LABYRINTH SCREW PUMP THEORY

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Summary. An analysis of existing methods of labyrinth screw pump design is presented. The physical model of its work process has been refined. An integrated method of calculating the flow characteristics of the pump has been developed. In contrast to known ones, it contains no empirical coefficients and takes into account the shape of the screw groove. New analytic expressions have been given for calculating instantaneous pump delivery and its irregularity coefficient. Dimensionless criteria have been offered to compare the characteristics of pumps with different shapes of the screw groove.

Keywords: labyrinth screw pump, characteristics, delivery, irregular delivery, screw groove shape, comparison criteria.

INTRODUCTION

The current level of technology and its further development is inextricably linked to intensifying the operation of units and hydraulic systems, increasing their performance and energy conservation. As regards labyrinth screw pumps (LSP) used widely for pumping corrosive media, heterogeneous and gas-liquid emulsions, allowing for high heads at low deliveries and working with low-viscosity corrosive liquids with a specific velocity of n_s = 10...40, which is particularly advantageous in comparison with centrifugal pumps, a topical task is developing a refined theory of the work process. Besides, these pumps are much easier to manufacture than vortex ones and are more reliable because mechanical friction of parts is absent.

ANALYSIS OF PUBLICATIONS

Currently, the number of theoretical studies in LSP is very limited. This is due to the novelty and complexity of pump hydrodynamic processes occurring therein. The vast majority of studies dealing with the work process in labyrinth screw channels are dedicated to the seal [1]. Study [2] discusses the work process in a labyrinth seal in the laminar regime, which was represented as two screw seals. The illegality of such analysis of the work process in a labyrinth screw seal was proved experimentally [3].

When considering the turbulent flow in an LSP [4], the primary differential equations used were Reynolds equations for turbulent flow without convective inertial terms that do not reflect the physics of processes. In addition, the theory expresses the average velocity in the form of power expansions for a stepwise radial coordinate with coefficients that can be determined only experimentally. In deriving the dependence of the pressure drop on the delivery, the authors used the equation of change of momentum in the pump groove. The result is a linear dependence of the pump head on delivery, this being in poorer agreement with experiment than the quadratic one [3].

Study [3] investigates LSP performance with a gas-liquid mixture. It presents the methodology for calculating the characteristics of an LSP working with a gas-liquid emulsion. The study mentions good agreement of theoretical calculations with experimental data; however, experimental results are not shown. In conducting experimental studies, gas and liquid are fed to the pump through separate ports (a vessel, in which they are mixed, is absent). This does not meet actual pump operating conditions. It is noted that the developed method needs to be rectified through the development and experimental investigation of such pumps. The maximum concentration of gas in the gas-liquid mixture, which the pump is able to deliver, has not been evaluated. No universal criteria for evaluating pump performance have been developed.

Work [5] compares the experimental characteristics of LSP and vortex pumps, and notes their similarity. On this basis, an attempt was made to consider the LSP work process, using the equation of angular momentum as this was done for the vortex pump [6]. However, in constructing the *H(q)* curve with this method, one needs to know two experimental factors, and the resulting experimental characteristic is a straight line, which is not true. Studies [3, 7] describe the work process and give the LSP design methodology. In describing the work process, the fluid flow in the pump is considered as flow of fluid between the developments of the screw and sleeve surfaces moving in opposite directions at a rotational speed equal to half the angular speed of the screw. The assumption is that the friction force occurring in the liquid forms pressure and friction forces (turbulent ones) on the protrusions of the screw and the sleeve. When building the *H(q)* characteristics with this method, the pump head coefficient is determined experimentally, and this limits the scope of the method being considered.

Work [8] proposes a new efficiency factor to analyse the effectiveness of hydraulic systems that take into account the power loss in such systems. Ways of improving pump delivery and reducing losses in the hydraulic system are considered. However, the proposed criteria are not suitable for analysing and comparing the characteristics of LSP with working members of various shapes.

