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Simulation of the working process of a dual-circuit downhole ejection system

Symulacja procesu pracy dwuobwodowego wgłębnego systemu wypływowego

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ABSTRACT: The article is devoted to a study of the flow's nature distribution in the above-bit ejection system as part of a series and parallel connection of two jet pumps designed to improve the technical and economic indicators of drilling production wells. Using the finite element method and computer programs, the critical value has been determined of the hydraulic resistance of the sludged section of the bottomhole, which provides an automatic change in the ejection system operation mode from injection to injection-suction. Regularities of changes in the hydrodynamic parameters of the ejection system have been established and the probability and conditions for the existence of non-operating modes of jet pumps have been analyzed. When conducting experimental studies, the influence of the flow rate of the working medium on the flow characteristic of a jet pump of a direct-flow and swirling working flow has also been determined. To swirl the flow, a plate with an inclination angle of the guide elements of 15° placed in the working nozzle of the jet pump has been used. The swirling of the working flow makes it possible to increase the injection ratio of the jet pump. The maximum increase in the value of the injection coefficient is 20.84% and corresponds to small (up to 10^4) Reynolds numbers of the working flow. The ratio between the maximum and minimum values of the injection coefficient of the flow characteristics of the jet pump when swirling the working flow is halved.

Key words: downhole ejection system, finite element method, flow swirl, injection ratio, relative head, above-bit jet pump.

STRESZCZENIE: Artykuł poświęcony jest badaniu charakteru przepływu w układzie z wypływem nad świdrem w ramach szeregowego i równoległego połączenia dwóch pomp strumieniowych, zaprojektowanych, aby poprawić wskaźniki techniczno-ekonomiczne wiercenia otworów produkcyjnych. Stosując metodę elementów skończonych oraz programy komputerowe, wyznaczono wartość krytyczną oporu hydraulicznego odcinka odwiertu, w którym przepływa płuczka ze zwiercinami, co zapewnia automatyczną zmianę trybu pracy układu wypływowego z tłoczenia na tłoczenie–ssanie. Ustalono prawidłowość zmian parametrów hydrodynamicznych układu wypływowego oraz przeanalizowano prawdopodobieństwo i warunki istnienia trybów, w których pompy strumieniowe nie pracują. Prowadząc badania eksperymentalne, określono wpływ natężenia przepływu czynnika roboczego na charakterystykę przepływową pompy strumieniowej o przepływie bezpośrednim i wirowym. Do zawirowania przepływu zastosowano płytkę o kącie nachylenia elementów prowadzących 15° umieszczoną w dyszy roboczej pompy strumieniowej. Zawirowanie przepływu roboczego umożliwia zwiększenie współczynnika tłoczenia pompy strumieniowej. Maksymalny wzrost wartości współczynnika tłoczenia wynosi 20,84% i odpowiada przepływowi roboczemu w zakresie małych (do 10^4) liczb Reynoldsa. Stosunek pomiędzy maksymalnymi i minimalnymi wartościami współczynnika tłoczenia w charakterystyce przepływowej pompy strumieniowej przy zawirowaniu przepływu roboczego zmniejsza się o połowę.

Słowa kluczowe: wgłębny system wypływowy, metoda elementów skończonych, zawirowanie przepływu, współczynnik tłoczenia, względna wysokość podnoszenia, pompa strumieniowa nad świdrem.

Introduction

The decline in world hydrocarbon reserves necessitates the improvement of technologies for the development of oil and gas fields. One of the non-traditional methods for increasing the efficiency of oil and gas production in difficult mining and geological conditions is the use of downhole ejection systems, which are currently used in the drilling (Kryzhanivskyi and Panevnyk, 2020), completion (Shaidakov et al., 2018) and operation (Verisokin et al., 2021) of wells. The effectiveness of the introduction of oil ejection technologies is determined by the accuracy of predicting the operating mode of a downhole

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jet pump. Modern analytical models of the working process of a jet pump usually use systems of differential equations for the motion of a viscous fluid, which are solved using numerical simulation and special computer programs (Qian et al., 2021). In the process of analyzing the interaction of mixed flows, the propagation of submerged flows are what are usually studied (Ivaniv et al., 2021), and the geometric dimensions of the elements of the hydraulic system (Portillo-Vélez et al., 2019) and their mutual orientation (Cherniuk et al., 2021) are optimized. The search for ways to improve the design of ejection systems has led to the emergence of computer models of multi-jet pumps (Xu et al., 2020).

