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Fundamentals of friction and wear

PREDICTING LIFETIME OF INTERNAL COMBUSTION ENGINE CRANKSHAFT JOURNAL BEARINGS AT THE DESIGN STAGE

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ABSTRACT

The purpose of the article is to predict the service life time of the connecting rod bearing of an internal combustion engine. The technique of calculation is based on lubrication hydrodynamic theory and the molecular-mechanical theory of friction and wear fatigue theory. It includes several stages.The first stepis to set the load conditions of the engine. The second one is the definition of the forces acting on the connecting rod bearing. The third step is the calculation of the hydromechanical characteristics of the bearing such as the minimum of thickness of the lubricating layer, the maximal hydrodynamic pressure, the loss of power for friction, and others. The fourth step is the definition of the duration of the zone, where liquid friction is violated. The fifth step is the calculation of bearing wear for 720 degrees of crankshaft rotation (loading cycle) for each mode. Then the wear is recalculated taking into account the duration of each mode of operation. The characteristics of the crank journal and bearing are taken into account. The diagram of the wear of the connecting rod bearing is based on the simulation results. This work has been carried out within financial support of Russian Foundation for Basic Research (project № 16-08-01020\16).

Keywords: service life, connecting rod bearing, wear.

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AIMS AND BACKGROUND

The prediction of the resource of friction units for wear and assessment of their performance at the stage of design and development of machines is an actual task. Statistical and computational methods are used to assess the resource of machine parts for wear. A statistical method is based on the processing of information on the wear of machine parts under operating conditions. This method allows to take into account such important factors as the feature of operation and repair of machinery. Calculation methods to assess resource allocation require a study of the friction joint of the kinematic chain, in which it operates. It is impossible to consider the impact of the entire system for the friction pair. The statistical method allows to automatically take into account the mutual coupling between the wear parts in the machine, but it requires a large amount of statistical data. These data allow to reasonably identify the factors that affect the wear and to improve calculation methods. Thus, computational and statistical methods complement each other in predicting the wear and life time of machine parts[1].

Crank mechanism is one of the basic units of internal combustion engines (ICE), which determine their reliability. Main and connecting rod bearings of the crankshaft engine are the most responsible tribo-units in crank mechanism. The assessment of the crankshaft bearings of internal combustion engines operating life time is an actual task during a design stage. However, it is very difficult to describe the process of wear, taking into account a large number of factors such as physical and chemical characteristics of materials of the bearing, temperature conditions, properties of a lubricant, nature of loading and others. The crankshaft bearings operate in the different modes of friction, from boundary (during a starting period) to liquid (during the main work), that also complicates forecasting of their operating life time.

The lifetime of the crankshaft bearings can be determined by the results of calculations with the known techniques[2-4], experimentally[5], and also by the experimental and theoretical methods[6,7].

In relation to bearings of the ICE crankshaft, it is possible to distinguish between the following known methods of definition of a lifetime[6]: the fatigue theory of wear according to I.V. Kragelsky; method of the company IBM; calculation of wear of interfaces by A.S. Pronikov; wear from view point of the thermofluctuation theory of strength by S.N. Zhurkov and S.B. Ratner; the power theory of wear by Flayshera; the structural and power theory of wear by JI. To I. Pogodayev; a wear assessment method according to statistical data.

The exhaustive review[6] of the most known experimental methods of definition of a resource of tribounit is submitted: a measuring method, a weight method, the analysis of wear particles in grease, a method of radioactive isotopes, the determination of wear from a profilogram of surfaces, a method of artificial bases.

Experimental and theoretical methods of definition of a resource are based on establishment of a contact zone, taking into account geometry of the bearing, loading, and elastic properties of bearing materials. Thus, wear intensity of materials is defined mainly experimentally[3]. The model of wear of plain bearings of bent shafts of ICE is proposed[6]. The contact zone is defined from the calculation of the bearing characteristics, on the basis of the hydrodynamic theory of lubrication. The wear intensities of shaft and bearing are determined according to the molecular and mechanical theory of friction and the fatigue theory of wear (by I.V. Kragelsky)[7]. Further speeds of wear of surfaces, thickness of a worn-out layer of a shaft and the liner, and a bearing resource pay off. Application of this model to determination of wear of the connecting rod and main bearings of the gasoline engine showed good qualitative and quantitatively coincidence with the results of the experiment. However, this technique does not provide the definition of the wear chart for bearing.

