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ANALYSIS AND VALIDATION OF THE DYNAMIC METHOD FOR DIAGNOSING DIESEL ENGINE CONNECTING ROD BEARINGS

Summary. The paper considers the problems related with diagnosing diesel engine crank gears, and presents most widely used methods for their diagnosis. The authors provide arguments in favour of dynamic health status identification of connecting-rod bearings in terms of the oil layer thickness. For the first time the dependence is obtained of the oil layer thickness in conrod bearings from the rotation frequency of the crankshaft and the pressure in the lubrication system.

1. INTRODUCTION

At present the internal combustion engine services market offers a wide range of diagnostics equipment and techniques [1]. In practice, effective instruments designed by the research and development organizations did not come into common use; as a result, the production is limited to a small number of these instruments. The fact causes challenges in the selection of a single set of instruments and equipment needed for an integrated assessment of the health status of internal combustion engines (ICE) using standard structural parameters. Currently the linear displacement method is used for resource diagnostics of the crank gear based on the measurements of the axial movement of the piston in the idling engine in the area of the top dead point (TDP) being under injected pressure and depression provided by the compressor and vacuum installation. However, this method does not meet the requirements because of significant errors and labour intensity needed for the measurement procedure. So far, the engines have been repaired taking into account the running distance between repairs. In many cases, vehicles go through a premature repair work, whereas the performance of a vehicle can be restored by simple adjustment procedures or relatively simple and cheap recovery works [2].

The organoleptic method is applied for listening to the sounds of the engine's internal parts using acoustic or electronic stethoscopes. This method has a number of disadvantages: 1) it can be applied to diagnose the engine malfunctions only by the wear-out elements, and 2) subjectivity in assessment.

The calibration method is based on identifying the character of oil leaking in the oil distributing passage. Reduction of oil pressure in the lubrication system is an unmistakable sign of wear in the couplings [3]. However, reducing oil pressure will most probably indicate trouble relating the oil pump, engine oil and operation modes. Thus, the pressure in the lubrication system can be used in diagnosis as a parameter for achieving the limiting coefficient.

The essence of the indicator method used for engine diagnostics consists in the spectral analysis of the contents of wear particles in the engine oil by means of a costly installation (such as MFS-5), which is more efficient and can be applied for diagnosing marine and locomotive diesel engines. Among the disadvantages of the given method are emission of the particles including oil waste through the exhaust system; deposition of these particles both in the oil purifier and other parts of the lubrication system, and the sampling method [4].

Diagnostics by the phase methods determines the condition of the coupling elements by the dependencies which characterize the interrelation of the clearances and the angular displacement of the crankshaft. The application domain of this method is limited because of the high costs of the diagnosing equipment and limited degree of its accuracy potential.

The vibro-acoustic method utilizes the same tools as the method of linear displacement, i.e. compressor / vacuum installation. Bumps in the crank gear couplings occur at peak pressures after depression in the space above the piston. Sensors are used to measure the vibration level, whereas the amplitude of the vibration impulse helps in identifying the technical condition of the couplings. The disadvantage of this method is related with the challenges in deriving signals resulting from collisions.

The methods of diagnosing connecting-rod bearings by monitoring the amount of clearance in the couplings along the axial movement of the piston in the TDP area under excess pressure and depression does not ensure squeezing of the lubricant present between the couplings. Meanwhile, these indicators depend on the tightness in the cylinder-piston group and gas distribution mechanism. We assume that the methods for diagnosing connecting-rod bearings by changing the piston position in the TDP at the operation and start-up modes (the dynamic method) are most promising. When using this method, we propose to use the difference in the inertia forces of the connecting rod and the piston set with the total motion resistance force as the piston driving force acting on the amount of clearance in connecting-rod bearings in the TDP area. Thus, there is no need in the bulky compressor / vacuum installation. By intensifying the alternate loads by changing the crankshaft rotation frequency allows for selecting better coupling clearances [5].