Despite a significant number of studies based on the use of modern numerical modeling methods, there are currently no works devoted to the analysis of the relationship between the elements of double-circuit bottom-hole ejection systems consisting of a parallel and serial connection of two jet pumps (Panevnik et al., 2018). Compressed conditions for the simultaneous use of two jet pumps and design features of a double-circuit ejection system do not allow improving the pressure-flow characteristic by swirling the injected flow (Panevnyk, 2021). Due to the limitation of the allowable performance of the mud pump, the values of operating flow rates when using a parallel and series connection of two jet pumps are much less than in traditional above-bit ejection assemblies. The lack of information on the dependence of the injection coefficient on the value of the working flow reduces the efficiency of using double-loop downhole ejection systems.

The aim of the present work is to study the natural hydraulic elements connections of a downhole dual-circuit ejection system based on the simulation of the working process and to experimentally establish the influence of the operating flow rate on the characteristics of the jet pump.

Downhole ejection system design

The device consists of an above-bit adapter located in the lower part of the drill string 1, with auxiliary 2, and main jet pumps 3 and slurry channels (Figure 1).

Removal of sludge and its transportation to the surface is carried out through the annular channel of the annulus 5, formed by the wall of the well 4, the above-bit adapter and the drill string 1. The lower part of the adapter is connected to the bit 6. The nozzle of the auxiliary (injection-suction) jet pump 2, designed to remove sludge from the above-bore area, is connected to the receiving chamber of the main (injection) jet pump 3, the diffuser – with the annulus, and the receiving chamber – with sludge channels and the overhead space. The nozzle of the main jet pump 3 is connected to the channel of the drill string 1, the diffuser is connected to the flushing channel of the bit 6, and the mixing chamber is connected to the nozzle of the auxiliary jet pump 2. The flushing liquid enters the nozzle of the main jet pump 3 through the channel of the drill string 1 by means of a diffuser and the flushing system of the drill bit 6. As a result of creating a low pressure zone in front of the mixing chamber, an additional amount of flushing solution is sucked through the flow path of the auxiliary jet pump 2 and sludge channels. The flow rate of the injected flow decreases with an increase in the degree of bottomhole sludge and an increase in the hydraulic resistance at the outlet of the jet pump 3.

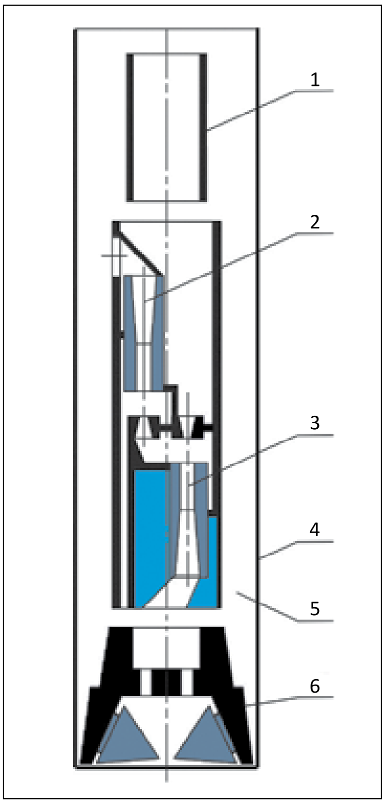

Figure 1. Schematic diagram of the downhole ejection system:
1 – drill string; 2 – auxiliary jet pump; 3 – main jet pump;
4 – borehole wall; 5 – annular space; 6 – drill bit

Rysunek 1. Schemat ideowy wgłębnego systemu wypływowego:
1 – przewód wiertniczy; 2 – pomocnicza pompa strumieniowa;
3 – główna pompa strumieniowa; 5 – ściana otworu; 5 – przestrzeń pierścieniowa; 6 – świder wiertniczy

After reaching the critical value of hydraulic resistance, the jet pump 3 stops sucking in the flushing solution, and the flow is redistributed through the channel of the drill string 1 in its receiving chamber. Part of the flow passes through the mixing chamber with the diffuser of the main jet pump 3, the flushing system of the bit space, and the part enters the nozzle of the auxiliary jet pump 2, passes its mixing chamber with the diffuser and is directed to the annulus channel. The suction of the flow by the auxiliary jet pump 2 is carried out through the sludge channel. At the same time, a difference in hydrodynamic pressures is created in the sludged section of the bottomhole and there arise conditions for the erosion of the sludge plug.

After cleaning the bottomhole and eliminating its excess sludge, there is resumed the preliminary circulation of the flushing solution, which is performed using the main jet pump 3.

According to the hydraulic scheme of the ejection system (Figure 2), when the main jet pump 4 is operating in the suction mode, at the point "a", the flow is connected with the flow rate Q_w directed by the mud pump and the flow with the flow rate Q_{e1} that is sucked in by the main jet pump 4.

