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RESEARCH

# Methods of Assessing the Resource of the Crankshaft Bearing of Internal Combustion Engine Based on the Calculation of Hydro-Mechanical Characteristics

I.G. Levanov[a], E.A. Zadorozhnaya[a], A.L. Dudnikov[a]

[a] *Department of Motor Transport, South Ural State University, Chelyabinsk, Russia.*

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Corresponding author:

*Igor Levanov*
*Department of Motor Transport,*
*South Ural State University,*
*Chelyabinsk,*
*Russia.*
*E-mail: il74-lig@rambler.ru*

A B S T R A C T

*The purpose of the article is to develop a tool to assess the theoretical resource crankshaft bearings of internal combustion engine. As a result, two methods for evaluating of the theoretical resource crankshaft bearings have been developed on the basis of the calculation of hydro-mechanical characteristics of bearings: the minimum film thickness and the extent of the zone of boundary friction. Under the theoretical resource of crankshaft bearing it is understood that during his work an increase of the radial clearance in the area of potential exposure (boundary friction) is over the limit. The first technique is based on the bearing life dependence 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 the resource of the crankshaft journal bearings at the design and finishing stage. However, some parameters of mathematical models have 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 bearing material on his life.*



## 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 in crank mechanism.

According to GOST R 27.002-2009 [1], resource (or operating life) of the crankshaft bearing of engine is the time interval (in hours or kilometers run) in 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 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 the crankshaft bearings of internal combustion engines operating life is an actual task at 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 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 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].

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

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

where: $[\Delta h]$ is the allowable wear [μm], and $\Delta h_{\lim}$ [μm] is limiting wear.

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

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

where: $t$ [h] is the settlement life cycle of a tribounit on achievement of allowable wear, $\gamma_1$, $\gamma_2$ [μm/h] are the wear speeds of the tribounit elements, and $T$ is the operating life of bearing.

Wear speed is defined by multiplication of the wear intensity $I_h$ with the slippage speed in contact piece $\upsilon_s$ [3].

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

In relation to bearings of the ICE 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 the company IBM; calculation of wear of interfaces by A.S. Pronikov; wear from positions 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.

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 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]. In work [6], the model of wear of plain bearings of bent shafts of DVS is offered. 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 to the results of the experiment. However, this technique does not provide the definition of the wear chart for bearing.

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

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

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

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 [9]:

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

Usually the following are used as criteria of bearings operability [9-12]: minimum 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 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 has to be bigger than the average sum of roughnesses of the interacting surfaces $R_{z1}$, $R_{z2}$:

$$h_{\text{lim},cr} > R_{z_1} + R_{z_2}\,. \tag{3}$$

Identification of the modes of friction is carried out from 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.

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) [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 efficiency in 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.

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

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 layer.
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 element (for example a rod journal) under the influence of external periodic loading. The trajectory is constructed from 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 et al. [9].

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-Shtibera, with existence of sources of lubricant supply (bores, flutes). Thus, the rheological properties of grease are taken into account [14]. Reynolds equation is solved by means of the adaptive multigrid algorithm, developed by the authors [14], which allows receiving the distribution of pressure in a lubricant layer within $10^{-4}$.

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.

## 4. DEFINITION OF THE RESOURCE OF CRANKSHAFT BEARINGS

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

$$R_h = (\Delta h_{lim} / \gamma)^{1/\beta_1}, \tag{4}$$

where: $\gamma$ [m/s] is the speed of wear and $\beta_1$ is the index.

Limiting wear $\Delta h_{lim}$ is reasonably 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 $\beta_1$ is the determined value and can be set as a function of minimum permissible lubricant film thickness $h_{lim,\,cr}$ and $h_{min}$ by:

$$\beta_1 = 1 + \left(h_{min} / h_{lim,cr}\right)^{n_I}. \tag{5}$$

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

Value of index $n_I$ in a formula (5) is approximately defined by the solution of the return task. That is, if a bearing resource is set with concrete parameters (in our case we will accept 10,000 hours) and with known distribution of minimum lubricant film thickness $h_{min}$ for a cycle, it is possible to define, at first the index $\beta_I$, and then the required size $n_I$.

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

Results of an assessment of a theoretical resource of the connecting rod bearing of 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 for the connecting rod bearing of diesel engine.

| Bearing width $B$, [mm] | Results of calculation | | | |
|---|---|---|---|---|
| | $h_{min}$ [μm] | $\beta_1$ | $\alpha_{h_{lim,cr}}$ [%] | $R_h$ [h] |
| Diameter of the 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]. The calculation of hydro-mechanical characteristics of the crankshaft bearing is the cornerstone of for the technique which is described above. A minimum lubricant film thickness $\alpha_{h_{lim,cr}}$ [%], that is the cycle duration where the hydrodynamic mode of friction is broken, is defined by the results of calculation.

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

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

$$\delta(\varphi,\beta) = I_h(\varphi,\beta) \cdot 2 \cdot \beta_c \cdot R_b. \tag{6}$$

Here $\varphi$ [deg] is the current value of a crank angle; $\beta_C$ [rad] is a contact half-angle; $\beta$ is a current angular coordinate that changes between – $\beta_C$ and $\beta_C$; $R_b$ [m] is the bearing radius. Linear wear intensity is defined as

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

Here $p\,(\varphi, \beta)$ [Pa] is the 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 of the second technique for the resource of the connecting rod bearing 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 Fig. 1.

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

| Bearing width *B*, [mm] | Results of calculation | | | |
|---|---|---|---|---|
| | $h_{min}$ [μm] | $\gamma$ [μm/h] | $\alpha_{h_{lim,cr}}$ [%] | $R_h$ [h] |
| *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 |

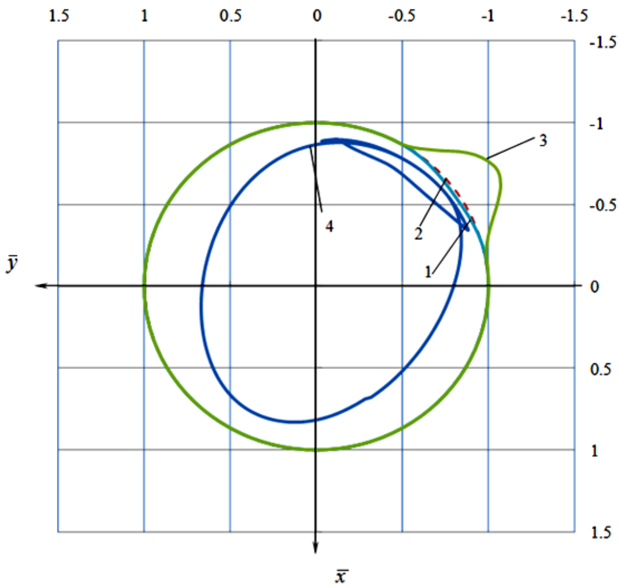


**Fig. 1.** Trajectory of the movement of the center of a rod journal in the bearing: 1 – theoretical wear diagram of the journal; 2 – unit circle; 3 – theoretical wear diagram of the bearing; 4 – trajectory of the journal.

The results of calculation characterize the resource of the connecting rod bearing with the engine in a 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 for the connecting rod bearing, it 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 with the experimental data. In each separate case, justification of limiting wear size, law of the wear intensity change, and wear speed is required.

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

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

In the next stage of the method is necessary to improve the methodology of calculation of hydro-bearing characteristics in the direction of accounting changes the geometry of the journal and bearing as a result of wear.

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