GEOMETRY OF THE WORKING PART OF AN EXCAVATOR TOOTH

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Studies of excavator tooth wear kinetics conducted earlier by the present authors [1, 2] showed that the main factor controlling wear platform dynamics is the physical-mechanical property of the rock. Wear platforms evolve in two stages.

Tooth wear acquired during the "critical" stage [2] has no significant influence on excavator performance in the mining and geologic conditions typical for the northeastern regions of Russia. Cutting elements can continue to be used up to the maximum permissible wear level specified by the manufacturer. In this respect, intensive wear during initial stages apparently reflects some design imperfection rather than the effects of the work adjustment process. Investigators have studied the causes and consequences of intense wear of excavator teeth, but there are still no basic criteria upon which to formulate general principles so as to improve the wear resistance of cutting elements as determined by their design [3-5].

An efficient way to raise the wear resistance of an excavator tooth is to devise the design parameters of the working component so as to ensure classical single-stage wear, bypassing the "critical" (pseudoadjustment) phase. We developed a new excavator tooth design which features heightened wear resistance.

The outline of the working component of the tooth and its dimensions were developed with due regard for the main characteristic points of the wear resistance curves of mass-produced wedge-shaped teeth. To attain a linear behavior for the wear process of such teeth with a rate equal to or less than what is observed during the second stage of wear with mass-produced teeth, we specified the design parameters corresponding to the beginning of the second phase, where the specific pressure from the standard force of the thrust mechanism drops to 10-12 MPa. Figure 1 plots pressure variations on the wear platforms of teeth of buckets used in common quarry excavators according to the following expression:

$$P = U_{\mathrm{p}} \frac{P_1}{D \cdot i},$$

where U_p is the width of the wear platform; P_1 is the rated force of the thrust mechanism; D and i are the length of the tooth cutting edge and the number of teeth on the bucket, respectively.

The curves show that there are certain pressure regions on wear platforms where rock resistance to teeth is equal to or greater than the force developed by the thrust mechanism. This loading pattern for cutting elements is observed on monolithic strong (e.g., permafrost) rocks. On the other hand, some materials resist cutting with a much weaker strength than the force developed by this thrust mechanism. To estimate the specific pressures formed when cutting elements interact with these materials, we plotted curves 1-4 by computing the pressure on the wear platforms of an ÉKG-5A excavator tooth at 0.8, 0.4, 0.2, and 0.1 of the rated thrust force.

On weak rocks the pressure variation pattern on the wear platform is the same, but the pressures and dimensions for the worn portion of teeth after the beginning of the second stage may be much smaller (sometimes by a considerable factor). This is clearly seen in Fig. 1. Zone I, crossing the curves, defines the parameters of the onset of the second stage of wear for teeth of different excavators and for different rock strengths (curves 1-4). For ÉKG-5A excavator teeth the starting point of the second wear stage obtained experimentally lies in zone I and corresponds to a pressure of $P' = 10\text{-}12$ MPa and a wear platform width of $U_{t_{cr}} \approx 45$ mm.

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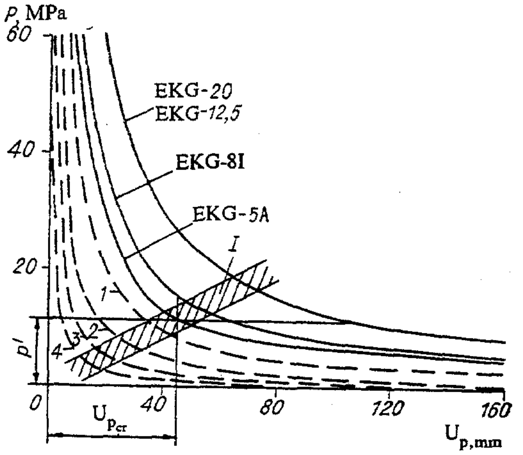


Fig. 1

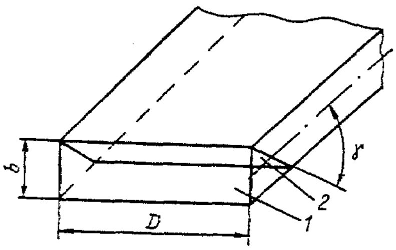


Fig. 2

Fig. 1. Pressure variation as a function of wear platform size (*1-4* — theoretical pressure curves on an ÉKG-5A excavator tooth wear platform when working rocks with resistance 0.8, 0.4, 0.2, and 0.1 of standard thrust force).

