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Studies of Equipment Stability for Drill String in Oil Fields Development

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***Abstract* — The current development state of oil and gas fields is characterized by the share increase of hard-to-recover reserves, which leads to new technological solutions for bottom-hole layouts during construction of directional wells. The paper presents analytical studies of the oscillator-turbulizer, and static equations for the initial and closed position of the valve. The results can be further used to develop various designs of hydraulic vibrators for drilling.**

Keywords — directional well; bottom hole assembly; oscillator-turbulator; oscillations; valve.

I. INTRODUCTION

The development of a significant part of oil and gas fields is fraught with considerable difficulties due to the increase in the share of hard-to-recover reserves [8,9]. In this regard, there is a constant search for new technological solutions for bottom-hole assembly in directional wells construction [4,5,7].

According to previously conducted experimental studies, longitudinal oscillations of the bit are effective means to increase the drilling rate in hard rocks. This was the basis to create methods to control the dynamics of a rock-cutting tool, vibration or impact-rotational drilling, begun, in particular, by G.I. Neudachniy, L.E. Graf, F.F. Voskresenskiy, D.D. Barkan, V.M. Slavsky, E.I. Tagiev and other researchers. Hydraulic hammers of both simple and double action were created, vibratory hammers and mechanical unbalance vibrators, which increase the productivity of drilling operations by 2 ... 3 times. However, characteristic of such vibration generators was a complicated construction with a lot of high-wear parts (springs, valve) and a large impact mass, which led to a quick failure of the bit.

In recent decades, various theoretical and experimental studies in this area have been carried out by specialists at Russian University of Oil and Gas named after Gubkin, Ufa State Petroleum Technical University, etc. [2,3,6,10,11].

Various different designs of vibration amplifiers were developed and applied, which, unlike the previously proposed structures, have controlled dynamic effects that do not lead to a sharp increase in the intensity of destruction of the bit supports: with a three-blade hydraulic actuator, a flip valve, adjustable dynamics, hydraulic amplifier, etc.

Foreign researchers are also interested in devices capable of influencing the dynamic load change on the bit. The device contained [12] a hydraulic load amplifier, in the form of a piston, periodically moving downward, and providing, at the same time, the regulation of the amplitude of the dynamic load due to damping, in particular in the jetting nozzles of the bit. The static load was provided, mainly, due to the pressure drop in the nozzles. At the same time, control of the position of the front face of the cutting part of the bit was ensured, periodic braking (up to a stop) of the downhole motor was warned. For example, in [12], it was noted that the application in the bottomhole assembly during oblique drilling of wells with medium and large zenith angles proposed by the developers of the bit load control device allowed increasing the mechanical drilling rate by 35% in the slip mode and 15% when using roller cone chisels The number of tool breakdowns in the well, especially those related to the parameter measurement systems during drilling (MWD), logging (LVD), and downhole motors, has decreased. In general, the number of violations decreased by 40-50%.

Thus, theoretical research and development of technological solutions to combine static load on the rock-breaking tool with a dynamic impulse, while maintaining continuous contact of the bit with the bottom and improving the quality of well drilling, is relevant and significant for oilfield services companies.

II. METHODS AND MATERIALS

At the department Drilling of Oil and Gas Wells Almetyevsk State Oil Institute, a well oscillator-turbulator has been developed and patented [1], (Fig. 1), which works as follows. Wash fluid is pumped from the surface by pumping units and passes through a string of pipes to the downhole oscillator. Through the passage channel, the jet of fluid enters the upper diffuser 4. The upper diffuser 4 performs the function of transferring a fluid from round section to square bushing 5. On the hub, the liquid jet moves through square section and enters the valve, which under its action begins to make oscillatory movements, bending then one or the other side to the inner wall of the housing, as a result of which, at certain points in time, the passage channel is blocked. This leads to oscillation of low-frequency oscillations of the washing liquid. The fluid after the transition from the valve 3 moves along the sleeve 5, and thereby enters the lower diffuser 4, which has a circular cross-section. Valve 3 is supported on axis 6. Cover 2 serves to connect an oscillator-turbulator with a downhole motor. On the case of the oscillator there are screw notches (grooves) that contribute to a better removal of sludge in the horizontal part of the well.

The limiting (extreme) position of the valve depends on its shape and size. The valve must be resistant, i.e. ability to return to the original vertical position after the termination of the deflecting forces.

The oscillator-turbulizer valve dynamics was investigated by numerical methods in Mathcad program with the following assumptions:

1. To describe the movement, it suffices to consider one cycle, from the beginning of the opening of the valve to its closing.
2. Fluid flow through the valve is assumed to be dependent linearly on the angle of rotation of the valve (maximum flow Q_{max} in the vertical position, minimum Q_{min} – in the limit/extreme position), i.e. $Q=Q_{max}-K\cdot\varphi$. In the particular case $Q_{min}=0$.
3. Valve mass, moment of inertia calculated in SolidWorks program for specific geometrical parameters and material characteristics.

