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

ҚАЗАҚСТАН РЕСПУБЛИКАСЫ
ҰЛТТЫҚ ҒЫЛЫМ АКАДЕМИЯСЫНЫҢ
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ИЗВЕСТИЯ

НАЦИОНАЛЬНОЙ АКАДЕМИИ НАУК
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SERIES
OF GEOLOGY AND TECHNICAL SCIENCES

2 (446)

MARCH – APRIL 2021

THE JOURNAL WAS FOUNDED IN 1940

PUBLISHED 6 TIMES A YEAR

ALMATY, NAS RK

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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 мекен-жайы: 050010, Алматы қ., Шевченко көш., 28, 219 бөл., тел.: 272-13-19, 272-13-18

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

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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, оф. 219, тел.: 272-13-19, 272-13-18

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

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Адрес типографии: ИП «Аруна», г. Алматы, ул. Муратбаева, 75.

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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, Almaty, 050010, tel. 272-13-19, 272-13-18,

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

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Address of printing house: ST "Aruna", 75, Muratbayev 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 2, Number 446 (2021), 198 – 204 **https://doi.org/10.32014/2021.2518-170X.53**

UDC 621. 664

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RESEARCH OF THE INTERNAL LEAKAGE PROCESS OF A LIQUID IN THE DESIGN OF A GEAR PUMP WITH A TWO-AXIAL CONNECTION

Abstract. This article presents the results of a study of the process of internal fluid leakage in the design of a gear pump with a biaxial connection. It is known that the volumetric efficiency mainly depends on the leakage of the working fluid through the gaps formed by the tooth heads and the pump casing, as well as between the end surfaces of the gears and the side walls of the casing. In addition, additional leaks occur along the contact line of the teeth. To reduce radial leaks, the gap between the gears and the pump casing is minimized, and to reduce end-leakage, the side walls are automatically pressed against the end surfaces of the gears by liquid under operating pressure. The design of the pump shows that the main internal fluid leaks occur in the gap between the gears and the housing. Naturally, these leaks require excessive energy consumption and, thereby, reduce the efficiency of the pump. The natural way to combat this circumstance is to reduce the marked gaps to a minimum. This is possible using an additional element - a biaxial sleeve. The dependence of the spring force on the fluid flow rate and the guaranteed gap between the ring gear and the pump casing has been analytically established. They are characterized by a drop in force with an increase in clearance and a rise in flow rate. This is due to an increase in the dynamic force in the gaps, which must be compensated by the spring.

Keywords: wear of rubbing surfaces, clearance, fluid leakage, biaxial connection, gear pump, eccentricity.

Introduction. The reliability and durability of hydraulic systems is largely determined by the ability of the working fluids used in them to help reduce wear on rubbing surfaces and prevent them from seizing. The wear of gear oil pumps and precision pairs of setting and control parts has a significant impact on pump performance and causes excessive power losses due to significant leaks through leaks resulting from wear [1].

The pressure developed by the pump when operating in a closed loop affects the volumetric capacity of the pump. As pressure rises, the volumetric capacity of the pump decreases. This is due to an increase in internal leakage of fluid from the discharge cavity of the pump to the suction, i.e., with a kind of slipping of the fluid. Since internal leakage is observed in all pumps, it is more convenient to judge the pump performance by the volume of liquid supplied at a given pressure per unit time. The amount of internal leakage depends on the type of pump and on the degree of wear it determines the volumetric efficiency of the pump [1].

The analysis showed that a huge number of scientific publications and works are devoted to general issues of the theory and practice of gear pumps. In particular, they are reflected in the works of Rybkin EA, Usova EA [2], Yudina E.M. [3], Osipova A.F. [4,5] and others [6,7,8,9,10]. In these works, it is noted that the main structural features of gear pumps are associated with the presence of gearing, which should meet the specifics of volumetric hydraulic machines [11,12,13,14].

Depending on the type of gearing, gear pumps can be of two types - external and internal, used in all engineering industries. The simplest type of gear pump is an external gearing device. It is a structure consisting of a housing and two gears. These wheels are engaged and are distinguished by their involute. Gear pumps and hydraulic motors due to their simple design and operational reliability are widespread in hydraulic drives of road cars [15,16]. Figure 1 shows a gear pump diagram.