An important characteristic of pumps, including LSP, is their irregular delivery determined by the irregular delivery factor [9]:

$$\delta_{í} \approx \delta_{t.í} + \delta_{s.í} \,, \tag{1}$$

where: $\delta_{\text{s. н}}$ is irregular delivery factor conditioned by compression of the working fluid (WF) in the cells of the pump during its transfer from the suction space to the decreasing one; $\delta_{\text{t. н}}$ is theoretical irregular delivery factor [9]:

$$\delta_{\text{t. н}} = \frac{q_{\max} - q_{\min}}{q_a} \,, \tag{2}$$

where: $q_{\max}$, $q_{\min}$ and q_a are respectively the maximum, minimum and average theoretical pump discharge.

Note that $\delta_{\text{s. н}}$ is calculated by a dependence similar to that in (2). In most papers devoted to design of screw pumps, the theoretical irregular delivery factor is taken equal to zero, whereas it was not determined at all for the LSP.

Thus, we can say that, presently, all methods of LSP design are based on using empirical relationships, the physical work process model has not been studied thoroughly and ignores the hydraulic resistance at the inlet and outlet of the screw grooves, and the flow between the LSP grooves. Little attention is paid to studying and improving the flow parts of such pumps. Fluid flow in the flow parts of these pumps has not been studied. No analytical dependence for calculating instantaneous pump delivery exists, nor are there any universal criteria for evaluating characteristics of LSP.

The above-stated determines the relevance of the paper aimed at solving an important scientific and practical problem, viz. development of the theory of LSP design.

PHYSICAL MODEL OF THE LABYRINTH SCREW PUMP WORKING PROCESS

WF energy is transferred in the pump due to rotation of the screw relative to the sleeve, resulting in force interaction between the fluid flowing over the screw and that flowing over the sleeve. The sleeve screw tips intensify this process. The conventional interface surface area between the screw and sleeve experiences a growing turbulent friction force enhanced by the centrifugal forces caused by rotation of the screw and the vortices formed by non-stationary WF flow, as well as by the vortex resulting from fluid leakage through the radial clearance between the sleeve and the screw grooves. Liquid viscosity is manifested as diffusion of vortices. Note that the turbulent friction forces and the vortices formation intensity depends directly on the geometrical parameters of the screw and sleeve, their clearance, the material properties (their roughness), the screw rotational speed and the parameters of the liquid being pumped.

Note that the nature of WF flow is not uniform over the entire length of the screw. The flow is formed at the groove inlet. Near the outlet, the flow pattern also changes because the groove opens. In any arbitrary cross section of the pump, the relative positions of the tips of the sleeve and screw threads periodically and continuously change. The frequency of change of tip positions - the main frequency of pulsations of the delivery rate in the given cross section – equals the product of screw rotational speed and the number of thread leads. This is confirmed by experimental studies of such pumps [3, 10].

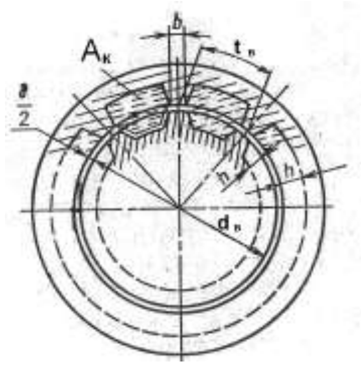
The friction forces occurring in the WF create pressure and friction forces on the tips of the screw and sleeve. With turbulent flow over the tips when the WF has relatively low viscosity, the main role is played by pressure forces perpendicular to the surface of thread tips. The components of this force in the axial direction determine the pump head.

Thus, during pump operation, the WF moves in two mutually opposite directions from pump inlet to its outlet (direct flow) in the screw grooves and is driven by rotation of the screw; from pump outlet to its inlet (reverse flow) through the clearance between the sleeve tip and the screw thread due to the pressure differential between the outlet and inlet of the pump, and the weight of the WF if the pump is mounted not horizontally.

