A research of bearing parameters' influence on the working capacity of lubrication system

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The working capacity of a hydrodynamic lubrication system is evaluated by a quantity of flow and pressure in all units of a system, and by value of oil film thickness, loss of power , maximum hydrodynamic pressure and temperature in all bearings of engine. This values are calculated relative to the turning angle of the engine crankshaft and their maximum, minimum and mean values per cycle are defined. The problem is solved by means of iteration method.

On the first stage the motion trajectory of the shaft centre in the plain bearings lubricated under pressure and loaded by forces variable in quantity and direction are calculated. Calculations are carried out by numerical methods taking into account the size and location of the lubrication sources (openings, grooves, pockets) for a number of constant values of a feed pressure.

In the process of hydrodynamic and thermal calculation the basic parametres of bearings which are a function of time (a turning angle of the engine crankshaft): minimum oil film thickness $h_{\min}$, maximum hydrodynamic pressure $P_{\max}$, friction losses of power N , the side-leakage flow of the bearing Q are determined. On the basis of calculation of momentary values of these parametres we define the mean values of $\overline{h}_{\min}$, $\overline{P}_{\max}$, $\overline{N}$, $\overline{Q}$ per loading cycle and a middle-integral temperature of a lubrication layer T.

The hydrodynamic and thermal calculations are based on the combined solution of a viscous liquid flow in thin oil film equation (Reynolds equation), a shaft centre motion equation and thermal balance equation.

A generalized Reynolds equation for pressures in thin oil film of dynamically loaded bearings is writen in the form:

$$\nabla\left(H^3\nabla\Pi\right) = M(\varphi)\,, \qquad M(\varphi) = 6\overline{\mu}_p\left(\Omega\frac{\partial H}{\partial\varphi} + \frac{\partial H}{\partial\tau}\right) \tag{1}$$

where H nondimensional oil film thickness; Π nondimensional hydrodynamic pressure; $\overline{\mu}_p$ nondimensional dynamic coefficient of a lubrication viscosity corresponding to a rated temperature ; Ω relative angular speed; τ relative time; φ circular coordinate of bearing.

The motion equation is put in the form of a loading balance eqution

$$\begin{aligned} R_X + F_X &= 0 \\ R_Y + F_Y &= 0 \end{aligned} \tag{2}$$

where R_X, R_Y nondimensional reactions of oil film; F_X, F_Y relative loads acting on the bearing.

The thermal balance equation can be represented in the form of

$$A_N = A_Q \tag{3}$$

where A_N a quantity of heat generated in the oil film; A_Q a quantity of heat lost with the side-leakage flow of the bearing.

The calculation algorithm of equations (1), (2), (3) is based on the numerical methods. To find the solution of Reynolds equation the finite difference method is used. The nonlinear equations for loading are solved by Fauler and Booker methods. The thermal balance equations should be handled for every step of the shaft centre trajectory or in every loading cycle after the trajectory calculation.

The results of hydrodynamic calculation of bearings are used for formulation of nonlinear macromodels of bearings and for calculation of hydrodynamic resistances in the oil film in clearances between bearing surfaces.

On the second stage the calculation of the lubrication system is carried out. Here the calculation algorithm is based on a power chain method. A block structure of model formulation and its solutions are interconnected, therefore they are considered from a single point of view caused by a common method of presentation of designed lubrication systems in the form of substituition schemes consisting of two-pole components.

In general case the component equations are nonlinear. Their forms are various and they depend on the required accuracy of study. The general form of component equations

$$\begin{aligned} i_i &= f_1\left(u_k, i_e, u_m, i_m, x_p, t\right) \\ u_e &= f_2\left(u_k, i_e, u_m, i_m, x_p, t\right) \end{aligned} \tag{4}$$

where k,e,p indices, having any value out of a multitude of component system names; i flow variable; u tension variable; x shift; t time.

While calculating the hydraulic lubrication system a lubrication flow (Q) is used as a flow variable, and a pressure difference (P) as a difference variable.

In case of numerical methods of a system solution the system motions are considered for a number of constant discrete time intervals. If a mathematic model is formulated for discrete time interval without a continuous system analog, formulation procedure is greatly simplified and the speed of solution rises. For this equation the elements are subjected to algebraization and linearization on the component level.

