Dynamics of the ground hydraulic hammer-pulsator

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Abstract. Various vibration mechanisms are used as vibration exciters for the beneficial use of vibration. By the type of drive, vibration mechanisms are divided into mechanical, electromechanical, hydraulic, pneumatic, vacuum-compression, electromagnetic and magnetostrictive. They are designed to excite forced vibrations that provide the required technological effect. Vibrators and vibrating hammers are currently used for vibration drilling of shallow wells in soft rocks, for vibration-rotating drilling in rocky and semi-rock formations, for driving and removing casing pipes and eliminating accidents associated with stuck drilling at the bottom of the well. The structural scheme of the ground hydraulic hammer-pulsator is designed for drilling seismic wells, the striker of which is a two-stage piston. The piston is connected to the atmosphere; the large piston is for studying the influence of the hydraulic hammer-pulsator on the drilling efficiency is equipped with a valve that closes the channels of the large piston in the upper position, and opens - in the lower position. A schematic diagram of a surface hydraulic hammer is shown to study the effect of a hydraulic hammer-pulsator on the drilling efficiency.

1. Introduction

Seismic drilling, carried out to obtain blast wells, is characterized by the penetration of small (usually up to 100 m), but significant diameter (up to 150 mm) wells. The following drilling rigs are applied for this purpose:

a) hard-to-reach areas (highly rugged, taiga, swampy, etc.) - URB-1S, URB-1V, URB-2T, AVT-T, UTTTB-T;

b) areas accessible for the movement of off-road vehicles - UKB2-100, URB-2L, URB-2A-2, AVBZ-100, USHB.

Portable drilling rig URB-1S is designed for drilling seismic wells using auger method, and annular and continuous bottom-well drilling with clay mud.

The URB-1V drilling rig is an exploration drilling rig of the normal row No. 1, self-propelled, on a caterpillar all-terrain vehicle, intended for drilling in hard-to-reach areas of seismic wells using the auger method, as well as annular and continuous bottom-well drilling with clay mud. The flush pump, driven by a V-belt drive from the drilling rig, is mounted on a single axle trailer with pneumatic wheels. The trailer is also used for transporting tools.
The drilling mechanisms are mounted on the GAZ-47 all-terrain vehicle. The drive of mechanisms, including the mud pump, is carried out from the propeller engine of the all-terrain vehicle, from which power is taken up to 50 kW.

The UKB2-100 drilling rig is designed for rotary drilling with bottom well washing up to 100 m deep. The drive motor and all mechanisms are fixed on a special frame, which allows the rig to be installed on ZIL-151 (131).

The USHB-T drilling rig (auger drilling on a tractor) is designed for drilling seismic wells with casing using a vibrator. Main mechanisms of the drilling rig are mounted on the S-80 tractor (T-100).

The drive of all mechanisms is carried out from the KDM-46 running tractor diesel engine, which has an all-mode speed controller in the range of 700–1000 rpm of the diesel engine, due to which the speed control of the winch, rotator and vibrator is possible. Hoisting operations are carried out by a mechanized winch using tackle equipment and a rotator, to which the lifted tool is attached.

The USHB-T-15 drilling rig is mounted on the S-80 (T-100) tractor and is designed for drilling seismic wells in an area inaccessible to vehicles by an auger method using a vibrator for planting and extracting casing pipes. Mechanical three-shaft vibrator is combined with a rotator; the frequency of vibrations created by the vibrator is from 400 to 1800 per minute.

Delves with a diameter of 600–1200 mm can be drilled under favorable conditions by the auger method to a depth of up to 50 m if the rig is equipped with a special tool.

The URB-2A drilling rig, self-propelled, is for drilling seismic wells with annular and continuous bottom wells with clay mud flushing. It is also used for drilling mapping and water wells.