PERFORMANCE OF THE LABYRINTH SCREW PUMP

In developing the methodology of LSP performance design, the authors relied on the physical model of the working process described above. The screw and sleeve were supposed to rotate in opposing directions. The centrifugal forces acting on the WF in the working space of the pump, which are proportional to the ratio of the pump flow section height and its mean radius, were neglected because of their small value. The WF flow in the groove was assumed quasistationary. Since the WF flow in the pump grooves is a developed turbulent one, it was considered stationary only relative to time-averaged velocity and pressure parameters and steady over a sufficiently long average period. Thus, the loss coefficient at the inlet and outlet of the pump and friction are determined by the same dependences as those for conventional hydraulic devices at steady-state fluid flow.

To simplify analysis of the pump work process, it was assumed that the screw and sleeve threads have the same shape and size, and are characterized by the hydraulic radius. The characteristic geometrical dimensions of the LSP are shown in Fig. 1.



$t_в$ – screw groove pitch in the cross section; b – screw tip width in the cross section; δ – diameter clearance between the screw and sleeve threads

Fig. 1 Characteristic geometric dimensions of labyrinth screw pump (cross section)

$t_в$ –шаг винтовой канавки в поперечном сечении; b – ширина верхушек винта в поперечном сечении; δ – диаметральный зазор между нарезками винта и втулки

Рис. 1 Характерные геометрические размеры лабиринтно-винтового насоса (поперечное сечение)

According to the physical model of the LSP work process, its average discharge is determined by the following relationship:

$$q_{\mathrm{a}} = z\left[q_1(n) - q_2(\Delta p_{\mathrm{vux}})\right], \tag{3}$$

where: q_1 is discharge due to displacement of fluid volume in the screw groove of the pump for one revolution of the screw; q_2 is discharge due to flow of fluid in the groove of the pump through the clearance formed by the sleeve thread tip and screw groove space under the influence of the pressure differential across the inlet and outlet of the pump Δp_{vux} and gravity in case of a non-horizontal position of the pump.

To obtain a universal dependence of the average pump discharge on various shapes of its working members, their dimensions were determined by hydraulic radius R_g.

The discharge due to movement of fluid volume in the screw groove of the pump per revolution of the screw was determined by the relationship:

$$q_1(n) = k_{\text{кw}} \mathrm{A}_{\text{к}} L_{\text{к}} \frac{n}{60}, \tag{4}$$

where: $k_{\text{кw}}$ is a factor accounting for the actual volume of the screw channel:

$$k_{\text{кw}} = \frac{\mathrm{A}_{\text{к}} L_{\text{к}} - \mathrm{A}_{\text{к}}\left(4R_{\mathrm{g}} + b\right) + \frac{4}{3}\pi\left(2R_{\mathrm{g}}\right)^3}{\mathrm{A}_{\text{к}} L_{\text{к}}}; \tag{5}$$

where: $A_к$ is screw channel area, m^2 (Fig. 1):

$$\mathrm{A}_{\text{к}} = 4\pi R_{\mathrm{g}}^2, \tag{6}$$

$L_к$ is helix length, m:

$$L_{\text{к}} = \sqrt{(\pi d_{\text{в}})^2 + s_{\text{в}}^{\,2}}. \tag{7}$$

The discharge due to the flow of fluid in the groove of the pump through the clearance formed by the sleeve thread tip and the screw groove space was calculated with the formula:

$$q_2(\Delta p_{\mathrm{vux}}) = \mu(\mathrm{Re})\left[\frac{\mathrm{A}_{\text{к}}}{2} + \delta R_{\mathrm{g}}\right]\sqrt{2g\left(\frac{p_{\mathrm{vux}} - p_{\mathrm{v}}}{\rho_{\mathrm{c}}\, g} + h_{\mathrm{vt}} + l_{\text{в}}\right)}, \tag{8}$$

