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# Investigation of hydraulic characteristics of a downhole jet pump

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**Abstract**

The paper considers the possibility of increasing the energy efficiency of downhole ejection systems by twisting the injected flow with inclined guide elements placed in the flow part of a jet pump. Based on the conservation laws for energy, momentum and flow continuity, there was developed a mathematical model of the vortex ejection system in the form of direct-flow working and helical injected jets, which allowed obtaining the equation of pressure and energy characteristics of a high-pressure jet pump. In the case of zero-twist angle of the injected flow, the obtained dependences take the form of classical known equations that characterize the working process of a direct-flow jet pump. In the process of experimental studies of hydraulic characteristics of a high-pressure jet pump for the case of the injected flow twisting, there was obtained an increase in the injection ratio, relative pressure and efficiency of the jet pump to 19.1, 16.9 and 21.3 %, respectively. The error obtained during the experimental verification of the pressure characteristic of the vortex jet pump does not exceed 9.5 %.

**Keywords:** *ejection system, energy efficiency, flow twist, injection ratio, jet pump.*

According to the BP Energy Outlook 2017, despite the fact that non-fossil fuels will provide half of the growth in energy demand by 2035, hydrocarbons and coal will remain its main sources and will account for 78 % of global consumption [1]. Meeting the needs of global energy consumption requires improved technologies for oil and gas production. The increasing complexity of conditions for the development of hydrocarbon fields necessitated the use of innovative technologies for drilling, operating and repairing oil and gas wells: attracting areas with non-degraded residual deposits by drilling of sidetracks, drilling under negative differential pressure during the primary opening of the productive horizon, reducing the time for well construction using top drive systems of drilling rigs, the use of alternative drives of equipment for mechanized oil production, the introduction of coiled tubing technologies and layouts for multi-stage hydraulic fracturing, minimization of negative consequences of "killing" wells in the process of implementing high-tech methods of repairing them under pressure (Snubbing), an increase in the exploitation of coastal oil and gas regions and the development of deposits of shale hydrocarbons.

A significant amount of innovative methods being implemented for the development of hydrocarbon deposits is associated with the possibility of using oil and gas ejection technologies. Placing a above-bit jet pump in the lower part of a drill string allows creating a local pressure drop at the downhole [2] while maintaining positive differential pressure in other sections of the well. At the same time, the natural permeability of the productive horizon is preserved during its initial opening. The use of combined assemblies makes it possible to increase the efficiency of using traditional methods of mechanized oil production [3]. A significant reduction in the cost of oil production is associated with the combustion of separated petroleum gas in the gas drive of a surface pumping unit of a downhole jet pump [4]. The joint arrangement of the jet pump and coiled tubing string made it possible to improve the quality of well cleaning after hydraulic fracturing [5]. The hydrojet method of operation allows increasing the production rate of wells at initial stages of the operation of shale oil fields [6] (Liberty Resources, USA). The compactness of the surface equipment of the downhole ejection system made it possible to use it in conditions of limited dimensions of offshore oil and gas platforms [7] (Weatherford International). The ability of the jet pump to create a significant reduction in pressure is used to restore the exploitation of depleted hydrocarbon fields [8].

The defining disadvantage of using downhole ejection systems is associated with peculiarities of the process of mixing flows in the flow path of the jet pump [9]. The presence of vortices accompanying the connection of flows with a significant difference in velocities and pressures is accompanied by significant hydraulic losses and is the reason for the low energy efficiency of ejection technologies. The negative influence of insufficient values of the jet pump efficiency is most fully manifested in the

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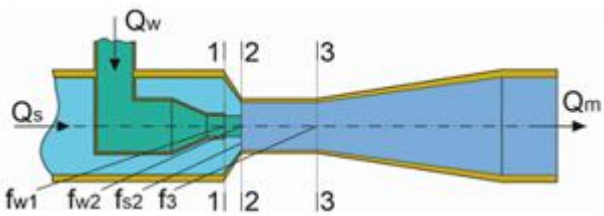
implementation of long-term ejection systems operations. Due to the simple design, versatility and prevalence of downhole ejection systems, the study of the working process of a jet pump, aimed at improving the energy performance of its use, is an urgent task.

