Vol. 40, No. 2 (2018) 300-310, DOI: 10.24874/ti.2018.40.02.13

Tribology in Industry

www.tribology.rs

RESEARCH

Diagnostics of Friction Bearings by Oil Pressure Parameters During Cycle-By-Cycle Loading

A.V. Gritsenko[a], E.A. Zadorozhnaya[a], V.D. Shepelev[a]

[a] *Department of Automobile Transport, Automobile and Tractor Faculty, South Ural State University, 76 Lenin Avenue, Chelyabinsk 454080, Russia.*

Keywords:

Engine
Friction bearings
Diagnostics
Crankshaft main and connecting rod bearings
Pressure parameters
Diagnostic parameters
Technical state

Corresponding author:

Vladimir Shepelev
Department of Automobile Transport, Automobile and Tractor Faculty, South Ural State University, 76 Lenin Avenue, Chelyabinsk 454080,Russia
E-mail: shepelevvd@susu.ru

A B S T R A C T

Failures of friction bearings of the crank mechanism comprise from 5 to 25 % of engine failures. The analysis of the main reasons for failures shows that the dominant reasons are the following: excess of loading conditions; severe operating conditions; non-observance of the periodic maintenance of the lubrication system; violation of the procedure and conditions of maintenance (contamination, oil residues, etc.); use of poor-quality oils and filters, etc. It is possible to prevent the growth of failures of friction bearings by a continuous monitoring of their complex technical state. For that purpose, we have supposed that the technical state of the crankshaft main bearings of the crank mechanism can be determined by measuring the pressures in the central oil line and calculating their difference in the cycles with the maximum load and without it at different engine crankshaft rotation frequency. As a result of the experimental work, we developed a method for in-place diagnostics of the state of friction bearings of the internal combustion engine, as well as an instrument that provides loading conditions for the bearings of the crank mechanism. We obtained an experimental dependence for determining the wear degree of the crankshaft main journal by the difference in the minimum pressure amplitudes of two adjacent cycles during the operation of the diagnosed cylinder in various modes.

1. INTRODUCTION

The performed analysis of statistical data on the number of failures in the bearings of the crank mechanism (CM) shows that they account for 5 to 25 % of engine failures [1–4]. The main reasons for these failures are:

1. Exceeding the loading conditions [5–7];
2. Severe operating conditions [8–10];
3. Macro- and micro-deviations from the correct geometric shape of the tribo-unit elements [11-13];
4. Non-observance of the periodic maintenance of the lubrication system [2,3,14];
5. Violation of the procedure and conditions of maintenance (contamination, oil residues, etc.) [15-17];
6. Use of poor-quality oils and filters, etc. [18,19].

One cannot do without observing all the listed measures. A continuous monitoring of the complex technical state of the CM bearings and the lubrication system may solve the problem [1,3,5]. The considered analysis of means, diagnostic methods and continuous monitoring shows their low functional capability and reliability [11,20,21]. The impossibility of built-in control (a diagnostic tool in the design of an automotive vehicle, which is a standard instrumental tool for diagnosing engine systems), when using most of these methods and means, makes it impossible to apply them [19-21]. A promising area in the field of engine diagnostics is the use of test methods for diagnosing nodes and parts in a dynamic mode [19,22]. Therefore, the purpose of the article is to develop a means and a method for continuous in-service monitoring of the technical state.

The analysis of the research results has shown that among all methods of diagnosing CM bearings and lubrication systems of the ICE, the most promising one is the diagnostics by the parameters of pressure pulsation in the central oil line. This method also allows us to assess the influence of technical state of individual components on the correct functioning of engines [19,20,22]. These pulsations are created by the elements of the lubrication system and the CM bearings, the pulsations reflect the change in the operational state of the units and elements, which makes it possible to consider and study the complex interrelationship of dynamically loaded elements in the operating mechanism-engine.

A large number of structural links in the CM influence the change in the shape of the pressure oscillogram in the central oil pipeline [10,15,23]. In this regard, the connection between structural and diagnostic parameters is quite complex and uncertain. To evaluate the uncertainty, private hypotheses were put forward: "It is possible to determine the technical state of the crankshaft main bearings of the CM by measuring pressures in the central oil pipeline and calculating their difference in the cycles with the maximum load and without it at different crankshaft engine rotation frequency" [19,22].

2. THEORETICAL RESEARCH

The amount of lubricant flowing into the ends of the bearing was determined from the dimensionless lubrication rate coefficients through the loaded q and unloaded q' bearing areas [15]:

$$Q_k = \frac{l_k \cdot d_k \cdot \delta_k \cdot \omega}{2}(q + q'), \tag{1}$$

where: l_k is bearing length, m; d_k is diameter of the bearing, m; δ_k is diametral clearance, m; ω is rotation frequency, rad./s.

Let us study the change in the pressure and oil flow through the crankshaft main bearings of the ICE. The coefficient of lubricant consumption through the unloaded bearing zone q' was determined by the formula [15]:

$$q' = \frac{P_k \cdot \delta_k^2}{12 \cdot \mu \cdot l_k^2 \cdot \omega}\left(\pi\left(1 + \frac{3}{2}\chi_k^2\right)\right) \tag{2}$$

where: χ_k is relative eccentricity; P_k is oil pressure at the inlet to the bearing, MPa; μ is dynamic viscosity, Pa·s.

