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# Simulation of a downhole jet-vortex pump's working process

## Symulacja procesu pracy wgłębnych wirowych pomp strumieniowych

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ABSTRACT: The article is devoted to the theoretical study of the operation process of the borehole ejection system as part of the tubing string, jet pump and packer installed below; the system implements the hydrojet method of oil well operation. The improved design of the jet pump contains inclined guiding elements placed in its receiving chamber for swirling the injected flow, which results in an increase in the efficiency of the borehole ejection system. Based on the law of conservation of liquid momentum in the mixing chamber of the jet pump and taking into account the inertial pressure component caused by the swirling of the injected flow, there is obtained the relative form of the equation of the ejection system pressure characteristic, the structure of which contains a component that determines the value of the additional dynamic head. According to the results obtained, the additional dynamic head caused by swirling of the injected flow is determined by the ratio of the geometric dimensions of the flow path of the jet pump, the angle of inclination of the elements for creating vortex flows, and the ratio of the power and reservoir fluids. In the case of asymmetric swirling of the injected flow, an increase in the value of the relative displacement of the jet pump decreases the value of the additional dynamic pressure. In order to study the effect of flow swirling on the energy characteristic of the ejection system, the pressure characteristic of the jet pump was transformed into the dependence of its efficiency on the injection coefficient. Jet pump models with the ratio of the cross-sectional areas of the mixing chamber and the nozzle of 5.012 and 6.464, respectively, were used to check the adequacy of the theoretical pressure and energy characteristics obtained during the simulation of the performance process of the concentric ejection system. The average error in the theoretical determination of the pressure and efficiency of the vortex jet does not exceed 8.65% and 6.48%, respectively.

Key words: downhole ejection system, oil jet pump, relative pressure, injection ratio, flow twist, efficiency.

STRESZCZENIE: Artykuł poświęcony jest teoretycznemu studium opisującemu proces pracy wgłębnego systemu wydobycia z odwiertu złożonego z kolumny rur wydobywczych, pompy strumieniowej i zainstalowanego poniżej pakera; przy użyciu tego systemu prowadzona jest eksploatacja odwiertów naftowych metodą hydrojet. Udoskonalona konstrukcja pompy strumieniowej zawiera skośne elementy prowadzące umieszczone w jej komorze zasilającej do zawirowania przepływu tłoczonego płynu, co skutkuje zwiększeniem wydajności systemu wydobycia z odwiertu. Na podstawie zasady zachowania pędu cieczy w komorze mieszania pompy strumieniowej oraz po uwzględnieniu bezwładnościowej składowej ciśnienia wywołanej zawirowaniem przepływu tłoczonego płynu otrzymuje się względną postać równania charakteryzującego ciśnienia w układzie ejekcyjnym, którego struktura zawiera składnik określający wartość dodatkowej wysokości podnoszenia. Zgodnie z uzyskanymi wynikami dodatkowa wysokość podnoszenia spowodowana zawirowaniem przepływu tłoczonego płynu jest określona przez: stosunek wymiarów geometrycznych toru przepływu pompy strumieniowej, kąt nachylenia elementów tworzących przepływ wirowy oraz stosunek płynów zasilających i złożowych. W przypadku asymetrycznego zawirowania przepływu tłoczonego płynu wzrost wartości względnej wyporności pompy strumieniowej powoduje zmniejszenie wartości dodatkowego ciśnienia dynamicznego. W celu zbadania wpływu zawirowania przepływu na charakterystykę energetyczną układu ejekcyjnego – charakterystykę ciśnieniową pompy strumieniowej przekształcono na zależność jej sprawności od współczynnika tłoczenia. Do sprawdzenia prawidłowości teoretycznych charakterystyk ciśnienia i energii uzyskanych podczas symulacji procesu pracy koncentrycznego układu ejekcyjnego wykorzystano modele pomp strumieniowych o stosunku powierzchni przekroju komory mieszania do dyszy odpowiednio 5,012 i 6,464. Średni błąd ciśnienia i wydajności strumienia wirowego wyznaczonych w sposób teoretyczny nie przekraczał odpowiednio 8,65% i 6,48%.

Słowa kluczowe: wgłębny system wydobywczy, pompa strumieniowa do wydobycia ropy naftowej, ciśnienie względne, współczynnik tłoczenia, przepływ skrętny, wydajność.