2. METHODOLOGY AND RESEARCH RESULTS

The dynamic method has been studied insufficiently. To prove its effectiveness, researchers at Yuri Gagarin State Technical University of Saratov developed a device and method for diagnosing connecting-rod bearings. The investigations refer the diagnosis modes and obtaining the data on the boundary conditions of the total clearance in the crank gear, oil film thickness between connecting-rod bearings at the moment of piston transposition at TDP, and its change patterns based on the mileage. Metrological characteristics of the device allowed us to conduct the measurements with the accuracy sufficient for the feedback of the method. Testing the device on the ICE confirmed the assumption about the efficiency of the dynamic method for the crank gear diagnosis as being most promising.

To study the speed rates and the optimum crankshaft rotation frequency in diagnosing internal combustion engines, it is necessary [6]:

- to determine the change patterns of oil film thickness in connecting-rod bearings at various speed rates and thermal modes of ICE operation;
- to identify the conditions for piston motion modes referred to connecting rods with the clearance level in the TDP area;
- to estimate the temperature impact on thermal expansion in the engine parts.

The calculations were made for the cylinders at engine-on modes, where injectors were replaced for diagnostic devices applied for sealing or depressurizing combustion chambers, in case of necessity.

2.1. Calculation of the oil layer thickness in connecting-rod bearings of high-power engines (as illustrated by KAMAZ diesel engines)

To derive the formula for the changes in the minimum thickness of oil film to connecting-rod bearings at the removed injector, we will use the kinetostatics method. As it follows from the d'Alembert principle

$$P_j^I + P_j^{II} + F_{ml} + F_{hydr} = 0, \tag{1}$$

where $P_j^I = -mw^2 r\cos\varphi$ is the first-order force of inertia, kN; $P_j^{II} = -mw^2 r\lambda\cos^2\varphi$ is the second-order force of inertia, kN; $F_{hydr} = \frac{\mu w}{\phi^2} ld\Phi_p$ is the force of the hydraulic lubrication layer, kN; F_{ml} is the force of mechanical friction losses, kN.

For analytical description of mechanical power losses at friction between the piston rings and the cylinder liner, we will use the analytical expression for mechanical losses, kN/m^2. For diesel engines [7]

$$P_m = 0,8 + 0,17 v_n\ , \tag{2}$$

where v_n is the piston speed, m/sec.

In [8], the intensity of mechanical losses for friction between the piston rings and the liner is given by (kN)

$$F_{ml} = 10\pi DH\left(0,8 + 0,17wr\left(\sin\phi + \frac{\lambda}{2}\sin^2\phi\right)\right) \quad . \tag{3}$$

Then

$$mw^2 r(\cos\varphi + \lambda\cos^2\varphi) - 10\pi DH\left(0,8 + 0,17wr(\sin\varphi + \frac{\lambda}{2}\sin^2\varphi)\right) = \frac{\mu w}{\psi^2} ld\Phi_p \quad . \tag{4}$$

Hence the load rate, which is a dimensionless function of the crankpin position in the bearing and the boundary of the lubrication layer

$$\Phi_p = \frac{\psi^2\left[mw^2 r(\cos\varphi + \lambda\cos^2\varphi - 10\pi DH(0,8 + 0,17wr(\sin\varphi + \frac{\lambda}{2}\sin^2\varphi)))\right]}{\mu wld} \tag{5}$$

There is empirical interconnection between the load factor Φ_p and the relative eccentricity λ. Using a set of application programs, we found the most valid interconnection between the two parameters (with the coefficient 0.86), which represents the exponential function

$$\lambda = A\Phi_p^{в} = 0,6\Phi_p^{0,18} \quad . \tag{6}$$

On the other hand, we know the analytical expression for the connection between the relative eccentricity and the minimum oil layer thickness in connecting-rod bearings [9]

$$h_{\min} = \delta(1-\lambda). \tag{7}$$

Then, the minimum oil film thickness in connecting rod bearings is given by

$$h_{\min} = \delta\left[1 - 0,6\left(\psi^2\frac{mw^2 r(\cos\varphi + \lambda\cos^2\varphi) - 10\pi DH(0,8 + 0,17wr(\sin\varphi + \frac{\lambda}{2}\sin^2\varphi)}{\mu wld}\right)^{0,18}\right], \tag{8}$$

where δ is the clearance in connecting-rod bearings, m; ψ is the coefficient characterizing the pin-liner fitting; m is the piston kit weight, kg; w is the crankshaft rotating frequency, sec^{-1}; r is the crank radius, m; φ is the crankshaft rotation angle; μ is dynamic oil viscosity, mPa·sec; l is the length of the connecting rod liner, m; d is the connecting rod liner diameter, m; D is the sleeve diameter, m; H is the height of the friction sleeve surface, m.

