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RESEARCH

Theoretical and Experimental Investigations of the Rotor Vibration Amplitude of the Turbocharger and Bearings Temperature

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A B S T R A C T

One of the most urgent issues of the modern world and domestic automobile and tractor production is the problem of the production of efficient and reliable turbochargers. The rotor bearings largely determine the reliable operation of the turbocharger. By increasing the degree of the forcing of the engine the turbocharger rotor speed and the load increases significantly. Working conditions of bearings also complicated because of the temperature rise. In this case the bearing of the turbine and the compressor bearing works in different thermal conditions. The definition of the thermal state of the bearings can be performed experimentally. However, to perform these studies the sophisticated experimental equipment must be used. Researchers can't perform experiments for each type of turbocharger. Therefore, the applying of the theoretical approaches becomes more relevant. The peculiarity of the considered problem is the design of the bearings, which are made in the form of multilayer bearings with floating rings. Such designs increase the number of the parameters that affect the behaviour of the rotor. For the calculation of the multilayer bearings and turbocharger rotor dynamics a method and calculation algorithm was developed. A plan of the experiment based on the orthogonal central composite plan was drawn up. The regression equations for rotor amplitude and bearing temperature were obtained. As variable parameters the clearances (external and internal), rotor speed, pressure and lubricant temperature were used. The results of the calculation were compared with experimental results obtained at the plant. Non-Newtonian properties of the lubricants were taken into account in the calculations. Comparative results showed good agreement. In this way the resulting function can be applied to studies of the similarly multilayer bearings without complicated experimental studies.

1. INTRODUCTION

The currently, the boost is the main way to achieve the current requirements for power ratings, economic and environmental indicators of internal combustion engines [1]. Manufacturing of efficient and reliable turbochargers is one of the most pressing issues of the modern world and domestic automobile and tractor production. The rotor bearings largely determine the reliable operation of the turbocharger. By increasing the degree of forcing engine the turbocharger rotor speed and the load of the rotor is significantly increased. Working conditions of bearings are also complicated because of the temperature rise [2-4].

Kucinschi B. and Fillon M. [5] presented a study, which shows the determination of the temperature distribution in the plain journal bearing, which is loaded with a constant force. The main focus was on hydrodynamic pressure, temperature distributions at the film/bush interface, oil flow rate, power losses and film thickness. The obtained results demonstrated that: for highly loaded bearings operating at low speed, critical parameters are film thickness and maximum pressure; for high speed operating bearings submitted to a light load, the shaft being almost centered, heating becomes very significant and maximum temperature becomes the critical parameter.

Pankaj Khatak and H.C. Garg [6] analyzed of the capillary compensated hybrid journal bearing by considering combined influence of thermal effects and a micropolar lubricant. The results obtained by numerically indicate that the bearing is significantly affected by increase in temperature. Hence, it is essential to consider the thermal effects for bearing operating with micropolar lubricant to produce realistic bearing characteristic data.

The shearing of lubricant at high loads and speed in journal bearings results in generation of large amounts of heat. The lubricant viscosity will decrease with increase in temperature. Many authors have studied the thermal effects of lubricant on the performance of hybrid journal bearings.

Moes H. et al. [7] presented a method of taking into account the thermal effects when simulating dynamically loaded, flexible, journal bearings. The method is an extension of Van der Tempel's model for a flexible, short journal bearing under severe dynamic duty, with partial circumferential supply grooves, cavitations and etc. It incorporates the heat generation due to the viscous dissipation and the cooling due to the axial and circumferential flow of lubricant. The results for the film thicknesses, the lubricant flow, the viscous dissipation and the temperature distribution in two specific connecting-rod bearing configurations were presented.

A theoretical model of slot entry hybrid bearing was developed by Sharma et al. [8], in which viscosity variation due to thermal effects was considered. It was reported that minimum fluid film thickness reduces for all the configurations of bearing due to thermal effects. They further reported that lubricant supply, stiffness, and damping coefficients change appreciably when thermal effects are considered.

