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Article

Theoretical Study of the Size and Amount of Abrasive Particles in Motor Oil Involved in the Wear Process of Cam Profile

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Abstract: The article presents the results of a theoretical study of the amount and size of abrasive particles in the motor oil that cause wear of the gas distribution mechanism and the profile part of cam. The reliability of the results was evaluated on the basis of the experimentally determined values of wear indicators of the input and output cams of the camshaft.

Key words: camshaft, cam profile, wear resistance, tappet, abrasive particles, wear value, friction coefficient, wear rate.

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Introduction

It is known that today internal combustion engines are widely used in agricultural machinery, industrial equipment, construction machinery, marine ships and many other mechanisms. Currently, there are more than 1.3 billion vehicles and machinery powered by internal combustion engines worldwide. In order to use them rationally and to achieve high quality and resource efficiency, the research and improvement of their engine performance is considered a priority.As a result of the statistical analysis of internal combustion engine malfunctions, it was found that 67% of malfunctions in the gas distribution mechanism, which is one of the main mechanisms of internal combustion engines, are caused by wear of parts involved in the friction process, and 47-48% of these malfunctions are caused by direct distribution faults in the shaft[1]. For this reason, it is important to develop a method of researching the wear resistance of the friction pairs consisting of the cams of the gas distribution shaft and the pusher sleeve(follower). At the same time, it is important to optimize the resource and smoothness of the distribution shaft cam - follower pair, the concentration of abrasive particles in the oil, and the coefficient of roughness on the surface of the cam in contact with the follower.

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Materials. Participant in the grinding process is to determine the largest size of the abrasive particles entering the pin-shaped slot formed by the connection of the ash profile and the spherical part of the pusher sleeve. The equivalent force of this friction pair, which resists the penetration of abrasive particles into the pin-shaped plug, is determined by the geometric sum of its tangential and normal forces. In this case, the force N_{f1} and N_{f2} acting on the abrasive are moved parallel according to the diagram shown in the following figure (Figure 1), and then the vector sum of the forces is determined by the parallelogram method. The equal acting force resisting the entry into the nick tries to squeeze the abrasive particles out of the nick, and its direction acts in the direction opposite to the direction of the equal-acting line of the frictional force between the abrasive particle and the friction surface. shows, and motivates the abrasive particle to move in the direction of the friction surface in the groove. And the normal force acting on the abrasive particle tries to sink the abrasive particle into the friction surface consisting of the ash profile and the bearing sleeve[2].

We will consider the following cases of the contact of abrasive particles located in the pore groove with the friction surfaces.

First, we consider the case where the abrasive particle interacts with the friction surfaces having the same radius of curvature $P_1 = P_2$ and $\alpha_1 = \alpha_2$.

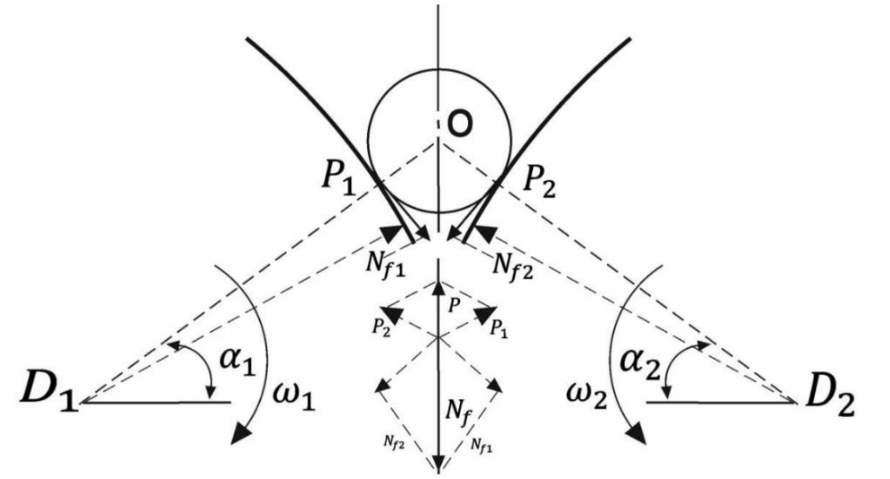

Figure - 1. Scheme for calculating the largest size of an abrasive particle entering the pin-shaped slot of a friction pair consisting of a sleeve profile and a pusher sleeve

Equivalent friction force between the abrasive particle and the friction surfaces,

$$N_f = \sqrt{N_{f1}^2 + N_{f2}^2 - 2N_{f1}N_{f2}\cos(90° + \alpha_1)}\ ,\ N; \qquad (1)$$

where N_{f1}, N_{f2} are the frictional forces between the surfaces of the friction pair consisting of the sleeve profile and the pusher sleeve and the abrasive particle, N; α_1 is the angle between the force vectors affecting the friction pair and the abrasive particle, grad.

The distance from the point of contact of the abrasive particle with the friction surfaces to the point of contact of the friction surfaces involved in the friction process is determined from Δ P1OO1, given in Figure 1:

$$OO_1 = \sqrt{P_1O_1^2 - P_1O^2} = \frac{\sqrt{4\rho d_{max} - d_{max}^2}}{2} \qquad (2)$$

where ρ is the radius of friction surfaces, m.

