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Forming Laminar Flow of Engine Oil Under Conditions of High-Speed Sliding Friction

V. I. Kubich, E. A. Zadorozhnaya and O. G. Cherneta

Abstract The state of engine oil flow in hydrodynamic couplings is determined not only by geometrical or thermo-mechanical parameters of the interacting friction surfaces, but also by the flow regime. Due to heavy operating conditions of plain bearings in some regimes, self-organization processes on friction surfaces are inefficient. This negatively affects the continuity and damage resistance of forming lubricants. The regime of lubricant flow against the friction surfaces changes during operation. The engine oil flow frequently makes a transition from a laminar flow to a turbulent state. The article reports the results of evaluating possible operating regimes and conditions which enable the engine oil flow in the plain bearing of the turbocharger shaft makes a transition to turbulence. This allowed us to determine the thicknesses of likely boundary layers about the radius of surface roughness and compare them with the total thickness of the possible lubricating layer in a journal-bushing coupling. The proposed approach makes it possible to evaluate the overall state of the oil flow from the point of view of its layer pattern and the motion pattern of its components, which is necessary to improve the methods for ensuring reliability of lubricating the bearing of a particular turbocharger.

Keywords Reynolds number · Turbulent flow · Turbocharger · Boundary layer · Lubricating formations · Oil component

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1 Introduction

The development of measures for improving the reliability of bearing assemblies, operating under the conditions of high-speed sliding friction, implies the availability of data that determine both the surface loading condition and manifestation of the rheological properties of lubricants flowing through clearances of certain sizes. The range of high shaft speed in the friction assemblies of automotive engines is quite wide, for example, from 3000 to 6000 min^{-1} in tribological assemblies of crankshafts and camshafts, and from 75,000 to 220,000 min^{-1} in turbocharger shafts. The state of engine oil flow in such couplings is determined not only by geometrical or thermo-mechanical coupling parameters of the interacting friction surfaces, but also by the flow regime. Due to heavy operating conditions of plain bearings in some regimes, self-organization processes on friction surfaces are inefficient, and this negatively affects the continuity and damage resistance of forming lubricants. This is due to mechanical damage caused by insufficient oil in the friction zone. It should be taken into account that oil is a flowable substance where changing regimes of flow against the friction surfaces in general and the elements of their surface contour in particular can happen, that is, a possible transition of engine oil flow into a turbulent state. A characteristic feature of the turbulent state is vortex motion of lubricant layers. This puts questions about the possibility of forming stable lubricant structures on friction surfaces and their functionality under extreme operating conditions. Answers to the formulated questions make it possible not only to expand the established scope of knowledge about the formation of protective boundary layers on the friction surfaces of engine parts limiting the operational life of critical friction assemblies, but also to develop measures for improving fluid lubricant control.

2 Literature Review

Presently, much attention is paid to the issues of engine oil flow through sliding friction assemblies of automotive engines and, in particular, through the bearing clearances of a crankshaft or a turbocharger shaft [1–7]. For example, [1] developed and presented methods for diagnosing main and connecting rod bearings for automobile and tractor engines by measuring pressure pulsations in the main oil gallery. Having developed some supplementary equipment, the authors managed to establish a correlation between the pressure signal quantity and the technical condition of main bearings. In [2–4, 6], the team of authors conducted a series of theoretical and full-scale experimental studies on installing supplementary equipment into the lubrication system to increase its operational reliability. The turbocharger was equipped with a hydroaccumulator and a brake device to reduce the rotor rundown time, and cooling and lubrication of the shaft during the engine shutdown by forcible inlet of engine oil to the friction zone using a supplementary pipe and hollow bolts. Research results showed that reduction in the rotor rundown time to 37–57 s, depending on the operational mode, excludes the possibility of semi-dry or dry friction modes, which leads to possible extension of the operational life of the turbocharger.

The research [5] presents a methodology for evaluating and interpreting the diagnostic information, based on the studies presented in [1]. The proposed methodology makes it possible to diagnose engines without disassembling them and to evaluate the wear degree of plain bearings in the slider-crank mechanism according to the parameters of pressure pulsation in the central oil gallery. In [7], the team of authors established the correlation between the rotational speed of a turbocharger and the lubrication criterion λ at different temperatures of engine oil. This research established the correlation between the radial force that loads the bearing and the shaft speed. In general, the obtained results indicate that lubricating formations fail always to withstand the effects of the coupled element, that is, the turbine shaft, caused by the magnitude of the radial force. And this, in turn, leads to switching to lubrication regimes that are adverse to the resistance of bearing material to wear.

