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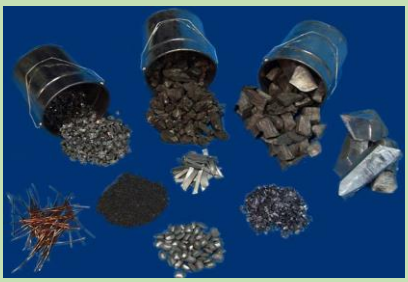







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The beautiful thing about learning is nobody can take it away from you—B. B. King

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AZERBAIJAN BAKU 2022

MÜNDƏRİCAT / TABLE OF CONTENTS

EQUIPMENT. TECHNOLOGIES. MATERIALS
VOLUME 09 ISSUE 01 2022

INTEGRATION OF AUTOMATIC BALL ADJUSTING DEVICE INTO THE GRINDING PROCESS SYSTEM OF «AZERBAIJAN INTERNATIONAL MINING COMPANY LIMITED»

DOI:

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ABSTRACT

While the technologies used in the manufacturing sector, as well as equipment and special technical means, are characterized by modernity, reliability and high productivity, they have undergone certain improvements and innovations from time to time. Meanwhile, the work done on the grinding process in SAG Mill mills installed at the AGL plant in the Kadabay contact area is of great importance.

The article presents the results of the integration of the automatic ball adjusting device into the system of SAG mills used in the grinding process.

Keywords: mining industry, grinding process, balls, wear, automatic loading, loading device.

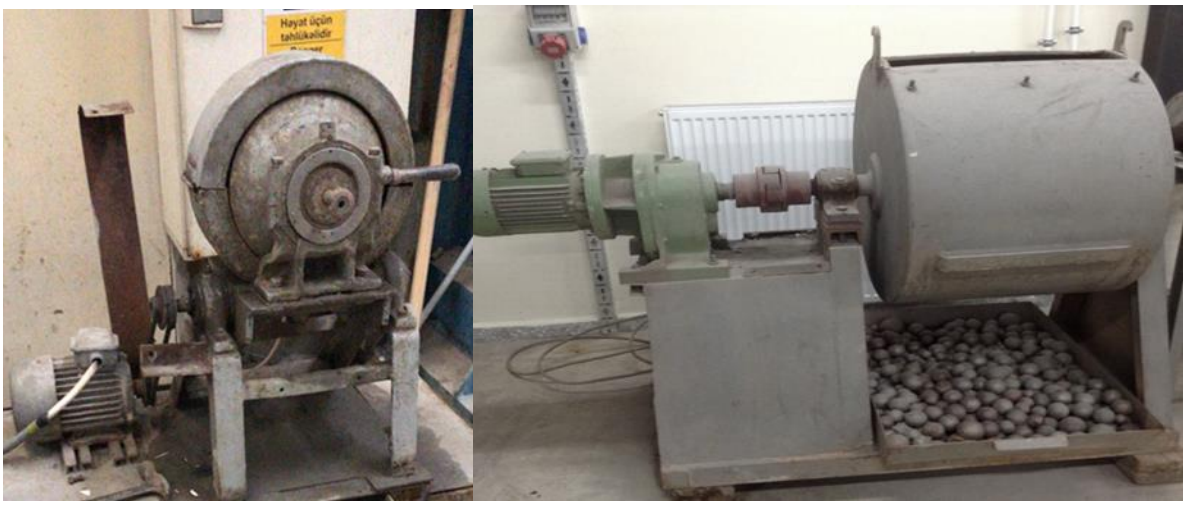
The actuality of the subject. Azerbaijan International Mining Company Limited (AIMCL) has been one of the leading companies in the non-oil sector of the country's economy since 2009 and is engaged in the exploitation of mineral resources. On the basis of the existing 30-year contract, AIMCL is located in Nakhchivan AR (Ordubad field), Tovuz (Qosha field), Gadabay (Gadabay, Ugur, Garadagh, Zafar, Kharkhar and Gadir fields), Aghdara (Gizilbulag and Demirli fields) and Zangilan fields (Vejnali) conducts mining operations.

Although the technologies used in the production areas, as well as equipment and special technical means, are distinguished by their modernity, reliability and high productivity, they have undergone certain improvements and innovations from time to time. Meanwhile, the work done on the grinding process in the SAG Mills installed at the AGL plant in the Gadabay contact area is of great importance.

As a result of our research, it was determined that in order to maintain a stable processing regime in the operation of these mills, the rate of grinding balls is increased by 1 ton every 24 hours. This led to a deterioration in the overall grinding performance of the mill after 96-120 hours of operation, resulting in a decrease in the volume of the ball due to abrasion. At the same time, the effect of variability of ore productivity on the intensity of abrasion of balls in the mill depending on the hardness of the ore supplied to the mill was determined. For these reasons, the change in the results of the grinding process of the SAG Mill had a serious impact on the quality of the output at the BALL Mill, where the next grinding process was carried out, resulting in a decline in the technical and economic performance of the final product.

The purpose of the work. Investigation of the dependence of quality indicators on the intermittent yield of balls in the grinding process in the mill type SAG Mill and the development and application of the relevant device.

Methodological base of the research. In accordance with the purpose of the study, the following parameters were comprehensively monitored and observed: SAG mill productivity, (60-110) tons / hour; balls consumption, (0.4-0.7) kg / ton; mill speed, (43.1-48.3) Hz, and % index of particles larger than 500 µm, (20-45%).
Experimental experiments were carried out at the test facilities of Gadabay plant and Hacettepe University (see Figure 1).



a) b)

Figure 1. Gadabay plant (a) test device; Hacettepe University (b) testing device.

The hardness of the ores accepted for testing varies. The hardness of the ore from the Ugur field was 3 according to the Mohs table, the hardness of the ore from the Gadabay field was 5 and the hardness of the ore from the underground Qadir field was 6-7 according to the Mohs.
Results obtained and their discussion. Figure 2 shows the dynamics of change of the studied parameters over six months.
As can be seen from the analysis of figure 2, the analysis of the grinding performance of the SAG Mill during the 160-day study revealed that when the percentage of particles larger than 500 µm (graphs 4) is more than 35%, 1 ton of ball is added to the SAG mill each time (graphs 2) was added. As a result, it temporarily increased the grinding performance.
These changes did not go unnoticed in subsequent processes. At the same time, it was found that when grinding high-strength ores in the mill, along with a decrease in productivity, there is an increase in ball consumption. Graphs 1 and 2 in figure 2 show this relationship.
The effect of ore hardness on ball consumption was carried out on a laboratory stand at the Gadabay plant and partly on the Losangels mill installed in the laboratory of Hacettepe University in Turkey. Based on the results of the experiments, it was determined that the dimensional diameters of the balls, depending on the hardness class of the ore masses extracted from different deposits from the integration into production, correspond to the graph below (see Figure 3).

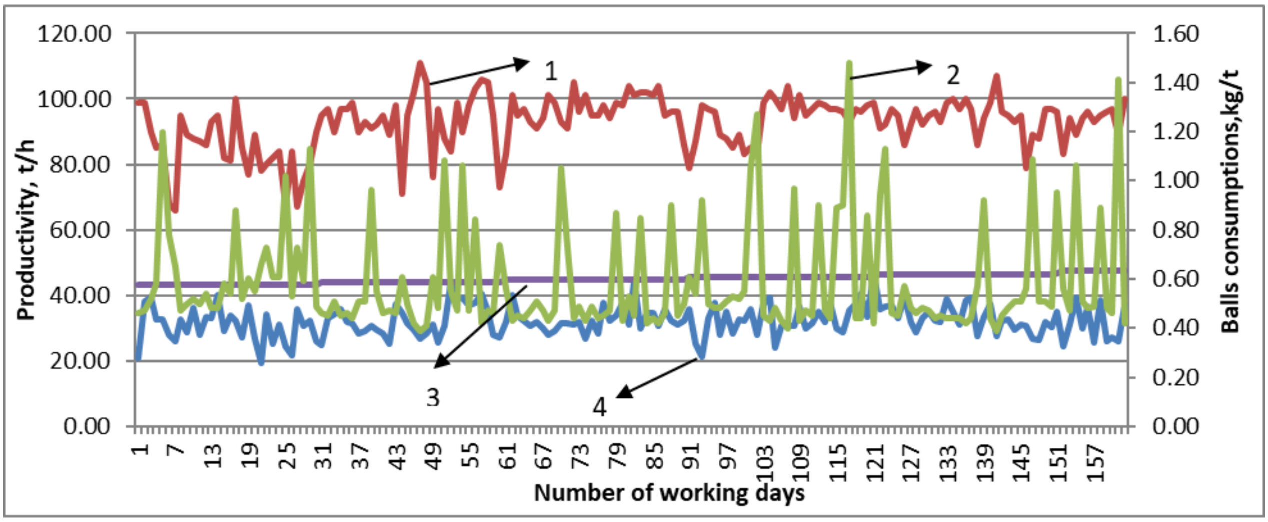


Figure 2. Mill productivity (1); ball consumption (2); mill speed (3), and the dynamics of change in the percentage of particles larger than 500 µm during operation (4).

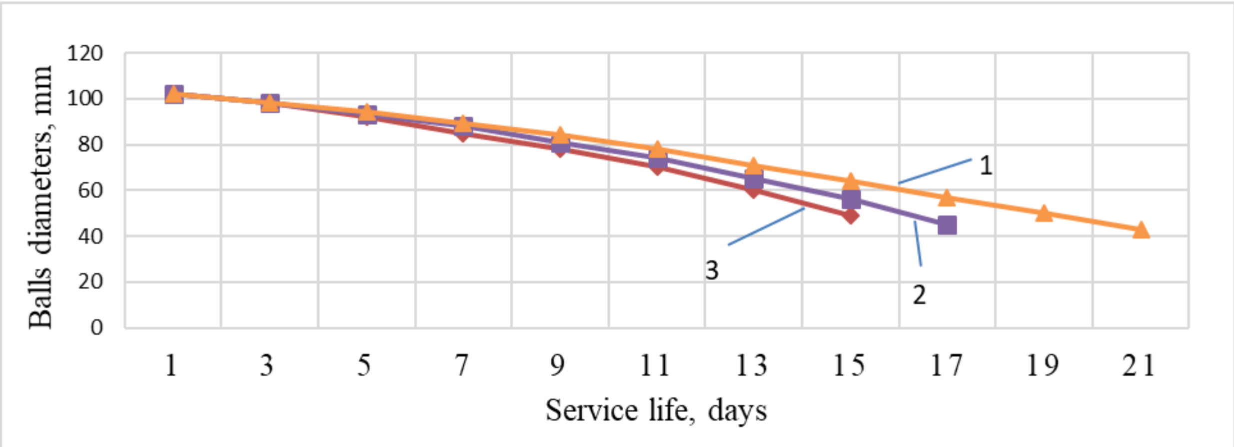


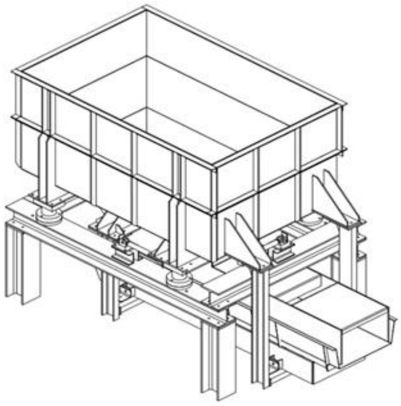
Figure 3. Taken from Ugur (1), Gadabay (2) and Gadir (3) fields dynamics of erosion of ores in the process of ore processing.

As can be seen from the analysis of figure 3, the changes in the diameters of the balls (as a result of abrasion) differ in the ores taken from different deposits. A comparative analysis of the results of laboratory and under real operating conditions tests showed that the change in the hardness of the ore between the Mohs table (3-7) is the consumption of the balls (0.48-0.6) kg/ton, respectively.

Thus, it can be concluded that the variability of the hardness of the processed ore has a significant impact on the consumption of balls in the mill. For this reason, it is very important to carry out this work continuously in order to transfer the ball inside the mill and keep the mass volume constant there. Thus, the failure to keep the volume of the ball inside the mill stable creates a serious basis for the optimization of quality and productivity processes in the next grinding stage BALL Mill. Since the grinding process at the BALL Mill is closed, the solution to the problem requires the SAG Mill.

Taking into account the above, the issue of establishing and applying a system for automatic regulation of ball yield to keep the volume of the ball constant, regardless of the hardness of the ore mined at the SAG Mill, was set and resolved. The function of the proposed automatic system was to periodically ball into the mill at appropriate intervals.

Auto Ball Loading Equation (ABLE) was developed by ATLAS CRUSHER of the Islamic Republic of Iran based on the initial conditions and constructive design set by us (see figure 4).



a)

b)

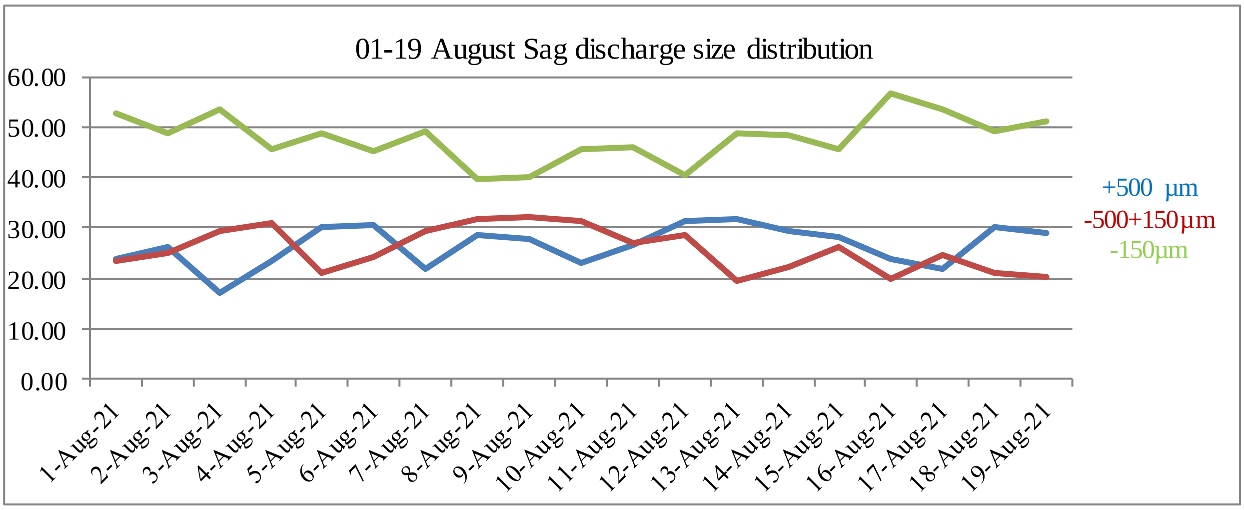
Figure 4. Scheme of automatic ball adjustment equipment (a) and its location in the AIMCL (b).

The equipment was adapted to the process in July-August 2021, and was commissioned in the second half of August. After commissioning, the mill periodically measured the ball volume change every 168 hours and compared the grinding parameters before installing the automatic ball adjustment equipment.

Figure 5 shows the results of the comparison. As can be seen from the graphs, after the automatic ball adjustment device was put into operation, the volume of the ball inside the SAG mill was stabilized and as a result, the grinding parameters were stabilized.

Thus, the final results of the research can be formulated as follows.

- The quality of the product obtained at the SAG Mills used in the grinding of ores in the mining industry is strongly dependent on the loss of mass due to the consumption of balls used there.
- In order to maintain the stability of the balls consumption within the SAG mills, the loading mode once a day cannot be considered satisfactory.
- It is expedient to transfer the supply of balls to the mills without interruption.
- The use of automatic ball adjustment equipment is an important factor in stabilizing the ball volume and increasing the grinding performance inside the SAG Mill.
- Also, supplying balls to the SAG Mill without interruption with Auto Ball Loading Equipment contributes greatly to minimizing the human factor and risks during the performance of work.



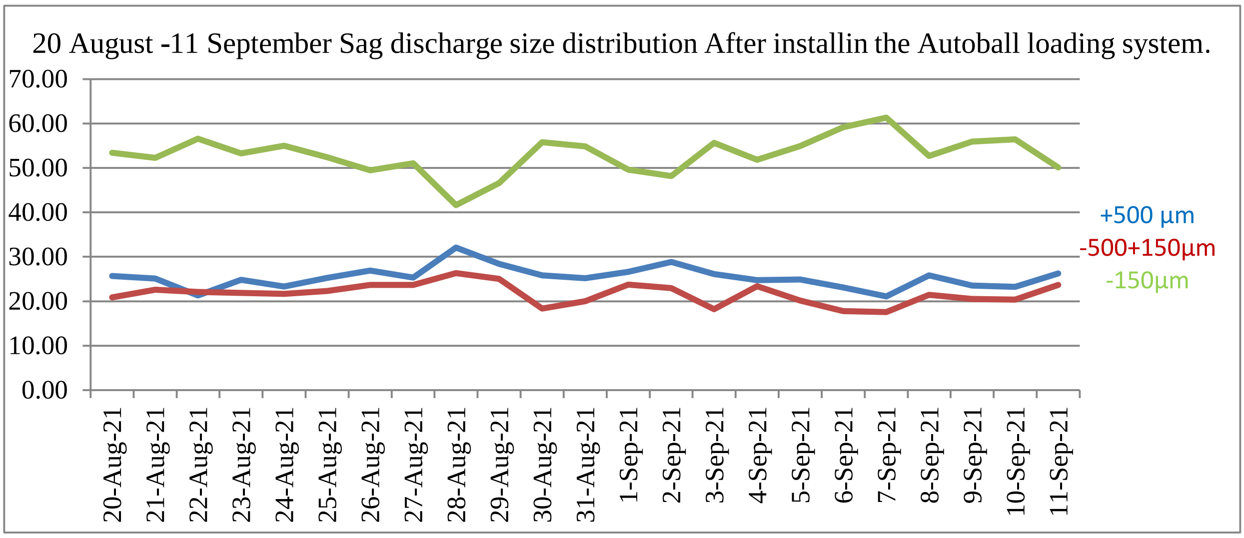


Figure 5. Comparison of grinding results.

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УСТРОЙСТВА ДЛЯ АВТОМАТИЧЕСКАНОЙ ПОДАЧИ МЕЛЮЩИХ ШАРОВ В МЕЛЬНИЦУ КОМПАНИИ “AZERBAIJAN INTERNATIONAL MINING COMPANY LIMITED”

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РЕЗЮМЕ

Хотя технологии, применяемые в производственной сфере, а также оборудование и специальные технические средства характеризуются современностью, надежностью и высокой производительностью, время от времени они подвергались определенным усовершенствованиям и новшествам. Между тем работа, проделанная по процессу измельчения в мельницах SAG Mill, установленных на заводе AGL в Гедабекском контактном районе, имеет большое значение.

В статье представлены результаты интеграции устройства автоматической загрузки мелющих шаров в систему мельниц SAG Mill, используемых в процессе измельчения.

Ключевые слова: горнодобывающая промышленность, процесс измельчения, шары, износ, автоматическая загрузка, загрузочное устройство.

“AZERBAIJAN INTERNATIONAL MINING COMPANY LIMITED” ŞİRKƏTİNDƏKİ ÜYÜDMƏ PROSESİ SİSTEMİNƏ AVTOMATİK KÜRƏ YÜKLƏMƏ QURĞUNUN İNTEQRASIYASI

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XÜLASƏ

İstehsal sahərərində tətbiq olunan texnologiyalar, eləcə də avadanlıqlar və xüsusi texniki vasitələr müasirliyi, etibarlığı və yüksək məhsuldarlıqları ilə seçilsələrdə vaxtaşırı müəyyən təklilləsdirlmələr və innovasion xarakterli mükəmmələşmələrə məruz qalmışdırlar. Bu sırada Gədəbəy kontakt sahəsindəki AGL zavodunda quraşdırılmış SAG Mill tipli dəyirmanlardakı üyütmə prosesi ilə bağlı görülmüş işlər böyük aktuallıq kəsb edir.

Məqalədə üyütmə prosesində tətbiq olunan SAG dəyirmanlarında avtomatik kürə tənzimləmə qurğusunun sistemə inteqrasiyasının nəticələri əks olunmuşdur.

Açar sözlər: dağ-mədən sənayesi, üyütmə prosesi, kürələr, yeyilmə, avtomatik yüklənmə, yükləmə qurğusu.

EQUIPMENT. TECHNOLOGIES. MATERIALS

VOLUME 09 ISSUE 01 2022

RESEARCH STABILITY OF MAST ROPE-PIN DRIVE OF DOWNHOLE SUCKER-ROD PUMP

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ABSTRACT

This scientific article is about the research stability of mast rope-pin drive of downhole sucker-rod pump. The design of downhole sucker-rod pump drive in the form of a mast, which is mounted directly on the column of the wellhead has been proposed. Calculations of the frequency of natural vibrations of the system from the rotation of the cranks during the breakage of the rods are performed, which completely eliminates the possibility of resonant phenomena that destroy the mast. This proves that the mast drive has a high stability margin.

Key words: downhole sucker-rod pump drive, mast, long stroke drive.

Topicality of the research: In view of the fact that in conditions of periodically flooded or hard-to-reach areas it is often not possible to build a reliable foundation for equipment, was proposed design of the drive downhole sucker-rod pump (DSRP) as a mast, which is installed directly on the column of the wellhead. The model of this construction was exhibited at All-Union Exhibition of National Economy (AUENE) in Moscow.

Methods: This design lacks such metal-intensive units as a balancer with a head, a frame, a mast, and application of a high mast, excludes flooding of the electric motor during floods, and also provides the maximum stroke length of the rod hanger point (BSP), i.e. $S_{max} = 2r$, where r - is a radius of rotation of the rod-rope connection point with cranks (figure 1) [1]. However, due to the limitation of the possibility of increasing the total length of the crank, it is undesirable to increase the radius of the crank more than 1.5 m, i.e. the maximum stroke length must not exceed 3 m.

In order to increase the stroke length of RHP, the design shown in figure 2 was proposed, in which the traverse is replaced by a central pulley mounted on the bottom flange of the mast on the same axis as the side pulleys. A rope is attached to this pulley, which in turn is connected to a rope hanger with a polished rod connected to a column of pump rods through a polished rod. This design can already be used for a long stroke DSRP drive, because here the diameter of the central pulley is selected in comparison with the diameters of the lateral pulleys as large as necessary to increase the stroke length of RHP. The fact that, according to the standard for downhole sucker-rod pumps, the maximum stroke length of the plunger reaches 6 m, there is also a possibility of creating a long stroke drive with the stroke length of RHP up to 6 m.

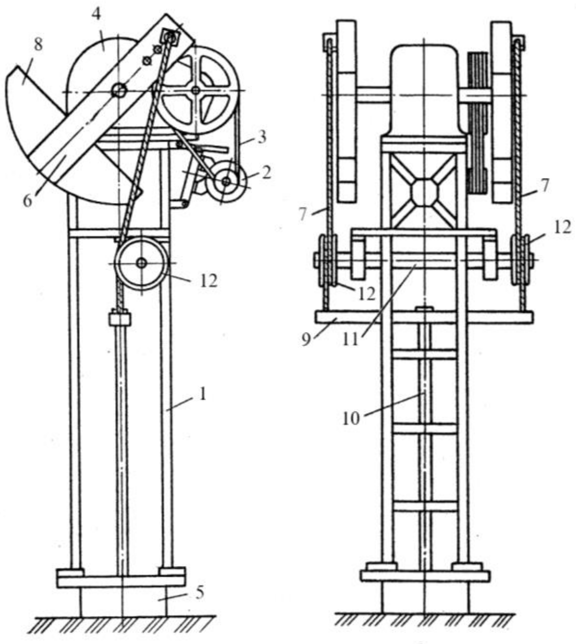

Figure 1. Schematic diagram of mast-type construction (AS 802608): 1 - mast; 2 - electric motor; 3 - V-belt drive; 4 - gearbox; 5 - wellhead string; 6 - cranks; 7 - connecting rope; 8 - counterweights; 9 - traverse; 10 - polished rod; 11 - axle; 12 – pulleys.

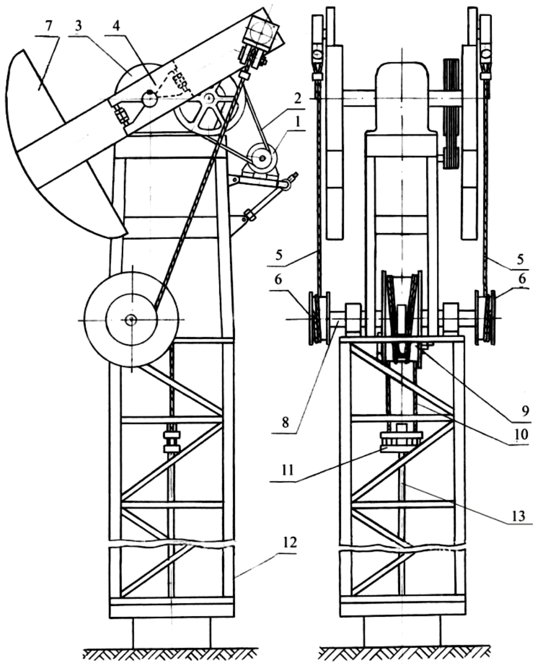

Figure 2. Schematic diagram of the long stroke drive of the mast-type DSRP: 1 - electric motor; 2 - V-belt transmission; 3 - gearbox; 4 - cranks; 5 - rope-crank; 6 - side pulleys; 7 -

counterweights; 8 - axle; 9 - central pulley; 10 - rope; 11 - rope suspension; 12 - mast; 13 - polished rod.

To make strength calculations of a mast drive, it is necessary to have a picture of the drive force loading during the cycle of its operation. The force factors, in turn, are largely determined by the kinematic characteristics of the actuator. During the development of DSRP drives, as a rule, a number of standard sizes of equipment is created, similar in geometrical parameters and differing in the load acting on RHP and other parameters. Therefore, when deriving formulas for determining the geometrical characteristics of the drive, the ratio of the pole distance length h to the crank radius value r (figure 3) is taken as a dimensionless constant, which serves as a criterion of geometrical similarity for this drive design:

$$\mu = \frac{h}{r} \tag{1}$$

To determine the kinematic parameters, the similarity criteria are:

$$\frac{S}{S_o}, \quad \frac{V}{\omega S_o}, \quad \frac{W}{\omega^2 S_o} \tag{2}$$

Here, S_o – is the value of the maximum stroke of the boom hanger point, $S_o = 2r$; ω – angular velocity of the crank rotation; S, V, W – are the current values of the stroke, velocity and acceleration of the boom hanger point.

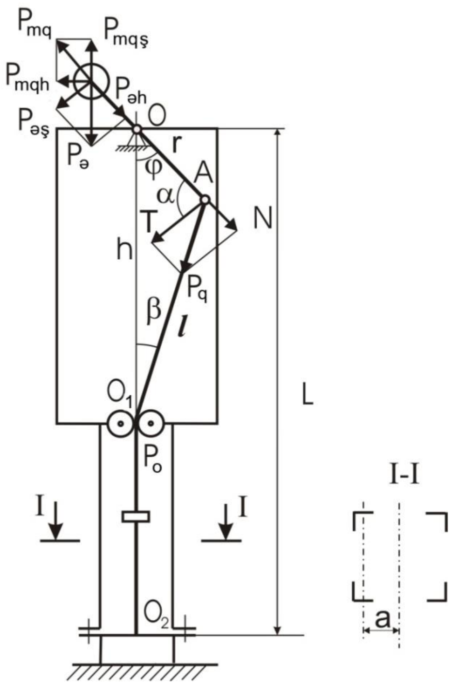

Figure 3. Kinematic diagram of a mast-type DSRP drive.

Using the kinematic diagram of the mast drive DSRP, write down the following dependencies:

$$\begin{cases} l\sin\beta = r\sin\varphi \\ r\cos\varphi + l\cos\beta = h \end{cases} \tag{3}$$

Here l – is the current length of the flexible connecting rod of variable length; β –angle between the flexible connecting rod and the vertical; φ – angle of crank rotation.
From this we define expressions for the current length of the flexible connecting rod of variable length and the angle between it and the vertical:

$$l = r\frac{\sin\varphi}{\sin\beta} \tag{4}$$

$$\cos\beta = \frac{h - r\cos\varphi}{l} = \frac{\frac{h}{r} - \cos\varphi}{l} = \frac{\mu - \cos\varphi}{l} \tag{5}$$

$$\beta = \arccos\frac{\mu - \cos\varphi}{l} \tag{6}$$

Considering that the minimum length of the flexible connecting rod is equal to $l_{min} = h - r$ and expressing the value of the crank radius through the maximum stroke using the criterion of geometric similarity, we obtain an expression for determining the movement of the boom suspension point depending on the crank rotation angle:

$$S = l - l_{min} = \frac{S_o\sin\varphi}{2\sin\beta} - h + \frac{S_o}{2} = \frac{S_o\sin\varphi}{2\sin\beta} - \frac{S_o}{2}(\mu - 1) = \frac{S_o}{2}\left(\frac{\sin\varphi}{\sin\beta} - \mu + 1\right) \tag{7}$$

The resulting expression allows us to write the first kinematic similarity criterion in the following form:

$$\frac{S}{S_o} = \frac{1}{2}\left(\frac{\sin\varphi}{\sin\beta} - \mu + 1\right) \tag{8}$$

When the crank rotates uniformly, the velocity and acceleration of the boom hanger point can be expressed as derivatives of the crank angle, i.e:

$$V = \omega\frac{dS}{d\varphi} \quad \text{and} \quad W = \omega^2\frac{dV}{d\varphi} \tag{9}$$

By differentiating the expression for the first kinematic criterion by the angle, we obtain an expression for the second (velocity) kinematic criterion:

$$\frac{V}{\omega S_o} = \frac{\cos\varphi - \frac{d\beta}{d\varphi}(\mu - \cos\varphi)}{2\sin\varphi} \quad \text{where} \quad \frac{d\beta}{d\varphi} = \frac{\mu\cos\varphi - 1}{1 + \mu^2 - 2\mu\cos\varphi} \tag{10}$$

For the third kinematic similarity criterion related to the acceleration of the boom point, differentiating the second kinematic similarity criterion by the angle of rotation of the crank, we obtain:

$$\frac{W}{\omega^2 S_o} = \frac{\left(1+\frac{d\beta}{d\varphi}\right)\sin\varphi + (\mu-\cos\varphi)\frac{d^2\beta}{d\varphi^2} + \left[\cos\varphi - (\mu-\cos\varphi)\frac{d\beta}{d\varphi}\right]\frac{d\beta}{d\varphi}\mathrm{ctg}\beta}{2\sin\beta} \tag{11}$$

where the second derivative of the angle β at φ:

$$\frac{d^2\beta}{d\varphi^2} = \frac{\mu(1-\mu^2)\sin\varphi}{(\mu^2 - 2\mu\cos\varphi 1)^2} \tag{12}$$

A program for calculating kinematic criteria at different values of geometric criterion was made and calculations were performed, the results of which were plotted in graphs (figure 4).

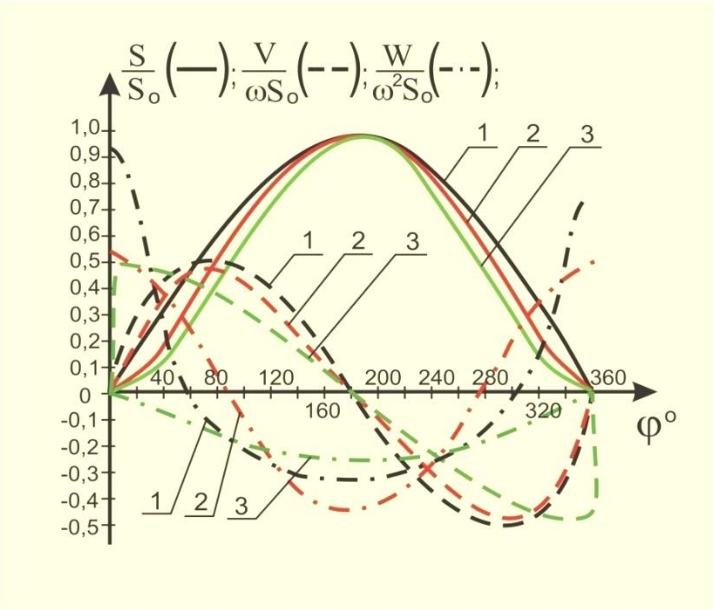

Figure 4. Diagram of variations of stroke length, speed and acceleration of the BSP of DSRP drive of mast type: 1 - r/h = 0,1; 2 - r/h = 0,5; 3 - r/h = 1.