Garg et al. [9] developed the slot entry hybrid bearing model to include the non-Newtonian and thermal effects of lubricant. It was observed that oil requirement in hybrid journal bearings was enhanced by increase in temperature and non-Newtonian effects in lubricant. These available studies indicate that performance characteristics of journal bearings are significantly affected by the temperature increase of the lubricant.

J. Yavorova's paper [10] presents an investigation of the characteristics of the bearing with finite length, taking into account the influence of Rabinovich's non-Newtonian rheological system and the elastic deformations of the support liner. The authors concluded that, in comparison with Newtonian lubricants, higher film pressures and capacities for lubricants with a dilatant were obtained. This work once again confirms the need to take into account the non-Newtonian properties of the lubricant.

Thermal analysis of finite journal bearings with micropolar lubrication has been performed by few researchers [11, 12]. Khonsari and Brewe [11] compared the isothermal results of the micropolar lubricated finite journal bearing with thermohydrodynamic solutions. They showed the significant reduction in load-carrying capacity of the journal bearings when thermal effects are considered.

Edgar J. Gunter presented some results of the linear and nonlinear dynamics of the rotor, which rotates in the rotating floating ring bearings for the typical turbocharger [13]. He computed the linearized stability of the system for the various ratios of inner and outer clearances. He also represented the analysis of the critical speed and showed several fluctuation forms of the rotor.

A. Tartara [14] had conducted experiments and concluded, that remarkable effects of stabilizing system can be expected, when the bearings with floating rings are used. Namely, the ring starts rotating as soon as the whirl occurs, and the further stable operation is realized as the journal speed increases. The stabilizing effect of the floating-ring bearing is conspicuous when the ratio between inner and outer clearances and the absolute clearances are large.

Additionally, researchers should take into account the mutual influence of multi-layered bearings on each other. For example, in paper [15] authors presented an algorithm for solving the problem of rotor dynamics. The authors take into account the flexibility of the rotor. An algorithm for solving thermal problem for the plain bearing is represented in papers [16,17].

However, experimental studies [18,19] had shown that the bearings of the turbine and compressor operate in different thermal conditions. The temperature difference for the turbine and compressor bearings can be 30…50 °C.

Studies on thermal aspects of bearings considered with different bearing temperatures are very limited.

Consequently, the evaluation of the thermal state of each bearing and accounting them thermal state in the calculation of the rotor dynamics is an urgent task. At the same time the position of the rotor relative to the bearing must be determined at each time point. The flexibility of the rotor must be taken into account.

An attempt has been made in this paper to seek more realistic functions for the amplitude of the oscillations of the rotor and the temperature in each bearing.

2. METHOD OF THE THEORETICAL ANALYSIS

The problem of the hydrodynamic theory of friction units is characterized by a set of methods for solving several interrelated tasks: the determination the conditions of stability and parameters of the journal nonlinear oscillations on the lubricating film; the calculation of its trajectory; the calculation of the field of hydrodynamic pressure in the lubricating layer, which separates the friction surfaces of the journal and bearing, taking into account an arbitrary law of their relative motion; the calculation of the temperature of the lubricating film.

The methodology of the dynamics calculation of the flexible non-symmetric rotor on the multi-layer sliding bearings is based on methods of integrating the motion equations of movable elements of the bearings and rotor. The motion equations include forces that are associated with the presence in the system "rotor – bearings" the lubricating layers having substantially nonlinear characteristics. Bearing with rotating rings containing two lubricant layers and the direction of movement of lubrication is shown at Fig. 1.

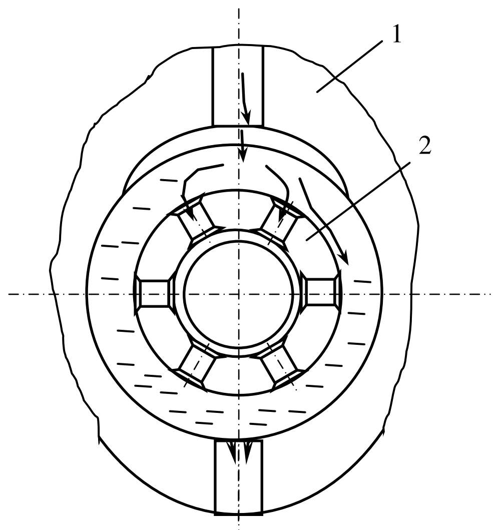

Fig. 1. Bearing with rotating rings: 1 – turbocharger housing; 2 – rotating rings.