One of the ways to optimize the operation process and increase the energy efficiency of using downhole ejection systems is swirling mixed flows, which can be carried out using inclined guide elements or by direct rotation of individual parts of the flow path of a jet pump. In the transfer of energy from a high-pressure working to a low-pressure injected flow, not only friction forces take part, but also centrifugal forces caused by the rotation of liquid particles, as a result of which jet devices acquire additional properties of dynamic pumps characteristic of the working process. In the process of analyzing the comparative efficiency of vortex jet devices, it is found that swirling the flow makes it possible to increase the injection coefficient up to 38.1 %, the efficiency up to 70 %, and the vacuum in the receiving chamber up to 40 % [10]. Despite a significant number of researches devoted to the working process of vortex jet pumps [11–14], the authors, as a rule, limit themselves to determining the experimental characteristics of ejection systems. The purpose of the studies, the results of which are presented in this work, is to simulate the working process of a downhole ejection system for the conditions of swirling the injected flow, determine the theoretical pressure and energy characteristics of a high-pressure jet pump, experimentally study its working process and test the obtained analytical dependencies.

According to the design diagram of the hydraulic model of the jet pump (Fig. 1), the working flow with the flow rate $Q_w$ enters the working nozzle of the jet pump.




*1–1 – outlet section of the working nozzle;*
*2–2 – inlet section of the mixing chamber;*
*3–3 – initial section of the mixing chamber*
$f_{w1}$ *– cross-sectional area of the working flow at the outlet of the working nozzle;*
$f_{w2}$ *– cross-sectional area of the working flow at the entrance to the mixing chamber;*
$f_{s2}$ *– cross-sectional area of the injected stream at the entrance to the mixing chamber;*
$f_3$ *– cross-sectional area of the mixing chamber*

**Figure 1 – The flow path of the jet pump**

Due to the high flow rate of a working stream, a low pressure region is created at the outlet of the working nozzle, as a result of which conditions arise for the addition of the injected stream with a flow rate $Q_s$. In the mixing chamber with a diffuser, there are restored the velocity and pressure profile, after which the mixed flow with the flow rate $Q_m$ enters the pressure line of the jet pump.

The equation of conservation of momentum in a fluid for the mixing chamber of a jet pump has the following form

$$\varphi_2\left(G_w v_{w2}+G_s v_{s2}\right)-\left(G_w+G_s\right)v_m = \\ = p_{m3} f_3 - p'_{s2} f_{s2} - p_{w2} f_{w2}\ , \tag{1}$$

where $\varphi_2$ is the velocity coefficient for the inlet section of the mixing chamber; $G_w, G_s$ are mass flow rates of the working and injected flows; $v_{w2}, v_{s2}, v_m$ are the velocities of the working flow at the inlet of the mixing chamber, the injected flow at the inlet of the mixing chamber and the mixed flow at the outlet of the mixing chamber; $p_{m3}, p'_{s2}, p_{w2}$ is pressure of the mixed flow at the outlet of the mixing chamber, injected swirl flow and working flow at the inlet of the mixing chamber; $f_{s2}, f_{w2}, f_3$ is the cross-sectional area of the injected flow at the inlet of the mixing chamber, the working flow at the inlet of the mixing chamber and the outlet section of the mixing chamber.

The hydraulic force $P'$ created by the swirling injected flow in the inlet section of the mixing chamber is determined by integrating the unit forces acting on the area of the elementary cross section

$$P' = p_{s2}\pi\left(r_c^2 - r_w^2\right) + \rho\omega_s^2\frac{\pi}{4}\left(r_c^4 - r_w^4\right), \tag{2}$$

where $p_{s2}$ is the injected direct flow pressure $\rho$ is the fluid density; $\omega_s$ is the angular velocity of injected flow rotation; $r_c, r_w$ are the radii of the mixing chamber and the working nozzle.

The flow rates in the characteristic sections of the jet pump are determined taking into account the density of the working $\rho_w$, injected $\rho_s$ and mixed $\rho_m$ flows

$$v_{w2} = \frac{G_w}{f_{w2}\rho_w}, \\ v_{s2} = \frac{G_s}{f_{s2}\rho_s}, \\ v_m = \frac{G_w + G_s}{f_3\rho_3} \tag{3}$$

The relationship between the hydrodynamic parameters of flows in the mixing chamber and the characteristic sections of the jet pump is carried out using obvious relationships

$$p_{m3} = p_m - \frac{\varphi_3^2\left(G_w + G_s\right)^2}{2 f_3^2 \rho_m}, \tag{4}$$

$$p_{w2} = p_w - \frac{G_w{}^2}{2\varphi_1^2 f_{w1}^2 \rho_w} - \Delta p_k\,, \tag{5}$$

$$p_{s2} = p_s - \Delta p_k\,, \tag{6}$$

where $p_m, p_s, p_w$ are pressure values of the mixed, injected and working flows; $\varphi_1$ is the velocity coefficient of the working nozzle; $\varphi_3$ is the velocity coefficient for the outlet section of the mixing chamber; $\Delta p_k$ is hydraulic losses in the flow between sections 1–1 and 2–2.