When studying the flow state of liquids and gases with different rheological properties in specific operational modes of couplings, it is always necessary to determine the Reynolds number. This will determine above all the laminarity or turbulence of layered flows in the total flow, washing the geometric profiles of surfaces, which determine, for example, the thicknesses of the boundary layers. Hydromechanical problems that are solved using the Reynolds number are considered in various applied and fundamental research and development tasks in different fields of technology, and were published in a number of scientific journals, including [8–11].

Thus, in [12, 13], the authors used the Reynolds number to evaluate the oil flow, which is fed to the friction assemblies of the crankshaft, camshaft, and turbocharger units at different viscosity and temperature parameters of engine oil, geometrical parameters of plain bearings, and operational speed parameters of engine samples. It was found that flow turbulization of the engine oil flow in bearing clearances is theoretically possible in certain operating modes. In [12, 13], the flow through operating clearances was considered. However, discontinuity of the oil flow, which leads to lubrication of surfaces with residual material in the clearance, was not considered, and the possibility of flow turbulization of the residual oil layers was not evaluated. This mode of interaction between surfaces is possible in the general case of delayed oil inlet to the friction zone, and in particular, when the mixed lubrication regime starts. The mathematical model of lubricant material presented in [14] took into account lubricant films in the form of adsorbed layers that are formed straight on the friction surfaces. The effect of polymolecular adsorption on the rheological behavior of lubricating oil in a thin layer was described in [15–17]. However, these research works failed to present an experimental study of friction assemblies in machines where the described lubricant flow regime would be implemented.

Thus, the results of the analyzed works indicate the relevance of evaluating the state of the engine oil flow in the engine friction assemblies, since it determines both the possibility of diagnosing these assemblies in general and the formation of lubricant films straight on the surfaces of engine parts.

2.1 Problem Statement, Goal Setting

The engine oil flow is a structural component of almost every tribological coupling with hydrodynamic, boundary, or mixed lubrication regimes according to the operating conditions. The plain bearings of the turbocharger shaft of automotive engines are considered as tribological "journal-bushing" couplings with a changing pattern of engine oil flow. However, such lubrication regimes are sometimes inconstant, since the thickness of the lubricating layer changes depending on a number of parameters. These are dynamic viscosity η, contact load N, and parameters of surface roughness $R_{a1,2}$.

If plain bearings, where the reverse flow zone is more extensive, have a large length, the static eccentricity is significant (ε > 0.3–0.4), and the Reynolds number is high (Re > 200), then vortex circuits are to be expected in the wide part of the lubricant layer, which disturbs the lubricant flow in the adjacent parts of the lubricant layer.

When Re is more than 2300, the flow is turbulent. Since lubrication systems of automotive engines usually use viscous oils, and the speed in the galleries is low (0.5 l/s), the oil flow is always laminar.

According to Couette, the flow in plain bearings is laminar at Re < 1900 and turbulent at Re > 1900. Radial bearings with hydrodynamic lubrication are calculated using the method of successive approximation or the method of variation by improving structural and operational factors in order to achieve low work wear and favorable energy costs taking into account the available oils and their viscosity.

The stability of a laminar flow is influenced by breakdown and cavitation of liquid lubricant. These phenomena significantly disturb the evenness of fluid flow and the boundary conditions of flow. Moving cavities and splashes cause significant disturbances in the form of random local eddies. Such eddies do not form the usual disordered turbulent flow, as they are of a larger scale, lower frequencies, and lower stability. Nevertheless, they can significantly increase the viscous resistance and change the acting hydromechanical forces.

The critical values of the Reynolds numbers also depend on the roughness of the bearing surface. Moreover, a rough surface has lower critical Reynolds numbers than a smooth surface. When the bearing surface is washed by oil, a turbulent boundary layer may be formed from the very beginning (there are no laminar zones and transition zones in this case).

Thus, according to [18], the minimum thickness of the lubricant layer, which maintains the conditions for hydrodynamic lubrication, is determined according to the expression:

$$h_{\min} = 0.57\frac{\eta \cdot U \cdot d \cdot l}{N}, \tag{1}$$

where d is the diameter of the bearing shaft, m; l is the bearing length, m.

Taking into account the established tribological approach to the evaluation of lubrication regimes according to the criterion λ [18], the parameters of expression (1) will determine its numerical values as follows:

$$\lambda = 0.57\frac{\eta \cdot U \cdot d \cdot l}{N\sqrt{R_{a1}^2 + R_{a2}^2}}, \tag{2}$$

where $R_{a1,2}$ is the arithmetic average value of roughness profile of the contacting surfaces.