where: h_{vt} is loss of pressure due to local resistances and friction in a channel formed by the sleeve thread tip and screw groove space found by relationship:

$$h_{\mathrm{vt}} = \left(\lambda(\mathrm{Re})\frac{L_{\text{к}}}{4R_{\mathrm{g}}} + \zeta_{\mathrm{v}} + \zeta_{\mathrm{vux}} + k_{\mathrm{ot}}\zeta_{\mathrm{otv}}\right)\frac{v_{\mathrm{ser.vux}}^2}{2g} \tag{9}$$

where: $v_{\mathrm{ser.vux}}$ is average WF velocity in the screw channel at the pump outlet found by relationship:

$$v_{\mathrm{ser.vux}} = \frac{q_{\mathrm{ser}}}{z\,\mathrm{A}_{\text{к}}}. \tag{10}$$

Note that, in formula (8) and in the sequel, the value of $l_{\text{в}}$, the screw length, was substituted if the pump position was not horizontal.

In formulas (8) and (9), the following notation was used: $\mu(\mathrm{Re})$ is loss coefficient in the clearance formed by the sleeve thread tip and screw groove space, which were determined based on the Reynolds number for the average WF velocity in the screw channel at the outlet of the pump according to formula [11]; the Reynolds number was determined from dependence $\mathrm{Re} = \frac{4 v_{\mathrm{ser.vux}} R_{\mathrm{g}}}{v_{\mathrm{t}}}$; v_{t} and ρ_{c} are WF kinematic viscosity and density, respectively accounting for the gas content, pressure and temperature, and calculated using dependences [12]; $\lambda(\mathrm{Re})$ is friction loss coefficient depending on Re for the average velocity of the WF in the screw channel at the pump outlet; ζ_{v}, ζ_{vux} and ζ_{otv} are, respectively, local loss coefficients at the inlet and outlet, and at sudden change in the flow section determined by the formulas [13]; $s_{\text{в}}$ is screw groove pitch; $l_{\text{в}}$ is screw length; k_{ot} is a factor accounting for the number of sudden changes in the flow section along the length of the screw groove calculated as:

$$k_{\text{ot}} = Int\left(l_{\text{в}}/s_{\text{в}}\right), \tag{11}$$

where: $Int\left(l_{\text{в}}/s_{\text{в}}\right)$ is the integer part of the number; $l_{\text{в}}/s_{\text{в}}$ is the nearest smaller number.

To ensure pump cavitation-free operation, the pressure at its inlet should be no less than the pressure of evaporation (for water, at 20 ºC the pressure of saturated steam is 2.3388 kPa; at 30 ºC, it is 4.2453 kPa) [14].

The method of calculating the pump discharge characteristic comprises the following sequence of actions [15, 16]:

First, one specifies the pump geometric parameters (the parameters of its working members), the parameters of the WF and the screw rotational speed.

The first stage of calculating with formula (4) was to determine the discharge defined by displacement of the volume of fluid in the screw groove of the pump per one screw revolution – $q_1(n)$.

The second stage was to specify the discharge defined by the flow of fluid found in the screw groove of the pump, which flows through the clearance formed by the sleeve thread tip and the screw groove space – $q_2(\Delta p_{\text{vux}})$. Note that the value of discharge $q_2(\Delta p_{\text{vux}})$ should not exceed the discharge $q_1(n)$. Formula (3) is used to find the average discharge at the pump outlet and formula (10) is used to find the average WF velocity in the screw channel at the pump outlet.

At the third stage, the Reynolds number is found using the average WF velocity in the screw channel at the pump outlet. Dependences (23) and (24) determine the coefficient of friction losses $\lambda(\text{Re})$, and formula (9) is used to find the pressure loss caused by local resistances and friction in a channel formed by the sleeve tip and screw groove space h_{vt}. The formula [11] is used to find the loss coefficient in the clearance formed by the sleeve thread tip and the screw groove space $\mu(\text{Re})$.

