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Parameters of the lubrication process during operational wear of the crankshaft bearings of automobile engines

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Abstract

The article discusses the dynamics of the lubrication process during the operational wear of the crankshaft bearings of automobile engines. All lubrication modes and wear processes are analyzed. The main attention is paid to the steady and unsteady modes of lubrication of the crankshaft bearings. The nature of the dynamics of changing lubrication regimes is substantiated through operational tribological parameters that characterize the integral degree of existence of the lubricating layer. A model is proposed for the relationship of these parameters with a number of factors, and the nature of their changes with a change in the speed of the crankshaft of the car engine is substantiated.

The results of bench tests of diesel engines of the KamAZ series, speed and load characteristics of the engine in terms of tribological operational parameters are presented. A graphical interpretation of the tribological parameter in the field of the load-speed mode of KamAZ engines is given.

Key words: car engine, crankshaft bearings, lubrication mode, tribological parameter, speed and load characteristics

Introduction

One of the movable mates that limit the life of the engine are the crankshaft bearings. They account for about 18% of the number of failures of all elements of the power units of KamAZ vehicles, the average time between failures is 12% of the engine life before overhaul, and the share of repair costs is over 45% [1,2].

Connecting rod and main bearings of the crankshaft of automotive internal combustion engines are fluid friction bearings lubricated under pressure. The external load vector acting on the crankshaft bearings varies not only in magnitude and direction, but also rotates relative to the crankshaft axis at a certain speed. The crankshaft not only rotates, but also moves relative to the bearings, squeezing out the oil. The thickness of the oil layer changes periodically. As the load increases, it decreases. The oil pressure in the supply line practically does not affect the pressure in the oil layer, but to a large extent affects the amount of oil pumped and the thermal state of the bearing [3,4].

The nature of the duration of the existence of the lubricating layer varies depending on the condition of the plain bearing. Therefore, it is fair to assume that such an operational tribological parameter as P_g, on the one hand, is sensitive to the technical condition of the bearing, and, on the other hand, affects the intensity of its wear [5,6].

In practice, when assessing the resource of car engines with identical and different numbers of connecting rod and main bearings of the crankshaft, it is important to identify the effect of various tribological parameters on the lubrication process of mating parts during operation.

Literature review

During the operation of the engine, the conditions for the operation of bearings in the liquid lubrication mode (hydrodynamic friction) must be ensured in the entire range of operating modes for which they are intended. Under conditions of liquid lubrication, the rubbing surfaces of the mating parts are separated by a continuous layer of lubricant material of considerable thickness, several times greater than the sum of the heights of microroughnesses of the working surfaces. The process of mechanical wear is practically absent, but the processes of fatigue wear, cavitation wear and liquid erosion take place. It is believed that the abrasive action of particles of mechanical impurities contained in engine oil for connecting rod and main bearings is weakened. Oil on its way to the bearings undergoes coarse and fine cleaning in oil filters, as well as in dirt traps located in the crankshaft cavities [7,8].

The contact interaction of friction surfaces occurs during boundary lubrication, when the thickness of the boundary lubricating layer is several molecular layers of lubricant, oriented in the direction of movement of the mating parts, and is commensurate with the sum of the heights of the microroughnesses of the contacting surfaces [9-11].

Boundary lubrication is determined by the properties of boundary lubricating layers arising from the interaction of the material of the rubbing surface and the lubricant as a result of physical adsorption or chemisorption. In this case, the bulk properties of the lubricant do not appear, and the physicochemical interactions on the surfaces determine the nature of friction and wear. With boundary lubrication, molecular-mechanical and corrosion-mechanical types of surface wear are realized. Wear intensity with boundary lubrication is 7-13 orders of magnitude higher than with liquid lubrication [12-14].

Violation of the liquid lubrication occurs when the friction surfaces approach each other so much that in the zones of the highest pressures the oil film breaks and the microroughnesses in the contact spots come into contact. In the case when the load is simultaneously perceived by the oil film and the contact surface irregularities, then this is the mixed lubrication mode.

In the case of the alternate appearance and disappearance of the oil film between the rubbing surfaces, a transient lubrication process occurs. The operation of plain bearings of the crankshaft of automobile engines under the conditions of a transient lubrication process was studied in [18-20]. The intensity of wear during the transitional lubrication process is determined by the ratio of the duration of non-contact and contact types of interactions. It is believed that if the duration of contact between the journal and the bearing is short (no more than 20% of the cycle time), then these critical positions may not be dangerous [21-23].

The transient lubrication process is a general case of interaction of surfaces in lubricated tribocouplings of machine parts. Particular cases are liquid, boundary and mixed types of lubrication, which are considered as steady. At the same time, the dynamic process of transition from one type of lubricant to another is practically not considered.

Steady-state lubrication mode – the work of tribocoupling of parts with constant indicators of the lubrication process over time: minimum thickness of the lubricating layer $h_{\min}$, average temperature of the lubricating layer T_M, friction coefficient f_{mp}, dynamic oil viscosity μ. A consistent set of steady state modes is a static characteristic of the tribocoupling of parts, usually represented as a dependence of process indicators on one of the parameters selected as an independent variable.

The main sign of an unsteady regime is a violation of the condition of constancy of the process indicators in time τ: $h_{\min} = f(\tau)$, $T_M = f(\tau)$, $f_{mp} = f(\tau)$, $\mu = f(\tau)$ etc. Unsteady modes are characterized by cycle average values of indicators that change during the period of transition from one steady state to another. The dynamic characteristic is a time-sequential set of unsteady modes, represented by the dependence of the performance indicators of tribocouplings of parts that change during the transition process [24-25].

Of the variety of transient processes, the most characteristic are the following:

– processes caused by a change in the speed of the v relative movement of the surfaces of tribocouplings of parts. The influencing parameters are the relative change $\delta_v = (v_2 - v_1)/v_H$, the period of change T_v and the nature of the change in speed $v = f(\tau)$;

– processes caused by a change in the external load N on the tribocoupling of parts. The influencing parameters are the relative change $\delta_N = (N_2 - N_1)/N_H$, the period of the change T_N and the nature of the change $N = f(\tau)$;

– processes caused by changes in the properties (eg viscosity) and supply parameters of the lubricant (eg pressure, flow, temperature). The influencing parameters in this case are the relative change of these parameters, the period and nature of the change.