At the top dead point ($\varphi = 0$)

$$h_{\min} = \delta\left[1 - 0{,}6\left(\frac{\psi^2(mw^2r(1+\lambda) - 8\pi DH)}{\mu wld}\right)^{0{,}18}\right]. \quad (9)$$

Considering [10] $\delta = \delta_0 \cdot e^{-bl}$, where l is the running, km; h_0, b are experimental parameters, e is the exponent

$$h = h_0 \cdot e^{-bl}. \quad (10)$$

The conditions corresponding to the piston motion relative to the connecting rod with sample clearances in the TDP area (based on the general equation for the piston dynamics), are based on Newton's second law. Considering the origin of coordinates on the crankshaft axis, "+" the rotation directed clockwise, and the motion "-" from the bottom of the dead point to the top, we get

$$m\ddot{S} = F; \quad (11)$$

$$\ddot{S} = -rw^2(\cos\varphi + \lambda\cos^2\varphi) + \Delta\ddot{S}; \quad (12)$$

$$F = R - F_n, \quad (13)$$

where $m, \ddot{S}$ is the weight and acceleration of the piston kg, m/sec^2; F is the resultant force acting on the piston, kN; $\Delta\ddot{S}$ is acceleration of the piston relative to the upper head of the connecting rod, m/sec^2; R is the force acting on the piston from the upper head of the connecting rod, kN; F_n is the resultant of resistance forces to the piston motion which is equal to the friction force in the coupling between the piston rings and cylinder liner, and the piston pin with the upper head in the connecting rod, kN.

Analyzing the equations (11)-(13), and taking into account the crank gear kinematics, it can be noted that the direct access to clearances in the transition of the piston towards the TDP occurs under $\Delta\ddot{S} = 0$, i.e. when the piston motion is predetermined only by the connecting rod motion. The given condition is fulfilled within the frequency range $0 \le w \le w_{0\max}$, where $w_{0\max}$ is the maximum frequency corresponding to transition of the piston at the TDP, from restricted to free motion.

We will define the frequency $w_{0\max}$ from the equations (11) - (13) and at the same time fulfil the requirements for the restricted ($\Delta\ddot{S} = 0$) and free ($R = 0$) motions of the piston towards the TDP. Here, at TDP the transfer acceleration module reaches the maximum $\left|\ddot{S}\right| = rw^2(1+\lambda)$, whereas the desired frequency, sec^{-1}

$$w_{0\max} = \sqrt{\frac{F_n}{mr(1+\lambda)}} \quad . \quad (14)$$

Using the formula (14), we take into account

$$F_n = 10\pi DH\left(0{,}8 + 0{,}17wr\left(\sin\varphi + \frac{\lambda}{2}\sin^2\varphi\right)\right) \quad . \quad (15)$$

When $\varphi = 0$; $F_n = 8\pi DH$, the resultant expression, sec^{-1}

$$w_{0\max} = \sqrt{\frac{8\pi DH}{mr(1+\lambda)}}, \quad (16)$$

where r is the crank radius, m.

Thus for KAMAZ engines, the crankshaft clearance must be measured with the crankshaft rotation frequency at 800 ... 850 min^{-1}. This is confirmed by the test results for warm engines. At in-creasing the crankshaft speed above the specified value, the clearances in the crankshaft gear are not increased.

We can now estimate the influence of the ICE temperature mode on thickness and lengthening of connecting rod liners under warm engine. Heat generation in the bearing

$$Q = fF_r w\frac{d}{2} = \psi\frac{\Phi_f}{\Phi_p}F_r w\frac{d}{2}, \tag{17}$$

where f is the coefficient of friction in the bearing; Φ_f is friction characteristics, involving the dimensionless function for of the crankpin position in the bearing, the base oil layer and the ratio l/d; Φ_p is load coefficient, which is the dimensionless function of the crankpin position in the bearing and the boundary of the base oil layer, depending on the ratio l/d; F_r is the base force of the oil layer, kN; d is the inside diameter of the liner, m; ψ is the coefficient for the sleeve bearing fitting.

