ISSN 2518-170X (Online),
ISSN 2224-5278 (Print)

ҚАЗАҚСТАН РЕСПУБЛИКАСЫ
ҰЛТТЫҚ ҒЫЛЫМ АКАДЕМИЯСЫНЫҢ
Satbayev University

Х А Б А Р Л А Р Ы

ИЗВЕСТИЯ

НАЦИОНАЛЬНОЙ АКАДЕМИИ НАУК
РЕСПУБЛИКИ КАЗАХСТАН
Satbayev University

N E W S

OF THE ACADEMY OF SCIENCES
OF THE REPUBLIC OF KAZAKHSTAN
Satbayev University

SERIES
OF GEOLOGY AND TECHNICAL SCIENCES

1 (445)

JANUARY – FEBRUARY 2021

THE JOURNAL WAS FOUNDED IN 1940

PUBLISHED 6 TIMES A YEAR

ALMATY, NAS RK

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Қазақстан Республикасы Ұлттық ғылым академиясы "ҚР ҰҒА Хабарлары. Геология және техникалық ғылымдар сериясы" ғылыми журналының Web of Science-тің жаңаланған нұсқасы Emerging Sources Citation Index-те индекстелуге қабылданғанын хабарлайды. Бұл индекстелу барысында Clarivate Analytics компаниясы журналды одан әрі the Science Citation Index Expanded, the Social Sciences Citation Index және the Arts & Humanities Citation Index-ке қабылдау мәселесін қарастыруда. Webof Science зерттеушілер, авторлар, баспашылар мен мекемелерге контент тереңдігі мен сапасын ұсынады. ҚР ҰҒА Хабарлары. Геология және техникалық ғылымдар сериясы Emerging Sources Citation Index-ке енуі біздің қоғамдастық үшін ең өзекті және беделді геология және техникалық ғылымдар бойынша контентке адалдығымызды білдіреді.

НАН РК сообщает, что научный журнал «Известия НАН РК. Серия геологии и технических наук» был принят для индексирования в Emerging Sources Citation Index, обновленной версии Web of Science. Содержание в этом индексировании находится в стадии рассмотрения компанией Clarivate Analytics для дальнейшего принятия журнала в the Science Citation Index Expanded, the Social Sciences Citation Index и the Arts & Humanities Citation Index. Web of Science предлагает качество и глубину контента для исследователей, авторов, издателей и учреждений. Включение Известия НАН РК. Серия геологии и технических наук в Emerging Sources Citation Index демонстрирует нашу приверженность к наиболее актуальному и влиятельному контенту по геологии и техническим наукам для нашего сообщества.

«ҚР ҰҒА Хабарлары. Геология және техникалық ғылымдар сериясы».

ISSN 2518-170X (Online),
ISSN 2224-5278 (Print)

Меншіктенуші: «Қазақстан Республикасының Ұлттық ғылым академиясы» РҚБ (Алматы қ.).

Қазақстан Республикасының Ақпарат және қоғамдық даму министрлігінің Ақпарат комитетінде 29.07.2020 ж. берілген **№ KZ39VPY00025420** мерзімдік басылым тіркеуіне қойылу туралы куәлік.

Тақырыптық бағыты: ***геология және техникалық ғылымдар бойынша мақалалар жариялау.***

Мерзімділігі: жылына 6 рет.
Тиражы: 300 дана.

Редакцияның мекенжайы: 050010, Алматы қ., Шевченко көш., 28, 219, 220 бөл., тел.: 272-13-19, 272-13-18,
http://www.geolog-technical.kz/index.php/en/

Типографияның мекенжайы: «NurNaz GRACE», Алматы қ., Рысқұлов көш., 103.

«Известия НАН РК. Серия геологии и технических наук».

ISSN 2518-170X (Online),
ISSN 2224-5278 (Print)

Собственник: Республиканское общественное объединение «Национальная академия наук Республики Казахстан (г. Алматы).

Свидетельство о постановке на учет периодического печатного издания в Комитете информации Министерства информации и общественного развития Республики Казахстан **№ KZ39VPY00025420**, выданное 29.07.2020 г.