Characteristics of the working environment:
flushing fluid flow rate:

- Q_{max}=0,035…0,045 m^3/s; Q_{min}=0,001 m^3/s;
flushing fluid density:
- ρ_J=1000…1200 kg/m^3;
fluid viscosity:
- $\mu=5\cdot10^{-3}\ldots20\cdot10^{-3}$ Pa·s.

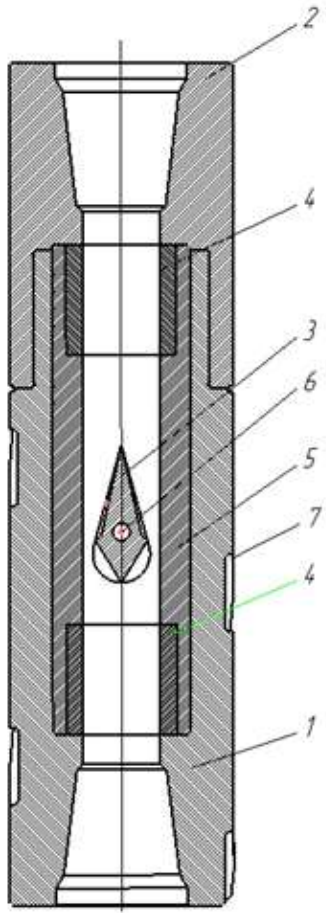

Fig. 1. Scheme of oscillator-turbulator: 1-body; 2- cover; 3 valve; 4 - upper, lower diffuser; 5 - sleeve; 6 - axis; 7 - notches.

III. STATIC EQUATION FOR INITIAL VALVE POSITION

Let us consider the work of the oscillator-turbulizer in detail.

The deflecting forces will be the pressure force of the flushing fluid (hydrodynamic force F_{hydrod}). The stabilizing forces (returning the valve to the initial vertical position) will be the force of gravity G and the buoyancy force Fa (Archimedes force). The mass center and pressure center of the buoyancy do not generally coincide (for example, a float with a sinker for a fishing rod; a balloon with a hanging basket for cargo).

Since the absolute value of $G>Fa$, it is necessary that the force of gravity "work" to open the valve (that is, return the valve to its original vertical position). This is possible if the gravity vector G is located to the left of the vertical axis (Y axis) passing through the axis of rotation (point O).

The initial position of the valve (at the initial moment of starting the valve into operation) is shown in Fig.2, where G is the valve weight, F_a is the buoyant force (Archimedes force), F_{hydrod} is the hydrodynamic force (pressure force of the flushing fluid), N is the reaction force of the body walls.

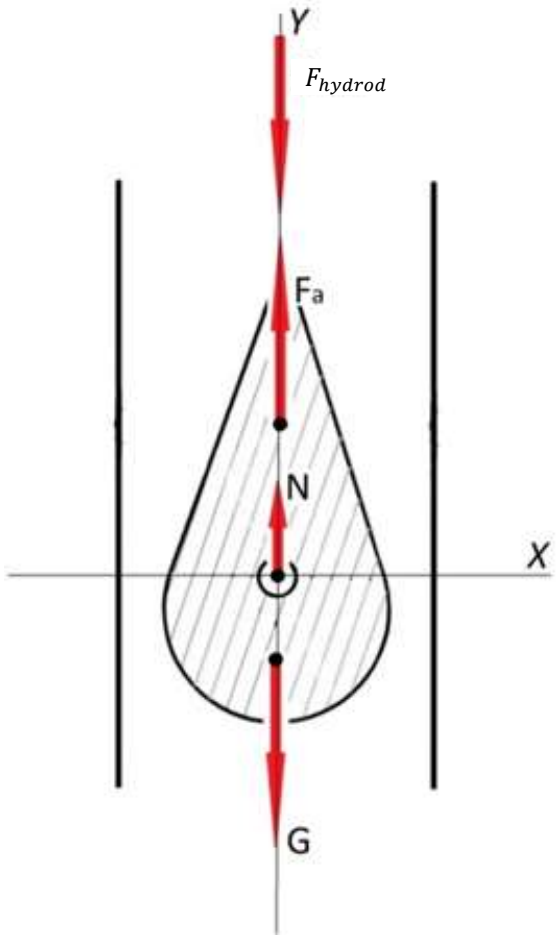

Fig. 2. Initial moment of the valve start-up.

We consider the direction of the Y axis upwards to be positive.

$-G + N + F_a - F_{hydrod} = 0.$

It follows that the reaction force in the support is equal to

$$N = G - F_a + F_{hydrod}. \quad (1)$$

Sliding friction force is $F_f = fN$, (2)

Where f – slip friction coefficient.