The process of swirling the injected flow is characterized by the ratios as follows

$$v=\frac{v_0}{\cos\alpha_s}, v=\frac{v_\theta}{\sin\alpha_s}, v_0=\frac{4Q_s}{\pi d_c^2}, v_0=\omega_s\frac{d_c}{2},$$

$$\omega_s=\frac{8}{\pi}\frac{Q_s}{d_c^3}tg\alpha_s, \frac{v_\theta}{v_0}=tg\alpha_s, \qquad (7)$$

where $v$ is the velocity of the swirling flow; $v_\theta, v_0$ are rotating and axial components of velocity; $\alpha_s$ is the angle of inclination of the guiding elements for swirling the injected flow; $d_c$ is the diameter of the mixing chamber of the jet pump.

When deriving the equation of the pressure characteristic of a high-pressure jet pump, dimensionless relations are used

$$h=\frac{p_m-p_s}{p_w-p_s},\; i=\frac{Q_s}{Q_w},\; K_p=\frac{f_3}{f_{w1}}, \frac{f_3-f_{w1}}{f_{w1}}=K_p^{-1}, \qquad (8)$$

where $h$ is the relative pressure of the jet pump; $i$ is the injection coefficient of the jet pump; $K_p$ is the basic geometric parameter of the jet pump.

According to the existing classification, a high-pressure jet pump corresponds to the value of the basic geometric parameter $K_p \le 4{,}0$.

Solution of equations (1)–(8) in accordance with the methodology of VTI of F.E. Dzerzhinsky [15], taking into account the peculiarities of screw flows movement, makes it possible to determine the pressure characteristic of a vortex high-pressure jet pump

$$h=\frac{\varphi_1^2}{K_p}\left[2\varphi_2\frac{f_{w1}}{f_{w2}}+2\varphi_2 i^2\frac{f_{w1}}{f_{s2}}-\left(2-\varphi_3^2\right)\frac{(1+i)^2}{K_p}\right]-$$

$$-\frac{\Delta p_k}{\Delta p_w}+\frac{2\varphi_1^2 i^2 tg^2\alpha_s\left(1+K_p^{-1}\right)}{\left(1+K_p^{0,5}\right)^2\left(K_p-1\right)}, \qquad (9)$$

where $\Delta p_w$ is the difference between the pressures of the working and injected flows, $\Delta p_w = p_w - p_s$.

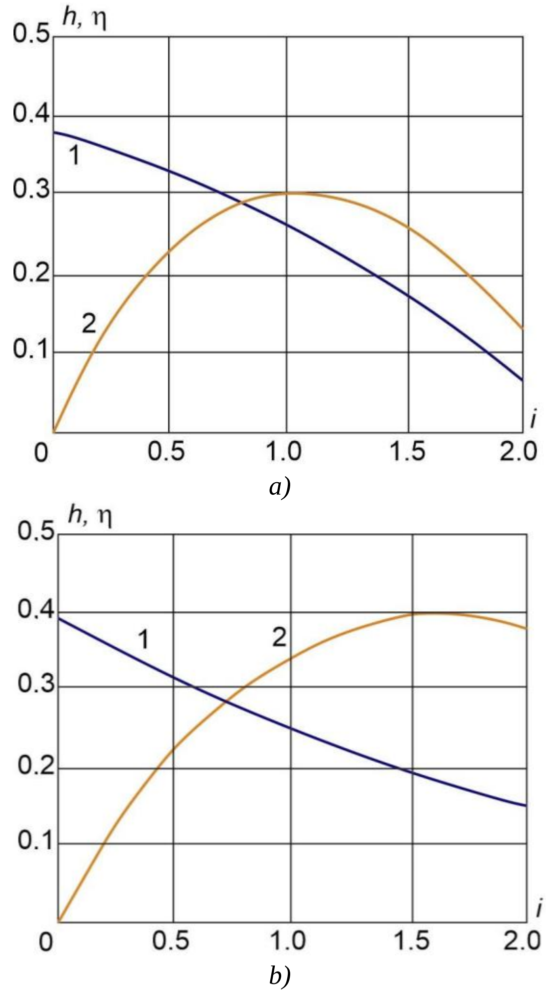
The last component of equation (9) determines the value of an additional dimensionless dynamic head created by swirling the injected flow. For zero values of the injection coefficient $i$ = 0, equation (9) turns into the equation obtained by N. E. Sokolov, N. M. Zinger [15].

