Possible ways to expand range of rock strength, destroyed by cutter bit

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Abstract. The strength analysis of the rotary tangential cutter in the conditions of rock destruction has shown that the stresses in the most dangerous section of the tool (at the junction of the head and shank of the holder) are distributed, mainly, near the surface. This makes it possible to increase the overall strength of the tool by making its holder in the form of a bimetallic construction, the shell of which is made of high-strength material, and the core of a highly viscous, not necessarily durable material. By the example of the RKS-2 it is shown that the use of a bimetallic holder extends the range of tool work on rocks with a strength up to \( f = 8 \).

1. Introduction

The mechanical method of rocks destruction, carried out with the help of shearer and tunneling combines, is one of the most common methods of mining. The working tool of the combine's executive body is a cutter bit intended for separating chips from the face of the rock by cutting as a result of continuous static impact and moving the tool.

At present, both in our country and abroad, the mechanical method is used for the destruction of rocks by strength only \( f = 7 - 10 \) on the scale of prof. M.M. Protodyakonov. Stronger rocks are destroyed explosively - more dangerous, laborious and less ecological than mechanical (Table 1).

Table 1. The maximum strength of a rock to be destroyed by tangential incisors of different manufacturers

| Company manufacturer | Sandvik | KMW | Mining tools |
|-----------------------|---------|-----|-------------|
| Cutter brand          | P8AA-3084-5690 | RKS-2 | RSH32-70L72/19 |
| \( f_{\text{max}} \)  | 7       | 7   | 7…10            |
| Holder diameter in the tool holder | 30 | 32 | 32 |
wear resistance, being subjected to intensive abrasive wear (Figure 1, a), which increases with the strength and abrasiveness of the rock, which leads to the subsequent breakdown of the carbide insert (Figure 1, b) with loss of the cutter efficiency.

Figure 1. Type of rotary tangential incisors after operation on rocks of medium strength and abrasiveness (a) and after breakage carbide inserts operated (b)

Thus, the development of the cutter holder design, which is distinguished by its high strength, wear resistance and satisfactory resistance to impact, is an important step in expanding the range of rock hardened by the mechanical method. Based on the stresses distribution in the body of the tool during its operation as a part of the tunneling combine, in this work the construction of the holder having the above properties is justified.

The turntable type fixed in the tool holder can be represented in the form of a rod of variable cross section. One end of it is clamped, and the concentrated components of the cutting force $Z_{\text{max}}$ and the feed rate $Y_{\text{max}}$ act on the other one during the contact with the rock (Figure 2), so that the cutter is subjected to simultaneous action of compressive and bending loads.

Figure 2. The scheme of the forces acting on the cutter (a) and the diagrams of longitudinal and shear forces and bending moment (b): 1 - reinforcing insert, 2 - cutting tool, 3 - tool holder, 4 fracturable rock, $Z_{\text{max}}$ - maximum cutting force, $Y_{\text{max}}$ - Maximum compressive force, $F_{\text{comp}}$ - compressive force, $F_{\text{bend}}$ - bending force, $\Theta$ - cutting angle
The result of the action of the compressive load is the appearance of normal stresses $\sigma_N$, which are distributed uniformly in the cross sections of the tool $\sigma$ (Figure 3) along its entire length.

![Figure 3. Diagrams of stress distribution in the cross section of the tool](image)

Under the action of a bending load, tangential stresses $\tau_{yz}$ arise in the cutter, varying in cross section height according to the law of a quadratic parabola, the normal stresses $\sigma_{\text{bend}}$ being maximal near the surface and increasing along the length of the tool, are proportional to the bending moment to the value $\sigma_{\text{max}}$.

2. Bimetallic cutter bit
Cutter RKS-2, the production of "Kopeysk Machine Works", has been selected as a calculation object, designed for use on rocks $f = 7$ [1, 2]. As shown by the calculation of stresses arising in different sections of the tool during its work on the rock of the maximum permissible strength, their maximum values are achieved in section A-A in Fig. 1, as a result of which the cross section should be considered as the most dangerous when calculating the holder for static strength. This conclusion is confirmed by the experience of operating rotary incisors [3, 4], which indicates that the breakage of the holders occurs, in most cases, precisely at the junction of their head and shank. At the same time, the value of $\sigma_{\text{bend}}$ at the attachment point of the holder in the fist (1350 MPa) is significantly (8-10 times) higher than the value of $\sigma_N$ (160 MPa) and $\tau_{yz}$ (105 MPa) [9].

Calculations showed that the maximum stresses $\sigma_{\text{max}} = \sigma_{\text{bend}} + \sigma_N$ are reached in the surface layers of the holder (Fig. 3). It was logical to assume that in the developed design of the holder, the surface layers material receiving the main load should have high strength properties (high value of the allowable stress $[\sigma_S]$) and hardness. Besides, the core, which does not experience significant static loads, can be much less strong (small value $[\sigma_C]$), but high viscosity (high impact strength), to compensate for impact loads acting on the cutter.

Such holder design can be achieved, for example, in two ways:
1. The application of surface hardening at a given depth for a holder made of steel, which is sufficiently viscous in the annealed state, and in hardened by high strength and hardness [8].
2. Manufacturing of the holder in the form of an integral bimetallic structure, in which a strong and hard shell, for example, of steel U8 in the quenched state, is connected to a viscous core (12Kh18N10T steel)[10].