To calculate the forces of the DSRP mast drive nodes, a calculation dynamogram is built using the method generally accepted for DSRP drives. The dynamogram determines the traction force P_o acting on the rope-crank in any position of the mechanism and allows you to set the value of counterweights P_c weight. Then the torque on the reducer shaft will be equal:

$$M_{rm} = T \cdot r - P_{hc} \cdot R = P_o \cdot r \cdot \sin(\varphi + \beta) - R \cdot P_c \sin\varphi \tag{13}$$

where P_{hc} – is the horizontal component of the force from the weight of the counterweights; R – is the radius of rotation of the center of mass of the counterweights.
The tractive force P_o and cyclic characteristics of the system make it possible to determine the necessary rope and sheave diameters, perform strength calculations of sheave axles and

bearings, cranks, gearbox shafts, metal construction, fasteners, etc. All these calculations are performed as part of general engineering design work.
At the same time, in comparison with other mechanical drives, the mast drive has a significant difference in the calculation plan, caused by the presence of large rotating masses, taken out on a comparatively large height.
During normal operation of the mast drive, the effect of these masses on stability is compensated by the traction force. However, in an emergency case, namely when the rope breaks, the horizontal component F_{hc} of the centrifugal force F_c from unbalanced rotating masses of counterweights leads to lateral swaying of the mast with the gearbox, electric motor, pulleys and other massive elements located on the top of the mast. Therefore, it is necessary to calculate the danger of resonance phenomena in the system, as well as the possibility of large deflections of the top of the mast from the vertical.
The horizontal component of centrifugal forces from unbalanced masses of rotating counterweights is equal:

$$F_{hc} = m_o\omega^2 R\sin\varphi \tag{14}$$

where m_o – is the total mass of the counterweights.

Then the differential equation of motion of the heavy mast top is written in the form:

$$m\frac{d^2x}{dr^2} + Cx = m_o\omega^2 R\, sin\varphi \tag{15}$$

Here x – is the deviation of the heavy mast top from the vertical; m – is the mass of the heavy mast top; C is the bending stiffness of the mast.

$$C = \frac{3EJ_x}{L^3} \tag{16}$$

where E – is the modulus of elasticity of the mast material; L – is the mast height; J_x – is the moment of inertia of the mast cross-section.
If the mast consists of 4 legs that are identical in cross-section, then its moment of inertia, reduced to the mast's median axis, is equal:

$$J_x = 4(J_{x1} + a^2F) \tag{17}$$

Here J_{x1} – is the moment of inertia of the cross section of the mast leg relative to its own axis; a – is the distance between the axes of the legs and the mast; F– is the cross-sectional area of the mast leg.
Resonance in the system comes in the case of equilibrium of frequencies of natural and forced vibrations.

Conclusions: As an example we considered a mast drive DSRP with the following parameters: L = 9 m, R = 1,23 m, a = 0,5 m. Bearing legs of the mast are provided from equal-sided angular rolled steel №14. The mast drive design corresponds to the parameters of a rocking-type machine SKD8-3-4000, the maximum stroke length of the point of suspension rods 3 meters, the mass of the mechanisms that make up the weighted top of the mast,

approximately equal to 15970 kg, and the mass of counterweights is determined by eight weights of 650 kg each. The maximum crank speed is equal to 12 revolutions per minute.
According to the given formulas our calculations showed that the frequency of natural vibrations of the system is 11.7 s^{-1}, which differs greatly from the frequency of forced vibrations and it excludes the possibility of the resonance events which are destroying the mast. The maximum deflection of the weighted top of the mast does not exceed 1 sm, i.e. mast drive DSRP has a high margin of stability.
The only disadvantage of mast-type DSRP drive, installed directly at the wellhead, is the difficulty of clearing the wellhead and the space above the wellhead required for the current well repair. However, when using the hoisting unit UPT-32, mounted on a caterpillar tractor and having a tower height of 18 m, it is quite possible, after disconnecting the drive from the wellhead, to install it away from the wellhead, as the total height of the mast-type drive at a maximum stroke length of 6 m, does not exceed 12 m. I suggest for reliable stability of the drive when it is installed on the ground to provide an additional frame at the base of the mast, increasing the area of contact with the ground surface.

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ИССЛЕДОВАНИЕ УСТОЙЧИВОСТИ МАЧТОВОГО КАНАТНО-ШКИВНОГО ПРИВОДА СКВАЖИННОГО ШТАНГОВОГО НАСОСА

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АННОТАЦИЯ

Данная научная статья об исследовании устойчивости мачтового канатно-шкивного привода скважинного штангового насоса. Была предложена конструкция привода скважинного штангового насоса в виде мачты, которая устанавливается непосредственно на колонне устья скважины. Выполнены расчеты частоты собственных колебаний системы от вращения кривошипов при обрыве штанг, которые полностью исключает возможность возникновения резонансных явлений, разрушающих мачту. Это доказывает, что мачтовый привод обладает высоким запасом устойчивости.
Ключевые слова: привод скважинного штангового насоса, мачта, длинноходовой привод.

DOR ŞƏKLİNDƏ QUYU ŞTANQLI NASOSUNUN DAYANIQLIĞININ TƏDQİQATI

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XÜLASƏ

Bu elmi məqalə dor şəklində olan quyu ştanqlı nasosunun dayanıqlığının tədqiqatı haqqındadır. Birbaşa quyu ağzının sütununda quraşdırılmış dor şəkilli quyu ştanqlı nasosun intiqalı üçün konstruksiya təklif edilmişdir. Nasos ştanqların qırılması zamanı çarxqolların fırlanmasından sistemin təbii vibrasiya tezliyinin hesablamaları aparılır ki, bu da doru məhv edən rezonans hadisələrinin ehtimalını tamamilə aradan qaldırır. Bu, dor tipli intiqalın yüksək dayanıqlığına malik olduğunu sübut edir.
Açar sözlər: quyu ştanqlı nasosunun intiqalı, dor, uzungedişli intiqal.

EQUIPMENT. TECHNOLOGIES. MATERIALS

VOLUME 09 ISSUE 01 2022

THE ACTION OF STRESSES AND DYNAMIC LOADS ON RODS FROM FIBERGLASS, WHEN CATCHING THEM BY THE BODY, WITH A ROD CATCHER

DOI:

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ABSTRACT

The article discusses the process of gripping the working surface of the spiral, the parts of the rod catcher by the outer surface of the body of fiberglass rods. A diagram of this capture is shown. The diagram shows the forces acting on the gripping surface, between the fiberglass rods and the inner surface of the helix. The greater roughness of the outer surface of the fiberglass rods contributes to their more efficient grip. The drawbacks are indicated in the capture and release of fiberglass rods, which, when designing these rods, are taken into account in the preparation of technical specifications.

Keywords: fiberglass rods, deep pumps, tool, work piece.

Introduction: In order to save metal, in recent years, fiberglass rods have been used instead of steel rods in field deep-well sucker-rod pumping units. These rods are recommended for serial production.

Under difficult conditions, the robots of the rods of deep-well pumping units in oil production wells, the probability of accidents with fiberglass rods is assumed. For this purpose, it is necessary to take into account the reliability of the grip of the rod catcher for the outer surface of the fiberglass rods, when eliminating accidents with these rods. As a result, it is necessary to consider the loads and stresses that arise in the process of gripping between the inner working surface of the part (spiral, collet) of the rod catcher and the outer surface of the fiberglass rod body.

Purpose of the study. Reliability of gripping with a rod catcher, provided for catching fiberglass rods during the elimination of an accident with deep well pumping units.

In the operation of oil fields, downhole pumps, in addition to conventional metal steel rods, are also used fiberglass rods. In terms of physical and mechanical properties, fiberglass rods are sharply different from metal ones. Operation of deep pumps with fiberglass rods in production wells for a certain period of time leads to their aging and mechanical wear. When the fiberglass rods wear out, the possibility of their breakage is possible, as a result of which an accident occurs associated with the abandonment of the deep pump in the well.

Mechanical wear occurs due to loads and stresses arising from the movement of the rods back and forth in a vertical position. The reciprocating movement of the rods leads to the occurrence of tensile and compressive loads and stresses, as a result of which fatigue stresses in the body of the rods. When the rods move up, tensile forces and stresses act. Downward movement of the bar exposes it to compressive forces and stresses.

It is also necessary to take into account when operating fiberglass rods, their mechanical characteristics. Fiberglass rods, submersible pumps, for operation in the conditions of oil wells, are accordingly calculated and selected according to their characteristics. The brand of fiberglass material of the rods is also selected.

During the operation of rods made of fiberglass material of the corresponding brand, they must have durability, which depends on the strength of the material of this brand.

The action of tensile and compressive loads on the fiberglass rods, respectively, occurs parallel to the fiber material of the rod body.

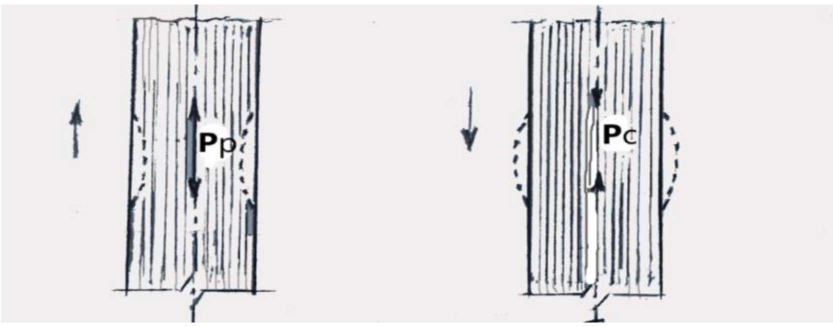

Figure 1. Fiberglass rod: a) action of tensile load Pp; b) the action of the compressive load Pc.

Figure 1 shows a part of the body of a fiberglass rod, which is deformed under the action of tensile loads Pp (a) and compressive loads Pc (b). Normal tensile and compressive stresses σp, σc will also act.

Let us consider the scheme of interaction of the working part of the gripping mechanism of the rod catcher, the spiral i.e. its internal grooved toothed image surface of the body of the fiberglass rod, shown in figure 2.

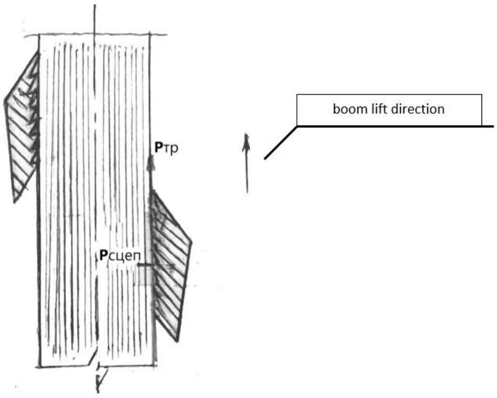

Figure 2. Scheme of the capture of the working surface of the part of the spiral, for the outer surface of the body of the fiberglass tissue.

As can be seen in figure 2, the grip of the fiberglass rod by the body occurs with the help of teeth (notches) on the inner surface of the spiral. In the process of gripping, the teeth of the spiral, overcoming the elastic deformation of the smooth surface of the fiberglass rod, by the adhesion force Pcsep., As shown in figure 2. As a result, friction force Ptr acts between the spiral teeth and the surface of the fiberglass rod, figure 2. The gripping efficiency depends on the specific roughness of the fiberglass rod, which is determined by the rod material itself and its operating conditions. That is, the more rough the surface of the fiberglass rods, the more reliable the grip on the rod body. It is also necessary to take into account, when designing a rod catcher, that is, its gripping part of the spiral, its inner surface diameter, which is calculated according to the outer diameter of the fiberglass rod body, with a corresponding interference fit. Corrosion resistance of fiberglass rods also contributes to better liquidation of an accident with deep pumps in the well.

After gripping the body of the fiberglass rods with a rod catcher, lift these rods and release their rod catcher. The release takes place according to the mechanism of this catching tool. Unlike metal rods, the process of freeing the fiberglass rods from the grip is easier due to the fiberglass material itself and the relatively smooth gripping surface of the rod.

The disadvantage when gripping and releasing fiberglass rods is their low strength, as well as their smoother surface, less amenable to effective gripping. Therefore, in the design and manufacture of fiberglass rods, it is necessary to draw up technical conditions that take into account these shortcomings and carry out quality control and testing of fiberglass rods.

Conclusion: The use in deep-well sucker rod borehole installations, instead of metal steel rods, fiberglass rods, has certain advantages, such as metal savings, anti-corrosion of fiberglass rods, their low weight, etc. The disadvantage of fiberglass rods is their relative low strength compared to steel rods. This leads to accidents with deep pumping units in oil production wells. When eliminating an accident with the help of a rod catcher tool, specifically for the reliability of gripping by the rod body, the inner working surface of the gripping element, i.e. spirals or collets, it is necessary to take into account the forces and stresses acting at the site of the cantact of the inner working surface of the gripping element of the head catcher, with the outer surface of the body of fiberglass rods.

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ДЕЙСТВИЕ НАПРЯЖЕНИЙ И ДИНАМИЧЕСКИХ НАГРУЗОК НА ШТАНГИ ИЗ СТЕКЛОВОЛОКНА ПРИ ЛОВЛИ ИХ ЗА ТЕЛО ШТАНГОЛОВИТЕЛЕМ

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АННОТАЦИЯ

В статье рассматривается процесс захвата рабочей поверхностью спирали детали штанголовителя за внешнюю поверхность тела стекловолокнистых штанг. Приведена схема этого захвата. На схеме указаны силы, действующие на поверхности захвата между стекловолокнистыми штангами и внутренней поверхностью спирали. Большая шероховатость внешней поверхности стекловолокнистых штанг, способствует более эффективному их захвату. Указываются недостатки при захвате и освобождении стекловолокнистых штанг, которые при конструировании этих штанг учитываются в составлении технических условий.
Ключевые слова: стекловолокнистые штанги, глубинные насосы, инструмент, рабочая деталь.

ŞÜŞƏ LİFLİ ŞTANQLARIN GÖVDƏDƏN ŞTANQTUTUCU ALƏTİ İLƏ TUTMA ZAMANI, GƏRGİNLİKLƏRİN VƏ DİNAMİK QÜVVƏLƏRİN TƏSİRİ

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XÜLASƏ

Məqalədə şüşə lifli ştanqların gövdəsinin xarici səthi üçün ştanqtutucu hissəsinin spiralının işçi səthinin tutulması prosesi müzakirə olunur. Bu tutulmanın sxemi verilmişdir. Sxemdə tutma səthində, şüşə lifli ştanqları və spiralın daxili səthi arasında hərəkət edən qüvvələri göstərir. Şüşə lifli ştanqların xarici səthinin böyük kələ-kötürlyü olması onların daha səmərəli tutmasına kömək edir. Şüşə lifli ştanqların tutulması və buraxılmasında olan çatışmamazlıqlar göstərilir, bu ştanqların tərtib edərkən, texniki şərtlərdə hazırlanmasında nəzərə alınır.
Açar sözlər: şüşə lifli, ştanglar, dərinlik nasosları, alət, işçi detal.

EQUIPMENT. TECHNOLOGIES. MATERIALS

VOLUME 09 ISSUE 01 2022

ANALYTICAL STUDY OF HYDRAULIC PRESSURE CAUSED BY LIQUID IN GATE VALVES

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ABSTRACT

There is an urgent need worldwide for efficient fluid control equipment. Some of the existing equipment has certain limitations for efficient fluid control under high pressure conditions. Thus, design modifications are required to solve this problem. This article presented a analytical method for designing and performing stress analysis for a high pressure valve used on a typical oil and gas wellhead with operating pressures up to 100 MPa. The most important components of a valve are the body and the valve (disc). The conversion design uses analytical method calculations to determine the stress and strain distribution in the critical components of a high pressure valve. Results from both analytical computational design and finite element analysis indicate reasonable agreement by measuring percent performance variance. Thus, the results of this research work confirm the reliability of the designed valves in accordance with engineering design estimates and, in turn, can be optimally improved in the manufacture of high pressure valves.

Keywords: gate valve, analytical method, validation, design and high pressure.

Introduction: Valves are static mechanical equipment that controls the flow and pressure within a system or process. They are essential components of a piping system that conveys liquids, gases, vapors, slurries etc. [1]. Heavy duty gate valves are normally used in oil wellhead configuration and installation. Because of its capability to control and withstand crude oil high flow rate and high pressure that is constantly experienced on the oil wellhead. Typical wellhead operating parameters includes: high pressure, flow rate, specific gravity, etc. This research work utilized computer aided design method to design and carryout critical stress analysis of a gate valve used on a typical oil and gas wellhead of working pressure up to 100 MPa. A Gate valve can be described as a type of valve that opens by raising a round or rectangular gate/wedge out of the path of the fluid and is operated by means of a threaded stem which connects the actuator (hand-wheel or motor) to the gate [1]. These valves are used for regulating flow, but many are not suited for this purpose, having been designed to be fully opened or closed [1]. Thus, when fully open, the typical gate valve has no obstruction in the flow path, resulting in a very low frictional loss [1] but the opposite become the case when it is fully closed. Hence, high frictional losses occur on the gate (the closing element of the valve) as a result of many obstructions in the fluid flow path. It is important to avoid or minimize the frictional losses at the design stage of the gate valve; therefore, the critical stress analysis of key elements of the gate valve is essentially identified in advance, before manufacturing of the gate valve. Hence, this study is bridging the gap between indigenous technology/engineering and acceptable international design methodology.

Although, it is a serious task for designers to design and estimate accurate stress distributions of any mechanical component; some limitations are encountered in the design. However, to overcome these limitations, computer aided design (CAD) methodology is an alternative. This method enables optimization of the design at the design stage, provides significant results prediction of stress distribution, and minimizes design time and cost, as well as providing efficient product.

Valves are designed with global codes and standards, for example the American Petroleum Institute Standards (API), American society of mechanical engineers (ASME), British standard (BS), America society of testing and materials (ASTM) and American National Standards Institute (ANSI) Deutsches Institut für Normung (DIN) etc. Thus, in order to achieve cost effective design, less design time and to produce accurate design performance outcome, CAD tools such as finite element analysis (FEA) is utilized extensively to analyze stress behaviors of the critical valve elements such as the gate, stem, valve body and bonnet. It helps in identifying any possible failures that may occur during operational life of the valve.

Proficient use of CAD tools will help increase the accuracy of the proposed locally design gate valve in geometry and analysis as well as design time reduction. Though, different size and pressure range of gate valves are found in the market. But constraints such as cost of importation to the local end users, failures in existing design due to old methodologies in design processes such as assumptions made in analytical calculations, manual drafting usually leads to design errors. For that reason, this research work aimed to address aforementioned constraints by proposing an effective, efficient and low-cost design for indigenous manufacturers and end users.

The materials used in this study consist of two components i.e. hardware and software components.

The hardware component involves a high speed performance computer while the software component involves a solid-work CAD tool. Analytical method was used to perform hand calculations for the design in term of sizes, dimensions and critical stresses that would act on the gate valve body, stem, gate (disc), bonnet and flanges. Computer simulation was also conducted to check for convergence of results between the hand design calculations and FEA simulation for the purpose of validating the gate valve optimal design credibility.

In a typical gate valve, the critical parts that experiences pressure of the fluid directly are the body, gate, stem and bonnet. These parts are described briefly with their design principles.

The body, sometimes called the shell, is the primary pressure boundary of a valve. It serves as the the body, sometimes called the shell, is the primary pressure boundary of a valve. It serves as the Narrowing of the fluid passage (Venturi effect) is also a common method for reducing the overall size and cost of a valve. In other instances, large ends are added to the valve for connection into a larger line [5]. When a cylindrical shell of a pressure vessel, hydraulic cylinder, valve and pipe is subjected to a very high internal fluid pressure Pi, then the walls of the cylinder must be made extremely heavy or thick [5]. In the design of thick cylindrical shell, lame’s equation is mostly considered; especially when ductile material with close or open ends is to be considered in accordance with the maximum normal stress theory of failures, the stresses σ_t is given as:

$$\sigma_t(max)\frac{p\,[(r_o)^2+(r_i)^2]}{(r_o)^2-(r_i)^2} \tag{1}$$

The above equation shows the maximum principal stress at the inner surface. The minimum principal stress at the outer surface is given as;

$$\frac{2p\,(r_o)^2}{2[(r_o)^2-(r_i)^2]} \tag{2}$$

Maximum shear stress τ is given as;

$$\tau_{max} = \frac{\sigma_{Tmax} - \sigma_{Tmin}}{2} \tag{3}$$

According to Lame's equation, the thickness of the pressure retaining vessel such as valve, is given as

$$t = r_i\left[\sqrt{\frac{\sigma_t+p}{\sigma_t-p}} - 1\right] \tag{4}$$

But when taking into consideration theories of failures, maximum energy strain failure criteria, the safe thickness is calculation from the expression;

$$\sigma^2 \geq \frac{2\sigma_r^2\left[K^4\left(1+\frac{1}{m}\right)+\left(1-\frac{1}{m}\right)\right]}{(K^2-1)^2} \tag{5}$$

Where $K = \frac{r_2}{r_1}$ from equation above, we can see that if the $P_i = \boldsymbol{\sigma}t$ or $P_i > \boldsymbol{\sigma}t$,

Then no thickness of the shell will prevent failure.

Since valve is a high pressure retaining vessel, the bonnet which serves as the end flange of the valve sometimes experience high pressure liquid when the gate is open or close in some cases. The bonnet of the valve is commonly bolted to the body. In the design of bonnet, the required thickness is gotten from Grasch of and Bach equation [5] given below:

$$t_1 = K\sqrt{\frac{P}{\sigma t}} \tag{6}$$

Graschof and Bach σ_t = allowable design stress, K_1 depends on the material and the holding methods, for cast iron, steel as shown in the table below:

Table 1. Bach constant

Material of the cover plate	Type of connection	Circular plate K_1
Cast iron	Freely supported	0.54
	Fixed	0.44
Mild Steel	Freely supported	0.42
	Fixed	0.35

It is one of the mostly found part at valve ends, it aids connections to the engaging pipe. A flanged joint may be made with flanges cast integral with the pipes or loose flanges welded or screwed. The figure below shows two cast iron pipes with integral flanges at their ends. The flanges are connected by means of bolts. The design calculations are adopted from a recommended manual [7].

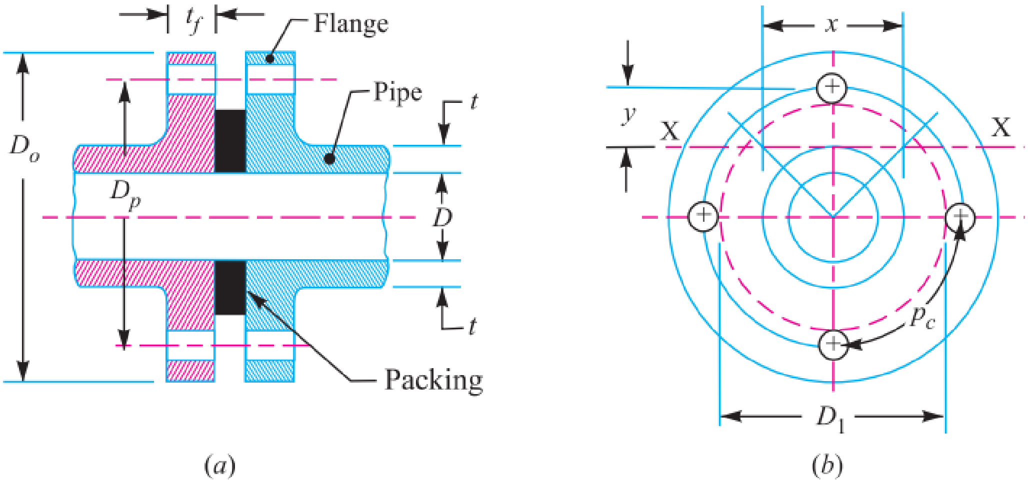

Figure 1. Flange dimension.

The following are the Dimensional consideration of the flange design:
Nominal diameter of bolts, d = 0.75 t + 10 mm
Number of bolts, n = 0.0275 D + 1.6
Thickness of flange, t_f = 1.5 t + 3 mm
Width of flange, B = 2.3 d
Outside diameter of flange, D_o= D + 2t + 2B
Pitch circle diameter of bolts, D_p = D + 2t + 2d + 12mm
Valves are constructed from a number of different materials based on the availability and cost of the material, ease of fabrication, resistance to corrosion and pressure, design metal temperature (DMT), and compatibility with the fluid controlled [6] (Luis and Julio, 2002). Materials used in the construction of valves shall conform to the specifications listed in API 600. According to the different types and requirements, the main performances of valves are sealing, strength [5-6]. When designing and selecting a valve, engineers must consider the basic parameters and the performances, the performance of the fluid including the fluid phase state (gas, liquid or contains solid particle), corrosiveness, viscosity, toxicity, flammable explosion hazard and radioactivity [4-6]
ASTM A216 Grade WCC is the most popular steel material used for valve bodies in moderate services such as air, saturated or superheated steam, noncorrosive liquids and gases. WCC is not used above 800F (427C) as the carbon rich phase might be converted to graphite. It can be welded without heat treatment unless nominal thickness exceeds 1-1/4 inches (32 mm).
Design alternatives and required objectives
Design consideration is based on numbers of selected alternatives; optimal selection is drawn from the lists of alternatives with special consideration to the objective functions as outlined below:

- Availability of material
- Environmental consideration

- Conformation to codes and standards
- Strength of materials

Table 2. Body and Bonnet materials

Material	Alloy Steel
Density	7850 kg/m3
Young Modulus	200 GPa
Yield Strength	6.20422e+008N/m^2
Tensile Strength	7.23826e+008 N/m^2

Table 3. Gate Material

Material	Alloy Steel 1.7014 (17CrS3)
Density	7850 kg/m3
Young Modulus	200 GPa
Yield Strength	4.50590e+008N/m^2
Tensile Strength	5e+008 N/m^2

The above data were used analytically by hand calculations and numerically by finite element analysis to complete the design of the high pressure gate valve. The result s obtain from the finite element analysis was validated by the analytical method outcomes in order to justify the credibility of the HPGV design.
The analytical method constitutes the traditional design hand calculations. Established formula was used to determine stress and deformation values for the design of the HPGV.
The variation of tensile stress and radial stress in relationship of wall thickness of the pressure retaining boundary of valve's body using lame's equation shown in fiqure1 the wall thickness was subdivided into five equal parts.

$$\sigma_t = \frac{p(r_i^2)}{r_0^2 - r_i^2}\left[1 + \frac{r_0^2}{x^2}\right]$$

Where σ_t - tensile stress, r_i -internal radius, r_0-external radius, $p = p_i$ - internal pressure, p_i = 103.44 MPa.
Using above equation, for wall thickness from 0-40 mm, the tangential stress tabulated below:
The tensile stress acting on a material is maximum at the inner surface of the pressure retaining body, while radial stress is maximum at the outer radius. However, increasing the thickness will help to prevent bursting failure in HPGV.
Conferring to this theory, the failure or yielding occurs at a point in a member when the distortion strain energy (also called shear strain energy) per unit volume in a bi-axial stress system reaches the limiting distortion energy (i.e. distortion energy at yield point) per unit volume as determined from a simple tension test. Therefore, equation below is used to test for yield strength of the Gate valve body i.e. maximum distortion energy.

$$(\sigma_{t1})^2 + (\sigma_{t2})^2 - 2\sigma_{t1} \times \sigma_{t2} = \left(\frac{\sigma_{yt}}{f_s}\right)^2 \quad (7)$$

Where σ_{yt} = yield strength, σ_{t1}, σ_{t2} = principal stresses
301620.6 +198737.6 – 489666.7 =10691.5
10691.5 < 1716100

Results: From the above failure analysis in line with the von misses failure theory; the body under analysis will not fail when subjected to the design maximum pressure of 103,44 Mpa.

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АНАЛИТИЧЕСКОЕ ИССЛЕДОВАНИЕ ГИДРАВЛИЧЕСКОГО ДАВЛЕНИЯ, ВЫЗВАННОГО ЖИДКОСТЬЮ В ПРЯМОТОЧНЫХ ЗАДВИЖКАХ

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АННОТАЦИЯ

Во всем мире существует острая потребность в эффективном оборудовании для контроля жидкости. Некоторые из существующего оборудования имеют определенные ограничения для эффективного контроля жидкости в условиях высокого давления. Таким образом, для решения этой проблемы требуются изменение конструкции. В этой статье представлен аналитический метод проектирования и проведения анализа напряжений для задвижек высокого давления, используемых на типичном устье нефтяной и газовой скважины с рабочим давлением до 100 МПа. Самыми важными компонентами прямоточной задвижки являются корпус и шибер (диск). В конструкции

переоборудования используются расчеты аналитического метода для определения распределения напряжений и деформаций в критических компонентах задвижки высокого давления. Результаты аналитического расчетного проектирования и анализа методом конечных элементов показывают разумное согласие путем измерения процентного отклонения производительности. Таким образом, результаты этого исследования подтверждают надежность разработанных прямоточных задвижек, которые соответствуют инженерно-конструкторским оценкам и, в свою очередь, могут быть оптимально усовершенствованы при производстве прямоточных задвижек высокого давления.
Ключевые слова: прямоточные задвижки, аналитический метод, оценка, конструкция и высокое давление.

DÜZAXINLI SİYİRTMƏLƏRDƏ MAYENİN YARATDIĞI HİDRAVLİK TƏZYİQİN ANALİTİK TƏDQİQİ

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XÜLASƏ

Bütün dünyada effektiv maye nəzarət avadanlığına təcili ehtiyac var. Mövcud avadanlıqların bəziləri yüksək təzyiq şəraitində mayenin effektiv idarə edilməsi üçün müəyyən məhdudiyyətlərə malikdir. Beləliklə, bu problemi həll etmək üçün dizayn modifikasiyasına ehtiyac var. Bu məqalədə iş təzyiqi 100 MPa-a qədər olan tipik neft və qaz quyusunun ağzında istifadə olunan yüksək təzyiqli qapaq klapanının dizaynı və gərginlik analizinin aparılması üçün analitik dizayn metodu təqdim edilmişdir. Düzaxınlı siyirtmələrin kritik komponentləri gövdə və şiberdir (disk). Ənənəvi dizayn analitik metoddan hesablamaları qəbul edir, lakin bu işdə yüksək təzyiqli siyirtmənin kritik komponentləri üzərində gərginliklərin və deformasiyaların paylanmasını müəyyən etmək lazımdır. Analitik dizayn hesablamalarından əldə edilən nəticələr onların ağlabatan razılaşmaları göstərir. Buna görə də, bu tədqiqat işinin nəticələri mühəndis dizayn qiymətləndirmələrinə cavab verən dizayn edilmiş düzaxınlı siyirtmələrin etibarlılığını təsdiqləyir və öz növbəsində yüksək təzyiqli düzaxınlı siyirtmələrin istehsalında optimal şəkildə inkişaf etdirilə bilər.
Açar sözlər: düzaxınlı siyirtmələr, analitik üsul, qiymətləndirmə, dizayn və yüksək təzyiq.

SYNTHESIS OF HEAT EXCHANGER NETWORKS CONSIDERING STEAM SUPPLY

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ABSTRACT

In process industries, the heating gap in heat exchanger networks (HENs) is normally compensated by the steam generated from a utility system, thus these two mutually influencing systems should be designed as a whole through establishing structural interrelationships. In this work, an improved stage-wise superstructure of HENs is proposed to integrate with a Rankine cycle-based utility system. Inner- and inter-stage heaters are considered in the new structure. Furthermore, the selection of steam in different levels is also investigated, extending the possibilities of steam utilization in HENs and generation in utility systems. The presented methodology is able to realize the optimal design of HENs by considering the supply and utilization of steam. Heaters' allocations, matches of streams, steam distribution and utilization are optimized accompanying with the trade-off amongst equipment investment, fuel consumption and power generation in objective, which is highly related to the final structure of the system. The optimization problem is formulated into a mixed-integer non-linear programming (MINLP) model and solved towards the lowest total annual cost (TAC) of the entire system. Finally, a case study with two scenarios is studied. The detailed results are given and analyzed to demonstrate the benefit from structural improvement.

Keywords: improved superstructure; HENs; utility system; steam heater; MINLP.