As follows from expressions (1, 2), the main parameters that determine the change of lubrication regimes under the conditions of steady regular roughness of interacting surfaces are the directly proportional sliding speed and viscosity, and the inversely proportional load and roughness parameter. In this case, the flow rate υ_c of the lubricant will be determined by the rotation speed of the tribo-assembly movable element in this case, by the sliding speed in contact $\upsilon_c = f(U)$. However, the movement of separate layers has different velocity gradients depending on the distance from the surfaces of the coupled elements.

Depending on the flow regime, there are laminar and turbulent boundary layers. The latter is characterized by intensive mixing, formation of eddies, velocity pulsations, significant transverse velocity, heat and mass exchange with the external flow [19]. Due to the smooth transition of the boundary layer to the flow core, it is assumed that the boundary layer ends at the thickness δ, where the rate differs from the external flow rate by 1% (Fig. 1).

As the wall is washed by the external flow with a slightly varying velocity, the thickness of the boundary layer δ increases. The layer is laminar at first (Section I). but under certain conditions, it begins to make a transition to turbulence due to the formation and intensification of pulsations. Section II, where this process occurs.

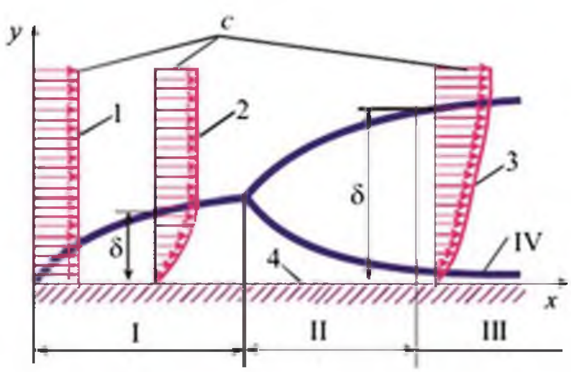

Fig. 1 Diagram of the boundary layer when the flat wall is washed by the flow: I—laminar regime; II—transition zone; III—turbulent regime; IV—laminar sublayer; 1—velocity profile of the ideal flow; 2—the same in the laminar layer; 3—the same in the turbulent layer; 4—the wall that is washed by the flow [19]

is called transitional, and in section III, the layer can be considered turbulent. However, there is also a thin laminar sublayer between the wall and the turbulent layer.

The thickness of the turbulent layer is remarkably larger than that of the preceding laminar layer. The turbulent layer has a more complete velocity profile than the laminar one (see the velocity profiles in Fig. 1), which is explained by intensive mixing inside the layer.

It is difficult to determine the characteristics of the non-laminar lubricant flow experimentally, because different regimes, ranging from ordered Taylor vortex flows to disordered turbulent flows, can intervene.

The vortex zone acts on the rest of the lubricant flow like a solid body of the same configuration that has been put in its layer. This reduces the wedge part of the lubricating layer in its widest part. The lubricant in this area makes a transition to turbulence only in the weakly loaded part of the lubricating layer, and therefore, the flow turbulization has little effect on the magnitude of hydromechanical forces acting, for example, on the shaft journal. Actually, the expanding part of the liquid lubricating layer is usually cavitated due to the lowering pressure in the main flow, and the lubricant part, where eddy currents could occur, breaks down.

Breakdown and cavitation of liquid lubricants—lubricating formations with variable rheological properties—have a great influence on the stability of the laminar flow. These phenomena significantly disturb the evenness of the fluid flow and the boundary conditions for the flow of its layers. Moving cavities and splashes cause significant disturbances in the form of random local eddies. Such eddies do not form the usual disordered turbulent flow, as they are of a larger scale, lower frequencies, and lower stability. Nevertheless, they can significantly increase the viscous resistance and change the acting hydromechanical forces.

The velocity gradients of the lubricating layers may be as follows.

First, they may be increasing $-\mathrm{d}\upsilon_c/\mathrm{d}h > 0$, which is predetermined by a preformed boundary layer, that is, lubricant formations of polyatomic carbon-hydrogen molecule chains. The latter are characterized by certain cohesive bonds within the structure and adhesive bonds with a set of active centers of the surfaces. These are polymolecular layers with a different shear strength.

Second, they may be constant $-\mathrm{d}\upsilon_c/\mathrm{d}h = 0$, when $\upsilon_c = 0$ or $\upsilon_c = \text{const} > 0$, which is predetermined:

- on the one hand, by the lacking boundary layers—when they have been destroyed for some time and have not been formed yet, which is caused by the critical values of contact loading parameters;
- on the other hand, by the laminar state of the flow when the lubricant films are formed again, which is characterized by the constant velocity of conditional points moving along the streamlines. In this case, the conditional points of the streamlines can be considered as the motion of lubricant components in the form of individual polyatomic molecules.