At the fourth stage, formula (8) is presented as:

$$p_{\text{vux}} - p_{\text{v}} = \Delta p_{\text{vux}} = \frac{q_2(\Delta p_{\text{vux}})^2 \rho_{\text{c}}}{2\mu(\text{Re})^2\left[\frac{A_{\text{к}}}{2} + \delta\, R_{\text{g}}\right]^2} - \rho_{\text{c}}\, g\left(h_{\text{vt}} + l_{\text{в}}\right) \tag{12}$$

The pressure at the pump inlet was assumed to be $p_{\text{v}} = 0$. The values found at stages two and three were substituted into formula (12), and the pressure at the pump outlet p_{vux} was found. Knowing q_{a} and p_{vux} yields the discharge characteristic point.

The fifth stage defines new discharge values $q_2(\Delta p_{\text{vux}})$, steps two to four are repeated and a new point of the discharge characteristic is found.

The calculation is repeated to obtain the required number of points for graph $H(q)$. Note that the maximum value of the pressure at the pump outlet is $q_1(n) = q_2(\Delta p_{\text{vux}})$.

The relative error between the calculated characteristics of the developed method and those obtained experimentally are given in [3, 10] and do not exceed 20 %. This discrepancy can be explained by the complexity of the vortices formation mechanism and fluid diffusion, and transfer of momentum at the screw-and-sleeve fluid interface. A unique advantage of the developed technique is that, as distinct from known methods, it does not contain empirical coefficients determined experimentally.

The power at the pump outlet is calculated as:

$$P = q_{\text{a}} \Delta p_{\text{vux}}\,. \tag{13}$$

LSP efficiency, according to the developed physical model, is calculated by the relationship:

$$\eta = \frac{q_{\text{a}}\ \Delta p_{\text{vux}}}{P_{\text{т}}} = \frac{30\, q_{\text{a}}}{\pi^2 R_{\text{g}}^2 d_{\text{в}} n \operatorname{ctg}\alpha}, \tag{14}$$

where: $P_{\text{т}}$ is theoretical capacity; $d_{\text{в}}$ is screw outer diameter; α is thread angle relative to the axis of the screw; n is screw rotational frequency, rev/min.

LSP IRREGULAR DELIVERY

The instantaneous discharge at the LSP outlet, depending on the angle of rotation of the screw, similar to formula (3), can be written as:

$$q_{\Sigma}(t) = z\left[q_1(\varphi(t)) - q_2(\Delta p_{\text{vux}})\right], \tag{15}$$

where: φ is screw angle of rotation.

We assumed that the screw thread tip varies according to a dependence describing the change of a short periodic triangular pulse [17], and the instantaneous discharge due to displacement of the volume of fluid in the screw groove of the pump was found by relationship:

$$q_1(\varphi(t)) = \text{A}_{\text{к}} L_{\text{к}} \varphi(t) - 4R_{\text{g}}\left(4R_{\text{g}} + b\right) h_{\text{в}}(\varphi(t)), \tag{16}$$

where: $h_{\text{в}}(\varphi)$ is sleeve thread tip change depending on the screw angle of rotation [17]:

$$h_{\text{в}}(\varphi)=\begin{cases}\dfrac{h_{\text{в}}}{k\pi}(\varphi+k\pi), & -k\pi\le\varphi<0,\\ \dfrac{h_{\text{в}}}{k\pi}(k\pi-\varphi), & 0\le\varphi<k\pi,\\ 0, & k\pi\le\varphi,<2\pi-k\pi,\end{cases}$$

$$k=\frac{4R_{\text{g}}+b}{L_{\text{к}}}\ ;\ k\ll 1. \qquad (17)$$

Note that the second term in equation (16) takes into account the discharge drop due to presence of the sleeve thread. Using $h_{\text{в}}(\varphi(t))$ found with (17) to compute the discharge, yields a slightly overrated result. However, this offsets to some extent the discharge drop due to WF compression in the screw groove.