Let's compose the ratio of oil consumption coefficients through the crankshaft main bearing under the maximum load and without load (assuming that the flow rates are ±20 %):

$$\begin{aligned} &\frac{P_1 \cdot \delta_1^2}{12 \cdot \mu \cdot l_1^2 \cdot \omega}\left(\pi\left(1 + \frac{3}{2}\chi_1^2\right)\right) = \\ &= \frac{P_2 \cdot \delta_2^2}{12 \cdot \mu \cdot l_2^2 \cdot \omega}\left(\pi\left(1 + \frac{3}{2}\chi_2^2\right)\right) \end{aligned} \tag{3}$$

Whereas in the initial expression $\delta_1 = \delta_2$ and other components d_b, l, b, μ are equal among themselves, considering the transformation, expression (3) will look as follows:

$$\frac{P_1}{P_2} = \frac{\left(1 + 1.5 \cdot \chi_2^2\right)}{\left(1 + 1.5 \cdot \chi_1^2\right)} \tag{4}$$

Thus, the larger the difference in the relative eccentricities at the idling crankshaft rotation frequency is, the larger is the ratio of the pressure values without load and under load. Thus, we found a sensitive diagnostic parameter for diagnosing the crankshaft main bearing – the ratio of the pressures during the operation of the bearing through a cycle with and without load.

The rotation frequency has a significant effect on the position of the shaft in the clearance space.

However, the total load on each journal of the shaft leads to its significant displacements relative to the clearance and an increase in its eccentricity.

We will analyze the dependence of the relative eccentricity on the rotation frequency and load conditions of the engine. For this purpose the authors' works [3,15,24] were used. The dependence of the relative eccentricity in the main bearing on the rotation frequency and load conditions is shown in Figs. 1a and 1b:

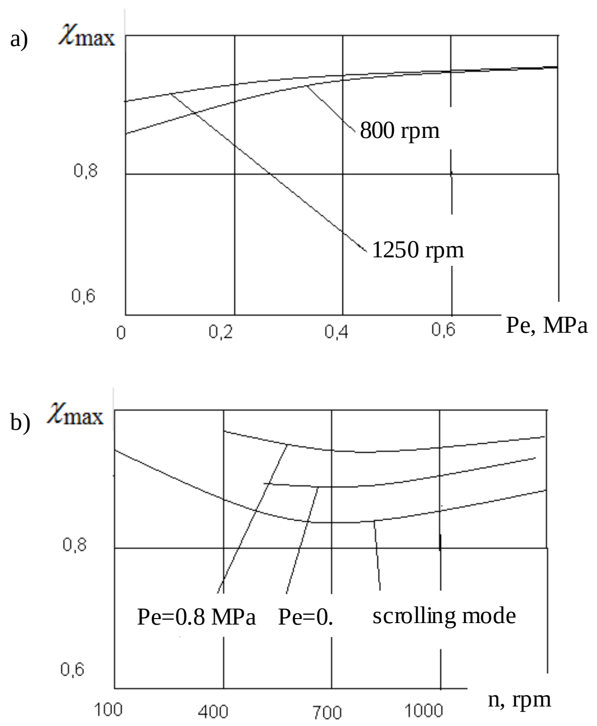

Fig. 1. Variation in relative eccentricity as a function of load Pe and engine rotation frequency n.

Figure 1 shows that at a engine rotation frequency of $n = 800$ rpm, the rate of change in relative eccentricity is greater than at $n = 1250$ rpmfor at the same values of the load on the engine Pe, MPa. Thus, the greater the ratio of the relative eccentricities at the lowest rotation frequency, the higher the ratio of the pressure values without load and under load. For example, according to Fig. 1, the value of the relative eccentricity at the idle rotation frequency and the load variation from zero to 0.4 MPa varies from 0.87 and 0.96. Substituting these values of the relative eccentricity into expression (4), we obtain:

$$\frac{P_1}{P_2} = \frac{(1 + 1{,}5 \cdot 0{,}96^2)}{(1 + 1{,}5 \cdot 0{,}87^2)} = \frac{2{,}3824}{2{,}13535} = 1{,}115$$

Thus, it is established that the ratio of pressures without load and under load with rotation frequency of 800 rpm is a maximum value of $P_1 / P_2 = 1{,}115$. This value was accepted as the most sensitive mode of diagnosing the main bearings. For the values of pressure under load and without load, the trajectory of the movement of the first main bearing at $n = 800$ rpm was considered [15] (Fig. 2):

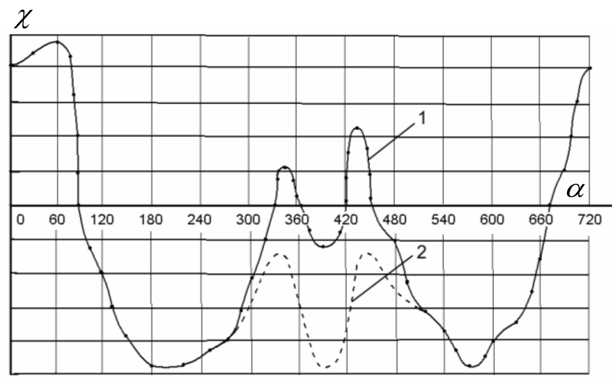

Fig. 2. Trajectory of the movement of the first main bearings, depending on the angle of rotation of the crankshaft α : 1 – no load; 2 – with load.

When the load on the motor is formed, the growth of the relative eccentricity results in an increase in oil flow through the ICE bearings [15] (Fig. 2). So in work [25] it is indicated that with increasing the load from 0 to 500 N·m the oil consumption through some bearings increases by 5-30 %. Feed pressure in each main bearing is via a lubrication system from the oil pump. However, at each instant of time, the shaft necks occupy a different position in the clearance space and the inlet holes can enter the loaded zone. The pressure in the lubrication system is considerably less than the hydrodynamic pressure in the loaded area of the lubricating layer. This leads to the "locking" of the hole and the countercurrent of the lubricant back to the lubrication system.

In the paper [19], the authors investigated the relationship between the pressure pulsation created by the oil pump A_p and the pressure pulsations from the oscillation of the crankshaft necks A_s. The results of the research showed that the pressure pulsations in the central oil line created by the oil pump at a rotation frequency of the crankshaft $n = 1200$ rpm are insignificant and equal to $A_p = 0{,}009 - 0{,}010$ MPa. At the same time, the amplitude of pressure pulsations from the oscillation of the necks of the crankshaft is $A_s = 0{,}04 - 0{,}06$ MPa (Fig. 3).