The heat carried by the engine oil from the friction unit, J

$$Q_1 = cq(t_2 - t_1) \tag{18}$$

where c is the volumetric heat capacity of oil, J/kg·^{0}C; q is oil consumption, J/kg ; t_1, t_2 is oil temperature at the inlet and outlet of the bearing, ^{0}C.

The heat transferred by the body of the bearing into external environment, J

$$Q_2 = k \cdot A \cdot (t_м - t_{cp}) \tag{19}$$

where k is the heat transfer coefficient; A is the bearing surface washed by oil, J/kg; t_{cp} is the temperature of the surrounding oil, ^{0}C; t_m is the average temperature of oil in the load area, ^{0}C.

According to the heat balance equation under steady bearing operation mode $Q = Q_1 + Q_2$. Hence $Q_1 = Q - Q_2$(J).

Using the formula (18), oil consumption is

$$q = \frac{Q_1}{c(t_2 - t_1)}. \tag{20}$$

Extension along the liner circle Δl and width Δb

$$\Delta l = l\alpha\Delta t, \quad \Delta b = b\alpha\Delta t. \tag{21}$$

where l is the length of the liner; α is the thermal expansion coefficient; Δt is the temperature difference; b is the width of the liner.

According to the work, Δl=0.94 μm, Δb=0.35 μm, whereas thickness is at 0.0148 μm.

Thus, the thickness of the liner under the engine operating at reasonable crankshaft speed does not affect the diagnosis data, as is seen from the experiments conducted for the cold and warm ICE.

2.2. The impact of pressure in the lubrication system and speed of the crankshaft in KAMAZ diesel engines on oil film thickness

To estimate the impact of pressure in the lubrication system by the minimal TOL and changing the crankshaft speed in ICE, we will use the hydrodynamic lubrication theory.

As the size of the clearances (capillary cracks) in connecting-rod bearings does not exceed 50 μm, the oil flow has laminar characteristics. Figure 1 shows the diagram of the oil flow in the connecting-rod bearing, where the piston is at TDP. Since the inner surface of the connecting-rod bearing and the crankpin is big enough compared to the oil film thickness, velocity distribution in the clearance is parabolic, and corresponds to the laminar flow.

Oil pressure in the lubrication system of the engine under the same amount of oil provided to connecting-rod bearings by the oil pump

$$P = \frac{Q^2 p}{2g\mu^2 F} \quad , \tag{22}$$

where F is the cross sectional area (m^2) of the clearance between the crankpin and bearing liners; ρ is the oil density, kg/m^3; μ is the coefficient of oil consumption; g is gravity acceleration, N/kg; Q is oil consumption, m^3.

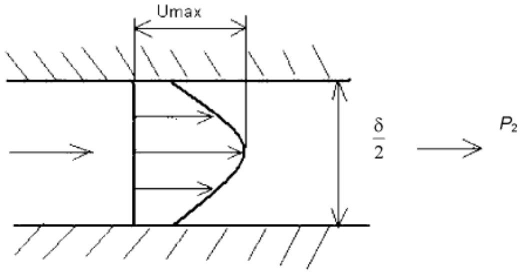

Fig. 1. The diagram of the engine oil flow under pressure between the crankpin and liner: U_{max} is the maximum flow rate, m/sec; δ/2 is the radial clearance, m; P_1 is the inlet pressure to the bearing, kN/m^2; P_2 is the pressure at the outlet of the bearing, kN/m^2

The cross-section area of the crack depends on the average size of the clearance δ, the radius r of the crankpin $F = 2\pi \frac{\delta \cdot r}{2} = \pi \cdot r \cdot \delta$, which means $F = k \cdot \delta$, where $k = \pi \cdot r$.

