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Internal combustion engine valve gear cam wear and its influence on valve gear and engine efficiency

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1. Introduction

An increase of internal combustion engines (ICE) power, economic and ecological parameters is connected with the improvement of the gas exchange processes that frequently are accompanied by the growth of valve gear part loadings, especially in a cam pair. In this connection damages of the considered pair, in particular, wear process of its working surfaces have become frequent. Researches executed before have shown, that the wear of cam pair and the corresponding change of the cam configuration lead to the increase in valve gear dynamic loadings and to possible force short violation of kinematic link, that limits serviceability of the mechanism. Besides the wear of cams profiles reduces average valves rises that leads to the reduction of their time-section, increase of hydraulic resistance in inlet and outlet, and also to the displacement of timings. Therefore, it is necessary to count, that the given problems, such as valve gear cam wear process forecasting in service and also influences of cam profile wear on valve gear serviceability and engine parameters, are actual.

2. Mathematical model

ICE valve gear cam-tappet wear intensity [1-4] where k is factor, taking into account the influence of additional parameters on a pair wear; σ is contact stress; HB is hardness of the material; μ is sliding friction factor; μ_S is ganging factor; R_{a1} и R_{a2} are arithmetical mean roughnesses of the profiles; h_b is boundary film thickness; ρ is reduced curvature radius of the surfaces in contact; η is dynamic oil viscosity at the temperature of making surfaces to contact; v_G is hydrodynamic effective speed, defined as the sum of a contact point migration speeds v_1 and v_2 accordingly on the cam and the follower; v_S is sliding velocity; F is linear loading in the contact; β_M is pressure exponent of oil viscosity; λ is oil heat conductivity; $Pe_{1,2}$ is average Pekle; α_T is the factor, describing oil viscosity dependence on temperature; v_S is relative sliding speed of the surfaces.

$$J_h = k\left[\frac{\sigma}{HB}\sqrt{\frac{\mu^3}{2\,\mu-\mu_S}}\right]\left[\frac{\sqrt{R_{a1}^2+R_{a2}^2}}{h_b+0.63\rho\left(\dfrac{\eta v_G}{F}\right)^{0.7}\left(\dfrac{F\beta_M}{\rho}\right)^{0.6}\left(\dfrac{\lambda Pe_{1.2}^{0.5}}{\alpha_T\eta v_S^2}\right)^{0.325}}\right] \qquad (1)$$

The values of reduced curvature radius, loading in contact, contact stress, speeds v_S and v_G are defined under the known formulas depending on the valve gear kinematic scheme [5]. Friction factor is defined on the appropriate empirical dependence, obtained for high-speed cam mechanisms [3]

$$f = \frac{0.065\left(10+lg\left(\dfrac{6\cdot 10^4 R_a}{E_r\rho}\right)\right)}{v^{0.07}v_G^{0.12}v_S^{0.2}} \qquad (2)$$

where R_a is roughness of the contacting body; E_r is reduced elastic modulus of the cam and follower; v is kinematic oil viscosity at conditions of entry to the contact.

Running loading F is defined as the division of the acting effort by length b of the contact. Average Pekle number $Pe_{1,2}$ characterizes the division of the stream heat in axial and cross-section directions and is calculated as arithmetic mean for the connected surfaces

$$Pe_{1,2} = \frac{b v_S c_{1,2}\gamma_{1,2}}{\lambda_{1,2}} \qquad (3)$$

where $c_{1,2}$, $\gamma_{1,2}$ and $\lambda_{1,2}$ are accordingly thermal capacity, density and heat conductivity factors of the cooperating details material (the index 1 concerns to a cam, 2 - to a tappet).

Linear wear of the cam profile surface for one loading cycle

$$\Delta h = J_h S_{fr} = J_h 2b_1\left|\frac{v_S}{v_1}\right| \qquad (4)$$

where S_{fr} is friction way; b_1 is halfwidth of Hertzian contact zone; v_1 is contact point migration speed on the cam. Further on the basis of (4) linear wear of the working cam surface depending on its rotation angle is calculated after several lading cycles during some time under the condition of tribological characteristics invariance.

