Designing Hydraulic Impact Devices for Low-Temperature Operation

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Abstract. The paper is devoted to the results of theoretical studying the hydraulic jackhammer operation under various temperature conditions and practical recommendations for their design and operation, considering the temperature factor.

1. Introduction
When working under the RF President grant MK-6405.2013.5, theoretical studying the hydraulic jackhammer operation was performed and its output characteristics were obtained in the form of temperature dependences of the impact frequency, energy, and power, and efficiency [1, 2, 3, 4, 5, 6, 7]. Temperature change primarily affects the viscosity characteristics of the pressure fluid, but since viscosity is controllable hardly during operation, the temperature dependence of the hydraulic hammer output characteristics is of practical interest. Also, considering that the hydraulic hammer comprises a hydro-pneumatic accumulator, then a temperature change will also affect the gas state in the gas cavity.

Hydraulic hammers are widely used in the development and extraction of minerals, crushing rocky and frozen soils, and loosening frozen soil [8]. Herewith, a hydraulic hammer installed on a base machine operates under low-temperature conditions, which affects its performance. This should be considered when designing hydraulic hammers to be operated under such conditions.

2. Developing the Technique
The developed technique for the engineering design of the parameters of a hydraulic hammer with a controlled power stroke chamber (Fig. 1) considers the temperature factor. The striker weight is determined from the condition that by the stroke end, it should reach the impact energy \( A \) and the permissible pre-impact velocity \( \nu \) = 8...10 m/s.

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\[
m_s = \frac{2A}{[\nu]^2}
\]

Pressure loss in the pressure line

\[
\Delta p_{p.l.} = \lambda_{p.l.} \frac{L_{p.l.} \nu_{p.l.}^2}{d_{p.l.}^2 \rho}
\]

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**Figure 1.** The Hydraulic Hammer Design Scheme:

1 - tool, 2 - striker, 3 - distributor, 4 - accumulator, 5 - reverse stroke chamber seal, 6 - power stroke chamber seal, $S_r$ - reverse stroke chamber area, $S_p$ - power stroke chamber area, $p_{p.l}$ and $p_{d.l}$ - pressure in pressure and discharge lines, respectively, $L_{run}$ - striker run, $d_1$-$d_3$ - diameters of the corresponding striker steps.

The loss ratio

$$\lambda_{p,l} = 0.316 \cdot 4 \frac{v(T)}{d_{p,l} v_{p,l}}$$

The striker's motion velocity during the power stroke is such that the flow rate $Q_n$ of the powering hydraulic station is not enough for the hydraulic hammer's normal operation. The flow rate deficit is compensated by a hydro-pneumatic accumulator. Consequently, in the pressure line, the fluid flow speed is

$$v_{p,l} = \frac{4Q_n}{\pi d_{p,l}^2}$$

The pressure in the hydraulic hammer chambers will depend on the network hydro-pneumatic accumulator state and vary within $p_{min}$ to $p_{max}$ depending on its filling with fluid

$$p_{p,l} = (p_n - \Delta p_{p,l}) \frac{e + 1}{2e}$$

where $e$ is the compression ratio equal to 1.2 for most hydraulic hammers.

$$e = \frac{p_{max}}{p_{min}}$$

The striker run length is determined by the permissible repulsive force acting on the base machine

$$L_{run} = \frac{A}{R_{rep} \frac{e + 1}{2e}}$$

The approximate repulsive force is chosen considering the impact energy

$$R_{rep} \sim (10...12)A$$

During the striker power stroke, friction force in the cups:

$$F_{fr,ps} = \pi (d_1 + d_3) k_d k_H H p_{p,l} f_{fr,c}$$

where $k_d$ is the cup diameter increasing coefficient (considers the cup collar thickness); $k_H$ is the cup height coefficient (considers the pressure distribution over the cup height only given its design
features); $f_{fr,c}$ is the friction factor; $d_1$ and $d_3$ are the diameters of the corresponding striker steps, $H$ is the cup height (since the striker step diameters differ insignificantly, all the cup design parameters can be assumed to be the same).

Working area

$$S_w = \frac{1}{p_{p.l.}} \left( \frac{A}{L_{run}} + F_{fr,ps} \right)$$

The average striker velocity during the power stroke

$$v_{av,run} = \frac{|v_s|}{2}.$$

The run time for which the striker moves to the run length from the extreme point to the moment of collision with the working tool is determined by the equation

$$t_{run} = \frac{m_s|v_s|}{p_{p.l.}S_wF_{fr,ps}}.$$

The accumulator working volume

$$\Delta V_a = S_wL_{run}.$$

The total volume of the accumulator’s gas cavity:

$$V_ga = e^{\log \left( \frac{1}{k_p} \right)n^{-1}} \cdot \Delta V_a \left( e^{\log \left( \frac{1}{k_p} \right)n^{-1}} - 1 \right)^{-1}$$

where $n$ is the polytropic index taken equal to 1.4 for hydro-pneumatic accumulators of hydraulic hammers; $k_p$ is the pre-charge factor.

The accumulator pre-charge factor

$$k_p = \frac{p_{vol}(T)}{p_{p.l.}},$$

where $p_{vol}$ is the pre-charge pressure.

When operating the hammer at low temperatures, it should be considered that in the general case, the pre-charge pressure and, consequently, the pre-charge factor depends on the gas temperature in the gas cavity. During the long-term operation, the pressure fluid heats up, causing the heat transfer to the gas (especially in diaphragm or bladder accumulators). To reduce the impact of the pre-charge factor on the hydraulic hammer output characteristics, the hydro-pneumatic accumulator gas cavity volume should be increased [6, 8].

