U.P.B. Sci. Bull., Series D, Vol. 87, Iss. 4, 2025 ISSN 1454-2358

INVESTIGATION OF THE EFFECT OF FLOW SWIRL ON THE PERFORMANCE CHARACTERISTICS OF A DOWNHOLE JET PUMP

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A mathematical model has been developed to describe the operation of a downhole ejector system under conditions of motive flow swirl. Mechanism of localized circulation of coaxial interacting streams has been proposed and theoretically justified. Experimental investigations have demonstrated that inducing swirl in the primary flow results in an increase in the relative pressure head and overall efficiency of the jet pump by 8.62% and 9.89%, respectively. Validation of the proposed mathematical model for the vortex-type jet pump revealed that the average deviation in the theoretical prediction of the relative pressure head under swirling flow conditions does not exceed 7.96%.

Keywords: Jet pump; ejector system; ejection ratio; relative pressure head; flow swirl.

1. Introduction

Jet devices are widely used in various industries including metallurgy, agriculture, chemical processing, and mining, due to their simple design, lack of moving parts, resistance to aggressive media, and adaptability to hard-to-reach locations. The utilization of the Venturi effect to intensify the mixing of multiphase components enables the generation of ultra-fine dispersions of powder-like materials in pharmaceutical production processes [1]. The integration of jet pumps as core components in refrigeration systems [2], serving as an alternative to conventional mechanical compressors, results in reduced energy consumption and enhanced overall system efficiency. Among the traditional areas of ejector technology development, the oil and gas industry remain particularly prominent. Jet devices have demonstrated efficacy in improving hydrocarbon recovery under complex operational conditions [3], and maintaining the intrinsic permeability of formation horizons during the initial penetration of hydrocarbon-bearing strata [4]. Furthermore, modern ejector system configurations facilitate

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the metered injection of corrosion inhibitors into surface pipelines within well product gathering systems [5].

One of the main challenges in applying ejector technology in oil and gas production is the inherently low energy efficiency of ejector systems. The enhancement of downhole jet pump designs and the optimization of their operational processes are, therefore, critical areas of research aimed at improving the energy efficiency of ejector-based production technologies. Improving the energy efficiency of downhole ejector systems is typically achieved by optimizing key geometric parameters, such as the area ratio between the mixing chamber and the motive nozzle [6], the entry angles of the interacting streams into the suction chamber, and the diffuser expansion angle [7], the spacing between the nozzle and the mixing chamber and the length of the chamber itself [8], as well as the radial offset of the motive nozzle [9]. Furthermore, it has been shown [10] that the efficiency of a jet pump may be improved by up to 30% through appropriate selection of its installation depth within the wellbore. In multijet pumps [11] and annular-type jet pumps [12], the reduction of vortex intensity and minimization of hydraulic losses during flow mixing are achieved by increasing the interfacial area between the interacting flows, which facilitates the equalization of velocity profiles. The use of multiple motive nozzles further contributes to the optimization of velocity distribution within the flow passage of the ejector system [13], thereby reducing the required length of the mixing chamber and the material intensity of the downhole jet pump design.

Recent advancements in ejector system design have led to the development of jet-vortex pumps that incorporate flow-guiding elements to induce swirl within the mixing section [14]. These inertial forces serve as an auxiliary mechanism contributing to the pressure drop within the flow path of the device [15]. In [16], it was demonstrated that the ratio between the surface area of the primary jet and that of the mixing chamber has a direct effect on the velocity distribution of the interacting flows.

The potential for improving the energy efficiency of downhole ejector systems through refinement of geometric and operational parameter selection methods for jet pumps has largely been exhausted. Further enhancement of the operational and energetic performance of ejector systems can be achieved by advancing the interaction mechanism between the mixed flows. Multijet and annular-type jet pumps, due to the small cross-sectional dimensions of the motive flow channels, exhibit increased sensitivity to the presence of mechanical impurities in the working medium – a factor that becomes increasingly critical during the late stages of hydrocarbon field development. The application of pulsating ejector systems employing intermittent injection of the motive fluid necessitates the incorporation of moving components, thereby negating one of the principal advantages of jet pump design – its mechanical simplicity and reliability.