Thus, the resulted formulas for the calculation of wear process intensity and linear wear of cam the pair allows to estimate directly the durability of this pair. On their basis the technique of movement law definition of the follower and tribological characteristics of cam pair are developed while in service from the view point of increasing profile wear. This technique includes, in particular, cyclically repeating procedures of double numerical differentiation of the tabular follower movement law on cam rotation angle with smoothing, calculation of the wear intensity and cam profile linear wear after the set time interval in the view of tribological characteristics and loading change, definition of the current cam profile in view of increasing wear.

Executing sequential calculation of the cam-tappet tribological characteristics and cam profile wear with the step, given on time, we obtain the dependence of cam profile change on time while in maintenance. It, in turn, enables to estimate wear process influence on valve gear dynamics, its work capacity, connected with unbreakable kinematic chain, and also on power and economic engine parameters. Thus it is possible to reveal limiting norms of cam wear and durability.

The described calculating technique is realized as a complex of the computer programs in the language Fortran. The value of factor k in the formula (1) was selected according the recommendations based on results of the experimental researches [6].

Based on numerical experiment the wear of the valve gear cam profile during 4000 operation hours has been investigated. Under the investigation automobile engine valve gear has a rocker with cylinder contact surface. The factor k in the formula (1) has been accepted $0{,}5 \cdot 10^{-7}$ according to recommendations [6]. Calculations were carried out on two high-speed modes: at values of angular cam rotation speed 157 and 280 rad/s. Comparative characteristics of the initial and worn out profiles for the first mode are submitted in Figs. 1 - 7, for the second - in Figs. 8 - 11. Besides in Figs. 12 - 17 curves of some parameters are submitted depending on time of wear process. Time calculation step was accepted to be 50 hours. Every time after this step the current wear was determined, characteristics of cam pair and wear intensity were anew specified. The executed numerical experiments have shown, that further reduction of the step does not allow to specify the calculation results.

The submitted figures allow to make a conclusion that characteristics of the initial and worn out cam profiles as a whole are similar. Essential differences are observed only at the ends of the profile. This is caused by the insignificant values of the follower displacement in this sections, but in computation algorithm displacement zeroing was accepted if it becomes less than zero as a result of wear process. However the cam profile ends are low loaded, and wear is insignificant here. Therefore wear process does not represent essential interest here.

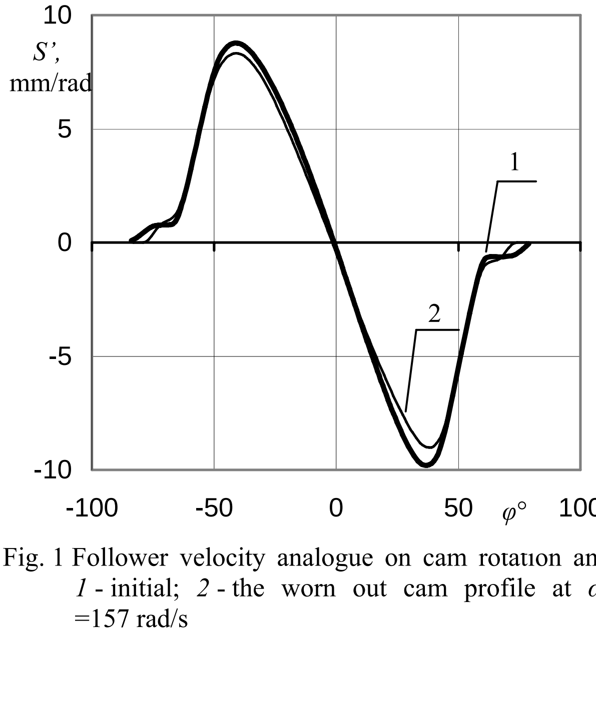

Fig. 1 Follower velocity analogue on cam rotation angle: *1* - initial; *2* - the worn out cam profile at ω = =157 rad/s