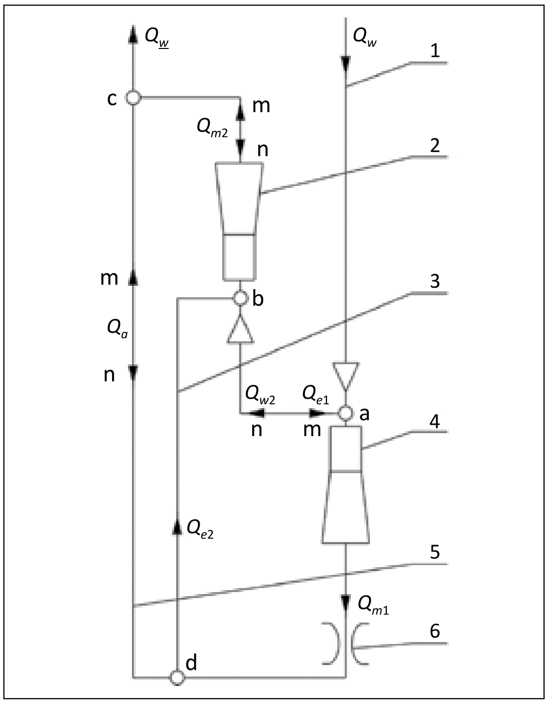

Figure 2. Hydraulic scheme of the ejection system: 1 – hydraulic channel of the drill string; 2 – auxiliary jet pump; 3 – sludge channel; 4 – main jet pump; 5 – hydraulic channel of the annulus; 6 – bit flushing system

Rysunek 2. Schemat hydrauliczny systemu wypływowego: 1 – kanał hydrauliczny przewodu wiertniczego; 2 – pomocnicza pompa strumieniowa; 3 – kanał płuczkowy; 4 – główna pompa strumieniowa; 5 – kanał hydrauliczny przestrzeni pierścieniowej; 6 – system przemywania świdra

The direction of "m" fluid movement in the diagram corresponds to the operation in the suction mode of the main pump 4, and the direction of "n" flows corresponds to the operation in the suction mode of the auxiliary jet pump 2. At the point "d", depending on the ratio of hydraulic resistances of the system elements, the mixed flow Q_{m1} and flow Q_a are joined coming from the annulus, or the mixed flow Q_{m1} is divided into flows with flow rates Q_a and Q_{e1}.

At point "b" there is a connection of flows with flow rates Q_{e2} and Q_{m2} (if the main jet pump 4 operates in the suction mode $Q_{e1} = Q_{e2} + Q_{m2}$) or flows with flow rates Q_{e2} and Q_{w2} (if the auxiliary jet pump 2 operates in the suction mode and there is fulfilled the ratio $Q_{m2} = Q_{e2} + Q_{w2}$).

At point "c" there is a mixed flow Q_{m2} split, i.e. $Q_{m2} = Q_a + Q_w$ (if the auxiliary jet pump 2 is in suction mode) or an annular channel split $Q_a = Q_w + Q_{m2}$.

Characterization of a downhole ejection system

Figure 3 shows a geometric model of a downhole ejection system built using Autodesk Inventor. The bit flushing system model is presented in the form of three flushing nozzles, and the hydraulic resistance to sludged bottomhole sections is taken into account using an equivalent diaphragm of a variable diameter.

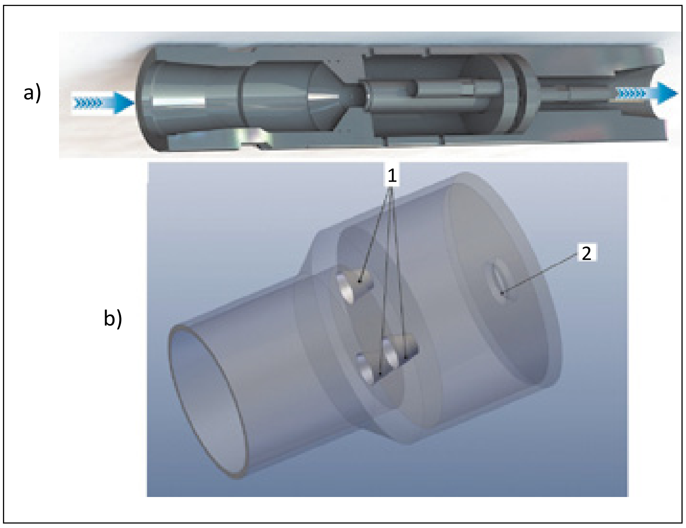

Figure 3. Geometrical model of the downhole ejection system (a) and a bit flushing system (b): 1 – bit flushing nozzles; 2 – equivalent pump orifice