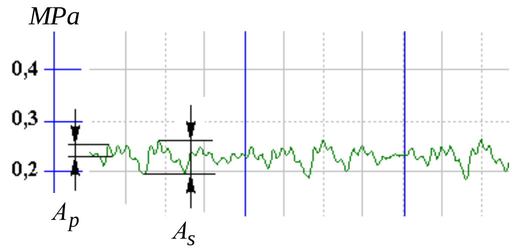

Fig. 3. Oscillogram of pressure pulsations created by the oil pump and the oscillation of the crankshaft necks.

The design capacity of the oil pump (Q_p) is determined as [26]:

$$Q_p = \frac{Q_{gt} \cdot n_f}{60} \quad (5)$$

where: Q_{gt} is volume of the cavities between the gear teeth, m3/r; n_f is rotation frequency of the pump gear, min-1.

According to expression (5), the oil feed by the pump in the central oil line changes linearly. The diagram of the oil feed to the connecting rod bearings is shown in Fig. 4.

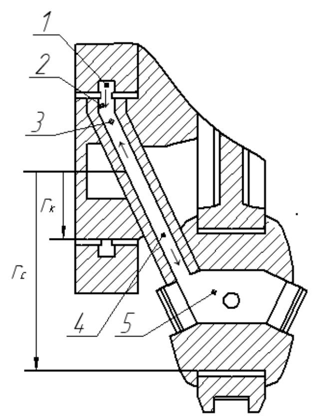

Fig. 4. Diagram of the oil feed to the connecting rod bearings: r_k – radius of the crankshaft main journal, r_c - rotation radius of the outlet holes in the connecting rod bearing (crank), 1 – annular groove, 2 – channel in the crankshaft main journal, 3 – inlet channel for feeding the crankshaft main journal, 4 – feed channel, 5 – rod cavity.

The pressure value in the central oil line (P) is determined by the expression [3, 15, 26]:

$$P = \frac{Q^2 \cdot \rho}{(\mu \cdot f)^2 \cdot 2} \quad (6)$$

where: Q is oil flow rate, kg/s; ρ is oil density, kg/m3; μ is flow coefficient; f is total area of sections, clearances, m2.

The changes in the pressure and carrying capacity are related to the effect of dynamic and static pressures in the channels of the crankshaft main and rod journals.

In connection with the effect of dynamic and static pressures in the channels of the crankshaft main and connecting rod bearings, the change in the carrying capacity through the crankshaft main and connecting rod bearing is also characterized by dynamic and static carrying capacities. Three regularities of the change $\mu \ f$ can be distinguished [19]:

1. The section of the static oil flow (up to 1200 min-1) through the crankshaft main and connecting rod bearing, which is characterized by the condition:

$$\mu_1 \cdot f_1 = \mu_{mc} \cdot f_{mc} + \mu_{rb} \cdot f_{rb} \quad (7)$$

where: $\mu_{mc} \cdot f_{mc}$ is static carrying capacity of the crankshaft main bearing, m^2; $\mu_{rb} \cdot f_{rb}$ is static carrying capacity of the connecting rod bearing, m^2.

2. The section of the dynamic oil flow (from 1200 min-1 to 1900 min-1) through the crankshaft main and connecting rod bearing, which is characterized by the condition:

$$\mu_2 \cdot f_2 = \mu_{mc} \cdot f_{mc} + \mu_{rb} \cdot f_{rb} + \mu_{tc} \cdot f_{tc} \quad (8)$$

where: $\mu_{tc} \cdot f_{tc}$ is total dynamic carrying capacity of the rod channel, m2.

$$\frac{1}{\mu_{tc} \cdot f_{tc}} = \frac{1}{\mu_{dc} \cdot f_{dc}} + \frac{1}{\mu_{dmc} \cdot f_{dmc}} \quad (9)$$

where: $\mu_{dc} \cdot f_{dc}$ is dynamic carrying capacity of the rod journal, m2; $\mu_{dmc} \cdot f_{dmc}$ is dynamic carrying capacity at the inlet to the channel of the crankshaft main journal, m2.

3. The section of the static and dynamic oil flow (over 1900 min-1) through the crankshaft main bearing and connecting rod bearing, which is dominated by the dynamic component at the inlet to the crankshaft main bearing, is characterized by the condition:

$$\mu_3 \cdot f_3 = \mu_{mc} \cdot f_{mc} + \mu_{tc} \cdot f_{tc} \quad (10)$$

Wherein, component (9) is negligibly small. The rod channel can be completely or partially locked.

Let us analyze the change in dynamic pressures. The value of the dynamic pressure in the rod channel (P^1_{rc}) is determined, MPa:

$$P^1_{rc} = \gamma \cdot r^2_{rc} \cdot \frac{\omega^2}{2 \cdot g} \quad (11)$$

where: r_{rc} is rotation radius of the output holes in the connecting rod bearing (crank) (r_{rc} =0.045 m (for ZMZ-4062)); γ is oil density (γ =900), kg/m^3; ω is rotation frequency, rad/s; g is free fall acceleration, rad/s^2.

Let us calculate the value of the dynamic pressure at the inlet to the channel in the crankshaft main journal (P^1_{dmc}), MPa:

$$P^1_{dmc} = \gamma \cdot r^2_{dmc} \cdot \frac{\omega^2}{2 \cdot g} \quad (12)$$

where: r_{dmc} is radius of the crankshaft main journal r_{dmc} =0.031 m (for ZMZ-4062).

The action of these two components is opposed. The component P^1_{rc} with an increasing engine crankshaft rotation frequency increases directly as the square of the rotation frequency of the shaft and tends to increase the oil flow through the connecting rod bearing, while P^1_{dmc} counteracts P^1_{rc} with an increasing engine crankshaft rotation frequency and tends to reduce the oil flow in the rod channel.