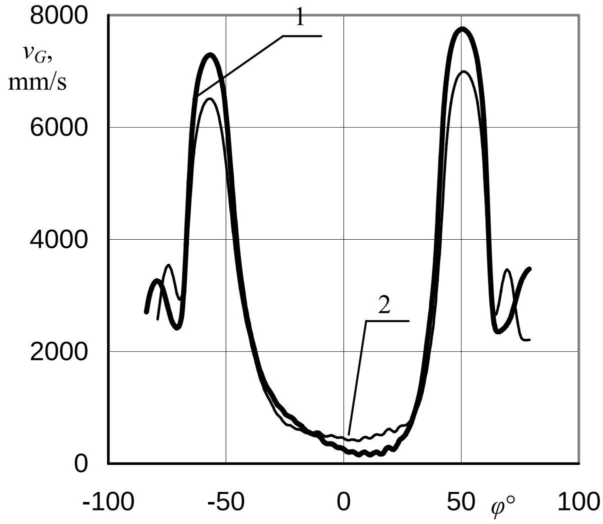

Fig. 2 Hydrodynamically efficient velocity on cam rotation angle at ω = 157 rad/s: *1* - initial, *2* - the worn out cam profile

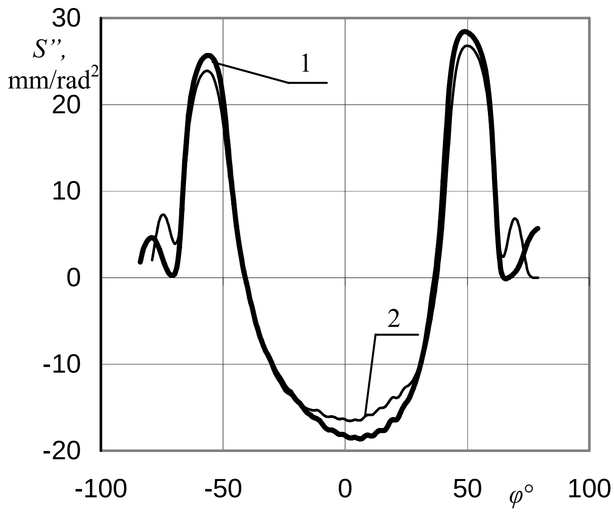

Fig. 3 Follower acceleration analogue on cam rotation angle: *1* - initial; *2* - the worn out cam profile at ω = =157 rad/s

During cam profile wear process at angular speed w = 157 rad/s, that approximately corresponds to a maximal engine torque, the maximal follower speeds on rise and sinking, and also its maximal positive and negative acceleration module (Figs. 1, 3) are slightly reduced. Hydrodynamically effective speed (Fig. 2) increases a little in its minimal values field that promotes reduction of the wear process intensity with the lapse of time.

Maximal values of a contact stress (Fig. 5) at the top of the cam are reduced owing to some increase of the cam profile curvature radius (Fig. 4). Thus the wear maximum (about 0.53 mm) is in the top of the cam zone (Fig. 7). In the same place wear process intensity J_h

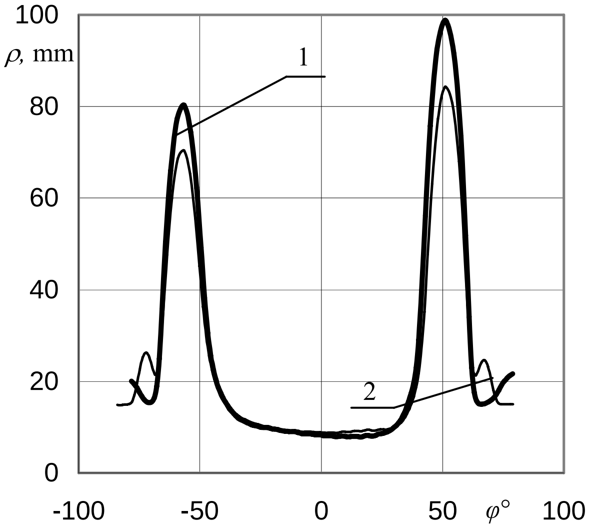

Fig. 4 Cam profile curvature radius on cam rotation angle: *1* - initial, *2* - the worn out cam profile at ω = =157 rad/s

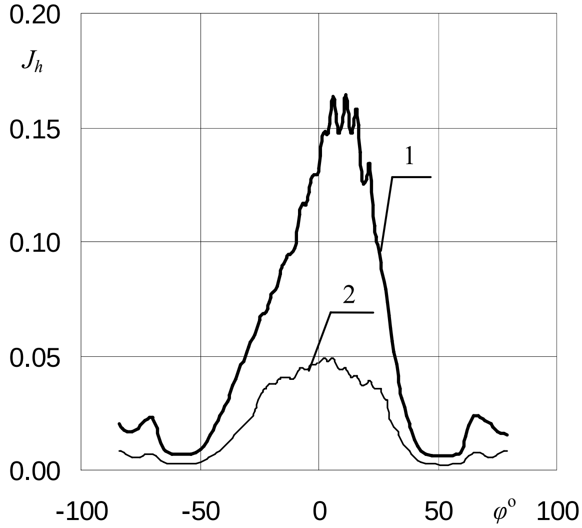

Fig. 6 Cam profile wear intensity on cam rotation angle: *1* - initial, *2* - the worn out cam profile at ω = =157 rad/s