The design feature of the URB-2A rig is double-drum winch. Both drums (one for feeding the tool while drilling, the other for tripping operations) are loosely mounted on a common shaft. The rotor table rotates on two thrust ball bearings and is equipped with a stopper to prevent rotation when screwing and unscrewing drill pipes.

The 11Gr drilling mud pump is installed on a frame with screws for tensioning the drive belts. Power take-off for the drive of the drilling rig mechanisms is carried out through the transfer case.

The tool feeding mechanism consists of transmission with a friction sleeve, a worm gear, a chain transmission, and a control. It provides up to 2000 kg down well pressure and full rotary kelly. The feed mechanism is driven by a V-belt from pulleys with a diameter of 140 mm of the rig main transmission.

The AVBZ-100 drilling rig is designed for drilling seismic, mapping wells and wells for water supply. The rotor is driven by the vehicle engine which it is mounted on. The rig is equipped with a metal mechanism using a push device.

The URB-2T drilling rig (exploration drilling rig of normal row No. 2 on a tractor) is mainly intended for drilling seismic and shallow mapping wells. It is recommended for use in hard-to-reach areas and in harsh climatic conditions.

The possibility of auger drilling, as well as drilling with a continuous and annular digging face with clay mud flushing or air blowing is provided in the design of the URB-2T drilling rig.

The URB-2A-2 drilling rig is for drilling geophysical (seismic) wells in a rotary way with cleaning the bottom well by flushing, blowing and transporting destroyed rock on the surface with augers.

The mechanisms included in the rig are mounted on their frame, attached to the ZIL-131 off-road chassis and are powered by its engine. The rig has a moving rotator with a hydraulic drive, which is used in the process of drilling, building up a drilling tool without breaking it off the bottom and performs, together with a hydraulic elevator, the work of lowering and lifting the tool and its feeding when drilling. The power and kinematics of the rotator also ensure the screwing and unscrewing of drill pipes, and consequently, there is no need for special mechanisms for this purpose. The rig control is fully hydraulic, including lifting and lowering the mast, which is mounted on the driller’s console.

Foreign companies “Mobile Drilling”, “GeoSPACE”, “Cardox”, “Webco”, “Akker Drill”, “Scar”, “Joy”, “Reich Drill”, “Byus Air” (USA), “Virt”, “Zeller Machine Factories” (Germany), as well as enterprises in Poland, the Czech Republic and Slovakia produce equipment that is widely used in the drilling of geological prospecting, geotechnical, and seismic wells.
Four basic standard sizes with a lifting capacity of 1.5–3; 3.5-5; 6-10 and 15-25 tons can be identified as a result of the analysis of technical characteristics among foreign drilling rigs. The first three sizes are intended for dry drilling with augers, spiral and spoon drills, as well as rotary drilling with flushing. Rigs with a lifting capacity of 15-25 t are mainly used in rotary drilling with flushing and blowing, as well as hammers.

The rotator drive is mostly carried out by means of hydraulic motors, which are sometimes used in combination with gearboxes. Two hydraulic motors of different power, included in the rotator kinematic diagram in parallel, are used to drive the rotator in the rigs of the “Virt” and “Zeller Machine Factories” companies (Germany). The rotator develops maximum torque at a reduced rotation speed in the case of simultaneous operation of two hydraulic motors. Turning off one of the hydraulic motors is accompanied by an increase in the number of the tool turnover with a simultaneous decrease in torque.

The mechanisms of the rigs, as a rule, are hydroficated and controlled remotely from the console. The rig is serviced by one or two people, owing to remote control and extensive mechanization of labor-intensive operations [1-3].

2. Materials and Methods

Various vibration mechanisms are applied as vibration exciters for the beneficial use of vibration. By the type of drive, vibration mechanisms are divided into mechanical, electromechanical, hydraulic, pneumatic, vacuum-compression, electromagnetic, and magnetostrictive.

Vibration mechanisms are divided into vibration (vibrators) and vibration-shock (vibration hammer) by the nature of the impact on the element attached to them.