Thus, in the formula $Q = \mu \cdot f \cdot \sqrt{\frac{2 \cdot P}{\rho}}$ for the central oil line we will substitute the values of the pump supply Q by expression (5), and the dynamic pressure P^1_{rc} and P^1_{dmc} calculated by expressions 11, 12. As a result, we will calculate the value of the carrying capacity $\mu \cdot f$ under the influence of the dynamic pressures P^1_{rc} and P^1_{dmc}.

As a result of calculating the carrying capacity for the three conditions, it has been established that the dynamic carrying capacity through the crankshaft main and connecting rod bearings takes significant values in the range of the crankshaft rotation frequency of 1200–1900 min^{-1}, which is illustrated by the dependences in Figs. 5 and 6.

As one can see from Fig. 5, the dynamic carrying capacity of the rod journal $\mu_{dc} \cdot f_{dc}$ ($\mu_{dc1} \cdot f_{dc1}$ – for the clearance in the rod journal of 0.05 mm; $\mu_{dc2} \cdot f_{dc2}$ – for the clearance in the rod journal 0.10 mm), m^2 increases with the growing engine crankshaft rotation frequency. The dynamic carrying capacity at the inlet to the channel of the crankshaft main journal $\mu_{dmc} \cdot f_{dmc}$, m^2, on the contrary, decreases with the growing engine crankshaft rotation frequency. Their cross-points ($\mu_{dc1} \cdot f_{dc1}$ with $\mu_{dmc} \cdot f_{dmc}$ at n =1610 min^{-1}) and ($\mu_{dc2} \cdot f_{dc2}$ with $\mu_{dmc} \cdot f_{dmc}$ at n=1520 min^{-1}) form the maxima of their total action. When the values of the dynamic carrying capacity of the rod and crankshaft main journals are substituted in expression (9), we obtain the dependence of the total dynamic carrying capacity $\mu_{tc} \cdot f_{tc} \cdot 10^{-6}$, m^2, on the engine crankshaft rotation frequency n, min^{-1} (Fig. 6).

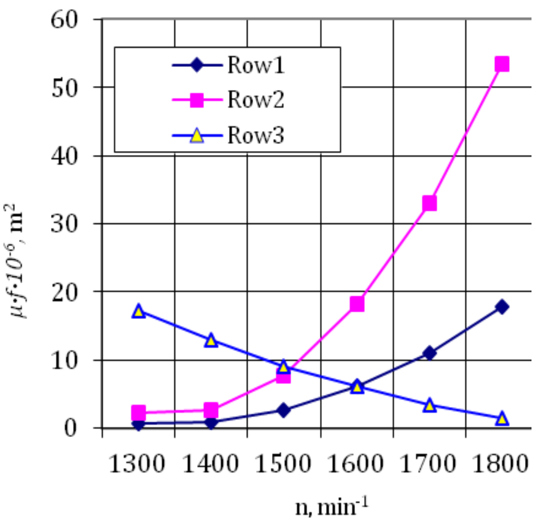

Fig. 5. Dependence of the dynamic carrying capacity $\mu \cdot f \cdot 10^{-6}$, m^2 on the engine crankshaft rotation frequency n, min^{-1}: row 1 – $\mu_{dc1} \cdot f_{dc1}$; row 2 – $\mu_{dc2} \cdot f_{dc2}$; row 3 – $\mu_{dmc} \cdot f_{dmc}$.

Upon the data analysis (Fig. 6) one can see that the larger the increase in the dynamic carrying capacity through the rod journal $\mu_{dc2} \cdot f_{dc2}$ is, the larger is the increment of $\mu_{tc} \cdot f_{tc} \cdot 10^{-6}$ ($\mu_{tc2} \cdot f_{tc2}$ versus $\mu_{tc1} \cdot f_{tc1}$).

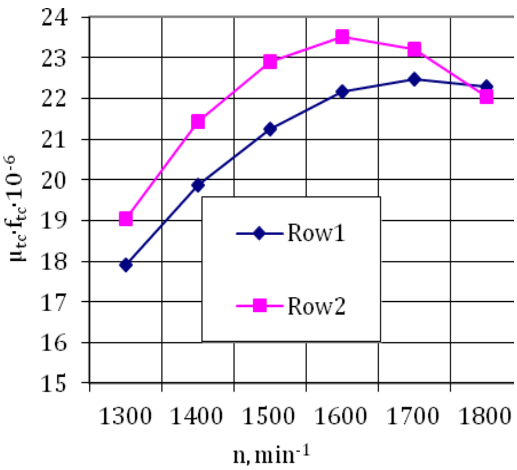

Fig. 6. Dependence of the total dynamic carrying capacity $\mu_{tc} \cdot f_{tc} \cdot 10^{-6}$, m², on the engine crankshaft rotation frequency n, min⁻¹: row 1 – $\mu_{tc1} \cdot f_{tc1}$; row 2 – $\mu_{tc2} \cdot f_{tc2}$.

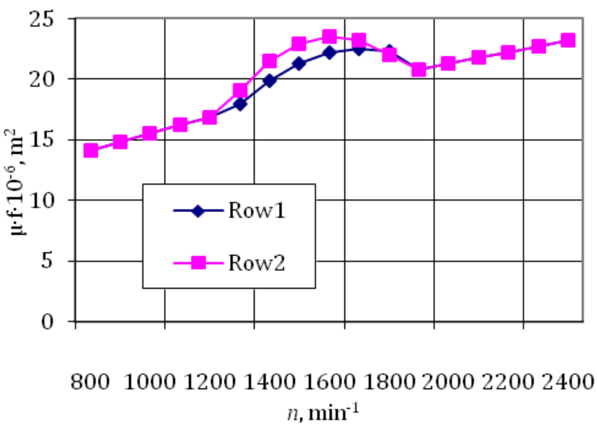

Fig. 7. Dependence of the total carrying capacity $\mu \cdot f \cdot 10^{-6}$, m² on the engine crankshaft rotation frequency: row 1 – Z_c =0.05 mm; row 2 – Z_c =0.10 mm.