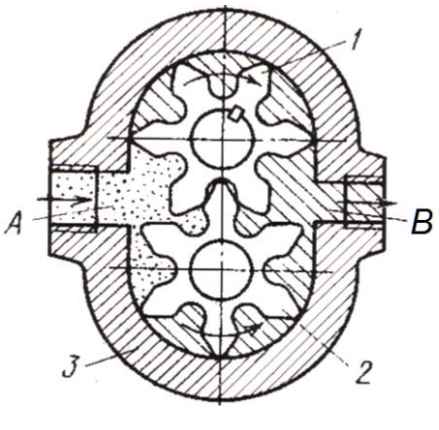
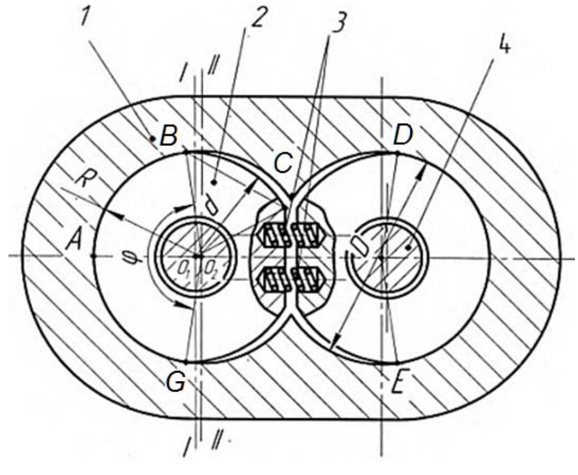


Figure 1 – Scheme of gear pump: 1, 2 – gears; 3 – case

The principle of operation of the gear pump is as follows. Two gears of equal width leading 1 and driven 2 are meshed and located in the housing 3 with a minimum radial clearance. To the end surfaces of the gears are adjacent the side walls of the pump. When the gears rotate, the fluid filling the cavities between the teeth is transferred by the gears along the inner surface of the housing (shown by arrows) from the suction cavity A to the discharge cavity B. Volumetric efficiency mainly depends on the leakage of the working fluid through the gaps formed by the tooth heads and the pump casing, as well as between the end surfaces of the gears and the side walls of the casing. In addition, additional leaks occur along the contact line of the teeth. To reduce radial leaks, the gap between the gears and the pump casing is minimized, and to reduce end-leakage, the side walls are automatically pressed against the end surfaces of the gears by liquid under operating pressure [17].

In all designs, dimensional bonds in the “shaft-bore” joint are generally considered. Such connections are the connection of the diameter of the support sleeve in the hole of the pump housing, the connection of the diameter instead of the teeth of the gear shaft in the hole of the pump housing, as well as the connection of the diameter of the shaft of the gear shaft in the hole of the support sleeve. These connections are traditional corresponding bore diameters standardized by existing standards. Structural analysis and technological requirements of the NS pumps showed that when designing the working unit of the NS pumps, it is possible to use biaxial joints [18]. Figure 2 shows the working element of the NS pump using a biaxial connection.

Figure 2 –
The working body of the NS pump using a biaxial connection:
1 – NS housing; 2 – supporting bushings; 3 – expandable elements; 4 – axis shaft - gears; D – the diameter of the hole in the housing; d – the outer diameter of the support sleeve;
R – the radius of the landing diameter of the supporting sleeve;
O_1 – diameter center of the support sleeve; O_2 – center radius of the bushing diameter of the support sleeve;
points A, B – the points of the arc of the landing diameter of the opening of the NS housing;
$\cup$ *BG* and $\cup$ *DE* – two arcs of two support bushings having a landing radius *R*: *φ* – angle covering arc $\cup$ *BG* and $\cup$ *DE*



From the design in figure 2 it follows that the supporting sleeve is pressed by the BG arc having a radius R to the opening of the NS housing. In this case, the radius R can be chosen equal to R = 0.5D. Such a choice of the value of R will allow us to combine for their average values the center of the hole in the NS housing with the center of the arc of radius R.

