A mathematical model of a vibrator with an axial striker mechanism

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Abstract. To be operated effectively by the production and injection wells, the nature of oil reservoir production is largely determined. The efficient operation of wells depends on geological and technological factors. This refers to their exploitation with oil output equal to the potential capabilities of the reservoir when it is fully served by the filtration process. The bottom-hole zone (BHZ) is in a non-equilibrium thermodynamic state of energy and mass exchange with the well and the reservoir. The contamination of the bottom-hole zone has a significant impact on the effectiveness of wells.

1. Introduction
The deterioration in properties of the bottom-hole zone is due to: the permeability of the well-killing and washing liquids in the process of underground repair of the wells; the penetration of mechanical impurities and corrosion products of metals during well-killing or washing; the deformation of rocks at the bottom hole during drilling; the reduction of penetrating and poriness with the increasing of effective stress; the reduction of the phase permeability for the liquid (oil) while the bottom-hole pressure reducing below the level of the saturation pressure of the oil reservoir by the gas; the reduction of the phase permeability for the oil from the water saturation of the reservoir during mining (using water-flooding operation, in the case of formation of water cones, etc.); the hydration of the particles of the argillaceous cement of the terrigenous reservoir when saturated it with fresh water; the precipitation and formation of asphaltene-resin-paraffin compounds of oil or salts from the produced water when temperature-pressure conditions are changing [1-5].

The analysis of the reasons affecting the permeability of geological rock in the bottom-hole zones of wells educed that the choking of the filtration rock channels with the solid particles of the argillaceous solution, the particles of drilled rock, sand, silt, etc. in the process of various technological operations reduce the relative permeability for oil by 5-6 times [6-10].

A downhole vibrator has been created for vibration-wave cleaning of the BHZ, and the design scheme of the vibrator is described [11-12].

2. Materials and methods
The downhole vibrator consists of a housing assembly, upper and lower adapters, a spring-loaded seat with a through-hole, a spring, a calibrated sleeve, inside which a balancer is installed, made at the
same time with a disk, which is attached through an axis. The housing assembly is provided with a shoulder.

The tool operates as follows. The downhole vibrator is lowered into the well, and after the circulation restoring, the jet of service fluid acts on the blade of the balancer, takes it out of balance, due to hydraulic forces, the balancer begins to sway from one extreme position to another. When the passage section of the seat is blocked by a disk, a water hammer occurs.

The spring stiffness is selected taking into account that during normal operation of the downhole vibrator, the seat is to be in its original position. If the seat channel is clogged with mechanical impurities, then under the influence of excessive pressure of the service fluid, the seat is pressed out and mechanical impurities are washed out.

3. Results and discussion

The mechanic process of the downhole vibrator.

To ensure the most stable forced vibrations for the vibrator, if possible, the following conditions should be fulfilled:
- the load on the seat-striker is equal to the pressure of the high-speed flow of the liquid on the counter surface of the seat plus the pressure drop in the inner hole of the seat-striker;
- the load on the seat-striker is greater than the spring's elastic force when it is pre-deformed.

The hydraulic losses in the inner hole of the seat-striker can be determined by a well-known formula:

\[ (P_1 - P_2) = V_1^2(2g)^{-1}(\xi_{thr} + \xi_{jc}D^2d^{-2}e^{-2}). \]  

where \( = \frac{d_{jc}}{d} \) is the coefficient of jet contraction; \( d_{jc} \) is the diameter of the jet contraction flow at the entrance of the chamber of the pendulum apparatus in the cavity of the seat-striker with a diameter \( d \); \( \xi_{thr} \) and \( \xi_{jc} \) are the coefficients of the local hydraulic resistance at the compression of strings and the sudden expansion of it at the exit from the cavity diameter \( d \), respectively; \( V_1 \) is the flow rate of the fluid in the cavity of the pendulum oscillations when applying the water hammer; \( d \) is the diameter of the inner seat-striker; \( D \) is the outer diameter of the seat-striker.

The operation scheme of the seat-striker is shown in figure 1.

We can rewrite the conditions for ensuring stable forced vibrations for the vibrator taking into account (1):

\[ P = \frac{v_1^2}{2g}F_k + (P_1 - P_2)F_k + \Delta P \pi d^2 \geq k(h_{fr} - h), \]  

where \( F_k \) is the area of the end surface of the seat-striker,

\[ F_k = \frac{\pi(D^2 - d^2)}{4}; \]  

\( k \) is the spring stiffness; \( h_{fr} \) is the height of the spring in a free state; \( h \) is the height of the springs after deformation during the vibrator assembling (figure 1) and under the additional influence of the dynamic pressure of the fluid flow; \( \Delta R \) is the variable load on the seat during periodic overlapping the holes \( d \) of the seat-striker of the pendulum valve and changing from zero to the value of \( \Delta P_{wb} \); \( \Delta R_{wb} \) is the amplitude of water hammer occurring at the time of overlapping the inner hole \( d \) of the seat-striker of the pendulum valve; \( D \) and \( d \) are the outer and inner diameters of the seat-striker, respectively.

