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METHODS OF ASSESSING THE RESOURCE BEARING CRANKSHAFT OF INTERNAL COMBUSTION ENGINE BASED ON THE CALCULATION OF HYDRO-MECHANICAL CHARACTERISTICS

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Abstract: *The purpose of the article is to develop a tool to assess the theoretical resource crankshaft bearings of internal combustion engine. As a result, have developed two methods for evaluating the theoretical resource crankshaft bearings on the basis of the calculation of hydromechanical characteristics of bearings: the minimum film thickness and extent of the zone boundary friction. Under the theoretical resource crankshaft bearing understood during his work to increase the radial clearance in the area of potential exposure (boundary friction) over the limit. The first technique is based on bearing life depending on the ratio between the minimum film thickness and its maximum allowable value. The second technique is based on the molecular-mechanical theory of friction and wear fatigue theory. Thus, these techniques may be used to estimate resource crankshaft journal bearings at the design stage and finishing. However, some parameters of mathematical models to be determined from the experimental test. The use of molecular-mechanical theory of friction and wear fatigue theory takes into account the influence of the physical and mechanical properties of a material bearing on his life.*

Keywords: *journal bearing, heavy-loaded friction units, hydro-mechanical characteristics, theoretical resource, wear.*

1. INTRODUCTION

Crank mechanism is one of the basic units of internal combustion (IC) engines, which determine their reliability. Main and connecting rod bearings of the crankshaft engine are the most responsible tribounits crank mechanism.

According to GOST R 27.002-2009 [1] resource (or operating life) bearing crankshaft of the engine is the time interval (in hours or kilometres run), in which the bearing functions to the limiting state. Limiting state of bearing crankshaft is a condition under which its continued operation is unacceptable or impractical for reasons of danger, economic or environmental. As a rule, the limit 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 reduce the pressure in the engine lubrication system.

The assessment of operating life bearings of the crankshaft at a design stage of internal combustion engines is an actual task. However, it is very difficult to describe 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. Bearings of the crankshaft work 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.

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

At the heart of many method of calculation of a resource of knots of friction and, in particular, bearings of the crankshaft the condition of working capacity lies [2-4]

$$[\Delta h] < \Delta h_{\lim} , \qquad (1)$$

where [Δh] is allowable wear [micron], $\Delta h_{\lim}$ is limiting wear [micron].

The condition of durability of the bearing [2] has an appearance

$$t = \frac{[\Delta h]}{\gamma_1 + \gamma_2} \geq T , \qquad (2)$$

where t [h] is settlement life cycle of a tribounit on achievement of allowable wear [hours], γ_1, γ_2 [μm/h] are speeds wear of elements of a tribounit, T of is operating life of bearing.

Wear speed in turn is defined by multiplication of I_{h} intensity wear on υ_{s} slippage speed in contact piece [3].

Justification and determination [Δh] and $\Delta h_{\lim}$ the bearing of the crankshaft is important.

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

In work [6] the exhaustive review of the most known experimental methods of definition of a resource of tribounit is also submitted: a measuring method, a weight method, the analysis of particles of wear in grease, a method of radioactive isotopes, determination of wear removal of 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, elastic properties of materials of the bearing. Thus intensity of wear of materials is defined mainly experimentally [3]. In work [6] the model of wear of plain bearers of bent shafts of DVS is offered. The contact zone is defined from calculation of characteristics of the bearing on the basis of the hydrodynamic theory of lubricant. Intensity of wear of a pin of a shaft and the 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 of friction, thickness of a worn-out layer of a pin of a shaft and the liner, and then and a bearing resource pay off. Application of this model to determination of wear the connecting rod and main bearers of the gasoline engine showed good qualitative and quantitatively coincidence to results of experiment. However, this technique doesn't provide definition of the chart of wear of the bearing.

Authors of work [8] presented a technique of creation of the theoretical chart of wear (theoretical wear lines) of main bearers of the crankshaft. Contact parameters of interface a shaft bearing pin also as in [6] decide on the basis of the solution of a contact task of the theory of elasticity on internal compression of two cylinders on close radiuses [7]. In work the flowchart of algorithm for creation of the chart of wear of the bearing, and also results of construction is submitted. Offered in [8] approach it can be used at design of bearings of the crankshaft. However, in work it isn't presented comparisons of theoretical wear lines to a resource of bearings of the engine.

Two techniques of an assessment of a resource of bearings of the crankshaft on the basis of calculation of their hydromechanical characteristics are presented in this article. Work [4] is the basis for the first technique. In the second techniques the approach developed in [8] is used.