Rysunek 3. Model geometryczny wgłębnego systemu wypływowego (a) i systemu przemywania świdra (b): 1 – dysze do przemywania świdra; 2 – równoważna kryza pompy

The characteristic of the jet pump hydraulic system is determined by the equations as follows:

$$h_1 = f(i_1) = \frac{P_{m1} - P_{e1}}{P_{w1} - P_{e1}} = \frac{\Delta P_b + \Delta P + \Delta P_{n2}}{\Delta P_{n1} + \Delta P_b + \Delta P + \Delta P_{n2}} \quad (1)$$

$$h_2 = f(i_2) = \frac{P_{m2} - P_{e2}}{P_{w2} - P_{e2}} = \frac{\Delta P_a}{\Delta P_{n2} + \Delta P_a} \quad (2)$$

where:

h_1, h_2 – the relative pressure generated by the main and auxiliary jet pumps, respectively,

i_1, i_2 – injection coefficients of the main and auxiliary jet pumps,

P_{m1}, P_{m2} – values of mixed flow pressures for the main and auxiliary jet pumps,

P_{e1}, P_{e2} – values of injected flow pressures for the main and auxiliary jet pumps,

P_{w1}, P_{w2} – values of workflow pressures for the main and auxiliary jet pumps,

ΔP_b, ΔP, ΔP_{n1}, ΔP_{n2}, ΔP_a – hydraulic losses in the bit washing system, equivalent diaphragm, nozzles of the main and auxiliary jet pumps and in the annular channel in the «cd» section (Figure 2).

The pressure of a mixed flow is determined at the outlet of the diffuser, the pressure of a working flow is determined in front of a working nozzle, and the pressure of an injected flow is determined in front of the receiving chamber of the jet pump. The jet pump injection coefficient characterizes the ratio of the flow rate of the injected and working flows.

In the process of simulating the workflow of above-bit jet pumps, the geometric model of the downhole ejection system and the flushing system of the bit was imported into the ANSYS software using the DesignModeler application module built into the Workbench platform. To enter the initial data, the values of the operating flow rate at the inlet of the model and the value of the pressure at the outlet of the assembly were used. Ansys Fluent Mosaic Meshing topology was used to speed up the process of building the mesh model. Figure 4 shows a structured grid model for the main jet pump, containing about four million finite elements of various shapes.

To analyze the model of the downhole ejection system, a set of three-dimensional components was created, combined into a single geometric structure. In the process of building the grid model finite elements of cubic and prismatic shape were used. Due to the significant number of elements of the working grid (1 816 804 finite elements) the "Virtual Topology" method was used in order to achieve an optimal balance between the accuracy of the results and the duration of the calculation operations. Model conditions are determined by the following parameters:

- solver velocity formulation (Absolute Velocity);
- solver time model (Steady-state);
- solver type (Pressure-Based Solver);
- viscous model for hydrodynamics calculations: "k-ε Realizable Turbulence Model" with "Scalable wall function";
- mass flow rate at the entrance of the model (28 253 kg/s);
- the pressure of the mixed flow at the outlet of the model in the annulus (43.24 MPa);
- the roughness of the walls of the model (200 microns).

The characteristics of the liquid used in the study correspond to parameters of the drilling mud, with a density of 1200 kg/m^3 and a dynamic viscosity of 0.0203 kg/(m · s).

To improve the accuracy of the mesh model of the downhole ejection system while maintaining an acceptable calculation time, the finite element mesh in volumes limited by complex geometry is made denser (Figure 4).

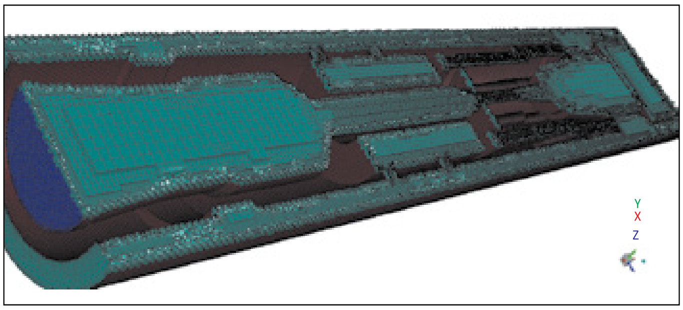

Figure 4. The mesh of finite elements of the model

Rysunek 4. Siatka elementów skończonych modelu

Figures 5, 6 show the distribution of velocities and pressures in the flow path of jet pumps for the case of an equivalent diaphragm with a diameter $d_d = 0.005$ m.

The diagrams obtained for different diameters of the equivalent diaphragm enabled us to determine the distribution of hydrodynamic parameters in the characteristic sections of the ejection system.

