Layout for drilling seismic wells by vibration method

L M Zaripova, M S Gabdrakhimov

Ufa State Petroleum Technological University, Branch of the University in the City of Oktyabrsky, 54a, Devonskaya St., Oktyabrsky, Republic of Bashkortostan, 452607, Russian Federation

E-mail: Lilyabert31@mail.ru

Abstract. More and more common methods are used to reduce the cost of drilling wells and extracting oil and gas. One of these methods is the increase of energy capacity of drilling rigs through the use of dynamic methods of drill bits loading. Various vibro-mechanisms are used as vibration exciters with helpful use of vibration. Vibro-mechanisms by type of drive are divided into mechanical, electromechanical and magnetostrictive. All of them are intended for the excitation of forced oscillations in the elements attached to them that provide the desired technological effect. The authors have carried out the work to solve the problem of studying the transfer of dynamic load on the drill string. In this case the effectiveness of bottomhole vibration is largely determined by layout parameters of the drilling tool. Oscillations of the drill string are described by a wave equation in partial derivatives. With the solution of the problem, conditions for the transfer of bottomhole vibration load are being determined. The solution shows that dynamic bottomhole load can be effectively transferred from the mouth up to 100 m; the frequency may be greater at lower striker weights.

1. Introduction

The rotary percussion drilling method began to develop and investigated by the following scientists G.I. Neudachnyi, L.E. Graf, F.F. Voskresenskii, D.D. Barkan, V.M. Slavskii, O.I. Tagiev, B.Z. Sultanov, and others who created various hydraulic hammers of both simple and double action, mechanical vibrators, vibratory hammers, which increase the mechanical speed of the drilling process of both soft and hard rocks [1-6].

Formulation and solution of the problem of studying the dynamic load transfer along the drill string. In this case, the effectiveness of bottomhole vibration effect is largely determined by the drilling tool layout parameters [7-10].

Oscillations of the drill string are described by a wave equation in partial derivatives:

$$\frac{d^2U}{dt^2} + b \frac{dU}{dt} + a^2 \frac{d^2U}{dx^2} = 0$$

(1)

where $U=U(x,t)$ – cross section offset of $x$ at the moment of $t$ time from static equilibrium position; $b$ – parameter characterizing the resistance forces acting on drill string; $a$ – distribution speed of oscillations in the drill string.

A vibrator of mass $M$ acts on upper end of the drill string with frequency $\omega$ and power $P$ (amplitude value):

$$EF \frac{dU}{dt} = M \frac{d^2U}{dt^2} - P \cos \omega t$$

(2)

where $E$ – Young’s modulus of pipe material; $F$ – cross-sectional area of pipe material.
Lower end of the drill string is of length $L$, rests on the bottomhole with stiffness and viscous resistance of $\mu$.

$$EF \frac{du}{dx} = cU - \mu \frac{du}{dt} \quad \text{(3)}$$

Equation (3) for integration is more convenient to present in the following form:

$$EF \frac{du}{dx} = M \frac{d^2u}{dt^2} - Pe^{i\omega t} \quad \text{(4)}$$

**Figure 1** Design layout of the drilling tool equipped with a ground hydraulic hammer: 1 - bottomhole of a well; 2 - bit; 3 - drill pipe string; 4 - hydraulic hammer

We will seek a forced solution in the following form:

$$U(x, t) = X(x) e^{i\omega t} \quad \text{(5)}$$

After substituting equation (5) into wave equation (1), we obtain:

$$-\omega^2 X(x) e^{i\omega t} + i\beta \omega X(x) e^{i\omega t} - a^2 \frac{d^2X}{dx^2} e^{i\omega t} = 0 \quad \text{(6)}$$

$$\frac{d^2X}{dx^2} + \frac{\omega^2 - i\beta \omega}{a^2} = 0 \quad \text{(7)}$$

