UDC 622.24 + 621.694.2

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# STUDY OF THE STRESS STATE OF THE DOWNHOLE JET PUMP HOUSING

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# ДОСЛІДЖЕННЯ НАПРУЖЕНОГО СТАНУ КОРПУСА СВЕРДЛОВИННОГО СТРУМИННОГО НАСОСА

**Purpose.** The purpose of the research is determination of boundary parameters of a downhole ejection system operation, analysis of critical stressed state of an ejector pump body and substantiation of the method of choosing its geometrical and strength properties.

**Methodology.** On the basis of using classical methods of hydromechanics, the Bernoulli equation, the Darcy-Weisbach equation and continuity of flow it has been established that the most difficult conditions of ejector pump body operation appear during its exploitation in a cavitation mode. Combined analysis of hydrodynamic cavitation occurrence condition and strength condition enabled us to evaluate the boundary stressed state of the ejector pump body.

**Findings.** In the process of modeling the ejector pump operation there is shown necessity to determine hydrodynamic parameters of flow which are suitable for extreme conditions of ejection system operation in a well. The obtained value of cavitation flow rate determines the waste of pressure in hydraulic canal of the annular space of a well and the maximum value of external pressure which influences the jet pump body. The succession of determination of stresses, which occur in the ejector pump body, and of minimum required, under strength conditions, thickness of its side has been shown.

**Originality.** Relationship between hydrodynamic and strength characteristics of individual elements in a downhole ejection system has been determined. The principles of evaluating the stressed state of ejection system elements form theoretical basis for choosing geometrical parameters of the ejector pump body.

**Practical value.** Reasoning the method for choosing the design parameters of a downhole ejector pump increases reliability of its usage and decreases probability of accidents, connected with failure of a bottom hole assembly. The suggested principle of combined analysis of hydrodynamic and strength characteristics can also be used for modeling the operating process of surface ejector pumps, which are the part of the system of extraction, gathering and preparation of oil and gas.

**Keywords:** *oil well, ejection system, jet pump, cavitation, stressed state, operating parameters*

**Introduction.** No moving parts, easy transfer and conversion of energy, the ability to operate a wide range of debit changes, small dimensions and low cost have resulted in the use of jet pumps in many oil and gas fields of Australia, England, Venezuela, Indonesia, Canada, China, Mexico, Russia, the United States of America. Today almost all the main processes of oil and gas field exploitation can be implemented using such ejection systems as drilling, emergency response, case cementing, development and study of wells, their exploitation, intensification of oil extraction methods, underground repair of wells, gathering and preparation of oil and gas. A significant advantage of downhole jet pumps is the possibility of their application under complex mining and technical conditions. In particular, the application of ejecting technologies improves the efficiency of drilling in abnormally low reservoir pressures, the drilling of horizontal sections of wells, emergency response related to the removing of underground equipment from the bottom part, elimination of stuck drill string, operating low-yield wells at final stages of oil and gas field development. The advantages of ejecting technologies are significant while developing oil and gas fields in remote inaccessible areas [1], and for extending the life of aging wells [2] whose products contain a significant amount of dissolved petroleum gas and solids if necessary.

Prevalence and wide application of ejecting technologies demonstrate their world importance. Growth in the use of downhole jet pumps requires further development of theoretical issues aimed at modeling ejection system workflow in the well. Improvement of mathematical models of downhole jet pump workflow allows us to increase the efficiency of oil and gas field development, and it is an urgent task.

**Analysis of the recent research.** Modern mathematical models of jet pump workflow are based on the study of hydraulic connections between the elements of the ejection system and determine their geometric dimensions, mutual orientation [3] and the nature of inclusion [4] in the bottomhole circuit of a well. The workflow of the ejection system is usually modeled as potential adjacent streams with different kinematic and hydrodynamic parameters, the values of which are aligned in the flow part of the jet unit. Besides, there are regulated a number of expected parameters, for example, the ratio of the working and injected flow pressure [5].The principle of minimizing the hydrodynamic parameters of coaxial flows is included in the choice of a borehole layout diametric dimension [6]. While optimizing the jet pump workflow in the work there is used [7] a criterion parameter as a performance index, which provides a necessary combination of kinematic and dynamic characteristics of streams in the bottomhole zone.