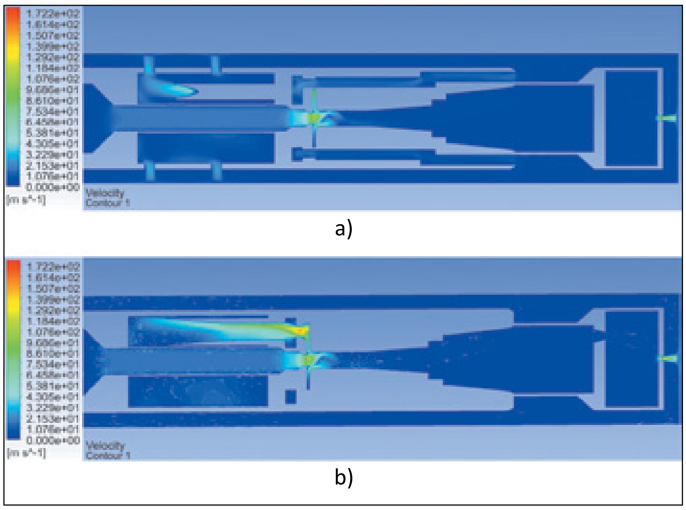

Figure 5. Velocity distribution in the flow path of the main (a) and auxiliary (b) jet pumps

Rysunek 5. Rozkład prędkości na ścieżce przepływu głównej (a) i pomocniczej (b) pompy strumieniowej

The pressure characteristics of the main and auxiliary jet pumps were obtained taking into account the possibility of their operation in the mode of negative values of the relative head and injection coefficient (Figure 7).

The operation of a jet pump in suction mode can be classified as a hydraulic machine work mode.

The numbers in brackets in Figure 7 correspond to the diameters of the equivalent diaphragm determined in 10^{-3} m. For comparison, the dotted line on the graph shows the classical theoretical characteristic of a jet pump, obtained using the methodology of Sokolov and Zinger (Sokolov and Zinger, 1989). The points corresponding to the diameters of the equivalent diaphragm 30×10^{-3} m and 100×10^{-3} m (curve I) characterize the operation of the main (pressure) jet pump in the suction mode, and the points corresponding to the diameters of the diaphragm 5×10^{-3} m; 8×10^{-3} m; 11×10^{-3} m and 27×10^{-3} m determine its operation in the mode of negative values of the injection coefficient (ejection system reverse mode of operation).

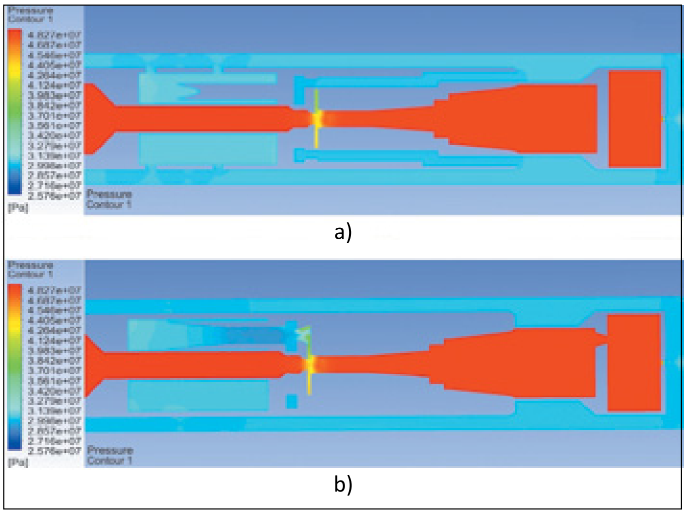

Figure 6. Pressure distribution in the flow path of the main (a) and auxiliary (b) jet pumps

Rysunek 6. Rozkład ciśnienia na ścieżce przepływu głównej (a) i pomocniczej (b) pompy strumieniowej

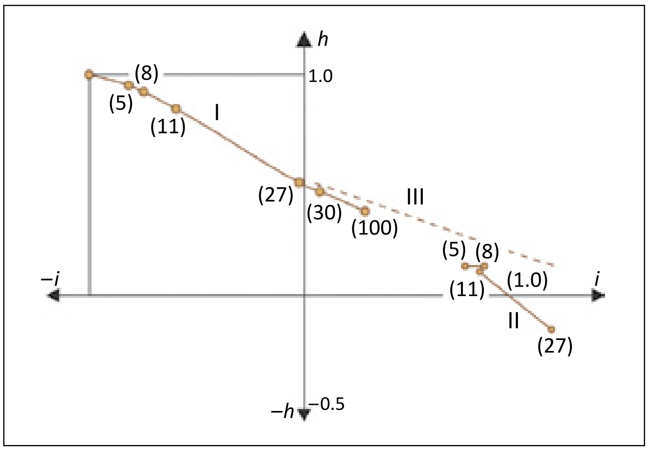

Figure 7. Dependence of the relative pressure on the value of the injection coefficient; I – the main jet pump, II – auxiliary jet pump, III – theoretical characteristic according to the basic method