– combined processes, accompanied by the simultaneous impact of changes in speed, load and properties and parameters of the lubricant supply. It is not only the main one in operational conditions, but also the most common case of transients.

The deviation of the lubrication regime from liquid lubrication has a significant effect on the intensification of wear processes. The degree and nature of the deviation are determined by the conditions under which the transient lubrication process takes place, the degree of adaptation of the bearing to it. This indicates the need to have a unified system for comparative analysis of the lubrication process indicators in various operating conditions, the degree of their adaptation to the latter. When analyzing the quality of lubrication processes in tribo-conjugated parts, it is advisable to use relative indicators – the ratio of the indicator during the transition process to its value under steady-state lubrication modes.

An important indicator characterizing the transitional lubrication process is its duration τ_t. This value is calculated from the moment the lubrication mode is changed until the new mode is established. The last moment is determined by the achievement of stable indicators of the lubrication process, corresponding to the new steady state. In this case, the transitional lubrication process can be both incomplete and completed. The duration parameter of the lubricating layer depends on the time τ : $P_g = f(\tau)$. If the τ_t ratio has been established in the tribo-couplings of the parts over time $P_g \cong 1$, then it is considered that the transient process has ended and the non-contact interaction mode has been established, if $P_g \cong 0$ – the contact interaction mode. If the process has not been established, then it is characterized by average values $\overline{P_g}$ changing during the transition from one steady friction mode to another ($0 < P_g < 1$).

The types of interaction of the rubbing surfaces of the tribocoupling parts are described by the Gersey-Striebeck diagram (Fig. 1), which is the dependence of the friction coefficient f_{mp} on the load-speed mode (N – radial load on the bearing, v – linear velocity of the mating surfaces) and the properties of the lubricant in it (μ – dynamic viscosity).

H. Chihos [24] considers this diagram together with the dependence of the lifetime of the lubricating layer P_g and defines three characteristic areas: I – the area of continuous non-contact interactions – liquid lubrication ($P_g = 1$); II – area of alternating contact-non-contact interaction – transient lubrication process ($0 < P_g < 1$); III – area of continuous contact interactions of surfaces ($P_g = 0$).

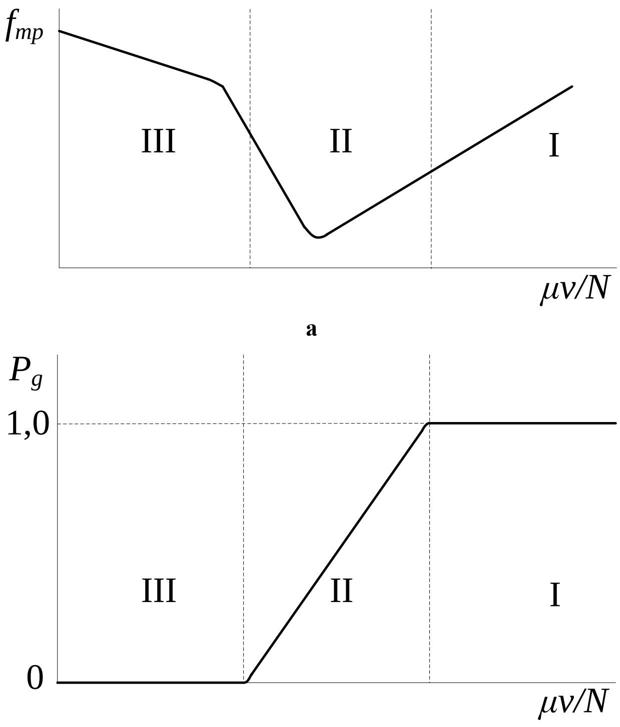

Fig. 1. Combination of the Gersey-Striebeck diagram (a) with the characteristic scheme of parameter change P_g (b).

An analysis of the lubrication process in the crankshaft bearings of automobile engines shows that:

– bearings are designed to operate in a liquid lubrication mode, which ensures minimal wear of the tribological interfaces of parts, however, in real operating conditions they can also operate in other lubrication modes – mixed, boundary, transitional lubrication process, when the wear rate of rubbing surfaces increases significantly;

– the lubrication mode is an important means of influencing the wear processes, and the violation of the liquid lubrication mode has a negative impact on the wear of the tribomechanical system, it is important to determine the conditions for establishing the liquid lubrication mode in the bearing;

– it is advisable to consider the operation of bearings from the standpoint of a transient lubrication process, which is a general case of the interaction of rubbing surfaces of mating parts.

Purpose

The purpose of the work is to substantiate the influence of tribological parameters on the lubrication process of tribocouplings of parts (rod and main bearings of the crankshaft) of the engine.

Results

For a generalized assessment of the lubrication process in the connecting rod and main bearings of the crankshaft, the method of equivalent electrical circuits [7] of automobile engines is used. (fig.2., fig.3) The equivalent electrical circuit is an electrical circuit where voltage is applied to the cylinder block and crankshaft toe, and variable electrical resistances correspond to movable mates in the crank mechanism, cylinder-piston group and gas distribution mechanism . The variability of the resistance is determined by the changing thickness of the lubricating layer with dielectric properties in the contact zone of the tribo-coupling of the parts, depending on the lubrication conditions.

For the implementation of bench tests, a scheme for connecting the electric current to parts of the CPG of diesel engines was proposed (fig. 2).