Electromechanical vibrators and vibrating hammers are most widespread in the conditions of geological exploration and engineering and geological surveys [4-12].

A schematic diagram of a surface hydraulic hammer is shown to study the effect of a hydraulic hammer-pulsator on the drilling efficiency (Figure 3).

The structural scheme of the ground hydraulic hammer-pulsator is designed for drilling seismic wells, the striker of which is a two-stage piston. The piston is connected to the atmosphere; the large piston is for studying the influence of the hydraulic hammer-pulsator on the drilling efficiency is equipped with a valve that closes the channels of the large piston in the upper position, and opens - in the lower position.

3. Results and Discussion

The design diagram of the hydraulic hammer-pulsator is shown in Figure 1.

Forces acting on a piston with a valve:

a) Total gravity force of the piston with the valve.

\[ G = P_1 + P_2 = (m_1 + m_2)g = Mg \]  

\[ M = m_1 + m_2 \]  

where \( P_1 \) – piston gravity force; \( P_2 \) – valve gravity force; \( M \) – total mass of piston and valve; \( G \) – total gravity force of piston with valve; \( g \) – gravity acceleration; \( m_1 \) – piston mass; \( m_2 \) – valve mass.
Figure 1. Design scheme of a hydraulic hammer-pulsator: 1 – upper stop; 2 – shank; 3 – valve; 4 – hammer piston; 5 – anvil

b) Atmospheric pressure force

\[ P_{at} = \frac{p_{at} \pi d^2}{4} \]  
where \( P_{at} \) – atmospheric pressure force; \( p_{at} \) – atmospheric pressure; \( d \) – shank diameter.

c) Vertical (top) hydraulic pressure force

\[ P_{lh} = \left( \frac{\pi D^2}{4} - \frac{\pi d^2}{4} \right) p_v \]  
where \( p_v \) – chamber pressure (above the valve); \( d \) – piston diameter.

d) Vertical (bottom) hydraulic pressure force

\[ P_{lh} = \frac{\pi D^2 P_b}{4} \]  
where \( P_b \) – pressure under the piston.

e) Valve spring force

\[ F_{el} = cx \]  
where \( c \) – spring stiffness coefficient; \( x \) – spring deformation value.

f) Friction force of the piston against body walls \( F_{fr1} \)

g) Friction force of the shank on body walls \( F_{fr2} \)

2. Differential equations of the piston motion together with the valve (along the x-axis - down) are compiled according to the design scheme of acting forces when the piston moves down:

\[ M \frac{d^2 x}{dt^2} = \sum F_{ix} \]  
\[ M \frac{d^2 x}{dt^2} = G + p am + P_{uh} - P_{lh} - F_{fr1} - F_{fr2} - el \]
where $M = m_1 + m_2$ – total mass (piston and valve); $x = x(t)$ – current coordinate defining the piston position.

Here, the total constant forces acting when the piston moves down:

\[ F_h = P_{lh} - P_{uh} \]  \hspace{1cm} (9)
\[ F_{fr} = F_{fr1} + F_{fr2} \]  \hspace{1cm} (10)

The initial equations (8) can be written in the form:

\[ \frac{d^2x}{dt^2} = g + a + f_h + f_{fr} - \frac{cx}{M} \]  \hspace{1cm} (11)

where $f_h$ - valve wall friction; $f_{fr}$ - shank friction on the body walls

Here, all forces (the equation right side) refer to the unit mass of the moving piston with a valve.

\[ g = \frac{G}{M} \]  \hspace{1cm} (12)
\[ a = \frac{P_at}{M} \]  \hspace{1cm} (13)
\[ f_h = \frac{F_h}{M} \]  \hspace{1cm} (14)
\[ f_{fr} = \frac{F_{fr}}{M} \]  \hspace{1cm} (15)

We combine the terms of the constants in the equation (11), $g + a + f_h + f_{fr} = C = Const$

Designating $\frac{c}{M} = k^2 - const$, the differential equation (11) takes the final form:

\[ \frac{d^2x}{dt^2} + k^2 x = C \]  \hspace{1cm} (16)

The differential equation (16), which determines the piston movement is a linear inhomogeneous differential equation of the second order with constant coefficients.