Rysunek 7. Zależność ciśnienia względnego od wartości współczynnika tłoczenia; I – główna pompa strumieniowa, II – pomocnicza pompa strumieniowa, III – charakterystyka teoretyczna zgodna z metodą podstawową

The last calculated point for the main jet pump (obtained for an equivalent diaphragm diameter of 5×10^{-3} m) is connected to the imaginary boundary point of the characteristic. For the auxiliary (injection-suction) jet pump (curve II), points corresponding to the diameters of the equivalent diaphragm 5×10^{-3} m, 8×10^{-3} m; 11×10^{-3} m characterize its operation in the suction mode. In this case, the operation of the jet pump occurs with "backwater". It should be noted that the operation of the jet pump in the mode of negative values of the injection factor $i < 0$ or negative values of relative pressure $h < 0$ are not typical. Comparison of the results of computer simulation with the theoretical classical pressure characteristic (curve III) confirms the well-known increase in the error of use of this technique (Sokolov and Zinger, 1989) with increasing value of the injection coefficient of the jet pump.

The ratio of conditions corresponding to the operating modes of the main and auxiliary jet pumps is most fully illustrated by the dependence of the flow rate of the injected flow Q_{e1} of the main pump (equivalent to the operating flow rate Q_{w2} of the auxiliary jet pump) on the value of hydraulic losses ΔP at the sludged bottomhole (Figure 8).

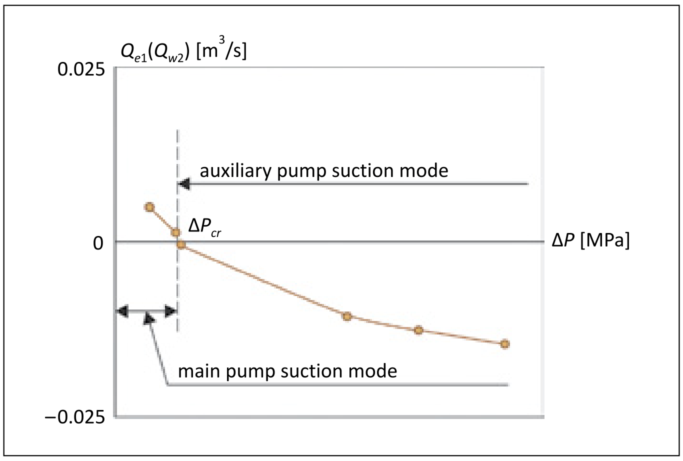

Figure 8. Dependence of the flow rate of an injected flow of the main jet pump (or the operating flow rate of an auxiliary pump) on the value of hydraulic losses in the sludged section of the bottom

Rysunek 8. Zależność natężenia przepływu głównej pompy strumieniowej (lub roboczego natężenia przepływu pompy pomocniczej) od wartości strat hydraulicznych w dolnym odcinku płuczkowym

The value of hydraulic losses in the sludged area of the bottomhole has been determined as the difference between the pressure of the mixed flow of the main jet pump and the pressure in the annular channel of the well (Figure 2) $\Delta P = P_{m1} - P_a$. For any values of hydraulic losses, the flow rate of the injected flow of the main jet pump is equal in absolute value to the flow rate of the working flow of an auxiliary jet pump $|Q_{e1}| = |Q_{w2}|$. Zero cost values $Q_{e1} = Q_{w2} = 0$ correspond to the critical value of hydraulic losses at the slurry bottomhole, the value of which, according to the calculations made, is $\Delta P_{cr} =$ $= 3.039$ MPa. Then, taking into account the obtained results, the values of hydraulic losses at the bottomhole from $\Delta P = 0$

to $\Delta P = 3.039$ MPa correspond to the operation in the suction mode of the main jet pump, and the value of hydraulic losses $\Delta P > 3.039$ MPa corresponds to the operation in the suction mode of an auxiliary jet pump.

Experimental study of ejection system characteristics

In the process of conducting experimental studies, the influence of the operating flow rate on the characteristics of the jet pump has been established. The experimental stand consists (Figure 9) of a working nozzle 1 and a mixing chamber with a diffuser 2 of two models of jet pumps, a suction manifold 3, a suction manifold 4 and a pressure line 5, a tank 6, a power pump 7, flow meters 8, 9, pressure gauges 10–12 and latches 13–15.