Introduction. With the continuous increase of energy consumption in industrial processes, the energy crisis has further intensified on account of the limited fossil energy reserves. Energy prices are also on the rise at the same time, which drives the pursuit of energy-saving technologies and methods. Heat exchanger networks (HENs) are an inevitable part in processing enterprises for heat recovery. Hot process streams that need to be cooled down and cold process streams that need to be heated up widely exist in process enterprises. Synthesis of HENs is able to obtain HEN structures with reasonable matches between these hot and cold process streams and recover waste heat to the greatest extent. Thus the additional utility consumption will be greatly reduced. On the other hand, utility systems based on steam power cycles are one of the major sources which can export multi-level steam and power to industrial enterprises from a single primary energy source. Energy consumption in utility systems accounts for a large proportion of total industrial energy consumption, which makes it meaningful to improve energy efficiency of utility system. These two parts, the HEN and the utility system, are involved the whole process of energy supply, energy consumption and energy saving. While in implementation, the utility system is usually designed independently with its connections with the processes ignored, so the simultaneous synthesis and optimization of HENs and utility systems should be improved.

Determining the use of utilities is always one important content in HEN synthesis, either within a sequential synthesis method or a simultaneous synthesis method. In pinch technology [1] and the trans-shipment model-based method [2], the consumption of cold and hot utilities is targeted by monitoring of pinch point locations, before designing the network structure in terms of paring hot and cold streams; in the superstructure-based simultaneous method [3], utility consumption is optimized by the trade-off with the capital costs of heat exchangers. Although the study of HEN synthesis has been developed for decades, there are still deficiencies. For example, most previous works on HEN synthesis only used one type of utility at the stream end and assumed that the temperatures of these utilities are able to satisfy all the hot and cold demands, while the fact is that the utility can be in many forms, such as flue gas, steam with different temperatures and hot water. Thus, studies have been launched to investigate the reasonable use of these utilities in HEN synthesis. Costa and Queiroz [4] introduced an extension of the problem table algorithm to optimize and analyze multiple utilities selection and utilization instead of using a grand composite curve. Although the advantages of multi-level utility selection are not analyzed, it is an effective way to optimize multi-level utility utilization with a table algorithm. Salama [5] developed a simple and direct numerical geometry-based technique to target optimal assignment of multiple utilities, but only the heat load was optimized without considering the equipment investment. Shenoy et al. [6] presented a cost-optimal targeting methodology considering the trade-off between energy consumption and equipment investment simultaneously. Optimal selections and loads for multiple utilities were determined based on pinch analysis and the cheapest utility principle. In essence it was still a sequential method, which makes it impossible to achieve a real trade-off between energy and equipment costs. Isafiade and Fraser [7] studied an interval-based MINLP superstructure where the intervals were defined according to the supply and target temperatures of hot or cold process streams. The superstructure model was then applied to HEN synthesis with multi-level available utilities, trading off operating and capital costs for each type of utility. Many nonlinear terms were ignored by mixing split streams at equal temperature, which made the obtained results improvable. Ponce-Ortega et al. [8] developed a stage-wise superstructure that allowed existence of intermediate placement of multiple utilities within each stage. Load and placement of multiple utilities were treated as optimization variables by employing disjunctive programming formulation, rather than regarding utilities as process streams and setting their capacity flowrates as optimization variables. Na et al. [9] proposed a modified superstructure that contained utility sub-stages between adjacent stages. Series utility locations were fixed to facilitate convergence. These studies greatly expanded the design space of the HEN structure and utilization of multiple utilities, but the selection of multiple utilities was mainly based on their price, and the interactions with other systems was not investigated in depth. Zhang et al. [10] explored a new HEN superstructure presentation named the stage-wise chessboard model for management of feasible research regions. A random walk algorithm was employed to lower the calculation load. The problem can be solved faster, but many possible matches are not included in the proposed superstructure because it was equivalent to a stage-wise superstructure with only one stage. Pavão et al. [11] considered the solving difficulty of complex mathematical models resulting from an enhanced stage-wise superstructure which includes the use of multiple utilities at single stream branches. An enhanced meta-heuristic solution method was presented to handle the complex mathematical model, and the superstructure had no difference from the superstructure in Ponce et al. [8] mentioned above. Ma et al. [12] launched multi-objective optimization of interplant HENs operated for multi-periods, which used steam as the heat transfer medium. This research analyzed the conflict between environmental impacts and exchanger investment according to the trade-off between utility consumption and exchanger areas in a case study, but the influence of multi-level steam distribution on the environmental and economic objectives was ignored. Besides, power generation in the utility system was not included in the objective function either, thus the HEN and utility system were not globally integrated.

HENs have been further integrated with utility systems based on their close interactions. Klemeš et al. [13] studied the simultaneous synthesis of a production process and a utility system based on pinch analysis methodology in a total site integration, but the optimization was carried out after integration within a single plant was completed in advance. Thus the heat recovery and utility system were not optimized simultaneously. Liew et al. [14] optimized the design and operation of a centralized utility system to adapt to shutdowns or process upsets, making a trade-off between operational adaptability and operating costs. Detailed synthesis of the HEN was not included either. Chen and Lin [15] proposed a MINLP model to design a steam network and heat recovery network simultaneously. Hot and cold utilities were only placed at the stream ends, while steam can be generated within an inner-stage to promote energy synergy among plants. Different from previous total site integrations, the heat demand of the HEN is unknown before the design stage, but this research did not consider the utilization of multiple types of utilities within the inner-stage. Hipólito-Valencia et al. [16] combined an organic Rankine cycle with HEN and converted waste heat into mechanical energy, which provided more inspiration for the combination of HEN and other processes. Goh et al. [17] synthesized a HEN and utility system simultaneously, but the minimum operating cost, hot and cold utility demands were determined in advance through multiple cascade automated targeting, without realizing the overall optimization. Luo et al. [18] integrated a HEN with a utility system, using the sensible heat of steam condensate to heat cold process streams. Besides, waste heat of the hot process streams was recovered to preheat boiler feedwater, but the combination was allocated at stream ends, which limited the optimization space compared with the superstructure considering inner-stage utility utilization. Martelli et al. [19] developed a two-stage sequential synthesis algorithm to solve the nonconvex MINLP problem derived from the simultaneous synthesis of HEN and utility systems. An isothermal mixing assumption was made and the utility systems were set as process streams, which will increase the problem-solving complexity. Elsido et al. [20] have worked on the simultaneous synthesis of utility systems, Rankine cycles and heat exchanger networks. However, their mathematical model was established based on a p-h superstructure but not a stage-wise superstructure. The main purpose was proposing an ad hoc bilevel decomposition method to improve solution efficiency but not optimizing the utility utilization through improving the superstructure. The simultaneous optimization of multi-plant heat integration using steam as intermediate fluid described in Chang et al. [21] reflected the utilization of energy within the inner-stage, but the steam used within the inner-stage was generated from process streams, and the utility system was not considered completely here.

Huang et al. [22] extended their research to the simultaneous optimization of a heat exchanger network, steam Rankine cycle and organic Rankine cycle, but the steam utilization and ORC evaporator were placed at stream ends, and inner- and inter-stage improvements were not considered in the superstructure.
Research about HEN synthesis and utility system is summarized here. On one hand, improvements have been made to provide more matching possibilities in HENs and obtain HEN configurations closer to an optimal solution. On the other hand, utility systems and HENs were considered simultaneously to strengthen the heat integration. In total site heat integration using the sequential method, the utility system was usually designed after the determination of heating demand, making it unable to obtain the optimal trade-off between equipment investment and operating cost. As for the mathematical programming method which can perform the design of HENs while optimizing the operation of the utility system, either the heaters were placed at stream ends, or the operating parameters and power generation of utility system were not included, even though the utilization of utilities was considered within the inner-stage. Thus, in order to achieve a better network management, an improved superstructure considering multiple utilities utilization within the inner- and inter-stage is presented and integrated with the utility system based on a Rankine cycle. Multiple utilities (steam in different pressure levels) are produced in cascade in the utility system with concurrently power generation. The structure of the HEN and the operation of the utility system are simultaneously influenced by the placement and distribution of multiple utilities, leading to a trade-off among capital costs, fuel costs and power generation profits. It should be noted that steam is not regarded as process stream but rather as an additional heating source in this study, so the established mathematical model corresponding to the special HEN superstructure including inner- and inter-stage heaters is different from previous formulations, by which the steam in any alternative level can be selected as long as the temperature difference demand is met. Finally, cases are illustrated to show the goals of this paper, demonstrating that multi-level steam selection and utilization within the inner- and inter-stage will provide a larger optimization space for steam distribution, power generation and fuel consumption of the utility system, achieving better economic performance of the whole system.

Problem statement: A synthesis of HEN involving the optimization of steam supply and utilization is desired in this study, and the entire problem can be described as follows: a set of hot and cold process streams which must be cooled and heated are given with their heat capacity flowrates, supply temperatures, target temperatures and heat transfer coefficients. To compensate the heating gap, a utility system is optimized to produce appropriate steam, in terms of multiple paralleled mains/levels with different pressures and as well temperatures. These steam branches are usually classified as high pressure steam, medium pressure steam, and low pressure steam (HPS, MPS, LPS), etc. In this study, a set of steam alternatives with certain pressures and temperatures are given for selection. The utility system follows the basic process of a Rankine cycle, but the properties and quantity of generated steam must be optimized towards the best benefit of the whole system. Additionally, all the cost-related parameters are also given in order to assess cold utility cost, fuel cost, equipment investment (for heat exchangers, boilers and turbines) and the profit from selling power. The integration problem aims at achieving the most cost-efficient configuration of steam supply HEN, by making trade-offs amongst the contribution of the mentioned economic sectors in the total annualized cost. For this purpose, the interaction between the two sub-systems must be comprehensively explored and the network details, such as fuel consumption, steam generation and allocation, generated power, stream matches, heat loads, operating temperatures, and area of heat exchangers must be optimized. In order to alleviate the solving difficulties caused by a complicated mathematical presentation, it is assumed that: (1) the heat capacity and heat transfer coefficient are constant throughout the whole process; (2) the heat exchange task is completed in a countercurrent heat exchanger; (3) all operations are adiabatic, ignoring the heat loss and mass loss; (4) transportation problems are not taken into account; (5) only the primary units in the utility system are considered, ignoring the auxiliary devices.

Method overview: To solve the mentioned synthesis problem, an optimization-based method is developed and presented in this study. Since a synthesis solution is characterized by two aspects: the network structure (referring mainly to where to exchange heat and whether generate the steam) and the network parameters (which should be quantified, such as flowrate, temperature, heat exchange areas), a superstructure embedding all potential network configurations is introduced as the first step of the method, to offer decision options for the network structure, then a mathematical model is formulated to perform the optimization, making an automatic determination for both network structure and network parameters. Although this method is developed from the general concepts of HEN and utility systems, the major contribution of this work is still worth mentioning, that is, the utilization of steam in the HEN is extended by considering various locations along streams, being in series or/and parallel with stream-stream heat exchange, and meanwhile not only with the given utility steam but having the generation and allocation of steam integrated, such that the interaction between HEN and utility system can be explored with more space. It is believed that better design solutions will be obtained by using this method.

Superstructure: Figure 1 presents the superstructure of the proposed integration problem. All potential configuration alternatives of the HEN are displayed by considering the utilization of steam. For the HEN side, two hot process streams and two cold streams are used to exhibit the features of the stage-wise superstructure. Since the heaters involving steam could be before, after and in parallel with stream-stream heat exchangers, new stages are introduced in the superstructure to fully explore the utilization of steam. It should be noticed that the stages are divided into two types—the inner-stage and the inter-stage—which are alternately arranged. Within each inner-stage, a cold process stream can be split into multiple branches to perform the heat exchange with hot process streams and steam simultaneously. After that, the branches are mixed non-isothermally into a main stream and go into the inter-stage, where only steam is allowed to heat the cold streams. Inter-stage heaters are allowed at both the start and end of cold streams, so the inter-stage number is one more than that of inner-stages. This arrangement could provide much more matching options between cold process streams and steam with limited stages, enriching the configurations of the superstructure. Hot streams are operated in the traditional manner, cooled down by cold process streams within inner-stages and cooling water at stream ends.

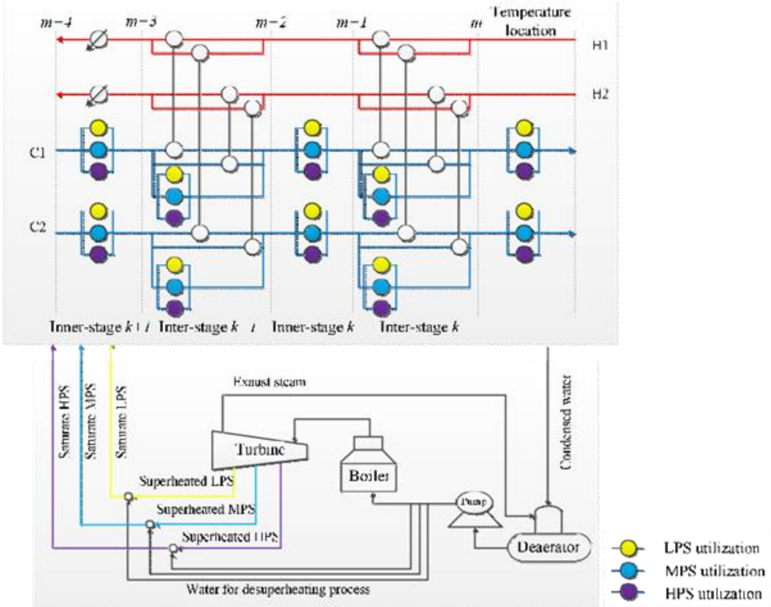

Figure 1. Superstructure for combined HEN and utility system.

The operation of utility system is in typical Rankine cycle: the highest pressure superheated steam is generated in a boiler through combusting fuel, and then sent into a turbine to generate multi-level superheated steam with turbine condensed steam extracted and condensed at the end, producing power at the same time. The superheated steam needs to be desuperheated by mixing with water from the deaerator before being sent to the HEN as saturated steam. After heating the process streams, the condensed water returns to the deaerator, and splits into two parts after the operation, part is sent to the boiler and evaporated into superheated steam, and the residue is mixed with multi-level superheated steam to obtain saturated steam, finishing a cycle. Although the flowsheet of the utility system is quite certain, the possibilities of steam generation and utilization in the HEN need to be determined within the design.

Based on the presented superstructure, a mathematical model can be formulated to determine the optimal network structures and network parameters. In this work, considering the deaerator is an energy-consuming unit, another system without deaerator is also investigated, e.g., using a chemical deaeration technique. The resultant variation in structure is that part of the condensed water is mixed with the corresponding superheated steam, while the rest flows back to the boiler directly.

Conclusions: An improved superstructure for HENs synthesis which contains the utilization of multiple utilities within inner- and inter-stages has been presented to integrate with the utility system. The coupling relationship between HENs and Rankine cycle-based utility systems is investigated and extended by considering various locations of steam heaters. The HEN structure has an impact on steam distribution and accordingly determines the operation of the utility system, thus in the study, the stream matches and heat transfer area of HEN, as well as the multi-level steam distribution, fuel consumption and power generation in utility system are all optimized to minimize the TAC of the whole system. Results are discussed and analyzed with case studies. The obtained economic benefits, 20.4% and 3.7% cost cuts for the cases separately, have demonstrated the method very well. It is concluded that the thermal property and quantity of steam have great effect on fuel consumption, power generation and heat transfer area, leading to an economic trade-off; relieving the location limit on steam heaters could create better solution, in terms of using more lower pressure steam not only at stream ends. Future research will be launched to develop this method to deal with the simultaneous synthesis problem of industrial park HEN and utility system with considering the selection of primary energy sources, e.g., fuel, solar, nuclear, wind, etc.

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СИНТЕЗ СЕТЕЙ ТЕПЛООБМЕННИКОВ С УЧЕТОМ ПОДАЧИ ПАРА

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АННОТАЦИЯ

В обрабатывающих производствах тепловой зазор в сетях теплообменников (HENS) обычно компенсируется паром, вырабатываемым из коммунальной системы, поэтому эти две взаимно влияющие системы должны проектироваться как единое целое путем установления

структурных взаимосвязей. В этой работе предлагается усовершенствованная поэтапная надстройка HENS для интеграции с системой полезности, основанной на цикле Ренкина. В новой конструкции предусмотрены внутриступенчатые и межступенчатые нагреватели. Кроме того, также исследуется выбор пара на разных уровнях, расширяя возможности использования пара у HENS и выработки в коммунальных системах. Представленная методология способна реализовать оптимальную конструкцию HENS с учетом подачи и использования пара. Распределение нагревателей, совпадение потоков, распределение и использование пара оптимизированы с учетом компромисса между инвестициями в оборудование, потреблением топлива и выработкой электроэнергии в объективе, что в значительной степени связано с окончательной структурой системы. Задача оптимизации формулируется в модели нелинейного программирования со смешанным целым числом (MINLP) и решается с целью достижения наименьшей общей годовой стоимости (TAC) всей системы. Наконец, изучается тематическое исследование с двумя сценариями. Подробные результаты приведены и проанализированы, чтобы продемонстрировать преимущества структурного улучшения.
Ключевые слова: усовершенствованная надстройка; HENS; инженерная система; паровой нагреватель; MINLP.

BUXAR TƏCHİZATINI NƏZƏRƏ ALINMAQLA İSTİLİKDƏYİŞDİRİCİLƏRİ ŞƏBƏKƏLƏRİNİN SİNTEZİ

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XÜLASƏ

İstilikdəyişdiricilər şəbəkələrdə (HENS) istilik boşluğu emal sənayesində adətən buxar sistemin istehsalı ilə kompensasiya edilir, belə ki, bu iki qarşılıqlı təsir sistemləri struktur qarşılıqlı əlaqələr yaradılması ilə bir bütöv dizayn kimi edilməlidir. Bu işdə renkin dövrünə əsaslanan kommunal sistemlə inteqrasiya üçün HENS-in təkmilləşdirilmiş mərhələli əlavəsi təklif olunur. Yeni dizaynda intraven öz və mərhələlərarası qızdırıcılar nəzərdə tutulmuşdur. Bundan əlavə, HENS-də buxarın istifadəsi və kommunal sistemlərdə istehsal imkanlarını genişləndirərək, müxtəlif səviyyələrdə cütlərin seçimi də araşdırılır. Təqdim olunan metodologiya buxarın verilməsi və istifadəsi nəzərə alınmaqla optimal HENS dizaynını həyata keçirə bilər. Qızdırıcıların paylanması, axınların üst-üstə düşməsi, paylanma və buxarın istifadəsi avadanlıqlara, yanacaq istehlakına və sistemin yekun strukturu ilə əhəmiyyətli dərəcədə əlaqəli olan obyektivdə elektrik enerjisinin istehsalına qoyulan investisiyalar arasında kompromis nəzərə alınmaqla optimallaşdırılmışdır. Optimallaşdırma məsələsi qarışıq bir tam (MİNLP) ilə qeyri-xətti proqramlaşdırma modelində hazırlanır və bütün sistemin ən az ümumi illik dəyərinə (TAC) nail olmaq üçün həll edilir. Nəhayət, iki ssenari üzrə tematik tədqiqat aparılır. Ətraflı nəticələr struktur təkmilləşdirilməsi faydaları nümayiş etdirmək üçün verilir və təhlil edilir.
Açar sözlər: təkmilləşdirilmiş əlavə; HENS; mühəndislik sistemi; buxarlandırıcı; MİNLP.

CONSEQUENCES OF ELECTROMAGNETIC DEFECTOSCOPY IN OIL AND GAS WELLS

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ABSTRACT

Currently, various methods and methods are used to diagnose the technical condition of wells, which make it possible to determine the most complete and reliable picture of its condition. This paper presents the results of assessing the technical condition of oil wells using the electromagnetic flaw detection method.
Keywords: oil and gas wells, diagnostics of technical condition, electromagnetic flaw detection, corrosion of the wall of the pipe column.

During the operation of oil and gas wells, the load-bearing elements and equipment components used in the process are exposed to excessive pressures, corrosion and mechanical wear. This in turn helps to reduce their resources and increase the probability of failures. In these conditions, the importance of diagnostic maintenance of the structural integrity and tightness of wells, accommodation of repair and restoration work, as well as increasing the service life of equipment and technical means increases.
Currently, various methods are used in the oil sector of Azerbaijan to diagnose the technical condition (DTC) of all categories of oil and gas wells, the essence of which is to determine the places of leak-tightness of the production column, pumping and compressor pipes (tubing), bottom and others [1]. Despite the fundamental difference in the DTC technology, they are united by one goal - to collect the most detailed information about the condition of the well, thanks to which the processes occurring during the operation of the well are predicted. The information base collected on the basis of the well's DTC is usually based on the results obtained by a combination of several technologies that allow us to determine the most complete and reliable picture of the well's condition. Thanks to this, a decision is formed on its further operation or repair, as well as complete liquidation.
Currently, the SOCAR oil company has more than six thousand oil and gas wells on its balance sheet, which are periodically plunged by DTC. About 15.2% of them have been liquidated, 3.91% are in conservation, and the rest are being exploited [2, 3].
Since 2005, the Department of Geophysics and Geology has been created in the structure of the SOCAR oil company to replace the software "Geophysics and Engineering Geology". This department carries out both on land and in the Azerbaijani water area of the Caspian Sea, carries out exploration and field-geophysical, as well as complex engineering and geological work.
The purpose of the study. Assessment of the technical condition of oil wells using the method of electromagnetic flaw detection.

Existing methods of DTC wells. There are several ways to diagnose a well, used both during drilling and during operation. Among such methods of assessing the condition of a well, the most widely used are geophysical technology (logging) and tele-inspection (TI).
Logging of wells is carried out using a geophysical probe, which allows you to fully examine the well, determine the degree of pipe wear, thanks to its own sensors, it measures the wall density, temperature, pressure and other parameters. Among the types of logging, the following are distinguished: electric (EC), electromagnetic (EMC), gamma-ray logging (GC), radioactive (RC), acoustic (AK), as well as such methods as thermal logging (TC) and inclinometry (NM) [4].
Since geophysical methods are characterized by high costs and to some extent limited information about the condition of the well, special interest is increasing in the tele-inspection (TI) method. This method differs not only in efficiency, but also allows you to visually observe the process, identify the integrity of pipes, the condition of seams, coupling connections and filters, the level of deposits, etc.
Modern systems for the tele-inspection (TI) assessment of the condition of wells are equipped with high-resolution color video cameras, which are able to detect the smallest defects of casing pipes, as well as to assess the amount and nature of deposits.
In this work, wells of the Bulla-More deposits were selected to assess the technical condition, as well as the sufficiency of the technical means used. Monitoring of the well condition was carried out by an electromagnetic flaw detector of the EMD-43 type.
The results of the study and their discussion. Initial tests were carried out in the Bulla-More field in the BD-126 well, covering a depth of 1998.9 -2381.5 meters. The aim was to determine the location of defects and damages in the technical column with diameters D_o x D_i = 127.0 x 108.6 mm, D_o x D_i = 339.7 x 313.6 mm and D_o x d_i = 473.1 x 446.1 mm.
It was found that at the maximum value, corrosion in a pipe with an outer diameter of 127.0 mm occurs at a site of 2188.2 mm and is 4.3%. Similar studies carried out in technical columns with diameters of 339.7 and 471.1 mm found that the maximum values of corrosion were observed in sections of 2366.8 and 2363.9 meters and were, respectively, 36.1% and 16.4%. There were no gaps in the coupling connections. Pic.1 shows the results of determining the state of the well at a depth of 2300.0-2381.5 m, using an electromagnetic flaw detector of the EMD-43 type.
As can be seen from the analysis of the results of experiments on assessing the diagnostic condition of wells shown in Pic.1, the estimated value of the maximum penetration of electromagnetic waves into a pipe with a diameter of 127 mm is the smallest (4.3%) and at the same time the wall thickness of the pipe has decreased from only 9.19 mm to 8.79 mm (i.e. 9.19-8.79=0.40 mm).
The results of the research also revealed that the loss of metals in a technical column with a diameter of 339.7 mm compared to a pipe column of 473.1 mm, despite the same operating conditions, differs in different degrees of penetration of electromagnetic waves. With the penetration of electromagnetic waves (36.1%) for a column with a diameter of 339.7 mm, the decrease in wall thickness is 13.06 -8.04 = 5.02 at a depth of 2366.8 m and is characterized by the intensity of corrosion. In pipes with a diameter of 473.1 mm and at a suspension depth of 2363.9 m, the pipe wall thickness changed less than 2.2 and amounted to 13.49-11.28=2.21 mm.

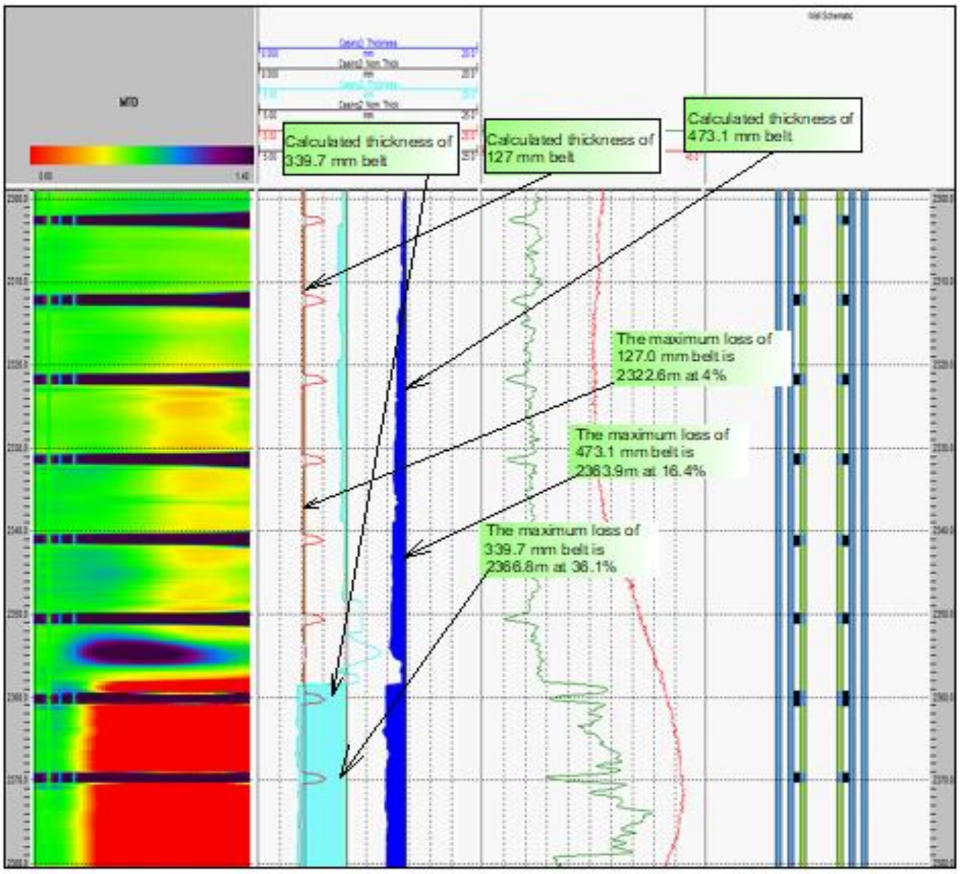

Figure 1. Results of the assessment of the diagnostic condition of the well at a depth of 2300.0-2381.5 m.

I interpret the results of studies on the penetration of electromagnetic waves, they are grouped into categories A, B, C, D and E (see Table 1).

Table 1. The results of the assessment of the corrosion condition in the well.

Penetration category	The amount of measurement of damage areas in pipes		
	Pipe diameter (d_o x d_i), mm		
	127,0 x 108,6	339,7 x 313,6	473,1 x 446,1
light (A)	40	10	26
Few (B)	-	27	10
Secondary (C)	-	-	3
High (D)	-	-	1
Intensive (E)	-	3	-

As can be seen from the data in Table 1, out of forty cases of measuring the corrosion condition of pipes, the best results are a pipe with a diameter of 127.0 mm and it is recommended for further operation. The conditions of the inner surfaces of pipes with diameters of 339.7 and 473.1 mm differ from the original. It was also found that the loss of metal along the cross-section of the pipes occurs unevenly (see figure 2) and increases as the well deepens.

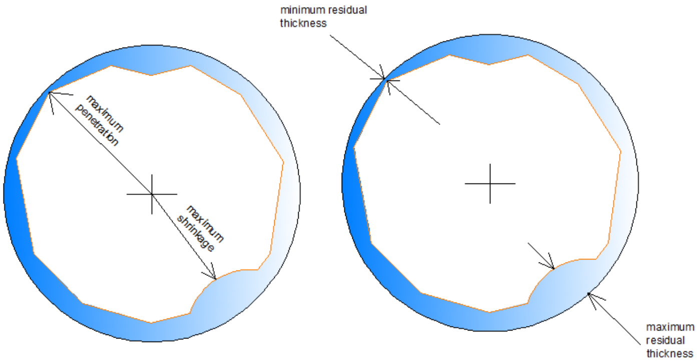

Figure 2. Pipe cross-section and the nature of metal loss along its perimeter.

Table 2 shows comparative data on the technical condition of the well provided by one of the foreign companies (Schlumberger). The analysis of these data showed their high proximity.

Table 2. Comparative results of gamma logage and gamma inclinometry.

Pipe diameter	Measurement results			
	Measurements conducted by SOCAR		Measurements carried out by a foreign company	
	Penetration of magnetic waves	Pipe wall thickness change, mm	Penetration of magnetic waves	Pipe wall thickness change, mm
127,0	4,3	0,40	4,28	0,395
339,7	36,1	5,20	35,12	5,123
473,1	16,4	2,21	16,53	2,23

Thus, based on the conducted research, the following main conclusions can be drawn.
- the method of electromagnetic flaw detection is an effective method and can be successfully applied to assess the technical condition of oil and gas wells.
- the corrosion condition of pipes with a diameter of 127.0 mm showed the best results, and it is recommended for further operation.
- the conditions of the inner surfaces of pipes with diameters of 339.7 and 473.1 mm differ significantly from the original.
- the loss of metal along the cross-section of the pipes occurs unevenly and increases as the well deepens.

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РЕЗУЛЬТАТЫ ЭЛЕКТРОМАГНИТНОЙ ДЕФЕКТОСКОПИИ В НЕФТЯННЫХ И ГАЗОВЫХ СКВАЖИНАХ

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АННОТАЦИЯ

В настоящее время для диагностики технического состояния скважин применяются различные методы и способы, которые позволяют определить наиболее полную и достоверную картину о ее состоянии. В работе приведены результаты оценки технического состояния нефтяных скважин с применением метода электромагнитной дефектоскопии.
Ключевые слова: нефтяные и газовые скважины, диагностики технического состояния, электромагнитная дефектоскопия, коррозия стенки колонны труб.

NEFT VƏ QAZ QUYULARINDA ELEKTROMAQNİT DEFEKTOSKOPİYANIN NƏTİCƏLƏRİ

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XÜLASƏ

Hazırda quyuların texniki vəziyyətinin diaqnostikası üçün müxtəlif metod və üsullar tətbiq edilir ki, bu da onun vəziyyəti haqqında ən dolğun və düzgün mənzərəni müəyyən etməyə imkan verir. Məqalədə elektromaqnit defektoskopiya metodundan istifadə etməklə neft quyularının texniki vəziyyətinin qiymətləndirilməsinin nəticələri verilmişdir.
Açar sözlər: neft və qaz quyuları, texniki vəziyyətin diaqnostikası, elektromaqnit defektoskopiya, boru sütununun divarının korroziyası.

THE CHOICE OF MATERIAL FOR THE MANUFACTURE OF SEALING ELEMENTS OF THE PACKER

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ABSTRACT

Despite numerous studies in the field of creating rubber products, to date, the choice of material for the manufacture of packer sealing elements has not found its solution. This issue becomes especially important drilling deep and ultra deep wells.

The article is devoted to the choice of a composite material based on a polymer with highly elastic fillers.

Keywords: Polymer matrix, nanoparticles, composite material, sealing element, packer.