The current ratios of relative pressure and injection ratio can be represented as the energy characteristics of the jet pump

$$\eta=\frac{hi}{1-h}, \qquad (10)$$

where $\eta$ is the efficiency of the ejection system.

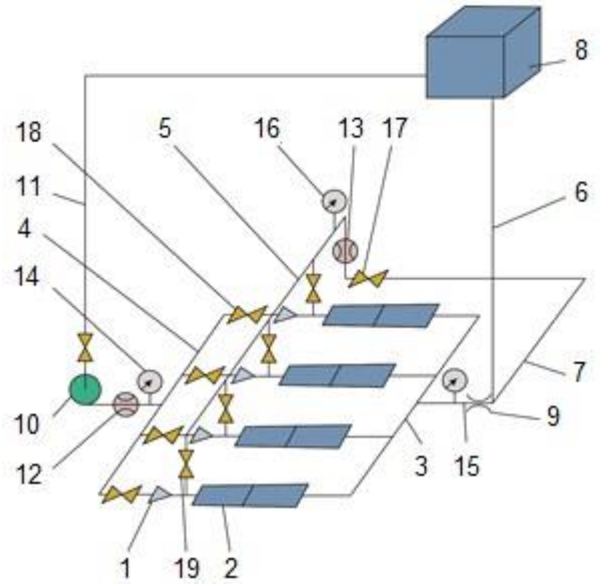
According to the results of comparative analysis (Fig. 2) of the energy characteristics obtained using equations (9), (10) it is established that the twisting of the injected flow allows to increase the maximum value of the efficiency of the jet pump by 25 %. According to the obtained results, the value of the dynamic pressure caused by the twisting of the injected flow and the injection coefficient of the jet pump are directly proportional.




*a) no swirling of the flow;*
*b) swirling of an injected flow*

**Figure 2 – Comparative analysis of pressure (1) and energy (2) characteristics of a high-pressure jet pump**

An experimental laboratory setup for studying the working process of a jet pump (Fig. 3) consists of several parallel jet pumps. The mixed flow after leaving the diffuser of the jet pump is divided into two parts: part of the mixed flow enters the reservoir 8 through the vertical hydraulic channel 6, and another part is directed to the suction line 5 of the laboratory installation along the bypass line 7. Placing the reservoir 8 above the level of the jet pump installation allows maintaining a stable hydrostatic pressure in the hydraulic system of the installation. The hydraulic load on the jet pump is formed using a throttling device 9.




*1 – working nozzle; 2 – mixing chamber with diffuser; 3 – outlet line; 4 – inlet line; 5 – suction line of the jet pump; 6 – vertical hydraulic channel; 7 – bypass line; 8 – tank; 9 – throttle device; 10 – centrifugal pump; 11 – suction line of the centrifugal pump; 12, 13 – water meter of working and injected flow; 14, 15, 16 – manometers for measuring the pressure of the working, injected and mixed flows; 17, 18, 19 – adjusting latches*

**Figure 3 – Laboratory setup for studying the working process of the jet pump**

The valve 17 allows one to change the operating mode of the pump and the flow rate of the injected flow. The valves 18 and 19 allow directing the flow rate of the working and injected flow to the jet pump, which is being studied. The design of the laboratory unit allows the study of both a single jet pump and parallel connection of several pumps.

During the experimental study of ejection system characteristics, there were used models of a high-pressure jet pump with different values of the main geometric parameter $K_p$, the distance between the working nozzle and the mixing chamber and the length of the mixing chamber (Table 1).

The relative distance between the working nozzle and the mixing chamber was defined as the ratio of the diameter $d_w$ of the working nozzle and the absolute distance $l_w$ to the 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 working nozzle $\overline{l}_c = l_c / d_w$.