Тематическая направленность: ***публикация статей по геологии и техническим наукам.***

Периодичность: 6 раз в год.
Тираж: 300 экземпляров.

Адрес редакции: 050010, г. Алматы, ул. Шевченко, 28, ком. 219, 220, тел.: 272-13-19, 272-13-18,
http://www.geolog-technical.kz/index.php/en/

Адрес типографии: «NurNaz GRACE», г. Алматы, ул. Рыскулова, 103.

News of the National Academy of Sciences of the Republic of Kazakhstan. Series of geology and technology sciences.

ISSN 2518-170X (Online),
ISSN 2224-5278 (Print)

Owner: RPA "National Academy of Sciences of the Republic of Kazakhstan" (Almaty).

The certificate of registration of a periodical printed publication in the Committee of information of the Ministry of Information and Social Development of the Republic of Kazakhstan **No. KZ39VPY00025420**, issued 29.07.2020.

Thematic scope: ***publication of papers on geology and technical sciences.***

Periodicity: 6 times a year.
Circulation: 300 copies.

Editorial address: 28, Shevchenko str., of. 219, 220, Almaty, 050010, tel. 272-13-19, 272-13-18,
http://www.geolog-technical.kz/index.php/en/

Address of printing house: «NurNaz GRACE», 103, Ryskulov str, Almaty.

N E W S
OF THE NATIONAL ACADEMY OF SCIENCES OF THE REPUBLIC OF KAZAKHSTAN
SERIES OF GEOLOGY AND TECHNICAL SCIENCES
ISSN 2224-5278
Volume 1, Number 445 (2021), 73 – 81 **https://doi.org/10.32014/2021.2518-170X.11**

UDC 628.32

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A MODEL FOR CALCULATING THE COLLECTION AND PUMPING OF RAIN AND MELT WATER IN THE DESIGN OF MULTIFUNCTIONAL BUILDINGS FOR PUBLIC AND INDUSTRIAL PURPOSES

Abstract. In drainage systems, it is often necessary to raise polluted rain and melt water to a low height. For example, pumping wastewater from one gravity collector to another, raising wastewater by several meters in order to reduce the deepening of the collecting sewer collector and, as a result, reduce capital costs for its construction. In such places, it may be more economically feasible to reduce such deepening. The novelty of the study is that the development of a drain can be fulfilled only if the filling of the throughput of the collectors is sufficiently ensured. The study shows that the possibilities of expanding collectors can be achieved based on compensation for energy consumption. The study outlined the necessity of allocating the device of a high-pressure pumping station with the subsequent laying of a pressure sewage collector to a common collector, or to the main sewer network, which in turn supplies rain and melt water to the main collector. The practical significance of the study is the widespread introduction of intermediate pumps of low pressure in sewerage networks is constrained by the lack of a sufficient range of reliable and economical low pressure pumps for contaminated liquids.

Key words: pumps, heads, treatment system, rain and melt water.

Introduction. Special purpose-made centrifugal pumps are mainly used for pumping wastewater. They are distinguished by the fact that they do not require preliminary wastewater treatment [1]. Due to the complexity of adjustment, the centrifugal blades of the discs operate most of the time with excess heads, which leads to waste of energy [2]. In order to reduce the likelihood of clogging, centrifugal impellers are made with fewer blades than pumps for clean water (most often 1-3 blades per disc) [3]. Fewer blades result in lower efficiency [4]. A centrifugal pump with a head up to 10 metres of water column for certain flow rates is hard to find these days [5]. The need to raise water to this height often occurs during the operation of sewage collection systems [6-9]. Therefore, these tasks require the use of pumps operating with excess heads. These excess heads then have to be extinguished by throttling or by building special extinguishing chambers (additional capital costs). Screw pumps can be used to raise rain and melt water to low heights (up to 8–10 meters) [10].

At the wastewater system, it becomes necessary to pump sludge from primary sedimentation tanks, as well as fine-grained sand from secondary sedimentation tanks. The lack of high speeds inside the screw pumps makes them promising for use as dredging pumps that drain rainwater into the collection network [11-14]. However, these pumps have not been widely used in the sewer industry [15-19]. This is due to the fact that traditional designs of screw pumps have a number of disadvantages that hinder their widespread adoption [20].

