Working equipment of the single-bucket excavator for the development of frozen ground

A B Letopolsky*, P A Korchagin and I A Teterina
Siberian State Automobile and Highway University (SibADI), 644089, Omsk, Mira Ave., 5 Russia

*Antoooon-85@mail.ru

Abstract. The article presents a variant of improving the design of the working equipment of a single-bucket excavator. The proposed technical solution allows the development and sampling of solid and frozen soil without changing the working equipment from the bucket to the hydraulic hammer. The result is achieved by installing two hydraulic hammers on the front part of the bucket. Presents calculations of the hydraulic system of the excavator and hydraulic hammers, allowing to confirm the performance of the proposed solution. The methodology of technological operations is determined taking into account the use of the proposed technical solution.

1. Introduction
Any construction equipment is part of the "machine - operator - road - environment" system and its properties are manifested in interaction with the elements of this system. Therefore, the significance of a certain operational property depends on the conditions in which this property manifests itself, in other words, on operating conditions [1]. In general, the operating conditions are determined by roadless, road, transport and climatic conditions. For the extraction of hydrocarbon resources in conditions characterized as difficult-climatic, it is necessary to develop new or modernize existing construction equipment, including by improving its working equipment [2].

The process of building infrastructure for mining in the Arctic zone of the Russian Federation provides for various types of earthworks, which requires a change of work equipment, an excavator bucket for a hydraulic hammer. The proposed design solution will allow avoiding this operation and, accordingly, reducing the time and financial costs for the re-equipment of construction equipment, thereby increasing operating efficiency [3].

2. Researching
The single-bucket excavator is a cyclic digging machine, the main working body of which is a bucket with a cutting edge or teeth that carry out cutting of the soil. The working cycle of the excavator consists of successive operations: digging the ground (cutting with a set of soil into the bucket), lifting and turning the bucket from the face, unloading the soil on vehicles or into the dump and reverse turning and lowering to the bottom [4].

The hydraulic hammer can act as the replaceable equipment of the excavator. This working equipment has a wide enough scope and is used for the destruction of frozen and solid soils, reinforced concrete and concrete structures, opening asphalt concrete pavements, as well as rocky soils [5].
The most common among hydraulic hammers, received hydropneumatic hammers. The use of pneumatic accumulators in hydraulic hammers allows you to create devices that provide the realization of impact energy in a wide range [6].

In order to optimize the workflow and reduce the time spent on changing work equipment, an improved design of the working body of a single-bucket excavator has been proposed. The design solution combines a bucket and two hydraulic hammers located on the front side of the bucket. General view of the working body is presented in Figure 1.

![Figure 1. Bucket with 2 hydraulic hammers: 1 - top view; 2 - side view](image)

The proposed version of the working equipment includes: 1 - rear wall; 2- stand for hydraulic hammer; 3- tooth; 4 - tooth bracket; 5 - left bracket; 6 - right bracket; 7 - hammer; 8 - side wall (right). The organization of work using the proposed equipment is presented in Figure 2.

![Figure 2. Issues of work carried out using the proposed equipment: 1 - Excavator works with hydraulic hammers, breaks frozen and solid ground; 2 - The excavator works with a bucket tool, collects the loosened soil and unloads it into the dump.](image)

To perform work with lower energy consumption, the hydropercussion device must have certain parameters set in accordance with existing recommendations (energy change, impact frequency when changing the type of work, etc.).
When soil is destroyed, the stroke of the moving parts of the hydraulic hammer is important, taking into account the parameters of the soil (especially with a direct impact on the ground) to increase the efficiency of its destruction. The stroke rate of the moving parts, in turn, depends on the frequency of impacts; the recommended stroke value of the moving parts of the hydraulic hammer for soil destruction is 75 ... 250 mm [7].

To confirm the performance of the proposed equipment and determine its parameters, the hydraulic system of the excavator's working equipment was calculated, taking into account the capabilities of the basic machine and the designation of a hydraulic hammer (soil compaction, frozen soil development, coating destruction, crushing of oversize, etc.) [8].

The working cycle of a hydropneumatic hammer consists of two periods: idling (platoon) of the peen, accompanied by gas compression in a pneumatic accumulator, and stroke (acceleration) of the peen, which occurs when the gas of the pneumatic accumulator expands [7].