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## Introduction

The late stage of oil field operation is characterised by a decrease in reservoir pressure and an increase in the content of gas and water in the well product, as a result of which the flow rates of dehydration and separation increase. Produced water contained in a well can have a high degree of mineralisation, which has an extremely negative effect on the performance of underground equipment. In order to increase the oil recovery factor, it is necessary to allocate additional funds for the use of secondary oil production methods. The presence of dissolved gas during well production reduces the efficiency of traditional oil production methods. As a result of a pressure drop in the bottomhole formation zone, there is disturbed thermodynamic equilibrium, accompanied by deposits of tar, asphaltenes and paraffins on the surfaces of oilfield equipment. If the productive horizon is composed of weakly-cemented anemic rocks at the late stages of production, the sand content in the well production increases. The complication of conditions for the use of borehole pumps reduces the overhaul period of their operation. Thus, at a late stage of an oil field's development, the costs of maintaining and operating oilfield equipment increase significantly and the volume of oil production decreases.

Attempts to preserve the profitability of production prompted the development of a hydrojet method for operating oil wells.

The change in the method of well operation is associated with a change in the amount of costs aimed at maintaining and operating surface and underground equipment. At the initial stages of field development, the cost of lifting oil by the hydrojet method of operation exceeds the cost when using other methods of mechanised oil production. In particular, the costs associated with pumping the power fluid into the well during the implementation of the hydrojet operation exceed the energy costs associated with the use of plunger pumps. With an increase in the life of the field, the costs associated with the current maintenance of wells equipped with plunger pumps increase. Taking into account the peculiarities of jet pump operation technology, the current costs of their maintenance remain practically unchanged over time.

The use of the hydrojet method of oil production in this case makes it possible to increase the duration of the oil well operation.

For a long time, it was believed that the hydrojet method of operation is marked by high energy costs associated with direct oil recovery and lower costs for routine maintenance of equipment compared to other methods of oil field development. This made it possible to recommend the use of a hydrojet method of operation for ageing fields, when oil production conditions become more complicated and the costs associated with well workover increase. Attempts to reduce the energy consumption of the hydrojet method of well operation led to the development of a surface pumping unit (for directing the power fluid to the nozzle of the jet pump), which is driven by a gas engine. According to Diverse Energy Systems and Liberty Resources (Muster and Clark, 2016), during industrial studies of wells in the Bakken oil field, the replacement of an electrically driven plunger pump with a gas-driven jet pump of a surface pumping unit reduced energy costs by $ 437,990. The hydrojet method of operation is characterised not only by a reduction in the cost of routine maintenance (repair) of wells, but also by a decrease in energy consumption of oil production (when using a gas pump with a surface pumping agent). This circumstance makes it possible to expand the area of application of ejection technologies and to use a jet pump at earlier stages of oil field development.

According to the author, the growth in the share of using downhole ejection systems is due to the development of two technologies that are decisive for the development of jet pumps:

- replacement of borehole pump parts in a hydraulic way, which made it possible to reduce the cost of its maintenance;
- development of a gas-driven surface pumping unit that uses gas separated by well production.

The use of a gas lead made it possible to reduce the operating costs of the hydrojet method of oil production and made it economically feasible to use it at earlier stages of field development.

The simple design and absence of moving parts make it possible to use downhole ejection systems in difficult geological and technical conditions. A significant advantage of the use of borehole jet pumps is the increase in the efficiency of drilling (Kryzhanivskyi and Panevnyk, 2020) and oil production (Zafarullah et al., 2021), resumption of ageing fields operation (Carpenter, 2020a), and the ability to produce heavy oil with a significant content of paraffin (Carpenter, 2020b). The design of a borehole jet pump allows its use in coiled tubing units (Qinglong et al., 2020), as a result of which is the increase in the efficiency of using both types of oil and gas field equipment.

A significant disadvantage of downhole jet pumps is the small value of the ejection system efficiency (Panevnyk and Panevnyk, 2020). The efficiency of the downhole ejection system can be improved by swirling the mixing flows (Zhu et al., 2012a; 2012b; Morrall et al., 2018). The purpose of the research is to simulate the characteristics of the downhole ejection system for the case of the swirling of the injected flow and to experimentally verify the characteristics of a jet vortex pump.

## Construction of oil jet pump

The borehole ejection system (Fig. 1) consists of the tubing (1), borehole wall (2), mixing chamber with a diffuser (3) and working nozzle (4).

To separate the areas of high and low pressure, the schematic diagram of the ejection system includes a packer (5) installed below the oil jet pump. Work fluid is created by a surface pumping unit and is directed into the annulus of the well. The work fluid flows through the annulus to the working nozzle (4), where the area of low pressure is formed due to the high-speed leakage of the fluid.