Third, they may be variable $-\mathrm{d}\upsilon_c/\mathrm{d}h \geq 0$, $\mathrm{d}\upsilon_c/\mathrm{d}h \leq 0$, which is predetermined by the transition from the laminar flow state of the lubricant components to the turbulent state.

It is an established fact that in case the laminar flow regime of lubricant fails, the friction moment acting on the journal and on the bearing increases. The hydrodynamic force supporting the journal slightly increases, but reliable measurements of this force for non-laminar lubricant flow regimes are unknown. There are very little data about the components of hydromechanical forces during lubricant fluctuations [20]. According to the known measurements, the pressure profiles of the turbulent lubricant flow do not differ significantly from those of the laminar flow and sometimes are within the margin of measurement error.

The next issue that is important for high-speed shafts in mechanical engineering structures is transition from the laminar flow regime of lubricant, for example, engine oil, to a turbulent one. The latter increases the moment of friction, enhances the heat production, and therefore, has a smaller flow of lubricant components than the laminar regime.

The probability, with which adhesive–cohesive bonds will be formed in new lubricant film structures of certain thicknesses, will be determined by the motion rate of lubricating medium components and, accordingly, by the time spent on energy exchange and interaction of force fields. It seems obvious that the lack of ordered motion of lubricating medium components with different molar masses will cause chaos in the formation of stable bonds. This is especially important for surface-active substances, which, for example:

- are responsible for coordinating the position of polyatomic molecules, for example, $CH_3(CH_2)$, with respect to the active centers on the metal surfaces;
- participate in their aggregation process in suspension and form a liquid crystal layer of the boundary film, healing the damaged layers on the surfaces [20].

Thus, the lack of ordered motion of the interaction medium components, that is, additives, can be represented with a model of electrohydrodynamic flows. The higher the intensity and heterogeneity of the field between the electrodes and the concentration of the polar phase in the oil volume are, the stronger they are [21]. We interpret this model as an ordered laminar motion of components, Fig. 2a, and a disordered turbulent motion of components, Fig. 2b, against a conditional surface with active centers. Moreover, the authors discuss the mechanism of boundary lubricating layer formation under the conditions of electromagnetic action on oil additives. The researchers consider a kind of inverse problem, that is, control of forced flow turbulization of the lubricant medium in order to intensify the mixing of lubricant components, which generally leads to increased physical adsorption of additive molecules on metal surfaces, the formation of a more durable multimolecular boundary layer of lubricant, and the improved tribotechnical indicators of friction couplings.

However, the lubricating medium in the form of engine oil does not consist only of additive compositions with surface-active substances, but also of base oil with

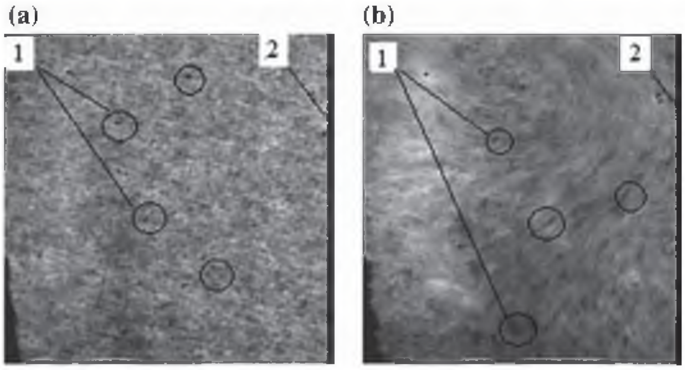

Fig. 2 Photomicrographs of a LiquiMoly 10W40 oil sample on glass in the interelectrode space, magnification—×1000: **a** voltage on the electrodes $U = 0$; **b** voltage on the electrodes $U = 500$ V; 1—medium components; 2—conditional metal surface

complex molecules that form its structure as a whole. This composition is certainly characterized by viscous friction forces, which make the fluid adhere to the surface of the washed element [22]; that is, they keep the oil components in the rest state. According to the theory of the boundary layer, created mainly by L. Prandtl, under certain flow conditions, the engine oil components form such a boundary layer, where the viscous friction and pressure forces are of the same order and the velocity increases as it gets farther from the body surface [22]. In any case, regardless of breakdown and repair of protective lubricant formations, there are boundary layers on the surfaces that are washed with oil. But it still remains unanswered: What inherent properties of intensity and time of action do they have under the conditions of change of the hydraulic state? It is also interesting to evaluate the percentage of the components of surface-active substances in the boundary layer during transitions from one lubrication regime to another. This knowledge is especially important for controlling the processes when restoring effective boundary layers in tribological systems of materials with different surface activity.

The boundary layer thickness δ is not a well-defined concept, since the layer boundary on the liquid side is not sharply outlined, but it can be determined as follows [22]:

$$\delta \approx \frac{l}{\sqrt{Re}}, \tag{3}$$

where l is the characteristic linear size of the washed element; Re is the Reynolds number.