The way of interconnection of components is described by topological equations (coupling equations),which are composed on the basis of conservation and continuity laws:

1) algebraical sum of flow variables for any node points of a substitution scheme is equal to nought

$$\sum_{q=1}^{n} i_q = 0 \tag{5}$$

where n a number of connecting lines that are centred in the node points (it is a continuity flow law for lubrication systems);

2) algebraical sum of tension variables for a substitution scheme of any contour is equal to nought

$$\sum_{p=1}^{m} u_p = 0 \tag{6}$$

where m number of connecting lines in a contour (it is formed on the basis of Bernully equation for lubrication systems).

Mathematical model of a lubrication system studied is calculated by a combined solution of component and topological equations. Having solved a static problem we get the distribution of pressure and flow in all node points of a lubrication system.

On the third stage the bearing parameters are calculated more accurately owing to the results of the second stage.

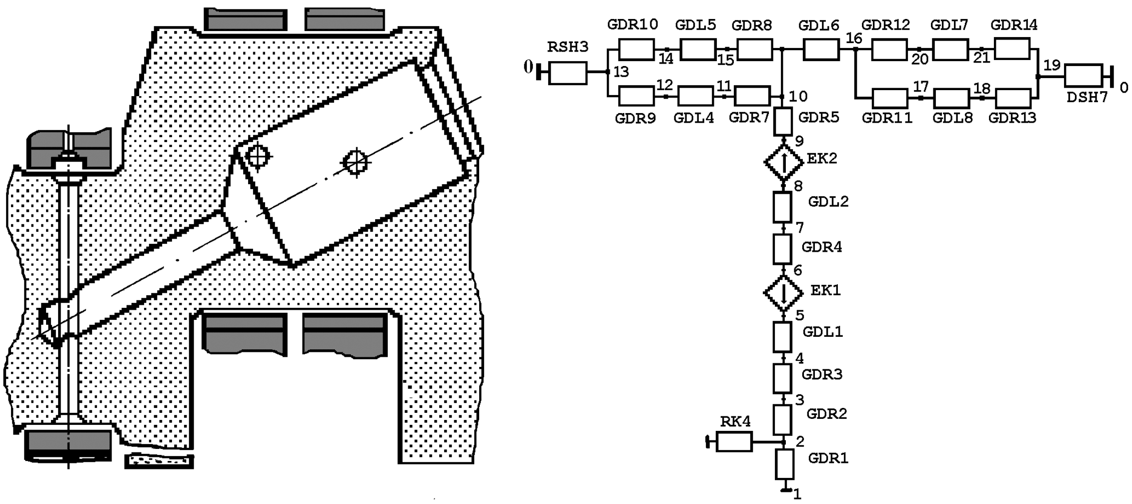
Consider the worked out method for calculation of librication system element in internal-combustion engine. On Fig.1,a the lubrication scheme for 4-th main and 3-rd and 7-th connecting rod bearings is given. The lubrication flow through system of grooves is modelled by a substitution scheme (see Fig.1,b). This scheme consists of two-pole components that are connected by means of node points. The component names of the substitution scheme are given in accordance with specification:

GDR... conductivity of local resistance; *GDL...* conductivity of a part of the groove or the canal; *RK4,RSH3,RSH7* values of resistance with side leakage lubrication flow of bearing; *EK...* inertial pressure in canals; *ED...* pressure sources.

The calculation results of instantaneous values of side-leakage flow of the 3-rd connecting rod bearing (Q_{sh}) and pressure oil feed for the 3-rd connecting rod bearing - P_{sh} for a series of radial clearances h_0 are presented on Fig.2 (1 - h_0=100mkm, 2 h_0=200mkm, 3 h_0=300mkm). Fig.3 shows the values of Q_{sh} and values of P_{sh} depending upon the feed pressure of the 4 th main bearing P_0 (1 P_0=0.45MPa, 2 P_0=0.25MPa, 3 P_0=0.15MPa).