Materials and methods. One of the main characteristics of any pump is its flow – the volume of liquid that the pump lifts per unit of time [21-25]. The main equation for determining the theoretical water transmission is the following (1)

$$Q_{teor} = Wan \tag{1}$$

where W – the volume of water in the interblade space of the pump, m^3; a – number of rotations; n – rotational velocity of the screw, min^{-1}.

A complex task is to determine the volume of water in the interblade space of the screw. It depends not only on the design of the screw, but also on the operating characteristics of the pump, such as the rotational speed of the screw, its angle of inclination, and the like. Consider a simplified diagram of a screw pump, which consists of a hub, on the cylindrical surface of which a tube of infinitely small cross-section is coiled along the helix. Let us choose two coordinate systems with a common origin at point O, located on the axis of the screw hub. The first system with traditional coordinate axes x, y, z is tied to the horizon, the second – α, β, γ (2) is tied to the axis of the screw hub. A line A-A is drawn so that it would be a line of the horizontal surface of the liquid in the space between turns. The parametric equation of the helix in coordinates α, β, γ is written as

$$\alpha = r_{zil} \cos\varphi, \beta = r_{zil} \sin\varphi, \gamma = \frac{h}{2\pi}\varphi \tag{2}$$

where h – lead of helix, cm.

Performing the transition from one coordinate system to another, determine the difference between points lying on this line over the horizontal surface passing through the origin of the coordinate system x, y, z. This difference is determined by the equation (3)

$$x = r_{zil} \cos\lambda\,(\varphi tg\lambda tg\varepsilon r_{zil} + \cos\varphi) \tag{3}$$

where r_{zil} – radius of the cylindrical surface of the hub, cm; φ – helical line angle; λ – angle of the screw above the horizon; $tg\varepsilon r_{zil}$ – tangent of an angle of helix pitch.

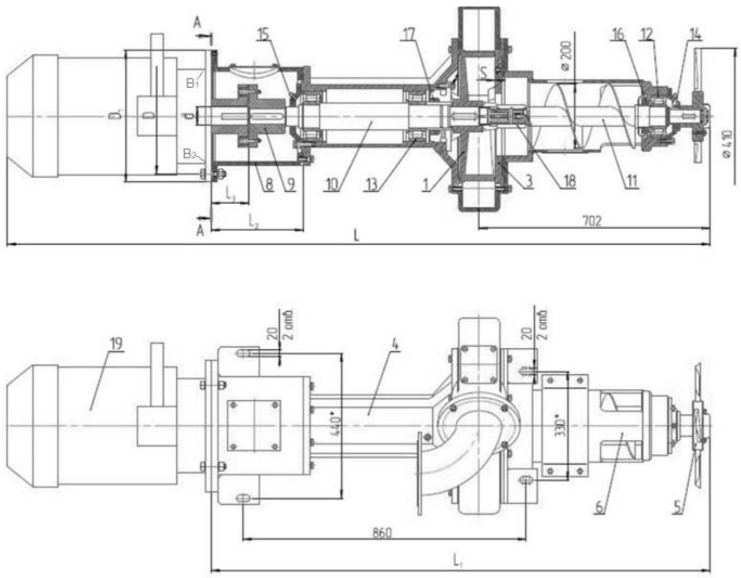

Figure 1 – Typical drawing of a screw pump

On a tube of infinitely small diameter, forming a helical line, there are two points B_1 and B_2, which have the same ordinates x_1 and x_2 of sections with the horizon line A-A. The point B_1 is determined by the helical line angle φ_1, point B_2 – by the rotational angle φ_2. Let there be a certain volume of liquid between these points. The point B_1 is the top bend point of the tube. In the case when the horizon line A-A rises above the point B_1, the liquid will begin to flow into the lower annular space. Therefore, this point is decisive for the liquid level when the maximum filling of the interblade space is observed. Since the point B_1 is a knick point, its coordinate x_1 has a maximum value over a certain interval. Taking this into account, from formula (3), the helical line angle φ_1 can be determined by equating the derivative (3) to zero (5–6)

$$dx = r_{zil} \cos\lambda\,(\varphi tg\lambda tg\varepsilon r_{zil} + \cos\varphi) = 0 \tag{4}$$