General solution of equation (7) is as follows:

$$X = Ae^{kx} + Be^{-kx} \quad \text{(8)}$$

where

$$k = \sqrt{-\beta \omega - \omega^2} \quad \text{(9)}$$

From boundary conditions (2), (3) after substitution (5) is followed:

$$EF \frac{dx}{dx} = -M\omega^2 X - P \quad \text{(10)}$$

$$EF \frac{dx}{dx} = -(c + i\omega \mu) X \quad \text{(11)}$$

Solving the equations of (10) and (11) after a series of transformations and substitutions, we determine the complex value of dynamic effort at the bottomhole:

According to the well-known decision the complex value of dynamic effort at the bottomhole is determined:

$$P_{zk} = EF \frac{dx}{dx} = EF(A e^{kL} - B e^{-kL}) e^{i\omega t}. \quad \text{(12)}$$

where $A$ and $B$ – constant integrations, $k$ – root of the characteristic equation.

The dynamic force amplitude is determined through the modulus of the force complex value:

$$Q(\omega) = |P_{zk}| = |EF(A e^{kL} - B e^{-kL})| \quad \text{(13)}$$

Decisively, the dynamic bottomhole load amplitude is determined by the formula:

$$Q(\omega) = |EF(A(\omega) e^{k(\omega)L} - B(\omega) e^{-k(\omega)L})| \quad \text{(14)}$$

Equation (14) was solved with the following data: $B = 10; 30; 90; 270; M=250; 500; 1000; 2000 \text{ kg}$; $L=25; 100; 250 \text{ m}$; $E=2.1 \times 10^{11} \text{ N/m}^2$; $D=0.146 \text{ m}$; $d=0.126 \text{ m}$; $P=3.1415926$; $M=250; 500; 1000 \text{ kg}$;
F=8.9723884656·10^8; \omega=1…1000 \text{ s}^{-1}; i=\sqrt{(-1)}; b=10; 30; 90; 270; k(\omega)=-\frac{(\omega)^2}{a^2} + i \cdot \omega \cdot b \cdot \frac{1}{a^2}; P= 10000 \text{ H}; a=5.1887452166\cdot10^3 \text{ m/s}; s=100000000 \text{ H/m}; \mu=100000\text{ Pa}\cdot\text{s}.

Results of the decisions are shown in Figures 2, 3 [5].

![Graph 1](image1.png)

**Figure 2.** Dependence of maximum dynamic bottomhole load on the depth at M = 1000kg, b=30.

![Graph 2](image2.png)

**Figure 3.** Effect of striker mass on the purity of maximum dynamic bottomhole load L=100 m, b=30.

Частота, Гц – frequency, Hz

The solution shows that dynamic bottomhole load can be effectively transferred from the mouth up to 100 m; the frequency may be greater at lower striker weights; parameter b obstructs the transfer of dynamic bottomhole load.

Theoretical studies of technical parameters influence of technological process, on the transfer of dynamic load from a hydraulic hammer installed on the drilling tool, on the bottomhole of a well have been conducted.

The research results indicate that with increasing the depth of the well, frequency of oscillations transmitted to the bit increases; maximum load decreases with increasing the drill string length, and with increasing the mass of the striker, oscillation frequency decreases.

2. Results and Discussion

A schematic diagram of the ground hydraulic hammer is shown in Figure 4 to study the effect of a hydraulic hammer on drilling efficiency.
A constructive scheme of a ground hydraulic hammer-pulsator is developed. 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 channels of the large piston in the upper position, and opens in the lower one.

The hydraulic hammer consists of a container 1 fitted with an upper adapter sub 2 and a lower 3, lateral drains outlets 4 and 5, slot 6, partitions 7 and 8, anvil 9 and a guard 10. Piston-hammer 11 is installed inside container 1 and has a shank adapter 12 and port-holes 13. The piston and shank adapter are equipped with congestions 14 and 15. A disk 16 is hafted on the shank adapter and is connected by a pin 17, which interacts with a partition 8 by means of a spring 18. The pin 17 is sealed in the partition by means of congestion 20.