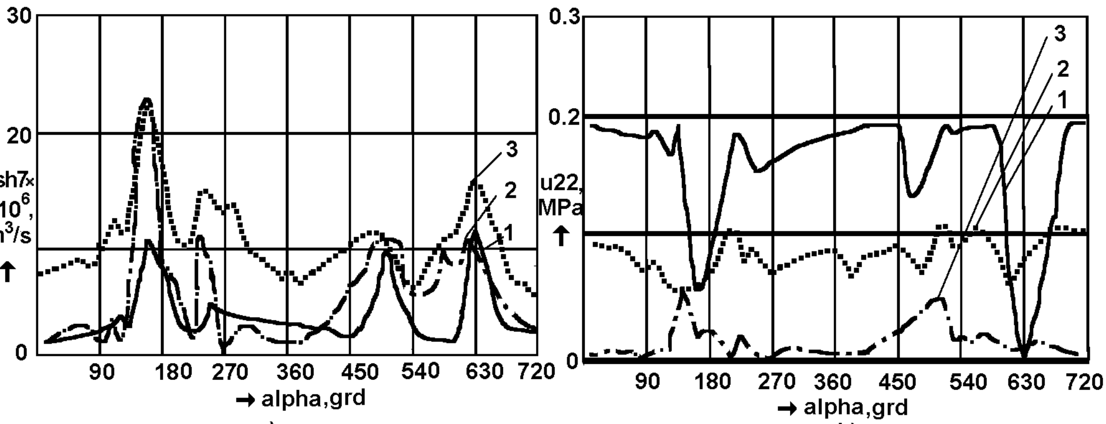
Fig.4 represents the calculation results for the 3-rd connecting rod bearing. The values of hydrodynamic parameteres are received for the same values of pressure P_0 and radial clearances. The comparison of calculation results and experimental data confirms the correctness of calculation method for hydraulic lubrication system.

The analysis of the results shows that the most advantageous from the point of view of working capacity are the operating modes with pressure 0.15 ... 0.25 MPa in the main lubrication canal and the radial clearances of 140...160 mkm. If pressure in the main lubrication canal lowers to 0.15 MPa or the radial-clearance rises to 160 mkm, there starts the mode of "lubricant hunger" bearings do not get lubricant.



a) b)

Fig.1 The crankshaft bearing lubrication system element: a) scheme of lubrication system for the 4-th main and 3-rd, 7-th connecting rod bearings; b) substitution scheme.



a) b)

Fig.2 The dependence of hydraulic characteristics upon radial clearances in bearings: a) the side-leakage flow of the 3-rd connecting rod bearing for the three values of radial clearances; b) feed pressure into the 3-rd connecting rod bearing for the three values of radial clearances.

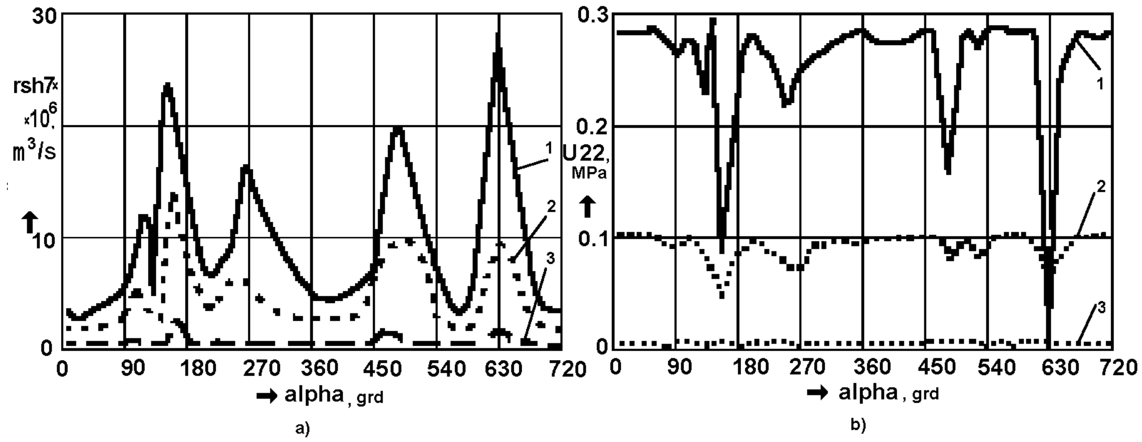


Fig.3 The dependence of hydraulic characteristics upon pressure in the main lubrication canal: a) the side-leakage flow of the 3-rd connecting rod bearing for the three values of pressure P_0; b) feed pressure into the 3-rd connecting rod bearing for the three values of pressure P_0.