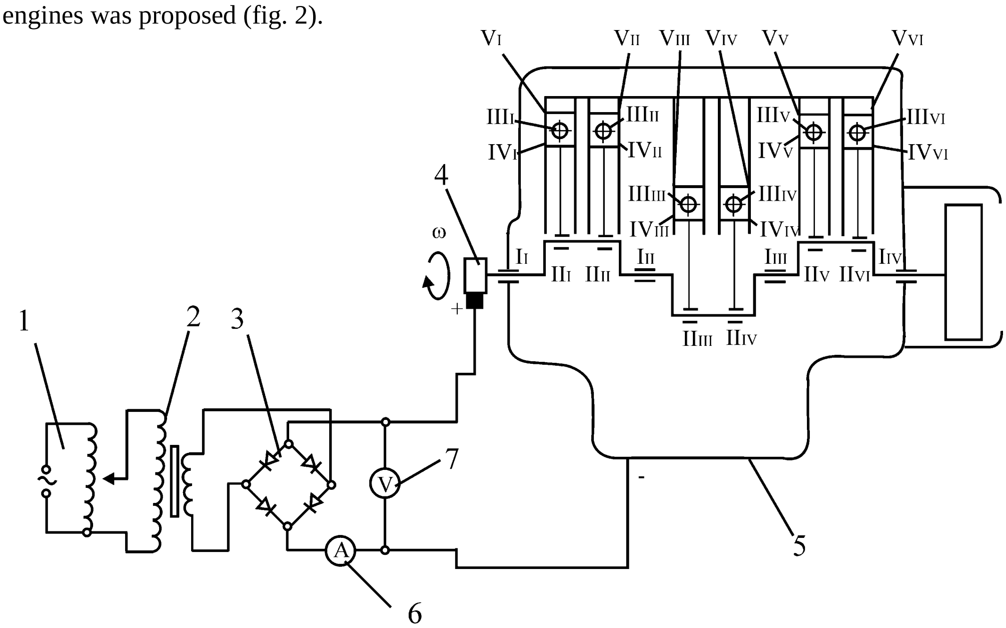

Fig. 2. The scheme of connecting electric current to parts of the CPG of diesel engines: 1 – current source; 2 – resistance for adjusting the current value; 3 – current rectifier; 4 – current collector, screwed in instead of a ratchet; 5 – engine to be run-in; 6 – ammeter; 7 – voltmeter: I – main bearings; II - connecting rod necks of the crankshaft and liners; III – piston fingers and bushings of the upper head of the connecting rods; IV – piston fingers and piston heads; V - cylinder liners, pistons and piston rings.

From the rectifier, electric current is supplied to the "plus" brush unit, and "minus" to the engine block. The brush unit is installed on the side of the oil pump drive pulley, for this it is necessary to unscrew the ratchet and screw in the copper shaft of the brush unit instead. The negative terminal is connected to the cylinder block at the place of attachment of the fuel filter.

The scheme of current distribution through the couplings of the engine (fig. 3) is a system of parallel chains, the first chain is the crankshaft, main bearings, cylinder block; the second chain – crankshaft, CPG, cylinder block. The current branches from the crankshaft to the main sliding bearings (resistance R_1), and through the connecting rods of the crankshaft to the connecting rods (resistance R_2), from the connecting rods to the piston fingers (resistance R_3), from the fingers to the piston heads (resistance R_4) and branches into two branches: from the piston to the sleeve (resistance R_7) and from the piston to the piston rings (resistance R_5) and further from the rings to the sleeve (resistance R_6).

The greatest resistance will be in the first chain (resistance R_1), because when the engine is running, the crankshaft seems to "float" in the oil medium in the main sliding bearings. The electric current will look for the path of least resistance according to Ohm's and Kirchhoff's law, then the largest part of it will pass through the second link - through the parts of the CPG.