Taking into account the change in the dynamic and static carrying capacities for the three conditions, we determined the dependence of the total carrying capacity $\mu \cdot f \cdot 10^{-6}$, m² on the engine crankshaft rotation frequency n, min⁻¹, (Fig. 7). When substituting the values of the total carrying capacity obtained from expressions (8, 9, 10, 11, 12) (Fig. 7) into expression (6) to calculate the pressures, we obtained the dependence of the pressure in the central oil line P, MPa, on the engine crankshaft rotation frequency n, min⁻¹, (Fig. 8).

Using the dependence in Fig. 8, we have established that the most sensitive diagnostic parameter in determining the technical state of the connecting rod bearing will be a decrease in the linearity of the pressure growth in the central oil line. The diagnostic mode is the range of the engine crankshaft rotation frequency from 1200 min⁻¹ to 1900 min⁻¹, at which there appears a decrease in the pressure growth linearity. This diagnostic mode depends on the geometric dimensions of the engine and requires specification for other types of engines.

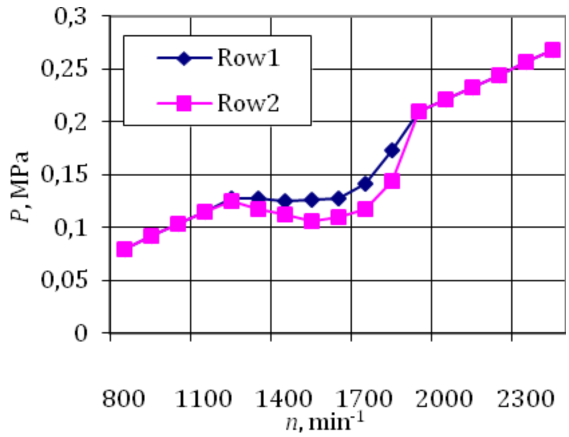

Fig. 8. Dependence of the pressure in the central oil line P, MPa, on the engine crankshaft rotation frequency n, min⁻¹: row 1 – P_1; row 2 - P_2.

We determined three regularities of the change in the carrying capacity $\mu \cdot f$ (7), (8), (10) through the crankshaft main bearing and connecting rod bearing, which makes it possible to establish the condition of the violation of the pressure increase linearity in the central oil line with increasing clearances in the bearings of the crank mechanism.

3. RESEARCH METHODOLOGY

We used a diagnostic complex for carrying out experimental studies; their configuration is described in detail in [19, 20, 22]. To measure the pressure pulsations, we used a D06M-3(U2) pressure sensor with the operating range of 0.06 ... 0.6 MPa [19]. During the metrological tests of the D06M-3(U2) pressure sensor, in order to reduce the interference from the ignition system and various external devices, we developed a digital strain gage amplifier (DSGA) with the gain ratio of 1000 [15, 24]. The DSGA was made as a separate unit on the pressure sensor and screened together with the pressure sensor [19]. A connecting fitting was made to connect the D06M-3(U2) sensor to the main oil line of the ZMZ-4062 engine.

A personal computer with a MT-10KM motor tester [20,22] was used in the study as registering equipment. The object of the tests was the crank mechanism and the lubrication system of the ZMZ-4062 engine. The ZMZ-4062 engine is a 4-cylinder, 16-valve engine with the 1-3-4-2 firing sequence, equipped with a microprocessor control system [20,22]. The engine was installed on a research stand with a 5.5 kW drive motor, a 4-speed gearbox able to crank the engine at the engine crankshaft rotation frequency: transmission 1 – 240 min^{-1}, transmission 2 – 480 min^{-1}, transmission 3 – 880 min^{-1}, transmission 4 – 1480 min^{-1} (Fig. 9).

Fig. 9. Research stand for testing the ZMZ-4062 engine.

Fig. 10. Switch of electromagnetic injectors (engine loader) in the operating conditions when diagnosing the ZMZ-4062 engine.

The research stand is equipped with a regular engine control system with a dashboard, which includes the following: an electronic engine control unit with connecting devices, an ignition switch, a fuse block and a relay. The main device that provides loading conditions for the CM bearings is the switch of electromagnetic injectors (engine loader) [19,20,22]. It is intended for diagnosing the mechanisms and systems of the internal combustion engine, and in the functioning mode - for diagnosing the ZMZ-4062 engine it is shown in Fig. 10.

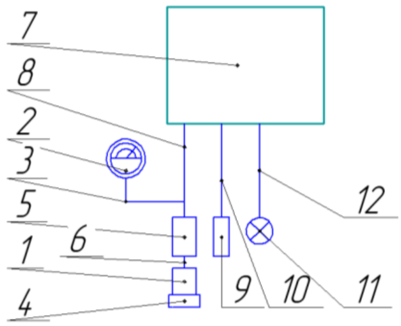

Fig. 11. Schematic diagram of the device for determining the technical state and the service life of internal combustion engines.

Fig. 12. Strain gauge pressure sensor connected to the central oil line by means of a connector and a signal amplifier integrated in one housing with the pressure sensor.

The device for determining the technical state and the service life of internal combustion engines (Figs. 11 and 12) consists of a strain gauge (piezoelectric) pressure sensor of type 1, the standard pressure indicator 2 (or indicator lamp) in the cabin, and connecting wiring of the nominal pressure indicator 3. The strain gauge (piezoelectric) pressure sensor is connected to the central oil line by means of connector 4. The signal amplifier 5 is integrated into one housing with the strain gauge (piezoelectric) pressure sensor and connected to it with the wiring 6. The standard modernized electronic engine control unit 7 has the wiring 8, through which the signal amplifier 5 is connected to the electronic engine control unit 7. The oil level sensor 9 is connected by a wiring 10 to the electronic control unit of the engine 7 and the CHECK ENGINE lamp 11 connected to the electronic engine control unit 7 by the wiring 12.