From which

$$tg\lambda tg\varepsilon r_{zil} - sin\,\varphi_1 = 0, sin\,\varphi_1 = tg\lambda tg\varepsilon r_{zil} \quad (5)$$

In order for the angle φ_1 to take positive values, it is necessary that the condition was met (6–7)

$$tg\lambda tg\varepsilon r_{zil} \leq 1 \quad (6)$$

or

$$\lambda + \varepsilon r_{zil} \leq 90^0 \quad (7)$$

Equation (7) determines the maximum tilt angle of the screw pump, at which it can work. When the angle is exceeded, the liquid will completely flow from the upper interblade space to the lower one. At the same time, the bigger the angle of elevation of the helical line εr_{zil}, the smaller should be the angle of installation of the auger to the horizon λ. Let us find the angle φ_2. The coordinates of the points B_1 and B_2 along the axis x are expressed by the equations (8–9)

$$x_1 = r_{zil}\,cos\,\lambda\,(\varphi_1 tg\lambda tg\varepsilon r_{zil} + cos\,\varphi_1) \quad (8)$$

$$x_2 = r_{zil}\,cos\,\lambda\,(\varphi_2 tg\lambda tg\varepsilon r_{zil} + cos\,\varphi_2) \quad (9)$$

Taking into account their congruence (10)

$$r_{zil}\,cos\,\lambda\,(\varphi_1 tg\lambda tg\varepsilon r_{zil} + cos\,\varphi_1) = r_{zil}\,cos\,\lambda\,(\varphi_2 tg\lambda tg\varepsilon r_{zil} + cos\,\varphi_2) \quad (10)$$

from which (11)

$$\frac{cos\,\varphi_1 - cos\,\varphi_2}{\varphi_1 - \varphi_2} = tg\lambda tg\varepsilon r_{zil} \quad (11)$$

The length of the arc holding water in the interblade space can be expressed by the equation (12)

$$S = \frac{r_{zil}(\varphi_1 - \varphi_2)}{cos\,\lambda} \quad (12)$$

In order to find the volume of liquid that is placed in the coil of the helical tube, it is necessary to multiply the length of the arc by the cross-sectional area of the tube. In a real pump, the pitch angles of the helical line at the inner and outer radius of the helical blade are different. The relationship between them can be expressed by the equation (13)

$$\frac{\lambda_r}{\varepsilon_R} = arctg\,\frac{r}{R} \quad (13)$$

where r – inner radius of the helical blade, cm; R – outer radius of the helical blade, cm. It can be seen that the angle of ascent of the blade along the inner radius is greater than along the outer one.

Calculating the volume of liquid in the interturn space, using the equation (14)

$$V = \int_{\varphi_1}^{\psi_1}\int_{r}^{r_{zil}} \Delta\gamma r_{zil} dr_{zil} d\varphi + \int_{\psi_1}^{\varphi_2}\int_{r}^{R} \Delta\gamma r_{zil} dr_{zil} d\varphi + \int_{\varphi_2}^{\psi_2}\int_{r}^{r_{zil}} \Delta\gamma r_{zil} dr_{zil} d\varphi \quad (14)$$

where ψ_1, ψ_2 – rotation angles of the helical line of the outer contour of the screw blade to the points of intersection with the horizon line A-A; r_{zil} – function (15) which can be found from equation (13), taking $\Delta\gamma = 0$

$$r_{zil} = \frac{tg\lambda}{cos\,\varphi}\left(\frac{x_1}{sin\,\lambda} - tg\varphi\right) \quad (15)$$