A promising approach to improving the energy exchange mechanism between the interacting flows lies in the implementation of inclined guide vanes for inducing swirl within the flow path of a vortex-assisted jet pump, followed by numerical and experimental analysis of its operational behavior.

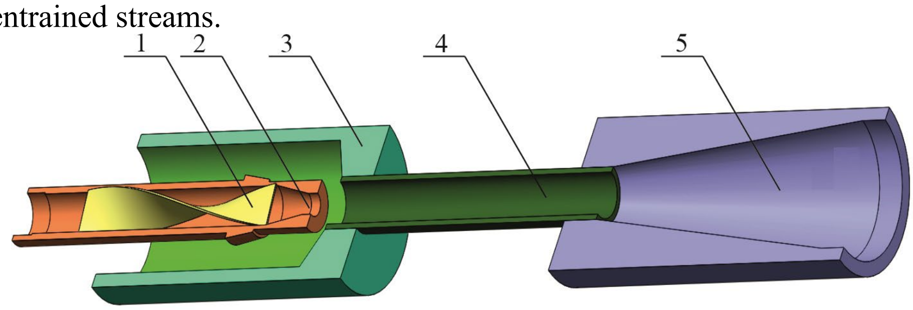
This study aims to develop and experimentally validate a mathematical model for a downhole ejector system that accounts for the swirling motion of the motive fluid.

2. Research Methodology

The investigation of the influence of flow swirl on the performance characteristics of a downhole jet pump was conducted in three sequential stages:

1. Development of the mathematical model and derivation of the pressure-head characteristic equation of the downhole jet pump under conditions of motive flow swirl, followed by analytical assessment of the resulting expressions.
2. Experimental investigation of the jet pump's operational behavior under flow swirling conditions within the working fluid.
3. Experimental validation of the model, aimed at assessing the adequacy of the derived pressure-head characteristic equation for the downhole jet pump operating with a swirling motive flow.

To induce localized swirling within the interacting flows, helical guide vanes were installed in the flow path of the jet pump (Fig.1). These vanes generated controlled swirl in the mixing chamber, thereby modifying the structure of the velocity field and enhancing the momentum exchange between the motive and entrained streams.



1 – guide vane; 2 – motive nozzle; 3 – suction chamber; 4 – mixing chamber; 5 - diffuser
Fig. 1. Schematic diagram of the installation of helical guide vanes in the flow section of the jet pump

The motive and entrained streams converge in the suction chamber of the jet pump. Within the mixing chamber, velocity profiles of the interacting flows are equalized, while in the diffuser, pressure recovery occurs. The presence of guide vanes within the motive nozzle increases the radius of the swirling primary jet in the radial direction relative to the pump axis. This radial displacement of the motive jet expands the interfacial area between the active and passive media, thereby enhancing the conditions for kinetic energy transfer from the motive to the entrained stream, ultimately resulting in an increase in the overall efficiency of the jet pump.

The angle of inclination of the screw guide elements is the main design parameter of the mechanism for twisting the flows in the flow part of the jet pump. An increase in the angle of inclination of the guide elements intensifies the rotation of their coaxial jets, while, however, the magnitude of the hydraulic losses caused by the rotation and deformation of the mixed flows increases. The presence of two opposing processes determines the existence of an optimal angle of inclination of the guide elements.

By design, devices for twisting flows in the flow part of a jet pump can be divided into two groups:

– placement of screw guide elements of the belt or screw type in the flow;

– flow direction using nozzles offset relative to the axis of the transit flow or placed at an angle.

An example of a device for twisting the working medium of the second group is the use of tangential supply of the working or injected flow.

Unlike devices of the first group, axial or angular displacement of the nozzles does not create additional pressure losses caused by a decrease in the normal cross-section of the hydraulic channel by screw guide elements placed in the flow. The mechanism of twisting flows with offset nozzles is thus more energy-efficient, but taking into account the design features, it requires the use of jet pumps with larger diametrical dimensions for its implementation. This circumstance must be taken into account when choosing the design of a jet-vortex pump when implementing specific production processes. In particular, in the design of downhole ejection systems, which are currently used for drilling, operating and repairing oil and gas wells, due to the restrictions imposed by the compressed conditions of use of jet pumps, the use of guide elements of the first group can be recommended for twisting mixed flows. If there are no restrictions on the maximum diameter of the jet pump, in the process of choosing a mechanism for twisting mixed flows, preference should be given to the use of offset nozzles.