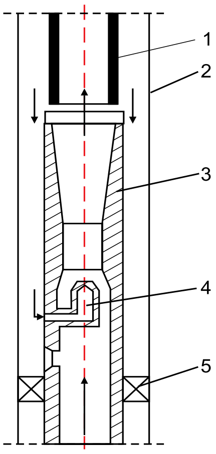



**Fig. 1.** Schematic diagram of the borehole ejection system: 1 – tubing string; 2 – borehole wall; 3 – mixing chamber and diffuser; 4 – working nozzle; 5 – packer

**Rys. 1.** Schemat ideowy systemu wydobywczego odwiertu eksploatacyjnego: 1 – kolumna rur wydobywczych; 2 – ściana odwiertu; 3 – komora mieszania i dyfuzor; 4 – dysza robocza; 5 – paker

The swirling of the mixed flows in the vortex ejection system allows increasing the injection coefficient and efficiency of the jet pump and decreasing the pressure in its suction line. The main advantage of using a vortex ejection system is determined by the type of technological process that is implemented using the jet pump. A high value of the injection ratio is a determining factor when choosing a jet pump design intended for flushing oil wells. The increase in the efficiency of the jet pump makes it possible to increase the efficiency of such long-term processes as direct oil recovery. Reducing the pressure in the suction line of the jet pump helps to reduce the duration of the stimulation treatment of the productive horizon.

The likelihood of cavitation in the process of interaction of the flow with the inclined guiding elements can be reduced by increasing the pressure in the suction line of the jet pump and decreasing the speed of fluid movement in the hydraulic channels of the vortex nozzle with an increase in the area of its flow sections. The value of the angle of inclination and the shape of the trailing edges of the elements for swirling the flow can also have a significant effect on the disruption of the flow continuity. The likelihood of cavitation effect in the flow path of a vortex jet pump decreases when using downhole ejection systems operated under conditions of significant (up to 40 MPa and more) hydrostatic pressures.

## Determination of jet pump characteristics for conditions of injected flow swirling

Analysis of modern designs of downhole ejection systems indicates the possibility of an asymmetric swirling of the flow when the center of the swirl is displaced relative to the axis of the jet pump.

The construction of a hydraulic model of the ejection system operation process (Sokolov and Zinger, 1989) is based on the use of the equation for conservation of the fluid momentum in the characteristic sections of the flow path of the jet pump (Fig. 2).

$$\begin{aligned}\varphi_2 (G_w V_{w1} + G_s V_{s2}) - (G_w + G_s) V_m = (P_{m3} - P'_{s2}) f_{s2} + \\ + (P_{m3} - P_{w1}) f_{w1} = P_{m3} f_3 - P'_{s2} f_{s2} - P_{w1} f_{w1}\end{aligned} \tag{1}$$

where: $\varphi_2$ = the velocity coefficient for the inlet section of the mixing chamber, $G_w$, $G_s$ = mass flow rates of the operating and injected flows, $V_{w1}$, $V_{s2}$, $V_m$ = velocities of the operating flow at the outlet of the nozzle, the injected flow at the inlet to the mixing chamber and the mixed flow at the outlet of the mixing chamber, $P_{m3}$, $P'_{s2}$, $P_{w1}$ = the pressures of the mixed flow at the outlet of the mixing chamber, injected swirl flow and working flow at the outlet of the nozzle, $f_{s2}$, $f_{w1}$, $f_3$ = a cross-sectional area of the injected flow at the mixing chamber inlet, the nozzle and the mixing chamber outlet.

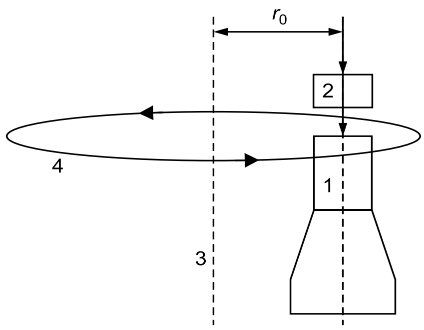



**Fig. 2.** Asymmetric swirling of the injected flow: 1 – mixing chamber and diffuser; 2 – working nozzle; 3 – axis of the well; 4 – trajectory of the liquid particles

**Rys. 2.** Asymetryczne zawirowanie przepływu tłoczonego płynu: 1 – komora mieszania i dyfuzor; 2 – dysza robocza; 3 – oś odwiertu; 4 – trajektoria cząsteczek cieczy

In the case of asymmetric swirling of the injected flow, the force in the inlet section of the mixing chamber of the jet pump is determined by the formula

$$F_s = P'_{s2} f_{s2} = \left( P_{s2} + \frac{\rho \omega_s^2 (r + r_0)^2}{2} \right) f_{s2} \tag{2}$$

where: $r_0$ = the distance between the axes of the well and the jet pump, $\omega_s$ = rotation velocity of fluid particles, $\rho$ = fluid density.