In [18, 19, 20, 21], dispersion fields of the diameters of the sleeve and holes for biaxial joints were shown. Depending on the task, the desired option for the mutual arrangement of tolerance fields of size R and size 0.5D can be selected.

Investigation of the process of internal fluid leakage. The design and scheme of the pump shows that the main internal fluid leaks occur in the gap between the gears and the housing. Naturally, these leaks require excessive energy consumption and, thus, reduce the efficiency of the pump [22]. The natural way to combat this circumstance is to reduce the marked gaps to a minimum. This is possible using an additional element - a biaxial sleeve. When studying the process of internal fluid leakage, the design scheme shown in figure 3 is used.

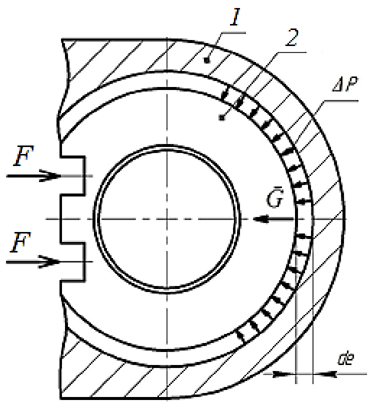


Figure 3 – Fluid leakage design scheme: 1 – pump housing; 2 – sleeve;
F – the force from the side of the compensating spring; Δp – the pressure drop during fluid leakage;
G – resultant forces acting on the sleeve

The most effective in the analysis of hydrodynamic losses during fluid motion is the Bernoulli equation [23,24]. For the case under consideration, we have:

$$\Delta P = \rho \frac{v^2}{2}$$

where ρ - the density of the liquid; υ - the fluid velocity in the gap.

With the steady laminar motion we have:

$$v = \frac{Q}{S'}$$

where *Q* – nominal fluid flow rate; S is the cross-sectional area of the gap.

Given that the force acting on the cross-sectional area of the sleeve *de* is equal to:

$$dG = \Delta p \cdot de$$

and integrating the resulting expression, we have:

$$G = \int_{S_B} \rho \frac{v^2}{2} de$$

where S_B – sectional area of the side surface of the sleeve:

$$S_B = he; e = \frac{S_B}{h}; de = \frac{dS_B}{h}$$

where *h* – sleeve surface width.

Then we get:

$$G = \rho \frac{v^2 e^2}{4} = \rho \frac{Q^2 e^2}{4S^2}$$

With a sufficiently small gap due to the eccentricity of the biaxial surface δ, an approximate value can be used:

$$S = e \cdot \delta$$

we get the dependence:

$$G = \rho \frac{Q^2}{4\delta^2}$$

The dependence of the force on the eccentricity value is obtained (see figure 4).

$$\rho = 890 \text{ kg/m}^3.$$

The formula determines the indicator G (H) for three parameters for the pumped liquid:

Q, m^3/sec	0,42 ×10^{-3} (25 l/min)			0,67 ×10^{-3} (40 l/min)			0,83 ×10^{-3} (50 l/min)		
δ, 10^{-3} м	0,4	1	2	0,4	1	2	0,4	1	2

I.

$$G_1 = \frac{890 \times 0{,}42^2 \times 10^{-6}}{4 \times 0{,}4^2 \times 10^{-6}} = 245 \text{ H}$$

$$G_2 = \frac{890 \times 0{,}42^2 \times 10^{-6}}{4 \times 1^2 \times 10^{-6}} = 39 \text{ H}$$

$$G_3 = \frac{890 \times 0{,}42^2 \times 10^{-6}}{4 \times 2^2 \times 10^{-6}} = 9{,}8 \text{ H}$$

II.

$$G_1 = \frac{890 \times 0{,}67^2 \times 10^{-6}}{4 \times 0{,}4^2 \times 10^{-6}} = 624 \text{ H}$$

$$G_2 = \frac{890 \times 0{,}67^2 \times 10^{-6}}{4 \times 1^2 \times 10^{-6}} = 99{,}8 \text{ H}$$

$$G_3 = \frac{890 \times 0{,}67^2 \times 10^{-6}}{4 \times 2^2 \times 10^{-6}} = 25 \text{ H}$$