The required value of the energy of a single blow for the working stroke of hydraulic hammers is determined by the formula [9]

$$ W = \frac{T}{\eta}, $$

where $W$ is the energy developed by the pneumatic accumulator during gas expansion, J; $T$ is the required kinetic energy of a single blow, J; $\eta$ is the efficiency of acceleration of a hydropercussion device, taking into account energy losses due to fluid flow and mechanical friction during acceleration of the striker (the value of $\eta$ can be taken 0.6 ... 0.8).

The energy developed by a pneumatic battery depends on its parameters and is calculated by the formula [10]

$$ W_p = P_g V_g \left( \frac{E_g^n - E_g}{E_g} \right), $$

where $W_p$ is the energy of the pneumatic accumulator, J; $P_g$ - maximum pressure of compressed gas in a pneumatic accumulator, Pa; $V_g$ is the volume occupied by the gas at pressure; $n$ is the indicator of polytropes, ($n = 1.25 ... 1.65$); $E_g$ - compression ratio of the reptile, ($E_g = 1.5 ... 3.5$).

The maximum gas pressure depends on the pressure of the pneumatic accumulator and is determined by the expression [9]

$$ P_g = P_{go} E_g^n, $$

where $P_{go}$ is the charging pressure, ($P_{go} \geq 0.5 ... 1.5$ MPa).

The mass of the peen (moving parts) of the hydraulic hammer is determined based on the required energy of a single blow (taken within 4 ... 8 m/s) by the formula [10]

$$ m = 2 \frac{T}{\theta^2}, $$

where $m$ is the peen mass, kg; $T$ is the kinetic energy of impact, J; $\theta^2$ - peen velocity at the moment of impact, m/s.

Such geometrical parameters of the hydraulic hammer as piston diameter, stem diameter, working areas of the charging, discharge planes depend on the maximum pressure value gas in the pneumatic accumulator and the nominal pressure in the hydraulic drive of the base machine. The maximum pressure in the charging plane of the hydraulic hammer should not exceed the rated pressure of the working fluid in the hydraulic drive of the base machine [11].

Piston and rod diameters are selected taking into account the used diameters of pistons, rods in hydraulic cylinders of excavators (for unification of sealing elements of rubber rings, cuffs), and also taking into account design parameters, for example, to provide the required mass of moving parts of a hydraulic hammer [12].

The working bore sections of the hydraulic hammer control unit are selected from the condition of ensuring minimum hydraulic pressure loss when the working fluid flows during the striking stroke.

The values of the radial deformation of the elastic locking element are used in the range of 2 ... 5 mm. The thickness of the elastic element is within 3 ... 7 mm [13].
Rubber is used as the material of the elastic element, for example, 110-68-1, modern elastomers, composite and other materials [13].

The frequency of hydraulic hammer strikes depends on the cycle time, which is affected by the stroke of the striker, the working area of the cocking cavity, and the pumping power of the base machine.

The time of the ideal working cycle of a hydraulic hammer (without taking into account the delay time of the platoon) is determined by the formula [14]

\[ T_c = \frac{S_c \cdot l}{Q} \cdot (1 + \frac{1}{k_a}) \], \hspace{1cm} (5)

where \( T_c \) is the cycle time, s; \( S_c \) - working (effective) area of the charging cavity, \( m^2 \); \( l \) - stroke of moving parts, m; \( Q \) - flow rate of working fluid, equal to the flow rate of the base machine pump, \( m^3 / h \); \( k_a \) - coefficient of asymmetry, \((k_a = 5 ... 10)\).

The frequency of blows is determined [14]

\[ n = \frac{1}{T_c} \]. \hspace{1cm} (6)

Having determined the energy and frequency of impacts, the effective impact power was determined by the formula

\[ N_p = T \cdot n. \]

The efficiency of the hydraulic hammer can be approximately determined by the ratio of the effective impact power to the rotary power of the pump of the base machine [14]

\[ \eta = \frac{N_{ep}}{N_p}, \] \hspace{1cm} (7)

where \( N_{ep} \) is the effective shock power, W; \( N_p \) is the useful power of the pump, W.