$f = 0{,}1 \ldots 0{,}2$ (for steel-steel friction pair);

$f = 0{,}1 \ldots 0{,}15$ (for steel-bronze friction pair);

$f = 0{,}05 \ldots 0{,}15$ (for для пары трения steel-cast iron friction pair);

$f = 0{,}07 \ldots 0{,}15$ (for cast iron-cast iron friction pair).

The force of sliding friction in the support must be taken into account when the valve moves (in dynamics).

IV. STATIC EQUATION FOR CLOSED VALVE POSITION

We considered the general case when the center of mass and the center of pressure of the buoyancy do not coincide.

The closed position of the valve is shown in Figure 3, where G is the valve weight, F_a is the buoyant force (Archimedes force), F_{hydrod} is the hydrodynamic force (pressure force of the flushing fluid), N_x, N_y, K_x are the support forces of the bearing, φ_{max} is the maximum angle of rotation valve, R_f is the shoulder of the hydrodynamic force, H is the width of the oscillator-turbulator body, O-axis of rotation, L is the distance of the action of the support force to the axis of rotation, b is the distance of the Archimedes force to the axis of rotation, a is the distance of the force of gravity rotation axis.

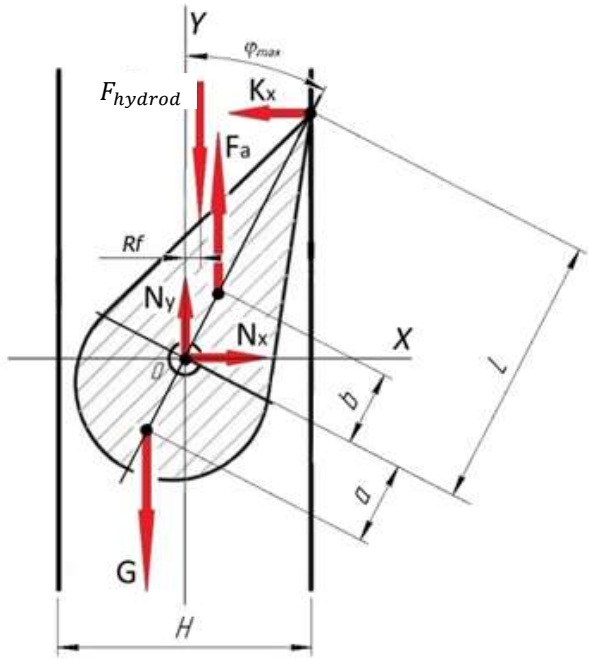

Fig. 3. Closed valve position.

Projection of forces on the X axis:

$$Nx - Kx = 0 \quad (3)$$

Projection of forces on the Y axis:

$$-G + F_a - F_{hydrod} + Ny = 0 \quad (4)$$

Moments of forces relative to point O (axis of rotation):

$$G \cdot a \cdot \sin\varphi_{max} + F_a \cdot b \cdot \sin\varphi_{max} + Kx \cdot L \cdot \cos\varphi_{max} - F_{hydrod} \cdot Rf = 0 \quad (5)$$

Here $\varphi_{max} = \arcsin(\frac{H}{2L})$.

From equations (3) - (5) you can find unknown reactions Nx, Ny, and Kx.

The reaction force of the support is always perpendicular to the contact surface of the bodies, therefore there will be no vertical projection of the reaction K ($Ky = 0$).

The force of friction arises if there is movement (or the possibility of such movement) of one body relative to another. Since the valve can only rotate around the axis (point O), there will be no frictional force at the point of contact of the valve with the body (sleeve).

If Kx>0 (the valve is pressed with some force against the wall of the body), then the valve will not be able to detach from the wall (will not be able to start moving).

Since there is always friction in the support during movement/rotation, the condition Kx<0 is necessary for guaranteed valve separation from the wall.

V. CONCLUSION

1. Corresponding valve profiling (upper and lower parts, i.e., above and below the axis of rotation) will help you to choose:

- necessary law of change in fluid flow through the valve;
- weight and size characteristics affecting gravity and the force of Archimedes;
- magnitude and point of application of hydrodynamic force;
- friction torque in the support (both through the change in the diameter of the axis, and the ratio of the forces acting on the valve).

2. In addition to the valve design, the design of the housing (sleeve) of the oscillator-turbulator is important. The change in the flow area affects the flow rate around the valve. As a result, there are changes in:

- the magnitude of the hydrodynamic force;
- hydraulic resistance coefficient (due to the mutual influence of the closely located elements of the device - the valve and the liner);
- the maximum angle of deviation of the valve from the vertical;
- mass and inertial characteristics of the valve (due to the possible variation of its thickness).

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