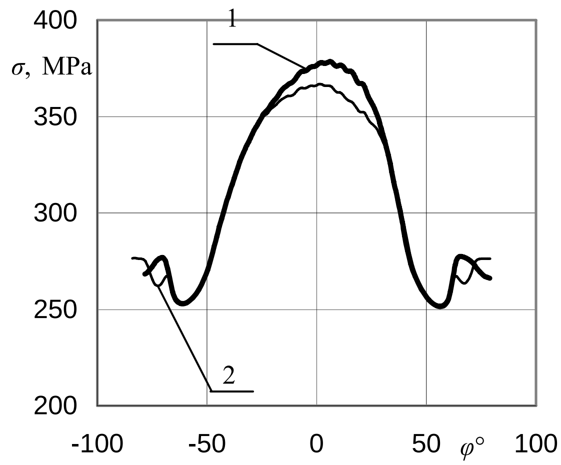

Fig. 5 Contact stress in cam pair on cam rotation angle: *1* - initial; *2* - the worn out cam profile at ω = =157 rad/s

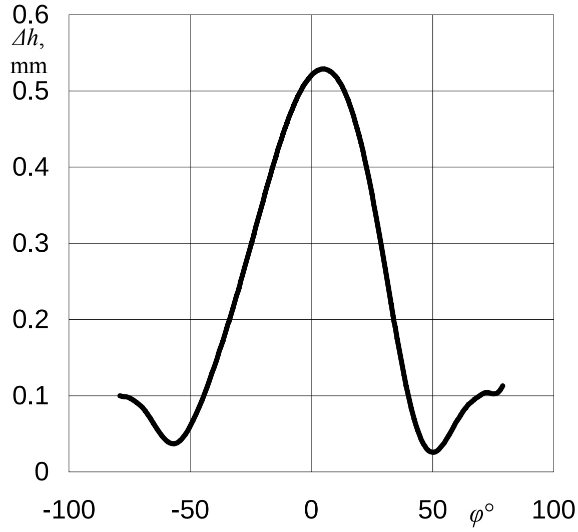

Fig. 7 Cam profile linear wear on cam rotation angle after 4000 hours at ω = 157 rad/s

(Fig. 6) is maximal also. It is necessary to note, that oscillatory character of J_h change dependent of the cam rotation angle at cam top zone is caused by the numerical procedures errors, namely, unstable process of double numerical differentiation of the follower displacement law. Thereof there are small errors of the basic tribological characteristics determination dependent on follower speed and acceleration, causing the necessity of resulting characteristics smoothings after time computation step. It is necessary to note also essential decrease of the cam profile wear intensity J_h of the worn out cam in comparison with initial (Fig. 6), caused, mainly, surfaces roughness reduction after extra earnings of the pair.

On nominal engine regime (cam rotation speed is ω = 280 rad/s) there is a different character of the cam profile wear process. There are two maxima of linear wear on angle cam rotation (Fig. 10): at φ approximately equal -20° and 28° (Δh = 0.18 mm). One of its principal causes is the reduction of contact stress σ (Fig. 11) at the top cam field in connection with the growth of negative acceleration module at ω increase.

Besides the follower negative acceleration, loading in the contact and contact stress at the cam top area continue to decrease during wear process (Figs. 9, 11) that leads to the reduction of valve springs effort reserve. Thus at certain stage of wear process there are the breaks of valve gear kinematic chain. Therefore the corresponding wear should be counted extremely admitted, limiting serviceability of the valve gear as a whole. The common decrease in wear process intensity and linear wear on mode ω = 280 rad/s in comparison with mode ω = 157 rad/s is caused, side by side with the reasons marked above, the change of tribological characteristics in connection with the increase in angular cam rotation speed.