After integrating the forces acting on the elementary annular section $2\pi r dr$ of the injected flow (taking into account that $r_0$ = const), we obtain

$$F_s = \int_{r_w}^{r_0}\left(P_{s2} + \frac{\rho\omega_s^2(r+r_0)^2}{2}\right)2\pi r dr = P_{s2}\pi(r_c^2 - r_w^2) + \\ +\rho\omega_s^2\pi\left(\frac{r_c^4 - r_w^4}{4} + 2r_0\frac{r_c^3 - r_w^3}{3} + r_0^2\frac{r_c^2 - r_w^2}{2}\right) \tag{3}$$

where: $r_c$, $r_w$ = the radius of the mixing chamber and the nozzle.

The flow rates in the characteristic sections of the jet pump are determined taking into account the specific density of the operating $\upsilon_w$, injected $\upsilon_s$ and mixed $\upsilon_m$ flows

$$V_{w1} = \frac{G_w}{f_{w1}}\upsilon_w \\ V_{s2} = \frac{G_s}{f_{s2}}\upsilon_s \\ V_m = \frac{G_w + G_s}{f_3}\upsilon_m \tag{4}$$

After substituting formulas (2), (3), (4) into equation (1), we obtain

$$\varphi_2(G_w V_{w1} + G_s V_{s2}) - (G_w + G_s)V_m = \\ = (P_{m3} - P'_{s2})f_{s2} + (P_{m3} - P_{w1})f_{w1} = \\ = P_{m3}f_3 - P'_{s2}f_{s2} - P_{w1}f_{w1} \tag{5}$$

The relationship between the hydrodynamic parameters of the mixing chamber and characteristic cross-sections of a jet pump is carried out using relationships

$$P_{m3} = P_m - \frac{\varphi_3^2 V_m^2}{2\upsilon_m} = P_m - \frac{\varphi_3^2(G_w + G_s)^2\upsilon_m}{2f_3^2} \tag{6}$$

$$P_{s2} = P_s - \frac{V_{s2}^2}{2\varphi_4^2\upsilon_s} = P_s - \frac{G_s^2\upsilon_s}{2\varphi_4^2 f_{s2}^2} \tag{7}$$

$$P_{w1} = P_w \frac{V_{w1}^2}{2\varphi_1^2\upsilon_w} = P_w - \frac{G_w^2\upsilon_w}{2\varphi_1^2 f_{w1}^2} \tag{8}$$

where: $P_m$, $P_s$, $P_w$ = the pressure of the mixed flow at the outlet of the jet pump and the pressure of the injected and operating flow at the inlet to the jet pump, $\varphi_1$, $\varphi_3$, $\varphi_4$ = the velocity factor for the nozzle, mixing chamber outlet and the suction port.

Using formulas (6)–(8) and the concept of the injection coefficient, which is determined by the ratio of injected $Q_s$ and operating $Q_w$ flow rates $i = Q_s/Q_w$ after obvious transformations, the equation (5) can be written in the form as follows

$$h = \frac{P_m - P_s}{P_w - P_s} = \varphi_1^2\frac{f_{w1}}{f_3}\cdot A + \frac{\rho\omega_s^2\pi}{G_w^2}\cdot \\ \cdot\left(\frac{r_c^4 - r_w^4}{4} + \frac{2}{3}r_0(r_c^3 - r_w^3) + \frac{1}{2}r_0^2(r_c^2 - r_w^2)\right) \\ A = 2\varphi_2 + \left(2\varphi_2 - \frac{1}{\varphi_4^2}\right)\frac{i^2}{K_p - 1}\frac{f_{w1}}{f_3 - f_w}\frac{\upsilon_s}{\upsilon_w} - \\ -(2-\varphi_3^2)(1+i)^2\frac{f_{w1}}{f_3}\frac{\upsilon_m}{\upsilon_w} \tag{9}$$

where: $h$ = relative pressure of the jet pump, $K_p$ = the basic geometric parameter of the jet pump, $K_p = (f_3/f_{w1})$.