Thus, the international experience in modeling ejection systems workflow is generally limited to the analysis of hydraulic phenomena in a flowing part of the jet pump, systematization of operating factors and choosing optimization criteria. Thus mechanical characteristics of deep equipment elements are not examined in the vast majority of the existing methods for choosing operating parameters of ejection systems. The work [8] presents the theoretical foundation of the joint review of hydraulic and mechanical parameters of ejection system elements. Based on Barlow`s formula for a thick cylindrical casing there is found a relationship between stresses in a housing material of the above-bit instrument and the flow of the drilling fluid for different structures of a pump circulation system and a flowing part of the jet pump. These studies help us to formulate the valuation principles of the boundary stress state in some elements of an ejection system, and results in its trouble-free operation.

**Unsolved aspects of the problem.** Features of the downhole jet pump workflow define more complex conditions of its operation compared with elements of other downhole equipment. Due to the high rate of the flow leakage from the working nozzle an area of low pressure in the receiving chamber of the jet pump is formed. Depression that occurs in the receiving chamber increases the difference of pressures acting on the body of the jet pump and increases the possibility of accidents related to the destruction of downhole equipment. The need to prevent accidents related to this equipment requires the research on the stress state of the jet pump housing, identification of critical conditions of borehole ejection system operation and the rationale for the choice of geometric and strength parameters of its elements. This problem is solved in the case of external placement of the jet pump in the bottomhole ejection system [8]. However, the majority of bottomhole ejection systems are characterized with an internal placement of jet pump parts. The stress state of jet pump housing parts was not considered for this type of the pump layout and it requires further studying.

**Objectives of the article.** The purpose of the article is to determine boundary parameters of the borehole ejection system operation, analyze the stress state of the jet pump housing, and justify the method for choosing its geometric and strength characteristics.

**Presentation of the main research.** The borehole ejection system (Fig. 1) consists of the working nozzle *3*, mixing chamber with a diffuser *4* and slurry channels *5* connected to the above-bit area, placed in the housing *2*. The working nozzle *3* is combined with a hydraulic channel of drill pipes *1*, and the slurry channel *5* – with a drilling system of the bit *6*. The drilling fluid flows through the drill pipe *1* to the working nozzle *3*, where the area of low pressure is formed due to the high-speed leakage of the fluid. Having passed the drilling system of the bit, part of the flow moves in an upward direction in the annular space formed by the drill string and the borehole wall *7*, and another part of the flow enters the mixing chamber with the jet pump diffuser *4* through the slurry channels *5*, where the velocity profiles of injected and working flows are leveled. The flow rate at the bottomhole exceeds the performance of a surface pumping unit driveline due to the additional circuit of drilling fluid circulation.

The isolation level of the maximum pressure difference acting on the jet pump housing is shown by the dotted line *a-a* in Fig. 1. The pressure values at points *1*, *2* are determined by the hydraulic resistance of hydraulic system elements. The minimum pressure value at point *1* is determined by operating the jet pump in a cavitation mode and meets the values of saturated vapor pressure of the drilling fluid $P_{1min} = P_{sv}$. The pressure value at point 2 is defined by the following formula

$$P_2 = P_h + \Delta P_c ,$$

where $P_h$ is a hydrostatic pressure in the well; $\Delta P_c$ stands for hydraulic losses in the annular channel, which is formed by the drill string and the borehole wall.

Taking into account the fact that the pressure of saturated vapors is much smaller than the hydrodynamic and hydrostatic pressures, it can be assumed that $P_{sv} \approx 0$. In this case the maximum difference in pressure acting on the jet pump housing wall is determined by the ratio

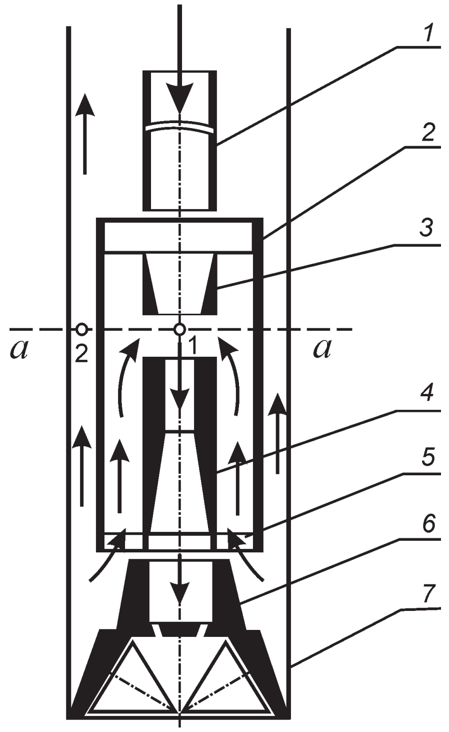


*Fig. 1. Schematic diagram of the borehole ejection system:*
*1 – drill string; 2 – housing; 3 – working nozzle; 4 – mixing chamber with a diffuser; 5 – slurry channels; 6 – bit; 7– borehole wall*