Changes of some cam pair parameters during wear process at angular cam rotation speed ω = 157 rad/s are submitted on Figs. 12 - 17. Their analysis allows to draw the following conclusions. First, there is essential decrease in wear process intensity at the initial stage of wear process, connected with extra earnings (reduction of the contacting surfaces roughness). Second, there is observed some increase in wear process intensity after approximately 1500 work hours because of material hardness reduction. At last, there is some decrease in wear process intensity in connection with the change of the cam pair tribological characteristics.

Further worn out cam profiles are used at working processes simulation in ICE cavities [7]. The simulation was carried out on the basis of quasi-stationarity hypothesis by numerical solving of differential equations, describing working body state in ICE elements. In result the correlation of valve gear parameters and characteristics with ICE working processes and its effective capacity is established. In particular, it was established, that after 12 thousand hours of the automobile engine operation the valve timing are noticeably reduced, and the maximal cam wear is 1.15 mm (Fig. 18). In outcome the effective engine power is reduced by 6.7%, and the specific effective fuel consumption increases approximately by 2 %.

Thus, the limitation on ICE parameters degradation, being set it is possible to define an extreme supposed wear of the cam pair and its durability.

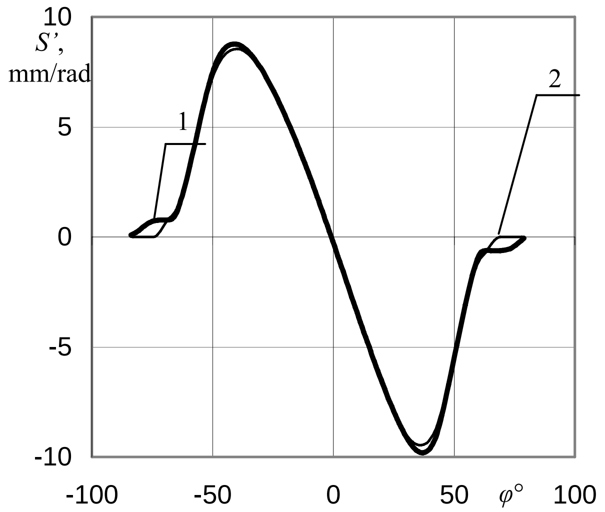

Fig. 8 Follower velocity analogue on cam rotation angle: *1* - initial; *2* - the worn out cam profile at ω = =280 rad/s

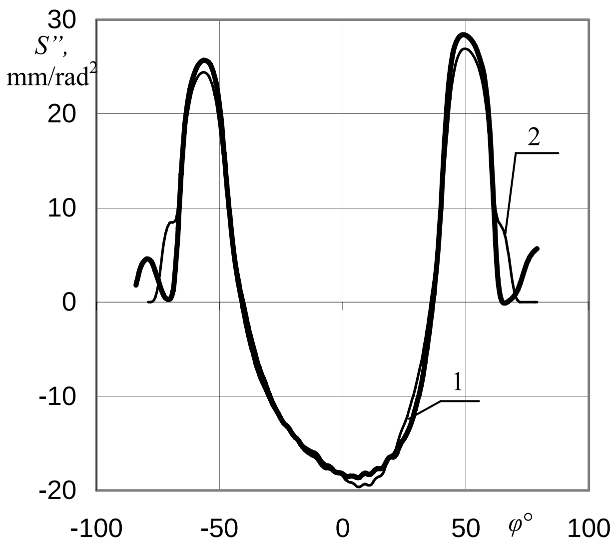

Fig. 9 Follower acceleration analogue on cam rotation angle: *1* - initial; *2* - the worn out cam profile at ω = =280 rad/s

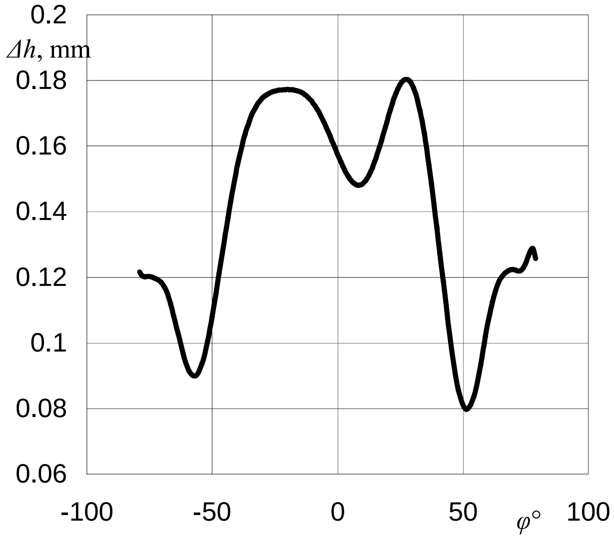

Fig. 10 Cam profile linear wear on cam rotation angle after 4000 hours at ω = 280 rad/s