Given the obvious transformations

$$\frac{f_{w1}}{f_3} = \frac{1}{K_p};\ \frac{f_3 - f_{w1}}{f_{w1}} = K_p^{-1} \tag{10}$$

we obtain

$$h = \frac{\varphi_1^2}{K_p}\cdot B + \frac{\varphi_1^2\rho\omega_s^2\pi f_{w_1}^2}{G_w^2\upsilon_w f_3}\cdot \\ \cdot\left(\frac{r_c^4 - r_w^4}{4} + \frac{2}{3}r_0(r_c^3 - r_w^3) + \frac{1}{2}r_0^2(r_c^2 - r_w^2)\right) \tag{11}$$

In equation (11) is denoted

$$B = 2\varphi_2 + \left(2\varphi_2 - \frac{1}{\varphi_4^2}\right)\frac{i^2}{K_p - 1}\frac{\upsilon_s}{\upsilon_w} - \\ -(2-\varphi_3^2)\frac{(1+i)^2}{K_p}\frac{\upsilon_m}{\upsilon_w} \tag{12}$$

Let us transform the component of equation (11), which determines the additional dynamic pressure caused by swirling of the injected flow using the dependences

$$h_d = \frac{\varphi_1^2\rho\omega_s^2 f_{w1}^2}{2G_w^2 f_3\upsilon_w\pi}a \tag{13}$$

where

$$a = \left((r_c^4 - r_w^4) + \frac{8}{3}r_0(r_c^3 - r_w^3) + 2r_0^2(r_c^2 - r_w^2)\right)\pi^2$$

In the case of a zero displacement of the injected flow swirling $r_0 = 0$, equation (13) turns into a composite of the equation, which determines the value of the additional dynamic head for symmetric swirling of the injected flow.

Let us determine the form of the equation for calculating the angular velocity of the asymmetric swirl of the injected flow.

The rotating component of the mean velocity of the injected flow taking into account Fig. 2 can be determined by the formula

$$V_{\theta s} = \frac{\omega_s \left( \frac{d_c}{2} + \frac{d_w}{2} + d_0 \right)}{2} \tag{14}$$

where: $d_c$, $d_w$ = diameter of the mixing chamber and the nozzle, $d_0 = 2r_0$.

Then the resulting velocity of the swirling injected flow can be determined by the relationship

$$V_s = \frac{V_{\theta s}}{\sin \alpha_s} = \frac{\omega_s \left( \frac{d_c}{2} + \frac{d_w}{2} + d_0 \right)}{2 \sin \alpha_s} \tag{15}$$

where: $\alpha_s$ = the inclination angle of guiding elements for swirling the injected flow.

Taking into account the axial component $V_{0s}$ we obtain

$$V_s = \frac{V_{0s}}{\cos \alpha_s} = \frac{4Q_s}{\pi (d_c^2 - d_w^2) \cos \alpha_s} \tag{16}$$

where: $Q_s$ = volumetric flow rate of the injected flow.

After joint solving of equations (15), (16), we obtain

$$\omega_s = \frac{8Q_s \sin \alpha_s}{\pi (d_c^2 - d_x^2) \cos \alpha_s \left( \frac{1}{2}\sqrt{K_p} + \frac{1}{2} + K_0 \right) d_w} \tag{17}$$

where: $K_0 = d_0 / d_w$.

Given the obvious relationships

$$\operatorname{tg} \alpha_s = \frac{\sin \alpha_s}{\cos \alpha_s} \;\; ; \;\; f_s \frac{\pi (d_c^2 - d_w^2)}{4}$$

we finally get

$$\omega_s = \frac{4Q_s \operatorname{tg} \alpha_s}{f_s (1 + K_p^{0.5} + 2K_0) d_w} \tag{18}$$

The analysis of the obtained equation shows that, under conditions of asymmetric swirling of the injected flow, the angular velocity of fluid particles rotation is lower than in the case of symmetric swirling. After substituting formula (18) into equation (13) and corresponding transformations, it can be written

$$h_d = \frac{2\varphi_1^2 i^2 \operatorname{tg}^2 \alpha_s}{(1 + K_p^{0.5} + 2K_0)^2 (K_p - 1)} \frac{a}{f_3 f_s} = \frac{2\varphi_1^2 i^2 \operatorname{tg}^2 \alpha_s}{(1 + K_p^{0.5} + 2K_0)^2 (K_p - 1)} A \tag{19}$$

where: $$A = \frac{a}{f_3 f_s} = \frac{(r_c^4 - r_w^4) + \frac{8}{3} r_0 (r_c^3 - r_w^3) + 2r_0^2 (r_c^2 - r_w^2)}{r_c^2 (r_c^2 - r_w^2)}$$

In the process of determining the form of relationship (19), the formulas for calculating the areas of the mixing chamber and the injected flow were taken into account.