For both cases, the ratio between the diameters of the hard shell ($D_S$) and the viscous core ($D_C$) can be determined from the diagram (Figure 4) and equality

$$D_c = \frac{D_s \cdot (\sigma'_k - \sigma_S)}{\left(\sigma'_k - \sigma_S\right)}.$$  \hspace{1cm} (1)

where $[\sigma_S]$, $[\sigma_C]$ are taken outside the yield strength of shell and core materials, MPa.
Figure 4. Distribution of stresses in the most dangerous section of the tool for the existing (on the left) and bimetallic (on the right) holder.

In the case of a shell made of steel U8 ([σ]_S = 1250 MPa), and the cores of 12Kh18N10T ([σ]_C = 300 MPa [6]), the diameters D_c, D_S are correlated as 1: 6.

The maximum strength of the rock, destroyed by the RKS-2 cutter, in which the holder is made in the form of a bimetallic construction (D_c:D_S = 1:6), was calculated from the design stresses arising in the longitudinal section of the tool under the action of compressive and bending forces when the tool was operated on rocks of different strength. The results were compared with the yield point of the existing material of the holder (steel 30KhGSA [7]) and the shell material (steel U8) of the proposed bimetallic structure.

In the calculation, the formulas and values of the parameters from [5] were used, based on the maximum cutting forces Z_max and the feed rate Y_max, acting on the crown cutter of the tunneling machine KP21, which experiences maximum loads:

$$Z_{\text{max}} = (1.5 \cdot K_a \cdot K_s \cdot K_p \cdot h_{\text{max}}^{1.6} \cdot K_t \cdot (1 + \frac{t_i}{h_{\text{max}}}) + 0.25 \cdot \mu \cdot F_{\text{blunt}}) \cdot k_{\text{max}} \cdot P_k,$$

where P_k - rocks contact strength (P_k = 44 \cdot \sqrt{f^3}), MPa; F_{\text{blunt}} - projection of the blunting area on the cutting plane (4 cm²); k_max - maximum overload factor (assumed to be 6); K_a - coefficient that takes into account the influence of the cutting angle of the tool when the rocks are destroyed (at \Theta = 45^\circ K_a = 1); K_s - coefficient, taking into account the influence of the shape of the cutting part of the tool (for rotary cutters, with the conical shape of the core and the head of the holder with a core diameter of 9 mm, K_s = 1); K_p - coefficient that takes into account the type of tool (for rotary cutters K_p = 1.5); t_i - cutting step (24 mm); h_{\text{max}} - maximum chip thickness (22.6 mm); \mu - cutting resistance coefficient \mu = \mu_t + 0.24 \cdot h_{\text{max}}, where \mu_t - coefficient of friction of the cutter about the rock (0.4).

$$Y_{\text{max}} = \left(0.4 + \frac{0.3}{h_{\text{max}}}\right) \cdot 1.5 \cdot K_a \cdot K_s \cdot K_p \cdot h_{\text{max}}^{1.6} \cdot K_t \cdot (1 + \frac{t_i}{h_{\text{max}}}) + 0.25 \cdot F_{\text{am}}) \cdot k_{\text{max}} \cdot P_k.$$

The resulting forces were converted into compressive and bending forces on the tool:

$$F_{\text{bend}} = Z_{\text{max}} \cdot \cos \Theta - Y_{\text{max}} \cdot \sin \Theta = 63kN;$$

$$F_{\text{comp}} = Z_{\text{max}} \cdot \cos \Theta + Y_{\text{max}} \cdot \sin \Theta = 131kN.$$
steel 30KhGSA (holder), as the existing material of the holder, and steel U8 – as the shell material of the proposed bimetallic construction. It can be seen from the comparison that with an allowable strength of the rock \( f \sim 7 \) for steel 30KhGSA, a bimetallic holder with a shell of steel U8 is operable at strength of up to \( \sim 8 \).

![Figure 5](image-url)

**Figure 5.** Stresses that arise in the body of the holder, depending on the strength of the rocks, in comparison with the permissible stresses for the analyzed steels

### 3. Auxiliary hub

Another possible way to expand the range of rock strength, which is destroyed by rotary incisors, is the use of an auxiliary bush, installed between the holder and the wall of the toolholder. The bush extends from the toolholder to the maximum possible height, limited by the maximum chip thickness and angle:

\[
[l_{as}] = l - \left( r_h \cdot \tan(\Theta) + \frac{h_{max} - (r_h - r_c) \cdot \cos(\Theta)}{\sin(\Theta)} \right),
\]

where \([l_{as}]\) – length of the protruding part of the auxiliary bush, mm; \(r_h\) – radius of the outer contour of the toolholder at the point of fastening of the tool in it.

For cutting in \( \Theta = 45^0 \) \([l_{as}] = 22 \) mm.

Installation of the protruding bush from the tool holder reduces the bending moment arm, thereby reducing the maximum bending load (Figure 6).
Figure 6. Stresses arising in the dangerous section of the holder with ($\sigma'_{\text{max}}$) and without ($\sigma_{\text{max}}$) the use of an auxiliary bush

As follows from the form of the curves in Fig. 6, the use of an auxiliary bush extends the range of fracturable rocks to $f = 10$ for a bimetallic tool holder and up to $f = 8$ for an existing one.

4. Conclusion
In conclusion, it is necessary to note that the proposed methods help to increase the strength of destructible rocks from 7 to 10 on the scale of M.M. Protodyakonov. In addition, the further research in this area is being carried out.

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