The device during operation is installed under the swivel. At initial moment, the spring 18 is expanded; the disk is in upper position. As the flushing fluid flows, the piston-hammer 11 and elements connected to it move upward due to the pressure difference under the piston and in the annulus. Movement occurs until the closure by disk 16 of port-holes 13 of the piston-hammer 11. As a result, a hydraulic shock happens, pressure above the piston sharply increases, the disk moves down and strikes the anvil. Before strike, the pin opens the disk, which, due to expansion of the spring, occupies the initial position. The piston moves upwards due to hydraulic forces. The cycle is repeated. The increase of indicators in rotary drilling during the drilling of seismic wells by this device is carried out at the expense of creating a dynamic load on the bit and pulsating flushing of bottomhole wells. With the use of a ground hydraulic hammer in the layout of the drilling tool, the well No. 61 was drilled with the creation of vibration.

To study the dynamics of a hydraulic hammer, a differential equation of the piston movement together with the valve is compiled (down):

$$M \frac{d^2x}{dt^2} = \sum F_{ix}$$

$$M \frac{d^2x}{dt^2} = G + P_{at} + P_{Hyb} - F_{fr1} - F_{fr2} - F_{el} - cx$$

where $M=m_1 + m_2$ – total mass of both the piston and valve; $x=x(t)$ – current coordinate defining piston position; $G$ – total piston gravity with valve; $P_{at}$ – atmospheric pressure force; $P_{HyT}$ (hydraulic top) – vertical (top) hydraulic pressure; $P_{Hyb}$ (hydraulic bottom) – vertical (bottom) hydraulic pressure; $F_{fr1}$ –
piston friction force against container walls; $F_{fr2}$ – shank adapter friction force against container walls; $F_{el}$ – elastic force of the valve spring; $c$ – spring stiffness.

Here the total constant forces acting when the piston moves down are equal to

$$F_{Hy} = P_{HyT} - P_{HyB} \quad (17)$$

Then initial equations (16) can be written in the following form:

$$\frac{d^2x}{dt^2} = g + a + f_r - \frac{cx}{M} \quad (19)$$

All forces (right side of the equation) refer to the mass unit of a moving piston with a valve, that is

$$g = \frac{G}{M}; \quad a = \frac{Pa}{M}; \quad f_{Hy} = \frac{F_{Hy}}{M}, f_{fr} = \frac{F_{fr}}{M}; \quad (20)$$

We combine the constant members of sum in equation (19) $g + a + f_r + f_{fr} = P - \text{Const.}$

Considering, that $\frac{c}{M} = k^2 \text{(const)}$, differential equation (19) takes the final form

$$\frac{d^2x}{dt^2} + k^2 x = P \quad (21)$$

The differential equation (20), which determines the downward movement of the piston, is a linear non-uniform second-order differential equation with constant coefficients. To integrate this differential equation, initial and boundary conditions are written: i.e at $t=0$

$$X_0 = 0, \quad V_{x_0} = 0 \quad (22)$$

This means (based on nature of the work of the piston with valve) that the piston was initially static and was located at the top guard.

The solution of equations (20), (21) determines the speed of the piston movement, which has the form:

$$V = \frac{P}{k^2} \sin(kT), \quad (23)$$

where time value $T$ is found by the formula

$$T = \frac{1}{k} \arccos\left(1 - \frac{ik^2}{P}\right), \quad (24)$$

where $l$ – piston stroke

Knowing the speed of $V$ piston at the moment of strike with the anvil, we determine the total impact pulse by the formula

$$S_{sh} = m(l + k_{col}) \frac{P}{k^2} \sin (kT) \quad (25)$$

where $k_{col}$ – collision coefficient at strike, $k_{col} = \frac{u}{u_0}$; $u$ – piston speed after strike.