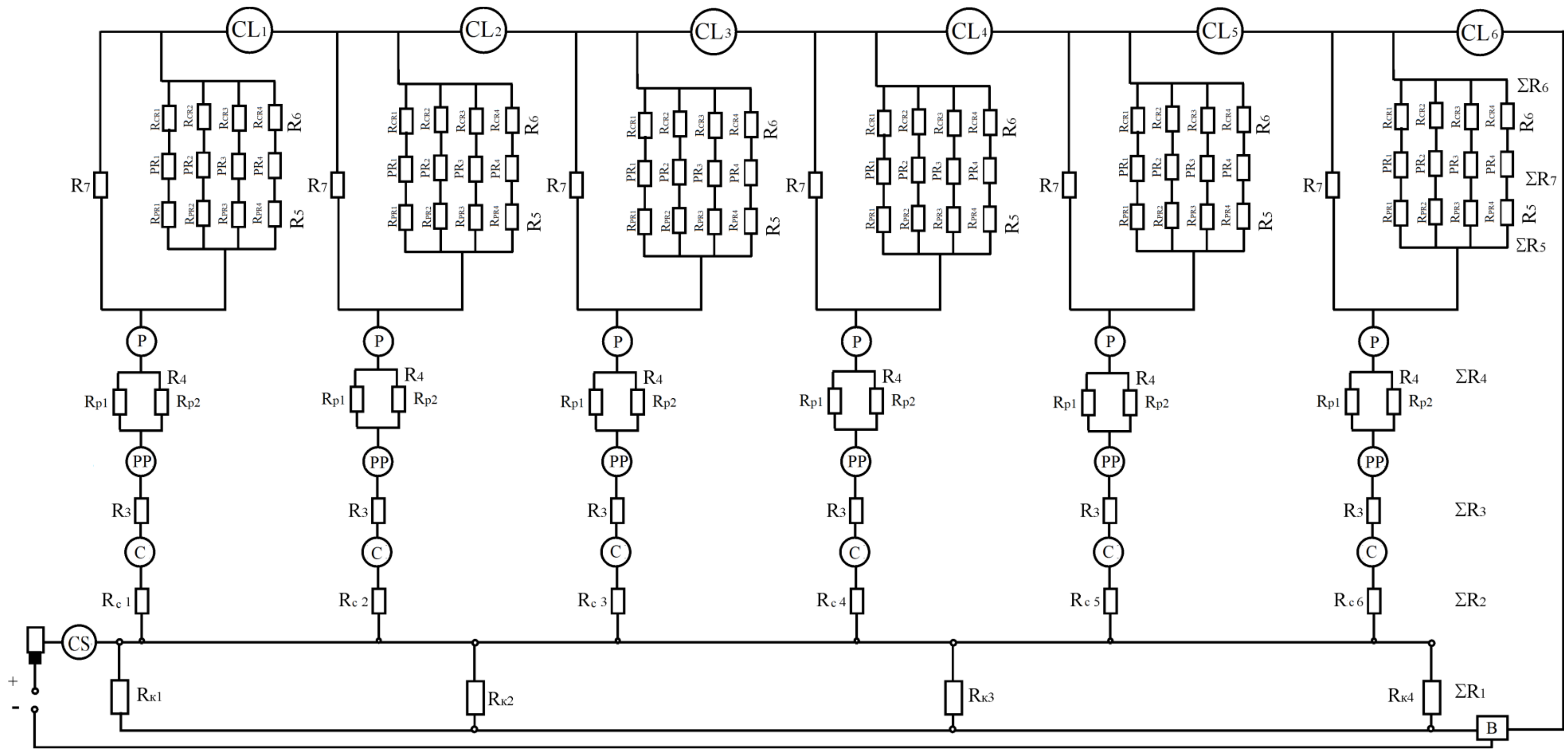

Fig. 3. Scheme of distribution of electric current on the parts of the CPG of diesel engines: CS – crankshaft; B – cylinder block (main bearings); C – connecting rod (connecting rod bushings); PP – piston pin; P – piston (piston heads); PR – piston rings; CL – cylinder liner; R_1 – total resistance between the root neck and the root liners; R_2 – total resistance between the connecting rod neck and connecting rod liners; R_3 – total resistance between the bushing of the upper head of the connecting rod and the piston fingers; R_4 – the total resistance between the piston fingers and the piston heads; R_5 – total resistance between the piston and the ring; R_6 – total resistance between the ring and the sleeve; R_7 is the total resistance between pistons and sleeves.

This is due to the fact that the oil film between the connecting rod neck and the connecting rod liner when the engine is running has a much smaller thickness than that between the main neck and the main slide bearing, that is, it has less resistance. In other elements of CPG parts, the thickness of the oil film is even smaller, since the rings perform a reciprocating movement relative to the mirror of the cylinder liner. Thus, the thickness of the oil film at TDC and TDC of the rings will be minimal to 0.1 μm. At the same time, the mode of marginal friction is observed. The maximum thickness of the oil film will be at the moment when the rings reach the maximum speed and will have a thickness within 10 microns, depending on the oil density. Therefore, the maximum part of the current will pass through the CPG parts when the piston is in the dead center position. At this moment, the current on the parts of the CPG will be distributed almost evenly in the engine, since at the same time the pistons will occupy different positions relative to the sleeve. The difference in the passage of current through the parts of the CPG for one revolution of the crankshaft will be up to 10%.

The resistance between the crankshaft and main bearings is a parallel chain of resistances and is calculated by the expression:

$$\frac{1}{R_1}=\frac{1}{R_{\kappa 1}}+\frac{1}{R_{\kappa 2}}+\frac{1}{R_{\kappa 3}}+\frac{1}{R_{\kappa 4}}.$$

Similarly, the total resistance between the crankshaft and connecting rods is calculated:

$$\frac{1}{R_2}=\frac{1}{R_{C1}}+\frac{1}{R_{C2}}+\frac{1}{R_{C3}}+\frac{1}{R_{C4}}+\frac{1}{R_{C5}}+\frac{1}{R_{C6}}.$$

Current and voltage were recorded using an ammeter and a voltmeter, and then the total resistance R was calculated according to Ohm's law $I=\frac{U}{R}$.

In modern automobile engines, there are at least 20...25 types of movable mates, and their total number depends on the design of the engine, and is 100...150 units and more. Lubrication regimes differ significantly from each other and depend on many factors: purpose, load-speed and thermal regimes, lubricant supply conditions, technical condition, etc.

An analysis of equivalent electrical circuits allows us to conclude that the probability of electric current passing between the cylinder block and the crankshaft is determined by the probability of metal contact in the movable mates of the crankshaft. The contact of the mating of bearing parts occurs when the lubricating layer is destroyed, the integral probability of the existence of a lubricating layer in the tribocoupling is equal to:

$$P_g^{\Sigma}=P_g^{tb}P_g^{gdm}P_g^{cgb}\prod_{i=1}^{i=k}P_{g.i}^{mb}\prod_{j=1}^{j=m}P_{g.j}^{ccp}, \quad (1)$$

where P_g^{tb} is the parameter P_g in the thrust main bearing; P_g^{gdm} – parameter P_g in the group of interfaces of parts of the gas distribution mechanism and its drive; P_g^{cgb} - parameter P_g in the group of interfaces of clutch and gearbox parts; $P_{g.i}^{mb}$ – parameter P_g in the *i*-th main bearing; $P_{g.j}^{ccp}$ – parameter P_g in the *j*-th group of the connecting rod bearing and the cylinder-piston group; *k* and *m* are the number of main bearings and groups of mating parts from the connecting rod bearing and the cylinder-piston group, respectively.

The analysis of formula (1) allows us to draw a conclusion about the relationship between the parameter P_g^{Σ} and the parameter values P_g in each group of mating parts. Moreover, if $P_g^{\Sigma}=1$, then this means the conditions of non-contact interaction in all groups of triboconjugations of parts ($P_g=1$), if $P_g^{\Sigma}=0$, then at least one of the groups has metal contact ($P_g=0$).

The operating conditions of the thrust bearing show that fluid friction predominates in it, and therefore the following condition can be assumed: $P_g^{tb}=1$.

The conditions for the passage of electric current from the crankshaft to the cylinder block through the drive of the gas distribution mechanism and the gas distribution mechanism itself make it possible to assume that $P_g^{gdm}=1$.