The method for determining the technical state and the service life of internal combustion engines is as follows. A device is connected to determine the technical state and the service life of internal combustion engines. The engine is started, while the oil pump starts to pump oil into the central oil line, and pressure pulsations are formed in the pipeline. The mobile device starts to move, while the engine remains in service. The pressure sensor measures the pressure pulsation amplitude in the central oil line within the area from the filter to the crankshaft bearings under load. The signal amplifier amplifies the signal in proportion to pressure pulsations. The amplified signal is fed to the electronic engine control unit and to the standard pressure indicator (or indicator light) in the cabin. The indicator displays an average pressure in the central oil line. The strain gauge (piezoelectric) pressure sensor measures the pressure pulsation amplitude for a number of rotation frequency and load engine conditions (for example, for 10 table values of the pressure amplitude that are stored in the memory of the electronic control unit). The electronic engine control unit compares the measured pressure pulsation amplitude in operation and the table one (reference, defined for the new engine), and using the comparison determines the technical state and the service life of the engine. Based on the comparison of the measured pressure pulsation amplitude in operation and the table one, the electronic control unit also generates a decision on the influence on the engine crankshaft rotation frequency. In case of an unacceptable decrease in the pressure pulsation value in the main line, the electronic control unit provides adaptive control of the crankshaft rotations towards their increase to ensure a trouble-free operation of the engine. At the same time, the oil level sensor measures the oil level in the engine crankcase to avoid the engine damage, since a decrease in the pressure pulsation amplitude may occur due to a violation of the oil level in the engine crankcase. The results of measuring the pressure pulsation amplitude in operation are stored in the memory of the electronic control unit and output as trouble codes to the dashboard via the CHECK ENGINE lamp. When the normal value of the pressure pulsation amplitude is restored, the error code is eliminated and the engine operates normally, while the engine crankshaft rotations are reduced to the minimum stable idling rotation frequency.

4. EXPERIMENTAL RESEARCH

When carrying out the experiments to determine the connection between the technical state of crankshaft main bearings and the value of the pressure signal, we have established that the most sensitive mode, in which the influence of the technical state of the crankshaft main bearings is maximum, is the mode at the engine crankshaft rotation frequency close to the idling rotation frequency for the ZMZ-4062 engine n =800 min^{-1} and (the load for the first cylinder - cylinders 2, 3, 4 are off). The first cylinder operates through a cycle when the throttle valve is fully opened. The diagnostic parameter was the difference in the amplitudes of the minimum pressure values of two adjacent cycles during the operation through a cycle, with and without load in the points corresponding to the start of the combustion cycle in the first cylinder [19]. It resulted in the following dependence (Fig. 13).

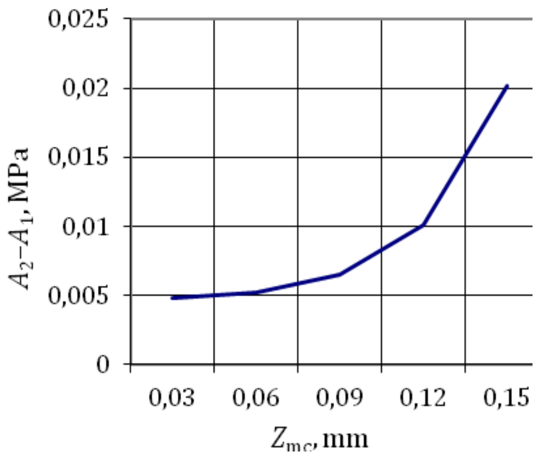

Fig. 13. Dependence of the difference in the minimum pressure amplitudes $A_2 - A_1$, MPa of the two adjacent cycles during the operation of the first cylinder (2nd, 3rd, 4th cylinders are off) through a cycle, with and without load on the technical state of the first crankshaft main bearing (values of the clearance Z_c, mm) when the throttle valve is fully opened (P_{op} =max; n =800 min^{-1}; Z_{cr} = 0.05 mm; t_m = 90 C (oil temperature)).

The relationship of these operational parameters is described by a third-order polynomial:

$$Z_c = 1.961 \cdot 10^5 \cdot (A_2 - A_1)^3 - \\ -7.725 \cdot 10^3 \cdot (A_2 - A_1)^2 + \\ +97.548 \cdot (A_2 - A_1) - 0.28 \qquad (13)$$

Thus, measuring the difference in the pressure amplitudes of the two adjacent cycles during the operation of the diagnosed cylinder (the other three are off) through a cycle, with and without load, using equation (13), it is possible to determine the actual wear of any crankshaft main bearing [19].

It has been established that the limiting value of the difference of the minimum pressure amplitudes under load and without load for culling the crankshaft main bearing at the engine crankshaft rotation frequency of n=800 min^{-1} is 0.02 MPa.

As a result of mathematical processing of the obtained experimental data for the ZMZ-4062 ICE, we composed a general regression equation connecting the maximum increment of the minimum pressure amplitude with the size of the clearances in the crankshaft main and rod journals [19]:

$$\Delta P = \left(-600 \cdot Z_{rj} + 115\right) \cdot Z_c^2 - \\ -\left(-197 \cdot Z_{rj} + 25.017\right) \cdot Z_c + \\ +\left(-11.66 \cdot Z_{rj} + 1.348\right) \qquad (14)$$

where: ΔP – maximum increment of the minimum pressure amplitude, MPa; Z_c – clearance in the crankshaft main journal, mm; Z_{rj} – clearance in the rod journal, mm.

In order to use the results of the multi-factor experiment, it is necessary to solve the inverse problem, to determine the general regression equation with regard to the clearance in the rod journal Z_{rj}, mm. The recalculation of the clearance in the rod journal Z_{rj} resulted in a general regression equation connecting the clearance in the rod journal with the maximum increment of the minimum pressure amplitude ΔP, MPa and the clearance in the crankshaft main journal Z_c, mm:

$$Z_{rj} = \left(243.889 \cdot Z_c^2 - 68.217 \cdot Z_c + 4.97\right) \cdot \Delta P - \\ -\left(34.898 \cdot Z_c^2 - 8.076 \cdot Z_c + 0.467\right) \qquad (15)$$

Experimental results of diagnosing crankshaft main and connecting rod bearings of KAMAZ-740.11 crank mechanism (EURO).