Results and Discussion. Conventional screw pumps are single or multi-sided screws that rotate at low frequency in a specially designed open tray, or in a closed tube. The screw pump has the following performance characteristics: *Q* – feed; *H* – height of liquid rise (water head); *N* – power consumption; *η* – efficiency. The design and operating characteristics of the pump: *D* – screw diameter (outer diameter of the blades); *d* – diameter of the screw shaft (screw hub); *S* – helix pitch distance of screw; *L* – length of the screw; *a* – number of rotations; *a* – angle of inclination to the horizon; *δ* – the gap between screw and tray; c – distance along the axis of the screw between two adjacent blades [26-28]. Based on the calculated data, the study proposes the following model for installing a screw pump. It is proposed to manufacture two screws with external diameters $D = 0.39m$ and $0.5m$ and the length $L = 4.6m$, and the ratio of diameters $d/D = 0.5$ each. The installation will make possible to change the angle of inclination of the screw a, its rotation frequency n, water level in the channel in front of the screw, as well as the gap between the screw and the tray in which it rotates δ. It is known that the total pump efficiency η_{povn} consists of its mechanical η_{mex}, hydraulic η_{gidr} and volumetric efficiency η_{obemn} and is determined by the equation (16)

$$\eta_{povn} = \eta_{mex}\eta_{gidr}\eta_{obemn} \tag{16}$$

Mechanical efficiency depends on the design of the pump. In the experimental setup, an electric drive with a single-stage reduction unit is used. For this design, the mechanical efficiency is determined at the level of $\eta_{mex} = 0.85$. The author does not recommend determining the volumetric and hydraulic efficiency separately, considering it a difficult task. It is proposed to find experimentally the product of the volumetric and hydraulic efficiency (23), calling it the general hydraulic efficiency.

$$\eta_{povn} = \eta_{gidr}\eta_{zag.gidr} \tag{17}$$

Then the total efficiency can be determined by the equation (18)

$$\eta_{povn} = \eta_{gidr}\eta_{zag.gidr} \tag{18}$$

The value $\eta_{zag.gidr}$ is found by determining the power developed by the pump and the power consumed by it minus mechanical losses (determined by the mechanical efficiency). At high speeds, splashing losses become significant. It is proposed to theoretically determine the amount of overflows through the gaps according to known equations. Partly as the flow rate during the outflow through a small hole in a thin wall at a constant pressure, and partly as the flow rate when flowing out through a large rectangular hole. For this, another method is proposed for determining the flows through the gaps, as well as another method for determining η_{gidr} and η_{obemn} of the pumps, then comparing it with the experimental Q_{dosl} under the same conditions, that was determined experimentally. The pump lifts the liquid in portions, the volume of which is equal to the volume of the section between adjacent blades of the screw. The section can be considered as a cylinder with a generatrix and oblique, but parallel to each other, bases [29-31].

Then the volume of liquid in one section of the pump, subject to its maximum filling, can be determined by the formula (19)

$$W = W_1 - W_2 \tag{19}$$

where W_1 – volume forming a cylindrical tray in which the auger and adjacent auger blades rotate, cm^3; W_2 – volume forming the submerged part of the inner shaft (hub) of the screw, cm^3.

These volumes can be determined by the equations (20–21)

$$W_1 = \frac{F_1+F_2}{2}c \tag{20}$$

$$W_1 = \frac{f_1+f_2}{2}c \tag{21}$$

where F_1,F_2 – the area of the wetted surface of the lower and upper blades, respectively, cm^2, f_1,f_2 – area of the conditionally wetted surfaces of the screw hub sections in the lower and upper parts of the section, cm^2.

Let us express the area F_1 and f_1 through the filling of the screw section in its lower part of H_1 and h_1. Taking into account the fact that at the maximum filling of the pump section we will obtain the ratio (22–23),

$$H_{max} = \frac{D+d}{2} \tag{22}$$

$$h_{max} = d \tag{23}$$

where H_{max},h_{max} – maximum filling of the section along the screw diameter and the diameter of the hub, respectively; D,d – screw and hub diameters, cm.

Taking into account equation (23) obtain (24–25)

$$\frac{2H_{max}}{D} - 1 = \frac{2(D+d)}{2D} = \frac{d}{D} \tag{24}$$

$$\frac{2h_{max}}{d} - 1 = 1 \tag{25}$$

Let us express the areas included in equations (20–21) through the diameters of the screw and hub, as a result obtain (26–27), the areas F_2 and f_2 are expressed by equations (30–31)

$$F_1 = \frac{D^2}{8}\left[2\pi - 2arccos\left(\frac{2H_1}{D} - 1\right)\right] + \frac{D^2}{8}sin\left[2arccos\left(\frac{2H_1}{D} - 1\right)\right] \tag{26}$$

$$f_1 = \frac{d^2}{8}\left[2\pi - 2arccos\left(\frac{2h_1}{d} - 1\right)\right] + \frac{d^2}{8}sin\left[2arccos\left(\frac{2h_1}{d} - 1\right)\right] \tag{27}$$