The parameter P_g^{ccp} can be estimated by the formula:

$$P_g^{ccp}=1-(1-P_g^{cb})(1-P_g^{ph})(1-P_g^{pp})^2(1-P_g^{pc}), \quad (2)$$

where P_g^{cb}, P_g^{ph}, P_g^{pp}, P_g^{pc} is the parameter P_g in the connecting rod bearing, respectively, of the tribo-couplings of the parts "piston pin-bushing of the piston head of the connecting rod", "piston pin-piston boss" and "piston-cylinder".

An analysis of formula (2) shows a complex dependence of the parameter P_g^{ccp} on the type of interaction in tribo-couplings of parts. If the parameter is in one of the tribological conjugations of the parts $P_g = 1$, then the values of the parameter P_g in other conjugations do not have a significant effect on the value of P_g^{ccp}.

The operating conditions of the tribocouples "piston pin-piston head sleeve", "piston pin-piston boss" and "piston-cylinder" indicate that they are dominated by boundary friction $P_g = 0$, and therefore the following relationship can be assumed:

$$(1-P_g^{cb})(1-P_g^{pp})^2(1-P_g^{pc}) = 1. \tag{3}$$

Hence it follows that $P_g^{ccp} = P_g^{cb}$. Based on the assumptions made, the P_g^{Σ} operational friction parameter is determined by the formula:

$$P_g^{\Sigma} = \prod_{i=1}^{i=k} P_{g.i}^{mb} \prod_{j=1}^{j=m} P_{g.j}^{cb}, \tag{4}$$

The indicator is P_g^{Σ} used for a comparative assessment of engines with identical numbers of connecting rod and crankshaft main bearings. To evaluate engines with different numbers of bearings, an "equivalent crankshaft bearing" model is proposed, which has a generalized assessment of the lubrication process in connecting rod and main bearings. For quantitative assessment, the parameter is used E_g - "integral degree of existence of the lubricating layer", the value of which is determined by the formula:

$$E_g = \sqrt[k+m]{\prod_{i=1}^{i=k} P_{g.i}^{mb} \prod_{j=1}^{j=m} P_{g.j}^{cb}}. \tag{5}$$

The value of the parameter E_g changes from the maximum value $(E_g)_{\max} = 1$, which characterizes the steady state of liquid lubrication in all crankshaft bearings, to the minimum value $(E_g)_{\max} = 0$, at which at least one bearing operates in the mode of boundary lubrication or dry friction. Intermediate values of the parameter $0 < E_g < 1$take place under the conditions of a transient lubrication process with successive alternation of liquid and boundary lubrication in time.

On fig. 4 shows the dependences of tribological parameters P_g^{Σ} and E_g on the speed of the crankshaft.

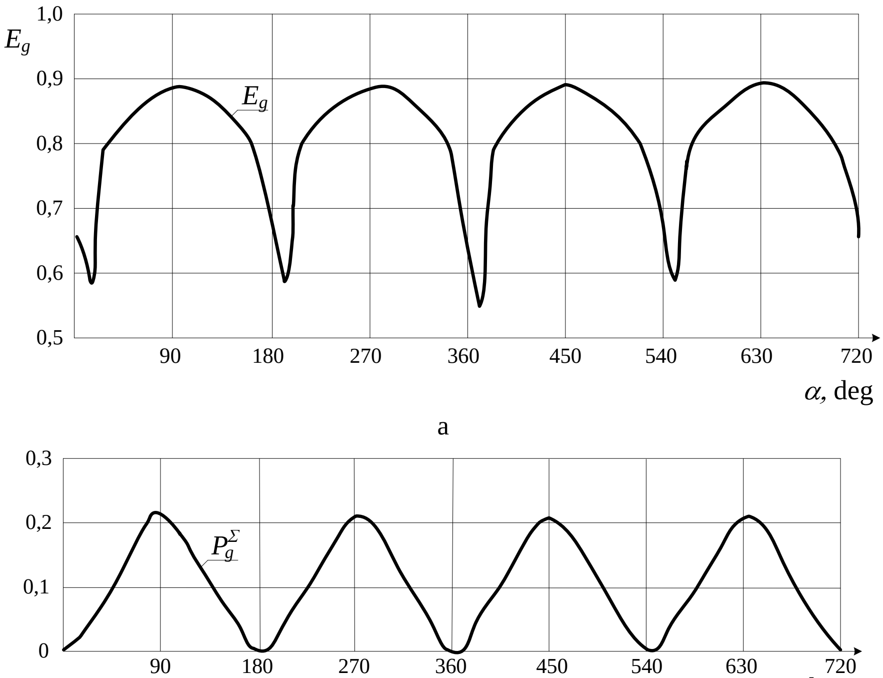

Fig. 4. Scheme of formation of parameters E_g (a) and P_g^{Σ} (b)

It is determined that under operating conditions the value of the parameter E_g depends on a number of factors:

$$E_g = E_g(l_b, d_b, M, n, \mu(T_M), T_{MP}, p_P, h_{кр}, \Delta, ...). \quad (6)$$

The factors involved in the model (6) change according to various patterns:

– bearing length l_b and bearing diameter d_b remain practically unchanged;

– the dynamic viscosity of the oil μ is determined by the viscosity-temperature properties of the oil $\mu(T_M)$ according to SAE, due to the aging of the base base, dilution with fuel, and the operation of additives;

– the crankshaft speed n and torque M vary over a wide range depending on the load and speed modes of the engine;

– the critical thickness of the lubricating layer $h_{кр}$ depends on the roughness and other micro- and macrogeometric parameters of the friction surfaces, is formed initially during manufacture and installation, decreases during running-in, changes insignificantly during the period of steady (normal) wear, and increases during accelerated wear;

– the diametrical clearance Δ is formed during manufacture and installation, increases due to wear of the journals and liners of the crankshaft, both at the running-in stage and during periods of steady (normal) and accelerated wear;

– oil pressure is p_P determined by the design and characteristics of the elements of the main oil line and the lubrication system, increases with increasing crankshaft speed, oil viscosity, and decreases with increasing oil filter contamination, wear of the oil pump and crankshaft bearings, overheating and dilution by fuel;

– the oil temperature T_M depends on the thermal state of the engine parts, the operation of the engine preheating system; in the starting mode during warm-up increases, while driving it depends on the load-speed mode of the engine and on the factors of warming and cooling the engine.

