Impact Factor: 9.9
ISSN-L: 2544-980X

Investigate of Wear Resistance Pair Consisting of Cam Profile of Gas Distribution Camshaft and Roller of Pusher

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Annotation: In the article, the wear resistance of the profile of the cam shaft of the internal combustion engine was studied. The dimensions of the modeled samples, the coefficient of acceleration of the wear test and the change of the test duration were determined. In this case, the wear surfaces of the intake and exhaust cams were divided into symmetrical parts, the wear process was studied for each part, and the values of the geometric dimensions and wear values were presented in tabular form.

Key words: camshaft, cam profile, wear resistance, pusher, contact width, contact area, roughness, wear rate.

Introduction

The study of the wear resistance of the friction pair consisting of the cam profile and the pusher sleeve of the engine gas distribution mechanism is a rather complex process, requiring the researcher to conduct extensive research into the dynamics and kinematics of the movement of the mechanisms. Therefore, the need arises to accelerate the testing process on a friction machine by modeling the wear of the friction pair consisting of the cam profile and the pusher sleeve with roller analogs. Because when studying the friction process, the roller analogs modeled for conducting such tests provide a higher angular velocity than the friction pairs of existing internal combustion engines. In this article, the characteristics of the wear process occurring in the intake and exhaust cams were studied depending on the joint movement of the cam and pusher sleeve. The values obtained as a result of the experiment were compared with the existing values, their practical significance was studied, and at the same time, the conclusions were interpreted as a practical result of the research.

Materials and research methods

In the friction and wear tests, samples made based on the indicators of the fixed-size parts of the cam profile are mounted on the lower spindle of the friction machine. The sample made based on the radius of curvature of the contact surface of the thrust sleeve has a fixed size, the sample made based on the radius of curvature of the cam profile, the kinematic scheme of which is shown in Figure 1, is mounted on the upper spindle of the MI-1M friction machine, and the sample made based on the radius of curvature of the spherical part of the thrust sleeve is mounted on the lower spindle of the friction machine.

Research results and discussion

The contact length of the position of these samples depends on the coverage angle and the radius of curvature of the position of the cam profile, and the contact length of the cam position can be determined by the following expression[1],

$$l_i = \frac{2 \cdot \alpha_i \cdot r_i}{\pi}. \qquad (1)$$

Where i is the angle of mutual coverage of the cam position and the friction surfaces of the pusher sleeve, degrees; ri is the radius of curvature of the cam position, mm.

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The ratio of the transmissions, determined from the ratio of the cam position to the radius of its cylindrical part, is given in Table 1, and its value is determined from the ratio of the radii of curvature of the contact surfaces of the pair of tested samples,

$$i_i = \frac{r_i}{r_s}. \qquad (2)$$

where rs is the radius of curvature of the cylindrical (1 and 6) sectors of the cam profile (Figures 1 and 2), and it is rs=21,5 mm, and when modeling its value, samples made on the basis of the cylindrical part of the distribution shaft have the smallest radius of curvature [1,2].

When modeling these cases through the center of rotation of the cam, the compression of the valve springs is equal to 10 mm, which is the amount of deformation of the valve spring when the cam completes a full rotation, equal to the amount of compression that occurs in cases 2 and 5 of the cam profile [3].

The radius of the cylinder of the valve, which passes through the center of rotation of the cam, is determined by adding the deformation of the valve spring (2 mm for each position) to the radius of 21.5 mm, and the point of intersection with the arc of the cam profile is determined, the distance from this point to the center of rotation of the cam is the radius of rotation of the cam profile at this point. The other radii of rotation of the cam profile in positions 2 and 5 are also determined in the same way.