The nonlinear reactions of the lubricating layer are determined by integrating the hydrodynamic pressures diagram. These diagrams are calculated at the each time step by numerical integration of the differential equation Reynolds.

Among the factors considered in the modeling of the rotor vibration at the bearings with floating the rotating ring (contains two lubricating layers, Fig. 1) is included the gyroscopic and inertial forces, and the kinematic excitation, resulting from the fluctuations of the foundation. The use of the multigrid method [20] for the numerical solution of the Reynolds equation allows one to take into account movable and immobile sources of lubrication in various forms, which are performed on the friction surfaces of the hull and bushings. These sources are taken into account by setting the boundary conditions for the pressures to the grid step.

The non-isothermicity of the lubricant flow is taken into account by the correction of its viscosity at each step of calculating the trajectory of the rotor and the floating bushings [16].

The degree of perfection of bearings with floating bushings is estimated by calculations of the stability characteristics of mobile elements and hydro-mechanical characteristics [21], which include friction losses, dissipation in lubrication, the thickness of the lubricating layer and temperature, the values of the hydrodynamic pressures. Those characteristics allow you to directly or indirectly estimate the heat release rate and the fatigue life of bearings, wear and tendency to galling of friction surfaces. The system of equations of the motion for a flexible asymmetric rotor, which leans on multilayer bearings, is presented by V. Prokopiev et al. [21], P. Taranenko et al. [15].

The lubricant is considered as a design element of any friction units. Non-Newtonian properties of the lubricant had been described by the function presented in [22]. The system of equations of motion of the rotor elements are integrated by the Runge-Kutta method. Variable step integration over time is automatically selected by using Merson amendment. An algorithm for solving the task of rotor dynamics has been presented in the article [22].

3. EXPERIMENT

The experiment was carried out by employees of RPA "TURBOTEKHNIKA". To estimate the thermal state of the rotor elements and bearings the thermocouples were used. Measurement range is 0 ... 350 °C. The installation scheme of thermocouples is shown in Fig. 2.

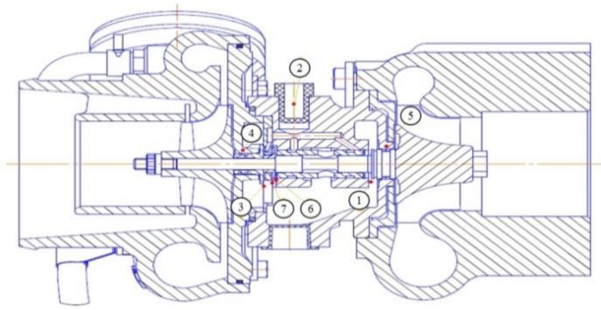

Fig. 2. The scheme of installation of thermocouples.

The figure shows: 1 is the draining the oil from the radial bearing from the turbine side; 2 is the oil supply to the bearing housing; 3 is the draining of oil from the thrust bearing from the compressor side; 4 is the seal on the compressor side; 5 is the seal on the side of the turbine; 6 is the draining of oil from the radial bearing from the compressor side; 7 is the draining the oil from the thrust bearing from the turbine side. Additional channels were made in the bearing housing to installation and attachment of thermocouples.

The measurements were carried out in 5 stages. The input parameters, varied during the tests, are given in Table 1.

The airflow through the compressor was measured at 5...6 points on each branch of the compressor's characteristic from the surging boundary to the stop point.

Table 1. The input parameters .

Parameter	Step number 1	2	3	4	5
Circumferential speed at the outer diameter of the compressor wheel, m/s	250 400 500 550	400 500	400 500	400 500	400 500
Oil temperature at the entrance to the bearing housing, ^{0}C	90	90	70	90	105
Input oil pressure, MPa	0,4	0,3	0,4	0,14	0,4

The parameters to be registered are:

- temperatures at the points of installation of thermocouples;
- the temperature at the oil drain from the turbocharger (bench thermocouple);
- degree of pressure increase in the compressor;
- the oil flow through the turbocharger;
- the air temperature at the compressor output;
- the gas temperature at the input to the turbine.