The initial and boundary conditions are written out to integrate the given differential equation: at $t = 0$

\[ X_0 = 0, \ V_{x0} = 0 \]  \hspace{1cm} (17)

This means (based on the piston work with the valve) that the piston was initially stationary and was at the upper stop.

The general solution of the differential equation (16) is defined as the sum of the general solution $x_1$ of the corresponding homogeneous equation (equation without the right-hand side) and the partial solution $x_2$ of the given equation (16) with the right-hand side.

\[ x(t) = x_1 + x_2 \]  \hspace{1cm} (18)

The general solution $x_1$ of the corresponding homogeneous equation has the form:

\[ x_1 = C_1 \cos(kt) + C_2 \sin(kt) \]  \hspace{1cm} (19)

Since the right-hand side of the differential equation (16) is a constant value ($C = const$), therefore, a particular solution $x_2$ of this differential equation in the form of some constant, $x_2 = A = const$.

Substituting this value $x_2 = A$ into the original differential equation (16)

\[ k^2 A = \Pi \]  \hspace{1cm} (20)

we reveal that $A = \frac{C}{k^2}$, that is

\[ x_2 = \frac{C}{k^2} \]  \hspace{1cm} (21)

In accordance with (19) equation, the general solution of the original differential equation (5.16) has the form:

\[ x(t) = C_1 \cos(kt) + C_2 \sin(kt) + \frac{C}{k^2} \]  \hspace{1cm} (22)

To determine the values of the integration constants $C_1$ and $C_2$, an equation that determines the speed of the piston is made, by time differentiation of the motion equation (5.22):

\[ V = \frac{dx}{dt} = -C_1 \sin(kt) + C_2 k \cos(kt) \]  \hspace{1cm} (23)

Substituting initial conditions (15) into equations (22) and (23), respectively, we obtain:
\[0 = C_1 + \frac{c}{k^2}\]  
\[0 = C_2k\]  
(24)  
(25)

From (25) we get that \(C_2 = 0\) (since \(k \neq 0\)), but from (24) we get \(C_1 = \frac{\pi}{k^2}\).

Substituting the obtained values of \(C_1\) and \(C_2\) into the general solution (22), we reveal the law of the piston motion

\[x = \frac{c}{k^2}(1 - \cos(kt)) + \frac{c}{k^2} \]  
(26)

or

\[x = \frac{c}{k^2}(1 - \cos(kt))\]  
(27)

Thus, the downward movement of the piston is determined by the equation:

\[x(t) = \frac{\pi}{k^2}(1 - \cos(kt))\]  
(28)

At the same time, we find the expression for the piston speed, differentiating the motion equation in time (28):

\[V = \frac{dx}{dt} = \frac{c}{k^2}[-(-k \sin(kt))] = \frac{c}{k^2}\]  
(29)

Downward speed of the piston takes the form:

\[V = \frac{c}{k^2}\sin(kt)\]  
(30)

Assuming that the time \(T\) of reaching the anvil by the piston is determined in the equation (28) \(x = 1; (1 = L - b)\) (Fig. 1), then

\[l = \frac{\pi}{k^2}(1 - \cos(kT))\]  
(31)

Then,

\[\cos(kT) = \frac{lk^2}{c}\]  
(32)

\[T = \frac{1}{k} \arccos(1 - \frac{lk^2}{c})\]  
(33)

Then the speed value of \(V\) piston movement at the moment of contact with the anvil (at the moment of the piston impact on the anvil) will be determined by the formula:

\[V = \frac{c}{k^2}\sin(kT)\]  
(34)

where the value of \(T\) time is found by the formula (33).