This approach in evaluating the state of engine oil flow near the washed surfaces theoretically determines the formation of a shielding boundary layer with favorable conditions for the formation of adhesive bonds of oil components with active metal centers. However, the velocity of layers in the boundary layer rapidly increases as it gets farther from the surface. This together with the variable velocity gradients of components in the bordering zone, which is typical of the flow turbulization effect, causes fragmentary blocking of their direct passage to the surface. Thus, a blocking layer of different density is formed.

The eccentricity of the bearing journal of the high-speed shaft, while it is not large ($\varepsilon < 0.3$), has almost no effect on the stability of the laminar flow. According to some data, such eccentricity slightly (within the range of 10–15%) reduces the critical value of the Taylor number, but, according to other data, it slightly increases it. A more significant eccentricity definitely prevents Taylor vortices, thereby stabilizing the laminar lubricant flow. The turbulence of the lubricant flow manifests itself mainly as an apparent increase in lubricant viscosity and a slight reduction in the relative length of the bearing.

Theoretically, the laminar flow in the lubricating layer is stable at any flow rate if the angular velocity of bushing rotation exceeds the angular velocity of journal rotation, when they rotate in the same direction. This can be explained by the fact that when the bushing rotates, the centrifugal forces press the lubricant to the moving surface. On the contrary, the loss of stability during journal rotation is due to the fact that the centrifugal forces push the lubricant from the rotating surface. Such a simplified idea of forces certainly does not explain all the peculiarities of the stability loss by the laminar flow.

The formation of a blocking layer is determined by the fact that oil components with lower velocities cannot overcome the resistance to the motion of components with high velocities. Based on the above it follows that the boundary layers have smaller thicknesses when the Reynolds number has large values, and, accordingly, the reserve of concentrated surface-active substances that causes restoration of the damaged boundary layers may not be sufficient. Basing on the length of one average molecule, the thickness of the boundary layers will be comparable with the heights of the forming polymolecular lubricant layers; for example, for a fatty acid molecule, it will be 1.7–2.3 nm. We propose to use the outlined approach to evaluate the state of engine oil flow in the plain bearing of turbochargers as a basis for developing measures of improving its reliability under extreme operating conditions.

Thus, from the point of view of evaluating the processes that cause the structuring of boundary layers shielding the interaction surfaces, it seems significant to take into account the velocities of the lubricant components that flow around the friction surfaces during their interaction. It is quite difficult to calculate the velocities of motion of the lubricating medium components in the lines of fluid flow, but it is not difficult to evaluate the flow state with the lines of fluid flow according to the Reynolds number for a plain bearing.

At this stage, the research aims to determine the possible operating regimes and conditions under which the engine oil flow in the plain bearing (s) of the turbocharger shaft transitions to turbulence with corresponding thicknesses of the likely boundary layers. The obtained results will allow us to evaluate the overall state of the oil flow from the point of view of its layer pattern and the motion pattern of its components.

3 Materials and Methods

To calculate the Reynolds number for the plain bearing, we used the mathematical expression [12, 13]:

$$\mathrm{Re} = \frac{\pi \cdot d \cdot n \cdot \rho \cdot S}{2 \cdot \mu_{ef}} \leq 41.3 \cdot \sqrt{\frac{d}{S}}, \tag{4}$$

where d is the shaft diameter, m; n is the turbocharger shaft speed, s^{-1}; ρ is the density of the engine oil, kg/m^3; μ_{ef} is the dynamic viscosity of the engine oil, Pa s; S is the diameter clearance, m.

Calculations were made in Microsoft Excel. The following values of interference parameters were used as initial data for calculations: the shaft (journal) diameter, m: d_{max} = 0.012; d_{min} = 0.006; the diameter clearance, µm: S_{nom} = 16; S_{manuf} = 58; S_{bound} = 75; the shaft speed, s^{-1}: n_{max} = 4100 (246,000 min^{-1}); n_{min} = 20 (1200 min^{-1}); the dynamic viscosity, Pa s: μ_{max} = 0.9; μ_{min} = 0.0024.

The values of dynamic viscosity were calculated for Lukoil Super 15 W-40 mineral engine oil according to the data given in Table 1.