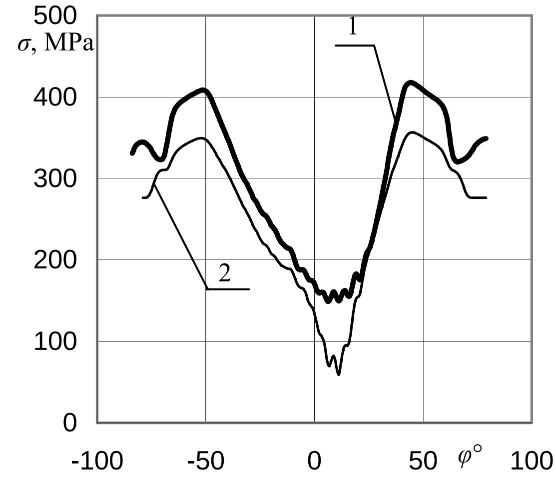

Fig. 11 Contact stress in cam pair on cam rotation angle: *1* - initial; *2* - the worn out cam profile at ω = =280 rad/s

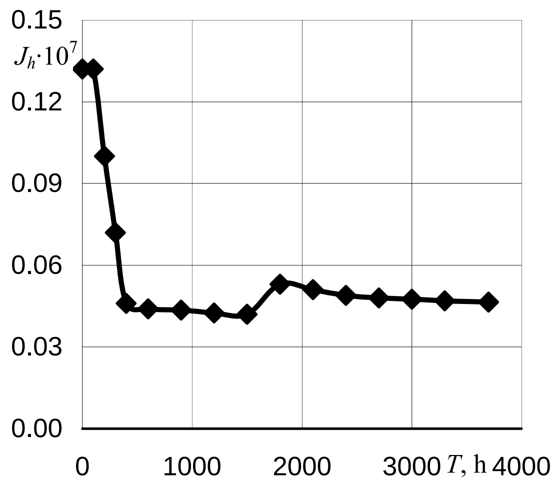

Fig. 12 Cam profile wear intensity at the top of the cam during wear process at ω = 157 rad/s

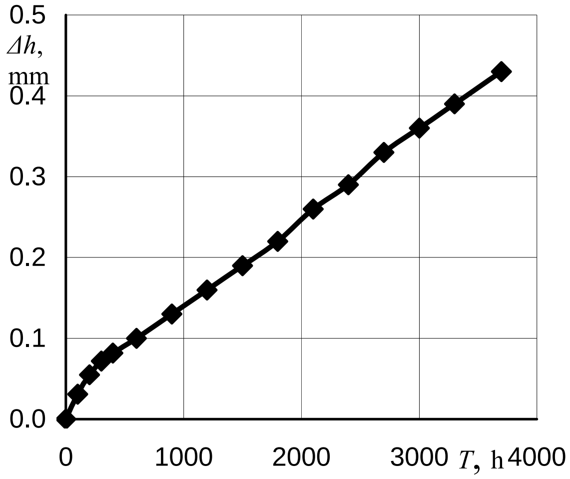

Fig. 13 Cam profile linear wear at the top of the cam during wear process at ω = 157 rad/s

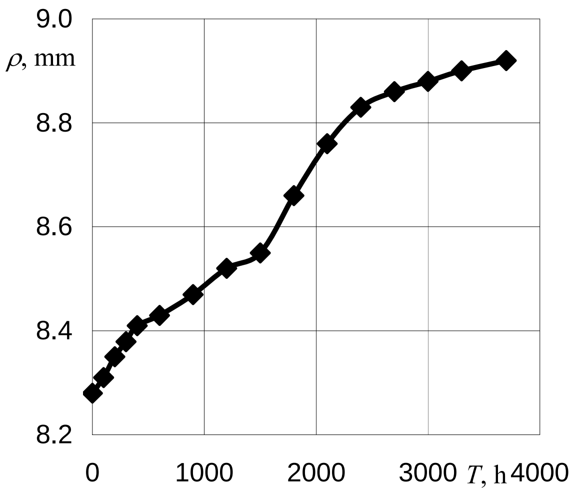

Fig. 14 Cam profile curvature radius at the top of the cam during wear process at ω=157 rad/s