During the reverse stroke of the striker, the accumulator should accumulate the required fluid reserve consumed during the power stroke, i.e., the accumulator charging time and the reverse stroke time should be equal.

The accumulator charging time

$$t_{ch} = \frac{\Delta V_a}{Q_n}.$$

The reverse stroke time during which the striker makes a translational motion from the initial (the tool) to the final point, while passing a distance equal to the run length, will be

$$t_{rs} = t_{ch}.$$

The average striker velocity during the reverse stroke

$$v_{av,rs} = \frac{t_{run}}{t_{rs}}.$$

The friction force in the cups during the reverse stroke of the striker
\[ F_{fr,rs} = [(d_1 + k_d)p_{p.l} + (d_3 + k_d)p_{d.l}]\pi k_H f_{fr,c} \]

To determine the reverse stroke chamber area, the equation of equilibrium of the acting forces should be expressed, considering the inertia forces and the pressure loss in the drainage line, the argument of which will be the reverse stroke chamber area. One of the equation roots will be the desired value:

\[
f(S_{rs}) = \frac{m_e v_{av,rs}}{t_{rs}} - p_{p.l} S_{rs} + F_{fr,rs} + \\
+0.158 \frac{L_{d.l}}{d_{d.l}} \rho \left( \frac{v(T)}{d_{d.l}} \right)^{0.25} \left( \frac{4v_{av,rs}}{\pi d_{d.l}} \right)^{7/4} (S_w + S_{rs})^{11/4}.
\]

According to previous studies and engineering experience, the desired value will be within 10^{-3}–10^{-4} m².

The power stroke chamber area:

\[ S_{ps} = S_{rs} + S_w. \]

Cycle time is the sum of the reverse and power stroke times:

\[ t_c = t_{run} + t_{rs}. \]

The impact frequency:

\[ n_{im} = \frac{1}{t_c}. \]

Efficiency:

\[ \eta = \frac{A_{n_{im}}}{p_n Q_n}. \]

3. Results

Based on the study results, the following recommendations have been formulated for the design and low-temperature operation of hydraulic hammers [4, 5], which allow improving the efficiency of the hydraulic jackhammer with a controlled power stroke chamber and, accordingly, the destruction of rocky and frozen soils [9]:

- An oil with the appropriate viscosity characteristics should be chosen depending on the minimum operating temperature, the work intensity, and the warming-up time. For the startup and operation at the lowest temperatures, it is advisable to choose a low viscous oil (VG-15). This oil will allow starting the base machine of the hydraulic hammer at the shortest warming up and operating it in a less intensive mode when the hydraulic hammer is not switched on frequently, which leads to its rapid cooling.

- For operation at highly negative temperatures, an oil with viscosity characteristics like that of SHELL Tellus Oils T-15 is suitable.

- For long-term low-temperature operation with oil heating before startup, to obtain high output characteristics, more viscous oils of the VG-22 viscosity grade are suitable such as SHELL Tellus OilsS-22.

- To reduce friction forces and improve the output parameters of a hydraulic hammer designed for low-temperature operation, the striker and body should mate with an increased clearance (at least 0.025–0.040 mm).

4. Recommendations

Depending on the operating temperature and the oil chosen, the hydraulic hammer can be started immediately after starting or warming up the base machine. Herewith, it should be considered that immediately upon starting the base machine, the hydraulic hammer output characteristics will be...
several times lower than when it is warmed up. Thus, the impact energy and the impact power may be more than 5 and 11 times lower, respectively [6].

It is advisable to warm up the hydraulic hammer after warming up the machine [3, 10, 11, 12, 13, 14]. Despite the high oil temperature in the base machine hydraulic system, after warming up, it should be considered that the hydraulic hammer itself and, accordingly, the oil in it remain cold, so the maximum performance will not be achieved immediately. The time of the total hydraulic hammer warming up depends mainly on the base machine warming-up time, which differs for different base machine types [10]. It should be considered that the pressure fluid temperature stabilizes (the heat balance occurs) at lower temperatures than those of normal operation and reduces with decreasing ambient temperature.

For effective low-temperature destruction of rocks, minerals, and other materials, it is advisable to startup a hydraulic hammer with impact energy at least 30% lower than its rated value [11, 15, 16, 17, 18].

The common algorithm for choosing the hydraulic hammer operating parameters under various temperature conditions is shown in Figure 1.

The following features of the low-temperature hydraulic hammer operation should also be considered. Low temperatures negatively affect rubber and other elastic parts of the hydraulic hammer such as seals [19]. For normal hydraulic hammer operation, seals should be used that can withstand high velocities of moving parts (up to 10 m/s) and low temperatures. The same requirements apply to the diaphragm of the hydro-pneumatic accumulator. Since the diaphragm should withstand significant loads, the requirements for it at low temperatures increase significantly. When using a piston hydro-pneumatic accumulator, the hydraulic hammer reliability improves [18, 20, 21].

As the temperature decreases, the impact toughness of the hammer manufacturing materials reduces. To avoid rapid wear and destruction of parts, the pressure fluid should be warmed up before starting the hammer. With decreasing temperature, the linear dimensions of the parts also decrease. In a hydraulic hammer, the fastening bolts and tightening remain cold since they are located on the outer
body surface, while the body and some other parts warm up, which leads to significant thermal deformations and increased stresses. The deformations should be compensated by reducing the tightening torque of the threaded connections.

The given results replenish the area of studying hydraulic jackhammers operating at low temperatures.

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