Assuming that the friction coefficients f_1 and f_2 between friction surfaces and abrasive particles are close to each other in terms of quantity, their value in the equation is defined as follows:

$$2f_1^2 f_2^2 = f_1^4 + f_2^4 \quad (3)$$

Solving the equation with respect to dmax, the largest size of the abrasive particle that enters the pin-shaped slot of the friction pair consisting of a bushing profile and a thrust sleeve:

$$d = \frac{\rho(f_1^2+f_2^2)}{1+f_1^2+f_2^2}, m. \quad (4)$$

An increase in the contact radius and friction coefficient of the sleeve profile and the pusher sleeve increases the possibility of increasing the size of the abrasive particles entering the plugging of the friction surfaces.

Research results.

If there is an oil film between the friction surfaces of the abrasive particle and the bearing profile and the pusher sleeve, the friction coefficient between this friction pair will decrease due to the grinding of the abrasive particle entering the slot.

$$\boldsymbol{d_{max}} = \boldsymbol{d_{max\,c}} - \boldsymbol{2} \cdot \boldsymbol{h_a}, m, \quad (5)$$

where h_a - is the thickness of the oil film between the friction surfaces of the sleeve profile and the pusher sleeve and the abrasive particle, mm.

When there is an oil film between the friction pair consisting of the abrasive particle and the cam profile and the pusher sleeve, the coefficient of friction in it is calculated and expressed as follows:

$$f_m = \sqrt{f^2{}_{can} \cdot \frac{(\rho_к+\rho_т)^2+\rho_к \cdot \rho_т \cdot f_{can}^2 - h_a^2}{\rho_к \cdot \rho_т}}; \quad (6)$$

where f_{can} is the coefficient of friction in dry friction.

The expression for calculating the size of the abrasive particle that can enter the slot of a friction pair consisting of cam profile and pusher sleeve[3]:

$$d_{max} = \rho_к \cdot \rho_t \cdot f_{men}^2 + (\rho_к + \rho_t)^2 \cdot f_{men}^2 - (\rho_к - \rho_t), mkm. \quad (7)$$

Table 1.

The variation of the estimated largest size of the abrasive particle entering the pin-shaped groove of the cam profile and the pusher sleeve in relation to the connection indicators of the cam profile and the pusher sleeve

Cam profile posi-tion	Radius of the position of the cam profile, mm		The coefficient of friction between the friction surface and the particle	The average speed of the cam profile mm/s	The average speed of an abrasive particle in a conical slot, m/s	The size of the abrasive particle entering the hole in the cam, µm	
	Input	Output				Input	Output
1	21,5	21,5	0,047	4,5	0,38	15,70	15,70
2	153,3	117,0	0,061	4,5	0,38	30,63	30,44
3	31,5	31,5	0,052	4,5	0,38	26,96	26,96
4	31,5	31,5	0,052	4,5	0,38	26,96	26,96
5	153,3	117,0	0,061	4,5	0,38	30,63	30,44
6	21,5	21,5	0,047	4,5	0,38	15,70	15,70

Thus, according to the calculation results presented in Table 3.8, when there is an oil film between the friction pair consisting of the cam profile and the pusher sleeve, the friction coefficient and the radius of curvature of the parts involved in the friction increase, the abrasive entering the pin-shaped slot and participating in the wear process causes the particle size to increase.

Determination of the amount of abrasive particles in the oil, which ensures the wear resistance of the cam profile. The maximum allowable wear of the cam profile $\mathbf{U_{ch}}$ consists of the wear in the presence of abrasive particles $\mathbf{U_{ish}}$ and the amount of wear that occurs without the presence of abrasive particles $\mathbf{U_{is}}$, i.e.:

$\mathbf{U_{ch} = U_{ish} + U_{is}}$.

The amount of wear with the presence of abrasive particles,

$$\mathbf{U_{ish} = U_{ch} - U_{is}}. \quad \mathbf{(8)}$$

Expression for calculating the amount of abrasives in the oil that provides the resource of the intake (exhaust) cam under the conditions of use:

$$\varepsilon_{k(\text{к,ч})} = \frac{(U_{ch(k,ch)} - U_{is(k,ch)}) \cdot H_{k,ch} \cdot n_p \cdot B_{k,ch} \cdot \gamma_m}{10{,}29 \cdot P \cdot d_{o\prime r}^2 \cdot \sigma_a \cdot G_{k,ch} \cdot n_{k,ch} \cdot \cdot \gamma_a \cdot k_v} \quad (9)$$

Calculations for cases 3 and 4, where the greatest wear occurs, are based on the following initial data:

U_{ch}=3,0 mm; U_{is}=2,3 mm; P=3000 soat; $H_к$=610 MPa; n_{pk}=24,0 k_v=1,003; $d_{o\prime r}$=0,012 mm; γ_a=1,9 г/см³; σ_a=103,9 **MPa**; G_{ch}=1,984; n_k=40 s^{-1}; γ_m=0,91 г/см³; k_v=1,003; B_k =0,764 mm.

$$\varepsilon_{k(k,ch)} = \frac{(3{,}0 - 2{,}3) \cdot 6100 \cdot 24{,}0 \cdot 0{,}764 \cdot 0{,}91}{10{,}298 \cdot 8000 \cdot 0{,}012^2 \cdot 103{,}9 \cdot 1{,}984 \cdot 40{,}0 \cdot 1{,}9 \cdot 1{,}003} = \frac{71248{,}2}{186268{,}6} = 0{,}38\%$$

Conclusions.

1. The wear in the profile part of the cams is different from each other in the output and input cams, this amount is different in the cam wear areas, and the largest wear occurs in the 3rd and 4th areas.

2.The value of the amount of wear on the profile of the cam is calculated depending on the amount of abrasive particles and changes proportionally to the pressure force transmitted through the pusher.

Literature used

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