The design of the jet pump: with a central or peripheral supply of the active flow obviously does not affect the choice of the type of mechanism for twisting mixed flows.

The construction of the hydraulic model describing the operating process of the ejector system is based on the application of the conservation of linear momentum within characteristic cross-sections of the jet pump flow path [17]

$$i = \frac{G_s}{G_w}; \quad (1+i) = \frac{G_w + G_s}{G_w} = \frac{G_m}{G_w} \tag{1}$$

$$\Delta P_m = P_m - P_s; \ \Delta P_w = P_w - P_s; \ h = \frac{\Delta P_m}{\Delta P_w} \tag{2}$$

$$K_p = \frac{f_3}{f_w}; \quad K_p - 1 = \frac{f_3 - f_w}{f_w} = \frac{f_s}{f_w} \tag{3}$$

where i, h, K_p are the entrainment ratio, the dimensionless pressure head, and the primary geometric parameter of the jet pump; G_w, G_s, G_m denote the mass flow rates of the motive, entrained, and mixed streams; $\Delta P_m, \Delta P_w$ are the pressure differential generated by the jet pump and the pressure drop of the motive flow; P_w, P_s, P_m denote the static pressures of the motive, entrained, and mixed flows; f_w, f_s, f_3 are the cross-sectional areas of the motive, entrained, and mixed streams.

The experimental study of the operational behavior of the downhole jet pump was conducted using a laboratory-scale test rig (Fig. 2).

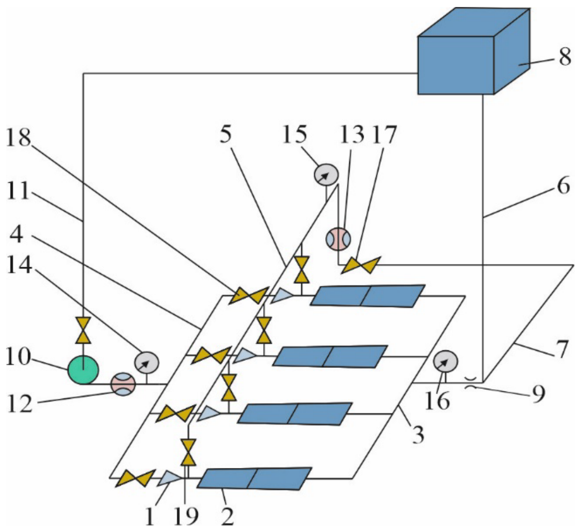


Fig. 2. Schematic diagram of the laboratory test rig

The setup comprises a model of the jet pump, consisting of a motive nozzle 1 and a mixing chamber with a diffuser 2, along with a discharge manifold

3, motive fluid manifold 4, and suction manifold 5. The system also includes discharge 6 and suction 7 pipelines, a combined suction–discharge reservoir 8, and a local flow restriction 9 designed to simulate the hydraulic resistance typically created by the bit's flushing system. A centrifugal pump 10 with a suction inlet 11 provides motive fluid circulation. Flow rates are measured using flow meters 12, 13, while pressure monitoring is carried out via manometers 14 – 16. Valves 17 and 18 regulate the flow rates of the motive and entrained streams, respectively. During the experimental campaign, measurements were taken for the volumetric flow rates of both the motive and entrained fluids, as well as for the static pressures of the motive, entrained, and mixed flows. The measured flow rates were expressed in the form of a dimensionless parameter – the entrainment ratio i (equation (1)), while the pressure data were utilized to compute the dimensionless pressure head h (equation (2)) of the jet pump. The obtained flow and pressure relationships enabled the derivation of an empirical dependence of the entrainment ratio on the Reynolds number of the motive flow, as well as the construction of the pressure-head characteristic curve of the jet pump.

The Reynolds number of the motive flow was determined using the following relation [14]

$$R_{ew} = \frac{V_w d_w}{\nu} \tag{4}$$

where V_w is the flow velocity at the outlet cross-section of the motive nozzle; d_w denote the diameter of the motive nozzle; ν is the kinematic viscosity of the working fluid.