Given dependence (16), the instantaneous discharge at the LSP outlet can be represented as:

$$q_{\Sigma}(\varphi(t))=z\,A_{\text{к}}L_{\text{к}}\varphi(t)-4\,R_{\text{g}}(4R_{\text{g}}+b)\sum_{0}^{z-1}h_{\text{гв}}(\varphi)\,[\varphi+k_z\beta]\,\varphi(t)-$$
$$-z\,\mu(\text{Re})\times\left[2\pi R_{\text{g}}^2+\delta\,R_{\text{g}}\right]\sqrt{2g\left(\frac{p_{\text{vux}}-p_{\text{v}}}{\rho_{\text{c}}\,g}+h_{\text{vt}}+l_{\text{в}}\right)}, \qquad (18)$$

where: k_z is a factor that successively takes the values 0, 1, 2,..,.(z - 1); β is the angle between the centres of the grooves, $\beta=2\pi/z$.

Note that formula (18) is valid under the condition:

$$z<Int\,(L_{\text{к}}/4R_{\text{g}}+b), \qquad (19)$$

where: $Int\,(L_{\text{к}}/4R_{\text{g}}+b)$ is the integer part of the number; $(L_{\text{к}}/4R_{\text{g}}+b)$ is the nearest smaller number.

Note that if condition (19) in the second term of equation (18) is not met, one should take into account the actual number of sleeve screw tips per one screw groove pitch. If condition (19) is not met when $z>Int\,(L_{\text{к}}/4R_{\text{g}}+b)$, that is, when there are two or more sleeve screw tips at the pump outlet at the same time, this is taken into account by the coefficient of the second term in equation (18). Then it was assumed that condition (19) holds. Note that Δp_{vux} is never zero because the pump inlet is always under vacuum.

The LSP irregular delivery coefficient conditioned by compression of the WF in the grooves of the pump, when it is transferred from the suction cavity to the discharge one, can be accepted to be $\delta_{\text{s. н}}\approx 0$ with sufficient precision for calculations. The validity of this approach stems from the fact that, according to the physical model developed for the pump work process, during pump operation the screw grooves communicate at all times with the discharge cavity. This is used for calculating the instantaneous discharge resulting transfer of the fluid volume found in the screw groove of the pump dependence (17), and yields a slightly overrated result. Hence, the overall irregular delivery coefficient for the LSP can be found from:

$$\delta_{\text{н}}\approx\delta_{\text{t. н}}. \qquad (20)$$

The coefficient of irregularity was found from formulas (3), (15) and (18), implying that $\sum_{0}^{z-1}h_{\text{в}}(\varphi)[\varphi+k_z\beta]=z\,h_{\text{в}}(\varphi_0)\approx 2\,z\,R_{\text{g}}$, and $\varphi(t)=n$.

Note that the error of such change does not exceed 0.01 %:

$$\delta_{\text{t. н1}}=\frac{2(4R_{\text{g}}+b)}{\pi\,L_{\text{к}}\bar{n}-(4R_{\text{g}}+b)}, \qquad (21)$$

where: $\bar{n}=n/n_0$ is relative rotational frequency of the screw, $n_0=1$ s^{-1}.

When $q_2(\Delta p_{vux})=0.5\,q_1(\varphi(t))$, irregular LSP delivery is given by:

$$\delta_{\text{t. н}}=\frac{4(4R_{\text{g}}+b)}{\pi\,L_{\text{к}}\bar{n}-2(4R_{\text{g}}+b)}. \qquad (22)$$

When $q_2(\Delta p_{\text{vux}})\approx 0$, the LSP irregular delivery coefficient is minimum.