At the first stage of experimental research, there is determined the influence of the operating flow of a jet pump on the value of the injection coefficient of the jet pump. The results of experimental studies are presented as the dependence of the injection coefficient value of the jet pump on the Reynolds number of the workflow $i = f\left(Re_w\right)$

$$Re_w = \frac{4Q_w}{\pi d_w \nu}\ , \qquad (11)$$

where $\nu$ is the coefficient of kinematic viscosity of the liquid.

The element for twisting the injected flow is made by 3D printing technology with a tilt angle α = 45° and its thickness of 2.5 mm. The presence of a central sleeve (to ensure the rigidity of the auger) reduced the normal cross-sectional area of the receiving chamber of the jet pump by 8.7 %. In addition, the presence of auger turns, the rotation of the liquid particles and the continuous change of the direction of movement of the injected flow additionally affect the hydraulic resistance of the flow part of the jet pump. The efficiency of twisting the injected flow obtained during the research exceeds the negative impact on the jet pump characteristics of the increase of the hydraulic resistance of its receiving chamber caused by the reduction of its cross-sectional area and the change of flow directions.

Dependences of the injection coefficient value on the Reynolds number of the workflow are approximated in the form of asymptotic functions (Fig. 4)

$$i = \frac{Re_w}{b + a\,Re_w}\ . \qquad (12)$$

The values of the empirical coefficients $a$, $b$ for the models of jet pumps used in the test with the angles of inclination of the guide elements $\alpha_s = 0$, $\alpha_s = 45^{\circ}$ are given in Table 2.

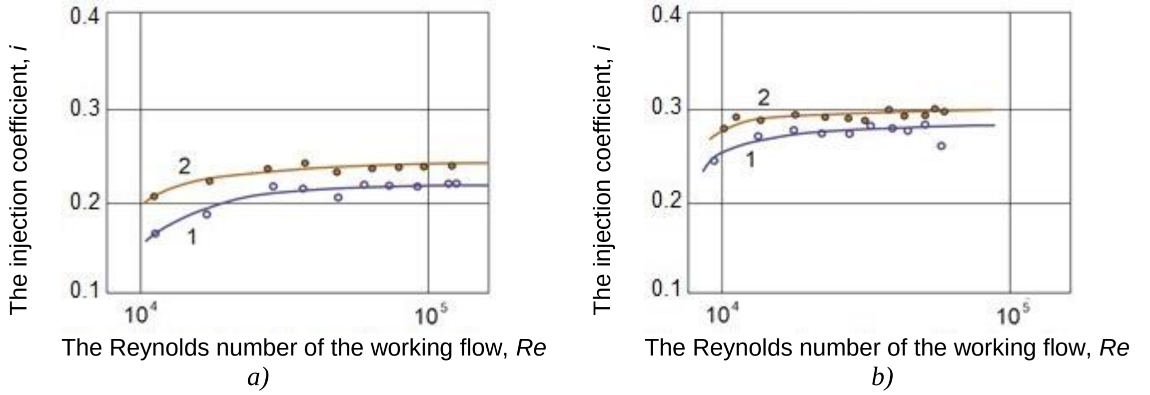
According to the obtained results, the maximum increase in the value of the injection coefficient caused by the twisting of the injected flow is 19.1 %. According to the dependences shown in Fig. 4, the effect of the twist of the injected flow on the value of the injection coefficient increases for small Reynolds numbers, i.e. for highly viscous liquids.

The regression equations for the pressure characteristics of the models of jet pumps (Fig. 5) are given in Table 3, and the values of the experimental coefficients – in Table 4.

Increasing the value of the injection coefficient reduces (Fig. 5) the discrepancy between the pressure characteristics of the direct-flow and vortex jet pump.

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

| Model | Diameter of a nozzle $d_w$, mm | Basic geometric parameter $K_p$ | Distance to the mixing chamber | | The relative length of the mixing chamber $\overline{l}_c$ |
|---|---|---|---|---|---|
| | | | Absolute $l_w$, mm | Relative $\overline{l}_w$ | |
| 1 | 7.7 | 3.795 | 7 | 1.10 | 15.6 |
| 2 | 8.1 | 3.429 | 10 | 0.81 | 14.8 |