As a result of the experimental studies, we obtained the dependences of the pressure change in the central oil line of the crankshaft main and connecting rod bearings of KAMAZ-740.11 crank mechanism (EURO) using the pressure sensor with a personal computer (electronic oscilloscope) (Figs. 13-15).

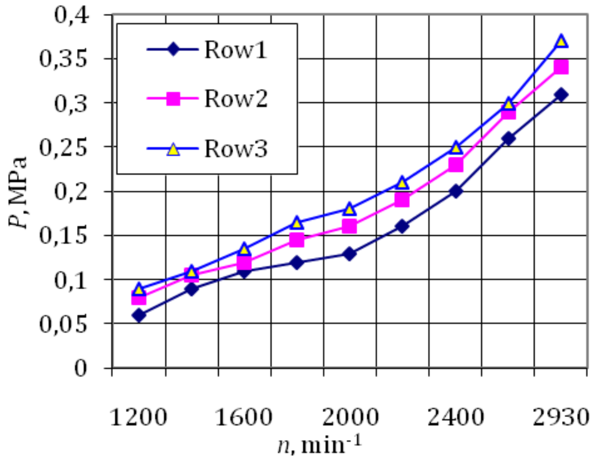

Fig. 14. Dependence of the pressure change in the central oil line on the ICE crankshaft rotation frequency without load. Clearances in the first crankshaft main journal: row 1 - 0.137 mm, row 2 - 0.132 mm, row 3 - 0.127 mm.

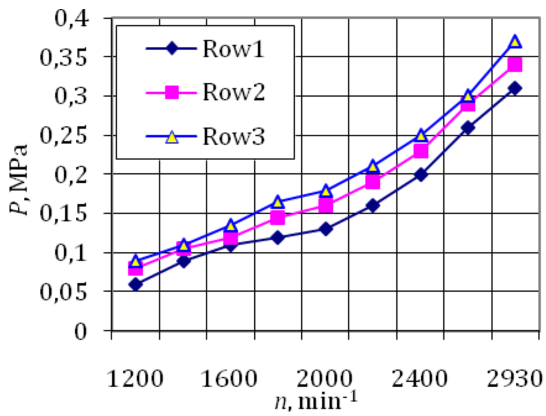

Fig. 15. Dependence of the pressure value in the central oil line on the ICE crankshaft rotation frequency under the load of 500 N m. Clearances in the first crankshaft main journal: row 1 – 0.137 mm, row 2 – 0.132 mm, row 3 – 0.127 mm. Clearance in the first rod journal - 0.08 mm.

As it can be seen from the diagram in Fig. 14, an increase in the clearance in the first crankshaft main bearing reduces the pressure value in the central oil line at different rotation frequency.

The maximum deviation is at 2000 min^{-1}. It creates the conditions for an earlier rupture of the continuity of the oil flow in the channels of the crank mechanism. It increases the probability of a failure of the bearings in the crank mechanism [19].

As a result of analyzing the dependence (Fig. 15), one can be assert that when the clearance is changed by 0.005 mm (rows 3 and 2), the pressure in the central oil line decreases more intensively, and its difference increases with an increasing frequency. An important factor of such dynamics is the load, which was 500 N m.

Analyzing the dependences in Figs. 14 and 15, one can determine that with an increase in the clearance in the first crankshaft main journal, the pressure in the central oil line decreases, and we can see that the intensity of pressure change increases with increasing frequency. Under load, the pressure is further reduced, which leads to disruption and rupture of the continuity of the oil flow at low ICE crankshaft rotation frequency, i.e. we can conclude that the wear of the crankshaft main journal causes a general decrease in the pressure in the central oil pipeline throughout the dependences (Figs. 14 and 15), whereas the wear of the rod journal leads to an increase in the nonlinearity in the range of 1600 ... 2600 min^{-1}, significantly increasing the probability of the bearings failure.

5. CONCLUSION

1. We offered and applied a method for in-place diagnosing the state of friction bearings of the internal combustion engine and developed an instrument providing loading conditions for the bearings of the crank mechanism (patent for invention No. 2344400) [19].

2. We experimentally obtained the dependence (12) for determining the wear degree of the crankshaft main journal Z_c, mm by the difference of the minimum pressure amplitudes $A_2 - A_1$, MPa of two adjacent cycles during the operation of the diagnosed cylinder through a cycle, with and without load at the crankshaft rotation frequency of the ZMZ-4062 engine n =800 min^{-1}. It has been established that the limiting value of the difference of the minimum pressure amplitudes under load and without it for culling the crankshaft main bearing is 0.02 MPa.

3. We developed a method for diagnosing the connecting rod bearings of the crank mechanism (patent for invention No. 2390746) and a method for extending their service life (patent No. 85958) [19,20,22].

4. We obtained the regression equation (15), which allows us to determine the clearance in the rod journal at the known clearance in the crankshaft main journal and the measured value of the maximum increment of the minimum pressure amplitude. This equation can be used to determine clearances in the crankshaft main and rod journals of any internal combustion engine.

Acknowledgement

South Ural State University is grateful for financial support of the Ministry of Education and Science of the Russian Federation (grant No 9.7881.2017/8.9).

REFERENCES

[1] Y. Chen, Y. Sun, D. Yang, *Investigations on the dynamic characteristics of a planar slider-crank mechanism for a high-speed press system that considers joint clearance,* Journal of Mechanical Science and Technology, vol. 31, iss. 1, pp. 75-85, 2017, doi: 10.1007/s12206-016-1209-z

[2] A. Harnoy, *Bearing design in machinery: engineering tribology and lubrication*, New York: CRC Press, 2002.