4. RESULTS OF EXPERIMENT

The results of the first stage showed that at a peripheral speed of 250 m/s, there is a significant difference in the temperature of the oil from the compressor and turbine side. From the turbine side, the oil temperature on the drain from the radial bearing is 6 ... 7° higher. At circumferential speeds of 400 ... 550 m/s, there is no significant difference.

The maximum temperature of the oil was 122 °C, that is, the gain relative to the inlet temperature is 30°. The maximum oil temperature in the thrust bearing was ~121 °C, and it does not significantly differ from the temperature in the radial bearing.

The air temperatures at the output of the compressor and gas at the input to the turbine are shown at Fig. 3.

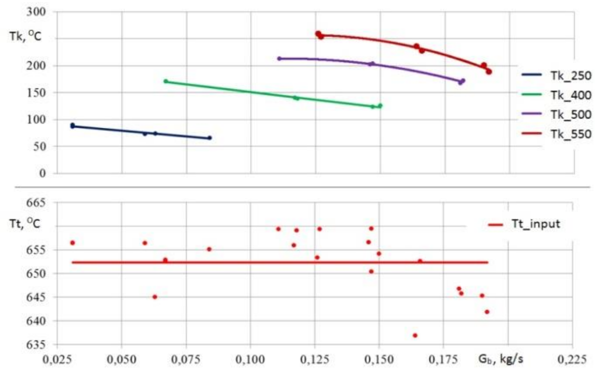

Fig. 3. Air temperature at the output of the compressor (Tk) and gas at the input to the turbine (Tt) from the air flow through the compressor Gb.

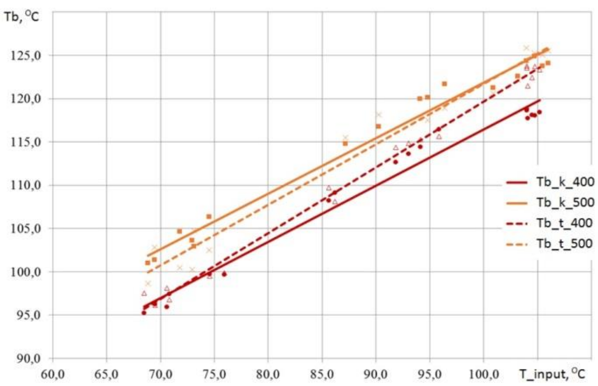

Fig. 4. The oil temperature (Tb) in the radial bearing.

At the second stage of the experiment, it was found that when the supply pressure of the lubricant increases, the temperature in the radial bearing increases to 5.5 °C.

The results of the measurements in the fourth stage when the oil supply pressure was reduced to 0.14 MPa showed that the increase the oil temperature on the drain from the radial bearing was 7...10 °C from the compressor side and 10...15 °C from the turbine side. The dependence of the oil temperature at the radial bearing drain on the oil temperature at the input to the turbocharger is shown in Fig. 4. The results of measurements in the fifth stage showed that the temperature of the oil on the drain from the radial bearing increased by an average of 12.5 °C.

5. RESULTS OF CALCULATION

A plan of the experiment based on the orthogonal central composite plan was drawn up. The regression equations for the function of the rotor amplitude and bearing temperature were obtained. As variable parameters the clearances (external and internal), rotor speed, pressure and lubricant temperature were used. The coefficients of the regression equation for the fluctuations amplitude of the journal and the bearing temperature are shown in Table 2.

Table 2. The coefficients of the regression equation.