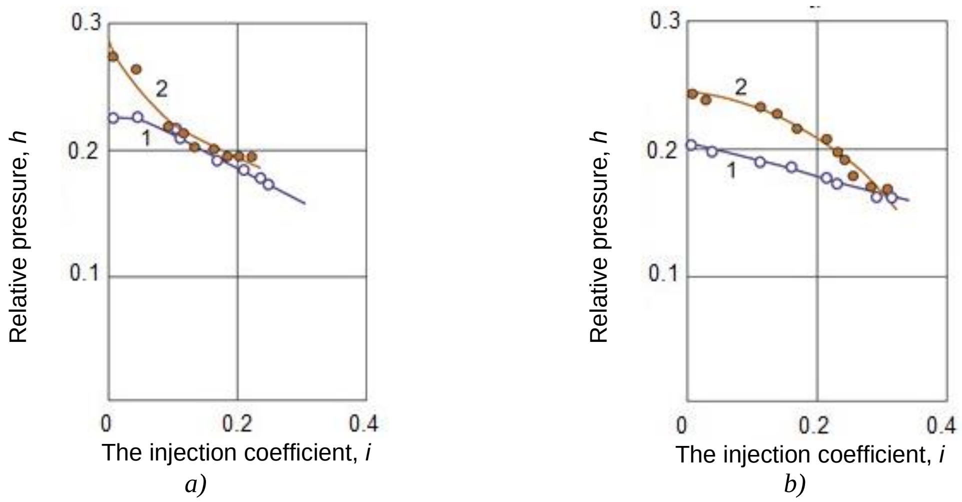



*a)* $K_p = 3.429$; *b)* $K_p = 3.795$

**Figure 4 – Dependence of the injection coefficient on the Reynolds number of the working flow of the straight-flow 1 and vortex jet pump 2 for various ratios of the geometric parameter $K_p$ of the jet pump**

**Table 2 – Coefficients values of empirical functions and correlation coefficients for direct-flow and vortex jet pumps**

| Model | Coefficients values | | | |
|---|---|---|---|---|
| | *a* | | *b* | |
| | $\alpha_s = 0$ | $\alpha_s = 45^{\circ}$ | $\alpha_s = 0$ | $\alpha_s = 45^{\circ}$ |
| 1 | 3.568 | 3.394 | 3878 | 2097 |
| 2 | 4.327 | 4.08 | 25934 | 12353 |




*a)* $K_p = 3.429$; *b)* $K_p = 3.795$

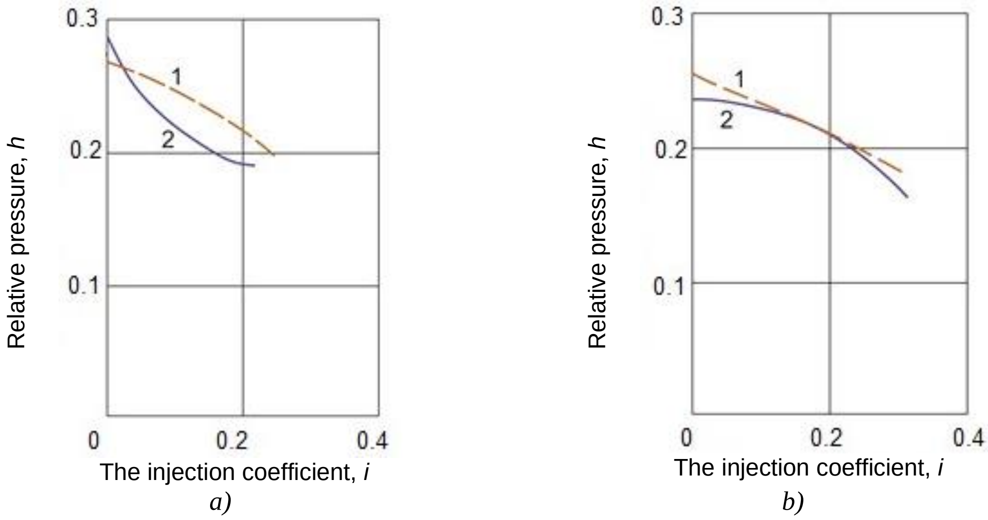
**Figure 5 – The pressure characteristic of a straight-flow 1 and vortex jet pump 2 for different ratios of the geometric parameter $K_p$ of a jet pump**

**Table 3 – Regression equation for pressure characteristics of jet pumps**

| Model | Straight flow | Swirling flow |
|---|---|---|
| 1 | $h = a + bi^3 + ce^i$ | $h = a + bi^2$ |
| 2 | $h = a + bi^2 lni$ | $\ln h = a + bi + ce^{-i}$ |