2. CRITERIA OF THE ASSESSMENT OF OPERABILITY OF BEARINGS OF THE CRANKSHAFT

Designs of bearings of crankshafts can be estimated comparison of parameters of settlement trajectories in which under the influence of the enclosed loadings the centers of pins, and a set of hydromechanical characteristics (HMC) to which the following belongs move [9]:

- minimum lubricant film thickness h_{min} [micron], and average $\bar{h}_{min}$ [micron] for a loading cycle engine values;
- the greatest and average values of hydrodynamic pressure for a loading cycle in a lubricant film sup p_{max}, p^*_{max} [MPa];
- friction losses $\bar{N}$ [W] and lubricant consumption in butts of the bearing $\bar{Q}$ [kg/s] and temperature $\bar{T}$ [°C] in a lubricant film for a cycle engine.

As criteria of operability of bearings usually use the following [9-12]: minimum permissible lubricant film thickness $h_{\lim,cr}$ and temperature in a working zone of the bearing, the relative extent of areas $\alpha_{h_{\lim,cr}}$, total for a loading cycle, where value of minimum lubricant film thickness h_{min} there is less than critical size $h_{\lim,cr}$.

Size $h_{\lim,cr}$ gets out of a condition of providing the hydrodynamic mode of friction in the bearing and there has to be more average sum of roughnesses of the interacting friction surfaces R_{z1}, R_{z2}:

$$h_{\lim,cr} > R_{z1} + R_{z2}\,. \qquad (3)$$

Identification of the modes of friction is carried out proceeding from conditions:

- $h_{min} < h_{\lim,cr}$ – boundary mode;
- $h_{min} = h_{\lim,cr}$ – commixed mode;
- $h_{min} > h_{\lim,cr}$ – hydrodynamic (liquid) mode.

It is known that short-term transition to area of the commixed lubricant isn't 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) [13]. Taking into account that the clear boundary between semi-liquid and boundary the modes of friction are very conditional, we will consider the bearing efficient in case extent doesn't exceed 20 %. In case of excess of this value the probability of emergence of a burr in the bearing sharply increases.

3. DESCRIPTION OF THE METHOD OF CALCULATION OF HYDROMECHANICAL CHARACTERISTICS OF BEARINGS OF THE CRANKSHAFT

Calculation of hydromechanical characteristics of bearings of the crankshaft is based on the solution of three interconnected tasks.

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

The problem of calculation of dynamics of the crankshaft bearing consists in creation of a trajectory of the movement of the center of mass of each mobile element (for example a rod journal) under the influence of external periodic loading. The trajectory is under construction on the coordinates received as a result of the solution of the equations of the movement. Integration of the equations of the movement is carried out by method of formulas of differentiation back described in works of Prokopyev et al. [9].

The field of hydrodynamic pressure necessary for calculation of reaction of a lubricant film is defined by integration of the equation of Reynolds under boundary conditions of Swift-Shtibera with existence on surfaces of friction of sources of supply of lubricant (bores, flutes). Thus rheological properties of grease are taken into account [14]. Reynolds's equation is solved by means of the adaptive multigrid algorithm developed by authors [14] which allows to receive distribution of pressure in a lubricant layer to within 10^{-4}.

For an 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 averages for a cycle of loading of values of the warmth dissipated in a lubricant film, and the warmth which is taken away by the lubricant escaping in butts.

4. DEFINITION OF THE RESOURCE OF CRANKSHAFT BEARINGS

The first method of definition of a resource of bearings of the crankshaft is based on use Rumb's dependences [9]. The resource of the bearing of the crankshaft is determined by calculated minimum values of thickness of a lubricant layer on a formula

$$R_h = (\Delta h_{\lim}/\gamma)^{1/\beta_l}, \quad (4)$$

where γ [m/s] is the speed of wear, β_l is the index.

Limiting wear reasonably $\Delta h_{\lim}$ is accepted in each separate case, for example, by the technique described in [14]. In our case $\Delta h_{\lim} = 30 \cdot 10^{-6}$ m and $\gamma = 10^{-13}$ m/s [3].

The index β_l is the determined size and its value can be set in function from the relation of minimum permissible lubricant film thickness $h_{\lim,cr}$ to $h_{\min}$ by:

$$\beta_l = 1 + (h_{\min}/h_{\lim,cr})^{n_l}. \quad (5)$$

In our case for the connecting-rod bearing crankshaft of diesel engine $h_{\lim,cr} = 1.9 \cdot 10^{-6}$ m [9].

Value of index n_l in a formula (5) approximately is defined by the solution of the return task. That is, if to set a bearing resource with concrete parameters (in our case we will accept 10000 hours) and to know distribution of minimum lubricant film thickness $h_{\min}$ for a cycle, it is possible to define at first index β_l then and required size n_l.

The index n_l has the range of values from – 2.0 to – 2.5 at $h_{\min}$ = 2.5…3.5 [micron] and $h_{\lim,cr} = 1.9 \cdot 10^{-6}$ m.

Results of an assessment of a theoretical resource of connecting rod bearing of the diesel engine with a diameter of cylinder of 130 mm and a piston stroke of 150 mm are presented in Table 1.