Knowing the speed of \(V\) piston at the moment of impact on the anvil, determined by the formula (34), it is possible to find the shock impulse (the impulse arising from the impact of \(N\) force during the impact time \(t\)):

\[S_{im} = \int_{0}^{T} N dt\]  
(35)

We have the issue of the rigid body (piston) impact, the motion is translational and rectilinear, on a fixed surface (anvil). The impact process is meant in two phases (Fig. 2): deformation phase and recovery phase. The deformation phase with \(t_1\) duration is counted from the moment of the impact start to the moment of the body greatest deformation, when the velocity changes from the initial, equal to \(V\) (according to formula (5)), to zero.
During $t_2$ recovery phase, the body from the moment of the greatest deformation of its separation from the surface at a rate partially restores its original shape under elastic impact. Therefore, the total shock impulse $S_{im}$ is determined by:

$$S_{im} = S_1 + S_2$$  \hspace{1cm} (36)

where $S_1$ – impact impulse of the surface reaction force (impact force) during the first phase of impact,

$$S_1 = \int_0^{r_1} N \, dt \hspace{1cm} (37)$$

$S_2$ – impact impulse of the surface reaction force during the second phase of impact,

$$S_2 = \int_0^{r_2} N \, dt \hspace{1cm} (38)$$

We apply the theorem on the change in the motion amount in the projection onto the normal to the surface during the first and second phases of the piston impact on the anvil:

a) $m_1 V_k - m_1 V_{an} = S_1$ that is $S_1 = m_1 V$  \hspace{1cm} (39)

b) $m_1 u - m_1 V_{an} = S_2$ that is $S_2 = m_1 u$  \hspace{1cm} (40)

Therefore, the total shock impulse during the impact time $\tau$ is equal to

$$S_{im} = S_1 + S_2 = m_1 V + m_1 V (1 + \frac{u}{V}) \hspace{1cm} (41)$$

Then,

$$S_{im} = m_1 V (1 + K_r) \hspace{1cm} (42)$$

where $m_1$ – piston mass (striker); $V$ – piston speed at the moment of the impact start, determined by the formula (35); $K_r$ – impact recovery coefficient,

$$K_r = \frac{u}{V} \hspace{1cm} (43)$$

$u$ – piston speed after impact; $V$ – piston speed before impact.

Taking into account the equation (35) for $V$ speed at the impact start, the formula for determining the shock impulse (43) takes the form:

$$S_{im} = m_1 (1 + k_b) \frac{\Pi}{K^2} \sin(kT) \hspace{1cm} (44)$$

The dependence of the impact impulse on the piston diameter.

The calculation is made by the following formulas:

a) impact impulse $S_{im}$ is calculated by the formula (45);

b) time for the piston to reach the anvil $T$ – by the formula (44).

The calculation is carried out according to the following parameters:
a) piston diameters D vary from 0.1 m to 0.2 m with a range of 0.025 m;
b) piston thickness b = 0.04 m;
c) recovery coefficient k_B = 1;
d) spring rate c varies from 200 to 400 N/m;
e) piston stroke L = 0.1 m;
f) shank diameter d = 0.04 m;
g) shank length l = 0.1 m.
The dependence graph of the impact impulse on the piston diameter (Fig. 3) is obtained as a result of the calculation. It shows that the impact momentum increases from 33 to 101 N · s with an increase in the piston diameter from 0.1 to 0.2 m at various spring rates.

The dependence of impact impulse on the piston mass.

The calculation is carried out according to the following formulas:
a) impact impulse $S_{im}$ is calculated by the formula (44);
b) time for the piston to reach the anvil $T$ – by the formula (34).

![Dependence graph of the impact impulse on the piston diameter](image)

**Figure 3.** Dependence graph of the impact impulse on the piston diameter

The calculation is carried out according to the following parameters:
a) piston mass $m_1$ varies from 2 to 10 kg with a range of 2 kg;
b) piston thickness b = 0.04 m;
c) recovery coefficient $k_r = 1$;
d) spring rate $c = 300$ N/m;
e) piston stroke $L = 0.1$ m;
f) shank diameter;
g) shank length.