To convert the pressure-head characteristic of the jet pump $h = f(i)$ into its energy characteristic $\eta = f(i)$, the following relation was employed [17]

$$\eta = \frac{hi}{1-h} \tag{5}$$

where η is the efficiency of the jet pump.

For the experimental investigation, a laboratory model of a jet pump was employed, for which the primary geometric parameter was set to K_p=6,464. The flow swirling was induced by a guide element fabricated in the form of a twisted plate installed within the hydraulic channel of the motive stream. During the experimental study, the average swirl angles of the motive flow were set to α=8°, α=15° и α=20° respectively.

3. Results and Discussion

3.1. Modeling Results

The equation of linear momentum conservation for the mixing chamber of the jet pump (Fig. 3), taking into account the swirling motion of the motive flow, is expressed as follows [17]

$$\varphi_2(G_w V_{w1} + G_s V_{s2}) - (G_w + G_s)V_m = (P_{m3} - P_{s2})f_{s2} + (P_{m3} - P'_{w_1})f_{w_1} = \\ = P_{m3}f_3 - P_{s2}f_{s2} - P'_{w_1}f_{w_1} \tag{6}$$

where φ_2 are the velocity coefficient at the inlet cross-section of the mixing chamber; V_{w1}, V_{s2}, V_m are the velocities of the motive flow at the nozzle exit, the entrained flow at the mixing chamber inlet, and the mixed flow at the outlet of the mixing chamber; P'_{w1}, P_{s2}, P_{m3} denote the pressure of the swirling motive flow, pressure of the entrained flow at the inlet of the mixing chamber, and pressure of the mixed flow at the outlet of the mixing chamber.

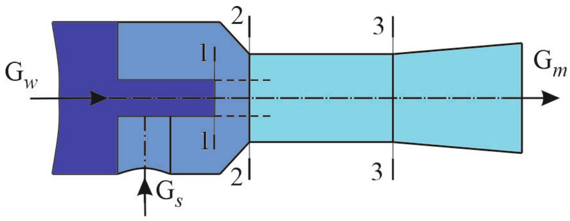


Fig. 3. Flow section of the jet pump

The force generated by the swirling motive flow at the inlet cross-section of the mixing chamber is determined by integrating the elemental forces acting on an elementary annular section, the area of which is given by $2\pi r dr$ [17]

$$P'_{w1}f_{w_1} = \left(P_{w1} + \rho\frac{\omega^2 r^2}{2}\right)f_{w_1} = \int_0^{r_w}\left(P_{w1} + \rho\frac{\omega^2 r^2}{2}\right)2\pi r dr = P_{w1}\pi r_w^2 + \rho\omega^2\frac{\pi}{4}r_w^4 = \\ = P_{w1}f_{w_1} + \rho\omega^2\frac{\pi}{4}r_w^4 \tag{7}$$

where P_{w1} is the pressure of the motive flow at the inlet cross-section of the mixing chamber under non-swirling conditions; ρ denote the density of the flow;

ω is the angular velocity of fluid particle rotation; r_w is the radius of the motive nozzle.

After integrating equation (7), determining the velocities and pressures at the characteristic cross-sections of the jet pump using the continuity equation and the energy conservation law in the form of the Bernoulli equation, and substituting the obtained results into equation (6), the expression for calculating the pressure-head characteristic of the jet pump under swirling flow conditions is obtained

$$h = \frac{\varphi_1^2}{K_p}\left[2\varphi_2 + \left(2\varphi_2 - \frac{1}{\varphi_4^2}\right)\frac{i^2}{K_p - 1} - (2-\varphi_3)^2\frac{(1+i)^2}{K_p} + \right] + \frac{\varphi_1^2}{K_p} tg^2\alpha \qquad (8)$$

where $\varphi_1, \varphi_3, \varphi_4$ are the velocity coefficients for the motive nozzle, the diffuser, and the inlet section of the mixing chamber; α is the inclination angle of the guide vanes.