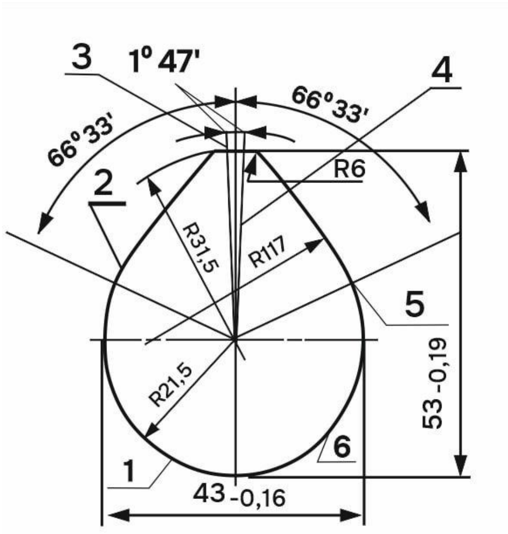

Figure 1. Layout of the characteristic parts of the input cam

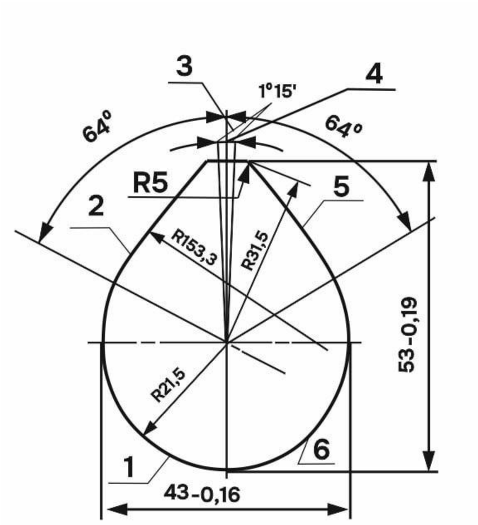

Figure 2. Layout of the characteristic parts of the output cam

The inlet profile of the cam was modeled by samples with a radius of curvature corresponding to the scheme presented in Figures 1 and 2, with friction pairs consisting of a pusher sleeve and an outlet cam profile and a pusher sleeve. Cases 1 and 6 of the cam profiles correspond to the cylindrical part of the cam profiles, and their radius is a distance equal to the length passing through the centre of the cam, equal to 21.5 mm.

To justify the size of the samples modeled on the basis of the cam profile and the pusher sleeve friction pair and to determine the coefficient of acceleration of wear testing. The camshaft cam profile was operated for a period corresponding to 1 engine oil change (250 hours). In it, the length of the samples involved in the friction process was determined for each position of the cam, based on the coverage angle, and the radius of the sample's melting point using expression (1).

The total value of the test acceleration coefficient Ki was determined from the product of the ratio of the rotation frequency of the sample tested on the friction machine to the rotation frequency of the distribution shaft bearing Kn, the abrasive particle content of the engine crankcase oil used in the friction machine under the conditions of use of the abrasive particle content Ka, and the ratio of the total length of the bearing profile to the length of the bearing profile Kp [3].

$$\gamma = \frac{Q}{2 \cdot \pi \cdot R \cdot b \cdot t \cdot \gamma_n} \qquad (3)$$

Where Q is the amount of wear measured in grams of the sample tested for wear by mass during a time equal to the duration t; R is the radius of curvature of the sample for which the wear rate is determined, mm; b is the length of the friction surface of the tested samples in contact, mm; n is the density of the sample material, g/mm3.

1 – jadval Results of wear test of the profile of samples made according to the conditions of the cam

№ o.n	The position of the sample in the cam profile	Amount of wear of cam, *mm*	Test load, *N*	Radius of the test specimen,*mm*	Acceleration coefficient of wear (general)	Speed of wear, mm/hour	
						For the cam profile	For a sample of the cam profile
For the input cam profile							
1	1, 6	0,41	264	21,5	19,1	0,000082	0,0016
2	2, 5-1	0,93	374	23,5	112,5	0,000184	0,0207
	2, 5-2	1,12	484	25,5	120,3	0,000224	0,0269
	2, 5-3	1,33	594	27,5	139,3	0,000264	0,0368
	2, 5-4	1,52	704	29,5	147,0	0,000321	0,0467
	2, 5-5	1,73	814	31,5	160,4	0,000344	0,0552
3	3, 4	2,50	814	31,5	1313,3	0,000504	0,6619
For the output cam profile							
1	1, 6	0,43	264	21,5	16,4	0,000088	0,0014
2	2, 5-1	0,97	374	23,5	96,4	0,000192	0,0185
	2, 5-2	1,13	484	25,5	103,1	0,000224	0,0231
	2, 5-3	1,34	594	27,5	110,9	0,000267	0,0296
	2, 5-4	1,54	704	29,5	119,9	0,000312	0,0374
	2, 5-5	1,76	814	31,5	130,5	0,000352	0,0459
3	3, 4	2,61	814	31,5	821,4	0,000522	0,4288

The determination of the wear rate of the camshaft profile presented in Table 1 was carried out on 6 sections of the camshaft profiles of the internal combustion engine, including 5 sections of equal length, corresponding to the profiles of sections 2 and 5, during the full service life of 5000 engine hours[4]. To calculate the wear rate of the camshaft profile, it was determined that the wear rate and wear rates of the samples modeling them, corresponding to each section of the camshaft profile, and corresponding to the data presented in Table 1, were determined, taking into account the number of tests conducted at intervals of 250 engine hours during the service life of 5000 hours and the wear rate of the profiles of the inlet and outlet camshafts during this period.

Conclusion

1. The total values of the wear test acceleration coefficient are the product of the components determined by the rotation frequency of the samples, the amount of abrasive particles in the oil, and the length of the camshaft in the state, and their values were found to be 16.5% higher for the intake camshaft than for the exhaust camshaft, regardless of the state of the camshaft profiles.
2. When the camshaft was tested for wear resistance in the engine structure for 250 hours and modeled with roller samples made on the basis of the test camshaft profile, the test duration for the intake camshaft was 11.21 hours, and for the exhaust camshaft - 13.41 hours, respectively, reducing the wear resistance test duration by 22.30 and 18.64 times.

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