Considering that the functional connections between the fillings in the upper and lower parts of the screw section have the form (28-29),

$$H_2 = H_1 - c\,tg\alpha \tag{28}$$

$$h_2 = h_1 - c\,tg\alpha \tag{29}$$

$$F_2 = \frac{D^2}{8}\left[2\pi - 2arccos\left(\frac{2(H_1 - c\,tg\alpha)}{D} - 1\right)\right] + \frac{D^2}{8}sin\left[2arccos\left(\frac{2(H_1 - c\,tg\alpha)}{D} - 1\right)\right] \tag{30}$$

$$f_2 = \frac{d^2}{8}\left[2\pi - 2arccos\left(\frac{2(h_1 - c\,tg\alpha)}{d} - 1\right)\right] + \frac{d^2}{8}sin\left[2arccos\left(\frac{2(h_1 - c\,tg\alpha)}{d} - 1\right)\right] \tag{31}$$

The theoretical pump feed Q_{teor}, defined as the product of the volume of one section at its maximum filling W_{max} by the number of sections N (32–33), raising water per unit of time

$$Q_{teor} = W_{max}N \tag{32}$$

$$N = an' \tag{33}$$

where a – number of blades in the screw, n' – its rotation frequency (c^{-1}).

Then the losses of the pumped liquid, for overflow through the gaps between the screw blades and the cylindrical tray, as well as overflow over the edge of the screw, will be determined by the equation (34)

$$Q_{peretik} = Q_{teor} - Q_{dosl} \tag{34}$$

The volumetric efficiency is determined by the equation (35)

$$\eta_{obemn} = \frac{Q_{dosl}}{Q_{teor}} \tag{35}$$

The proposed model makes possible to determine the flow rate of the screw pump Q^1_{dosl} with the following parameters: screw diameter $D^1 = 0.39m$, crew hub diameter $d^1 = 0.195m$, screw filling $\frac{h^1}{D^1} = 0.75$, angle of inclination to the horizon $\alpha^1 = 30^0$, gap between the screw and the tray $\delta^1 = 5.2\,mm$. The screw rotation frequency was changed in the range $n = 75 \div 120$ rpm (min^{-1}). Calculating the volumetric η_{obemn} and hydraulic η_{gidr} efficiency of this pump at different speeds, using equations (25–41) and the experimental values of its delivery. The calculation will be carried out for all flow rates in litres per second for speeds from 75 to 120 min^{-1} at 5 min^{-1} intervals (table 1).

First, express the screw speed in rotations per second $n'(c^{-1})$. The number of emptying sections during the operation of the screw in one second for a speed of $75min^{-1}$ will be $N_{(390)75} = 3 \times 1.25 = 3.75pc$. For other speeds, look at the value $N_{(390)}$ in the table 1.

Table 1 – Components of the efficiency of a screw pump with a screw diameter of 390 mm

n (min^{-1})	$n^{'}(s^{-1})$	N $(sekz/s)$	Q_{teor} (l/s)	Q_{dosl} (l/s)	$Q_{peretik}$ (l/s)	η_{obemn}	η_{gidr}	$\eta_{zag\ gidr}$	η_{povn}
75	1.25	3.75	28.12	15.0	13.12	0.53	0.96	0.51	0.43
80	1.33	3.99	29.93	17.0	12.93	0.57	0.93	0.53	0.45
85	1.42	4.26	31.95	19.0	12.95	0.59	0.92	0.54	0.46
90	1.5	4.5	33.75	20.0	13.75	0.59	0.93	0.55	0.47
95	1.58	4.74	35.55	21.5	14.05	0.60	0.88	0.53	0.45
100	1.67	5.01	37.58	22.5	15.08	0.60	0.85	0.51	0.43
105	1.75	5.25	39.38	24.0	15.38	0.61	0.82	0.50	0.42
110	1.83	5.49	41.18	25.5	15.68	0.62	0.76	0.47	0.40
115	1.92	5.76	43.2	27.0	16.2	0.63	0.68	0.43	0.37
120	2.0	6.0	45.0	28.0	17.0	0.62	0.61	0.38	0.32