On the other hand, the oil film thickness is associated with viscosity of the engine oil by means of the ratio, μm

$$h = \frac{d^2 \cdot \eta \cdot n}{18{,}36 \cdot p^1 \cdot \delta \cdot c} \quad , \tag{23}$$

where d is the crankpin diameter, μm; δ is diametrical clearance, μm; η is oil viscosity, sSt; n is the crankshaft speed, sec^{-1}; p^1 is the pressure acting on the projected area of the bearing, kN.

On deriving the diametrical clearance from the formulas (22) and (23), on equating the right-hand sides, we get

$$\frac{d^2 \cdot \eta \cdot n}{18{,}36 \cdot h \cdot p^1 \cdot c} = \frac{Q}{\mu \cdot \pi \cdot r} \sqrt{\frac{\rho}{2P \cdot g}} \quad . \tag{24}$$

Fig. 2 schematically shows the coupling "crankpin - liner" with the oversized clearance. Since any radial bearing has a natural lubricating wedge due to specifics of the clearance, then at high speeds, we find the lubricant being under increased pressure supporting the shaft and separating it completely from the bearing liner. Depending on parameters and variables, the bearing friction accounts for the Sommerfeld number in the formula [10]

$$S_0 = \frac{p^1 \cdot \psi_r^2}{\eta \cdot \omega} \quad , \tag{25}$$

where $p^1 = \frac{F_n}{2r \cdot L}$ is the load, acting on the projected area of the bearing, kN/m^2; $\psi_r = \frac{C_r}{r}$ is the relation of radial clearance to the crankshaft radius; η is oil viscosity; $\omega = \frac{v}{r}$ is angular velocity, sec^{-1}.

According to the Vogelpohl theory [11], the friction coefficient can be approximately expressed as

$$f = \frac{3\psi_r}{c} \text{ under } S_0 < 1, \; f = \frac{3\psi_r^2}{c^{\frac{1}{2}}} \; under \; S_0 > 1 \quad . \tag{26}$$

Hence, it follows that $f \approx \dfrac{\eta \cdot v \cdot L}{F_N}$ under $S_0 < 1$, $f \approx \left(\dfrac{\eta \cdot v \cdot L}{F_n}\right)^{\frac{1}{2}}$ under $S_0 > 1$.

In Fig. 3, the parameter $\dfrac{\eta \cdot v}{F_N}$ is used for the abscissa of the Stribeck curve, which shows the characteristics of the lubrication system.

Under mode 1 of the Shtribek curve and surface mode of the crankpin and liner are separated by continuous lubrication, with the thickness exceeding the roughness of the surfaces. The resistance to motion is caused by the internal lubricated friction. At this mode, the tribological behaviour is determined by the fluid mechanics methods. Since the given system consists of tight contacts, the dependence of viscosity on pressure may be ignored. This mode corresponds to surface fatigue wear, cavitation wear or fluid erosion. Under mode 2, there is a possibility for the wear mechanisms of four groups: surface fatigue, abrasive, adhesive, and corrosion and mechanical wear. However, these processes are influenced by the lubricant film and can be modified under its impact.

If the operation conditions of the system at mode 2 are shifted to the left along the Shtribek curve, the number of interacting irregularities within the contact area increases, and the film thickness decreases to several monomolecular layers. At mode 3 of boundary lubrication, the 3D flow properties of the lubricant are invalidated, whereas the load is considered as deformation of irregularities which result in scoring and loose running of the bearing liners.

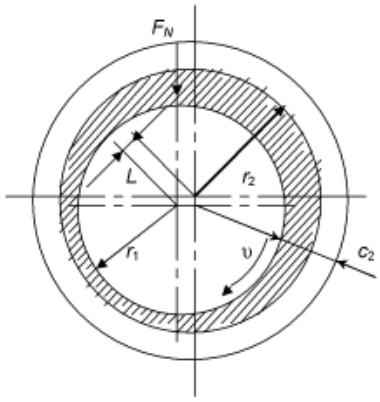

Fig. 2. The scheme for the geometry of the connecting rod bearing: F_N - the load; v - the speed; L - eccentricity; r_1 - the radius of the shaft; r_2 - the radius of the bushing; $c_2 = r_2 - r_1$ - the radial clearance

