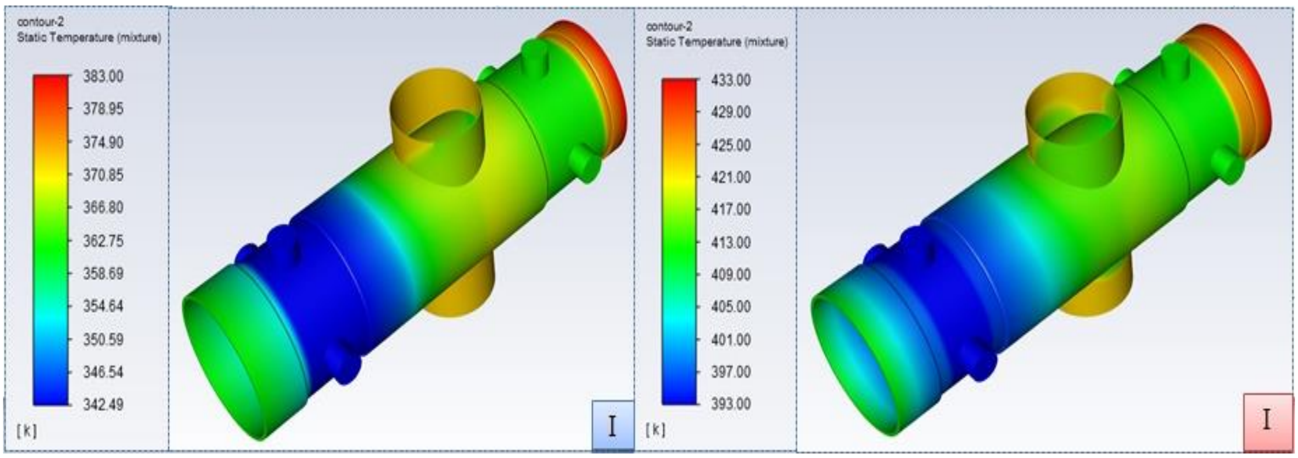

Fig. 6 Comparison of the distribution of thermal fields for different thermal conditions. I, II – thermal mode

Рис. 6. Сравнение распределения тепловых полей для различных тепловых условий. I, II - тепловой режим

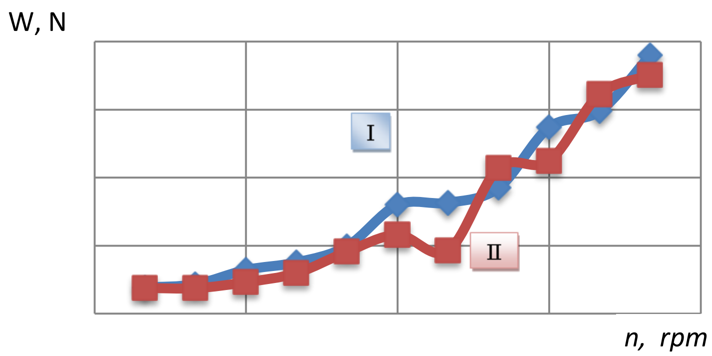

Fig. 7. Dependence of bearing capacity on thermal conditions and rotor speed

Рис. 7. Зависимость несущей способности от теплового режима и частоты вращения ротора

Conclusion

In this work, the simulation of physical processes in the lubricant layer of the real design of the turbocharger radial bearing is carried out. The obtained temperature fields indicate uneven thermal loading of the turbine and compressor bearings. This fact significantly affects the position of the flexible asymmetric rotor in the gap space. The following results were obtained:

• The volume fraction of steam prevails in the **I** thermal mode of operation.

• The pressure distribution depends on the eccentricity, which depends on the thermal operating conditions of the bearing and the rotor speed.

• In most cases, the pressure on the compressor side is higher than on the turbine side due to different fluid parameters.

• In thermal mode **II**, the flow from the inlets is distributed less efficiently than in the first one, which affects the temperature distribution in the layer.

• The maximum bearing capacity is achieved at thermal mode **I**, which indicates the need to operate the turbocharger under normal temperature conditions.

Further studies are aimed at taking into account the misalignment of the shaft relative to the bearing, as well as calculating the heat transfer through the turbocharger housing in order to take into account the obtained temperatures when modelling the hydrodynamics in the lubricating layer.

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