Coefficients	The fluctuations amplitude of the journal	The bearing temperature
b_0	-12,7034	88,54901
b_1	0,788164	-4,5208
b_2	-0,46852	7,919623
b_3	0,00019644	0,00285239
b_4	-12,4847	17,74234
b_5	0,174299	-0,12758
b_{12}	-0,03059	0,040477
b_{23}	-7,9513·10-6	-8,7008·10-6
b_{34}	0,000545	0,000367
b_{45}	0,083428	-0,02839
b_{11}	-0,00854	0,048233
b_{22}	0,088224	0,36404
b_{33}	-7,7·10-9	-7,3·10-8
b_{44}	-8,93225	-44,9515
b_{55}	-0,00109	0,005644

Figures 5 show the precession amplitude of the rotor and the bearing temperature by external and internal bearing clearances. In this case the oil pressure is 0,05 MPa, temperature is 95 °C.

With increasing the rotor speed, the precession amplitude of the rotor decreases, the oil temperature increases. In the range of rotor speed from 14500 c^{-1} to 19100, the changes of estimation parameters are slightly less than 0.5 microns for precession amplitude of the rotor and less than 10°C for the bearing temperature.
When the clearances were changing within tolerance the maximum amplitude of the rotor precession is about 7,5...11 μm, the maximum bearing temperature is about 110...120 °C.

Figure 5 shows the dependence of the precession amplitude of the rotor and oil temperature at the output of the bearing on the outer and inner clearances in the bearing at various rotational speeds of the rotor. The oil pressure is 0.05 MPa, the temperature is 95 °C. At the Fig. 5 the internal clearance is assumed to be constant and equal to the maximum values.

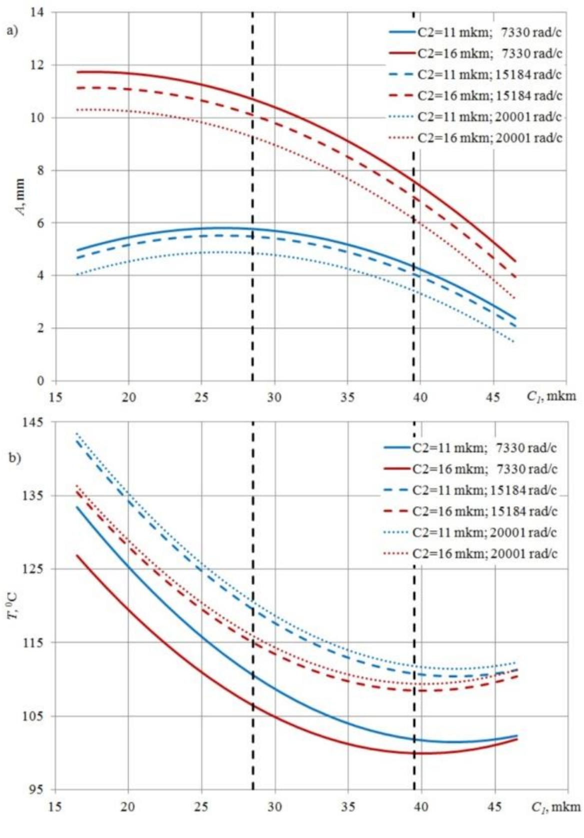

Fig. 5. The precession amplitude of the rotor (a) and the bearing temperature, (b) depending on the outside clearance at various rotor speeds.

The results of the calculation showed that the amplitude of the precession of the rotor decreases with increasing the external clearance. The temperature at the exit from the bearing is minimal in the range of external clearances of 40...45 μm and it grows when the clearance is changed.

With increasing the rotor speed, the precession amplitude decreases, the oil temperature at the output increases. If the clearances vary within the tolerance, the maximum amplitude of precession of the rotor is 7.5...11 μm, the maximum temperature at the exit from the bearing is 110...120 °C.

Figure 6 shows the dependences of the estimated bearing parameters on the ratio of the outer clearance to the internal clearance (C1/C2).