The diameter clearance was selected based on the fact that it increases from the rated value to the boundary value, as the bearing life diminishes. The shaft diameters were selected based on the design features of turbochargers, determined by the geometrical dimensions of the main parts and the maximum turbine shaft speed. Thus, two average diameters of shafts were selected for analytical studies: d_{max} = 0.012 m for turbochargers with maximum shaft speed from 100,000 to 125,000 min^{-1}, for example, TCR-6 (01), TCR7-00.1 (700-1118010), K-27-543-01, etc., with an operating range of 90,000–100,000 min^{-1}; d_{min} = 0.006 m for turbochargers with maximum shaft speed of 150,000 min^{-1} and higher, for example, TCR-4, TCR5, Garret G25-555, Garret 15445, etc., with an operating range of 160,000–180,000 min^{-1}.

The behavior of the Reynolds number value was modeled in Microsoft Excel according to expression (4). To visualize the line of oil transition into the turbulent state and determine the regimes and conditions for maintaining this state, response surfaces were built.

Table 1 Initial data

Parameter	Numerical values							
Temperature T (°C)	0	20	40	60	80	100	120	140
Kinematic viscosity γ, mm^2/s	1000	215	70	26	18	8	5	3
Density ρ*, kg/m^3	908	893	878	863	848	833	818	803
Dynamic viscosity μ_{eff}, Pa s	0.9	0.19	0.061	0.022	0.015	0.0066	0.0041	0.0024

Note *the density was calculated according to the expression ρ = 0.908–7.5 × 10^{-4} T, which was obtained by processing the data of laboratory research on the change in lubricant mass per volume unit under a thermal effect

To determine the thickness of a likely boundary layer, expression (3) was used. The numerical value of the radius of the surface roughness curve r was used as the characteristic linear size of the washed body. The numerical values of the parameter r are determined by the profile recording data of the working surfaces of the journal and the bushing. To make theoretical calculations, we used the values r_1 = 30 μm, for example, for the surface of a steel shaft, and r_2 = 13 μm, for example, for the surface of a bronze bushing that correspond to the cylindrical surface finish at $R_{z\max}$ = 1.2 μm and the internal grinding at $R_{z\max}$ = 4.72 μm.

4 Results

Figure 3 shows response surfaces for the Reynolds number with the range of shaft rotation speed n = 20–2760 s^{-1} (1200–165,600 min^{-1})

Tables 2 and 3 show turbocharger regimes that maintain the oil flow turbulization transition. Table 4 shows numerical values of the boundary layer.

5 Discussion

Tables 1, 2, 3, and 4 show the calculation results that indicate the following.

Oil flow does not make a transition to turbulence when the operating clearance S is 16 μm, regardless of the shaft diameters, viscosity change, and shaft speed. The thickness of the boundary layer varies from 1.8 to 9.5 μm on the surface with a larger asperity radius, and from 0.78 to 4.1 μm with a smaller asperity radius. This determines favorable conditions for the interaction of the oil components both between themselves and with the surfaces, especially at low shaft speed.

When the operating clearance S is 58 μm and the shaft diameter d is 0.006 m, the oil flow makes a transition to turbulence when its rotation speed ranges from 138,000 min^{-1} (with the oil viscosity of 0.0024 Pa s, which corresponds to a

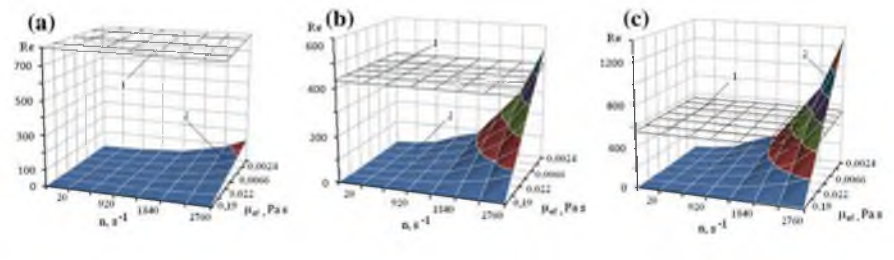

Fig. 3 Response surfaces for the Reynolds number: 1—line of transition into the turbulent state; 2—the response surface; **a** when d = 0.006 m; S = 16 μm; **b** when d = 0.006 m; S = 58 μm; **c** when d = 0.012 m; S = 75 μm

Table 2 Numerical values of the parameters when the shaft diameter $d = 0.006$ m

Clearance S (μm)	Dynamic viscosity μ (Pa s)	Shaft speed n, $\min^{-1}$	Reynolds number Re
16	0.0024–0.9	1200–246,000	799* $Re_i < 799$
	$Re_{max} = 206$		
58	0.0066–0.9	1200–246,000	420*
	0.0041–0.9	1200–165,000	$Re_i < 420$, $Re_{max} = 749$
	0.0024–0.0041**	138,000–246,000**	$Re_i > 420$
75	0.0066–0.9	1200–246,000	369*
	0.0041–0.9	1200–138,000	$Re_i < 369$, $Re_{max} = 969$
	0.0024–0.0041**	110,400–246,000**	$Re_i > 369$