The last term in Equation (8) represents the magnitude of the additional pressure head induced by the swirling motion of the motive flow. Based on Equation (8), a directly proportional relationship (Fig. 4) between the pressure head and efficiency of the jet pump and the inclination angle of the swirl-inducing guide vanes was obtained.

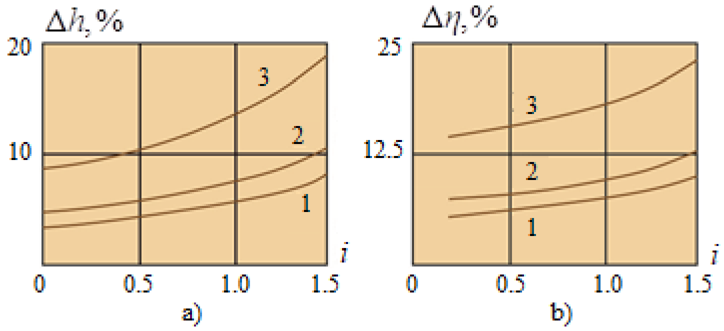


Fig. 4. Influence of the guide vane inclination angle on the enhancement of pressure head (a) and hydraulic efficiency (b) of the jet pump: 1 – α=20°; 2 – α=25°; 3 – α=30°

The value of the primary geometric parameter K_p of the downhole jet pump has an inverse effect on the pressure-head and energy performance characteristics of the ejector system.

3.2. Experimental Results

The experimental dependencies of the entrainment ratio on the Reynolds number of the motive flow, obtained for the investigated swirl angles, exhibit a nonlinear increasing trend (Fig. 5).

Considering that the pressure head in a vortex-type jet pump is generated primarily due to the combined action of viscous and centrifugal forces, it may be hypothesized that with increasing Reynolds number, the influence of flow swirl diminishes, and the entrainment ratio approaches an asymptotic constant value. Given the physical nature of the pressure generation mechanism, it is reasonable to approximate the experimental results using an empirical function with a horizontal asymptote, such as a hyperbolic relationship of the form $i = R_{ew}/\left(b_0 + b_1 R_{ew}\right)$.

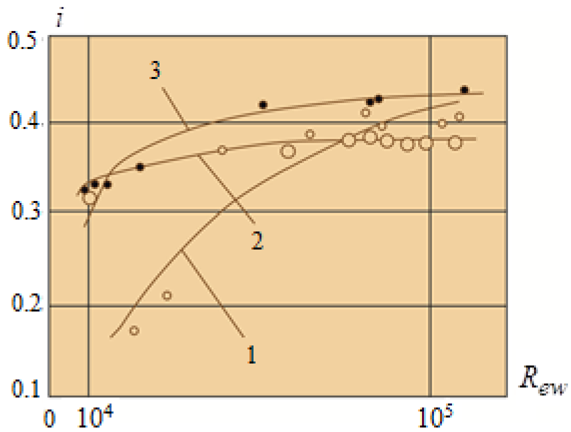


Fig. 5. Effect of motive flow rate on the entrainment ratio at different guide vane inclination angles: 1 – α =0°; 2 – α =8°; 3 – α=15°

The coefficients of the regression equations b_0, b_1 and the correlation coefficients r are presented in table 1.

Table 1

Empirical coefficient values for different guide angles

α, °	b_o	b_1	r
0	7281	1.6931	0.9611
8	3762	2.6094	0.9994
15	10220	2.231	0.9995

The maximum deviation between the dependencies $i = f\left(R_{ew}\right)$, obtained for swirling and non-swirling motive flows is observed at low Reynolds numbers.

The influence of flow swirl diminishes with increasing Reynolds number. An increase in the swirl angle leads to a higher entrainment ratio of the jet pump.

The investigation of the influence of flow swirl on the pressure-head characteristic of the jet pump was carried out in two stages. In the first stage, the pressure and flow rate relationships were determined at characteristic cross-sections of the jet pump using a conventional straight-through motive nozzle. In the second stage, a twisted plate with a guide vane inclination angle of α=8° or α=20° (Fig. 6) was installed in the hydraulic channel supplying the motive flow.

During statistical analysis, performed using appropriate computational tools, empirical dependencies were derived in the form of combined algebraic and exponential functions with fractional exponents.