Thus, the integral degree of existence of the lubricating layer E_g in the crankshaft bearings depends on the oil temperature and the load-speed mode, and these dependencies have their own characteristics at the stages of running-in, steady-state (normal) and accelerated wear.

During the running-in of bearings, the factors $h_{кр}$ and are variables Δ.

Under the same factors, the conditions of the load-speed mode of operation, the thermal state of the engine and the properties of the engine oil, the values of the factors M, n, T_{MP} and $\mu(T_M)$ in model (6) are unchanged, which makes it possible to determine the values of the parameter E_g depending on the factors of the technical condition of the bearings using the dependence $E_g = E_g(h_{кр}, \Delta)$.

The operational wear of bearings depends on a variable factor Δ. The validity of diagnosing the crankshaft bearings is determined by the fact that under the same operating modes, the thermal state of the engine and the properties of the engine oil, the values of the parameters M, n, $T_{МП}$, $h_{кр}$ and $\mu(T_M)$ in model (6) are unchanged, and it becomes possible to determine the diametrical clearance using the established dependence $\Delta = \Delta(E_g)$.

When the engine is warming up at idle, the values of the factors Δ and $h_{кр}$, as well as the viscosity-temperature characteristic of the oil $\mu(T_M)$ in model (6) are unchanged, and the factors n, $T_{МП}$ and $p_П$ depend on the time τ in the start mode. This allows you to determine the parameter values E_g depending on the values of the crankshaft speed n and oil temperature T_{MP} using the model $E_g = E_g(T_{MP}, n, \tau)$.

When the engine is running, the factors M and n are variables. With the same thermal and technical conditions of the engine and the properties of engine oil, the values of the parameters T_{MP}, $\mu(T_M)$, Δ and $h_{кр}$ in model (6) are unchanged and it becomes possible to determine the value of the parameter E_g depending on the operating mode factors (M and n) using the model $E_g = E_g(M, n)$.

The basis of experimental studies of the lubrication process in the crankshaft bearings, depending on the load-speed mode of the engine on the stand, is an enlarged model containing input (torque on the crankshaft M, crankshaft speed at idle n) and output (indicator E_g and dependencies $E_g = E_g(M, n)$ and $W_i = W_i(M, n)$ variables.

The test object was the KamAZ-740.14-300 diesel engine, which is used on KamAZ-53212, 43353, 53229, 65115 vehicles, and is identical in design of the lubrication system and crank mechanism with engines of other modifications KamAZ-740.11-240, 740.13 -260 used on many vehicles. The tests were carried out in the engine testing laboratory of the ERM department. The engine was installed on the stand of the company "AVL" with a hydraulic brake company "SCHENCK". The test engine was run-in, and the operating time at the time of testing was about 1200 moto-hours.

When testing the engine, coolant temperatures were maintained from 80ºC to 85ºC and oil from 75ºC to 80ºC. Steady operating modes were sequentially set at crankshaft speed n = 1000, 1400, 1800, 2200 and 2400 min $^{-1}$ with a stepwise change in torque M at each frequency from 100 to 1000 N·m with a step of 100 N·m.

The measurements were carried out in the forward and reverse directions. The duration of measurement in each mode was 30 s after an exposure of 30 s.

The results of parameter measurements E_g in each operating mode were averaged. The obtained load and speed characteristics of the engine in terms of the parameter E_g made it possible to draw the following conclusions:

– with an increase in the load on the engine at a constant crankshaft speed, the parameter decreases, which indicates a progressive deterioration of the liquid lubrication (fig. 5);

– with an increase in the shaft rotation frequency at a constant torque, the dependence of the parameter has a parabolic form (fig. 6) with a maximum characterizing the best lubrication conditions in the frequency range from 1250 to 1550 min $^{-1}$.

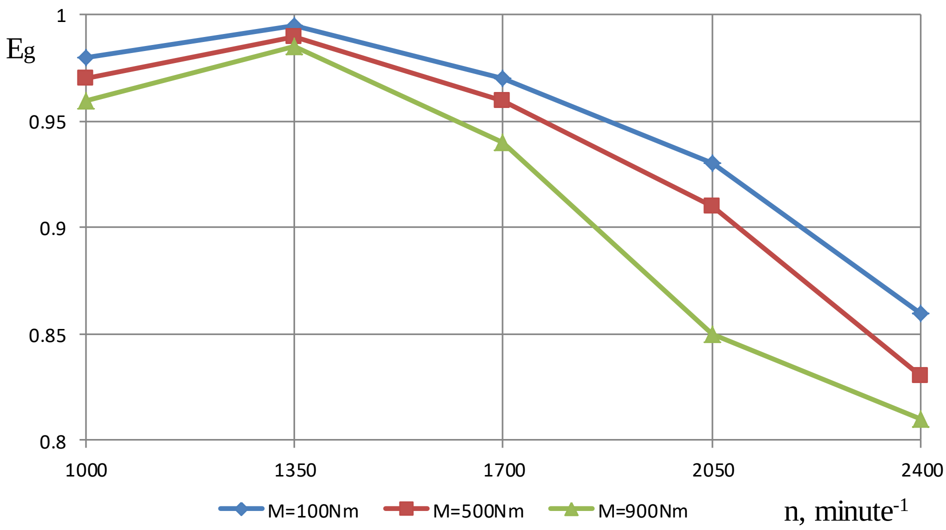

Fig. 5. Motor speed characteristics by parameter E_g (*n*)