$\Delta P_{max} = P_2$. We can write the equation for determining the boundary value of external pressure acting on the body of the jet pump, considering the basic equation of hydrostatics, continuity equation of flow and Darcy-Weisbach equation

$$\Delta P_{\max} = \rho g H_p + \frac{8}{\pi^2} \frac{\rho \lambda H_p Q_{pcav}^2}{\left(D_b - d_s\right)^3 \left(D_b + d_s\right)^2},$$

where ρ is the density of the washing solution; $g$ is the acceleration of gravity; $H_p$ is the depth of the jet pump placement in the well; $\lambda$ is the coefficient of the linear hydraulic resistance; $Q_{pcav}$ is the workflow rate (performance of a drilling pump) corresponding to the cavitation operation mode of the jet pump; $D_b$, $d_s$ are the borehole diameter and the outer diameter of the drill string respectively.

The magnitude of the workflow rate that corresponds to the occurrence of cavitation in the working nozzle of the jet pump can be determined using the Bernoulli equation for specific sections of the borehole ejection system.

Let us write the Bernoulli equation for the section placed in the drill string directly before the flow of drilling fluid enters the flowing part of a jet pump and the output section of a working nozzle (Fig. 2).

$$Z_1 + \frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} = Z_2 + \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + h_{1-2}, \quad (1)$$

where $Z_1$, $Z_2$ are geometric markings of sections position relative to the plane of comparison; $P_1$, $P_2$ are the values of fluid pressure in the chosen sections; $\alpha_1$, $\alpha_2$ are Coriolis coefficients; $V_1$, $V_2$ are the values of the flow velocity in the chosen sections; $h_{1\text{-}2}$ are fluid pressure flow rate between the sections.

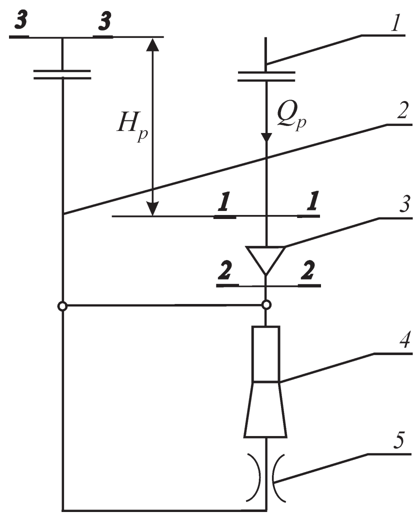


*Fig. 2. Hydraulic circuit of the borehole ejection system:*
*1 – drill string channel; 2 – channel of the annular space; 3 – working nozzle; 4 – mixing chamber with a diffuser; 5 – circulation system of the bit*

Having analyzed the equation (1) for a turbulent fluid motion we can write that

$$Z_1 = Z_2;$$
$$\alpha_1 = \alpha_2 = 1;$$
$$h_{1-2} = h_p = \frac{\Delta P_p}{\rho g},$$

where $h_p$, $\Delta P_p$ are fluid pressure flow rates in the working nozzle of the jet pump.

Having made the reductions we can solve the equation (1) relatively to the pressure $P_2$

$$P_2 = P_1 + \frac{\rho}{2}\left(V_1^2 - V_2^2\right) - \Delta P_p. \quad (2)$$

The value of pressure $P_1$ can be determined using the Bernoulli equation for sections 1–1, 3–3 (Fig. 2)

$$Z_1 + \frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} = Z_3 + \frac{P_3}{\rho g} + \alpha_3 \frac{V_3^2}{2g} + h_{1-3}, \quad (3)$$

where $Z_3$ is a geometric marking of the cross section position 3–3 relative to the plane of comparison; $P_3$, $V_3$ are the flow rate and pressure in the cross section 3–3; $\alpha_3$ is Coriolis coefficient for the cross section 3–3; $h_{1\text{-}3}$ is the pressure loss of the flow between cross sections 1–1 and 3–3.