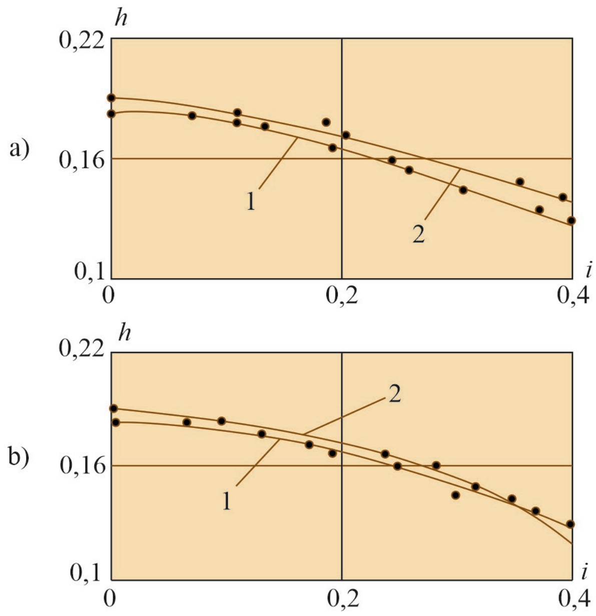


Fig. 6. Pressure-head characteristic of the jet pump for non-swirling (1) and swirling (2) motive flow: a) swirl angle α =8°; b) swirl angle α =20°

The results of the experimental investigations for the straight-through motive nozzle were approximated by the following empirical expression

$$h^{-1} = a + bi^2 \tag{9}$$

where the values of the empirical constants are a=5.4533; b=15.0362. According to the results of the statistical evaluation of the pairwise correlation, the correlation coefficient r for the nonlinear regression equation is r=0.9922.

In the case of flow swirl induced by a twisted plate with a guide vane inclination angle of α=8°, the experimental data were approximated by the following relationship

$$h^{0,5} = a + bi^{1,5} \tag{10}$$

where a=0.4348; b= -0.2438 (with a corresponding correlation coefficient of r=0.9452).

A similar dependency for the case where the motive flow was swirled using a twisted plate with an inclination angle of α=20° is given by

$$h = a + bi + ci^2 + de^i + ee^{-i} \tag{11}$$

The constant coefficients of equation (11) are as follows: a=104.2494; b=-16.1825; c=51.247; d=-43.946; e=-60.1142, and the correlation coefficient is r=0,9852.

Based on the obtained experimental results, the introduction of swirl into the motive flow led to an increase in the dimensionless pressure head of the jet pump:

– by 8.62% when using guide vanes with an inclination angle of α=8°;
– by 3.48% when using guide vanes with an inclination angle of α=20°.

After converting the pressure-head characteristic $h = f(i)$ into the energy efficiency characteristic $\eta = f(i)$ (using equation (5)), the increase in jet pump efficiency was determined as follows:

– by 9.89% for an inclination angle of α=8°;
– by 4.14% for an inclination angle of α=20°.

Considering the obtained results, it is recommended to use guide vanes with an inclination angle of α=8° for inducing swirl in the motive flow in the design of downhole jet pumps. Given the prolonged operational periods typical for oil and gas field exploitation, even a modest increase in jet pump efficiency can lead to a substantial reduction in energy consumption and the production cost per well.

3.3. Comparative Analysis of Theoretical and Experimental Pressure–Head Characteristics of the Jet Pump

The validity of the developed mathematical model describing the operating process of the jet pump was assessed by comparing its theoretical pressure-head characteristic (calculated using equation (8)) with the experimentally obtained pressure-head values (Fig. 7). The study revealed that the average deviation in theoretical predictions of the jet pump's pressure head was δh=7.96% for a swirl angle of α=8°, and δh=6.38% for α= 20° (Fig. 8). The notable deviation in the theoretical pressure-head predictions is attributed to the limitations of the baseline methodology used in the development of the mathematical model describing the jet pump operation under swirling flow conditions.