In the case of symmetric swirling of the injected flow $r_0 = 0$, therefore $A = 1 + K_p^{-1}$. After substitution of the values $K_0 = 0$, $A = 1 + K_p^{-1}$ into the equation (19) we obtain the formula for determining the additional head caused by symmetric swirling of the injected flow.

Let's compare the values of additional heads caused by asymmetric and symmetric swirling of the injected flow. We divide the amount of additional dynamic head caused by asymmetric swirling (equation (19)) by the amount of additional dynamic head caused by symmetric swirling

$$\bar{h}_d = \frac{A}{\left( 1 + \frac{2K_0}{1 + K_p^{0.5}} \right)^2 (1 + K_p^{-1})} \tag{20}$$

In case $K_0 = 0$, $A = 1 + K_p^{-1}$ relation (20) takes the value $\bar{h}_d = 1$.

Considering that the asymmetric swirling of the injected flow reduces the angular velocity of fluid particles rotation in comparison with the symmetric swirling, the ratio of the magnitude $\bar{h}_d$ throughout the entire range of variation of the distance between the axes of the well and the jet pump $r_0$ takes on a value less than one.

An increase in the swirl asymmetry (or relative displacement of the jet pump $r_0$) decreases the amount of additional dynamic head. An increase in the value of the main geometric parameter $K_p$ increases the discrepancy between the additional heads caused by asymmetric and symmetric swirling of the injected flow.

## Experimental verification of the characteristics of a jet vortex pump

The hydraulic diagram for the experimental study of the jet pump characteristics is shown in Figure 3. The laboratory unit consists of a jet pump in the form of a mixing chamber (1) connected to a diffuser, and the nozzle (2). The jet pump is connected to the pressure line (3), suction line (4) and the vessel installation height (5). The pumping of the flow is carried out using a centrifugal pump (6), the suction line (7) of which is connected to the reservoir (5), and the initial line is connected to the work nozzle (2). The flow rate of the working and injected flows is determined by flow meters (8), (9).

$$\eta = hi / (1 - h) \tag{21}$$

Pressure values of the working, injected and mixed flows are determined by manometers (manovacuum meters) (10),

(11) and (12). The gate valve (13) allows us to change the flow rate of the injected flow. The change of the jet pump injection coefficient is achieved by adjusting the degree of opening of valve (13), which changes the hydraulic resistance of the suction line (4) and the operating mode of the injection system.

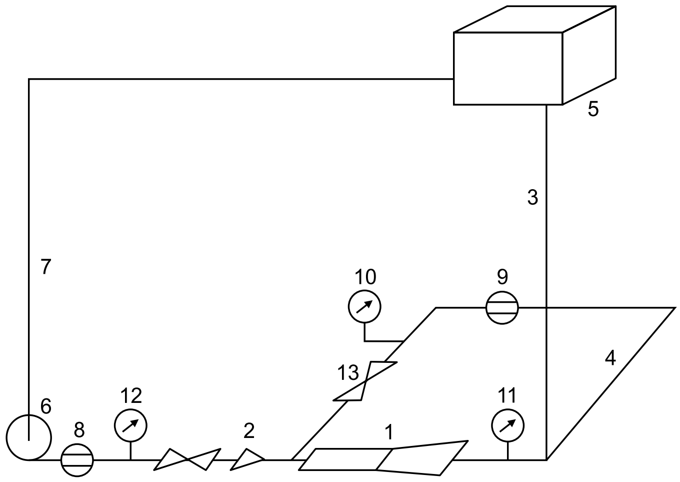



**Fig. 3.** Laboratory setup for investigating the operating process of a jet pump in conditions of swirling injected flow (description in the text)

**Rys. 3.** Stanowisko laboratoryjne do badania procesu pracy pompy strumieniowej w warunkach przepływu wirowego (opis w tekście)

Installing the reservoir (5) above the level of the jet pump allows maintaining a constant hydrostatic pressure in the system. The mixing chamber (1) with the diffuser and the suction line (4) form a closed circulation loop with the injected flow rate, the value of which is determined by the operating mode of the jet pump. After leaving the diffuser of the jet pump, the mixed flow is divided: part of the flow with a flow rate equal to the flow rate of the working flow moves upward along the pressure line (3), and another part forms the injected flow, which is directed to the suction line (4).