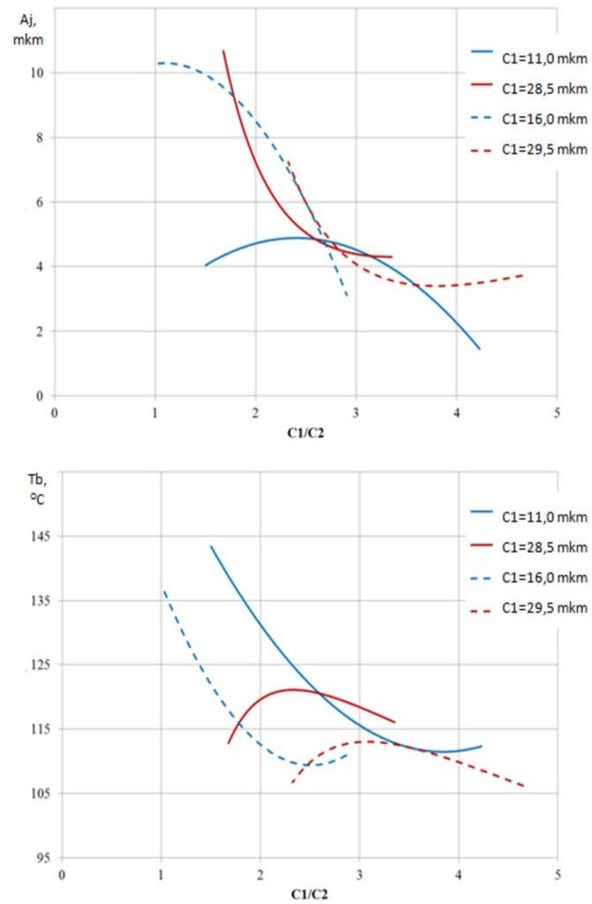

Fig. 6. Dependence of the estimated bearing parameters on the clearance ratio.

It can be seen from the graph that the optimal interval for the clearance is 2.5 ... 3.

At the last stage, the results of the calculation of the bearing temperature were compared with the experimental results. The results of comparing the calculated bearing temperature with the results of the experiment are shown in Figs. 7 and 8.

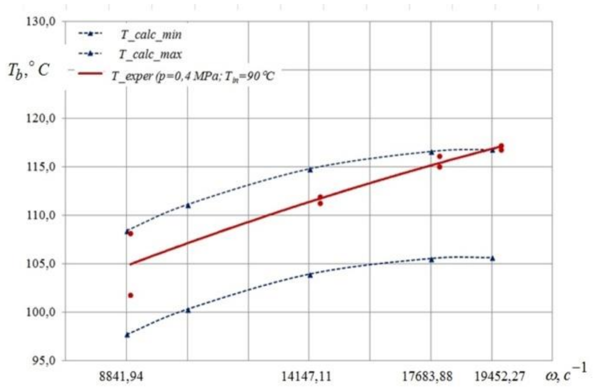

Fig. 7. The dependence of the bearing temperature on the rotation speed: p=0,4 MPa; Tin=90 °C.

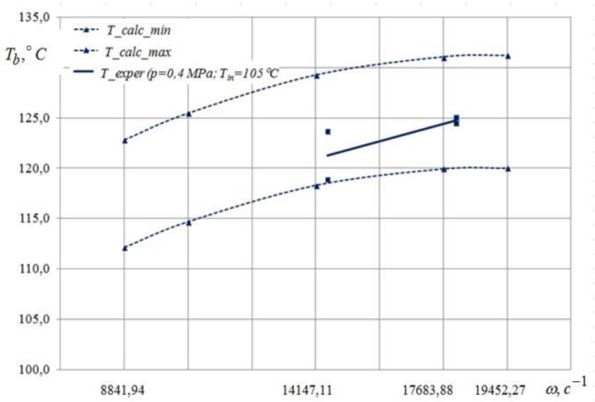

Fig. 8. The dependence of the bearing temperature on the rotation speed of the rotor. p=0,4 MPa; Tin=105 °C.

Comparative results showed good agreement.

6. CONCLUSION

The method of calculating the dynamics of the flexible asymmetric rotor on the radial plain bearing can be used for research of the turbocharger with other size. Regression equations were obtained for the rotor amplitude and bearing temperature, and were verified. Calculation results showed that the precession amplitude of the rotor at all modes is less than 11 microns. This indicates that the direct contact of the rotor and the radial bearing rings is absent. The increment of temperature in the bearing is 15...25 °C and only slightly dependent on the lubricant supply conditions.

The regression function can be applied to studies of the similarly multilayer bearings without complicated experimental studies.

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