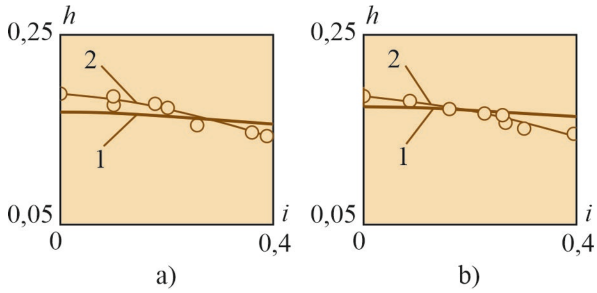


Fig. 7. Comparison of the theoretical (1) and experimental (2) pressure-head characteristics of the jet pump: a) swirl angle of the motive flow α=8°; b) swirl angle of the motive flow α=20°

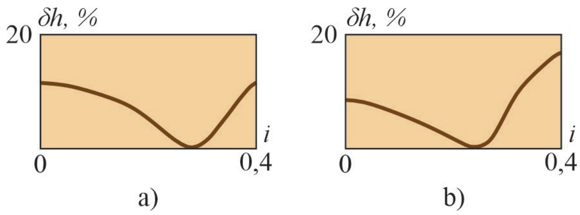


Fig. 8. Absolute values of errors in theoretical determination of jet pump head: a) angle of twist of the working flow α=8°; b) angle of twist of the working flow α=20°

3.4. Analysis of the interaction of hydromechanical characteristics of ejection system elements under conditions of swirling mixed flows

In addition to the impact on the hydraulic characteristics of the ejection system, the use of screw guide elements can intensify the process of hydroabrasive wear of the internal surfaces of the jet pump. The twisting of the working medium causes the appearance of a circular component of the velocities of the mixed flows, as a result of which the sliding of mechanical particles along the internal limiting surfaces of the flow part of the jet pump changes to impact interaction, which is accompanied by an intensification of the process of hydroabrasive wear of the elements of the ejection system. The use of screw guide elements in the design of the jet pump, thus, can have a negative effect in the case of its long-term use.

The intensification of the process of hydroabrasive wear of the internal surfaces of the jet pump necessitates the use of filters for cleaning the working medium from mechanical impurities in the designs of ejection systems in which swirling of flows is implemented. In the process of manufacturing jet-vortex pumps, it may be recommended to use materials resistant to hydroabrasive wear. Swirling of flows reduces the path of equalization of thermal characteristics when mixing liquids of different temperatures. This causes an increase in the temperature of the internal surface of the receiving chamber compared to direct-flow jet devices. In the case of a high temperature of the receiving chamber, the probability of its thermal deformation and increased wear of the internal surface increases. Given the increased probability of damage to the parts of the jet pump in the case of swirling of high-temperature flows, it is necessary to take a larger wall thickness of the receiving chamber, and for its manufacture it is advisable to use materials that are able to maintain operability under the influence of the temperature factor.

The swirling of the mixed flows changes the nature of the pressure distribution in the flow part of the jet pump. The minimum pressure value is set on the axis of the jet pump, and the maximum – in the wall areas of the jet pump housing. The magnitude of the pressure change is directly proportional to the rotation speed, radius and density of the mixed flows. When the flows rotate, the pressure on the inner surface of the working nozzle decreases, and on the inner surfaces of the receiving chamber, mixing chamber and diffuser – increases. In accordance with the change in pressure, the tensile stresses in the material of the working nozzle decrease and the tensile stresses in the walls of the receiving chamber, mixing chamber and diffuser increase. Due to the action of viscous forces, the speed of rotation of the working medium slows down and the uneven pressure distribution caused by the swirling of the mixed flows decreases. At a certain distance from the guide elements for twisting the working medium along the flow, the nature of the pressure and stress distribution acquires values that are characteristic of direct-flow jet pumps. The length of the section of the influence of the flow rotation on the nature of the pressure and stress distribution is determined by the hydrodynamic parameters and physical properties of the working medium. Taking into account the peculiarities of the pressure and flow distribution, the receiving chamber of the jet-vortex pump must have a greater margin of safety than in the case of direct-flow jet pumps.

The influence of the angle of rotation of the mixed flows on the mechanical characteristics of the jet pump parts will be assessed using the concept of the inertial pressure component due to the rotation of the working medium.