When the pendulum periodically closing to momentary overlaps the flow passage section with a diameter of \( d \) in the seat-striker, a hydraulic hammer occurs, which affects the seat with a variable load of amplitude \( \Delta P \). The value of the water hammer of \( \Delta P_{wb} \) can be determined by the formula of N.E. Zhukovsky: the disturbing variable load (on the seat) at \( t < \tau = \frac{2L}{c} \) (the case of installing vibrators in a well located hundreds of meters away from the bottom-hole or at the surface):

\[ \Delta P_{wb} = \rho c V_1; \]  

when \( t < \tau = \frac{2L}{c} \)

\[ \Delta P_{wb} = \rho c(V_1 - V_0), \]  

(5)
where \( c \) is the speed of the strike wave (for practical calculations, without taking into account the compressibility of fluid and elasticity of pipe walls, equal to the speed of sound); \( L \) is the length of the section of the column from the placement of the vibrator to the drilling bit jet nozzles (the source of flow rate); \( t \) is the overlap time of flow (in one cycle of the pendulum oscillation, the flow is blocked twice), \( t = (2 \nu)^{-1} \); \( p \) is the fluid density; \( V_{fh} \) is the average rate of the fluid hole of the seat; \( V_1 \) is the flow rate of the fluid in the cavity of the pendulum oscillations when applying the water hammer.

\[
\frac{V_1 - V}{t} = \frac{V_{fh} \Delta}{t}.
\]  

(6)

where \( V \) is the rate of the fluid flow.

Since in this case, \( t > \tau \), i.e. the hydraulic strike is not immediate and, properly, incomplete, its value will be slightly less and equal to

\[
\Delta P = \frac{F_{k \Delta P_{wh} \tau}}{t}.
\]  

(7)

In this case, formula (7) for determining the value of the disturbing variable load that is acting on the seat-striker when the pendulum valve periodically overlaps the hole can be converted into an approximate relationship

\[
\Delta P = \frac{2\rho LV_1^2}{t} F_k = \frac{4\rho LV_1^2}{g} F_k.
\]  

(8)

Figure 2 shows the results obtained by the formula (8) for calculating the dependence of the amplitude of the disturbing variable load \( \Delta P \) and the frequency of forced vibrations acting on the seat-

**Figure 1.** The scheme of the seat-striker: \( P_1 \) is the hydraulic load on the seat during water hammer; \( P_2 \) is the hydraulic load during water hammer under the seat; \( V_{fh} \) is the average rate of the fluid hole of the seat; \( V_1 \) is the flow rate of the fluid in the cavity of the pendulum oscillations when applying the water hammer; \( V_2 \) is the flow rate of fluid when applying the water hammer in the cavity under the seat; \( h \) is the height of the springs after deformation; \( k \) is the stiffness of the spring; \( d \) is the diameter of the inner seat-striker; \( D \) is the outer diameter of the seat-striker.

With a smooth change in the flow rate in the column (the high non-reactivity with a large length of it) during the time \( \tau \) of the loss rate,

\[
(V_1 - V) = \frac{V_{fh} \Delta}{t}.
\]

(6)

where \( V \) is the rate of the fluid flow.

Since in this case, \( t > \tau \), i.e. the hydraulic strike is not immediate and, properly, incomplete, its value will be slightly less and equal to

\[
\Delta P = \frac{F_{k \Delta P_{wh} \tau}}{t}.
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\]  

(8)
striker when the pendulum of the downhole vibrator moves from the flow of the washing liquid $Q$. The assumed placement of the vibrator from the bottom of the well is $L = 2.5$ m.

The end surface area of the valve seat is $F_k = \pi(0.1^2 - 0.03^2) = 0.0071 \text{ m}^2$.

The rate of the liquid flow in the cavity of the pendulum oscillations when applying the water hammer is equal to the rate of the liquid flow ($V_1 = V$). The density of the liquid $p = 10^3 \text{ kg/m}^3$.

**Figure 2.** The influence of the flow rate $Q$ on the frequency of the forced vibrations $v_{12}$ and the amplitude of the disturbing load $\Delta P$ of the pendulum vibrator, m: $l_1 = 0.052$; $l_2 = 0.064$; $l_3 = 0.032$.

4. **Conclusion**

Thus, the obtained values of the variable load $\Delta P$ are comparable with the results of bench studies of downhole vibrators.

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