[3] V.N. Anisimov, *Models of lubricated contact pairs formed by dynamically loaded deformable solids,* Chelyabinsk: SUSU, 2002. (in Russian)

[4] M.S. Sangha, J. B. Gomm, D. Yu, *Neural network fault classification of transient data in an automotive engine air path,* International Journal of Modelling Identification and Control, vol. 3, iss. 2, pp. 148-155, 2008, doi: 10.1504/IJMIC.2008.019352

[5] P.C. Mishra, *Modeling for friction of four stroke four cylinder in-line petrol engine*, Tribology in Industry, vol. 35, no. 3, pp. 237-245, 2013.

[6] P. Nikolakopoulos, A. Zavos, *Slew bearings damage detection using hilbert huang transformation and acoustic methods*, Tribology in Industry, vol. 37, no. 2, pp. 170-175, 2015.

[7] B.S. Ünlü, E. Atik, *Determination of friction coefficient in journal bearings*, Materials and Design, vol. 28, iss. 3, pp. 973-977, 2007, doi: 10.1016/j.matdes.2005.09.022

[8] B. Mănescu, D. Ionuţ, S. Nicolae-Doru, P. Nicolae, C. Adrian, P. Dinel, *Aspects in the synthesis of a variable compression ratio mechanism,* in the IOP Conference Series: Materials Science and Engineering, vol. 252, no. 1, 2017, doi: 10.1088/1757-899X/252/1/012075

[9] V.T. Phung, M. Pacas, *Sensorless harmonic speed control and detection of bearing faults in repetitive mechanical systems*, in the 2017 IEEE 3rd International Future Energy Electronics Conference and ECCE Asia, IFEEC - ECCE Asia, pp 1646-1651, 2017, doi: 10.1109/IFEEC.2017.7992294

[10] I.G. Levanov, E.A. Zadorozhnaya, A.L. Dudnikov, *Methods of assessing the resource of the crankshaft bearing of internal combustion engine based on the calculation of hydro-mechanical characteristics*, Tribology in Industry, vol. 37, no. 3, pp. 360-365, 2015.

[11] P.J. Blau, *On the nature of running-in.* Tribology International, vol. 38, iss. 11-12, 1007-1012. 2005, doi: 10.1016/j.triboint.2005.07.020

[12] J.C. Campbell, *Cylinder bore surface roughness in internal combustion engines: Its appreciation and control.* Wear, vol. 19, iss. 2, pp. 163-168, 1972, doi: 10.1016/0043-1648(72)90302-X

[13] J. Sun, C. Gui, Z. Wang, *Research on elasto-hydrodynamic lubrication of a crankshaft bearing with a rough surface considering crankshaft deformation,* Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, vol. 222, iss. 12, pp. 2403-2414, 2008, doi: 10.1243/09544070JAUTO781

[14] Yu.A. Guryanov, *Portable means of express diagnostics of ICE by oil parameters*, Repair, Restoration, Modernization, no. 10, pp. 11-16, 2006. (in Russian)

[15] V.I. Surkin, B.V. Kurchatov, *Lubrication of tractor diesel engines*, Chelyabinsk, 2009. (in Russian)

[16] G. Jing, X. Zhang, B. Wang, C. Xu, X. Wang, Z. Zhao, *Multiple-axle fatigue analysis of a star engine master conrod. Neiranji Gongcheng*, Chinese Internal Combustion Engine Engineering, vol. 38, no. 1, pp. 102-108, 2017, doi: 10.13949/j.cnki.nrjgc.2017.01.017

[17] E.A. Zadorozhnaya, S.V. Cherneyko, M.I. Kurochkin, N.A. Lukovich, *A study the axial and radial rotor stability of the turbo machinery with allowance the geometry of the surface and properties of the lubricating fluid*, Tribology in Industry, vol. 37, no. 4, pp. 445-463, 2015.

[18] K.I. Khansuarov, V.G. Tseitlin, *Technique of measuring pressure, flow, quantity and level of liquid, gas and steam,* Moscow: Publishing House of Standards, 1990. (in Russian)

[19] A.V. Gritsenko, S.S. Kukov, *Diagnostics of crankshaft main bearings of the crank mechanism by the pressure parameters in the central oil line*, Bulletin of Krasnoyarsk State Agrarian University, Krasnoyarsk, no. 3, pp. 143-147, 2009. (in Russian)

[20] A.V. Gritsenko, S.S. Kukov, K.A. Tsyganov, A.V. Gorbunov, Patent No. 2474715 RUG01 M 15/00, *A method for determining the technical state of an internal combustion engine and an electronic device for its implementation*, Claimed on 12.10.11, Published on 10.02.13, Bul. no 4, no. 2011141374. (in Russian)

[21] Yu.A. Guryanov, Patent No. 2082150 RUG 01 N 3/56, *Method for monitoring the wear of friction units,* Claimed on 26.02.93, Published on 20.06.97, Bul. no.17, no. 93009570. (in Russian)

[22] A.V. Gritsenko, S.S. Kukov, Patent No. 2399898 RU G 01 M 15/09, *Method of in-place diagnostics of the degree of wear of bearings of an internal combustion engine,* Claimed on 22.06.09, Published on 20.09.10, Bul. no. 26, no. 2009123720. (in Russian)

[23] N. Nikolic, T. Torovic, Z. Antonic, *A procedure for constructing a theoretical wear diagram of IC engine crankshaft main bearings,* Mechanism and Machine Theory, vol. 58, pp. 120-136, 2012, doi: 10.1016/j.mechmachtheory.2012.07.009

[24] V.N. Prokopiev, V.N. Anisimov, *To the calculation of optimal clearances of unsteady loaded crankshaft bearings*, Cars, tractors and engines, vol. 214, pp. 46-55, 1978. (in Russian)

[25] A.S. Denisov, A.T. Kulakov, *Ensuring the reliability of automotive tractor engines*, Saratov: Saratov State Technical University, 2007. (in Russian)

[26] E.M. Fedyakov, V.K. Koltakov, E.E. Bogdatiev, *Measurement of pressure variables*, Moscow: Publishing House of Standards, 1982. (in Russian)