The inertial pressure component is determined by the well-known formula [17]

$$P_i = \frac{\rho \omega^2 r_w^2}{2} \tag{12}$$

Using elementary relations between the axial and rotational components of the helical flow velocity, the dependence for determining the angular velocity of the flows can be represented as [17]

$$\omega = \frac{Q}{\pi r_w^3} tg\alpha \qquad (13)$$

where Q is the volumetric flow rate of the working stream.

Then, after substituting relation (13) into formula (12), we obtain

$$P_i = \frac{\rho Q^2}{2\pi^2 r_w^4} tg^2\alpha \qquad (14)$$

The change in the inertial component of pressure (Fig. 9) reflects the influence of the angle of inclination of the guide elements on the nature of the stress distribution in the parts of the jet pump.

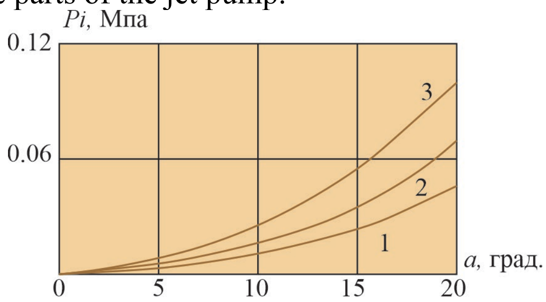


Fig. 9. Dependence of the inertial pressure component caused by the swirling of mixed flows on the angle of inclination of the guide elements and the flow rate of the working flow Q:
1 – Q =4×10^{-3} м^3/с; 2 – Q= 5×10^{-3} м^3/с; 3 – Q =6×10^{-3} м^3/с

Given the nature of the change in the inertial component of pressure, an increase in the angle of inclination of the guide elements for swirling mixed flows contributes to an increase in stresses in the body elements of the jet pump.

The level of turbulent pulsations can have an additional effect on the strength of the jet pump elements. Due to the significant difference in the speeds of the mixed flows between the working and injected flow, a shear layer of the liquid appears, which is characterized by the presence of vortex zones and pulsations of speeds and pressures. Pressure pulsations cause dynamic loads on the internal surfaces of the jet pump parts. Given the stochastic nature of the process, the place of formation of vortex zones is random. The level of turbulence increases under conditions of swirling mixed flows. This circumstance can cause additional dynamic stresses in the parts of jet-vortex pumps compared to traditional designs of direct-flow ejection systems. It should be noted that the shear layer and vortex zones are localized in the first half of the mixing chamber. For this section of the mixing chamber, the impact of dynamic loads will be maximum.

4. Conclusions

1. One of the promising approaches to enhancing the efficiency of downhole ejector systems is the optimization of the operating cycle of the jet device by inducing swirl in the interacting streams within its flow section.

2. The structure of the pressure-head characteristic equation of the jet device has been improved by incorporating the momentum balance of coaxial jets exhibiting a parabolic pressure distribution, which enables the effect of localized swirling of the motive stream to be taken into account.

3. It has been experimentally confirmed that inducing swirl in the motive flow enhances the entrainment ratio, the dimensionless pressure head, and the efficiency of the jet pump:

– the maximum increase in the entrainment ratio is observed at low Reynolds numbers, particularly under conditions associated with the production of high-viscosity crude oils;

– the maximum increase in the dimensionless pressure head of the jet pump reaches 8.62%;

– the maximum increase in the efficiency of the jet pump reaches 9.89%.

4. When employing the mathematical model of the jet device's operation proposed in this study, the maximum average deviation in the theoretical prediction of the dimensionless pressure head under swirling flow conditions was found to be 7.96%.

5. The interaction of the hydromechanical characteristics of the elements of the ejection system under the conditions of swirling of mixed flows has been established:

– the presence of a rotating component of the velocity of the mixed flows intensifies the process of hydro abrasive wear of the elements of the ejection system;

– in the case of swirling of high-temperature flows, the maximum temperature of the flow part of the jet pump corresponds to the initial section of the mixing chamber;

– swirling of flows causes a decrease in stresses in the working nozzle and an increase in stresses in the walls of the receiving chamber, mixing chamber and jet pump diffuser;

– due to an increase in the level of turbulence in the mixing area of swirled flows, the dynamic load on the initial section of the jet pump mixing chamber increases.

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