In Refs 8 a technique of creation of the theoretical wear chart (theoretical wear diagrams) of main crankshaft bearings is represented. Contact parameters of the interface between shaft and bearing is chosen as in Refs 6, on the basis of the solution of a contact task of the theory of elasticity on internal compression of two cylinders of close radii[7]. The flowchart of algorithm for creation of the bearing wear chart, and also results of construction are submitted. An approach offered in Refs 8 can be used for the design of crankshaft bearings. However, in this work it is not represented the comparison of theoretical wear lines to a lifetime of the engine bearings.

Zakharov[9] has created a model of the journal bearing of the ICE on a computer using the method of imitation modeling. This model reproduces the engine operating conditions, the working conditions of crankshaft bearing in these modes and other factors. However, this method is difficult to apply in the design phase of the internal combustion engine, because it requires a large amount of source data.

This work briefly describes the methodology of calculating lifetime of the connecting rod bearing of an ICE, which can be used at an early stage of design engine. The methodology is based on the works in Refs 6 and 11. We performed a lifetime calculation of the connecting rod bearing and compared the calculated values of the lifetime of the connecting rod bearing with the experimental data of other authors.

RESULTS AND DISCUSSION

Load conditions of the engine. In order to determine the lifetime of the connecting rod bearing of a crankshaft of the ICE it is necessary to know the mode of engine operation and the duration of operation in this mode. The modes of operation, specified in the standards for testing engines can be used. For example, the standard in Refs 12 requires monitoring of indicators of reliability and stability of parameters of diesel engines under normal test duration 800 hours of repetitive cycles. The operation of the engine and duration are shown in Table 1.

Table 1. Modes of operation of the diesel engines when testing for reliability

Mode	The duration of one cycle, min	
	diesel tractor	diesel agricultural harvester or other self-propelled agricultural machine
1. The maximal frequency of rotation of idling	10	10
2. The maximum of torque	10	–
3. Nominal frequency of rotation and position controls of the speed controller corresponding to the total fuel supply	210	10
4. 90% of power according to claim 3 of this table (when working for the regulatory branch regulatory characteristics)	–	210

The standard[13] provides tests for proper duration from 250 to 1000 hours, depending on engine size. In this mode the full flow of fuel at full engine speed is essential. We propose to calculate the lifetime of the connecting rod bearing at the design stage of the engine: this is the regime corresponding to the test reliability.

Defenition of the forces acting on the connecting rod bearing. We determined the gas loads on the basis of the calculation of the engine process using the law burnout of I.I. Vibbe[14]. The task dynamics is described in detail in Refs 15 and it is used to calculate the forces acting on the connecting rod bearing. The coordinate system used in the calculation is shown in Fig. 1. The forces acting on the considered connecting rod bearing V8 engine is represented in Fig. 2.

The connecting rod bearing parameters used in this work are given in Table 2.

Calculation of the hydro-mechanical characteristics of the bearing. Designs of the crankshaft bearings can be estimated by comparison of the parameters of settlement trajectories, in which under the influence of the enclosed loadings, the centers of pins move and a set of hydro-mechanical characteristics (HMC) to which the following belongs[11]:

– minimum of bearing oil film thickness $h_{\min}$ [m], and average $\bar{h}_{\min}$ [m] value for a loading cycle of the engine;

– the greatest and average value $p_{\max}$, $p^*_{\max}$ [MPa] of the hydrodynamic pressure in oil film for a loading cycle;

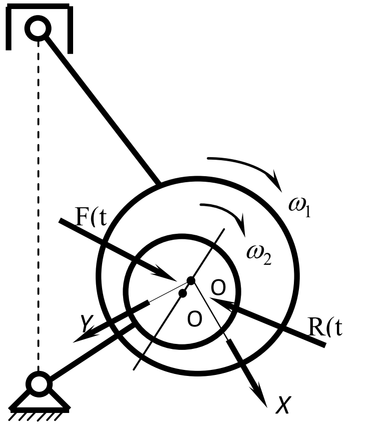

Fig. 1. Coordinate system used in the theoretical calculations for the connecting rod bearing