For analyzing the components of equation (3) it should be considered that the plane of comparison goes through the cross section 1–1, and the cross section 3–3 is connected with the atmosphere (Fig. 2)

$$Z_1 = 0 \,;\; Z_3 = H_p;\; P_3 = 0;\; \alpha_1 = \alpha_3 = 1;$$

$$h_{1-3} = h_p + h_b + h_c = \frac{\Delta P_p + \Delta P_b + \Delta P_c}{\rho g}, \qquad (4)$$

where $h_b$, $h_c$ are pressure losses in the circulation system of the bit and the channel of the annular space, the depth of which equals the depth of the jet pump placement in the well; $\Delta P_b$ stands for pressure losses in the circulation system of the bit.

Using the correlation (4) we can write the equation (3) in the following form

$$P_1 = \rho g H_p + \frac{\rho}{2}\left(V_3^2 - V_1^2\right) + \Delta P + \Delta P_b + \Delta P_c \,. \qquad (5)$$

Having substituted the equation (5) in the formula (2) we obtain

$$P_2 = \rho g H_p + \frac{\rho}{2}\left(V_3^2 - V_2^2\right) + \Delta P_b + \Delta P_c \,. \qquad (6)$$

The components of the equation (6) are defined by the obvious correlations

$$V_2 = \frac{4Q_p}{\pi d_p^2};$$

$$V_3 = \frac{4Q_p}{\pi\left(D_b^2 - d_s^2\right)}; \qquad (7)$$

$$\Delta P_b = \frac{8}{\pi^2}\frac{\rho Q_p^2}{N^2 \mu_b^2 d_b^4};\quad \Delta P_c = \frac{8}{\pi^2}\frac{\rho \lambda H_p Q_p^2}{\left(D_b - d_s\right)^3\left(D_b + d_s\right)^2},$$

where $d_p$, $d_b$ are the diameters of the working nozzles of the jet pump and circulation system of the bit; $\mu_b$ is the coefficient of loss of the bit circulation system nozzles; $N$ is the number of nozzles in the circulation system of the bit.

Taking into account the accepted assumption $P_2 = P_{sv} \approx 0$ and the correlation $Q_p = Q_{pcav}$ after substituting the (7) in equation (6) we can write

$$gH_p + AQ_{pcav}^2 = 0, \qquad (8)$$

where

$$A = \frac{8}{\pi^2}\left(\frac{1}{d_{\partial}^4} - \frac{1}{\left(D_b^2 - d_s^2\right)^2} - \frac{1}{N^2\mu_b^2 d_b^4} - \frac{\lambda H_p}{\left(D_b - d_s\right)^3\left(D_b + d_s\right)^2}\right). \qquad (9)$$

Having solved the equation (8) relative to the value of $Q_{pcav}$ we obtain the formula for determining the workflow loss which corresponds to the emergence of cavitation in the working nozzle of the jet pump

$$Q_{pcav} = \left(\frac{gH}{A}\right)^{0.5}. \qquad (10)$$

The system of equations (9, 10) is solved by the method of successive approximations because the coefficient $\lambda$, in its turn, depends on the workflow value $Q_{pcav}$. The procedure of calculating the coefficient $\lambda$ involves the preliminary calculation of the flow velocity and the Reynolds number based on the standard procedure.

Based on the equations (9, 10) there is built a dependence of the cavitation loss of the workflow on the depth of the jet pump placement in the well and the diameter of its working nozzle (Fig. 3). The analysis of the characteristics presented in Fig. 3 shows the directly proportional dependence of the cavitation flow on the depth of ejection system placement in the well and the inverse dependence of the flow on the diameter of the working nozzle of the jet pump.

The value of the cavitation working flow is used to determine the wall thickness of the jet pump housing that meets the requirements of strength. Using Lame theory it can be determined that the largest absolute values of hoop stresses occur on the inner surface of the jet pump housing. Their absolute value is calculated by the following formula

$$\sigma_\theta = \frac{2R_1^2 \Delta P_{\max}}{R_1^2 - R^2},$$

where $\sigma_\theta$ stands for hoop stresses in the jet pump housing; $R_1$, $R$ are inner and outer diameters of the housing.