Table 1. The results of assessment of the resource connecting rod bearing diesel engine

Bearing width B [mm]	Results of calculation			
	$h_{\min}$ [micron]	β_l	$\alpha_{h_{\lim,cr}}$ [%]	R_h [hour]
Diameter bearing D = 85 mm				
33	5.317	1.1276	17.083	3449.9
38	6.107	1.0967	14.861	5466.1
43	6.822	1.0775	13.055	7376.0
48	7.497	1.0642	11.527	9138.5
D = 95 mm				
33	5.739	1.1096	14.861	4502.2
38	6.606	1.0827	12.777	6799.4
43	7.350	1.0668	10.883	8761.7
48	8.090	1.0551	8.888	10604.7
D = 105 mm				
33	6.103	1.0969	13.194	5455.5
38	7.012	1.0734	10.972	7879.5
43	7.861	1.0584	8.611	10049.4
48	8.597	1.0488	5.416	11779.8

The second method is based on works [6,8]. Also calculation of hydromechanical characteristics of the bearing of the crankshaft for a technique which is described above is the cornerstone of a method. By results of calculation minimum lubricant film thickness $\alpha_{h_{\lim,cr}}$ [%] is defined, that is the cycle duration where the hydrodynamic mode of friction is broken.

Unlike work [8] chart of wear is under construction only if $h_{\min} < h_{\lim,cr}$.

The conditional wear depth $\delta(\varphi, \beta)$ [m] on each step of calculation [8] is offered to be defined as

$$\delta(\varphi,\beta) = I_h(\varphi,\beta) \cdot 2 \cdot \beta_C \cdot R_b. \quad (6)$$

Here φ [deg] is current value of a crank angle; β_C [rad] is a contact half-angle; β is a current angular coordinate that changes between $-\beta_C$ and β_C; R_b [m] is the bearing radius.

Linear intensity of wear is offered to be defined as

$$I_h(\varphi,\beta) = k \cdot p(\varphi,\beta)^m . \qquad (7)$$

Here $p(\varphi, \beta)$ [Pa] is contact pressure; k = $7 \cdot 10^{-12}$ Pa^{-1} is a constant; m = 0.0511 is an exponent.

Values $k = 7 \cdot 10^{-12}$ Pa^{-1} and m = 0.0511 are also received experimentally for contact piece like “roller-pad”.

Results of an assessment on the second technique of a resource of the connecting rod bearing crankshaft are presented in Table 2. A theoretical wear diagram and a trajectory of the movement of the center of a rod journal in the bearing are submitted in Figure 1.

Table 2. The results of assessment of the resource connecting rod bearing diesel engine

Bearing width B [mm]	Results of calculation			
	h_{min} [micron]	γ [micron/h]	$\alpha_{h_{lim,cr}}$ [%]	R_h [hour]
D = 95 mm				
33	5.739	0.0297	14.8	1008.1
38	6.606	0.0285	12.7	1049.4
43	7.350	0.0274	10.8	1094.2
48	8.090	0.0252	8.8	1188.9

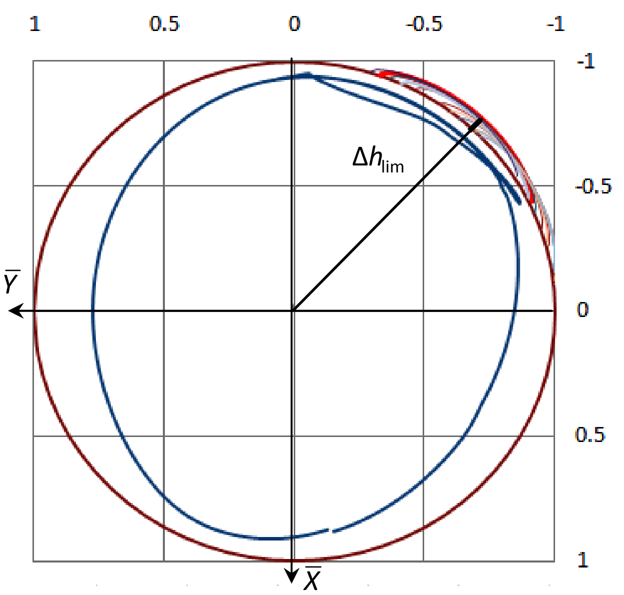

Figure 1. Trajectory of the movement of the center of a rod journal in the bearing

The results of calculation characterize resource connecting rod bearing with the engine at steady-state maximum torque. The engine operates in this mode only part-time in actual use. Thus, in order to obtain a more accurate value of the theoretical resource connecting rod bearing is advisable to take into account the nature of the loading of the engine under real operating conditions.

5. CONCLUSION

Results of calculations for both techniques need further comparison to experimental data. In each separate case justification of sizes limiting wear, the law of change of intensity wear, wear speed is required.

Though already now it is possible to tell that techniques can be used for an assessment of a resource of bearings of the crankshaft at a design stage.

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