Fig. 2. Working part of a cutting element with wedge angle 180° (*1* — cutting edge with area S_0; b — edge width; D — length; *2* — wear platform surface area S_{p2}; γ — wear platform slope angle.

At a given size of the working part of the tool, the stage of critical wear or pseudoadjustment is virtually absent on rocks and grounds with low strength, while tools experience intense two-stage wear on strong/hard rocks. In different mining and geologic conditions, it is obviously convenient to work with interchangeable tools. It is currently impossible to control the force parameters on the working element of an excavator. The operator observes the work of the machine visually, watching its motion and bucket filling. The loads acting upon working elements and teeth thus depend not only on rock resistance to cutting, but largely on operator skill and experience. An efficient and rational approach to devising working tooth component parameters is to consider the power of excavator drives.

The area of the cutting edge for a rectangular cutting profile with a 180° sharpening can be calculated from the pressure on the wear platform (see Fig. 2) corresponding to the onset of the second wear stage:

$$P = \frac{P_1}{S_{p2} \cdot i},$$

where P is the pressure on the wear platform when platform dimensions correspond to the beginning of the second stage; P_1 is the rated thrust force of the excavator (vertical component of the cutting force); S_{p2} is the wear platform area at the second stage onset; i is the number of teeth on the excavator bucket.

The wear platform is defined in terms of the cutting edge area as

$$S_{p2} = \frac{S_0}{\sin\gamma},$$

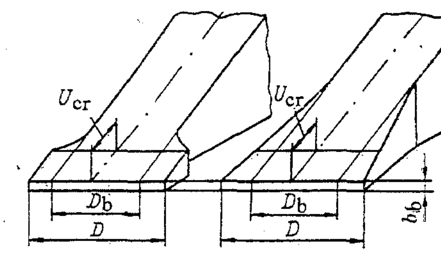
where γ is the wear platform slope angle relative to the back facet of the cutting profile; S_0 is cutting edge area.

The pressure on the wear platform can be expressed as

$$P = \frac{P_1 \cdot \sin\gamma}{S_0 \cdot i}.$$

The area of the cutting edge which provides the desired wear pattern for the cutting element is defined from the same formula:

$$S_0 = \frac{P_1 \cdot \sin\gamma}{P_1' \cdot i}.$$



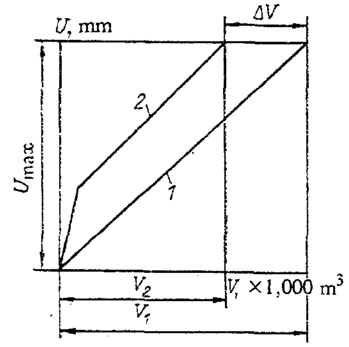


Fig. 3

Fig. 4

Fig. 3. Cutting elements with heightened wear resistance (U_{cr} — linear wear corresponding to first critical stage; b_b and D_b — basic width and length of cutting edge of wedge tooth; D — calculated length of cutting edge).

Fig. 4. Design-controlled wear resistance of wedge-shaped cutting elements (*1, 2* — variation of linear wear for a tooth with an expanded part and a standard tooth, respectively; U_{max} — maximum permissible wear; ΔV — increased operation resource of new tooth design.

Considering that the cutting edge area is linked to the wear platform by the preceding relation, we can formulate simple technological conditions for improving the design of the working component of standard wedge-shaped teeth in terms of optimal length of the tooth cutting edge as

$$D = \frac{S_{p2} \cdot \sin \gamma}{b},$$

where D is the optimal cutting edge length which provides steady single-stage wear of cutting elements; b is the actual (basic) width of the cutting edge of mass-produced wedge teeth; S_{p2} is the area of the platform corresponding to the onset of steady wear; and γ is the angle of the slope of the wear platform with respect to tooth longitudinal axis.