The pressure value of the working, mixed and injected flows, as well as the flow rate of the working and injected flows are measured for different degrees of valve (13) opening. The flow rate of the working flow is kept constant.

The determination of experimental ratios of relative heads and flow rates was carried out for a direct-flow and vortex jet pump. In the case of an experimental study of a vortex jet pump, inclined elements made using 3D printing technology were installed in its receiving chamber.

After determining pressure of the working $P_w$, mixed $P_m$ and injected $P_s$ flows there are determined dimensionless indicators: relative pressure $h = (P_m - P_s) / (P_w - P_s)$ and relative flow (injection ratio) $i = Q_s / Q_w$. After determining the relative pressure and injection ratio the efficiency of jet pump is determined.

The adequacy of the theoretical pressure characteristics was checked for jet pumps, the geometrical dimensions of which are given in Table 1.

The relative distance between the nozzle and the mixing chamber was determined as the ratio of a nozzle diameter $d_w$ and an absolute distance $l_w$ to a mixing chamber $\overline{l}_w = d_w / l_w$. The relative length of the mixing chamber was calculated as the ratio of its absolute length $l_c$ and the diameter $d_w$ of the nozzle $\overline{l}_c = l_c / d_w$.

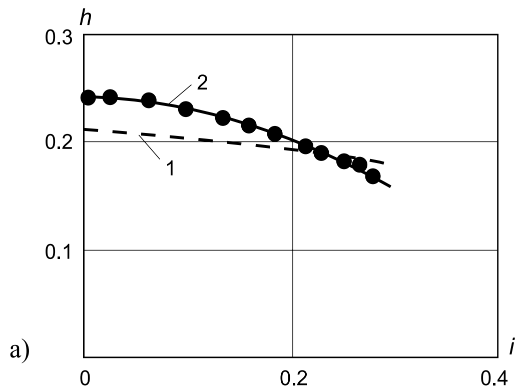
**Table 1.** Geometric dimensions of jet pump models

**Tabela 1.** Wymiary geometryczne modeli pomp strumieniowych

| N° | $K_p$ | $\overline{l}_w$ | $\overline{l}_c$ |
|---|---|---|---|
| 1 | 6.464 | 1.53 | 20.3 |
| 2 | 5.012 | 1.04 | 17.9 |

In the process of theoretically determining the pressure of the jet pump, equation (19) was used for the case of swirling of the injected flow by guiding elements with an angle of inclination $\alpha_s = 45°$ and the distance between the axes of the well and the jet pump $r_0 = 0$.

Theoretical values of pressures for each studied jet pump model are compared with the empirical values determined using the regression equations based on the results of experimental studies and are shown in Table 2 (Fig. 4).



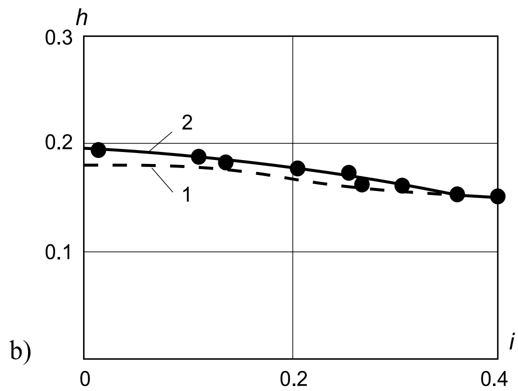




**Fig. 4.** Theoretical (1) and experimental (2) pressure characteristics of the jet pump for different values of the geometric parameter $K_p = 5.012$ (a); $K_p = 6.464$ (b)

**Rys. 4.** Teoretyczne (1) i eksperymentalne (2) charakterystyki ciśnienia wyznaczone dla pompy strumieniowej dla różnych wartości parametru geometrycznego $K_p = 5{,}012$ (a); $K_p = 6.464$ (b)

**Table 2.** The values of empirical coefficients of the regression equations for the pressure characteristics of jet pumps

**Tabela 2.** Wartości współczynników empirycznych równań regresji dla charakterystyk ciśnieniowych pomp strumieniowych

| $K_p$ | *a* | *b* | *c* |
|---|---|---|---|
| 6.464 | 0.1884 | –0.4642 | 1.2492 |
| 5.012 | 0.2365 | –0.9642 | – |

The regression equations for the pressure characteristics of jet pumps have the form:

for $K_p = 5.012$ $\quad h = a + bi^2$

for $K_p = 6.464$ $\quad h = a + bi^2 + ci^4$

Comparative analysis of the theoretical and experimental pressure characteristics of the jet pump for various values of the geometric parameter $K_p$ (Fig. 4) have shown that the average error in the theoretical determination of the pressure varies in the range from $\delta\bar{h} = 5.74\%$ to $\delta\bar{h} = 8.65\%$.