The relevance of the topic: Drilling oil and gas wells, as well as when performing various repair and restoration works, the situation when it is necessary to conduct independent studies of individual sections of the well for the inflow of oil, gas or water, as well as to carry out separate development of several horizons arises. Special devices packers are used for this purpose. They serve to seal the inter-tube space, holes in boreholes, long pipes, etc. According to the functional purpose, the main property of the packer is the ability to withstand significant pressure drops, thereby ensuring not only the reliability of specific equipment, but also the industrial safety of the entire process both during drilling and during operation. Currently packers of various designs with a diameter from 88 to 245 mm are used in the oil and gas fields of the republic, for casing pipes □ 114 273 mm, which provide a pressure drop of 14, 21, 35, 50 and 70 MPa. An important step in choosing the type and design of the packer is the correct determination of the material grade of the sealing elements (UE). The rational choice of elastomeric materials to ensure high tightness is an important task, especially to ensure the long and reliable service of oil and gas field equipment (NGPO) used in the extraction, processing and transportation of high-pressure natural gas and oil with a high content of dissolved gases.

By analyzing the research results, the operating conditions of packers and the main causes of failures showed that about 50-60% of failures are directly or indirectly related to sealing elements, 20-25% to the anchorage node, 20-25% to the rest. As can be seen from the variety of the above-mentioned findings, various types of destruction are dominant due to changes in the physical and mechanical properties of their rubber elements. Failures of rubber products lead not only to a decrease in the profitability of oil production and transportation, but also to an increase in the frequency of accidents, direct losses from which are ten times higher than the cost of the packers themselves.

Based on this, very strict requirements are imposed on the sealing elements of packers in terms of hardness, accumulation of residual compression deformation, resistance to the working environment, etc.

The purpose of the work is to select the material for the sealing elements of the packer, ensuring reliability during their operation.

The choice of the formulation of the composite material and the methodological basis for conducting experimental research: Currently, rubbers based on nitrile butadiene rubbers (BNCs) are used for the manufacture of sealing elements of packers. More than twenty BNC brands with a wide range of regulated properties are known in the industry [4, 5]. However, sealing elements made of the specified material do not always meet the requirements imposed on them.
There are known technologies for producing a new generation of BNCs based on them and various rubber mixtures, characterized by high properties of exposure to aggressive media and physics-mechanical characteristics [6, 7]. The new generations of BNCs are also characterized by high ecological purity, since sulfonate emulsifiers have been replaced with biodegradable paraffinate and tallate emulsifiers in the technology of their production [7, 8].
Based on this goal, the following components were selected for the structure of the composite material: a mixture of synthetic butadiene-nitrile and hydrogenated butadiene-nitrile rubber (BNC+GBNA), vulcanizer, stabilizer, technological additives and copper and iron nanoparticles. The content of nitrilacrylic acid (NAC) in nitrile butadiene rubber was 17-19%.
Table 1 shows the characteristics of the components included in the composition of the composite material for the manufacture of UE packers.

Table 2. Physical and mechanical properties of the studied composite materials.

Name of indicators	Composition of prototypes, %		
	K-1	K-2	K-3
Matrix (BNC+BNC)	30/34	32/32	34/30
The mass fraction of the antioxidant	1,0	1,0	1,0
Δm isooctan-toluene vulcanizer	30	30	30
Carbon black	1,5	1,5	1,5
Stabilizer (Novantox)	1,0	1,0	1,0
Technological additive (Softener PC-1)	1,5	1,5	1,5
Nanoparticle	1,0	1,0	1,0

The viscosity was determined on a laboratory viscometer, in accordance with the requirements of GOST 10722-84, the physical and mechanical properties of composite materials were carried out: Shore A hardness according to GOST 263-75, resistance to aggressive environment according to GOST 9.030-74, wear resistance, elasticity and aging resistance were also determined.

The results of the study and their discussion: It is known from field practice that in order to seal the annular space, the sealing cuffs of the packer expand radially under the influence of axial load and seal the annular space. Usually, the outer diameter of the sealing cuffs in the transport (uncompressed) position is 10-30 mm smaller than the inner diameter of the casing pipe in which the packer is installed. At the same time, it is very important, along with the high strength characteristics of the UE, to ensure good deformability.
Figure 1 shows the results of a study to assess the ability of elastic deformation of prototypes.

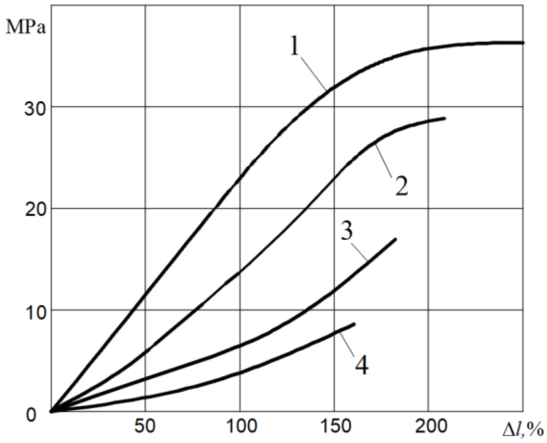

Figure 1. Results of the study to assess the ability of elastic deformation of prototypes:
1 - composite material K-1, 2 - composite material K-2, 3 - composite material K-3, 4 - without nanofill K-4.

As can be seen from the graphs presented in figure 1, the deformability of the sample from the K-1 composition is characterized by a high value, while the deformability of the sample without nanodubments is quite low.
Table 2 shows the results of studies on the properties of composite materials, as well as without the use of nanofillers (K-4).

Table 2. Physical and mechanical properties of the studied composite materials.

Property name	Unit of measurement	Symbols	Composition of prototypes			
			K-1	K-2	K-3	K-4
Muny viscosity at 100^{0}C	Conventional unit	*MБ, 100°C*	76	68	65	55
Elongation at break	%	ε_p	250	210	180	150
Hardness	Conditional unit by Shore	*H*	78	73	70	65
Rebound elasticity	%	*S*	18	16	14	12

In addition, it has been established that composite materials filled with nanoparticles have comparatively (without filled BNC) high physicomechanical and elastic properties.
Thus, based on the results of the conducted research, the following conclusions can be formed:

- composite material with a formulation corresponding to K-1 is characterized by high physical, mechanical and elastic properties, as well as increased resistance of rubbers to aggressive media, which allows its use as a material for the manufacture of sealing elements for packers;
- the nanoparticle factor in the composition of the composite material contributes to the formation of a stable structure with increased characteristics.

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ВЫБОР МАТЕРИАЛА ДЛЯ ИЗГОТОВЛЕНИЯ УПЛОТНИТЕЛЬНЫХ ЭЛЕМЕНТОВ ПАКЕРА

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АННОТАЦИЯ

Несмотря на многочисленные исследования в области создания резинотехнических изделий, до настоящего время выбор материала для изготовления уплотнительных элементов пакера не нашел своего решения. При бурении глубоких и сверхглубоких скважин этот вопрос становится

особенно важным. Статья посвящена выбору композиционного материала на основе полимера с высокоэластичными наполнителями.
Ключевые слова: полимерная матрица, наночастицы, композиционный материал, уплотнительный элемент, пакер.

PAKERİN KİPLƏNDİRİCİ ELEMENTLƏRİN İSTEHSALI ÜÇÜN MATERİALIN SEÇİLMƏSİ

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XÜLASƏ

Rezintexniki məmulatın yaradılması sahəsində çoxsaylı tədqiqatlara baxmayaraq, pakerın kipləşdirici elementlərinin istehsalı üçün material seçimi bu günə qədər öz həllini tapmayıb. Dərin və çox dərin quyuların qazılmasında bu məsələ xüsusilə vacib məsələyə çevrilir. Məqalə yüksək elastikli doldurucularla polimer əsasında kompozisiya materiallarının seçiminə həsr olunmuşdur.
Açar sözlər: polimer matris, nanohissəciklər, kompozisiya materialı, kiplǝndirici element, paker.

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INCREASING THE EFFICIENCY OF WATER INJECTED TO LONG-TERM DEVELOPED FIELDS

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ABSTRACT

The article presents the results and analysis of the application of water injection metods to enhance oil recovery/ It is noted that recently, instead of increasinq oil recovery, the injected waters lead to waterinq of the reservoirs.
The ways of reducinq the water cut of the reservoirs are shown. The need to improve water floodinq metods is noted.
Keywords: oil fields, water supply, oil recovery.

The actuality of the subject: The development of residual oil reserves concentrated in long-term oil fields is one of the most pressing problems of energy supply now and in the next decade. With the application of the second method of injection, the final oil recovery factor in the range of 0.25-0.45 does not ensure the extraction of oil resources. Reserves that are difficult to extract using this method from industrial inspections account for about 55-75% of the initial geological reserves. The high coverage of reservoirs in the oil-saturated state of reservoirs requires the study of many additional issues, the causes and nature of the formation and formation of wells, the solution of the optimal application in the field development system to increase the efficiency of water flow regulation and oil yield.
As most of Azerbaijan's oil fields have been in development for a long time, wells are starting to run out of oil as a result of declining oil production and rising irrigation rates. Today, the number of such oil and gas wells is in the thousands. In this context, the intensification and increase of oil production is of great importance.Today, the volume of water pumped to stabilize the formation pressure continues to increase.
The purpose of the work. Development of an effective method to prevent irrigation in oil wells.
Level of study of existing research on the problem. In order to prevent irrigation, such an impact method should be applied to solve the following problems as a result of this work:
- prolongation of well development period;
- object of operation as a result of selective isolation of wet layers regulation of development of oil non-homogeneous reservoirs;
- reduction of external water production by saving energy consumption;
- protection of natural resources and subsoil.
Experience in the development of fields by irrigation and research scientists working in this field - Abacov MT [1], Boxerman A.A. [2], Gorbachev A.T. and Kovalyov A.Q. [3],

Kisilenko B.E. [4], Marxasin İ.L. [5], Sayfullin Z.Q. [6], Surguchev M.L. [7] and b. show that the reason for the low efficiency of irrigation is the sharp difference in the viscosities of the compressible and compressed fluids, the improper movement of the water-oil contact, and thus the formation of non-developed zones.

The formation of the viscosity ratio leads to the formation of aquifers from the first day of field development, and thus a sharp decrease in the coverage of the formation. This occurs at all compression rates applied to the fields as the difference between the viscosities of the oil and water increases. It is noted that during the development of high-viscosity oil fields by irrigation, there is a low annual rate of resource development, rapid irrigation of production wells, large volumes of oil water and, consequently, a decrease in oil recovery.

The characteristic of high-viscosity oils is that they have non-Newtonian properties and adversely affect the development of the field [8]. If the viscosity of Newtonian oils remains constant at constant pressure and temperature, the structural viscosity of non-Newtonian oils varies with time, depending on the nature of the movement.

The physical nature of the rheological anomaly of non-Newtonian fluids is explained as follows: the liquid forms an internal molecular surface structure that can disintegrate as a result of deformation and change its properties during flow. This case - non-Newtonian filtration of viscous-anomalous fluids in porous media - Mirzajanzade A.Kh. [9,10], Qorbunov A.T. [11], Kovalyov A.Q. [12], Devlikamov V.V. [13] has been researched by.

Non-Newtonian oil fields in Azerbaijan include Binagadi, Balakhani-Sabunchu-Ramana, Kurovdag, Gum Adasi, Mud Pilpilasi and other fields containing 10% asphaltene, 15% paraffin and more than 60% resin. The active components mentioned in the oil reduce the filtration rate, which is characterized by poor mobility, affecting its filtration properties, especially when squeezed through cold water. Studies show that the active components in the oil affect the filtration rate and the displacement of oil and water in the porous medium. For example, the paraffin, asphaltene, and resin components in the oil increase the initial movement gradient on the rock surface and the thickness of the high-viscosity adsorption-solvate layer. This causes the oil droplet to adhere to the rock surface, making it difficult for the anomalous layer to disperse through the injected water.

It is known that regular injection of cold water to restore pressure causes such a layer to cool at long distances from the injection well. As a result, the initial pressure gradient is formed as a result of the colmotion of paraffin crystals, which in the low-permeability multilayer areas - zones of cooled, immobile, non-Newtonian oil.

Barisov Y.P.[14] show that the different filtration properties of the productive strata lead to an uneven distribution of the injected water across the cross-section, resulting in the development of high-permeability layers. The water injected into these reservoirs attacks the bottom of production wells, which causes premature watering of the well product. On the other hand, the inflow of cold water cools the formation and creates stagnant zones.

Marxasin İ.L. [5] shows that filtration failure occurs when the filtered oil contains asphaltene. In this case, the permeability of oil is 26 times less than the permeability of air.

Mirzajanzade A.X.[9] note that at all values of the pressure drop applied during the filtration of viscous oils, the movement of oil is weak in the parts of the formation with small pores. The increase in pores with such oil movement removes them from active filtration, thereby

reducing the oil permeability of the formation. The volume of pores with poor mobility at a constant pressure gradient increases with increasing structural and mechanical properties.
Some authors [15,16] explain the poor development of fields during irrigation with micro-macroheterogeneity of the layer. Tahirov N.C. [17], Jeltov Y.V. and Kovalyov A.Q. [18], Fazliyev R.T [19] show that complete displacement of oil from the micro-macro-non-porous environment is possible under the influence of capillary forces. Water injected into a micro-porous medium enters the small pores under the influence of capillary forces without entering the high pores of the oil in the form of separated droplets.
In macroporous porous medium, Ban A., Bogomolova A.F. [20] and b. capillary flow occurs between layers. However, the rapid attack of water on high-permeability areas of the porous environment prevents low-permeability areas from being affected. In this regard, the ratio of capillary and hydrodynamic forces determines the coverage of the micro-macro-non-porous environment with irrigation, which characterizes the form of water-oil contact.
Thus, the formation of oil slime and the reason for their inactivity are capillary forces or, in general, the surface contact of the rock with the formation fluid.
It is shown [21] that the distribution of residual oil in the reservoir also depends on the wetting of the rock. In high hydrophilic reservoirs, most of the residual oil is stored in the narrowing of large pores. In hydrophobic reservoirs, the oil is distributed in a thin layer on the rock surface. Most of the strata are in a state of moderate irrigation, during which the oil drips on the surface of the rock.
Ametov İ.M. [22]. It is noted that at zero value of wetting, it is very strong, regardless of the thickness of the oil layer absorbed on the rock. At any other value of Islam, the oil layer can change its thickness depending on the following expression.

$$\sigma_{bc/su} - \sigma_{bc/n} = \sigma_{su/n} \cos\theta$$

Here $\sigma_{bc/su}$, $\sigma_{bc/n}$, $\sigma_{s/n}$ - solid body-water, solid body-oil and water, respectively - surface tension at the oil boundary, θ - wetting angle.
The change is due to the fact that the differentiation of the oil layer into droplets occurs rapidly, depending on the surface tension on the one hand, and the viscosity on the other, the shape of the droplets is determined by the above formula, and the adhesion force is determined by the following formula.

$$A = \sigma_{su/n} \left(1 + \cos\theta \right)$$

This process is observed regardless of whether any of the squeezed or squeezed liquids wet the rock surface.
Analyzing the above, it can be concluded that the irrigation method needs to be improved to apply to fields with difficult-to-extract oil reserves, taking into account the geological and physical parameters - the rheological properties of the oil, the characteristics of the reservoirs and the heterogeneity of the formation.
Recently, due to the insufficient oil-bearing capacity of the injected water, great importance has been attached to increasing the efficiency of the methods used abroad and in our country and increasing the oil recovery factor by developing new ones. In this regard, the extraction of

oil from developing fields with the help of progressive methods is one of the important issues of the national economy.

Some progress has been made in this area, and many physicochemical methods based on SAM (surfactants), acids, alkalis and solvents have been developed. However, due to the fact that the impact coverage of the reservoirs remains small, the oil recovery factor of the reservoirs does not reach the required level. Coverage of the formation volume depends mainly on the geological structure of the field, the non-uniform nature of the formation rocks, the physical and chemical properties of the fluids in the formation and the efficiency of oil field development. The most influential factor is heterogeneity in conductivity. In order to increase the current and final oil yield, productive formation impact methods use various modifications of injection - along the field, behind the pipeline, inside the contour, etc. relies on artificial irrigation of collectors by applying systems. The increase in the impact coverage of the strata is due to the use of selective, nest (hearth) and periodic irrigation systems that ensure the efficient use of injected water energy, high pressure in the injected water line and the selection of the optimal well network.

Extraction of oil from lithological homogeneous strata with significant non-recoverable reserves and located in water-oil zones is even more difficult.

The main way to increase the oil yield of a field that is in the last period of development with areas that have been extensively washed away by water is to regulate the flow of water. In oilfield practice, the prevention of water infiltration into production wells should begin from the time the field is developed. However, the lack of the required chemical reagents reduces the efficiency of this work, allowing the work to be carried out only in the wellbore area.

It is known that due to the high degree of heterogeneity of productive strata and the nature of the liquids that saturate them, the bulk of oil reserves are obtained during the operation of production wells by irrigation. When the water content in the extracted liquid reaches 96-98%, the operation of wells is not economically viable, they are either stopped working or water isolation is carried out. Therefore, using all methods to increase the oil recovery factor, the maximum yield should be obtained before watering the wells. It should be noted that 50-70% of the balance reserve occurs as a result of the attack of water or injected working agent on low-permeability sludges and narrowed areas of high-permeability zones. Full coverage of the strata is not possible with the use of the most modern methods of irrigation - cyclic irrigation, diversion, rapid liquid production, as well as the use of physicochemical methods using various reagents.

Irrigation of reservoirs and reduction of water in production from production wells are the main problems facing researchers. At present, in order to increase the efficiency of water injection oil field development, there are a number of technologies to increase the coverage of the formation and to regulate the infiltration flow.

An analysis of the results of a large number of studies conducted to prevent water flow in wells did not reveal that any of them had a complex effect.

The main feature of water-oil displacement is characterized by a "resistance factor" [16]:

$$R_f = \lambda_{su} / \lambda_n$$

where lsu and ln are the coefficients of mobility of water and oil.

The mobility coefficient is characterized by the ratio of the permeability of the rock to the viscosity of the flowing fluid:

$$\lambda = K / \mu$$

The ratio of the coefficients of mobility of the combined flows of different fluids is determined by the volume velocity of their individual flows. The compression state of the oil can be considered pistonless if the compression agent does not flow to the production wells, in other words, the resistance factor will take the following value in each layer:

$$Rf \geq 1$$

At present, there is no doubt that in order to absorb the residual oil from the reservoir, any pressure applied must repel the capillary forces that create the pressure gradient. Only then does the residual oil move within the formation.

At present, there is no doubt that in order to absorb the residual oil from the reservoir, any pressure applied must repel the capillary forces that create the pressure gradient. Only then does the residual oil move within the formation.

$$N_k = \mu_{su}\,\vartheta / m\sigma$$

In order to assess this situation, it can be assumed that the capillary number (Nk) is a parameter that characterizes the ratio of viscosity and capillary:

Here, the viscosity of the compressed fluid, Pa · s, u - linear filtration rate of liquid, m / s, surface tension at the boundary of the oil compressor, N / m, m - porosity coefficient, Nk = 10-6 is assumed for normal starting conditions.

It is noted that the flow of water within the formation passes along the main flow line, without entering the oil-saturated areas. As a result, water attacks the production wells, which causes a decrease in the pressure gradient in the area of contour tension. If the pressure gradient decreases

leads to an increase in the structural and mechanical properties of the oil and a weakening of the oil flow. These factors lead to the formation of stagnant zones in the stratum.

It is known that not all formation fluids move during the filtration process. The main factor influencing the filtration of a liquid, as mentioned above, is its viscosity. Under the same conditions, the higher the viscosity of the liquid, the smaller the moving volume of the liquid. The structural and mechanical properties of oils also affect their moving volumes at different pressure gradients.

The effect of structural and mechanical properties of oil on the formation of stagnant zones in the formation during the development of non-Newtonian oil fields was studied. In the study of irrigation of anomalous oil fields, the method of mathematical modeling was applied using the generalized Darcy law for each phase and the system of equations for two-phase filtration of immiscible liquids.

Kreyq F.F. [23] shows that as a result of cold water injected during the irrigation of a multilayer field, the movement of oil in low-permeability layers will be weakened due to the uneven displacement of oil from the layers of different permeability. In this regard, as the impact of the structural and mechanical properties of oil in such strata increases, it will lead to an increase in additional oil loss in stagnant zones.

Most oil fields are heterogeneous in conductivity. Inhomogeneity can vary both linearly and across the field. Therefore, although the pressure gradient remains constant, the leakage resistance will vary in different areas of the field. Depending on the shape of the heterogeneity and their location, the filtration rate will also vary. In this regard, the degree of development (oil yield) will be sharply selected in different areas, depending on the heterogeneity.

These factors, together with the structural and mechanical properties of the oil, will affect the size of the highly stagnant zones (not covered by the process).

The effect of formation heterogeneity on the development of oil fields has been developed by many researchers. For the first time in this field Borisov Y.P. [14] systematization of the concept of heterogeneity, the effect of various parameters of heterogeneity of collectors on the development of the field.

Employees of the All-Union Scientific Research Institute of Petroleum [24] developed a methodology for the study of productive strata based on the analysis of large amounts of mining data from the Romashkin field. According to them, heterogeneity refers to the permeability, porosity, effective thickness, oil saturation, etc. of the formation along the field and cross section. should be understood.

The break in the formation has a great impact on the development of the field. If a large part of the formation is composed of isolated lenses, this will not only affect the final oil recovery coefficient by reducing the coverage ratio, but will also affect the current oil production by reducing the impact on the reservoir.

It is known that the limit value of pressure at constant temperature depends on the composition of the oil, gas saturation and permeability. Asphaltene and resin, which have the greatest impact on the structural and mechanical properties of oil, the amount of which can be approximated by the luminosity coefficient in different areas of the formation. Therefore, to determine the limit value of the pressure gradient, a map of the distribution of the transmittance coefficient is used to map the distribution of the luminance coefficient in the field. These maps allow you to map the distribution of the boundary value of the calculated pressure gradient.

During the irrigation of the formation, as a result of the fact that the productive layer has different filtration properties, the injected water is improperly absorbed by the cross section. As a result, first of all, high-permeability layers are quickly involved in development. In these layers, water moves to the bottom areas of production wells, as a result of which the product of the well is subject to irrigation. On the other hand, the injected cold water increases the viscosity of the formation oil, creating conditions for the formation of stagnant zones. The size of these zones depends on the amount of heavy components in the oil and the permeability of the rocks.

n order to increase the effectiveness of the methods used to increase the oil yield of reservoirs with heterogeneous reservoirs, their application in the intended direction should be adjusted. This requires the identification of areas of active compression, as well as areas of poor development and stagnation.

It is known that even collectors perceived as homogeneous are in fact heterogeneous. They can consist of several layers with different conductivities.

In real stratum inhomogeneous strata, the efficiency of oil compression through water does not depend on the degree of heterogeneity of the stratum and the viscosity of the oil, provided that there is a hydrodynamic connection between the strata. In this case, the mechanism of oil compression is complicated by capillary and hydrodynamic flow. Many theoretical and experimental works have been devoted to this problem.

The result: Thus, based on the analysis of the given review, it can be concluded that
A new technology must be developed to reduce the intensive movement of the injected agent into the washed areas of the formation, and to ensure that the oil, which has hitherto been unaffected, enters the remaining areas in the form of stagnant zones.

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УВЕЛИЧЕНИЕ ЭФФЕКТИВНОСТИ ЗАКАЧИВАЕМОЙ ВОДЫ В ДЛИТЕЛЬНО РАЗРАБАТЫВАЕМЫХ МЕСТОРОЖДЕНИЙ

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РЕЗЮМЕ

В статье приводятся результаты и анализ применения методов закачки воды с целью увеличения нефтеотдачи пластов. Отмечается, что последнее время вместо того, чтобы увеличивать нефтеотдачу, закачиваемые воды приводят к обводнению пластов.
Показаны пути уменьшение обводненности пластов и о необходимости усовершенствование методов заводнения.
Ключевые слова: нефтяной пласт, обводнение, нефтеотдача.

UZUN MÜDDƏT İSTİSMARDA OLAN YATAQLARA VURULAN SUYUN EFFEKTLİYİNİN ARTİRİLMASİ

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XÜLASƏ

Məqalədə uzun müddət istismarda olan neft yataqlarına su vurma üsulunu tətbiq etməklə neftveriminin yüksəldilməsindən bəhs edilir. Qeyd edilir ki, son zamanlar vurulan suyun neftverimini artırmaq əvəzinə layın sulaşdırılmasına səbəb olduğu aydınlaşır. Layların sulaşmasının qarşısının alınması üçün su vurma üsulunun təkmilləşdirilməsinin vacib olduğu göstərilir.

Açar sözlər: neft yatağıq, sulaşma, neft verimi.

EFFECT OF KEROSENE MASS FLOW RATE AND AVERAGE TEMPERATURE IN DIRECT CONTACT HEAT EXCHANGER

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ABSTRACT

Industrial plants are huge consumers of energy, these plants are installing heat exchangers in an effort to reduce energy consumption, and so improve operating efficiencies. Limited numbers of experimental and numerical investigations have dealt with the parameters affecting the heat transfer aspects in single phase direct contact heat exchangers which may be selected for their high thermal efficiency and minimum capital investment. In oil refineries energy from hot temperature oil streams can be recovered by transfer directly to a cheap coolant liquid in liquid-liquid direct contact heat exchangers. The heat recovered from these heat exchangers has different applications including preheating boiler feed water and preheating wash water. Heat recovery from hot temperature refinery products using direct contact heat exchanger throughout a theoretical phenomenological study is central to the theme of this paper. Kerosene-water system has been chosen. The effect of the heating fluid inlet temperature (65-97.50)°C, and mass flow rate (25 to 45) kg/s on direct contact heat exchanger design parameters and heat transfer characteristics were investigated theoretically throughout nine cases. Correlations of heat recovered from the system as well as design and operating characteristics of the heat exchanger were estimated. Increasing kerosene flow rate found to associate directly with increasing the contact surface area, number of plates, number of channels per pass and pressure drop, while when the heat exchanger is designed to operates at high kerosene inlet temperature, big heat exchangers with large areas, high number of plates and channels per pass are needed for efficient heat exchanger performance. Optimization and modeling the effect of kerosene operating variables on heat recovered was conducted using Response Surface Methodology. The results showed that an optimum heat recovery value of 6.8782 megawatt could be achieved for kerosene optimum inlet temperature (91.82°C), and mass flow rate (50.11 kg/s).
Keywords: direct contact heat exchanger; kerosene and water system, kerosene mass flow rate and inlet temperature.

Introduction: In direct contact heat exchangers, the two fluid streams come into direct contact, the heat exchange process takes place, and the heat exchange is separated after completion. If we compare recuperators and regenerators with direct contact heat exchangers, we can see that very high heat transfer rates can be achieved in single-contact heat exchangers, the construction of the exchanger is relatively inexpensive, and there is no contamination problem due to lack of heat transfer between the two fluids. [3]

Mixer-type heat exchangers have some advantages and disadvantages.
Advantages:
1. Low corrosion, pollution
2. Heat transfer is higher
3. Low pressure drop
Disadvantage:
1. The pressures of the fluids supplied to the nozzle must be the same
2. The heat exchange that results from the mixing of two liquids, and the separation of liquids after heat exchange, is a problem.
3. In this type, the mixing of two liquids is short-lived in some places. In some places, the mixing of two liquids is long-lasting. As a result, more heat is exchanged when the two fluids in contact.

Immiscible Fluid Exchangers: In this type, two immiscible fluid streams are brought into direct contact. These fluids may be single-phase fluids, or they may involve condensation or vaporization. Condensation of organic vapors and oil vapors with water or air are typical examples.

Gas–Liquid Exchangers: In this type, one fluid is a gas (more commonly, air) and the other a low-pressure liquid (more commonly, water) and are readily separable after the energy exchange. In either cooling of liquid (water) or humidification of gas (air) applications, liquid partially evaporates and the vapor is carried away with the gas. In these exchangers, more than 90% of the energy transfer is by virtue of mass transfer (due to the evaporation of the liquid), and convective heat transfer is a minor mechanism. A ‘‘wet’’ (water) cooling tower with forced- or natural-draft airflow is the most common application. Other applications are the air-conditioning spray chamber, spray drier, spray tower, and spray pond. [9]

Liquid–Vapor Exchangers: In this type, typically steam is partially or fully condensed using cooling water, or water is heated with waste steam through direct contact in the exchanger. Noncondensables and residual steam and hot water are the outlet streams. Common examples are desuperheaters and open feedwater heaters (also known as deaeraters) in power plants. [6]

Effect of Kerosene Mass Flow Rate: The estimated correlations for the effect of mass flow rate of kerosene at constant kerosene inlet temperature (70°C) are shown in Figures (4-9). An increase in mass flow rate of kerosene resulted in an increase in heat exchanger performance (heat recovered) as shown in Figure 8. The reason behind that is attributed to that high mass flow rate of kerosene will promote turbulence and good thermal contact/transfer. However, with higher values of flow rate, adequate importance should be given to heat exchanger design characteristics to achieving optimum performance. As seen, increasing kerosene flow rate is associated directly with increasing the contact surface area (Figure 6). Moreover, increasing kerosene mass flow rate resulted in increasing the pressure drop (Figure 9). It is well noticed that as a fluid flows through a heat exchanger there will normally be a pressure drop in the direction of the flow. The pressure drop is usually affected by type of flow (laminar or turbulent) and the passage geometry. The reason behind increasing the pressure drop with increasing kerosene mass flow rate may be due to the ununiformed flow rate distribution at high flow rate values. High pressure drop will lead to high velocity at the exit of the heat exchanger that resulted in erosion problems and increase the pumping costs. It is worthy to mention that the effectiveness of heat transfer is gauged by how well getting returns for what spending. Hence, the increased pressure gradient is usually outweighed by a decrease in required passage length so the overall pressure drop remains acceptable (Bassiouny and Martin, 1984). [1]

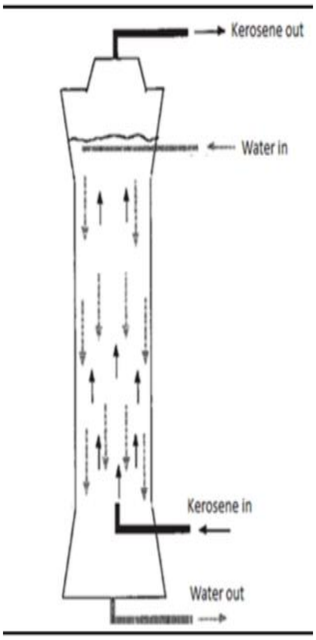


Figure 1. Spray direct contact heat exchanger.

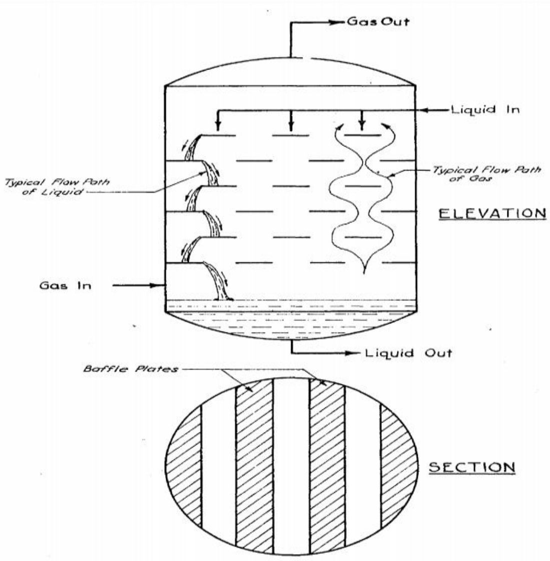


Figure 2. Baffle direct contact heat exchanger.

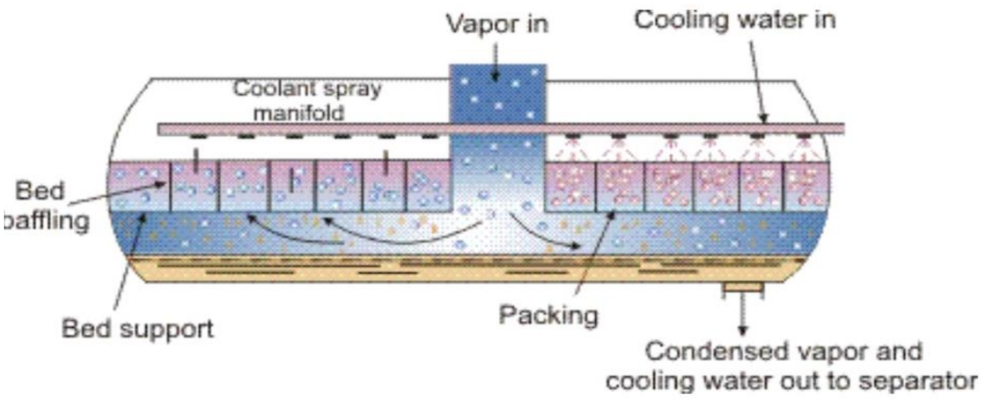


Figure 3. Packed direct contact heat exchanger.