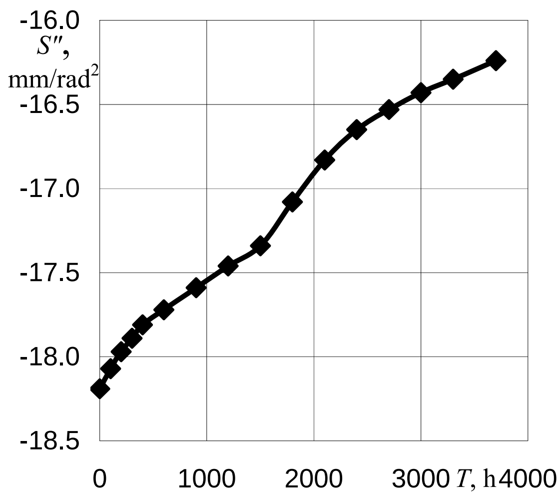

Fig. 15 Follower negative acceleration at the top of the cam during wear process at ω=157 rad/s

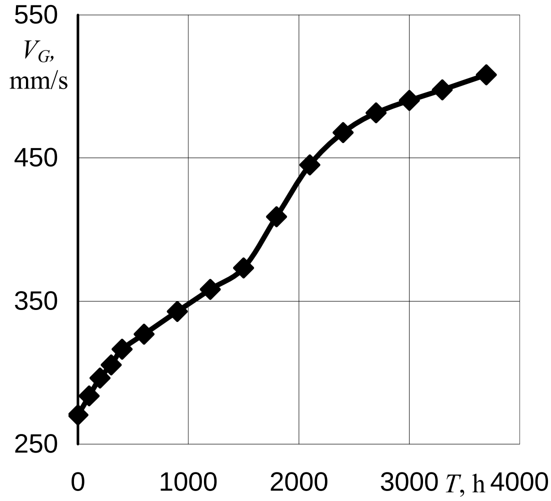

Fig. 16 Hydrodynamically efficient velocity at the top of the cam during wear process at ω=157 rad/s

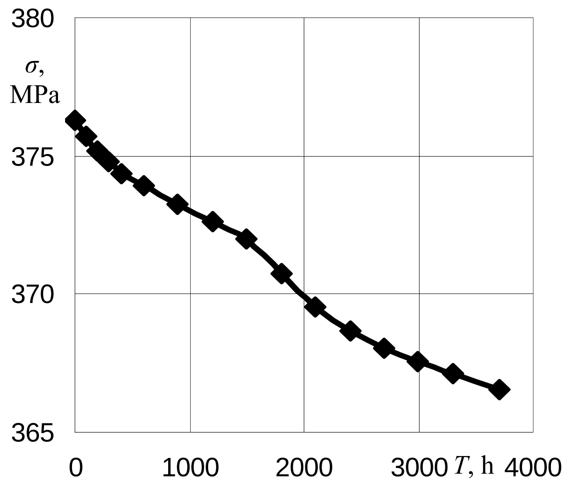

Fig. 17 Contact stress at the top of the cam during wear process at ω=157 rad/s

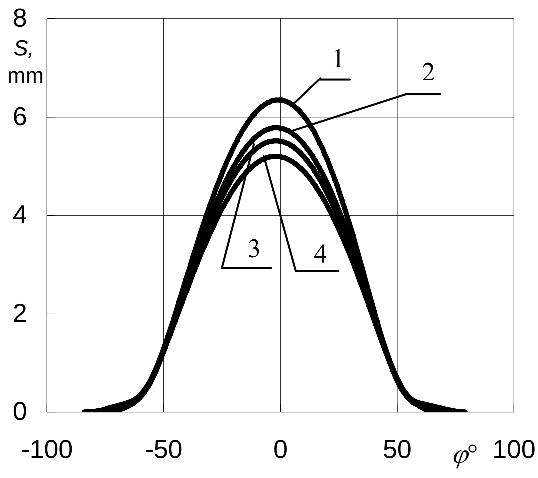

Fig. 18 Follower displacement depending on cam rotation angle: *1* - initial cam profile; *2* - after 4000 operation hours; *3* - after 8000 operation hours; *4* - after 12000 operation hours