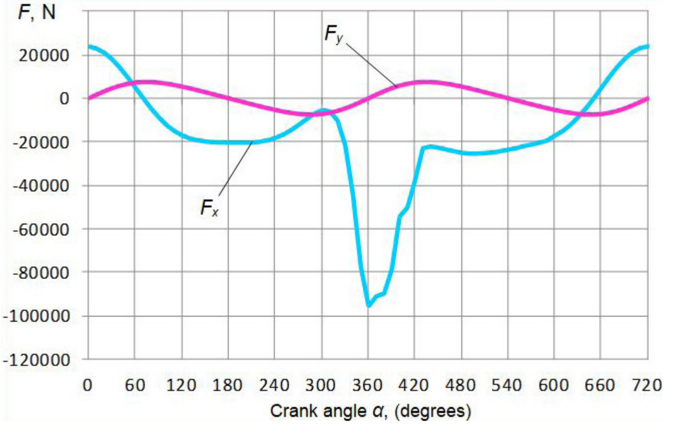

Fig. 2. Calculated connecting rod bearing loads

– friction losses $\overline{N}$ [W] and oil consumption in butts of the bearing $\overline{Q}$ [kg/s] and temperature $\overline{T}$ [°C] in oil film for a cycle of the engine.

Calculation of the hydro-mechanical characteristics of the crankshaft bearings is based on the solution of the three interconnected tasks.

1. Calculation of the dynamics of mobile elements of knot of friction.
2. Determination of the forces of hydrodynamic pressure in a lubricant film.
3. Assessment of the thermal condition of bearing.

The problem in calculation of the dynamics of the crankshaft bearing consists in creation of a trajectory of the movement of the center of mass of each mobile

Table 2. Connecting rod bearing parameters

Dimensions	Connecting rod
Diameter, D (m)	0,08588
Radius, R (m)	0,0425
Width, B (m)	0,029
Radial Clearance, C (m)	$44 \cdot 10^{-6}$
Temperature lubrication, T (Celsius degree)	90
Oil SAE Grade	10W-40
Angular frequency of rotation of the crankshaft, ω_0 (s^{-1})	230,38

element (for example a rod journal) under the influence of external periodic loading. The trajectory is constructed based on the coordinates, received as a result of the equations of movement. Integration of the equations of movement is carried out by method of formulas of differentiation back, described in works of Prokopyev[11] et al.

The field of hydrodynamic pressure, necessary for the calculation of reaction of a lubricant film, is defined by integration of the Reynolds equation under the boundary conditions of Swift-Shtiber, with existence of sources of lubricant supply (bores, flutes). Thus, the rheological properties of grease are taken into account[16]. Reynolds equation is solved by means of the adaptive multigrid algorithm, developed by the authors[16], which allows receiving the distribution of pressure in a lubricant layer within 10^{-4}.

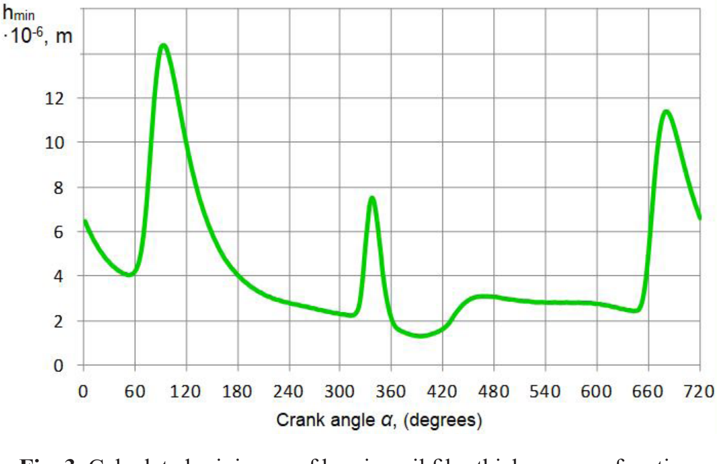

Fig. 3. Calculated minimum of bearing oil film thickness as a function of crank angle for the connecting rod bearing

For the assessment of a thermal condition of the bearing, the isothermal approach, based on the solution of the equation of thermal balance, is used. This equation reflects equality of average values of the warmth dissipated in a lubricant film, and the warmth, which is taken away by the lubricant escaping in butts.

Figure 3 shows calculated minimum of bearing oil film thickness as a function of crank angle for engine loads of 100 percent of the wide-open throttle load.

Definition of the duration of the zone, where liquid friction is violated. Usually the following are used as criteria of bearings operability[10,11,17,18]: minimal permissible lubricant film thickness $h_{\text{lim,cr}}$ and temperature in a working zone of the bearing, the relative extent of areas $\alpha_{h_{\text{lim,cr}}}$, total for a loading cycle, where the value of minimum of lubricant film thickness $h_{\min}$ is less than critical $h_{\text{lim,cr}}$.