Adjustment of the working flow rate is carried out by changing the degree of opening of the valve 14. For each position of the locking element of the latch 14, using flow meters 8 and 9, the flow rates of the working Q_w and injected Q_e flows are determined, and using pressure gauges 10–12, the pressure value of the working P_w, injected P_e and mixed P_m flows are determined respectively. The results obtained are presented in the form of a relative head $h = (P_m - P_e)/(P_w - P_e)$ and an injection coefficient $i = Q_e / Q_w$ of a jet pump. Experimental studies were carried out for a jet pump with a ratio of the squares of the mixing chamber and the working nozzle diameters, respectively $K_p = 3.429$ and $K_p = 3.795$. In order to study the possibility of creating a vortex flow in the downhole ejection system, a twisted plate was installed in the working nozzle of the jet pump with an angle of inclination of the guide surfaces $\alpha = 15°$.

The influence of the working medium swirl on the value of the injection coefficient was established by comparing the experimental dependences $i = f(R_{ew})$ (where R_{ew} is the Reynolds number of the working flow) obtained for the direct-flow and swirling jets (Figure 10).

Taking into account the physical content of the process of creating pressure for approximating the results of experimental studies, it is advisable to choose an empirical function in the form of a horizontal asymptote, for example, a hyperbolic dependence of the following form:

$$i = \frac{R_{ew}}{b_0 + b_1 R_{ew}} \tag{3}$$

The values of the coefficients b_0 and b_1 of the empirical function and the correlation coefficients r for the swirl angles of the working flow $\alpha = 0$ and $\alpha = 15°$ are given in Table 1.

The dependencies $i = f(R_{ew})$ for any values of the geometric parameter K_p and swirl angles α of the working flow are nonlinear ascending (Figure 10).

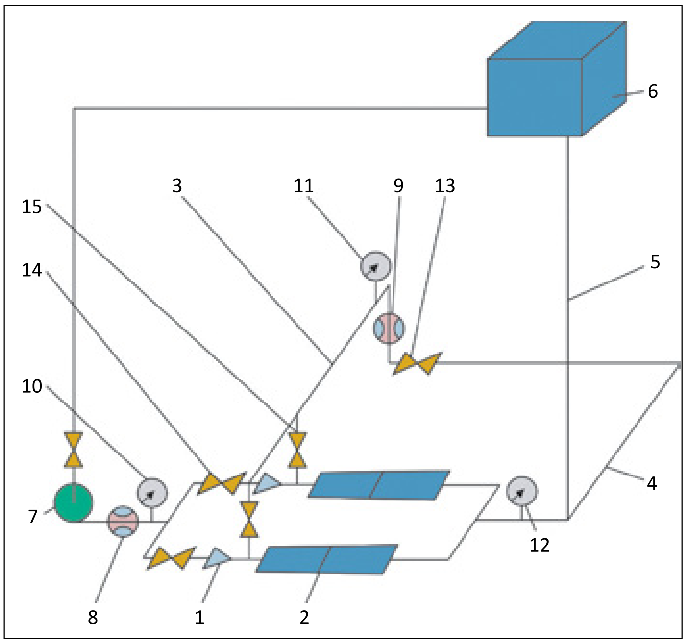

Figure 9. Experimental stand for the study of jet pumps
Rysunek 9. Stanowisko eksperymentalne do badania pomp strumieniowych

Table 1. Regression and correlation equation coefficients
Tabela 1. Współczynniki równania regresji i korelacji

Geometric parameter, K_p	Coefficients values					
	b_0		b_1		r	
	$\alpha = 0$	$\alpha = 15°$	$\alpha = 0$	$\alpha = 15°$	$\alpha = 0$	$\alpha = 15°$
3.429	11 195	4659	3.2271	3.2541	0.9978	0.9965
3.795	20 014	7705	3.3370	3.4560	0.9970	0.9976

The maximum discrepancy between the dependences $i = f(R_{ew})$ obtained for swirling and direct-flow working flows takes place for small values of the Reynolds number and equals: for a jet pump with a geometric parameter $K_p = 3.429$ – 16.98%, for a pump with a geometric parameter $K_p = 3.795$ – 20.84%. The impact of swirling of the working flow decreases with an increase in the Reynolds number, which is associated with a change in the ratio of the values of the axial and circular flow velocities. For a jet pump with the geometric parameter $K_p = 3.429$, the discrepancy between the maximum i_{max} and minimum i_{min} values of the injection coefficient for direct-flow and swirling flow is 27% and 13.55%, respectively. For a pump with geometrical parameter $K_p = 3.795$, the difference between the maximum i_{max} and minimum i_{min} values of the injection coefficient for direct-flow and swirling flow is 33.32% and 16.22%.