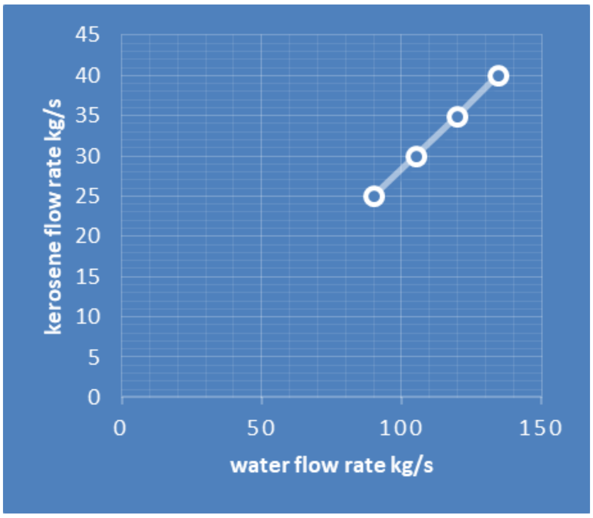


Figure 4. Variation of kerosene mass flow rate with the water flow rate.

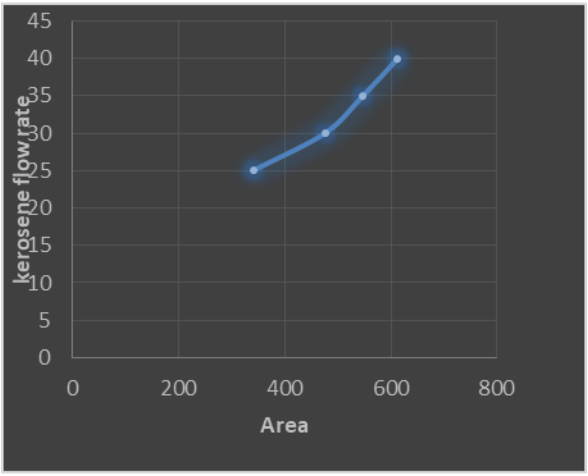


Figure 5. Variation of kerosene mass flow rate with the surface area of the heat exchanger.

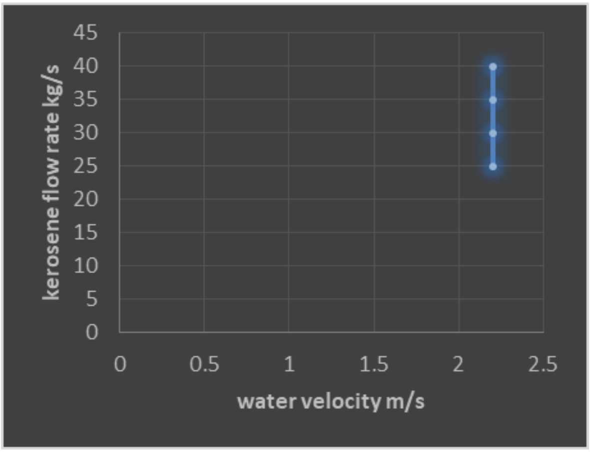


Figure 6. Variation of kerosene mass flow rate with the water velocity.

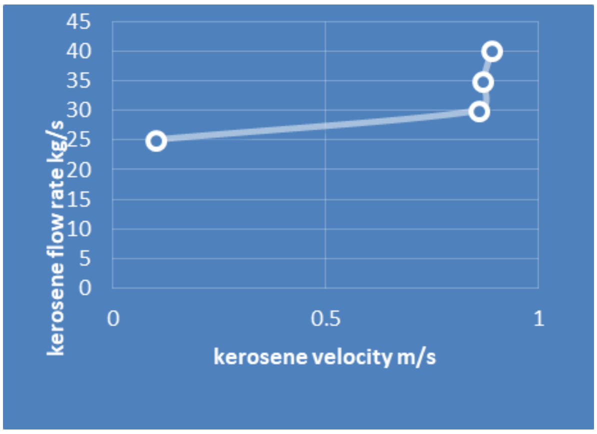


Figure 7. Variation of kerosene mass flow rate with kerosene velocity.

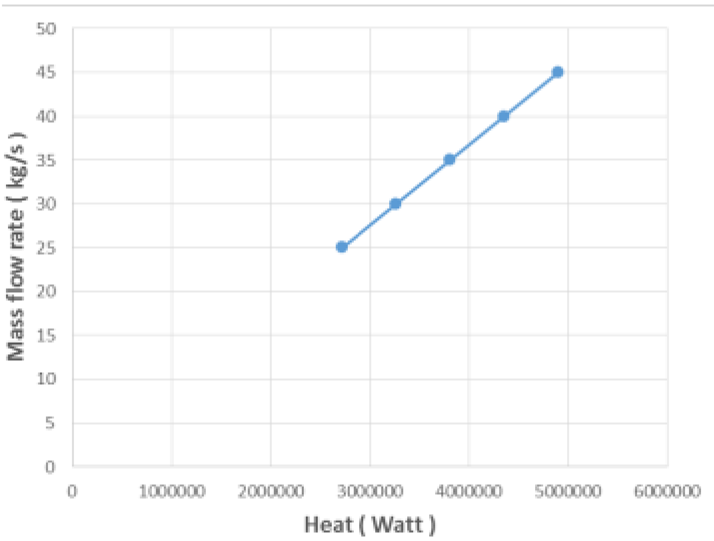


Figure 8. Variation of kerosene mass flow rate with the heat recovered.

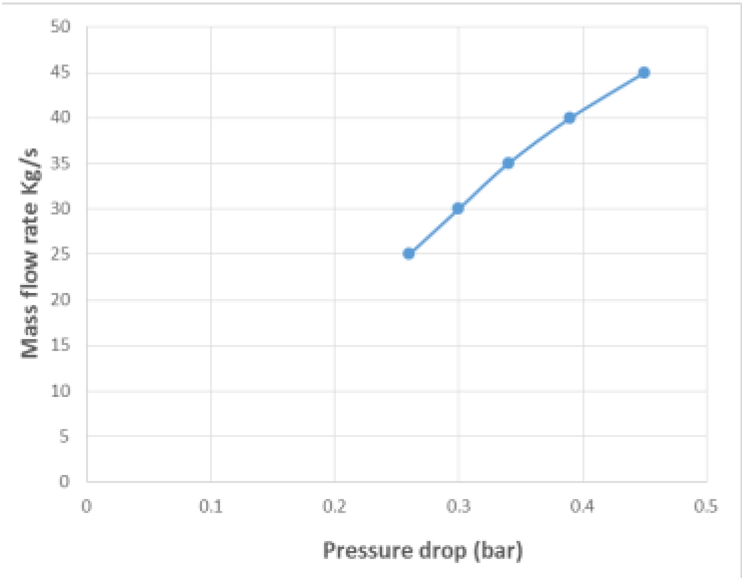


Figure 9. Variation of kerosene mass flow rate with the pressure drop.

Effect of Kerosene Average Temperature: The estimated correlations for the effect of kerosene inlet temperature at constant kerosene mass flow rate (25) kg/s are shown in Figures (10-12). At constant mass flow rate of kerosene the heat recovered increases when kerosene temperature increases as shown in figure 7. Generally, any alterations in the stream temperature will create a variation in the exchanger duty and log mean temperature difference (Dahran, 2017). When the heat exchanger is designed to operates at high kerosene inlet temperature, big heat exchangers with large areas (Figure 10), kerosene velocity (Figure 11) and water velocity (Figure 12) are needed for efficient heat exchanger performance. On another hand as kerosene inlet temperature increases pressure drop decreases owing to decreasing the viscosity of kerosene with increasing kerosene inlet temperature. However, when the operating variables limits are exceeded, the design characteristics will show dramatic changes (Babu, 2004).

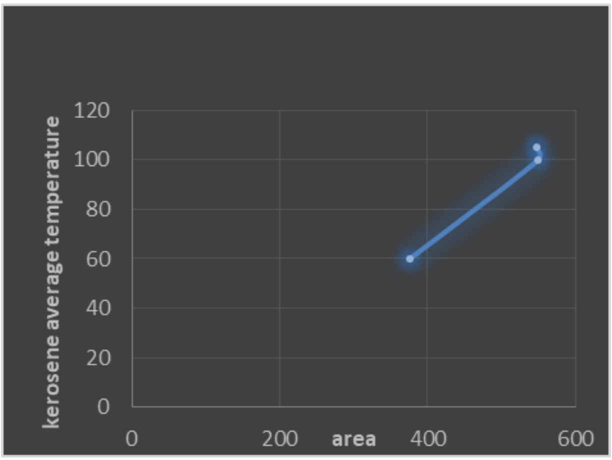


Figure 10. Variation of kerosene temperature with surface area.

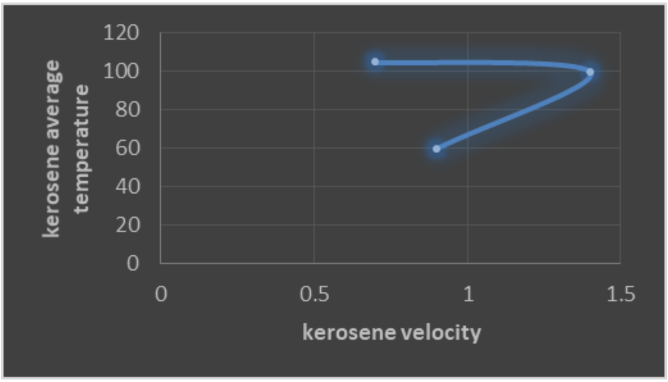


Figure 11. Variation of kerosene temperature with kerosene velocity.

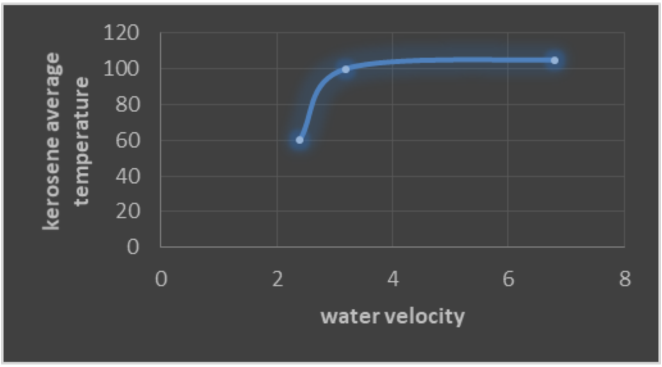


Figure 12. Variation of kerosene temperature with water velocity.

Response Surface Analysis Results: Response Surface Methodology was applied in order to model and optimize the effect of kerosene operating variables including inlet temperature, outlet temperatures and mass flow rate on the heat could be recovered from the heat transfer process. An experimental design was adopted with 16 experiments to study the effect of kerosene inlet temperature ranged (58.18-91.8)°C, kerosene outlet temperature ranged (26.59-43.4)°C, and kerosene mass flow rate ranged (24.88-50.1) kg/s on rate of heat transfer (heat recovered). The heat recovered corresponding to each experiment was calculated theoretically and applied to the software for analysis. Table 1 listed the levels of independent variables and the response for the 10 runs. The results of the heat recovered for all the runs were performed and reported in second-order polynomial multiple linear regression models. The response surface analysis is shown in figure 12 which illustrates the standardized Pareto chart, the main effect plot (general trends), and the estimated response surface. The vertical line on Pareto chart determines the effects that are statistically significant. The standardized effect is the estimated effect divided by its standard error. Hence a low standardized effect can mean either a low effect of the parameter or a large actual error. Pareto chart of standardized effects was calculated in order to show significant effects of all variables (linear, quadratic and interactions between variables). The vertical line represents the limit between the significant and insignificant effectregarding the response. The length of each parameter characterizes the absolute importance of the estimated effects. Moreover, the colour of the squares indicates whether the effect is positive or negative. The Pareto chart in figure 13A showed very strong significant effect of the three variables on the heat recovered. However, the inlet temperature of kerosene seemed of top most significant followed by mass flow rate and outlet temperature.

Table 1. Levels of indepent variables and responses

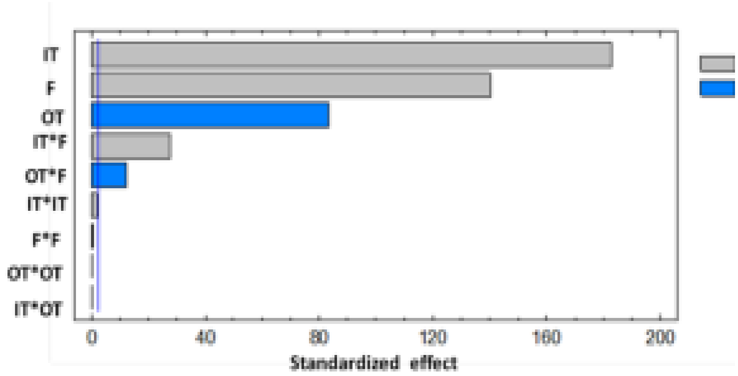
Experiment No.	Inlet temp. °C	Outlet Temp. °C	Flow rate kg/s	Specific heat J/kg.K	Heat Recovered (megawatt)
1	65	30	45	2051.53	3.23
2	85	30	30	2101.77	3.47
3	75	35	24.88	2093.40	2.08
*2	75	35	37.5	2093.34	3.14
6	58.18	35	37.5	2072.47	1.80
7	75	35	50.11	2093.40	4.20
8	91.82	35	37.5	2135.27	4.55
9	75	26.59	37.5	2089.21	3.79
10	85	40	30	2135.27	2.88

The main effects plot (Figure 13B) shows that the amount of heat recovered increases with increasing kerosene inlet temperature and flow rate and decreases with the increasing the out let temperature. Figure 13C shows the estimated response surface at constant kerosene flow rate (37.5) kg/s.

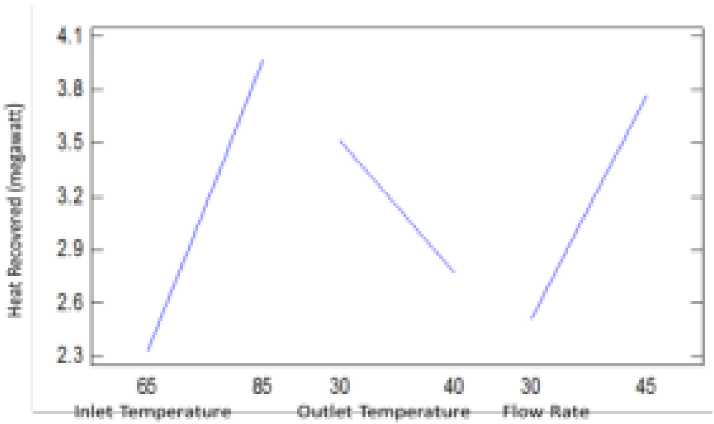
The polynomial empirical model, regression coefficients R2 (adjusted to d.f.), and the optimized values of the heat recovered were estimated from response surface analysis (RSA). The mathematical model correlates the heat recovered with kerosene operating variables is expressed in equation

2. Heat Recovered (megawatt) = 0.559097 - 0.01395·IT - 0.00536·OT - 0.0074·F + 0.000098· IT^2 + 0.0·IT·OT + 0.0022·IT·F + 0.0000398· OT^2 - 0.00193·OT·F - 0.000045· F^2 (2)

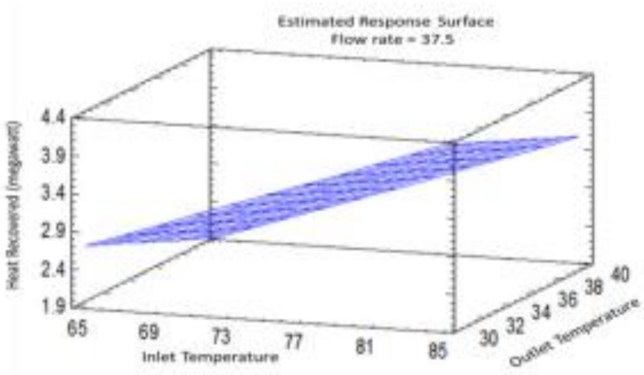
Where, IT is the kerosene inlet temperature, OT is the outlet kerosene temperature and F is the kerosene mass flow rate. Very high regression coefficient was estimated (R-squared = 99.98) which confirms that the model is of high accuracy and capable to perfectly generalize between input and output parameters with reasonable good predictions. An optimum value of heat (6.8782 megawatt) could be estimated from the model for an optimum operating parameters of the heat exchanger at kerosene inlet temperature 91.8°C, kerosene outlet temperature 26.81°C, and kerosene mass flow rate 50.1 kg/s as estimated from RSA. The optimized parameters for kerosene are listed in table 2.



A. Pareto chart for heat recovered



B. General trends



C. Response surface

Figure 13. Pareto chart (A) - general trends (B) - response surface (C) - for effect of kerosene inlet temperature - outlet temperature and flow rate on the heat recovered.

Table 2. Optimum parameters for heat recovery from RSA

Optimum heat recovery = 6.87823 (megawatt) = 6878230 Joule/sec	
Kerosene operating variables	Optimum value
Inlet Temperature	91.82 (°C)
Outlet Temperature	26.81 (°C)
Mass Flow Rate	50.11 (kg/s)

Conclusions: Studies related to the applications of recoverable heat to produce additional power using direct contact heat exchangers are very important from economic and environmental point of view. In oil industry, direct contact heat exchangers are important heat source for providing additional power. Waste heat recovery could take place through a thermal approach that uses the hot temperature oil streams to heat up water and partially fulfill heating or hot water needs of the industrial site. To recover heat from these heat exchangers efficiently, the heat exchangers are needed to be designed in such a way that it can handle the heat load with reasonable size, weight and pressure drop. Upon investigating how physical, design and heat transfer characteristics of single phase kerosene-water direct contact heat exchange system influenced the heat exchangers effectiveness, the design parameters were of significant impact. Also, Statgraphics plus software seemed very applicable to optimize and model the amount of heat recovered as well as kerosene operating variables. With an optimum design of the heat exchanger an optimum heat value (6.88 megawatt) could be recovered when kerosene enters the heat at 91.18°C and flow rate of 50.1 kg/s.

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ЭФФЕКТИВНЫЙ МАССОВОЙ РАСХОД И СРЕДНЯЯ ТЕМПЕРАТУРА КЕРОСИНА В ТЕПЛООБМЕННИКАХ СМЕШЕНИЯ

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АННОТАЦИЯ

Промышленные предприятия являются огромными потребителями энергии, на этих предприятиях устанавливаются теплообменники, чтобы снизить потребление энергии и тем самым повысить эффективность работы. Ограниченное число экспериментальных и численных исследований касалось параметров, влияющих на аспекты теплопередачи в однофазных теплообменниках с прямым контактом, которые могут быть выбраны из-за их высокой тепловой эффективности и минимальных капитальных вложений. На нефтеперерабатывающих заводах энергия из потоков масла с высокой температурой может быть восстановлена путем передачи непосредственно в дешевую охлаждающую жидкость в теплообменниках прямого контакта жидкость-жидкость. Тепло, получаемое от этих теплообменников, находит различные применения, включая предварительный нагрев питательной воды котлакеросин-вода. Влияние температуры нагревательной жидкости на входе (65-97,50)°C и массового расхода (от 25 до 45) кг/с на конструктивные параметры теплообменника прямого контакта и характеристики теплопередачи было теоретически исследовано в девяти случаях. Были оценены соотношения тепла, отводимого из системы, а также конструктивные и эксплуатационные характеристики теплообменника. Установлено, что увеличение расхода керосина напрямую связано с увеличением площади контактной поверхности, количества пластин, количества каналов за проход и перепада давления, в то время как, когда теплообменник предназначен для работы при высокой температуре керосина на входе, для эффективной

работы теплообменника необходимы большие теплообменники с большой площадью, большим количеством пластин и каналов за проход. Оптимизация и моделирование влияния рабочих параметров керосина на утилизацию тепла проводились с использованием методологии поверхности отклика. Результаты показали, что для оптимальной температуры на входе керосина (91,82°C) и массового расхода (50,11 кг/с) может быть достигнуто оптимальное значение рекуперации тепла в размере 6,8782 мегаватт.
Ключевые слова: теплообменник смешения, система керосина и воды, массовый расход керосина и температура на входе.

BİR BAŞA KONTAKLI İSTLİKDƏYİŞDİRCİLƏRDƏ KEROSİNİN EFFEKTİV KÜTLƏ SƏRFİ VƏ ORTALAMA TEMPERATURU

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XÜLASƏ

Böyük enerji istehlak edən müəssələrdə sərf olunan enerji düzgün istifadə olunmasına görə istilikdəyişdirici aparatlardan istifadə edilir. Səbəb olaraq bunu demək olarki daha az enerjidən istifadə ediləcək maksimum effektivlil əldə olunsun. Tək fazalı olan istilikdıyişdiricilərdə minimal kapital qoyuluşu ilə maksimum termal effektivlik alınsın. Tək fazalı dedikdə maye ilə maye arasında olur. Ümumiyyətlə istilik mübadiləsi qızmar maye axın tərəfindən istilik verilir, soyuq axın tərəfindən həmin istilik udulur və nəticədə qızmar axın soyuq vəziyyətdə ştuserdən çıxır, soyuq axın isə qızmar vəziyyətdə ştuserdən xarici olunur. Bu məqalə kerosin və su sistemi seçilmışdır. Əsas məsələ bir başa kontaklı istilikdəyişdirici aparatlarda kerosinin effektiv kütlə sərfi və ortalama temperaturu seçilməsidir. Bunun üçün Kerosinin giriş temperaturunu (65-97.50)°C, kütlə sərfini isə (25-45) kg/s götürüləcək bir başa kontaktı istilikdəyişdircinin parametrlərini və istilik mübadiləsinin xarakterlərinin tapılması. Kerosinin kütlə sərfi, və giriş temperaturunu dəyişməsi nəticəsində bu zaman suyun kütlə şərfi, sürətinin və suyun ortalama temperaturunu və təzyiq itkisinin, kerosinin sürəti, istilik dəyişdiricinin en kəsiyinin sahəsinin aslıqlarının üzərində araşdırılması aparılmışdır. Optimalışdırılma və moderzasiya aparılması üçün Sahə metodologiyasından istifadə olunaraq tapılır. Bu metod sahəsində optimal istilik miqdarı 6.8782 megawatt seçilir, kerosinin optimal giriş temperaturu(91,82°C) , optimal kütlə sərfi isə(50.11 kg/s) tapılmışdır.
Açar sözlər: bir başa kontaklı istilik dəyişdiricisi, kerosin və su sistemi, kerosinin kütlə sərfi və giriş temperaturu.

DESIGN OF CENTRIFUGAL COMPRESSOR IMPELLER FOR OPTIMAL EFFICIENCY

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ABSTRACT

This research present a case steady namely design of centrifugal air compressor impeller for power station. This compressor is a dynamic compress which depends on a rotating impeller to compress the air. Impeller is the most important part of the centrifugal compressor components. Detail design calculation of centrifugal compressor impeller is described in this research. This study contains a complete set of detail drawing for blade profile of impeller. It can be used at sites which flow rate is 0.1275 m3 /s and 45561 rpm. The required data are collected from Ywama Power Station which is located in Yangon. For the given capacity, the inlet and outlet diameter are 0.054 m and 0.17 m and the number of blade is 19.
Keywords: centrifugal compressor, impeller, velocity, diameter, width.

Introduction: A compressor is a piece of machinery that compresses a fluid, a liquid of a gas that flows in the compressor into greater pressure. During the past 30 years, the centrifugal compressor because of its simplicity and larger capacity/size ratio, compares to the reciprocating machines, became much more popular for use in process plants that were growing in size. Centrifugal compressor is one of the oldest turbo machinery, widely used in various industries like, aviation, oil and gas, refrigeration, etc. Centrifugal compressor is a radial turbo machine, which compresses air or gas with the action of centrifugal force. During the Second World War, the centrifugal compressors were used by British and American fighter aircrafts, as a part of early development of gas turbine engines. Later, during the 1950s, a large number of turboprop, turbofan, turbo-shafts and auxiliary power units started using the centrifugal compressors for air compression due to their high pressure raising capability in a single stage and their robustness in case of foreign object damage.

Centrifugal compressors are a key piece of equipment for modern production. Among the components of the centrifugal compressor, the impeller is a pivotal part as it is used to transform kinetic energy into pressure energy. Impeller is an active part that adds energy to the fluid, its geometry plays a major role in the centrifugal compressors performance. An impeller is a wheel or rotor which is provided with a series of backward curved blades or vanes. It is mounted on a shaft which is couple to on external source of energy which imparts the required energy to the impeller there by making it to the rotate. The impellers may be classified as;

- Shrouded or closed impeller,
- Semi-open impeller and
- Open impeller.

Specification data:
Inlet pressure, P1 = 1030.765 kPa
Inlet temperature, T1 = 380 K
Rated speed, N = 45561 rpm
Outlet pressure, P2 = 1799.6 kPa
Outlet temperature, T2 = 448 K
Capacity, Q = 0.1275 m3 /s
Air mass flow rate, m = 1.5907 kg/s

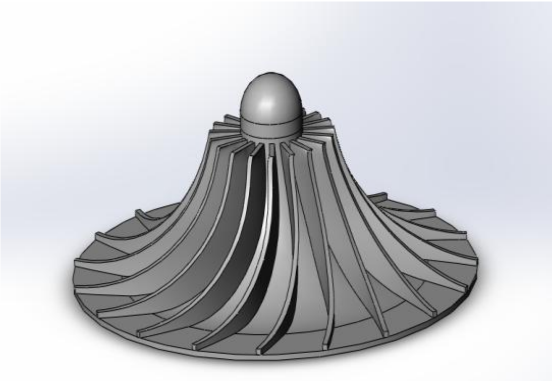

Figure 1. Impeller of centrifugal compressor.

Centrifugal compressor with these specifications has been installed on Ywama Power Station at Insein Township, Yangon, Myanmar.
Methodology: The thermodynamics law fundamental to an understanding of compressor operation is the ideal gas law, which is expressed in equation from the follows;

$$Pv = ZRT \tag{1}$$

The general form of the thermodynamic head equation for a polytropic process is

$$H_p = ZRT_1 \frac{N}{n-1}\left[\left(r_p\right)^{\frac{n-1}{n}} - 1\right] \tag{2}$$

This equation drives from integrating the steady-state, steady flow work equation given by:

$$Hp = \int vdp \tag{3}$$

The polytropic process is of form:

$$Pv^n = \text{constant} \tag{4}$$

A. Impeller Inlet Dimension: Before the impeller dimensions can be fixed, the shaft must first be approximated. The shaft diameter based upon torque alone is given by the equation:

$$D_s = \sqrt[3]{\frac{16T}{\pi S_s}} \tag{5}$$

The eye diameter Do may be found from the continuity equation:

$$\frac{\pi}{4}{D_0}^2 - \frac{\pi}{4}{D_h}^2 = \frac{Q}{V_0} \tag{6}$$

$$D_0\sqrt{\frac{4\times Q}{\pi\times V_0}+D_H{}^2} \tag{7}$$

The mean diameter of the vane inlet is made slightly greater than the impeller eye diameter.
Speed of sound of gas, a:

$$a=\sqrt{k\times g\times R\times T_1} \tag{8}$$

The impeller inlet hub Mach number is 0.2 to 1 for compressible fluid. The value of Mach number is 0.3(assumed).
The impeller absolute velocity equation is:

$$V_0=\mathrm{M}\times\mathrm{a} \tag{9}$$

The air enters the impeller eye to tip in the axial direction and prewhirl angle is zero, so that V1=Vf1 and is made slightly greater than Vo.
Impeller Outlet diameter, D2:

$$D_2=\frac{60\times\sqrt{H_p\times g}}{\pi\times n\times\sqrt{K}} \tag{10}$$

Enthalpy and Efficiency:
The greater the number of vanes, the smaller the slip, i.e. the more nearly Vω2 approaches U2. It is necessary in design to assume a value for the slip factor σ;

$$\sigma=\frac{V_{\omega2}}{U_2} \tag{11}$$

The analytical design of impeller inlet result data are expressed by graphs.

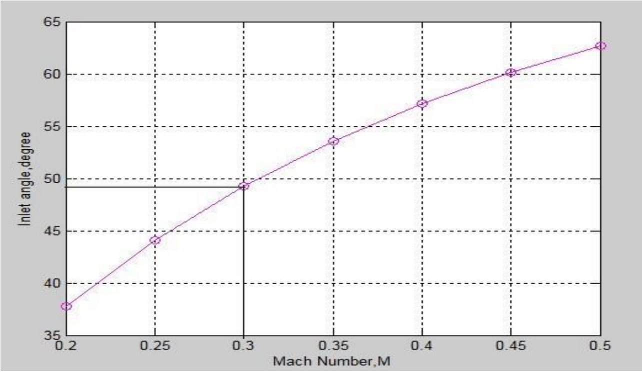

Figure 2. Mach number and Blade inlet angle.

The relation between the blade inlet angle and Mach number are illustrated in figure 2. This graph shows the larger the blade inlet angle, the higher the Mach number.

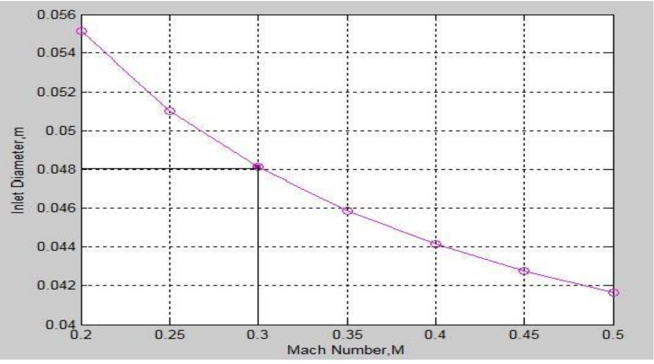

Figure 3. Mach number and Inlet diameter.

The relation between the inlet diameter and Mach number are illustrated in figure 3. This graph shows the smaller the inlet diameter, the higher the Mach number.

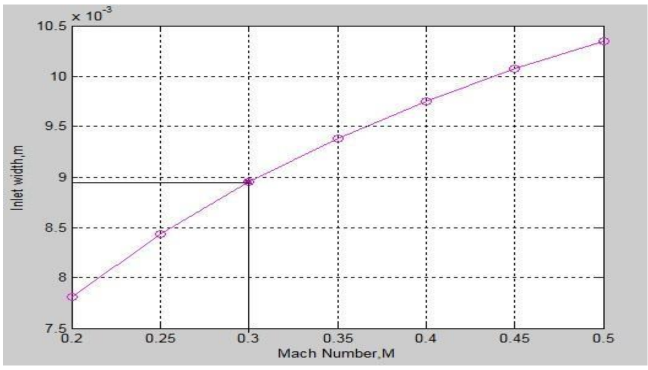

Figure 4. Mach number and Inlet width.

The relation between the inlet diameter and Mach number are illustrated in figure 4. This graph shows the greater the inlet diameter, the higher the Mach number. Based on these conditions Mach number, M is 0.3, at that point of data is nearly equal to the actual of the centrifugal compressor impeller inlet. Therefore, the design is satisfied for at that point of data.

Figure 5 shows inlet and outlet velocity triangle ofcentrifugal compressor impeller.

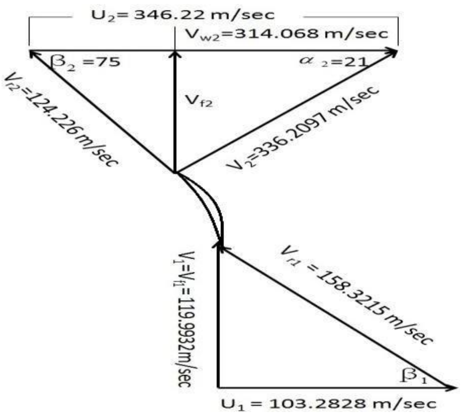

Figure 5. Inlet and outlet velocity triangle of impeller.