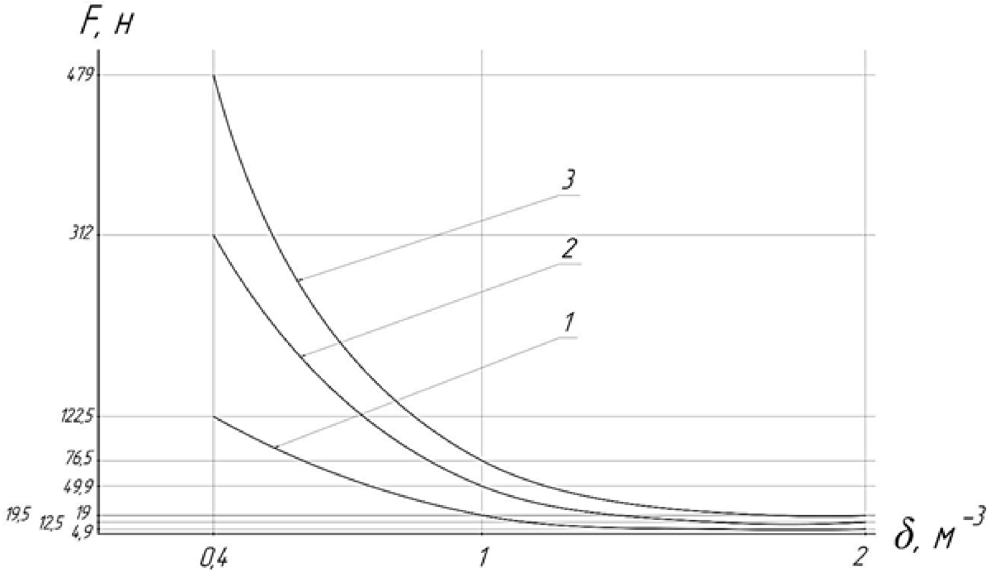
III.

$$G_1 = \frac{890 \times 0{,}83^2 \times 10^{-6}}{4 \times 0{,}4^2 \times 10^{-6}} = 958 \text{ H}$$

$$G_2 = \frac{890 \times 0{,}83^2 \times 10^{-6}}{4 \times 1^2 \times 10^{-6}} = 153 \text{ H}$$

$$G_3 = \frac{890 \times 0{,}83^2 \times 10^{-6}}{4 \times 2^2 \times 10^{-6}} = 38 \text{ H}$$

Q, m^3/sec	0,42 ×10^{-3} (25 l/min)			0,67 ×10^{-3} (40 l/min)			0,83 ×10^{-3} (50 l/min)		
δ, 10^{-3} m	0,4	1	2	0,4	1	2	0,4	1	2
G, H	245	39	9,8	624	99,8	25	958	153	38



Consumption Q: 1 – Q = 0,4 m^3/s; 2 – Q = 1 m^3/s; 3 – Q = 2 m^3/s

Figure 4 – The dependence of the force acting on the sleeve G on the eccentricity of the biaxial connection δ for various values of flow Q

To ensure guaranteed pressing of the sleeve to the surface of the housing, according to the design scheme, the condition must be provided

$$2F > G$$

Using Hooke's Law for Elastic Strength

$$F = C\lambda$$

where C – stiffness coefficient; λ is the amount of compression of the spring, we get:

$$C\lambda > \frac{G}{2}; C\lambda > \rho\frac{Q^2}{8\delta^2}$$

Conclusions:

1. The dependence of the spring force on the fluid flow rate and the guaranteed gap between the ring gear and the pump casing has been analytically established. They are characterized by a drop in force with an increase in the gap and an increase with an increase in flow. This is due to an increase in the dynamic force in the gaps, which must be compensated by the spring.

2. The above dependences allow choosing the characteristics of the spring at the pump design stage with guaranteed gaps between the ring gear and the pump casing and at specified costs.