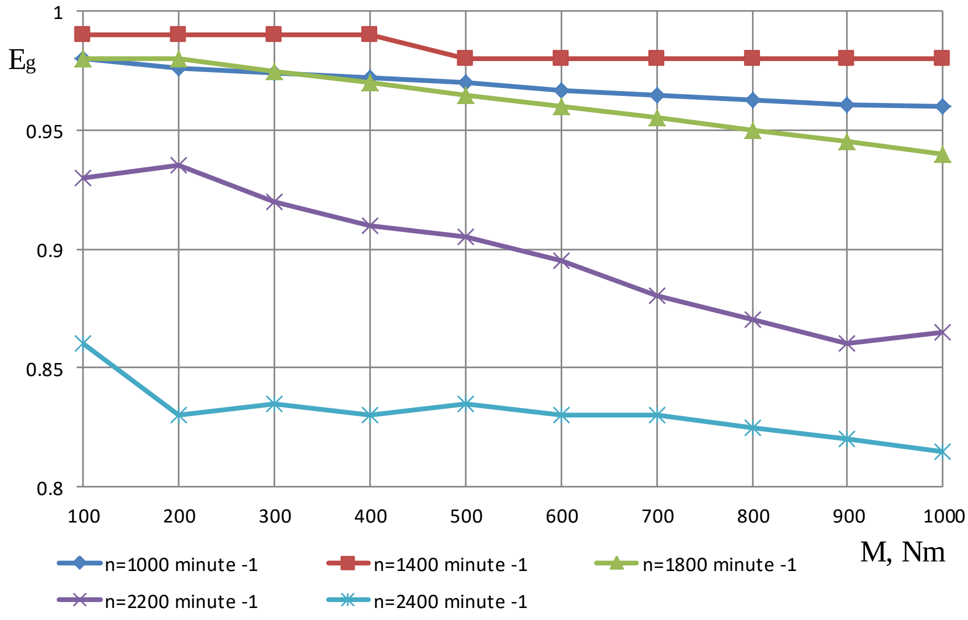

Fig. 6. Engine load characteristics by parameter E_g (*M*)

For further analysis of the lubrication process in KamAZ engines, the values of rotational speed and torque were recalculated into relative values according to the formulas:

$$n_0 = 100(n_x / n_{nom}), \tag{7}$$

$$M_0 = 100(M_x / M_{nom}), \tag{8}$$

where M_0 and n_0 are the relative values of torque and crankshaft speed, %: n_{nom} =2600 min $^{-1}$ and $M_{ном}$ =1000 Nm are the maximum absolute values of the indicators.

The experimental data $E_g(M,n)$ were approximated using PC applications, and a polynomial model of the form was obtained:

$$E_g = a + bn_0 + cM_0 + dn_0^2 + eM_0^2 + fn_0M_0, \tag{9}$$

where *a*, *b*, *c*, *d*, *e*, *f* – the coefficients of the model, the values of which are: *a* = 0.6966; *b* = 0.01107; *c* = 0.0004430; *d* = -0.0001012; *e* = -2.2734 10 $^{-6}$; *f* = -9.3841 10 $^{-6}$.

The coefficient of determination of the model was r^2 = 0.974, the standard error was 0.0104, which indicates a sufficient quality of the approximation.

Using model (9), the multifactorial characteristics of the engine were built according to the parameter of the integral degree of existence of the lubricating layer E_g in the crankshaft bearings (fig. 7 and 8).

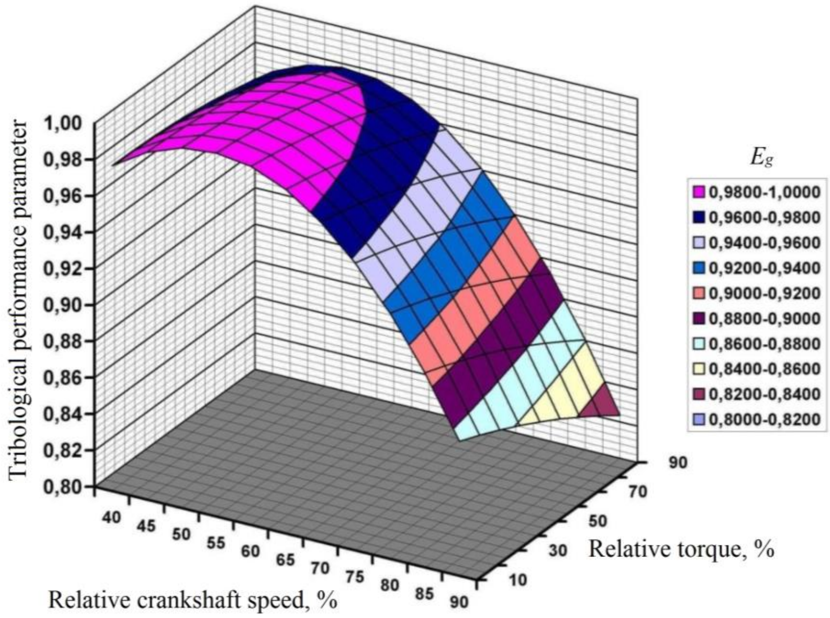

Fig. 7. Graphical interpretation of equation (9) by parameter E_g in the field of load-speed mode of engines of the KamAZ series

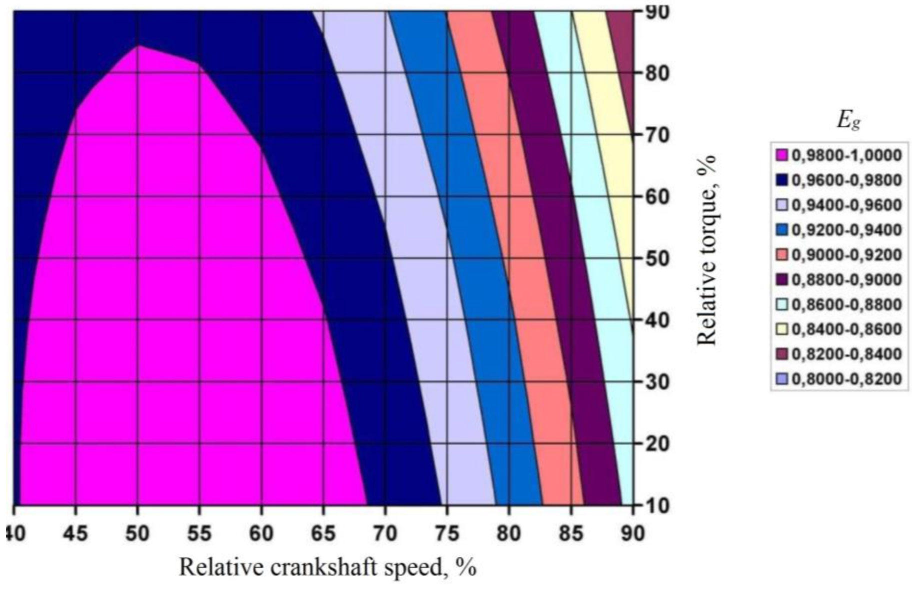

Fig. 8. Areas of parameter levels E_g in the field of the load-speed mode of the KamAZ engine