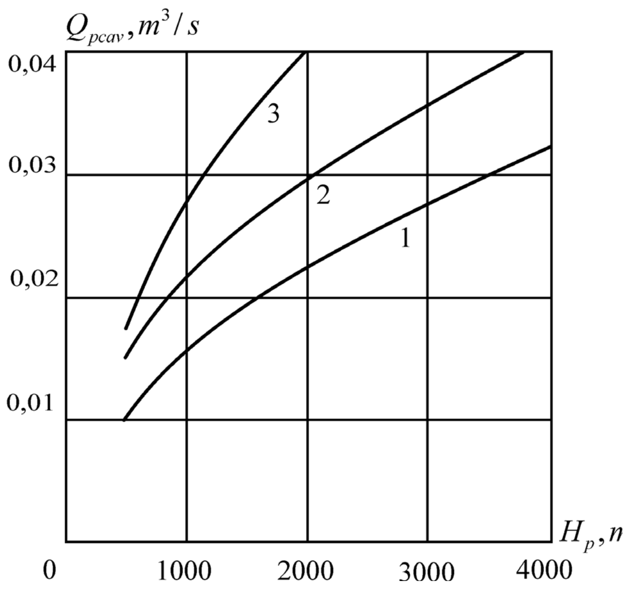


*Fig. 3. The dependence of the cavitation flow on the depth of the jet pump placement and the diameter of its nozzle:*
*1 – $d_p$ = 0.011 m; 2 – $d_p$ = 0.012 m; 3 – $d_p$ = 0.013 m*

The strength condition has the following form

$$\Delta P_{\max}\frac{2R_1^2}{R_1^2 - R^2} \le \frac{\sigma_f}{n},$$

where $\sigma_f$ is the boundary of material fluidity; $n$ is the coefficient of the safety factor.

Then the wall thickness of the jet pump housing can be calculated by the following formula

$$\delta = R\left(\sqrt{\frac{\sigma_f/n}{\sigma_f/n - 2\Delta P_{\max}}} - 1\right). \quad (11)$$

Let us perform a numerical analysis of equation (11) for a specific engineering problem. Fig. 4 shows the dependence of relative wall thickness of the jet pump housing when the components of equation (11) take values $\sigma_f = 760$ MPa (steel 445ХГМА), $n = 2$.

The dependence $\delta / R = f(H_p)$ has a growing nonlinear character and can be used for substantiating the choice of the necessary wall thickness of the downhole jet pump housing.

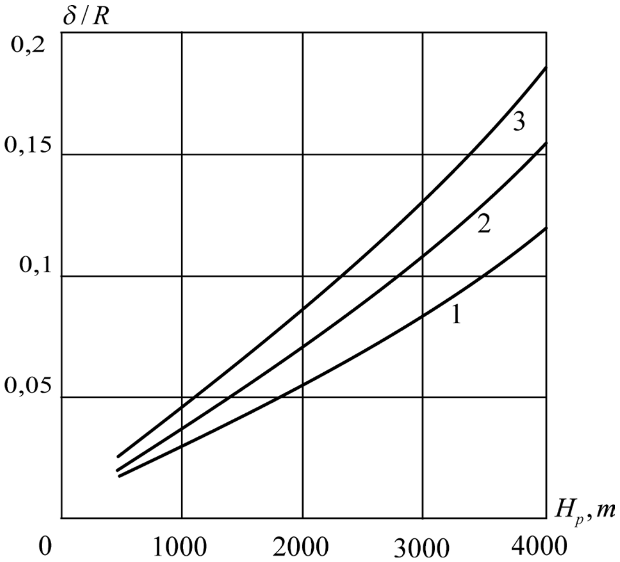


*Fig. 4. The dependence of the minimum relative thickness of the jet pump housing on the depth of its placement in the well and the density of the drilling fluid: 1 – $\rho = 1000$ kg/m$^3$; 2 – $\rho = 1200$ kg/m$^3$; 3 – $\rho = 1400$ kg/m$^3$*

**Conclusions and recommendations for further research in the area.**

1. Peculiarities of the borehole ejection system workflow increase the probability of destruction of the jet pump housing.
2. The research on the boundary stress state of the jet pump housing requires a joint analysis of the safety factor and parameters of the cavitation operation mode of the ejection system.
3. The value of the required thickness of the jet pump housing is directly proportional to the depth of its placement in the well and the density of the drilling fluid.
4. The prospect for further research is to substantiate the method for choosing the strength characteristics of jet pump surface elements, which are part of the systems of production, collection and preparation of oil and gas.