Value of the $h_{\text{lim,cr}}$ is obtained from the condition of providing the hydrodynamic mode of friction in the bearing, and it has to be bigger than the average sum of roughnesses of the interacting surfaces R_{z1}, R_{z2}:

$$h_{\text{lim,cr}} > R_{z_1} + R_{z_2} \, . \tag{1}$$

Identification of the modes of friction is carried out using the conditions:

$h_{\min} < h_{\text{lim,cr}}$ – boundary mode;

$h_{\min} = h_{\text{lim,cr}}$ – commixed mode;

$h_{\min} > h_{\text{lim,cr}}$ – hydrodynamic (liquid) mode.

If the calculated minimal bearing oil film thickness in the bearing is less than or equal to the minimum of permissible lubricant film thickness (Fig. 4), the

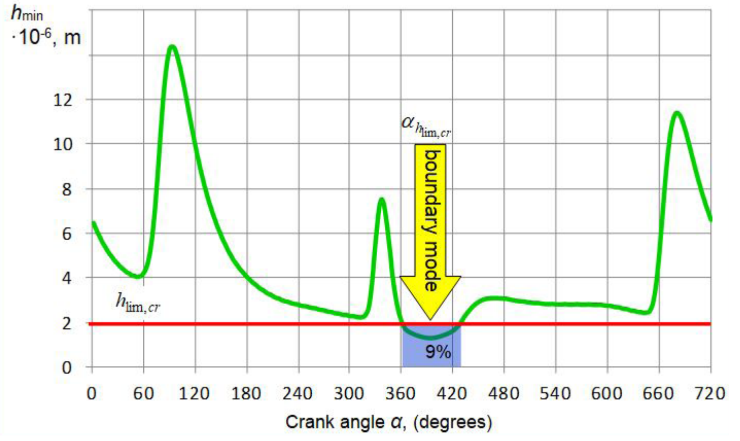

Fig. 4. Definition of the duration of the zone where liquid friction is violated

hydrodynamic friction regime at the moment is broken and the contact interaction of the rod journal and bearing occurs. In our case minimal permissible lubricant film thickness[10] $h_{lim,cr}$ is $1{,}9 \cdot 10^{-6}$ m.

It is known that short-term transition to the area of commixed lubrication is not dangerous to the bearing, if duration of contact of a crankshaft journal with a surface of the liner is small (no more than 20 % of time of a cycle)[19]. Taking into account that the clear boundary between semi-liquid and boundary the modes of friction are very conditional, we will consider the bearing efficiency in the case, when the extent does not exceed 20 %. In case of excess of this value, the probability of emergence of a burr in the bearing sharply increases.

Calculation of wear and lifetime connecting rod bearing. According to GOST R 27.002-2009 [20], lifetime (or operating life) of the crankshaft bearing of engine is the time interval (in hours or kilometers run) during which the bearing functions in the limiting state. Limiting state of the crankshaft bearing is a condition, under which its continued operation is unacceptable or impractical for reasons of danger, economic or environmental. As a rule, the limiting state of crankshaft bearings is characterized by an increase in clearances above acceptable limits as a result of wear. This often leads to a knock on the crank mechanism and reduces the pressure in the engine lubrication system.

The model of the journal-bearing contact is based on Refs 6 and 8. We propose to build wear diagram only in the boundary mode of friction, when $h_{min} < h_{lim,cr}$ unlike Refs 8.

The conditional wear depth $\delta(\alpha, \beta)$ [m] in each step of calculation[8] is defined as follows:

$$\delta(\alpha, \beta) = I_h(\alpha, \beta) \cdot \Delta s \,. \quad (2)$$

Here α is the current value of a crank angle; β is a current angular coordinate that changes between β_c and β_c; β_c [rad] is a contact half-angle; Δs is the path friction, which can be found as[21]

$$\Delta s = R_2 \cdot dt \cdot \left| -\omega_{1_i} + \omega_{1_{i-1}} \right| , \quad (3)$$

where dt the duration of the current step according to the crankshaft angle, $\omega_{1_i}, \omega_{1_{i-1}}$ absolute angular velocity of rotation of the bearing on the current and the previous step according to the crankshaft angle, respectively.

Linear wear rate $I_h(\alpha, \beta)$ is calculated according to the formulas described in Refs 6 and 7.