The net power of the pump is determined by the formula

\[ N_n = P_{pr} \cdot q, \] \hspace{1cm} (8)

where \( P_{pr} \) is the average pressure in the platoon cavity, defined as the arithmetic average for the beginning and end of the platoon period, Pa; \( q \) - pump flow, \( m^3/s \).

The power of the hydraulic drive is determined by the specified loads and speeds of the hydraulic motors, providing the actuator drive. The net power of the hydraulic motor of reciprocating action (hydraulic cylinder) is calculated by the formula [15]

\[ N_{mp} = F \cdot \vartheta, \] \hspace{1cm} (9)

where \( N_{mp} \) - hydraulic motor power, kW; \( F \) - force on the hydraulic cylinder rod, kN; \( \vartheta \) - the speed of movement of the rod m/s.

The resulting pump power is determined from the power of the hydraulic motor, taking into account the energy loss during its transfer from the pump to the hydraulic motor [15]

\[ N_{pp} = k_f \cdot k_s \cdot N_{mp}, \] \hspace{1cm} (10)

where \( N_{pp} \) - pump power, kW; \( k_f \) - factor of force, (1.1 ... 1.3); \( k_s \) - safety factor for speed, (1.1 ... 1.3); \( N_{mp} \) - hydraulic motor power, kW.

Knowing the necessary full capacity of the pump, and given that the useful capacity of the pump is related to the nominal pressure and flow dependency

\[ N_{pp} = p_n \cdot Q_n, \] \hspace{1cm} (11)

pump volume can be determined by the shape

\[ Q_n = \frac{N_{pp}}{p_n}, \] \hspace{1cm} (12)

\[ q_n = \frac{N_{pp}}{p_n n_p}, \] \hspace{1cm} (13)

where \( p_n \) is the nominal pressure, MPa; \( q_n \) - pump displacement, dm3; \( n_p \) - the frequency of rotation of the pump shaft, s\(^{-1}\).

In the proposed work equipment, the structural elements of hydropercussion devices with hydraulic cylinders were unified with other elements of the basic machine.
In accordance with the technical characteristics of the selected pump, the actual pump flow has been clarified [9,16]

\[ Q_{pa} = \dot{V}_{pa} \cdot n_{pa} \cdot \eta_v \]  

(14)

where \( Q_{pa} \) is the actual pump flow, dm³/s; \( \dot{V}_{pa} \) - the actual working volume of the pump, dm³; \( n_{pa} \) - the actual frequency of rotation of the pump shaft, s⁻¹; \( \eta_v \) - the volumetric efficiency of the pump.

The calculated values of the internal diameters of the suction, pressure and discharge lines are determined from the equation of continuity of fluid flow [10,17]

\[ d_p = \sqrt{\frac{4 \times 10^{-3} \cdot Q_{pa}}{\pi \cdot v_f}} \]  

(15)

where \( d_p \) is the calculated value of the internal diameter of the hydroline, m; \( Q_{pa} \) - actual fluid flow rate, dm³/s; \( v_f \) - velocity of the fluid in the hydroline, m/s.

In accordance with GOST 8734-75, the standard values of the inner diameters of the hydrolines are selected with a wall thickness of 3mm: the diameter of the suction line (\( d_{sl} = 32 \) mm); the diameter of the pressure line (\( d_{pl} = 16 \) mm); the diameter of the drainage line (\( d_{dl} = 26 \) mm) [18].

The actual speed of the fluid in the suction, pressure and discharge lines is determined by the formula [19]

\[ v_f = \frac{4 \times 10^{-3} \cdot Q_{pa}}{\pi \cdot d^2} \]  

(16)

where \( d \) - the actual value of the internal diameter of the hydroline, m.

3. Conclusions
In the course of the theoretical studies, calculated dependencies were obtained to determine the parameters of the installed hydraulic hammers. The above formulas allow you to calculate the time of the operating cycle, the frequency of impacts and the shock power of installed hydraulic hammers.

The obtained calculated dependences suggest that the proposed design of the working equipment of a single-bucket excavator meets all the necessary requirements and operational standards applicable to the development of frozen and solid soils, and earthworks.

The proposed design solution increases the performance and operational reliability of the working equipment of the excavator in the development of frozen and solid soils, as well as asphalt concrete and cements concrete pavements.

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