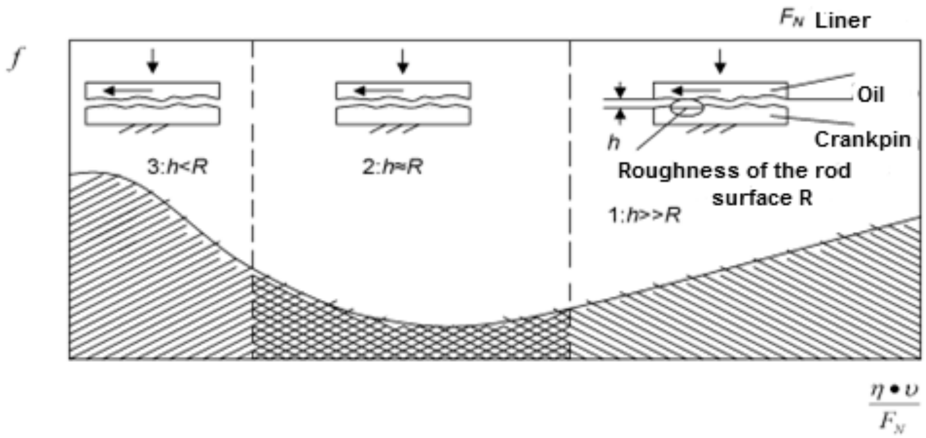

Fig. 3. The Stribeck curve and lubrication modes of the connecting-rod bearings: 1 is hydrodynamic lubrication; 2 is mixed lubrication; 3 is boundary lubrication

When calculating the oil film thickness by means of the proposed empirical dependencies (16), we can confirm that upgrading the efficiency of the oil pump performance is beneficial for the lubrication ratio of the crankshaft couplings. On the other hand, application of turbochargers for the high-power KAMAZ diesel engines allowed for increasing the capacity values and loads on the parts of the cylinder-piston group and the crank mechanism.

This can be confirmed if we analyze the expression (24) relating the influence of the crankshaft speed on the oil film thickness, the pressure in the lubrication system, and oil pressure in the bearing couplings by replacing the expression into the coefficient K (as in the case of KAMAZ diesels)

$$K = \frac{d^2 \cdot \eta \cdot \mu \cdot \pi}{18{,}36 \cdot c \cdot Q} \cdot \sqrt{\frac{2g}{\rho}} \quad . \tag{27}$$

Then oil film thickness, μm

$$h = K \cdot \frac{n \cdot \sqrt{P}}{p^1} \quad . \tag{28}$$

At the constant heat, speed and pressure, acting on the projected area of the bearing, the relation $\frac{n}{p^1}$ can be expressed as K_1. Then the oil film thickness is, μm

$$h = K \cdot K_1 \cdot \sqrt{P} = 47{,}4\sqrt{P} \quad . \tag{29}$$

The dependence of the oil film thickness on the pressure in the lubrication system can be characterized by the dependence shown in Fig. 4.

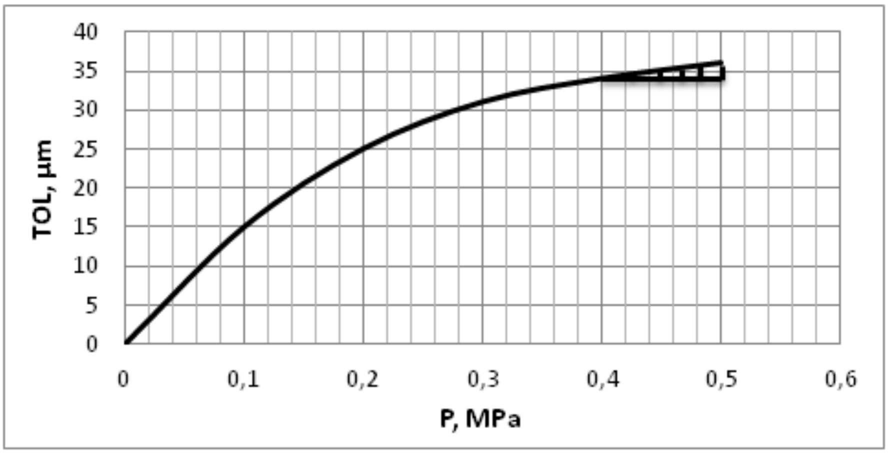

Fig. 4. The dependence of the oil film thickness on the pressure in the lubricating system