The total increase of the wear Δh_n in each point of the contact area is found as follows:

$$\Delta h_n \leftarrow \Delta h_n + \delta(\alpha, \beta). \qquad (4)$$

Wear is summed at each calculation step of the loading cycle bearing. The maximum of wear value $\Delta h_{w,\max}$ and the angular coordinate is then determined.

The lifetime bearing was defined as follows:

$$R_h = \frac{\Delta h_{\lim}}{\upsilon_{w,\max}}. \qquad (5)$$

The limit wear $\Delta h_{\lim}$ is an important parameter and determines the life of the bearing.

The limit wear of a bearing is defined as:

$$\Delta h_{\lim} = C_{\lim} - C. \qquad (6)$$

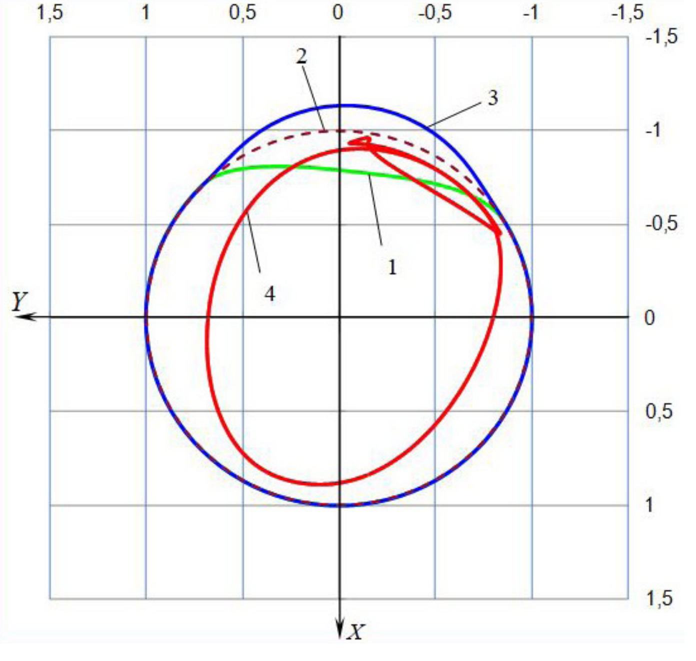

Fig. 5. Trajectory of the movement of the center of a rod journal in the bearing: 1 – theoretical wear diagram of the journal; 2 – unit circle; 3 – theoretical wear diagram of the bearing; 4 – trajectory of the journal

Where $C_{\lim}$ is the maximum of allowable radial clearance in the bearing. The choice of values is justified in Refs 22. In our case, maximum of allowable radial clearance is equal to $200 \cdot 10^{-6}$ m.

The maximal speed of wear of a bearing is defined as

$$\upsilon_{w,\max} = \frac{\Delta h_{w,\max}}{T_{\text{acceleration cycle}}} 3600. \quad (7)$$

Diagram of the wear surfaces of the insert and the shaft are constructed on the basis of the results of calculation of the wear distribution on the angular coordinate of the bearing[8]. Figure 5 shows the theoretical wear diagram and a trajectory of the movement of the center of a rod journal in the bearing for engine loads of 100 percent of the wide-open throttle load. The results of the calculation of the lifetime of the connecting rod bearing is represented in table 3.

Table 3. Results of calculation

Results of calculation			
$h_{\min} \cdot 10^{-6}$, m	$\upsilon_{w,\max} \cdot 10^{-6}$ [m/h]	$\alpha_{h\lim,cr}$ [%]	R_h[hour]
1,318	0,1250	9,0	1248

CONCLUSION

The obtained value of the lifetime of the connecting rod bearing is in good agreement with known experimental data. Refs 22 represents the results of motor tests of the V8 engine, for which we performed the calculation of the lifetime of the connecting rod bearing. Refs 23 stated that after working V8 engine for 1050 hours as the connecting rod bearing, after the test is satisfactory, except for the 6th bearing on which tuff was discovered.

This methodology allows for early design stage of the engine to predict the reliability of connecting rod bearings.

The further development of the method should be done in the direction of the surface of the calculation of the fatigue life time of the liner, because it is one of the three main types of wear of crankshaft bearings[5].

In the next stage of the method it is necessary to improve the methodology of calculation of hydro-bearing characteristics in the direction of accounting for changes in the geometry of the journal and bearing as a result of wear.This section should be formatted as main headings and text, and numbered.

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