If condition (19) is not met, that is, when there are two or more sleeve screw tips at the pump outlet at the same time, the irregular delivery coefficient increases. Note that when $q_1(\varphi(t))=q_2(\Delta p_{\text{vux}})$, the discharge at the LSP outlet is $q_{\Sigma}(t)=0$, and the irregular delivery coefficient has no sense.

Analysis of relationship (22) has shown that the irregular delivery coefficient of the LSP increases with hydraulic radius R_{g} and the width of the screw tips in cross section b, and is inversely proportional to the length of helical line $L_{\text{к}}$ and screw rotational speed n. The resulting minimum value of the irregular delivery coefficient of the LSP, calculated with formula (21), is $\delta_{\text{t. н1}}=3.897\cdot 10^{-4}$, being an order of magnitude less than in screw pumps [18, 19]. This finding is consistent with the experimental and computational studies presented in [3, 10, 20, and 21].

ACCOUNTING FOR THE SHAPE OF THE SCREW GROOVE IN LSP CHARACTERISTICS

Presently, there is no general analytical dependence for finding the coefficient of hydraulic friction λ that would simultaneously take into account the fluid flow regime and the shape of the channel. Hydraulic friction coefficients for the laminar flow regime are calculated using the known formula [22]:

$$\lambda = \frac{\mathrm{A}_f}{\mathrm{Re}}, \tag{23}$$

where: A_f is shape factor for laminar WF motion (Table 1).

Table 1

Shape factors [23]

Shape	Shape factor, flow regime	
	Laminar, A_f	Turbulent, K_f
Circle with the diameter *d*	64	0.11
Square with side *a*	57	0.098
Equilateral triangle with side *a*	53	0.091
Rectangle with an aspect ratio *a*/*b* = 0.1	85	0.15
a/*b* = 0.2	76	0.13
a/*b* = 0.25	73	0.12
a/*b* = 0.33	69	0.118
a/*b* = 0.5	62	0.10

Hydraulic friction coefficients for turbulent flow are determined with formula [23]:

$$\lambda = K_f \left(\frac{\Delta}{4R_\mathrm{g}} + \frac{68}{\mathrm{Re}} \right)^{0.25}, \tag{24}$$

where: K_f is shape factor for WF turbulent motion (Table 1); Δ is average height of roughness protrusions.

To calculate hydraulic losses associated with LSP inlet and outlet flow, we used dependences well known in fluid and gas mechanics to calculate the leakage flows between the screw grooves for the forward and reverse flow of fluid and proposed to introduce, by analogy with vortex diodes and jet resistor elements, the diodity factor:

$$\mathrm{D} = \zeta_{\mathrm{obr}} / \zeta_{\mathrm{pr}}, \tag{25}$$

where: ζ_{pr}, ζ_{obr} is coefficient of drag between the screw grooves for liquid flow in the forward and reverse directions, respectively.

When considering the leakage flow of fluid between the edges that separate the screw grooves, friction losses were neglected because such flow was small. When considering the leakage flow towards the pump discharge as narrowing and expanding at angle α, which, in the first approximation, can be taken as sudden, the resistance to direct fluid flow is determined by the relationship:

$$\zeta_{\mathrm{pr}} = \zeta_{1\mathrm{zv}} + \zeta_{1\mathrm{ros}}, \tag{26}$$

where: $\zeta_{1\mathrm{zv}}$ and $\zeta_{1\mathrm{ros}}$ are, respectively, the drag coefficient of the liquid narrowing and expanding at angle α determined by the diagrams in [13].

The drag coefficient for the reverse flow of fluid is given by:

$$\zeta_{\mathrm{obr}} = \zeta_{2\mathrm{zv}} + \zeta_{2\mathrm{ros}}, \tag{27}$$

where: $\zeta_{2\mathrm{zv}}$ and $\zeta_{2\mathrm{ros}}$ are drag coefficients, respectively, for fluid narrowing and expanding at angle α, [13].