Based on the expression (24), and taking into account (28) under the diagnostic crankshaft speed

$$K_2 = n \cdot \sqrt{P} \quad . \tag{30}$$

Then the oil film thickness, μm

$$h = K \cdot K_2 \cdot \frac{1}{p^1} = 1800 \cdot \frac{1}{p^1} \quad . \tag{31}$$

The characteristic features relating the increase of pressure over connecting-rod bearings in high-power KAMAZ diesel engines which result in decrease of the oil film thickness, are provided in Fig. 5 taking into account (29). The dependence of the oil film thickness on the crankshaft speed, based on the expression (31) and considering $K_3 = \frac{\sqrt{P}}{p^1}$, is given by, μm

$$h = K \cdot K_3 \cdot n = 0{,}025h \quad . \tag{32}$$

The dependence of the oil film thickness on the crankshaft speed and pressure in the lubrication system is shown in Fig. 7.

3. CONCLUSIONS

Based on the analytical dependencies (29), (30), (32), and the obtained resultant surface of interference parameters (Fig.7), as in the case of high-power KAMAZ diesel engines, we can conclude as follows:

1. Pressure increase in the lubrication system of KAMAZ diesel engine from 0.4 to 0.5 MPa will allow for increasing the oil film thickness by 4 µm.
2. Reduction in the crankshaft speed from 2600 to 2200 rpm will reduce the oil film thickness by 10 µm.
3. Pressure increase over the crankshaft bearing coupling at improving the engine output from 210 to 260 HP, will reduce the oil film thickness by 3 µm.

Thus, the research suggests that there is tendency for decreasing the oil film thickness in high-power KAMAZ internal combustion engines, and therefore, for decreasing the resource of the considered couplings and the whole engine. Moreover, at sudden changes in the amplitude of rotating speed of the crankshaft, we assume the possibility for the transition of the couplings in connecting-rod bearings with the crankpin into the boundary lubrication mode. Changes in the design and technology of combustion engines of the new KAMAZ family ensured improvement of their technical characteristics, whereas their operational reliability is still to be upgraded.

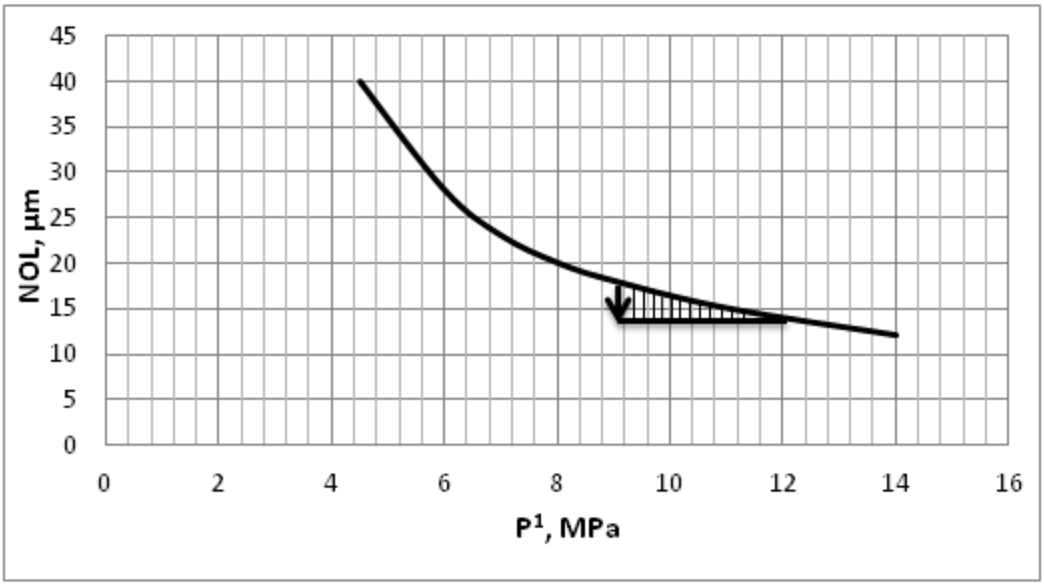

Fig. 5. The dependence of oil film thickness on pressure acting on connecting-rod bearings