Calculation of the dependence of the impact pulse on the piston diameter is carried out according to the following parameters: piston diameters $D$ range from 0.1m to 0.2m with a range of 0.025m; piston thickness $b=0.04$m; shank adapter length $l=0.1$m.

As a result of the calculation, the dependence of impact pulse on piston diameter (Figure 5) is obtained, which shows that the impact pulse increases from 33 H·s to 101 H·s with an increase in the piston diameter from 0.1 m to 0.2 m with different spring stiffness.

![Figure 5](image-url)
Calculation of the dependence of the impact pulse on the piston stroke length is carried out according to the following parameters: piston stroke length L varies from 0.05 to 0.25 m; piston thickness b=0.04 m; collision coefficient $k_{col} = 1$; spring stiffness $s = 300$ H/m; shank adapter diameter $d = 0.04$ m; shank adapter length $l = 0.1$ m.

As a result of the calculation, the dependence of impact pulse on piston stroke length (Figure 6) is obtained, which shows that the impact pulse increases from 40 to 147 Н·s with an increase in the piston stroke length from 0.05 to 0.25 m with different piston mass.

![Figure 6. Dependence of impact pulse on piston stroke length at different values of piston mass](image)

The results of laboratory studies of the hydraulic hammer model, the main elements of which were a cylinder, a piston and a pin, selected on the basis of geometric modelling. The test model was carried out on the stand, which had a circulating system, instruments for monitoring the parameters of fluid under and above the piston, a device for measuring the axial force of the reverse stroke of the rod.

When testing a model of a hydraulic hammer, the following questions were investigated: working efficiency of the hydraulic hammer model; selection of optimal diametric dimensions of the piston bores; ratio of the diameter of the piston and rod; effect of piston and rod friction force on model working efficiency; study of the dependence of the rod stroke return force on the piston cross-section area. In the study, the number of experiments required for static processing in order to obtain results was carried out. For this, the values of arithmetic mean and root-mean-square deviations were determined. Main geometrical dimensions of the hydraulic hammer were specified according to the results of tests of the hydraulic hammer model: adopted piston diameter - 145 mm; rod diameter - 30 mm; rod length - 40 mm; the ratio of areas of the piston and holes - 7.

Theoretical studies of technical parameters influence of technological process, on the transfer of dynamic load from a hydraulic hammer installed on the drilling tool, on the bottomhole of a well have been conducted.

The research results indicate that with increasing the depth of the well, frequency of oscillations transmitted to the bit increases; maximum load decreases with increasing the drill string length, and with increasing the mass of the striker, oscillation frequency decreases.

By solving the problem, we determine how the bottomhole load is transmitted. Experimental drilling conditions are given below. The test was carried out while drilling a well with a depth of 40m No. 61, profile No.1240 of Sorochnyskaia Square. The drilling was carried out by the drilling rig УББ -2A-2, drilling pump НБ-50, flushing fluid and technical water, pressure on the riser 2 .. 5 MPa.

Tool layout: drill pipes with a diameter of 60.3 mm, a two-blade bit with a diameter of 190 mm, and a ground hydraulic hammer was installed between the swivel and the drill string.

The installation was equipped with a combined rotor-swivel, rotor speed of 140 rpm (r/min). The load on the bit was created by a hydraulic vibration device 1.2 .. 2.6 t.f. Geological section of the well was composed of sediments of quaternary age, represented by loams and sands.
3. Conclusion
As a result of drilling a well by a vibration layout, 40 m of rocks for 1 spudding drilling is drilled. Drilling time is 20 minutes; mechanical speed is 2 m/min. We have compared mechanical speeds of neighboring wells drilled in the same area to assess the results of experienced drilling with the use of a ground hydraulic hammer. The average mechanical speed of the estimated wells is 1.64 m/min.

An increase in the mechanical drilling rate by 18% has been obtained as a result of experienced drilling, established when drilling the well No. 61 using a ground hydraulic hammer.

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