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ЕКІ ОСЬТІ ҚОСЫЛЫСЫ БАР ТІСТЕГЕРШІКТІ СОРҒЫНЫҢ КОНСТРУКЦИЯСЫНДАҒЫ СҰЙЫҚТЫҚТЫҢ ІШКІ АҒУ ПРОЦЕСІН ЗЕРТТЕУ

Аннотация. Мақалада екі осьті қосылысы бар тістегершікті сорғының конструкциясында сұйықтықтың ішкі ағуын зерттеу нәтижелері келтіріледі.

Көлемді ПӘК негізінен тістердің бас тиектерімен және сорғы корпусымен, сондай-ақ, тістегершіктің бүйір беті мен корпустың бүйір қабырғалары арасында пайда болған саңылаулар арқылы жұмыс сұйықтығының ағып кетуіне байланысты. Сонымен қатар, тістің түйісуі бойынша ағулар қосымша пайда болады. Радиалды ағуларды азайту үшін, тістегершіктер мен сорғы корпусы арасындағы саңылауды ең аз шамада жасайды, ал бүйір қабырғалары жұмыс қысымымен тістегершіктің бүйір бетіне сұйықтықпен автоматты түрде қысылады. Сорғының дизайны сұйықтықтың негізгі ішкі ағуы редукторлар мен корпус арасындағы алшақтықта болатындығын көрсетеді. Әрине, бұл ағып кетулер шамадан тыс энергия шығынын қажет етеді және сол арқылы сорғының тиімділігін төмендетеді. Бұл жағдаймен күресудің табиғи әдісі - белгіленген олқылықтарды азайту. Бұл қосымша элементтің көмегімен мүмкін-биаксиалды жең.

Серіппенің күштің сұйықтықтың шығынына және тістің тәжі мен сорғы корпусы арасындағы кепілді саңылауға тәуелділігі аналитикалық түрде анықталды. Олар саңылау ұлғайған кезде күштің құлауымен және шығынның ұлғаюымен сипатталады. Бұл серіппемен басылатын саңылаулардағы динамикалық күштің өсуімен байланысты.

Түйін сөздер: жанасатын беттердің тозуы, саңылау, сұйықтықтың ағуы, екі осьті қосылыс, тістегершікті сорғы, эксцентриктілік.

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ИССЛЕДОВАНИЕ ПРОЦЕССА ВНУТРЕННЕЙ УТЕЧКИ ЖИДКОСТИ В КОНСТРУКЦИИ НАСОСА ШЕСТЕРЕННОГО С ДВУХОСНЫМ СОЕДИНЕНИЕМ

Аннотация. В статье представлены результаты исследования процесса внутренней утечки жидкости в конструкции шестеренчатого насоса с двухосным соединением. Известно, что объемный КПД в основном зависит от утечки рабочей жидкости через зазоры, образованные головками зубьев и корпусом насоса, а также между торцевыми поверхностями шестерен и боковыми стенками корпуса. Кроме того, по линии соприкосновения зубьев возникают дополнительные утечки. Для уменьшения радиальных утечек зазор между шестернями и корпусом насоса сводится к минимуму, а для уменьшения торцевых утечек боковые стенки автоматически прижимаются к торцевым поверхностям шестерен жидкостью под рабочим давлением. Конструкция насоса показывает, что основные внутренние утечки жидкости происходят в зазоре между шестернями и корпусом. Естественно, эти утечки требуют чрезмерного расхода энергии и тем самым снижают КПД насоса. Естественный способ борьбы с этим обстоятельством – свести отмеченные пробелы к минимуму. Это возможно с помощью дополнительного элемента – двухосной втулки.

Аналитически установлена зависимость усилия пружины от расхода жидкости и гарантированного зазора между зубчатым венцом и корпусом насоса. Они характеризуются падением силы при увеличении зазора и возрастанием при увеличении расхода. Это обусловлено возрастанием динамической силы в зазорах, которая должна компенсироваться пружиной.

Ключевые слова: износ трущихся поверхностей, зазор, утечка жидкости, двухосное соединение, насос шестеренный, эксцентриситет.

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

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

Подписано в печать 15.04.2021.
Формат 70x881/8. Бумага офсетная. Печать – ризограф.
13,0 п.л. Тираж 300. Заказ 2.

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