Figure 3 offers technological concepts for reduction of cutting element wear dynamics based on mass-produced wedge-shaped teeth. The length of the expanded part of a tooth (D) should be not less than critical linear wear U_{cr}. After the expanded part is worn off, a tooth acquires the natural size of the platform corresponding to the second stage of steady wear.

This design wears according to a linear relationship (Fig. 4) with an intensity equal to that of the second stage of wear of mass-produced teeth (parallel portions of plots). After attaining maximum wear, teeth would have extended service life, expressed in an increased volume of excavated rock (ΔV).

We should pay special attention to creating teeth with heightened wear resistance without modifying the basic dimensions or shape of the working component. This is important, because this form is easier and less expensive to manufacture. We developed the universal geometry for the working part of an excavator tooth based on calculations of the optimal width of the cutting edge while retaining the main dimensions of standard teeth designs.* The tooth with the new working component geometry (Fig. 5) has cutting edge 1, linear segment of back face 2, and curvilinear part 3. The front face is formed of two linear segments 4 and 5. The linear segment of back face 2 is parallel to tooth longitudinal axis 6, situated at distance Γ from the axis [6]. The plane of the cutting edge is situated at an angle greater than 90° to the cutting plane. This helps form a steady compaction core on the plane of the cutting edge, which partly protects it from wear. The cutting edge width is found from an empiric relationship:

*We took the tooth design developed by the Institute of Heavy Machinery (Uralmash Production Association) for the basic prototype.

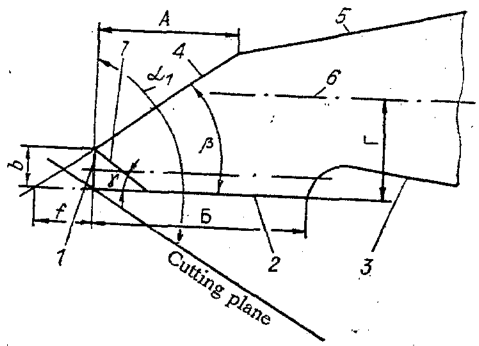


Fig. 5. Design of the working part of a tooth with optimal parameters (*1* — cutting edge; *2* — linear portion of the back facet; *3* — curvilinear back facet; *4, 5* — segments of the front facet; *6* — longitudinal tooth axis; *7* — wear platform; b = cutting edge width; α_1 = initial cutting angle; β = wedge angle; f = distance between wedge angle vertex and cutting edge; A and B = dimensions of linear segments of front and back facets, respectively; Γ = displacement of back facet segment from tooth axis; γ = wear platform slope angle.

$$b = \frac{P_1 \cdot \sin\gamma}{P \cdot D \cdot i},$$

where b is an efficient width of the cutting edge; P_1 is the excavator thrust force, which consists of the weight of bucket and the stick, and the force developed by the thrust mechanism; γ is the slope angle of the wear platform relative to the tooth axis (or the linear segment of the back facet); D is the length of the cutting edge; P' is the pressure on the wear platform at the beginning of the second stage; and i is the number of the teeth on the bucket.

Cutting edge 1 should be at distance

$$f = \frac{P_1 \cdot \sin\gamma}{\beta},$$

from the vertex of the wedge angle, where β is the wedge angle of the working part of the tooth.

Literary data indicate that a change of the cutting angle (more precisely, the back angle, which depends on the cutting angle) greatly affects the intrusion force of cutting elements. When the back angle of a tooth is increased, the energy capacity of its intrusion into the ground tends to decrease [7]. We formulated the new tooth geometry taking this factor into account. Accordingly, linear segment 2 or back facet 3 is parallel to tooth axis 6, which allowed us to increase the back angle by a factor of 2.0-2.5 compared with the mass-produced model. To reduce the wear of the horizontal component of the cutting parameter of the excavator bucket, we shifted segment 2 of the back facet (and thus cutting edge 1) by value Γ from the tooth axis. This position of the elements of the tooth working component relative to the bucket cutting edge reduces the rate of wear because the distance between the tooth cutting plane and the bucket edge cutting plane is increased by 70-80% for a given length of the working tooth part protruding beyond the bucket. For this tooth design, we defined the relationship which can be used to calculate the dimensions of the working elements (Table 1).