The lack of geometric, kinematic and dynamic similarity of jet pump designs used in computer modeling, experimental studies and influence of the Reynolds number of the workflow on the ejection coefficient, allowed us to limit only the qualitative comparison of the results. In particular, a 1.36-fold increase

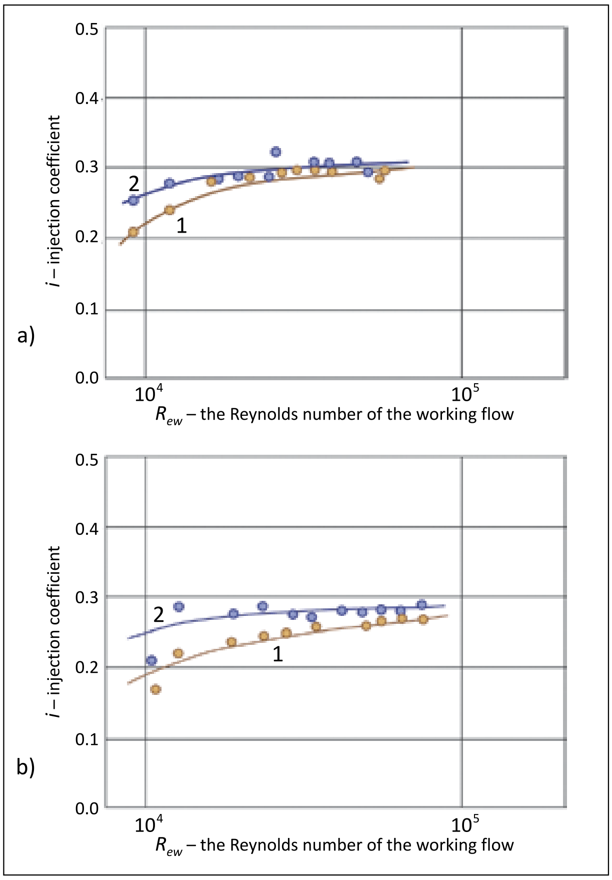

Figure 10. Dependence of the injection coefficient on the Reynolds number of the working flow for a jet pump with the value of the relative geometric parameter $K_p = 3.429$ (a) and $K_p = 3.795$ (b): 1 – direct-working flow; 2 – swirling workflow

Rysunek 10. Zależność współczynnika tłoczenia od liczby Reynoldsa przepływu roboczego dla pompy strumieniowej z wartością względnego parametru geometrycznego $K_p = 3{,}429$ (a) i $K_p = 3{,}795$ (b): 1 – bezpośredni przepływ roboczy; 2 – wirujący przepływ roboczy

in the Reynolds number of the workflow causes a 15.3% increase in the ejection coefficient based on computer simulations and an 8.1% increase in experimental studies.

Conclusions

1. The characteristics of the downhole dual-circuit ejection system in the injection jet pump, the nozzle of which is connected to the channel of the drill string, the diffuser – to the washing channel of the bit, and the mixing chamber – to the nozzle of the injection-suction jet pump, the diffuser of which is connected to the annulus channel, and the receiving chamber – to the overhead space have been studied:
   - the boundaries of operating modes areas of the injection and injection-suction jet pump and the conditions for automatic change (transition conditions) of the type of downhole ejection system have been established;
   - the operating mode of the suction jet pump demonstrates larger values of the injection coefficient than the pressure jet pump, which is due to different hydraulic resistance of the elements of individual circuits of the ejection system;
   - the influence of the degree of bottomhole sludge and the value of hydraulic resistance of its sludged section on the nature of the distribution of flows in the elements of the ejection system has been established;
   - the probability of the existence of non-operating modes of operation of a downhole jet pump has been analyzed: the modes of a negative injection coefficient (reverse mode of operation) and negative relative pressure (the mode of operation with "backup pressure").
2. The regularities of influence of the Reynolds number value of the working flow on the value of the injection coefficient of the jet pump have been established:
   - the injection coefficient and the Reynolds number of the working flow are associated with a non-linear ascending dependence;
   - swirling of the working flow allows us to increase the value of the injection coefficient. The maximum obtained increase in the value of the injection coefficient is 20.84%;
   - the effect of the working flow swirling decreases with an increase in the Reynolds number;
   - swirling of the working flow makes it possible to halve the discrepancy between the maximum and minimum values of the jet pump injection coefficient.

The task of further research is an industrial verification of the nature of the change in the kinematic and hydrodynamic parameters of the elements of a downhole dual-circuit ejection system as part of a parallel-series connection of an injection and injection-suction jet pump.

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