Inlet blade angle, β_1

$$\beta_1 = tan^{-1}\frac{V_1}{U_1} \tag{12}$$

Outlet blade angle, β_2

The compressor industry commonly uses a backward leading blade with angle, β_2 of between about 55-75 deg. The blade outlet angle of 75 deg is maximum power position. Therefore, the maximum design condition the outlet blade angle, β_2 =75 deg.

Theoretical and numerical results:

A. Theoretical Results

Table 1

Calculated data of impeller

no	Design Parameter	Symbol	Values	Units
1	Polytropic head	H_p	67.411	kNm/kg
2	Torque	T	32.906	N-m
3	Speed of sound	a	390.221	m/s
4	Mach number	M_1	0.3	-
5	Inlet velocity	U_1	103.283	m/s
6	Absolute velocity at inlet	V_1	119.993	m/s
7	Relative velocity at inlet	V_{r1}	158.322	m/s
8	Inlet blade angle	β_1	49.5	deg
9	Outlet velocity	U_2	346.22	m/s
10	Absolute velocity at outlet	V_2	336.209	m/s
11	Relative velocity at outlet	V_{r2}	124.226	m/s
12	Outlet blade angle	β_2	75	deg
13	Outlet Mach no;	M_2	0.8	-
14	efficiency	η	94	%

B. Numerical Results

Pressure Distribution

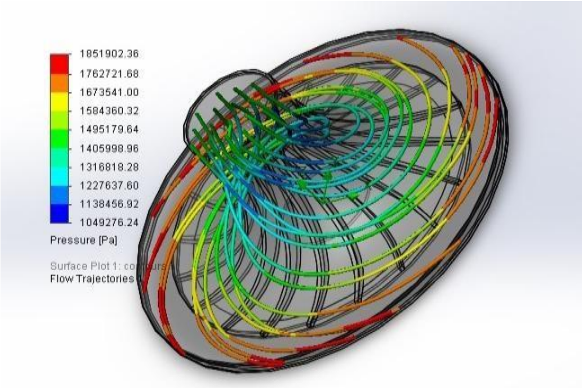

Figure 6. Pressure distribution of centrifugal compressor impeller (flow trajectories).

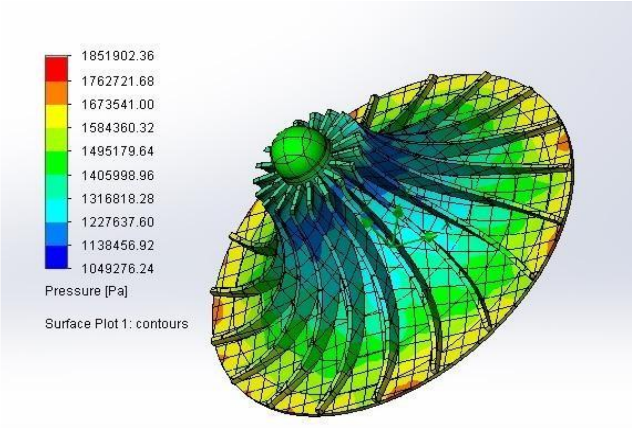

Figure 7. Pressure distribution of centrifugal compressor impeller (surface plots).

Figure 6 and 7 show the pressure distribution of centrifugal compressor impeller by using Solid Works software. To run these flow simulations the input data are mass flow rate. The existing value of impeller inlet and outlet pressure is nearly equal with numerical research value.

Conclusions: Centrifugal compressors are compressible flow machine. Centrifugal compressor from 'Ywama Power Station' is designed in this paper. This paper is attempted to design a single stage centrifugal compressor from 'Ywama Power Station'. The design of impeller in this paper is inlet and outlet diameter and number of blades and blade width. This paper describes the inlet and outlet velocity triangle. And this research shows the pressure distribution by using SolidWorks software. Types of compressor are used in many services and power generation. Centrifugal compressors compress air to raise pressure at high speed. This compressor use also main component in aircraft gas turbine. In this paper, the centrifugal compressor is designed with data from 'Ywama Power Station'. The centrifugal compressor is used an auxiliary component.

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КОНСТРУКЦИЯ КРЫЛЬЧАТКИ ЦЕНТРОБЕЖНОГО КОМПРЕССОРА ДЛЯ ОПТИМАЛЬНОЙ ЭФФЕКТИВНОСТИ

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АННОТАЦИЯ

Центробежный компрессор представляет собой высокоскоростную вращающуюся крыльчатку. Их приводы осуществляются электрическими, паровыми или газотурбинными двигателями. Эффективность центробежного компрессора зависит от размера крыльчатки, скорости её вращения и массы. На основании научных теорий были рассчитаны размеры входной и выходной частей крыльчатки, а также угол наклона лопастей крыльчатки и выбран наиболее оптимальный вариант. Затем, исходя из обусловленных данных, в SolidWorks была создана трехмерная форма крыльчатки, которая и была протестирована на производительность.
Ключевые слова: центробежный компрессор, крыльчатка, продуктивность, мощность, диаметр крыльчатки, масса крыльчатки.

OPTİMAL SƏMƏRƏLİLİK ÜÇÜN MƏRKƏZDƏNQAÇMA KOMPRESSORUN İŞÇİ ÇARXININ KONSTRUKSİYASI

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XÜLASƏ

Mərkəzdənqaçma kompressoru yüksək sürətlə fırlanan işçi çarxa malik intiqqallı qurğudur. Onların intiqalları elektrikli, buxar turbinli və ya qaz turbinli mühərriklər tərəfindən idarə olunur. Mərkəzdənqaçma kompressorun məhsuldarlığı işçi çarxın ölçüsündən və fırlanma sürətindən həmçinin işçi çarxın kütləsindən yüksək dərəcədə asılıdır. Buna görədə işçi çarxın

optimal formada layihələndirilməsi ən əsas məsələlərdən biridir. Bunun üçün işçi çarxın giriş və çıxış hissələrinin ölçüsü və həmçinin işçi çarx üzərində yerləşən pərlərin yerləşmə bucağı elmi nəzəriyyələr əsasında hesablanaraq ən optimal versiya seçilmişdir. Daha sonra təyin olunan qiymətlər əsasında işçi çarxın 3 ölçülü forması SolidWorks proqramında yaradılaraq məhsuldarlıq sınaqdan keçirilmişdir.
Açar sözlər: mərkəzdənqaçma kompressoru, işçi çarx, məhsuldarlıq, işçi çarxın kütləsi, işçi çarxın diametri.

INCREASING THE EFFICIENCY OF RESTORATION OF CYLINDRICAL PARTS ON THE BASIS OF COMPLEX PROCESSING METHOD

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ABSTRACT

The main reason for the failure of oilfield equipment is corrosion. The price of the processing share for mechanical processing of the shaft of the axial-piston pump has been determined. The processing of the worn cylindrical shaft of the pump went through four stages. The most important task of modern repair production is the creation, development and application of new high- efficiency technology, as well as increasing labor productivity, quality of work, reducing material and energy consumption of operations for the restoration of obsolete parts.
Keywords: Axial-piston pump, shaft mechanical processing share, repair work, wear, cylindrical shaft, mechanical processing processes.

Relevance of the issue: Modern repair production forms the most powerful sphere of the economy in terms of its strength, functions and carries out the production of recycling machinery. In connection with the practical significance and prospects for the restoration of worn-out machine parts on the basis of a complex method of processing, restoration technologies and the insufficient development of their equipment tools, the issue posed is very relevant. Taking into account the modern trend of the widespread use of durable, wear-resistant and hard-to-process materials in renewable production, as well as efforts to reduce the costs of energy and material resources, the importance of solving these problems becomes even more urgent.

Modern repair production combines the operation of special technical areas within a large number of independent repair plants and factories. The main work of repair companies is to ensure economic efficiency of machines and longevity, quality of parts. It should be noted that in operation, often the operating time of the repaired details is different from the normative operating time of the detail.According to scientific research works and experience of heavy equipment repair, the manufacturer requires better quality and longevity of the machines and spare parts produced at the plant. Thus, the most important task of modern repair production is the creation, development and application of a new highly efficient technology to production, as well as to take into account the increase in labor productivity, the quality of work performed, the reduction of material and energy consumption of operations for the restoration of worn parts.[1]

The main reason for the failure of oil equipment is eating. Below is the analysis of the recovered part of the axial piston pump – cylindrical shaft.[2]

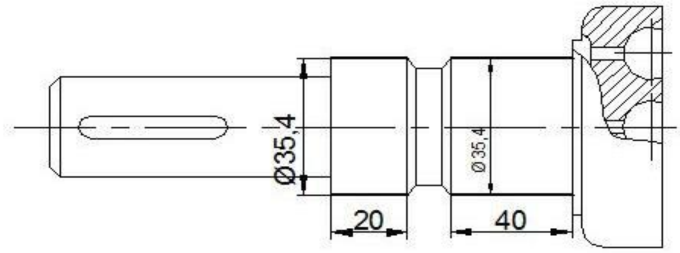

Figure 1. Scheme of the cylindrical shaft of the axial piston pump.

Based on the structure and operating conditions of the detail under consideration, the dimensions were performed in two mutually perpendicular planes in longitudinal parallel to the axis of the valve according to the scheme shown in Figure 2.

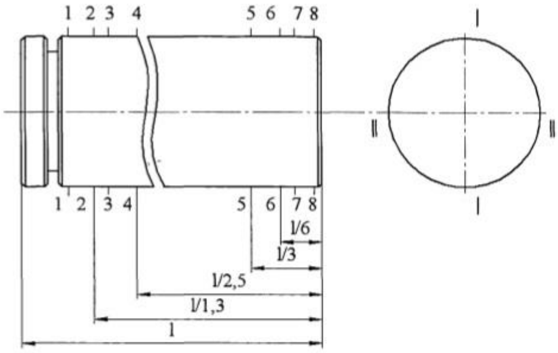

Figure 2. The breakdown scheme of the shaft to determine the magnitude of the meal.

It is necessary to determine the price of the processing share for mechanical processing of the shaft of the axial-piston pump.The cost of processing shares depends on many factors, including the size of the surface eaten, its shape (cylinder instead of cone, oval instead of Circle, etc.).[al-Bukhari, Ahmad, at-tirmithi, at-Tabarani and Abu Nu'aym]s.[3] therefore, the high precision and roughness class required for mechanical processing is indicated on the part.For mechanical processing of external and internal rotation surfaces, the reporting formula of processing Stakes is adopted

$$2z_{\min} = 2\left(R_{zi} + T_{i1} + \sqrt{\rho_{i-1}^2 + \varepsilon_i^2}\right) \tag{1}$$

here Rzi - surface roughness, Rz=50 mkm ; Ti1 – defective thickness, Ti1= 50 mkm; ρi-1 – there are spatial errors, mkm; it is defined as follows:

$$\rho_{i-1} = \sqrt{\rho_{s\ddot{u}r}^2 + \rho_{kor}^2} \tag{2}$$

where $\rho_{s\ddot{u}r}$ – drift of the index or axis, mm; $\rho_{s\ddot{u}r}$ = 0,025; ρ_{kor} -error in bending of the axis of the surface during heating.

Δ = 0,1 mkm – curvature of the axis, falling to a length of 1 mm.

Common spatial error

$$\rho_{i-1} = \sqrt{25^2 + 22{,}5^2} = 33 \text{ mkm}$$

$$\varepsilon_i = \sqrt{\varepsilon_b^2 + \varepsilon_{bz}^2 + \varepsilon_{tz}^2}, \quad (3)$$

where ε_i – is the error arising from the placement of the paste; ε_{bz}- error due to the base of the paste, ε_{bz} = 50 mkm; ε_{tz} – microns; εtz - error caused by the design or other devices, ε_{tz} = 0.

Thus, the error caused by the installation

where ε_i – pastah deployment error; ε_{bz} – pastah optimization error, ε_{bz} = 50 mkm; etz-design or other devices error, etz = 0.

Thus, the error caused by the installation

$$\varepsilon_i = \sqrt{50^2} = 50 \text{ mkm}$$

The smallest share of processing will be as follows

$$z_{\min} = 50 + 50 + \sqrt{33^2 + 50^2} = 160 = 0{,}16 \text{ mm}$$

The largest processing share

$$2z_{\max} = 2z_{\min} \pm \sigma_{i-1} + \sigma_i, \quad (4)$$

σ_{i-1} – the size of the surface, σ_{i-1}=0,032 mm.

σ_i – the default setting of the initial operation, σ_i =0,01 mm.

$$z_{\max} = 0{,}32 + 0{,}032 + 0{,}01 = 0{,}362 \text{ mm}$$

Processing process of eaten shaft of axial-piston pump consists of 4 stages:

1-rough polishing (before gas-flame eruption);

2 - gas - flame spraying;3-rough polishing;4-clean polishing.

Rough polishing operation

For the polishing operation, we select the polishing machine brand 3m151.

Main technical characteristics of machine tools:

The largest dimensions of the pedestal placed in the bench (mm):

diameter-200 mm; length-700 mm

rotation speed of pardach stone-1590 rpm.

the largest dimensions of the Diamond Stone (mm) outer diameter-600; height-80;

rotation speed of the part – 50 ÷ 500 rpm.

adjustment of rotation of the part –without batteries.

Taking into account the technical characteristics of the masonry, we select the parameters of the masonry: diameter of the masonry Dp = 250mm, height B = 50mm and speed of the masonry masonry np = 1590 rpm. The speed of the pardax stone on it:

$$Vp = \frac{\pi Dn}{1000 \cdot 60} = \frac{3{,}14 \cdot 250 \cdot 1590}{1000 \cdot 60} = 20{,}8 \mathrm{m/s} \tag{5}$$

We accept the rotation speed of the part Vh =20 m/min and determine the number of rotational periods of the part from the following expression:

$$\mathrm{n} = \frac{1000 \cdot \mathrm{V}}{\pi \mathrm{D}} = \frac{1000 \cdot 20{,}8}{3{,}14 \cdot 250} = 26{,}4 \text{ per/min} \tag{6}$$

The share of mechanical processing for primary processing of the working surface of the plunger: a =0,9 mm.

Determine the depth of cutting in the initial processing editik. Based on the selected bench

Sr =0,05 mm^2/ on the go.

Longitudinal stores S = (0,5-0,8) B = (0,5-0,8)50 =25-40 mm/period

We accept: S = 30 mm/period

Calculation of the time norm

Machine time during polishing of external cylindrical surfaces

$$T_\alpha = \frac{L}{S \cdot n_h} \cdot \frac{a}{S_r} \cdot K \text{, (min)} \tag{7}$$

here L - movement length of the instrument, mm;

$$L = L_h - (0{,}2\text{-}0{,}4)\ 50 = 40 - (10\text{-}20) = 30 \text{ mm}$$

a - one-way total processing share, mm; *a =0,9 mm*

S - longitudinal penetration of the poldach stone, S = 30 mm / period

S_r - radial yield of cut stone (cutting depth); S_r = 0.05 mm / period

n_h - number of cycles of the part, rev / min; $n_h = 26{,}44\ rev\ /\ min$

K- measurement coefficient; for initial processing. *K=1,4*

For initial polishing operation

$$T = \frac{30 \cdot 0{,}9}{30 \cdot 26{,}4 \cdot 0{,}05} \cdot 1{,}4 = 0{,}96\ min$$

Gas-flame coating process

Time spent on the process of gas – flame coating:

$$T = \frac{60 \cdot \pi \cdot D_H \cdot l \cdot h \cdot \gamma \cdot k}{10^6 \cdot G \cdot K_H}, min \tag{8}$$

$D_н$ = 40 mm;

l = 40 mm;

h = 1,2 mm;

γ = 6,7 q/sm^3;

G = 5,0 kg/saat - equipment productivity;

k = 1,8;

$k_н$ = 0,7.

Time spent on the process of gas – flame coating:

$$T = \frac{40 \cdot 3{,}14 \cdot 120 \cdot 40 \cdot 1{,}8 \cdot 6{,}8 \cdot 1{,}1}{10 \cdot 5 \cdot 0{,}7} = 2{,}32\ min$$

Rough polishing operation

$$L = L_h - (0{,}2\text{-}0{,}3)\ B = 40 - (10\text{-}15) = 30\ mm$$

a - one-way total processing share, mm; *a =0,3 mm*

S - longitudinal penetration of the poldach stone, S = 30 mm/period

S_r - radial yield of cut stone (cutting depth); S_r = 0.05 mm/ mileage

n_h - number of periods of part, döv/dəq; n_h =26,54 $dövr/dəq$

K- measurement coefficient; for initial processing. *K=1,4*

Time norm for rough polishing operation

$$T = \frac{30 \cdot 0{,}3}{30 \cdot 26{,}4 \cdot 0{,}01} \cdot 1{,}4 = 1{,}59\ min$$

Clean polishing operation

$$L = L_h - (0{,}2\text{-}0{,}3)\ B = 40 - (10\text{-}15) = 30\ mm$$

a - one-way total processing share, mm; *a =0,1 mm*

S - longitudinal shopping of the gilded stone, S = 30 mm/period

S_r - radial flow of cut stone (cutting depth);

S_r = 0.005 mm/ departure

n_h - number of periods of part, döv/dəq; n_h =26,4 $period\ /\ min$

K- measurement coefficient; for initial processing. *K=1,6*

Clean polishing operation

$$T = \frac{30 \cdot 0{,}1}{30 \cdot 26{,}4 \cdot 0{,}002} \cdot 1{,}6 = 3{,}03\ min$$

Number time norm

$$T_{\alpha d} = T_{\alpha} + T_k + T_{texn.v} + T_f = T_{op} + 7\% T_{op}\ . \qquad (9)$$

Here T_a -main technological time.

T_k - auxiliary time ; T_k =20% T_a

T_{op} - operative time. $T_{op} = T_a + T_k$

T_{tqv} - maintenance time.

T_f – time of personal demand.

Main technological time:

$$T_a = 0{,}96+1{,}59+3{,}03 = 5{,}58\ min$$

Auxiliary time:

$$T_k = 20\%\ T_a = 1{,}16 \text{ min}$$

Operative time:

$$T_{op} = 5{,}58 + 1{,}16 = 6{,}74 \text{ min}$$

Maintenance and personal demand times:

$$T_{tqv} + T_f = 0{,}2 \cdot 6{,}74 = 1{,}35 \text{ min}$$

Number of time norm related to shaft mechanical processing

$$T_{əd} = 5{,}58 + 1{,}16 + 1{,}35 = 8{,}09 \text{ min}$$

Total number of time norm taking into account the amount of time spent on spraying of the regenerated surface of Shaft:

$$T_{əd} = 8{,}09 + 2{,}32 = 10{,}41 \text{ min}$$

Conclusion: Recovery method for restoration of cylindrical shaft on the basis of complex processing method of axial piston pump has been substantiated and selected, restoration technology has been developed that takes into account the processes of pre-and post-restoration mechanical processing of Shaft.Calculating the shares of mechanical processing, the smallest processing share was obtained at prices of 0,16 mm, and the largest processing share 0,362 mm.The error received from the installation was found by calculating 50 mcm.

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ПОВЫШЕНИЕ ЭФФЕКТИВНОСТИ ВОССТАНОВЛЕНИЯ ДЕТАЛЕЙ ЦИЛИНДРОВ НА ОСНОВЕ КОМПЛЕКСНОГО МЕТОДА ОБРАБОТКИ

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АННОТАЦИЯ

Основной причиной выхода из строя нефтепромыслового оборудования является коррозия. Определена цена доли обработки на механическую обработку вала аксиально-поршневого насоса. Обработка изношенного цилиндрического вала насоса прошла четыре этапа. Важнейшей задачей современного ремонтного производства является создание, освоение и новой высокоэффективной техники, а также повышение производительности, качества работ, снижение материало- и энергоемкости операций по восстановлению морально устаревших деталей.
Ключевые слова: аксиально-поршневой насос, доля механической обработки вала, ремонтные работы, износ, цилиндрический вал, процессы механической обработки.

KOMPLEKS EMAL METODU ƏSASINDA SİLİNDRİK HİSSƏLƏRİN BƏRPASININ EFFEKTİVLİYİNİN ARTIRILMASI

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XÜLASƏ

Neft-mədən avadanlıqlarının sıradan çıxmasının əsas səbəbi yeyilmədir.Aksial porşenli nasosun bərpa edilən hissəsi – silindrik valı üçün aparılan təhlillər verilmişdir. Aksial-porşenli nasosun valının mexaniki emalı üçün emal payının qiyməti təyin edilib. Nasosun yeyilmiş silindrik valının emal prosesi dörd mərhələdən keçmişdir. Müasir təmir istehsalatının ən mühüm vəzifəsi yeni yüksək səmərəli texnologiyanın yaradılması, inkişafı və istehsalata tətbiqidir, həmçinin əmək məhsuldarlığının artımını, görülən işlərin keyfiyyətini, köhnəlmiş hissələrin bərpası üçün əməliyyatların material və enerji sərfiyyatının azaldılmasını nəzərə almaq lazımdır.
Açar sözlər: aksial-porşenli nasos, valın mexaniki emal payı, təmir müəssisələrinin işi, yeyilmə, silindrik val, mexaniki emal prosesləri.

EQUIPMENT. TECHNOLOGIES. MATERIALS

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INVESTIGATIONS OF THE MAIN CAUSES OF FAILURES OF PARTS AND ASSEMBLIES OF OILFIELD PUMPS

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ABSTRACT

Currently, the extraction of hydrocarbons is carried out from deep layers, and this in turn requires an increase in the bearing capacity of both drilling and operational equipment. The group of these equipment includes oilfield pumps, the hydraulic part of which has low operating time. The article presents the results of studies of the main causes of failures of parts and assemblies of oilfield pumps.

Keywords: oilfield pumps, failure, failure distribution, wear, breakdown.

Relevance of the topic: Currently, more than 80 oil and gas fields have been explored and in large part are being exploited on land and in the Azerbaijani sector of the Caspian Sea [1]. Oil and gas fields located on land, especially on the Absheron Peninsula, are at a late stage of exploitation [2]. The main share of oil and gas production in the republic is carried out at offshore fields, where a large number of different equipment are used, including oilfield and mud pumps. It has been established [3] that many parts and assemblies of these pumps during operation are subjected to intense wear, frequent breakdowns, various nature of corrosion effects, which leads to their failure.

To increase the service life and performance characteristics of oil and gas field equipment (OGFE) both their structural improvement and to increase the characteristics of the materials used are required. An example is a comparison of the U8-4 drill pump with the U8-6m pump, where the latter improved the design of the parts, which increased the service life. However, there is still a drawback in the form of a low service life of the valves.

There are other disadvantages in such pumps, in the form of mutual influence of one part on another such as: rod-seals, piston-cylinder bushing.

The purpose of the work: Investigation of the performance of oilfield pumps and the establishment of the main causes of their failures and the development of recommendations for their elimination.

Methods of conducting investigation. Increasing the operational depth of oil wells requires an increase in the bearing capacity of the equipment. The groups of these equipment include oilfield pumps, the main indicators of which are the provision of high pressures, pressure and supply.

Despite the current breakthrough in the field of production and processing of materials, the hydraulic part of oilfield pumps has low operating times for failure. The main reasons for these failures arise as a result of the wear of the abrasive layer, waterjet, hydroerosion-abrasive, shock -abrasive wear of their working surfaces

Oilfield pumps operate at high pressures, which creates a large load on the structural parts. With heavy loads, there are also large forces of interaction between the parts of the same unit, which leads to wear. With the time of operation of the equipment under heavy loads, wear and tear also increases. When the sealing part of the structure is destroyed, the liquid penetrates into the rest of the pump and acts on them under high pressure, which leads to the immediate failure of all equipment.
To date, the three-plunger pump H5-160 is used in the oil fields of Azerbaijan, the principle of operation of which is based on the conversion of mechanical energy into the energy of the fluid flow by cyclically changing the volume of the working chamber. This pump is used when pumping process water into a well, washing sand plugs, as well as during major repairs of wells, is used for pumping cementing fluid.
The main problems of this type of pumps are: a low reliability reserve of the elements of the hydraulic part, the ability of the structure to resist the emerging loads and resistance to pumped liquids. To reduce the load on the parts of the unit and prevent leaks, various types of seals are used. In this pump there are cuff seals that help reduce the friction force between the plunger and the cylinder, but there is also a lack of these seals, in the form of tightening abrasive particles under the cuffs, leading to an increase in the intensity of plunger wear.

Discussion of research results: Table 1 shows the results of statistical processing of information on the operating time of pump parts of the H5-160 brand.
When testing the investigated pump H5-160, it was found that under pressure 20 ... 25 MPa, the average operating mean time to failure of the discharge valves is 130.5 h., and the suction valve is 172 h., which is a small resource. The mean time between failures of other parts of the hydraulic part of the pump also became known.
Data on the remaining parts: the mean time between failures of the plunger is - 257.3 h., the plunger seals according to the Weibull distribution law - 45.5 h., the valve plate according to the law of logarithmic-normal distribution is 305 h. In general, the hydraulic part has a mean time between failures of 13.9 h., according to Weibull's law of distribution.

Table 1. Resource requirements by component

Detail	Law of distribution	Statistical distribution parameter	
		Operating time to failure, h.	Coefficient of variation h.
Piston	Exponential	98	0,83
Cylinder bushings	Weibull	204	0,52
Stocks	Logarithmic-normal	106	0,48
Valves	Exponential	73	0,77

Results of statistical processing of information on the operating time of the H5-160 pump parts.
Thus, the results of the conducted research, as well as a broad review of the literature, allowed us to establish that the piston and valve assemblies of pumps are very vulnerable and have a small operating time for failure. Increasing their service life is relevant and requires careful study.

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ИССЛЕДОВАНИЯ ОСНОВНЫХ ПРИЧИН ОТКАЗОВ ДЕТАЛЕЙ И УЗЛОВ НЕФТЕПРОМЫСЛОВЫХ НАСОСОВ

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АННОТАЦИЯ

В настоящее время добыча углеводородов осуществляется из глубоких пластов, а это в свою очередь требует повышение несущих способностей как бурового, так и эксплуатационного оборудования. В группу этих оборудований входят нефтепромысловые насосы, гидравлическая часть которых имеют низкие наработки. В статье приводятся результаты исследований основных причин отказов деталей и узлов нефтепромысловых насосов.
Ключевые слова: нефтепромысловые насосы**,** отказ, законы распределения отказов, износ, поломки.

NEFT-MƏDƏN NASOSLARININ HİSSƏ VƏ DÜYÜNLƏRİNİN ƏSAS SIRADAN ÇIXMASININ SƏBƏBLƏRİNİN TƏDQİQİ

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XÜLASƏ

Hazırda karbohidrogen hasilatı dərin laylardan həyata keçirilir və bu da öz növbəsində həm qazma, həm də istismar avadanlıqlarının iş qabiliyyətinin yüksıldilməsini tələb edir. Hidravlik hissəsi aşağı resurslara malik olan neft-mədən nasosları bu avadanlıqlar qrupuna daxildir. Məqalədə neft-mədən nasoslarının hissələrinin və birləşmələrinin nasazlığının əsas səbəbləri ilə bağlı tədqiqatların nəticələri təqdim olunur.
Açar sözlər: Neft yatağı nasosları, nasazlıq, nasazlığın paylanması qanunları, aşınma, nasazlıqlar.

EQUIPMENT. TECHNOLOGIES. MATERIALS

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INCREASE THE RELIABILITY OF A WELL HEAD

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ABSTRACT

The article describes the completion of drilling a well, lowering and cementing the production line, as well as after digging the cement plug in the casing (the height of the rise of the cement behind the belt if it is determined that there is a cement plug in the casing when determining with an electric thermometer)used to hang safety belts during wellhead equipment information on casing heads, their types and the study of the sealing element.
Keywords: casing head, casing hanger, hermetic knots, casing spool, rubber sealant, metal sealant.

Introduction: Different operations are used in the drilling and operation of wells. After the wells are drilled to the projected depth, protective casings of various sizes are installed on the wellbore. After all the casings have been lowered into the well, the belt heads are gradually installed at the wellhead and the space between the casings is sealed. During drilling, a preventive device with anti-spill equipment is installed on the casing heads, and during operation, a pipe head and fountain fittings are installed.

Relevance of the case: The casing head must also provide complete tightness at the expected pressure, as it is part of the discharge equipment to prevent formation energy in the wells. Due to the complex operating conditions of the well heads, they are exposed to temperature, pressure, abrasive corrosion and corrosive environment when used in different environments. These forces affect the performance of the sealing nodes of the belt head. One of the most pressing issues is the study of the failure of the sealing nodes of the well heads and their inability to perform their function.

The purpose of the work: During the operation of seals made of durable material under actual operating conditions to ensure the effective operation of the sealing nodes of various shapes and types of well heads, produced domestically and abroad, operated during the drilling and production of oil and gas wells and one of the main components of anti-spill equipment is to ensure reliability.
The casing head installed during drilling and production in wells is one of the components of anti-spill equipment. Designed to perform many operations in terms of construction. One of the main functions of the well heads is to seal and seal the annular space between the guide, intermediate and service casing. Well heads perform a variety of technological operations, including sealing the space behind the casing, keeping the casing hanging at the wellhead, controlling and interfering with the space behind the casing if necessary, installing the

fountain armature on the flange and installing it on the anti-spill equipment. One of the main components of the well heads is the casing hanger, which hangs the belts lowered into the wellbore and seals the wellhead to the space to ensure reliable isolation from the environment. There are two types of pipe heads designed for hanging and sealing technical pipelines. The first type of OKM (pipeline suspension with clutch suspension); the second is the well heads with OKK (with suspension of the pipelines).

Casings of this design allow to apply a specially prepared paste or self-hardening plastic mass in order to restore the tightness in the space between the casing hanging at the wellhead, if the tightness is violated. The well head (OKM) for the clutch hanger is designed for a pressure of 14 MPa. The main parts of this type of construction consist of a body, a clutch hanger, fixing (strop) screws, a tap and a manometer [1].

The well head (OKK) for wedge suspension is designed for pressures of 21, 35 and 70 MPa. This type of construction itself consists of casings of different sizes.

In all types of casing heads, the parts include a casing suspension (wedge or coupling), a sealing knot and a manifold of the casing spool (figure 1).

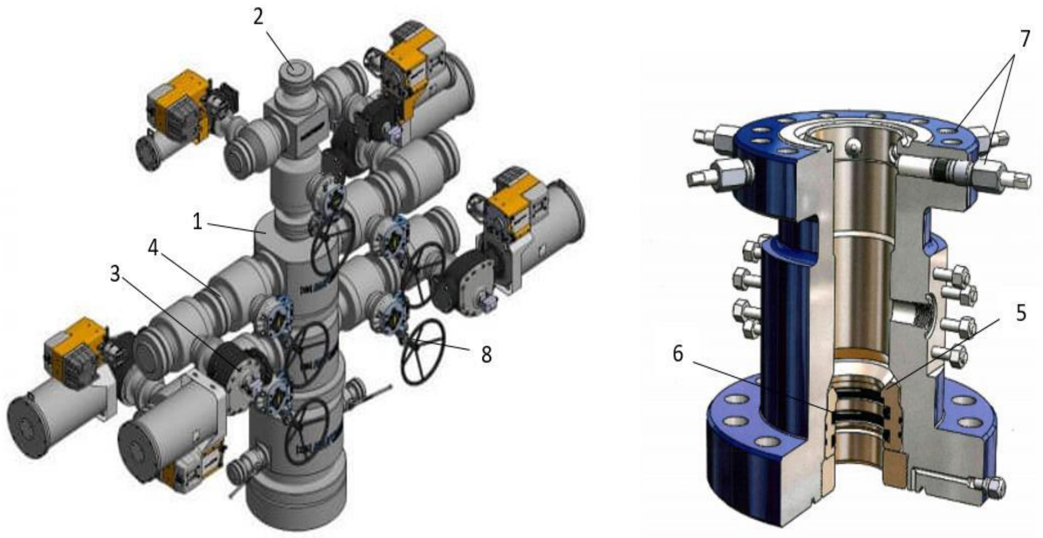

Figure 1. Casing hanger: 1-body; 2-casing sealant; 3-manometr; 4- casing hanger maniphold; 5-casing hanger; 6-sealant mechanism; 7-tightening strops; 8- shut-off tap.

A sealant rubber is installed in the casing spool to ensure tightness in the space between the production casings and the protective casings (figure 2). Sealants are made of two types of material (rubber and metal). In order to prevent the flow of products from the well, a gap of 120° is provided inside. The equipment manufactured by “CAMERON”, the largest manufacturer of casing heads in the world, is provided with the following method:

Initially installed seals, metal rings are placed separately; then the load received by the suspension while hanging from the wedge suspension acts on the rubber cap. In this case, the necessary stickiness is obtained.