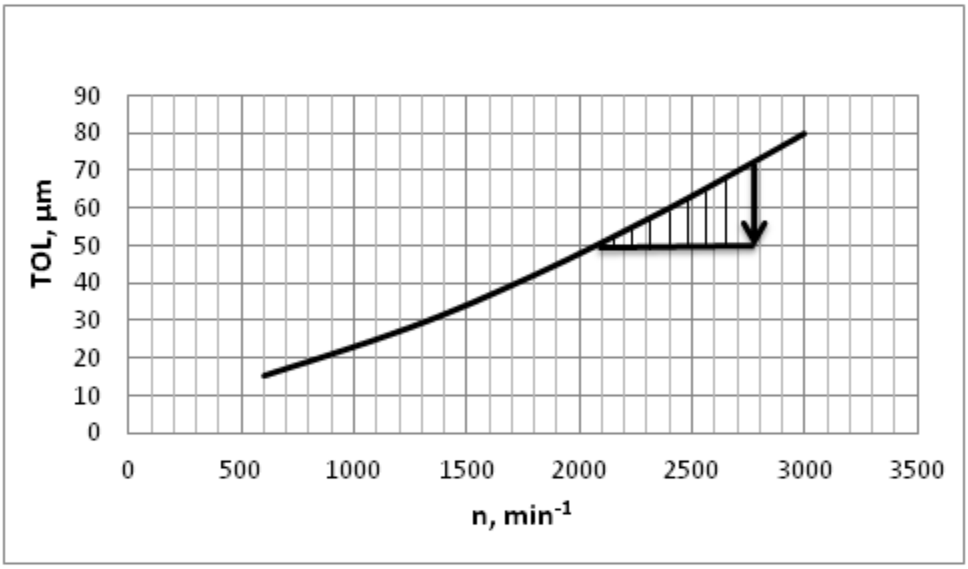

Fig. 6. The dependence of oil film thickness on the crankshaft rotation speed

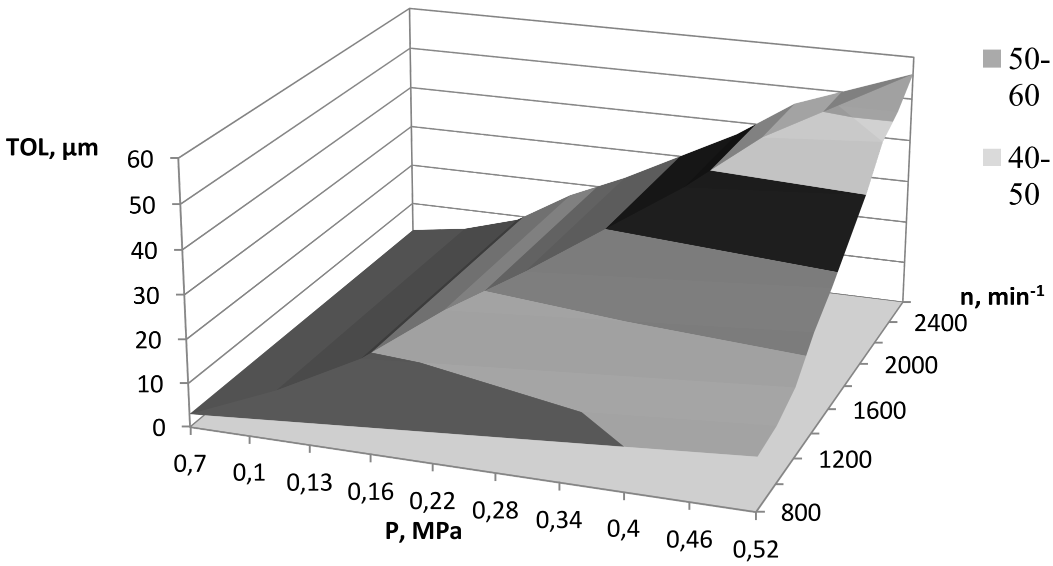

Fig. 7. The dependence of oil film thickness in connecting-rod bearings on the crankshaft rotation speed and pressure in the lubricating system

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