The length of the cutting edge (D) and the number of teeth (i) are chosen depending on the design of the excavator working element and the general machine specifications.

Figure 6 shows theoretical curves of the formation of wear platform dimensions as a function of linear wear for an ÉKG-5A excavator. We can see that at zero wear the design with efficient parameters has a wear platform of $\approx$50 mm. The design produced by the Uralmash Production Association attains the desired dimensions only after significant linear wear U_{cr}. With further wear (Fig. 6, zone II) the wear platform evolves less rapidly and the wear rate is approximately equal to that of

TABLE 1

Parameter	Dimensions of working component elements	Calcualted parameters of ÉKG-5A excavator tooth (4, 6)
Cutting edge width	$b = \frac{P_1 \cdot \sin\gamma}{P' \cdot D \cdot i}$	$b = 28$ mm
Distance between vertex of tooth wedge angle and cutting edge	$f = \frac{P_1 \cdot \sin\gamma}{P' \cdot D \cdot i \cdot \mathrm{tg}\,\beta}$	$f = 40$ mm
Wedge angle of working part of tooth	$\beta = 33 \ldots 37°$	$\beta = 35°$
Wear platform slope angle	$\gamma = 40 \ldots 45°$	$\gamma = 40°$
Length of linear segment of back facet	$B = (4{,}5—5{,}8)b$	$B = 150$ mm
Length of linear segment of front facet	$A = (3{,}0—4{,}0)b$	$A = 100$ mm
Displacement of linear segment of back facet from tooth axis	$\Gamma = (2{,}0—3{,}5)b$	$\Gamma = 70$ mm

the tooth made by Uralmash. In other words, during wear stages, wear resistance in zones I and II accumulates a certain reserve because the wear platform of a new tooth develops more rapidly in the initial stages (up to 50 mm). Subsequently, a change in wear platform size has no significant influence on wear rate.

The principles for working component development and parameter evaluation which make it possible to predict and control the wear dynamics of cutting elements were tested in real industrial conditions. The Orotukan Mining Machine Factory of the Severovostokzoloto Gold Mining Production Association manufactured a test batch of excavator teeth following the new design. They were tested at the Yagodnin, Berelekh, and other mining enterprises. Valuable test results were obtained at the Korba facility at the Burkandya Mine (Berelekh Enterprise), where teeth were used to excavate highly abrasive frozen rocks consisting of granite, sandstone, and clay schist fragments at surrounding air temperatures of $-45°C$.

Analysis of the design-related wear resistance of these experimental teeth indicated that, with the new working component geometry they experience single-stage wear with a resistance at least 40% higher than that of standard wedge teeth (Fig. 7). Experimental data were analyzed to define linear wear U as a function of work output V (thousand m^3) for new teeth. It was expressed by a first-degree regression equation:

$$U = 8.57 + 5.62V$$

with a correlation coefficient of 0.980. The test confirmed the basic design principles of cutting elements based on an empiric relationship of the working element geometry with pressure on the wear platform expressed in terms of the force characteristics of excavator working components.

Cutting edge width can be defined from

$$b = U_{P_{cr}} \cdot \sin\gamma, \tag{1}$$

where ..... is the wear is the wear platform for the onset of steady-state tooth wear, which is defined from a chart (see Fig. 1); γ is the wear platform slope angle.