Unlike the well heads produced by the US company CAMERON, the casing spool produced by AZINMASH (Institute of Petroleum Engineering) are sealed by flange joints.

Metal seals are used in such hermetic joints, which distinguishes it from other well heads. Sealing rings made of metal materials are placed on the casing together with the appropriate rubber sealants.

The shape and size of such sealants are very small compared to each other, due to the fact that there may be very few differences in the geometric dimensions of the casings to be operated. These rubber seals are installed on both sides of the rings made of metal material in accordance with the dimensions of the specified casings. The reason for the use of such metal sealants is that they are resistant to high temperatures in the case of any accident or fire at the wellhead. When installing hermetic knots, they are placed in opposite directions and on top of each other with a metal ring inside the casing spool. During production, the protrusion on the sealant was taken into account for placement inside and outside. The protrusion inside the sealant is used to seal the technical belts. The protrusion placed on the outside is placed on the inside of the casing spool. The purpose of this process is to create compression due to the weight of the casing spool when installing the well head on the sealant and to achieve tightness between the casings. [2].

Modern methods have been designed to seal the space between the casings in the well heads and are currently being implemented in production.

Examples include:

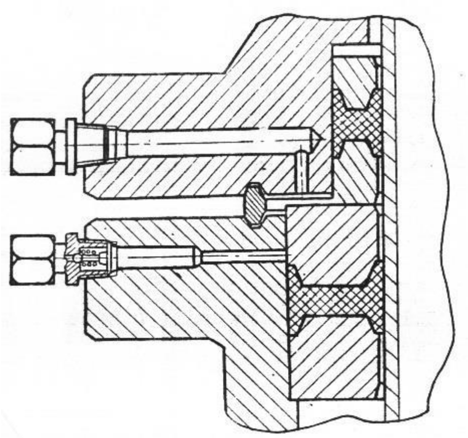

Figure 2. Sealing knot. Many well heads are fitted with rubber caps to seal the space between the casings.

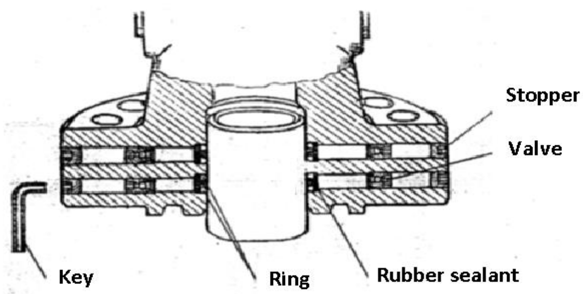

Figure 3. Inter-casings space sealant.

Such designed structures are mainly used to seal the space between the casings in the last row of wells of 2, 3 and 4 rows with the protrusion at the wellhead. It is possible to weld the technical belts to the pipe head to seal the space between the inner belts. However, this method of sealing is not preferred due to safety and other reasons.
The following are examples of these obstacles.

- Initially, welding materials with different properties depending on the well drilling process can cause many technological difficulties. However, the seams of the weld may be of poor quality.
- Another reason is the difficulty of installing the well head, which is an integral part of the welded pipe head of the technical pipelines.
- Another reason is that changing the high temperature regime of the wells is likely to cause the technical casings to move in the vertical direction. This causes the weld joint to lose quality. For this reason, the tightness of the space behind the relevant casing may be compromised. In this case, during the process of repairing the equipment inside the well, there may be leaks in the welding joint during the injection of the washing solution into the well under high pressure, which can be attributed to its unreliability.

In addition to casing heads, the “Sattarkhan” plant produces rubber seals made of corrosion-resistant and highresistance rubber, which are resistant to corrosion and temperatures up to 150 ° C.
The sealing operation between the casings is performed in the following order [3].
Casings of 168, 219 and 299 mm, suspended from the casing spool by means of a casing hanger, are sealed with rubber sealants along their diameters. (Picture 4). Sealing with these sealants prevents the effects of pressure from the top and bottom on the casing hanger. To improve the longevity and reliability of the well head and improve the sealing process, a special self-tightening paste is applied to the seals of the last protective 168 and 219 mm casings. The 377 mm casing is sealed only at the top, which means that the product is less likely to flow behind the initial casing during the expected pressure [4].

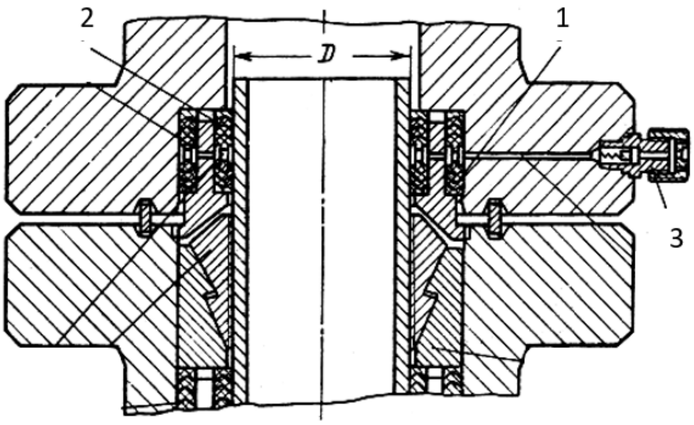

Figure 4. Sealing mechanism of the technical casing in the casing spool.

In well heads of this design, the sealing in the space between the casings without the use of welded joints is achieved by applying self-tightening pastes and sealing rubber caps. This method is used when using ANG type rubber seals sealing of technical casings suspended

from the casing hanger at the wellhead. Initially, sealants must be installed. In the next stage, the hardening of the rubber seals must be ensured.
The tensile force (ε) expected to perform the sealing function of the ANG type rubber cap, calculated by the leaf opening angle α / 2, is calculated by the following formula:

$$\varepsilon = C - \frac{Dinn - Dout}{2} = C - C1$$

In this formula: C - the cross-sectional area of the rubber cap in the free state, mm;
C_1 - the cross-sectional area of the rubber seal in the installed position, mm;
D_{inn} - diameter inside the casing hanger, mm;
D_{out} - diameter outside the casing hanger, mm.
It has been determined from the experiments and results that, expected tensile stiffness (ε) of the used rubber seals for technical casings suspended from the well head is determined based on actual dimensions. The main function of rubber sealants, which provide sealing, should be carried out in the range of 0,5-1,5 mm.
The degree of expansion of the petals loses their size before tightening during installation in place of the sealing rubber. ($\alpha_1 > \alpha$). As a result, the petals of the rubber sealant point upwards and cause the rubber to deform.
In order to ensure that the sealing elements are in a complete condition, the sequence of installation of the casing hanger inside the casing spool must be taken into account.

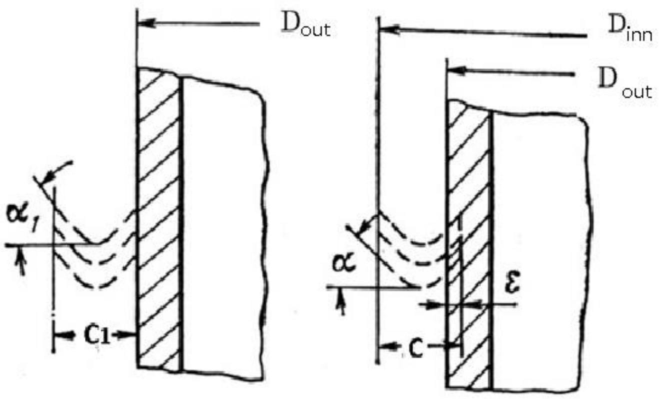

Figure 5. Sealing scheme of sealing elements.

As a result of research, it was determined that, the force required to install sealing sleeves with a diameter of 372 mm is 2-3 kN. The tension of the rubbers decreases slightly as their diameter decreases. As a result, the opening angle α of the petals changes slightly and this results in less power being used to install the cuffs. Dozens of kilograms of force are required to install sealing sleeves with a diameter of 168 mm and less [5].
Hanging mechanism for holding pipes in some structural casing heads and the sealing knot for sealing is produced in a combined design. In this type of casing heads, it is possible to perform both functions (suspension and sealing) at a single junction. Such a design provides both ease of use and is considered to be more constructive (figure 6).

As it is known, the operating conditions of seat belts are very complicated. They can be subjected to various influences while maintaining the tightness of the wellhead during operation. The most important example of these effects is temperature. Excessive temperature rise for any reason violates the tightness of the parts that ensure tightness in the casing head. For example, the temperature generated during operation affects the metal body, pins, cuffs and other details, which can lead to a number of conditions, such as violation of the tightness and compression of the flange.

There are also cases of elongation deformations and tensile forces in the details as a result of high temperatures. This is usually due to the fact that the protective pipeline is not cemented (in cases where the diameter of the pipe is 219-168 mm). For this reason, it is recommended to take the strength reserve factor in the range of 2.5-3 in the reports made during the construction of the casing spool [6].

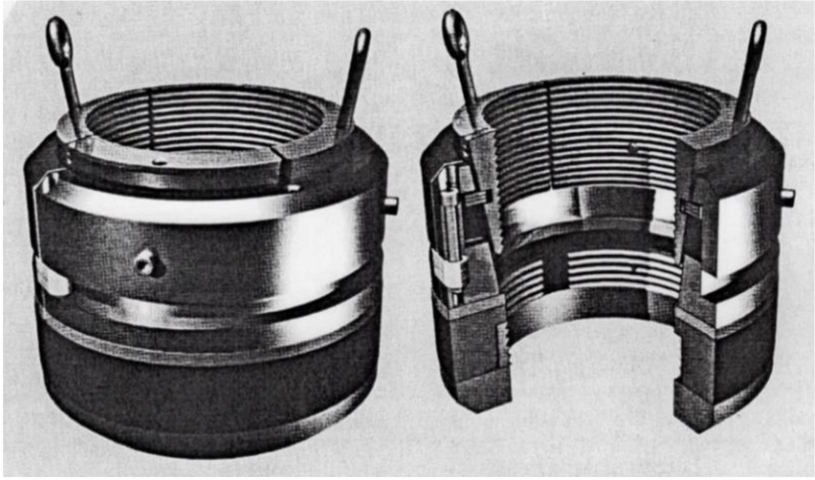

Figure 6. Combined suspension - sealing mechanism.

Result: 1. The material from which the casing head seal is made has been investigated.

2. The operation of the casing spool and wellhead, which is one of the main components of the drilled and operated well, depends on the operating pressure, well regime parameters and the type of pipes used, and the appropriate option is selected according to the existing conditions.

3. The study found that the use of metal rings with a new design seal reduces the load on the coupling groove. This prevents the clutch suspension from malfunctioning quickly, as well as its suitability for operation at high pressures and high temperatures. On the other hand, the advantage of using metal rings as a sealant is that they are durable because they are metal in the case of a fire.

4. The most effective of the various types of systems designed to seal between the pipes in the protective casing spools is selected and applied.

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ПОВЫШЕНИЕ НАДЕЖНОСТИ КОЛОННОЙ ГОЛОВКИ

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АННОТАЦИЯ

Статья дает информацию о действиях после завершение бурильных работ нефтегазовых скважинах и цементирования межколонного пространства после опускание в скважину технической колонны, а также после бурения буровым долотом цементного накопления внутри колонны (если цементное накопление произошло, электротермометром уточняется высота цементного пробки в межколонном пространстве) во время монтажа на устье скважины противовыбросового обуродования, удерживающие и обеспечивающие подвеску технических колонн на устье скважины типы колонных головок, колонных подвесок и узлов герметизаций.
Ключевые слова: колонные головки, колонные подвески, узлы герметизаций, колонные обвязки, резиновые герметизаторы, металлические герметизаторы.

KƏMƏR BAŞLIĞININ ETİRBALIĞINI YÜKSƏLDİLMƏSİ

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XÜLASƏ

Məqalə neft-qaz quyularında qazma əməliyyatını bitirdikdən və texniki kəmərləri quyuya endirib kəmər arxası fəzanı sementlədikdən sonra, həmçinin kəmər daxilində yığılmış sement qapanmasını qazma baltası ilə qazıdıqdan sonra (kəmərlər arası fəzada sement tıxacının hündürlüyü elektro-termometr vasitəsilə müayinə etdikdə kəmər daxilində sement qapanması baş verdiyi təyin olunarsa) quyu ağzında atqıya qarşı avadanlıqları quraşdırılarkən texniki kəmərlərin quyu ağzında saxlayan və asılmasını təmin edən kəmər başlıqları, növləri, kəmər asqısı və kipləndirici düyünlərinin tədqiqatı haqqında məlumat verilir.
Açar sözlər: kəmər başlığı, kəmər asqısı, hermetikləşdirici düyünlər, kəmər sarğısı, rezin kipləşdirici, metal kipləşdirici.

DYNAMICS OF SCREENS WITH COMPLEX VIBRATIONS USED IN THE OIL AND GAS INDUSTRY

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ABSTRACT

In the presented work, the differential equation of motion of the elements of the screen is compiled with generalized coordinates. Since the device has two degrees of freedom during vibration, two nonlinear differential equations have been written. Since it depends on the specific frequency, as well as the mass of drilling fluid and the angular velocity of the vibrating frame, it can be concluded that a vibrating screen can work more effectively at a certain value of the amplitude of the oscillation.

Keywords: Drilling fluid, vibrating screens, kinetic energy, generalized coordinates, Lagrange equation, angular velocity, amplitude.

It is known that various types of surface equipment are widely used in the oil and gas industry to clean drilling mud from drilled rocks. The use of vibrating screens is one of the mechanical methods used to clean drilling mud in Azerbaijan and other oil-producing countries. As an example SWECO MM4 and high-performance MM4 vibrating screens having elleptical oscillation (Figure 1.1 and Figure 1.2) can be shown as an example for such screens. It should be noted that the main parts of the main parts of the equipment operating mechanically and used to clean drilling fluid from various types of solid rocks are grids and springs.
The main purpose of this work is to build a mathematical model for the movement of the main elements of the vibrating screens of a complex oscillating nature used in the oil and gas industry. To solve this problem in the presented work, the differential equation of motion of the elements of the device is constructed with generalized coordinates. Since the device has two degrees of freedom during vibration, two nonlinear differential equations are written. Since this coordinate, in turn, depends on the specific frequency of the oscillation, as well as the mass of the drilling mud and the angular velocity of the vibrating frame, it can be concluded that at a certain value of the amplitude of the oscillation, the vibrating screen can work more effectively. Drilling mud performs various and at the same time complex tasks in drilling wells. These are called technological tasks because the solution used in the drilling process carries out various operations in the drilling wells technology. Each technological task of the drilling mud has a specific purpose in drilling wells. Lifting of rock particles excavated from the well bottom, protection of the well wall from collapsing, destruction of rocks in the well bottom, delivery of kinetic energy to the wellbore engine, etc. are the technological tasks of the drilling mud. In order to perform various technological tasks, the drilling mud must

have a certain quality. To obtain the required quality, the drilling mud must be chemically treated and, if necessary, aggravated. The drilling mud must ensure the drilling of the well at high speed and safely, removing the high formation product from the well.
Rock particles entering the drilling mud cause deterioration of the technological quality of the solution, deterioration of the technical and economic performance of the drilling and many complications. For this reason, it is very important to clean the drilling mud from harmful impurities.
Various complex devices are used to clean the drilling mud from the sludge. These devices include shaking screens, hydrocyclone sludge separators (sand and clay separators), centrifuges.
Selected equipment other than the centrifuge must ensure that the maximum amount of solution required during drilling is removed. In the drilling mud cycle, the cleaning equipment is installed in the following sequence: gas separator - shaking sieves - degasser (gas cleaner) - hydrocyclone sludge separator - separator centrifuge.
As it is known, the size of clay particles in the drilling mud varies from one micrometer to tens, the size of powdered barite is 0.5-0.75 microns and the size of the cuttings is from 10 to 25 microns. Only solid particles larger than 150 μm in the drilling mud can be removed very efficiently by means of vibrating screens (BC-1, BC-2). These vibrating screens are manufactured by Savako and Baroid. When using these sieves, the cleaning rate of the drilling mud is 50%.
The current state of drilling requires that more attention should be paid to the cleaning of drilling mud from waste. Poor cleaning of drilling fluids can lead to serious accidents. Currently, two types of equipment are widely used in equipment for cleaning drilling mud [1-5]:
- mechanical cleaning;
- hard rock fragmentation equipment.

Mechanical cleaning is carried out with the help of separation devices (screens). The separation device used to clean the excavated rock from solid particles of clay solutions is one of the main parts of the drilling equipment. The cleaning process is closely related to the rational operation of the separation unit, technological parameters, strength and energy intensity. To manage these parameters, it is necessary to create a mathematical model of the separation device. At least the approximate equation of the separation device has not been found in the references / 6-10 /.
It is known that during the purification of clay solutions, the process occurs during the rotation of the unbalanced inertial element. The operation of a separation device describes the complex movement of a force around a horizontal line from the rotation of an unbalanced element.
From this, it can be concluded that the movement of the separation device belongs to two degrees of freedom. As a general independent coordinate, let us consider the unbalanced element rotating around the horizontal axis to the angle φ and the motion of the vertical z vibrator with all the elements of the separation device.
Let's consider the scheme of movement of the separation device (Figure 1.1). The separation device (screen-1) has a modern design, it cleans the solution even in 0.16 mm clays. The vibrating frame is provided with two sequentially installed grids. To make our work easier,

let's assume that the grids are connected horizontally. The degree to which the separation unit purifies the solution depends on the operation of its individual elements.

Figure 1.1. “SWECO MM4” type screen having elleptical ostillation.

Figure 1.2. High performance “MM4” vibrating screen.

The aim of the work is to create a mathematical model that describes the movement of the main elements of the separation device and allows to regulate the rational vibration.
It is known that vibration is carried out with the help of energy of reducers mounted on the vibrator.
Let's compile the second type of Lagrange's equation:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\varphi}}\right) - \frac{\partial T}{\partial \varphi} = Q_{\varphi} \qquad \frac{d}{dt}\left(\frac{\partial T}{\partial \dot{z}}\right) - \frac{\partial T}{\partial z} = Q_{z} \tag{1.1}$$

Here φ is are the angle of rotation of the unbalanced element, z - the vertical motion of the vibro frame, T - the kinetic energy of the vibrating grid, $\dot{\varphi}$ and $\dot{z}$ - generalized velocities, Q_φ and Q_z - generalized forces.
The kinetic energy of the vibration is as follows:

$$T = A\dot{\varphi}^2 + B\ddot{z}^2 - m_1 r\dot{z}\dot{\varphi}sin\varphi \,, \tag{1.2}$$

Performing various mathematical operations, equations (1.1) and (1.2) can be written as follows:

$$2A\ddot{\varphi}^2 - m_1 r\dot{z}\dot{\varphi}sin\varphi = Q_\varphi,$$

$$2B\ddot{z} - m_1 z(\ddot{\varphi}sin\varphi + \dot{\varphi}^2 cos^2\varphi) = Q_z \,, \tag{1.3}$$

here,

$$A = \frac{J_1 + m_1 r^2}{2} \,; B = \frac{m_1 + m_2}{2}, Q_z = cz \,, \tag{1.4}$$

m_1 is the weight of the unbalanced element, m_2 is the weight of the solution with the vibrator, r is the eccentricity of the imbalance, J_1 is the moment of inertia with respect to the axis of rotation of the rotor of the electric motor.
(1.3) and (1.4) are the equations of the mathematical model of a vibrating screen device. In these equations, the angular velocity is a variable, which complicates its integration. Analysis of the operation of the separation device showed that the angular velocity of the imbalance can be considered as a constant quantity. In this case, if the coefficients of resistance acting on the device are not considered, the dependence of the movement along the horizontal z axis on the vertical harmonic oscillations from the ω-angular velocity can be written as follows:

$$(m_1 + m_2)\ddot{z} + cz = m_1\omega^2 cos\omega t, \tag{1.5}$$

Here $m_1\omega^2$ is the amplitude of the excitation force.
Taking into account the initial initial conditions, the solution of equation (1.5) can be written as follows:

$$z = \frac{D}{n^2 - \omega^2}(cos\omega t - cosnt) \tag{1.6}$$

where $D = \frac{\omega^2 r m_1}{m_1 + m_2}$, n is the specific oscillation frequency of the vibrating screen device.
Considering the effect of an excitatory force on the vertical movement of a vibrating frame, we get.

$$z = \frac{D}{n^2 - \omega^2} cos\omega t \,, \tag{1.7}$$

For this purpose, calculations have been made with different loads and different clay solutions according to formula (1.7). The calculation is mainly reviewed for the cases encountered in practice. Three of them are given in the form (1.3). Visual observations of the separation unit operation have shown that when the amplitude of the oscillation is 10 mm, the cleaning of drilling mud from waste is more effective. And this happens when

$$D = 8{,}98\ m,\ \ n = 63s^{-1},\ \ \omega = 54s^{-1}$$

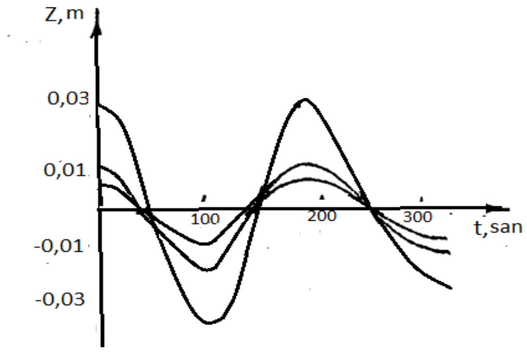

Figure 1.3.

Results: The dynamics of vibrating screens of complex oscillating nature used in the oil and gas industry has been considered.

1. Differential equations with generalized coordinates of the motion vibrating screens elements of complex oscillating nature used in the oil and gas industry have been compiled.
2. During the oscillation of vibrating screens with a degree of freedom equal to two, the amplitude of the oscillation has been determined and it was shown that at this value of the amplitude, the vibrating sieve can work more rationally.

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ДИНАМИКА СИТ ИМЕЮЩИХ СЛОЖНЫХ КОЛЕБАНИЙ, ПРИМЕНЯЕМЫХ В НЕФТЕГАЗОВОЙ ОТРАСЛИ

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АННОТАЦИЯ

В представленной работе составляется дифференциальное уравнение движения с обобщенными координатами. Поскольку устройство имеет две степени свободы при вибрации, то записываются два нелинейных дифференциальных уравнения. Так как, оно зависит от конкретной частоты, а также по массе бурового раствора и угловой скорости вибрационной рамы можно сделать вывод, что вибросито может работать более эффективно при определенном значении амплитуды колебаний.
Ключевые слова: Буровой раствор, вибросита, кинетическая энергия, обобщенные координаты, уравнение Лагранжа, угловая скорость, амплитуда.

NEFT-QAZ-MƏDƏN SƏNAYESİNDƏ İSTİFADƏ OLUNAN MÜRƏKKƏB RƏQSİ XARAKTERƏ MALİK ƏLƏKLƏRİN DİNAMİKASI

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XÜLASƏ

Təqdim olunan işdə ələyin elementlərinin hərəkətinin diferensial tənliyi ümumiləşmiş koordinatlarla tərtib edilir.Titrəmə zamanı qurğu iki sərbəstlik dərəcəsinə malik olduğundan iki qeyri xətti diferensial tənlik yazılır və analitik üsulla həmin tənliklər, xətti tənliklər sisteminə gətirilərək titrəyişli çərçivənin üfüqi hərəkətinin koordinatı təyin edilir.Bu koordinat öz növbəsində rəqsin xüsusi tezliyindən, eyni zamanda qazıma məhlullarının kütləsindən və vibroçərçivənin bucaq sürətindən asılı olduğundan belə nəticəyə gəlmək olur ki, rəqsin amplitudasının müəyyən qiymətində titrəyişli ələk daha effektli şəkildə işləyə bilər.
Açar sözlər: Qazıma məhlulu, titrəyişli ələklər, kinetik enerji, ümumiləşmiş koordinatlar, Laqranj tənliyi, bucaq sürəti, amplituda.

EQUIPMENT. TECHNOLOGIES. MATERIALS

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INVESTIGATION OF POLYMER NANOCOMPOSITIONS ON THE BASIS OF LOW DENSITY POLYETHYLENE (LDPE) AND FILLER MIXED

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ABSTRACT

Electrical conductivity, strength of nano compositions based on low density polyethylene and filler nanotube limestone rocks, etc. the effect of properties was applied.

Keywords: thermoplastic, polymers, composites, materials.

Introduction: The addition of low-density polyethylene (LDPE) and its chemically based nano-limestone limestone contributes to the electrical conductivity, strength, etc. of nanocomposites, properties. The effect of an electrically conductive filler is explained by the formation of a conductive structure of its particles in the polymer.

The purpose of the study: To increase the conductivity, strength, etc. when using nanotubes based on low-density polyethylene (LDPE) and inorganic fillers in the polymer composite material improvement of properties is shown. For this reason, the study of the impact of the factors considered appropriate.

The main part: One of the main directions of increasing the role of polymer compositions in the development of technical progress is the use of new sources of raw materials in the creation of these compositions.

The use of inorganic fillers such as chalky limestone as a filler and activator in compositions based on elastomers is widespread.

It is possible to obtain a multi-component system with durable properties, high physical and mechanical properties, and the ability to maintain these properties under static and dynamic load for a long time.

The properties of such systems depend on the substances introduced into the system from the polymers and the processing processes of the system.

It is known that polyolefins are formed from the polymerization of olefins and olifins on hydrocarbons containing double bonds, and one of the most common types of polyolefins is polyethylene.

Because ethylene is difficult to polymerize, polymerization takes place in a solid form.

Low-density polyethylene (LDPE) is mainly a product of Sumgayit PU, is the most widely used dielectric among synthetic polymers, and its feature is that these compositions are obtained on the basis of polymers with very convenient, large-volume production.

Due to its low moisture and moisture permeability, LDPE does not change its plasticity even after long-term storage in water. The polymerization process is carried out under a pressure of 150-300 MPa.
Carried out at a temperature of 240-280 ° C.
The process of modification and high temperature (125-160 ° C) processing of LDPE is carried out in the composition.
In the composition of LDPE (Garadagh and Dashsalakhli) a mineral filler with nano-limestone limestone was taken. (Table 1)
The density of nano-shaped limestone was 2.5 / 2.0% in kg / m^3 2008/2680 porosity, 1000 m^2 / kg in specific surface area. ASPE has the following parameters.
GOST-16337 type brand 15803-020, density (P) kg / m^3-919.0 ..; crystallization 40%, alloy flow rate (ALL) gr / 10 min.-2.1; drying stress (G gr), MPa-12.0: relative elongation at break (Eqr) 660%, melting temperature (Ta), 103- 105 ° C. Specific volume electrical resistance (P) Ohm · m · 1 · 10 The tangent of the angle of dielectric loss (tg, 1MHs) · 2.25 is shown.
It is known that electrical resistance is divided into groups
1. High conductivity p <100 Om · m
2. Moderate in conductivity p = 10^2-10^7 Om · m
3. Low conductivity p> 10^8 Om · m
Products made of nanocomposite materials are exposed to many external factors (temperature, deformation, electric field, mechanical load, etc.) during operation. Therefore, the study of nanocomposites against the influence of external factors when applying electrical properties is important both practically and scientifically.
When polymer nanocomposites are used for technical purposes, low hole voltage values are obtained because the uniformity of the electric field is not ensured.

Table 1. Electrical resistance of mineral fillers.

Minerals	Specific gravity gr / cm^3	Electrical resistance Om-m
Limestone	2,30-2,70	3,0-10^5
Chalk	2,56-2,70	2,63-2,73

Fillers consisting of LDPE and nano-limestone limestones are loaded into the mixer. After mixing the polymer at 150 ° C for 5-10 minutes, fillers are added and the mixing can continue for 10 minutes.
After the mixing process, the compositions are cooled to room temperature.
Preparation of compositions from the extruder is possible at 180-220 ° C. The mixture processed in the extruder for 5-7 minutes can be used after heating at 105-110 ° C for 10 minutes.
For the study of physical and mechanical properties, it is possible to prepare samples by direct pressing at a pressure of 12-15 MPa at 180-220 ° C and casting under pressure at 160-220 ° C (Table 2).
The composition of aggregates, the main physical and chemical properties of which are determined by known methods in the central laboratories of Sumgayit Polymer Building Materials Plant and Ganja Clay-Soil Plant (Table 3).

Table 2. Physical and mechanical properties of mineral fillers

№	Names of indicators	Indicators
		Chalk-shaped lime
1	Density, kg / m³	2008÷2680
2	Porosity,%	2,5÷2,0
3	Special surface,m²/kq	1000
4	Durability in water-saturated state,MPa	_

Table 3. Chemical composition of mineral fillers

№	Ingredients	Quantity,% by weight
		Chalk-shaped lime
1	SiO_2	0,02÷0,65
2	Al_2O_3	0,05÷0,25
3	Fe_2O_3	0,24÷0,16
4	CaO	53,2÷55,28
5	MgO	0,15÷31,4
6	SO_3	0,8
7	K_2O, MgO, TiO2 ,FeO,MnO, P_2O_5	-

Results: 1. Nanocomposites based on low density polyethylene (ASPE) and filler were obtained.
2. The high elasticity of the composition based on (ASPE) allows the widespread use of natural fillers with large deposits in our country.
3. Processed compositions are cheaper because they are processed at lower pressure temperatures.
4. The results are intended to show clearly that they are multi-component in the production of electrically conductive compositions.

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ИССЛЕДОВАНИЕ ПОЛИМЕРНЫХ НАНОКОМПОЗИЦИЙ НА ОСНОВЕ ПОЛИЭТИЛЕНА (ПЭНП) НИЗКОЙ ПЛОТНОСТИ И СМЕСИ НАПОЛНИТЕЛЕЙ

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АННОТАЦИЯ

Применение электропроводимости и прочности нанокомпозиций на основе полиэтилена низкой плотности (ПЭНП) и наполнителя в форме наномела из породы известняка.

Ключевые слова: термопласты, композиционные материалы.

AŞAĞI SIXLIQLI POLİETİLEN (ASPE) VƏ DOLDURUCU QARIŞIQLARI ƏSASINDA POLİMER NANOKOMPOZİSİYALARIN TƏDQİQİ

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XÜLASƏ

Aşağı sıxlıqlı polietilen (ASPE) və doldurucu nanotəbaşirşəkilli əhəngdaşı süxurları əsasında nanokompozisiyaların elektrik keçiriciliyi,möhkəmliyi və s. xassələrin təsiri tətbiq edilib.

Açar sözlər: termoplastlar, kompozisiya materialları.

ENHANCING PERFORMANCE OF MECHANICALLY OPERATED PACKER

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ABSTRACT

During hydraulic fracturing operations of low-permeability reservoirs, packers are the key component to ensure the success of multistage fracturing. Packers enable sections of the wellbore to be sealed of and separately fractured by hydraulic pressure, one at a time, while the remainder of the wellbore is not afected. However, reliable sealing properties of the packer rubber are required to meet the high-pressure and high-temperature (HPHT) conditions of reservoirs (such as 70 MPa and 170 °C). In this study, the structures of the packer rubber with two diferent materials are optimized numerically by ABAQUS and validated by experiments. The optimization process starts from the packer rubber with a conventional structure, and then, the weakest spots are identifed by ABAQUS and improved by slightly varying its structure. This process is iterative, and the fnal optimized structure of a single rubber barrel with expanding back-up rings is achieved. For the structure of three rubber barrels with metallic protective covers, both HNBR and AFLAS fail under HPHT conditions. For the fnal optimized structure, the packer rubber made of AFLAS can work better under HPHT than that made of HNBR which ruptures after setting. The results show that the optimized structure of a single rubber barrel with expanding back-up rings and the material AFLAS are a good combination for the packer rubber playing an excellent sealing performance in multistage fracturing in horizontal wells.

Keywords: packer rubber, sealing property, structure optimization, hydrogenated nitrile–butadiene rubber (HNBR), AFLAS rubber.

Introduction: At present, many oil felds in World have come into the middle and late stages of production. In order to guarantee the stability of the oil and gas production, unconventional oil and gas reservoirs and low-permeability reservoirs are the major goals of exploration and development. However, because of their low permeability and poor production, it is necessary to fracture the oil and gas wells. Staged fracturing completion technology in horizontal wells is an advanced and efective measure of enhancing oil and gas production and recovery ratios during the exploration of low-permeability reservoirs, and packers are key equipment to complete multistage fracturing successfully. However, high demands on the materials and the structure of the packer rubber are required to meet the high-pressure and high-temperature (HPHT) conditions of reservoirs.