To determine the volume of one section of the pump, calculate the necessary areas of the wetted walls of the section using equations (32-37):

$$F_{2(390)} = \frac{0.39^2}{8}\left[2 \times 3.14 - 2arccos\ (\frac{2\times(0.2925-0.13tg30°}{0.39} - 1)\right] + \\ + \frac{0.39^2}{8} sin\left[2arccos\ (\frac{2\times(2925-0.13tg30°)}{D} - 1)\right] = 0.0684(m^2) \quad (36)$$

$$f_{1(390)} = \frac{0.195^2}{8}\left[2 \times 3.14 - 2\arccos\ (\frac{2 \times 0.195}{0.195} - 1)\right] + \\ + \frac{0.195^2}{8} sin\left[2arccos\left(\frac{2\times 0.195}{0.195} - 1\right)\right] = 0.0298(m^2) \quad (37)$$

$$f_{2(390)} = \frac{0.195^2}{8}\left[2 \times 3.14 - 2arccos\left(\frac{2 \times (0.195 - 0.13tg30^0)}{0.195} - 1\right)\right] + \\ \frac{0.195^2}{8} sin\left[2arccos\left(\frac{2\times(0.195-0.13tg30^0)}{0.195} - 1\right)\right] = 0.0192(m^2) \quad (38)$$

Determine the volume of one section, subject to the maximum filling by the equations (20–21):

$$W_{1(390)} = \frac{0.096+0.0684}{2} 0.13 = 0.0107(m^3) \quad (39)$$

$$W_{2(390)} = \frac{0.0298+0.0192}{2} 0.13 = 0.0032(m^3) \quad (40)$$

$$W_{max(390)} = 0.0107 - 0.0032 = 0.0075(m^3) \quad (41)$$

Determine the theoretical pump flow for a speed of 75 $min^{-1}$by the equation (32)

$$Q_{teor(390)75} = 7{,}5 \times 3{,}75 = 28{,}12(l/s) \quad (42)$$

The value of overflows through the gaps for the screw rotation frequency of 75 min^{-1}, determined by the equation (34), will be (Table 1):

$$Q_{peretik(390)75} = 28.12 - 15 = 13.12(l/s) \quad (43)$$

The volumetric efficiency determined by equation (35), will be:

$$\eta_{obemn(390)75} = \frac{15}{28.12} = 0.53 \quad (44)$$

The obtained data and the overall hydraulic efficiency are written in the Table 1, that is, η_{obemn} x η_{gidr}, from where the hydraulic efficiency from the equation (17) is determined:

$$\eta_{gidr(390)75} = \frac{\eta_{zag\ gidr(390)75}}{\eta_{obemn(390)75}} = \frac{0.51}{0.53} = 0.96 \quad (45)$$

The total pump efficiency, calculated by equation (22), will be:

$$\eta_{povn(390)75} = \eta_{obemn(390)75} \times \eta_{gidr(390)75} \times \eta_{mex} = 0.53 \times 0.96 \times 0.85 = 0.43 \quad (46)$$

At high rotational speeds (more than $115\ min^{-1}$) the amount of liquid increases, the filling of the pump sections is splattered and decreases through the dynamic curvature of the free surface of the liquid, and the volumetric efficiency decreases. The hydraulic efficiency is sufficiently high at the low speeds.

Conclusion. The expediency of using screw pumps for pumping rain and melt water and recirculating activated sludge at treatment facilities has been substantiated, which ensures operation without significant excess heads, as well as without significant dynamic loads. A mathematical model of the pump operation has been developed and an improved method for calculating the components of the efficiency of screw pumps has been proposed, on the basis of which the structure of energy forms is shown. The constructive improvements to increase the overall efficiency of the pump have been developed. Based on predictive calculations, it is shown that it is possible to pump rain and melt water, and activated sludge with an efficiency that exceeds existing analogues by 30% without reducing the pump's performance.