Note *boundary value of the Reynolds number *Re* of flow turbulization, lower value–maximum value under flow turbulization; **values under flow turbulization

Table 3 Numerical values of the parameters when the shaft diameter $d = 0.012$ m

Clearance S, μm	Dynamic viscosity μ_{eff}, Pa s	Shaft speed n, s^{-1}	Reynolds number Re
16	0.0024–0.9	20–4100	1131* $Re_i < 1131$
	$Re_{max} = 413$		
58	0.0066–0.9	1200–246,000	594*
	0.0041–0.9	1200–138,000	$Re_i < 594$, $Re_{max} = 1498$
	0.0024–0.0041**	138,000–246,000**	$Re_i > 594$
75	0.0015–0.9	1200–246,000	522*
	0.0066–0.9	1200–138,000	$Re_i < 522$, $Re_{max} = 1930$
	0.0024–0.0041**	138,000–246,000**	$Re_i > 522$

Note *boundary value of the Reynolds number *Re* of flow turbulization, lower value–maximum value under flow turbulization; **values under flow turbulization

temperature of 140 °C) to 193,000 $\min^{-1}$ and higher (with a viscosity range between 0.0041 and 0.0024 Pa s, which corresponds to a temperature range between 130 and 140 °C). These ranges of operating parameters of the oil layer in TCR bearings are dangerous, because the contact lubricating layer sharply decreases. This causes the oil burning effect, plaque and sludge formation. Under these operating conditions, the resource reduces by 2–10 times. It may lead to the fusing of the working surfaces of the journal and the bearing, as well as complete jamming of the rotating TCR assemblies. The average thicknesses of the boundary layers are 1.4 and 0.6 μm, which is 1.7 and 1.8 times smaller than under more

Table 4 Numerical values of the boundary layer δ, μm

Clearance S, μm	Diameter d = 0.006 m		Diameter d = 0.012 m	
	Asperity radius r_1 = 30 μm	Asperity radius r_2 = 13 μm	Asperity radius r_1 = 30 μm	Asperity radius r_2 = 13 μm
16	2.5–9.5*	1.1–4.1*	1.8–9.5*	0.78–4.1*
58	1.46–9.5	0.63–4.1	1.23–9.5	0.53–4.1
	1.33–1.46**	0.6–0.63**	0.94–1.23**	0.41–0.53**
75	1.56–9.5	1.41–4.1	1.31–9.5	0.57–4.1
	1.17–1.56**	0.44–0.5**	0.83–1.31**	0.36–0.57**

Note *values were determined for the range of the Reynolds number Re from 10 to Re_{bound}; **values under flow turbulization (for the clearances of 58, 75 μm, the values in the numerator with zero flow turbulization)

favorable conditions. In this case, the theoretical range of flow turbulization is imposed on the possible operating mode of the turbocharger. This means that the discussed effects, consisting in transition of the oil flow to turbulence, its discontinuity, and cavitation phenomena, are possible. In a coupling with the same clearance, but with a shaft diameter of d = 0.012 m, the oil flow makes a transition to turbulence when the oil viscosities are the same, but the shaft speed is 110,400 min^{-1} and higher. However, turbochargers with a shaft diameter of about 12 mm and more do not operate within this speed range. Therefore, it is unnecessary to consider the mode of flow turbulization.

When the operating clearance S is 75 μm and the shaft diameter d is 0.006 m, the oil flow makes a transition to turbulence:

- when rotation speed is 110,400 min^{-1} and higher, oil viscosity is 0.0024–0.0041 Pa·s;
- when the rotation speed is 246,000 min^{-1} and the oil viscosity is 0.0066 Pa s, the temperature is 100 °C.

The thicknesses of the boundary layers get reduced by an average of 6–13%. In this case, the theoretical range of flow turbulization is not only imposed on the operating mode of the turbocharger, but also expands toward a decrease in the shaft rotation speed. Reduction in boundary lubricant layers and discontinuity of flow with the operating clearance S = 75 μm lead to an increase in dangerous ranges of TCR shaft speed. Approximate calculations showed that this can lead to a decrease in resource parameters by 1.1–1.6 times in comparison with the TCR operating with clearances of 16, 58 μm.

In a coupling with the same clearance, but with a shaft diameter of d = 0.012 m, the oil flow makes a transition to turbulence:

- when the rotation speed is 82,800 min^{-1} and higher, and the oil viscosity is 0.0024 Pa s;
- when the rotation speed is 110,400 min^{-1} and higher, and the oil viscosity is 0.0024–0.0041 Pa s;

The thicknesses of the boundary layers also get reduced by an average of 6–13%. This interaction geometry is characterized by the range of oil flow transition to turbulence that is imposed on the operating mode of the turbocharger.