The analysis of the multifactorial characteristics of the engine by the parameter E_g makes it possible to determine the areas of the load-speed mode of operation, in which the lubrication mode of the crankshaft bearings with different levels of the parameter values is provided E_g. It can be seen that the area of the load-speed mode with a high level of parameter values $E_g \geq 0{,}98$ is: for the crankshaft speed from 40% to 65% and torque from 10% to 50%; crankshaft speed from 45% ... 60% and torque from 50% to 80%.

Model (9) and multifactorial characteristics give an idea of the suitability of the engine to the operating mode in terms of the parameter of the integral degree of existence of the lubricating layer E_g in the crankshaft bearings.

Of practical interest is the assessment of the suitability of the engine to the operating mode in terms of the integral wear resistance W_{I} of the crankshaft bearings. The multifactor characteristic of the engine (fig. 9) shows the distribution of parameter values W_{I} in the areas of the load-speed mode.

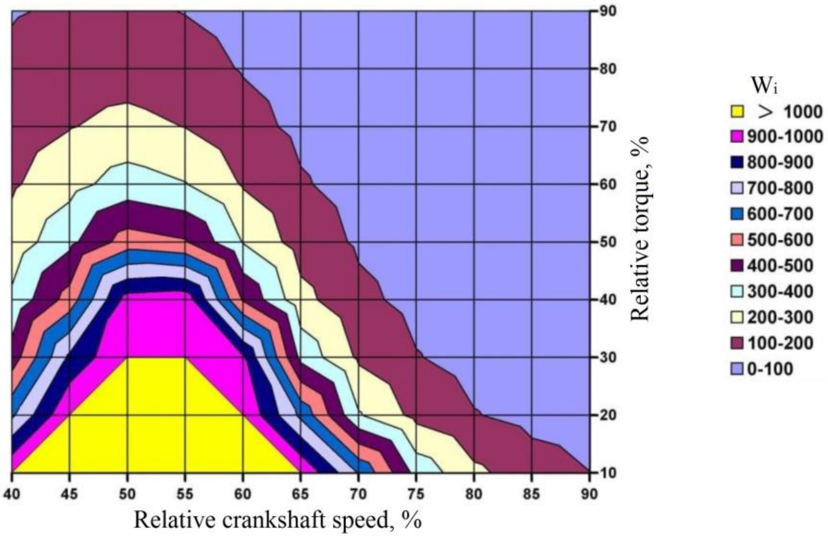

Fig. 9. Multifactorial characteristic of the engine by parameter W_{I} in the areas of load-speed mode

The resulting graphical display shows that: high wear resistance of the crankshaft bearings is $W_{\mathrm{I}} > 1000$ provided in the region of relative speed from 45% to 60% and relative torque from 10% to 30%. As the load-speed mode expands, the relative torque equal to 30% or more and the relative speed of 65% or more reduce the wear resistance of bearings. The speed range of 45% to 60% corresponds to an average piston speed in the range of 4.7 to 6.2 m/s. These values are close to the speed range of 5...7 m/s, which provide "wear-free" speeds of the crankshaft of the engine.

Conclusions

1. A technique has been developed for the experimental study of the lubrication process in the bearings of the crankshaft of an automobile engine during bench tests.

2. The regularities of the parameter of the integral degree of existence of the lubricating layer in the crankshaft bearings from the load-speed mode of operation of the KamAZ-740.14-300 engine have been established, which made it possible to find the regularities of the wear resistance index of the crankshaft bearings also from the load-speed mode of the engine.

3. It was revealed that the maximum wear resistance of the bearings is provided in the region of the relative value of the crankshaft speed from 45% to 60% and the relative value of the torque from 10% to 30%; and it is on average 20...25 times higher compared to wear resistance in other modes; as the load-speed mode expands from a relative torque of 30% and from a relative crankshaft speed of 65%, the wear resistance of the bearings decreases sharply.

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Аулін В.В., Лисенко С.В., Гриньків А.В., Ляшук О.Л., Гупка А.Б., Лівіцький О.М. Параметри мастильного процесу при експлуатаційному зношуванні підшипників колінчастого валу автомобільних двигунів

У статті розглядається динаміка мастильного процесу при експлуатаційному зношуванні підшипників колінчастого валу автомобільних двигунів. Проаналізовано всі режими змащення та процеси зношування. Основна увага приділена режимам змащення підшипників колінчастого валу, що встановився і не встановився. Характер динаміки зміни режимів змащення обґрунтовується через експлуатаційні трибологічні параметри, які характеризують інтегральний ступінь існування мастильного шару. Запропоновано модель зв'язку цих параметрів з цілого ряду факторів, а також обґрунтовано характер їх змін із зміною частоти обертання колінчастого валу двигуна автомобіля.

Наведено результати стендових випробувань дизелів серії КамАЗ, швидкісних та навантажувальних характеристик двигуна за трибологічними експлуатаційними параметрами. Дано графічну інтерпретацію трибологічного параметра в полі навантажувально-швидкісного режиму двигунів серії КамАЗ.

Ключові слова: автомобільний двигун, підшипники колінчастого валу, режим змащення, трибологічний параметр, швидкісні та навантажувальні характеристики