Using the empiric relationship (see Table 1) and rearranging it with substitution of numeric values of P' based on tests, we can estimate the cutting edge width as

$$b = 0{,}0536\frac{P_1}{D \cdot i}, \tag{2}$$

which guarantees single-stage wear in any operation conditions. This is achieved because the expression contains a constant coefficient (pressure $P' = 12$ MPa) which serves as the main criterion. At the beginning of linear wear, a specific pressure of ≈ 12 MPa operates on the wear platform of a cutting element designed according to this formula.

This analytic technique is simple and reliable because it provides a wear resistance margin. This is important for excavator teeth used to cut bedrock, which are usually rapidly blunted. The sharpness of teeth in this case is of little practical importance because the wear intensity is extremely high. A similar positive effect of this method can be expected for teeth used to cut frozen rocks, which are characterized by strong resistance to intrusion leading to back facet wear and formation of a wear platform that "sharpens" the teeth (reduces cutting edge).

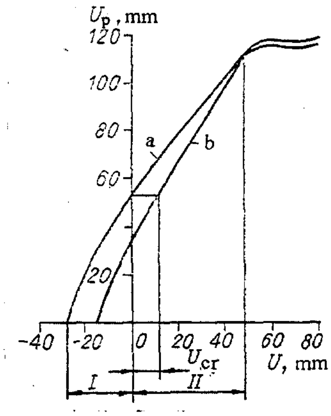


Fig. 6

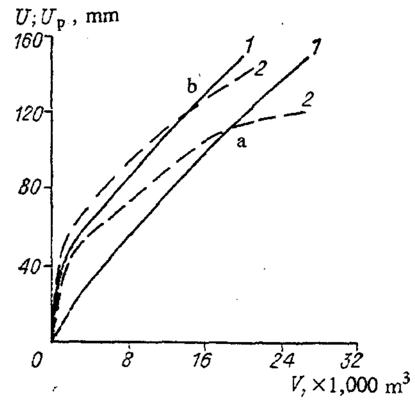


Fig. 7

Fig. 6. Variation of wear platform dimensions as a function of linear wear (a — formation of wear platform of cutting element with improved parameters of working part; b — formation of wear platform of Uralmash tooth design; I, II — development zones of wear platform before onset of linear wear (I) and during linear wear (II).

Fig. 7. Design-controlled wear resistance of ÉKG-5A excavator tooth (wear dynamics of new tooth design (a) and standard wedge tooth (b); *1, 2* — linear wear and wear platform formation, respectively).

TABLE 2

Characteristic	Excavator			
	ÉKG-5A(4,6)	ÉKG-8I	ÉKG-12,5	ÉKG-20
Excavator parameters:				
thurst mechanism force, MN	0,205	0,37	0,60	0,70
weight of bucket with stick, tons	20,0	26,5	45,0	81,5
total pressure force, MN	0,405	0,635	1,05	1,565
number of teeth per bucket, units	5	5	5	6
Conventional teeth:				
basic length of cutting edge, mm	150	180	180	220
basic width of cutting edge, mm				
wedge-shaped teeth with wedge angle 20-30°	12	16(40)	20	45 *
teeth with curved back and front facets	(16)	—	—	—
New design:				
calculated width of cutting edge, mm				
according to (1)	28	34	44	44
according to (2)	28	37	63	64

*Wedge angle of working component 38°.

The teeth parameters formulated with our method and experimentally tested can be compared with currently used models on quarry excavators (Table 2).

The data in Table 2 clearly show that most current designs do not meet the needs of frozen rock excavation. Our studies and calculations confirm this observation. Standard teeth with increased cutting edge dimensions (given in parentheses) are now used on a limited basis because they are only available on excavators manufactured by the Uralmash and Izhorskii factories.

We should note that the improvement in tooth wear resistance which can be attained by efficient design has certain limitations. This potential is restricted by the influence of cutting element size and on the energy intensiveness of the rock breaking process, and requires special investigation.

The tooth design described in the present paper was adopted by the Uralmash factory and the Krasnoyarsk Industrial Association for manufacture of ÉKG-12, ÉKG-5, and ÉKG-15 excavators, and by the Orotukan Mining Equipment Factory of the Severovostokzoloto Gold Mining Production Association.

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