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**Мета.** Визначення граничних параметрів експлуатації свердловинної ежекційної системи, аналіз критичного напруженого стану корпуса струминного насоса та обґрунтування методу вибору його геометричних і міцнісних характеристик.

**Методика.** На основі використання класичних методів гідромеханіки та рівнянь Бернуллі, Дарсі-Вейсбаха й неперервності потоку встановлено, що найбільш важкі умови роботи корпуса струминного насоса виникають під час його експлуатації в кавітаційному режимі. Спільний аналіз умови виникнення гідродинамічної кавітації та умови міцності дав змогу оцінити граничний напружений стан корпуса струминного насоса.

**Результати.** У процесі моделювання робочого процесу струминного насоса показана необхідність визначення гідродинамічних параметрів потоку, що відповідають екстремальним умовам експлуатації ежекційної системи у свердловині. Отримане значення кавітаційної витрати робочого потоку визначає втрати напору в гідравлічному каналі затрубного простору свердловини й максимальну величину зовнішнього тиску, що діє на корпус струминного насоса. Приведена послідовність визначення напружень, що виникають у корпусі струминного насоса й мінімально необхідної за умов міцності товщини його стінки.

**Наукова новизна.** Полягає у встановленні взаємозв'язку між гідродинамічними й міцнісними характеристиками окремих елементів свердловинної ежекційної системи. Сформульовані принципи оцінки напруженого стану елементів ежекційної системи є теоретичною основою для вибору геометричних параметрів корпуса струминного насоса.

**Практична значимість.** Обґрунтування методу вибору конструкторських параметрів свердловинного струминного насоса підвищує надійність його використання й зменшує ймовірність аварій, що пов'язані з руйнуванням глибинної компоновки. Запропонований принцип спільного аналізу гідродинамічних і міцнісних характеристик може використовуватися також при моделюванні робочого процесу поверхневих струминних насосів, що входять до складу систем видобування, збору й підготовки нафти та газу.

**Ключові слова:** *нафтова свердловина, ежекційна система, струминний насос, кавітація, напружений стан, параметри експлуатації*

**Цель.** Определение предельных параметров эксплуатации скважинной эжекционной системы, анализ критического напряжённого состояния корпуса струйного насоса и обоснование метода выбора его геометрических и прочностных характеристик.

**Методика.** На основе использования классических методов гидромеханики и уравнений Бернулли, Дарси-Вейсбаха и непрерывности потока установлено, что наиболее тяжёлые условия работы корпуса струйного насоса возникают при его эксплуатации в кавитационном режиме. Совместный анализ условий возникновения гидродинамической кавитации и условия прочности позволил оценить предельное напряжённое состояние корпуса струйного насоса.

**Результаты.** В процессе моделирования рабочего процесса струйного насоса показана необходимость определения гидродинамических параметров потока, соответствующих экстремальным условиям эксплуатации эжекционной системы в скважине. Полученное значение кавитационного расхода рабочего потока определяет потери напора в гидравлическом канале затрубного пространства скважины и максимальную величину внешнего давления, действующего на корпус струйного насоса. Приведена последовательность определения напряжений возникающих в корпусе струйного насоса и минимально необходимой по условиям прочности толщины его стенки.

**Научная новизна.** Заключается в определении взаимосвязи между гидродинамическими и прочностными характеристиками отдельных элементов скважинной эжекционной системы. Сформулированные принципы оценки напряжённого состояния элементов эжекционной системы являются теоретической основой выбора геометрических параметров корпуса струйного насоса.

**Практическая значимость.** Обоснование метода выбора конструкторских параметров скважинного струйного насоса повышает надёжность его использования и уменьшает вероятность возникновения аварий, связанных с разрушением глубинной компоновки. Предложенный принцип совместного анализа гидродинамических и прочностных характеристик может использоваться также при моделировании рабочего процесса скважинных струйных насосов, входящих в состав систем добычи, сбора и подготовки нефти и газа.

**Ключевые слова:** *нефтяная скважина, эжекционная система, струйный насос, кавитация, напряжённое состояние, параметры эксплуатациио*

*Рекомендовано до публікації докт. техн. наук І. І. Чудиком. Дата надходження рукопису 18.07.16.*