To account for diodity in the LSP discharge characteristic, the coefficient accounting for the number of sudden changes in flow area along the length of screw grooves k_{ot}, which is calculated with formula (11), should be divided by diodity factor *D*. Analysis of the shape of LSP screw grooves performed in [23 – 25] allowed improving the shape of pump working members [26, 27].

CRITERIA FOR COMPARISON OF LSP CHARACTERISTICS

When calculating LSP performance, it is necessary to compare the characteristics of LSPs with working members having different shapes. Using the hydraulic radius of screw groove R_g allows for a partial analysis. However, analysing with R_g is challenging. Partly, this problem can be addressed by taking into account the hydraulic resistance coefficient for friction λ, which takes into account the shape of screw grooves. However, its use fails to account for all the geometrical parameters of LSP working members. Note that commercial LSPs tend to have a semi-circular shape of screw grooves, so the most important consideration is this shape.

For a comparative evaluation of performance of LSPs with working members of different shapes, we proposed to use relative diameter $\tilde{d} = d_{в} / R_{g}$, where $d_{в}$ is diameter of the screw, and the specific LSP parameters we have introduced [28]. Specific head is the head that accounts for a unit of relative length per one unit screw thread lead, and is calculated as:

$$\tilde{H} = \frac{H}{z\, l_{в} / R_{g}} \ . \tag{28}$$

Similar to the specific head, we introduced specific discharge $\tilde{q}$; specific capacity $\tilde{P}$; and specific efficiency $\tilde{\eta}$ calculated with the following dependences:

$$\tilde{q} = \frac{q}{z\, l_{в} / R_{g}} \ ; \ \tilde{P} = \frac{P}{z\, l_{в} / R_{g}} \ ; \ \tilde{\eta} = \frac{\eta}{z\, l_{в} / R_{g}} \ . \tag{29}$$

The proposed dimensionless criteria allow analysing the influence of the geometric shape of screw grooves on LSP performance, optimising them and obtaining a line of LSPs with high efficiencies.

CONCLUSIONS

Review of the literature has shown that no modern theory of LSP design exists. Currently available theories are based on using empirical relationships, and the physical model of the work process fails to fully account for the loss of friction in the LSP and the leakage flow between the grooves.

We have rectified the physical model of the LSP work process with regard to pressure loss at the pump inlet and outlet and the leakage flow across the radial clearance between the sleeve and the screw grooves.

We first developed an integrated methodology for calculating the LSP discharge characteristic. In contrast to known ones, it has no empirical coefficients found experimentally. It takes into account as a whole the geometric and operating parameters of the pump, the nonstationary nature of WF flow and its parameters. An analytical dependence for calculating the instantaneous delivery and LSP irregular delivery coefficient was obtained.

To evaluate the influence of the shape of working members on LSP performance, we suggested using the hydraulic radius of the screw groove, the relative diameter, specific head, specific discharge, specific capacity, efficiency and the diodity coefficient.

The theory developed at the LSP design phase will allow analysing the influence of the geometric shape of screw grooves on LSP performance, optimise them, and obtain a line of high-efficiency LSPs.

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ТЕОРИЯ ЛАБИРИНТНО-ВИНТОВЫХ НАСОСОВ

Аннотация: Приведен анализ существующих методик расчета лабиринтно-винтовых насосов. Уточнена физическая модель его рабочего процесса. Разработана интегральная методика расчета расходной характеристики насоса, которая в отличие от известных, не содержит эмпирических коэффициентов и учитывает форму винтовой канавки. Приведены новые аналитические зависимости для расчета мгновенной подачи насоса и коэффициента ее неравномерности. Предложены безразмерные критерии для сравнения характеристик насоса с разной формой винтовой канавки.

Ключевые слова: лабиринтно-винтовой насос, характеристики, подача, неравномерность подачи, форма винтовой канавки, критерии сравнения.