As a result of the calculation, a graph of the piston mass is obtained (Fig. 4). It demonstrates that the impact impulse increases from 13 to 93 N · s with an increase in the piston diameter from 2 to 10 kg at different pressures above the valve.
Figure 4. Dependence graph of the impact impulse on the piston mass

The dependence of the impact impulse on the piston stroke length.

The calculation is carried out according to the following formulas:

a) impact impulse $S_{im}$ is calculated by the formula (45);
b) time for the piston to reach the anvil $T$ – by the formula (34).

The calculation is carried out according to the following parameters:

a) piston stroke length $L$ varies from 0.05 to 0.25 m;
b) piston thickness $b = 0.04$ m;
c) recovery coefficient $K_r = 1$;
d) spring rate $c = 300$ N / m;
e) piston diameter $D = 0.15$ m;
f) shank diameter $d = 0.04$ m;
g) shank length $l = 0.1$ m.

As a result of the calculation, a dependence graph of the impact impulse on the piston stroke length is obtained (Fig. 5), which shows that the impact impulse increases from 40 to 147 with an increase in the length of the piston stroke from 0.05 m to 0.25 m with different piston mass.

The dependence of the impact impulse on pressure above the valve.

The calculation is carried out according to the following formulas:

a) impact impulse $S_{im}$ is calculated by the formula (45);
b) time for the piston to reach the anvil $T$ – by the formula (34).

The calculation is carried out according to the following parameters:

a) piston stroke length $L = 0.1$ m;
b) piston thickness $u = 0.04$ m;
c) recovery coefficient $K_r = 1$;
d) spring rate $c = 300$ N / m;
e) piston diameter $D = 0.15$ m;
f) shank diameter $d = 0.04$ m;
g) shank length $l = 0.1$ m;
h) pressure above the valve varies from 1.0 to 3.0 MPa.

As a result of the calculation, a dependence graph of the impact impulse on pressure above the valve is obtained (Fig. 6), which shows that the impact impulse increases from 14 to 73 with an increase in pressure from 0.5 to 3 MPa with different piston diameters.
Fig. 5. Dependence graph of the impact impulse on piston stroke length

Fig. 6. Dependence graph of the impact impulse on pressure above the valve

4. Conclusion
The theoretical analysis of operating parameters of the hydraulic hammer made it possible to create a simple diagram of the mechanism for the fluid movement distribution in the channels during the operating process. Due to the use of two moving systems, it was possible to avoid the application of movable valve elements in the design, which are prone to rapid wear in the environment of the drilling fluid used as a working agent when drilling a well.
The diagram of a ground hydraulic hammer-pulsator is given; the striker is a two-stage piston. In this case, the piston is connected to the atmosphere, the large piston is equipped with a valve, which closes the channels of the large piston in the upper position, and it opens in the lower position.

The dynamics of the operation of a ground hydraulic hammer-pulsator is worked out, the transfer of impact loads from the piston to the anvil is described, analytical methods for calculating the impact loads of a ground hydraulic hammer-pump are presented.

The impact impulse increases with an increase in the piston diameter: with a piston diameter of 0.125 m, the impact impulse is 43 N ∙ s, with a piston diameter of 0.175 m; the impact impulse is 62 N ∙ s with a spring rate of 300 N/m.

With an increase in the piston stroke length, the impact impulse increases: with a piston stroke length of 0.1 m, the impact impulse is 74 N ∙ s, with a piston stroke length of 0.2 m; the impact impulse is 105 with a piston mass of 8 kg.

With increasing pressure above the valve, the impact impulse increases: at a pressure of 1.5 MPa, the impact impulse is 34 N ∙ s, at a pressure of 2.5 MPa; the impact impulse is 48 with a piston diameter of 0.15 m.

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