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ҚОҒАМДЫҚ ЖӘНЕ ӨНДІРІСТІК МАҚСАТТАҒЫ КӨПФУНКЦИОНАЛДЫ ҒИМАРАТТАРДЫ ЖОБАЛАУ КЕЗІНДЕГІ ЖАУЫН ЖӘНЕ ЕРІГЕН СУДЫ ЖИНАУ МЕН СОРУДЫ ЕСЕПТЕУГЕ АРНАЛҒАН ҮЛГІ

Аннотация. Су тарту жүйелерінде ластанған жаңбыр мен еріген суды шамалы биіктікке көтеру қажет. Мысалы ретінде ағын суды өздігінен ағатын бір резервуардан басқасына айдау, резервуар тереңдігін азайту және соның салдарынан құрылысына кететін шығынды азайту мақсатында ағынды суды бірнеше метрге көтеруді атап өтуге болады. Мұндай аудандарда тереңдікті азайту экономика тұрғысынан тиімді болуы мүмкін. Зерттеудің ғылыми жаңалығы су қоймаларын жеткілікті деңгейде толтыру арқылы қалыптастыруға болатындығымен айқындалады. Зерттеу жұмысы коллекторларды кеңейту мүмкіндігі энергия компенсациясы негізінде қолжетімді болатындығын көрсетті. Зерттеу қысымды сорғы станциясының жалпы коллектормен немесе негізгі кәріз желісіне кәріз қысымын төсеу арқылы таралу қажеттігін көрсетті әрі бұл жаңбыр мен еріген суды негізгі коллекторға айдауды қамтамасыз етеді. Зерттеудің практикалық маңыздылығы кәріз желілеріне төмен қысымды аралық сорғыларды кеңінен енгізу болып есептеледі, бұл ластанған сұйықтық үшін сенімді және үнемді төмен қысымды сорғылардың жеткілікті ассортиментінің болмауымен шектеледі.

Түйін сөздер: сорғы, тегеурін, тазарту жүйесі, жауын-шашын және қар суы.

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МОДЕЛЬ РАСЧЕТА СБОРА И ПЕРЕКАЧКИ ДОЖДЕВЫХ И ТАЛЫХ ВОД ПРИ ПРОЕКТИРОВАНИИ МНОГОФУНКЦИОНАЛЬНЫХ ЗДАНИЙ ОБЩЕСТВЕННОГО И ПРОМЫШЛЕННОГО НАЗНАЧЕНИЙ

Аннотация. В дренажных системах часто приходится поднимать на небольшую высоту загрязненные дождевые и талые воды. Например, перекачивание сточных вод из одного гравитационного коллектора в другой, подъем сточных вод на несколько метров с целью уменьшения углубления коллектора коллекторной канализации и, как следствие, снижения капитальных затрат на его строительство. В таких местах экономически более целесообразно уменьшить такое углубление. Новизна исследования заключается в том, что разработка дренажа может быть выполнена только при условии достаточного обеспечения заполнения пропускной способности коллекторов. Исследование показывает, что возможности расширения коллекторов могут быть достигнуты на основе компенсации энергопотребления. В исследовании обозначена необходи-

мость выделения устройства насосной станции высокого давления с последующей прокладкой напорного канализационного коллектора к общему коллектору или к магистральной канализационной сети, которая в свою очередь подает дождевую и талую воду в магистральный коллектор. Практическая значимость исследования заключается в том, что широкое внедрение промежуточных насосов низкого давления в канализационные сети сдерживается отсутствием достаточного ассортимента надежных и экономичных насосов низкого давления для загрязненных жидкостей.

Ключевые слова: насосы, напоры, система очистки, дождевая и талая вода.

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ISSN 2518-170X (Online), ISSN 2224-5278 (Print)

http://www.geolog-technical.kz/index.php/en/

Редакторы *М. С. Ахметова, Д. С. Аленов, А. Ахметова*
Верстка *Д. А. Абдрахимовой*

Подписано в печать 01.02.2021.
Формат 70x881/8. Бумага офсетная. Печать – ризограф.
12,75 п.л. Тираж 300. Заказ 1.

Национальная академия наук РК
050010, Алматы, ул. Шевченко 28, т. 272-13-19, 272-13-18