An increase in the value of the main geometric parameter of the jet pump (Fig. 4b) causes an increase in the flow rate and the Reynolds numbers of the mixed flows. The actual values of the velocity coefficients for the individual elements of the flow path of the jet pump, which were taken unchanged in the theoretical determination of its pressure characteristic, approach constant values with an increase in the Reynolds number. This circumstance explains the decrease in the error in the theoretical determination of the pressure characteristics of the jet pump with an increase in the value of the main geometric parameter.

Comparison of theoretical and experimental values of efficiency has been carried out for a jet pump with the value of the main geometric parameter $K_p = 6.464$ (Fig. 5).

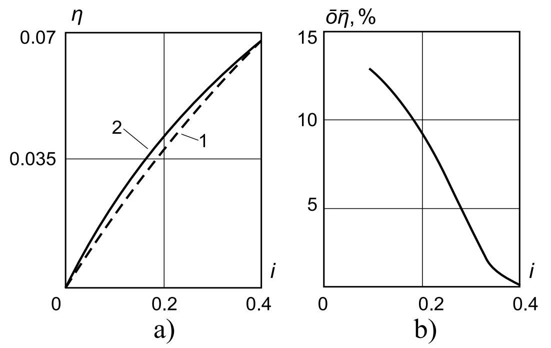



**Fig. 5.** Comparison of energy characteristics of the jet pump: a) theoretical (1) and experimental (2) dependences of the efficiency on the injection coefficient; b) relative error of theoretical and experimental values of efficiency

**Rys. 5.** Porównanie charakterystyk energetycznych pompy strumieniowej wyznaczonych: a) teoretycznie (1) i eksperymentalnie (2) zależności sprawności od współczynnika tłoczenia; b) błąd względny wartości sprawności wyznaczonych w sposób teoretyczny i eksperymentalny

Theoretical and experimental values of the efficiency were calculated using the formula (21). Theoretical values of the pressure were used to determine the theoretical values of the efficiency. According to the results of the experimental studies, the average error in determining the theoretical values of the efficiency is $\delta\bar{\eta} = 6.48\%$.

The discrepancy between theoretical and experimental values of the relative pressure and efficiency is largely explained by the design of the laboratory unit and peculiarities of the jet pump operating process under conditions of the swirling flow.

The use of a power drive in the form of a centrifugal pump in the design of a laboratory unit makes its flow rate dependent on the hydraulic resistance of the flowing part of the jet pump. In fact, the tests investigate the joint operation of a centrifugal and jet pump. The flow swirling in the flow path of the jet pump reduces the amount of energy required to mix the working and injected flows.

Reduction of energy losses during mixing of flows and hydraulic resistance of the flowing part of the jet pump increases the flow rate of the working flow, which is created by the centrifugal pump due to the lack of rigid characteristics for this type of hydraulic machine. The increase in operating flow rates, in its turn, affects the operating process of the jet pump and the value of its relative pressure.

## Conclusions

The use of the hydrojet method of oil recovery at the late stages of oil field development enables us to increase the overhaul period and serviceability of underground equipment, reduce the cost of well production, maintain production profitability and increase the exploitation time of depleted hydrocarbon deposits. The increase in the energy efficiency of the hydrojet method of oil extraction is largely due to the development of technologies for replacing parts of a downhole pump with a hydraulic method and the use of a gas pump for a surface pumping unit and makes it economically justified to use it at earlier stages of oil field exploitation. The combined use of hydrojet and coiled tubing units allows implementing the advantages of using both layouts in a single unit.

Under conditions of asymmetric swirling of the injected flow, the angular velocity of fluid particles rotation and the additional dynamic pressure created by the jet pump take on lower values than in the case of symmetric swirling. An increase in the swirl asymmetry, that is, an increase in the value of the relative displacement of the jet pump, decreases the value of additional dynamic pressure. The discrepancy between additional pressures caused by the asymmetric and symmetric swirling of the injected flow and the value of the main geometric parameter of the jet pump are related to each other by a direct proportional relationship:

- the average error of the theoretical determination of the pressure characteristic of a vortex jet pump varies in the range from 5.74% to 8.65% depending on the value of the main geometric parameter;
- the average error in determining the theoretical values of the jet pump efficiency is 6.48%.

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