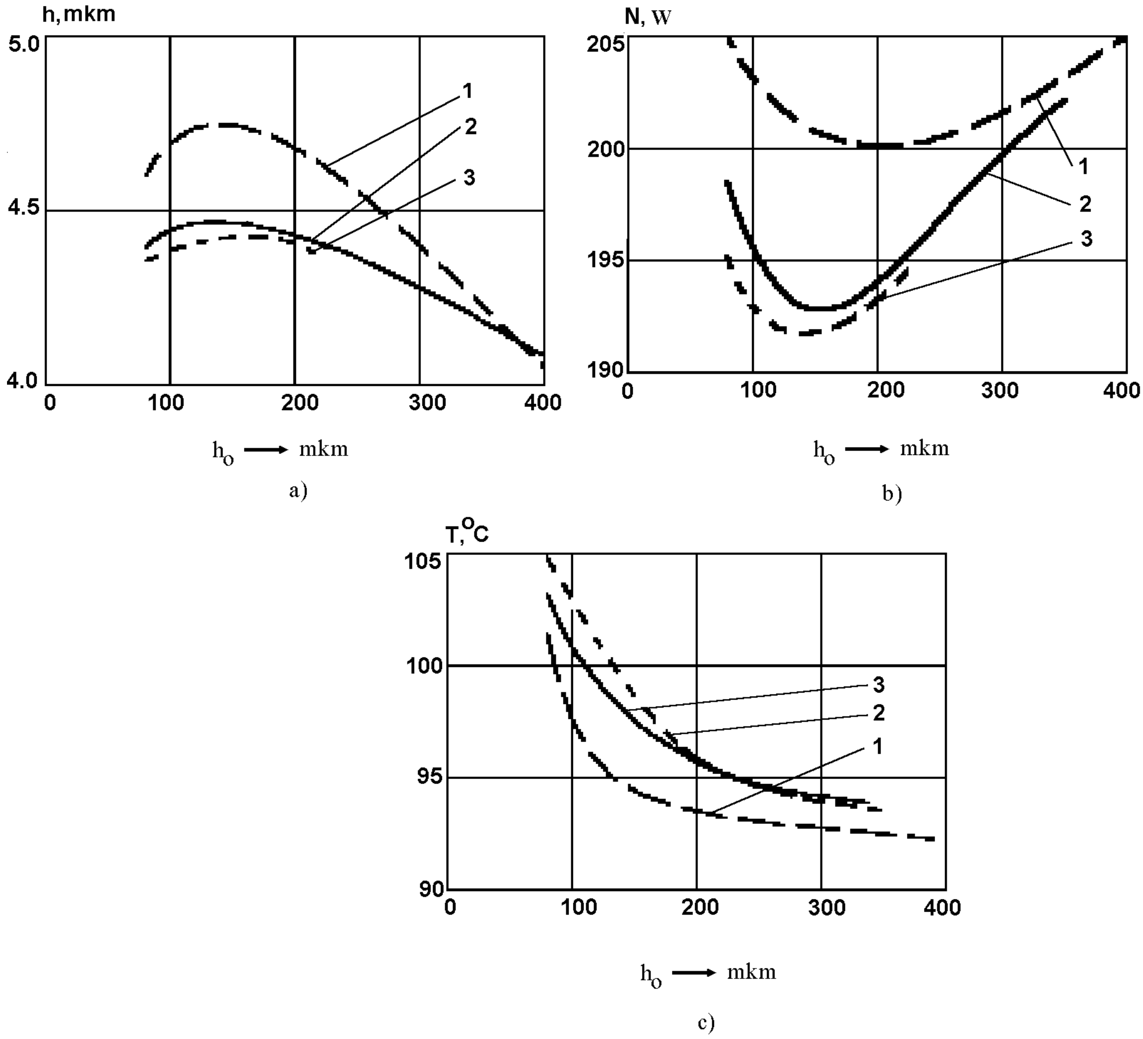


Fig.4 The dependence of hydrodynamic parametres of oil film upon the bearing radial-clearances: a) mean values of oil film thickness; b) friction losses of power in the connecting rod bearing; c) middle-integral temperature in the oil film.