3. Experimental

For the identification of the computation of cam pair tribological characteristics the experimental plant including loading machine and valve gear of the automobile engine have been developed. With its help the tests of valve gear including dynamics research, loading of the mechanism and wear process of its interfaces are carried out. At the given stage the results of cam wear process are received at the cam rotation angular speed w = 157 rad/s after 300 hours of tests. Its results allow to draw a conclusion on satisfactory concurrence of the wear process character on the given high-speed mode (Fig 19).

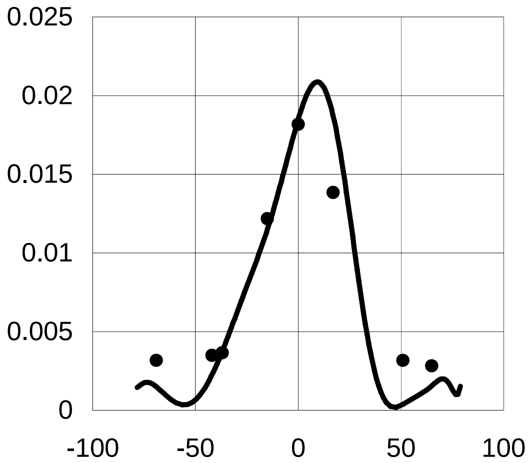

Fig. 19 Cam profile linear wear on cam rotation angle: calculation (line) and experiment (points)

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VIDAUS DEGIMO VARIKLIO VOŽTUVO MECHANIZMO KUMŠTELIO DILIMAS IR JO ĮTAKA VOŽTUVO MECHANIZMO IR VARIKLIO EFEKTYVUMUI

R e z i u m ė

Straipsnyje pateikiamas metodas vidaus degimo variklio vožtuvo mechanizmo kumštelio profilio dilimui apskaičiuoti. Skaičiavimai atliekami atsižvelgiant į vožtuvo mechanizmo kinematinį tipą, jėgos veiksnius, medžiagų savybes, hidrodinamines kumštelio ir stūmiklio kontakto tepimo sąlygas, taip pat tribologinių parametrų pasikeitimus eksploatuojant. Be to, ištirta kumštelio dilimo įtaka variklio darbo rodikliams. Tai leidžia prognozuoti skirstymo velenų darbingumą ir ilgaamžiškumą tiek juos projektuojant, tiek eksploatuojant.

A. Vasilyev, E. Deynichenko, D. Popov

INTERNAL COMBUSTION ENGINE VALVE GEAR CAM WEAR AND ITS INFLUENCE ON VALVE GEAR AND ENGINE EFFICIENCY

S u m m a r y

In this paper the method of cam profile wear calculation in internal combustion engine (ICE) valve gear is submitted. The calculation is executed with the account of the valve gear kinematic type, operating force factors, hydrodynamic lubrication conditions in the cam pair materials properties and also tribological characteristics changes during the maintenance. Besides the influence of cam wear process on automobile engine efficiency is investigated. It gives the possibility to predict camshaft's serviceability and durability both at the design stage and at the stage of operation

А Васильев, Е. Дейниченко, Д. Попов

ИЗНОС КУЛАЧКА КЛАПАННОГО МЕХАНИЗМА ДВИГАТЕЛЯ ВНУТРЕННЕГО СГОРАНИЯ И ЕГО ВЛИЯНИЕ НА КЛАПАННЫЙ МЕХАНИЗМ И ЭФФЕКТИВНОСТЬ ДВИГАТЕЛЯ

Р е з ю м е

В данной статье представлен метод расчёта износа профиля кулачка в клапанном механизме двигателя внутреннего сгорания. Расчёт выполняется с учётом кинематического типа клапанного механизма, действующих силовых факторов, гидродинамических условий смазки в сопряжении кулачок-толкатель, свойств материалов, а также изменения трибологических характеристик в течение эксплуатации. Кроме того, исследовано влияние процесса изнашивания кулачка на показатели работы двигателя. Это даёт возможность прогнозировать работоспособность и долговечность распределительных валов, как на стадии проектирования, так и эксплуатации.

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