However, in operational modes, the turbocharger bearings have radial clearances that are caused by the efficiency of the oil flows, as well as the direction and magnitude of the radial loading force. These clearances are not equal and even in cross-section.

To evaluate the correlations between the thicknesses of the boundary layers on the shaft and journal (bearing) surfaces and the thickness of the lubricating layer between them in operational modes of the turbocharger, the calculations were made according to expression (1), see Table 5. In this case, we used the radial force value N = 20H that was calculated in [7]. This force loads the friction surface of the plain bearing of the GT1544 turbocharger.

The analysis of changes in thickness fields of the lubricating layer in the coupling and the boundary layers on the friction surfaces of the shaft and bushing (Tables 3, 4, and 5) shows the following.

When operational clearances in the coupling are 58 and 75 μm and the shaft diameter d = 0.006 m, there is no possibility of forming stable boundary layers on the surface of the bushing and the shaft in the zone of maximum contact. This is determined by the fact that the average range of thickness of the lubricating layer under possible transition of oil flow to turbulence is 4–5 times smaller than the theoretical boundary layer on the shaft surface and 1.5–2.5 times on the journal surface. Such a correlation seems to be with a "preload." The conditions for clearly defined thicknesses of the boundary layers, that is, with a "clearance," are more favorable when the oil temperature is less than 100 °C. The area of preload work poses a threat by reducing the resource parameters of TCR assemblies by 1.1–2 times.

When operational clearances in the coupling are 58 and 75 μm and the shaft diameter d = 0.012 m, there is a possibility of forming stable boundary layers on the surface of the bushing and the shaft in the zone of maximum contact. This is determined by the fact that the average range of the lubricating layer thickness in the zone of maximum contact is approximately equal to the sum of the possible thicknesses of the boundary layers on the surface of the shaft and the journal. This

Table 5 Effective thickness of the lubricating layer h_{min}

Parameter	Conditions: d = 0.006; l = 0.015 m; N = 20 N							
	n = 110400 min^{-1}, υ_s = 34.66 m/s				n = 165,600 min^{-1}, υ_s = 51.81 m/s			
M_{eff}, Pa s	0.0024	0.0041	0.015	0.022	0.0024	0.0041	0.015	0.022
T, °C	140	120	80	60	140	120	80	60
h_{min}, μm	0.21	0.36	1.33	1.95	0.3	0.54	1.99	2.9
	Conditions: d = 0.012; l = 0.025 m; N = 20 N							
	n = 82,800 min^{-1}, υ_s = 51.99 m/s				n = 110,400 min^{-1}, υ_s = 51.99 m/s			
h_{min}, μm	1.06	1.8	6.66	9.7	1.4	2.43	8.89	13.8

correlation seems "transitional" with a defined "clearance" under the oil temperature lower than 110 °C. Thus, the work zone above 110 °C can be called transitional, and transition to turbulence is more likely.

In general, the obtained results indicate that the hydraulic tension of the layers formed of engine oil components on the surfaces of the shaft and the bushing may vary depending on the degree of its turbulence and the geometric parameters of the contact. The oil is under threat with a danger of developing additives, overheating of oil layers, or accelerated wear in relation to the bushing and the shaft. And this, in turn, determines the pattern of forming damage-resistant bonds between elementary polyatomic molecules of oil and active centers, that is, ions of metals of the bushing and shaft materials. At the same time, the time component of the contact interaction is decisive for the motion pattern of the components of engine oil when changing modes of force and speed loading of its layers. It must be taken into account when forecasting surface effects when using, for example, repairing and restoring products to improve the technical condition of the elements to be restored.

Further studies can be directed to a comparison of the calculation results with experimental data. For this purpose, experimental stands for research of turbochargers developed by Gritsenko [23–26] and others can be used.

6 Conclusions

The considered approaches allowed us to calculate the estimate of possible operational modes and conditions under which the engine oil flow in the plain bearing(s) of the turbocharger shaft makes a transition to turbulence, which allowed us to determine the thicknesses of probable boundary layers on the surfaces of the shaft and the bushing, and compare them with the thickness of a possible lubricating layer under the loading conditions. It has been established that the boundary layers can have different thickness and correlate with the "preload," the "clearance," and be "transitional" depending on the considered parameters. The considered technique can be applied to evaluate the state of engine oil flow in the predicted diameter clearance of any turbocharger. The obtained results may call for ways of ensuring reliability of bearing(s) lubrication in a particular turbocharger. Moreover, the obtained data can be used to simulate the effects of accelerated wear, reduced reliability of frictional couples, and undesired friction modes.

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