**Table 4 – The values of empirical coefficients of regression equations for the pressure characteristics of jet pumps**

| Model | Straight flow | | | Swirling flow | | |
|---|---|---|---|---|---|---|
| | *a* | *b* | *c* | *a* | *b* | *c* |
| 1 | 0.3108 | -0.00304 | -0.1086 | 0.2388 | 0.8502 | - |
| 2 | 0.2252 | 0.62970 | - | -14.5140 | 10.1045 | 13.2394 |




*a)* $K_p$ *= 3.429; b)* $K_p$ *= 3.795*

**Figure 6 – Theoretical (1) and experimental (2) pressure characteristics of the jet pump for different values of the geometric parameter $K_p$**

Analysis of the obtained experimental pressure characteristics (Fig. 5) indicates that the maximum increase in pressure caused by swirling of the injected flow is 9.2 % for a pump with a geometric parameter value $K_p$ = 3.429 and 16.9 % for a pump with a geometric parameter value $K_p$ = 3.795.

Dependences of the efficiency on the injection efficiency of the jet pump are obtained using the empirical pressure characteristics given in Table 4. The maximum increase in efficiency caused by the swirling of the injected flow corresponds to the model of a jet pump with a geometric parameter $K_p$ = 3.795 and equals 21.3 %.

The verification of the adequacy of the theoretical pressure characteristics was carried out for both models of a high-pressure jet pump under study. In the process of theoretical determination of the pressure, equation (9) was used for the case of injected flow swirling by guiding elements with an angle of inclination $\alpha_s$ =45 $^o$ (Fig. 6).

Comparative analysis of theoretical and experimental pressure characteristics of the jet pump for various values of the geometric parameter $K_p$ (Fig. 6) has shown that the average error in the theoretical determination of the pressure is $\delta h$ = 9.5 % for the pump with the geometric parameter value $K_p$ = 3.429 and $\delta h$ = 4.9 % for the pump with the geometric parameter value $K_p$ =3.795.

**Conclusions**

Based on the use of the law of conservation of energy, momentum in fluid in a closed volume of the mixing chamber and flow continuity, an equation for the pressure characteristic of a downhole high-pressure vortex jet pump is obtained for conditions of symmetric swirling of the injected flow. The structure of the obtained equation contains a component that determines the value of the additional dynamic pressure created by swirling the mixed flows in the flow path of the jet pump. In the case of a zero angle of symmetric swirling of the injected flow, an analytical dependence of the jet-vortex ejection system is proposed, which takes the form of the classical equation of the pressure characteristic of a straight-flow jet pump.

Based on experimental studies of a downhole high-pressure vortex jet pump for conditions of symmetric swirling of the injected flow, an increase in the value of the injection coefficient was obtained up to 19.1 %, pressure – up to 16.9 %, efficiency – up to 21.3 %.

In the process of checking the adequacy of the proposed pressure characteristics of the downhole high-pressure vortex jet pump, it is found that the maximum average error of the theoretical determination of the pressure does not exceed 9.5 %.

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## Дослідження гідравлічних характеристик свердловинного струминного насоса

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В роботі розглянута можливість підвищення енергетичної ефективності свердловинних ежекційних систем шляхом закручування інжектованого потоку розміщеними в проточній частині струминного насоса похилими направляючими елементами. На основі законів збереження енергії, імпульса та суцільності потоку розроблена математична модель робочого процесу вихрової ежекційної системи у вигляді прямоточного робочого та гвинтового інжектованого струменів, яка дозволила отримати рівняння напірної та енергетичної характеристик високонапірного струминного насоса. У випадку нульового кута закручування інжектованого потоку отримані залежності набувають вигляду відомих рівнянь, які характеризують робочий процес прямоточного струминного насоса. В процесі експериментальних досліджень гідравлічних характеристик високонапірного струминного насоса для випадку закручування інжектованого потоку отримано зростання коефіцієнта інжекції, відносного напору та коефіцієнта корисної дії струминного насоса відповідно до 19.1, 16.9 та 21.3 %. Отримана при експериментальній перевірці напірної характеристики вихрового струминного насоса похибка не перевищує 9.5 %.

Ключові слова: *відносний напір, ежекційна система, коефіцієнт інжекції, закручування потоку, енергетична ефективність.*