At present, most scholars have investigated the material properties of packer rubber by the use of dumbbellshaped specimens and simple cylindrical specimens, and their conclusions indicated that many rubber materials such as HNBR and AFLAS have better pressure resistance and high-temperature resistance. However, the infuence of the actual structures and shapes on sealing performance of packer rubber is seriously considered. The packer rubber made of high-performance material may have poor sealing performance under conditions of high pressure and high temperature in low-permeability reservoirs because of an unreasonable structure. In this paper, the study begins with packer rubber with a conventional structure and optimizes the materials and the structure of the packer rubber by ABAQUS software simulation and experimental research. In the experiments, rubber sleeves are installed on a packer and a function test scheme of the packer is designed which can simulate the packer setting and its working conditions.

Theory basis: Selection of the rubber materials. The selection of rubber materials is a key factor in the packer design, because the maximum operating diferential pressure and the maximum operating temperature of the packer are primarily limited by the material of the rubber barrel. The main rubber materials of the compression packer are fuorine rubber, nitrile rubber, hydrogenated nitrile rubber and AFLAS rubber (a new fuorine rubber based on tetrafuoroethylene and propylene copolymer), the advantages and disadvantages of these rubber materials are given in Table 1, and hydrogenated nitrile–butadiene rubber (HNBR) and AFLAS fuoro rubber are considered in the following sections

Order number	Hardness H, IRHD	Elastic modulus E_0, MPa	Material coefficients of rubber	
			C_{10}	C_{01}
1	60	4.42	0.491	0.294
2	65	5.54	0.616	0.307
3	70	6.96	0.774	0.387
4	75	8.75	0.972	0.486
5	80	10.98	1.221	0.610
6	85	13.80	1.533	0.767
7	90	17.33	1.926	0.963
8	95	21.77	2.420	1.410

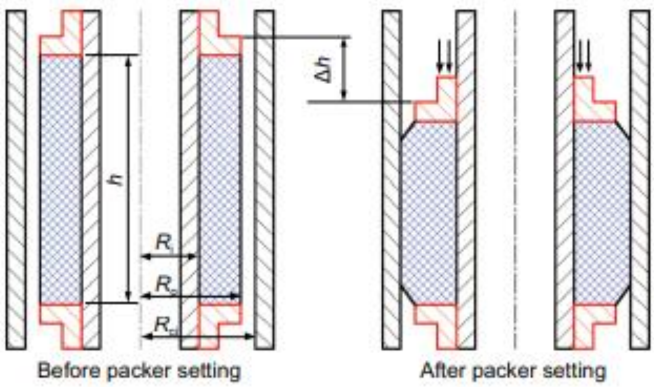

Figure 1. Force analysis diagram of the packer rubber before and after setting.

Calculation of the packer setting force: The calculation model of the packer setting force is shown in figure 1. In the packer setting, a compressive force (that is packer setting force) acts on the head face of the annulus packer rubber and compresses the packer rubber, which leads to the packer rubber expanding radially and contacting with the inne wall of the casing. In order to achieve a good sealing efect, the setting force should further compress the packer rubber to make sure that the friction force between the packer rubber and the casing due to the contact pressure is greater than the operating diferential pressure. The packer setting force includes two parts: the compressive force required to make the sealing element contact with the inner wall of the casing and the compressive force to make the sealing element expand from contacting with the inner wall of the casing to meet the required sealing efect

Conventional structure: As the key component of the packer, the structure of the rubber directly infuences the sealing properties of the packer. As shown in figure 2, the structure of the regular packer seal unit is a combination of three rubber barrels, which are separated by metal calking rings.

R_{ci}, mm	ΔP, MPa	Δh, mm	P, MPa	P_c, MPa
76.20	70	41.15	22.73	2.4
77.40	70	56.25	27.00	4.0
78.55	70	69.25	31.50	5.7
79.85	70	82.50	36.75	7.8

R_{ci}, mm	ε_z	Δh, mm	F_c, kN	$F_{\Delta P}$, kN	F, kN	P, MPa
76.20	0.165	41.15	9.302	79.962	89.264	22.73
77.40	0.225	56.25	13.649	92.511	106.164	27.00
78.55	0.277	69.25	17.936	105.789	123.725	31.50
79.85	0.330	82.50	22.927	122.374	144.253	36.75

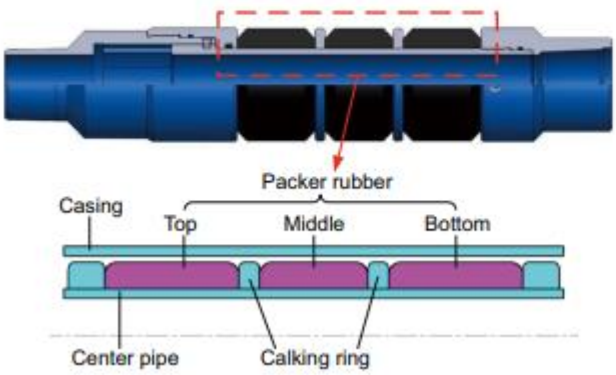

Figure 1. Structure of three regular rubber barrels.

In order to study the sealing properties and structure optimization of the packer rubber at high pressures and high temperatures, the process of the packer rubber setting is simulated using the fnite element software ABAQUS to provide a theoretical basis for structure optimization before experiments. In the simulation, some assumptions are made as follows without afecting the function of the packer: The packer is always located in the center of the casing and is symmetric around the centerline of the casing; the efect of the packer's self-weight to simulation results is neglected; only the packer rubber is analyzed, and the bottom calking ring, the center pipe and the casing are stationary in the simulation.

Structure optimization plan: Single rubber barrel with expanding back-up rings:
Although the composite construction of three rubber barrels with metallic protective covers can prevent the shoulder protrusion, the contact pressures of three rubber barrels are very diferent. Therefore, the new structure of a single rubber barrel with expanding back-up rings (shown in figure 15) is selected in place of the construction of three rubber barrels, because the middle rubber barrel plays a major sealing role. In the new structure, the material of the single rubber barrel is still AFLAS rubber and there are two metal conical rings at both ends of the rubber barrel instead of the top and bottom rubber barrels in the construction of three rubber barrels to bear the thrust for setting. There are two sets of expanding back-up rings among the conical rings and the rubber barrel for preventing the shoulder protrusion. One set of expanding back-up rings is made of an outer ring and an inner ring. The two rings are embedded in each other to ensure good contacts, and there is a small gap on each ring which is convenient for expanding. A dovetail groove is cut in the inner wall of the rubber barrel.

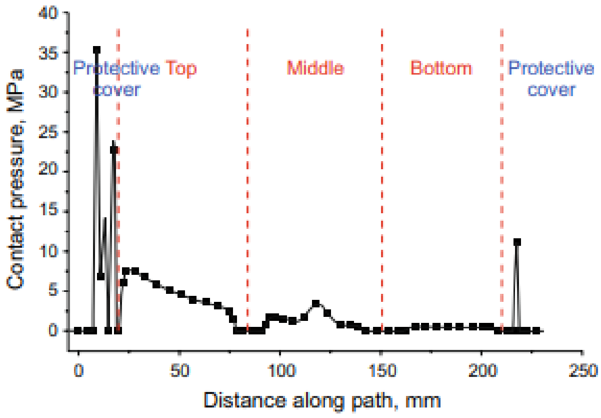

Figure 3. Contact pressure distribution of the new structure with metallic protective cove.

The hardness of the rubber barrel is 80 IRHD. The materials of the conical rings and the center pipe are 35CrMo, whose elastic modulus is 2.16×105 MPa and Poisson's ratio is 0.286. The material of the casing is 45 steel, and the materials of back-up rings are 20 steel. The elastic moduli of the casing and the back-up rings are 2.06×105 MPa, and their Poisson's ratios are 0.3. The possible contact among each part before and after deformation is set. The friction coefcient among metal parts is 0.1, and that between the packer rubber and the metal parts is set as 0.3. The bottom conical ring, the center pipe and the casing are stationary. A pressure of 68.8 MPa is applied to the end face of the top conical ring to compress the packer rubber. The results simulated by ABAQUS are shown in figure 17. As shown in figure 18, the contact pressure between the casing and the packer rubber with expanding back-up rings (the average value is 9.75 MPa, and the maximum value is 16.43 MPa) is larger than those of the conventional structure and the new structure with protective covers, and no apparent peak value is shown in the distribution curve of the contact pressure, which illustrates that the shoulder protrusion of the packer rubber does not occur and the sealing properties are much better.

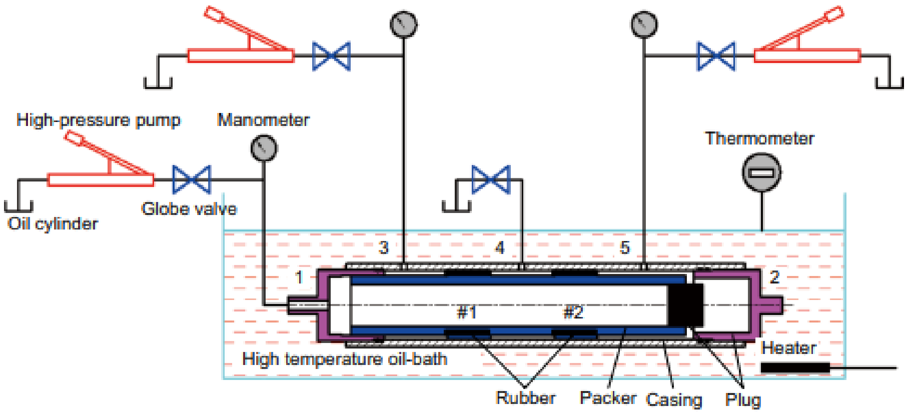

Figure 4. Schematic diagram of the performance test.

Figure 5. Experimental equipment.

Conclusions: Based on ABAQUS software simulation and experimental studies of the packer setting, we performed the material selection and the structure optimization of the packer rubber for working well under high temperature and high pressure, and the following conclusions can be drawn: (1) Two structure optimization schemes of the packer rubber are proposed. In each structure, both materials HNBR and AFLAS are tested. The results show that AFLAS rubber has greater resistance to high pressure (70 MPa) and high temperature (170 °C) than HNBR. In the structure of three rubber barrels with metallic protective covers, both HNBR and AFLAS are subjected to varying degrees of damage which cause poor sealing performances under high pressure and high temperature although the performance of AFLAS is better than that of HNBR. In the structure of a single rubber barrel with expanding back-up rings, the packer rubber made of HNBR is also ruptured but the packer rubber made of AFLAS works well under high temperature and high pressure after setting.

Figure 6. Packer rubber made of AFLAS after tests.

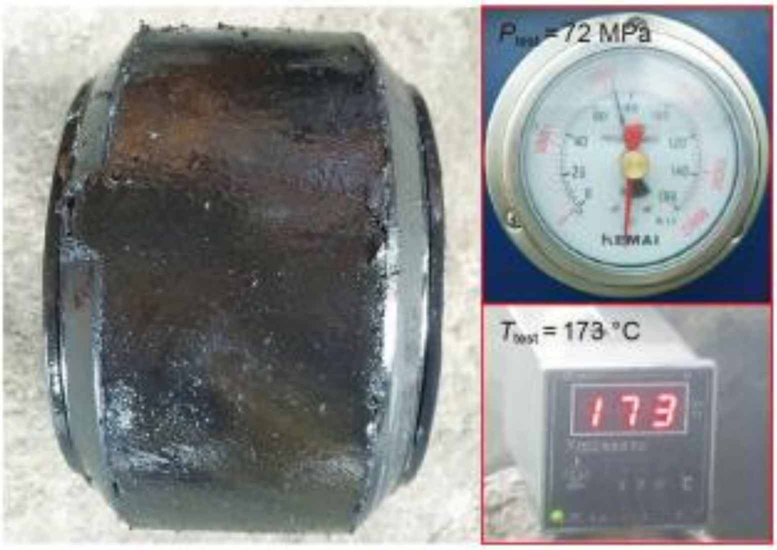

Figure 7. New packer rubber made of AFLAS with expanding back-up rings after the test.

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ПОВЫШЕНИЕ РАБОТОСПОСОБНОСТИ ПАКЕРА С МЕХАНИЧЕСКИМ ПРИВОДОМ

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АННОТАЦИЯ

При проведении операций по гидроразрыву низкопроницаемых коллекторов пакеры являются ключевым компонентом, обеспечивающим успех многостадийного ГРП. Пакеры позволяют герметизировать участки ствола скважины и проводить гидроразрыв пласта по отдельности, по очереди, при этом остальная часть ствола скважины не затрагивается. Однако для соответствия условиям высокого давления и высокой температуры (HPHT) в пластах (например, 70 МПа и 170°C) требуются надежные герметизирующие свойства пакерной резины. В данном исследовании структуры пакерной резины из двух различных материалов оптимизированы численно с помощью ABAQUS и подтверждены экспериментами. Процесс оптимизации начинается с резины пакера с обычной структурой, затем ABAQUS выявляет наиболее слабые места и улучшает их, слегка изменяя структуру. Этот процесс является итерационным, и в итоге оптимизируется структура одного резинового ствола с расширяющимися опорными кольцами. Для структуры из трех резиновых стволов с металлическими защитными оболочками, как HNBR, так и AFLAS разрушаются в условиях HPHT. Для окончательной оптимизированной структуры пакерная резина из AFLAS может лучше работать в условиях HPHT, чем резина из HNBR, которая разрывается после схватывания. Результаты показывают, что оптимизированная структура одного резинового ствола с расширяющимися опорными кольцами и материал

AFLAS являются хорошей комбинацией для пакерной резины, обеспечивающей отличную герметичность при многостадийном ГРП в горизонтальных скважинах

Ключевые слова: пакерная резина, герметизирующие свойства, оптимизация структуры, гидрогенизированный нитрил-бутадиеновый каучук (HPHT), AFLAS.

MEXANİKİ İDARƏ OLUNAN PAKERİN İŞ QABİLİYYƏTİNİN YÜKSƏLDİLMƏSİ

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XÜLASƏ

Məqalə məzmununda neft-qaz quyuları qazılan zaman,sementləmə prosesində,o cumlədən quyuların istismar zamanı istifadə olunan pakerlər və onların əsas elementləri olan kipləndirici elementlərin araşdırması haqqında məlumatlar əks olunmuşdur. Bununla birlikdə, laylardakı yüksək təzyiq və yüksək temperatur (HPHT) şərtlərinə cavab vermək üçün (məsələn, 70 MPa və 170 ° C), paket kauçukunun etibarlı sızdırmazlıq xüsusiyyətləri tələb olunur. Bu işdə, iki fərqli materialdan hazırlanmış bir qablaşdırıcı kauçukun quruluşu ABAQUS istifadə edərək ədədi olaraq optimallaşdırılır və təcrübələrlə təsdiqlənir. Optimizasiya prosesi adi bir quruluşlu bir paket kauçuk ilə başlayır, sonra ABAQUS ən zəif nöqtələri müəyyənləşdirir və onları yaxşılaşdırır, quruluşu biraz dəyişdirir. Bu proses təkrarlanır və nəticədə genişlənən istinad halqaları ilə bir rezin magistralın quruluşu optimallaşdırılır. HPHT altında həm HNBR, həm də AFLAS metal qoruyucu qabıqları olan üç kauçuk lüləli bir quruluş üçün məhv edilir. Son optimallaşdırılmış quruluş üçün AFLAS qablaşdırma kauçuk, HNBR kauçukundan daha yaxşı HPHT altında işləyə bilər və bu da tutulduqdan sonra zədə alır. Nəticələr göstərir ki, genişlənən rezin halqaları və AFLAS materialı ilə bir rezin çəngəlin optimallaşdırılmış quruluşu, üfüqi quyularda çox mərhələli hidravlik qırılma zamanı əla sızdırmazlıq təmin edən bir qablaşdırıcı kauçuk üçün yaxşı bir birləşmədir.

Açar sözlər: paker rezini, kipləndirici xüsusiyyəti, strukturun optimallaşdırılması, hidrogenləşdirilmiş nitril-butadien rezin (HNBR), AFLAS rezini.

RESEARCH OPERATION OF PISTON PUMP SEALING UNITS

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ABSTRACT

The article presents information about the study of pumps used in drilling wells, cementing protective belts, pumping clay into wells while drilling, developing productive horizons, as well as their operation, their sealing units. During piston pump operation, the seal assemblies are subjected to fluid pressure from both sides of the piston. Liquid and slurry handling equipment operates under high loads such as erosion, corrosion, cavitation and abrasion. Such cases have a significant impact on the degradation of pump performance, corrosion of the pump sliding surfaces, leading to a decrease in FIE, an increase in power consumption, and an increase in the overall service life for calculations. Failure to address these issues will jeopardize the integrity of the pumping equipment and result in equipment failure.

Keywords: piston pump, valve pair, valve seat, valve plate, valve seal, plunger.

The workability of the hydraulic part of the pump is determined by the longevity of its seals. Violation of the tightness of the seal leads to the failure of the whole part - the sealant and the metal details associated with it. Therefore, the future considerations should be done to increase the longevity of the seals in order to improve pump's workability.

Equipment for transporting liquids and suspensions operates under high loads such as erosion, corrosion, cavitation and abrasion. Such cases have a significant impact on the deterioration of the pumps operating characteristics, corrosion of pump's working surfaces, leading to a decrease in efficiency factor, and an increase in power consumption, as well as an increase in total service life during the calculations. Dereliction to address these issues could jeopardize the integrity of the pumping equipment and result in its failure . The solution can be achieved by replacing the equipment, but it always comes at a high cost and requires a long preparation time [1].

Failure of piston pumps operation until the time of abandonment results by the termination of workability of the hydraulic parts and the detailed nodes. The reliability of the referenced part depends on the wear resistance of the cylinder-piston group's details and the cylindrical carvings of the nodes as well as valve pairs, piston, rod, valve seal, hydraulic box, and also cylindrical carving seal. The details of the valve pairs of drilling pumps break down faster than the details of the cylinder-piston group. The high wear intensity in the hydraulic part of piston pump is due to the presence of abrasive particles in injected drilling fluid. The effect of high contact stresses on the working surfaces of the main part and nodes, as well as associated joints, moving and stationary surfaces - piston seals, valve plates and sealing surfaces which lead to decrease the intensity to wear at the end [2].

Sealing is the prevention of liquid flow or gas from one sealing chamber to another or to the external environment. The purpose of sealing is to maximize the sealing of the fluid leak, as

leakage is harmful economically and due to safety requirements, as well as the transport of abrasive fluids at high pressures leads to failure (refusal) of the entire sealing node as a result of hydroabrasive corrosion.
The operating conditions for the degree of movement of the seal and the change in pressure are given in Table 1.

Table 1. Working conditions of sealing nodes of the hydraulic part

Sealing nodes	The degree of movement of the seal	The nature of the change in pressure
carved pistons stocks, plungers	Moving, variable moving	Does not change
valves	Moving, periodic variable moving	Does not change
Covers of the suction valves Sealing element of the valves Placing the piston in the stock Cylindrical carvings (sleeve seals) Cylindrical covers/ caps	Fixed, frequently dismantled	It varies from suction pressure to working pressure
Caps of the discharge valves		Does not change
Flanged joints of the fixed parts	Fixed assemblies	Suction pressure
Flanged joints of the receiving part		
Diaphragms of pneumocompensators, pistons	Moving separators	Working pressure

The piston-cylinder carving of the pump is being placed in very difficult condition. The piston moves back and forth inside the groove according to the variable speed limit. In the latter position, the piston stops and changes direction (figure 1).

Figure 1. Piston pump.

The piston of the pump (figure 2) consists of a vulcanized steel core (1) with a sealing rubber sleeve (2) on both sides. Typically, the removable sleeves (figure 2, c) are mounted on 1 smooth sprayed core of the piston and, in turn, are fastened with metal washers 3 held in place by spring rings 2 in the annular slits of the core to prevent discharge.

However, experience to use the pistons in drilling pumps show that the presence of such technological holes in the core has a negative impact on the performance of pistons.

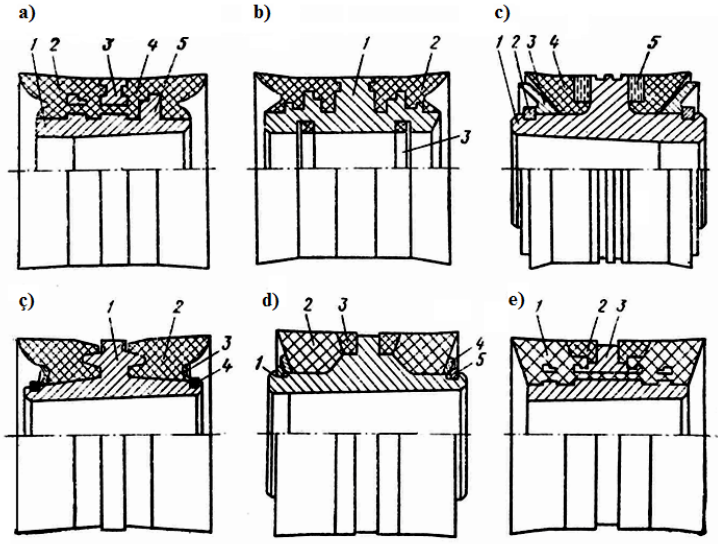

Figure 2. Drawing of pump piston: 1-steel core, 2- rubber sleeve; 3- washers; 4-spring ring; 5- plastic ring.

The design of the piston shown in figure 2 has a washer 3 and a removable sleeve 2 which is fastened with spring rings. There is a ring-shaped additives on the 1 support belt of the core.
In the classic construction of non-removable pistons, plastic ring joints (figure 2, e) were being used. Prior to pouring the rubber into the metal core, plastic rings 2 are installed to form the sealing sleeves 1.
Automatic valves are used on both sides of the opening device which is opened due to pressure difference, in piston pumps The valve, under the influence of its own weight and closed by the force of the spring, prevents the leakage of fluid injected in the reverse direction of the piston or plunger.
The suction and discharge valves of the pumps (figure 3, a) are usually interchangeable. The valve sealing element is pressed against the pump of the valve box, and its hole is covered by the plate when the valve is closed. The inner surface of the sealing element hole is the transverse guide of the plate.
The valve node of the cementing pump unit (figure 3, b, c) consists of a plate sealant. The plate is pressed into the sealing element by means of a spring. The four stamped protrusions of the plate are the lower guides. The body of the hydraulic box is sealed with a rubber ring with a circular cross-section. The inclination angle of the sealing element's seat surface and the plate is 30 degree [3].
The removable sealing ring of the valve is mounted on a special protrusion of the sealing element, then it is held by a steel ring.
As a result of the research, a new valve design of KCK type (figure 3, e) was developed, which differs from the series valves issued according to the current standards by having a plate support not only on the conical seat, but also on the horizontal plane of the rib partition. Due to the fact that the sleeves of the valve are made of IRP-1396-2 rubber, the bulky sealing

part is in the form of a drop, it is impossible to squeeze it at a pressure up to 13 MPa. The valve plate and sealing element are made of 40HNMA steel with a hardness of HB = 48-53, reinforced by one of the coating methods, which increases their strength and allows them to have a large depth of support surface.

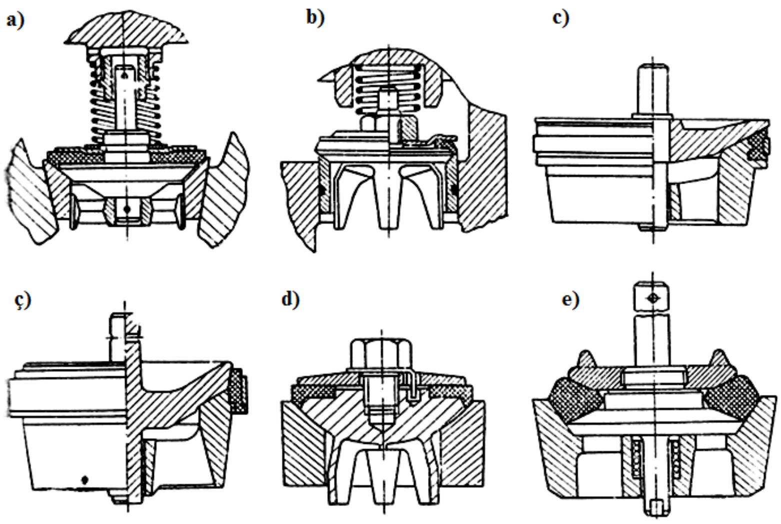

Figure 3. Valves of reciprocating pumps.a. У 8-4 drilling pump valve; b. valve of 9T cementing pump unit;c, ç. drilling pump valve's sealing element; d. 4P-700 brand pump valve; e. KSK type valve for U8-6M brand pump.

Various designed seals are used to separate both slits of the pump cylinder and to seal the gap between the valve box body and cylindrical groove in reciprocating pumps. The main feature of these seals is the work of the sealant during periodically varying pressure forces in the direction of absorption and injection of fluid on one side or the other [4].

Due to the lack of cylindrical engraving seals with rubber rings compressed by the edge of the engraving, it was necessary to use the construction (figure 3, ç) in which the engraving was compressed with two separate crowns. Similar sealing and such fastening of cylindrical engraving was used by Uralmash plant in the construction of U8-7M high pressure drilling pump (figure 4, a, b).

The free space presence allows to compensate for inaccurate details and preparation of details. Considering the rubber rings do not touch the edges of the sealed crack, they lead to prevent the seal from collapsing, as the rubber of the ring cannot flow through the sealed crack.

The main requirement for correct operation of such a seal is to limit the size of H during installation of the seal until it deforms the sealing ring. The incompressive condition of rubber determines the stability of sealing rings volume at any deformational degree. It follows that the cross section of sealing rings does not change during deformation. If sealing rings total cross-section before deformation is as follows:

$$S = \frac{\pi d^2}{4} n, \tag{1}$$

and the total area of the cross section of the sealing ring, which completely fills the initial free space during deformation is:

$$S = bH \,, \tag{2}$$

then we calculate the smallest value of H by the following formula:

$$\frac{\pi d^2}{4} n = bH_{\min} \tag{3}$$

Where d is the cross-sectional diameter of the sealing rings; n - number of rings in the seal; b is the largest possible radial size of the deformed ring.
If we assume that b slightly differs from d, then we obtain

$$H_{\min} \geq 0{,}785dn \tag{4}$$

Large cross-section circular rings were used in cylindrical engraving seals on 12Gr and BrN-1 drilling pumps of Volgograd Barricade plant [5].
Three types of covers are avaliable in reciprocating pumps; cylindrical, discharge and suction valve covers are available. A common feature of these covers is that they are immobile and often dismantled. Frequent disassembly places special requirements on the wear resistance of the sealing parts when replacing pumps' worn parts.
The sealing node of the plunger and the seal of the cylindrical cover work under the same conditions, the difference is that the installation does not place any requirements against wearing. Hydraulic box of the pump and the body of the seal is rarely dismantled, instead, in some designs of pumps, the complexity of its dismantling places high demands on the quality of sealing. As in the case of cylindrical cap seals, the difficulty of eliminating air intake without sealing is due to the impossibility of compressing the sealing sleeve.

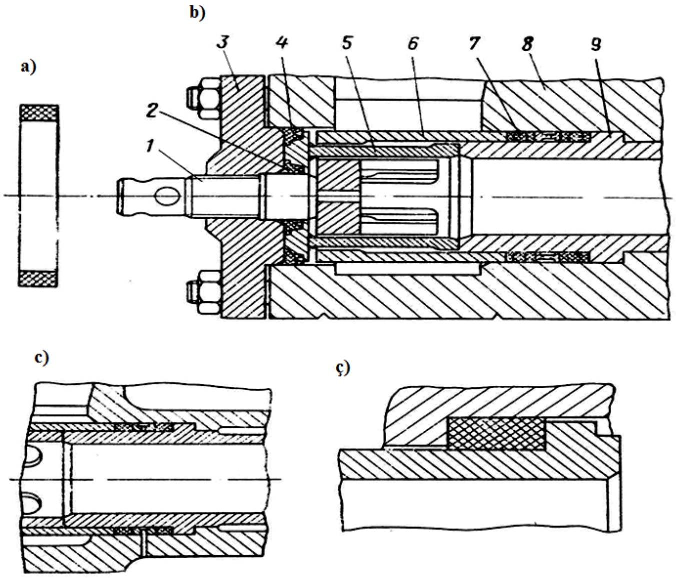

Figure 4. Sealing of cylindrical carving. standard drilling pump carving sealing (sealant); b) Sealing of U8-6M pump's carving; c) fastening node and sealing of U8-7M pump carving; ç) 9T pump carving sealing node.

Results: During the operation of the pump, sealing is subject to the pressure of the fluid on both sides of the piston. In the absence of required tension, the resistance decreases under the influence of various variable forces, and the tightness of the joining surfaces in the parts is violated.

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ИССЛЕДОВАНИЕ РАБОТЫ УПЛОТНИТЕЛЬНЫХ УЗЛОВ ПОРШНЕВЫХ НАСОСОВ

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АННОТАЦИЯ

В статье представлены сведения об исследовании насосов и их герметизирующих узлах, применяемых при бурении скважин, цементировании обсадных колонн, закачке глинистого раствора в скважины при бурении, освоении продуктивных горизонтов, а также используемых при эксплуатации. Во время работы поршневого насоса узлы уплотнения подвергаются давлению потока жидкости с обеих сторон поршня. Оборудование для транспортировки жидкостей и суспензий работает при высоких нагрузках, таких как эрозия, коррозия, кавитация и истирание. Такие случаи оказывают существенное влияние на ухудшение рабочих характеристик насосов, коррозию рабочих поверхностей насоса, приводящую к снижению КПД, увеличению потребляемой мощности, увеличению общего срока службы при расчетах. Неспособность решить эти проблемы поставит под угрозу целостность насосного оборудования и приведет к его выходу из строя.

Ключевые слова: поршневой насос, плунжер, клапанная пара, седло клапана, уплотнение клапана, тарелка клапана.

PİSTONLU NASOSLARIN KİPLƏNDİRİCİ DÜYÜNLƏRİNİN İŞİNİN TƏDQİQİ

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XÜLASƏ

Məqalədə quyuların qazılmasında, qoruyucu kəmərlərin sementlənməsində, qazma zamanı quyulara gili məhlulun vurulmasında, məhsuldar horizontların mənimsənilməsində, həmçinin istismarında istifadə edilən nasoslar, onların kipləndirici düyünlərinin işinin tədqiqi barədə məlumat verilir. Pistonlu nasosun işi zamanı kipləndirici düyünlər pistonun hər iki tərəfində maye axınının təzyiqinə məruz qalır. Maye və suspenziyaları nəql edən avadanlıqlar erroziya, korroziya, kavitasiya və sıyrılma kimi yüksək yüklənmələr şəraitində işləyirlər. Bu kimi hallar nasosların işçi xarakteristikalarının pisləşməsinə, nasosun işçi səthlərinin yeyilməsinə əhəmiyyətli təsir göstərərək, FİƏ-nin aşağı düşməsinə, elektrik enerjisi sərfinin yüksəlməsinə, hesablamalar zamanı ümumi xidmət müddətində cəm istismar müddətinin artmasına gətirib çıxarır. Bu problemlərin həll olunmaması nasos avadanlığının tamlığını təhlükə qarşısında qoyaraq nəticədə onun sınmasına gətirir.

Açar sözlər: pistonlu nasos, plunjer, klapan cütü, klapanın yəhəri, klapanın kipləndirilməsi, klapan boşqabı.

DESIGN OF CENTRIFUGAL COMPRESSOR IMPELLER FOR OPTIMAL EFFICIENCY

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ABSTRACT

A new methodology is proposed to use comprehensive up-to-date commercial software tools for Heat Exchanger Network (HEN) reliability modelling and optimisation. The idea behind this proposal is that to apply the combination of specific HEN optimisation and reliability software packages has several advantages over the commonly used approach. There is a variety of features that need to be taken into account to choose the right software tool. The HEN design has a significant impact on reliability issues and this should be considered. There are many related issues and features - the robustness, the type of welding, the increment of maximum mechanical resistance, the impact on manufacturing costs, reduction of lost opportunity costs caused by exchanger outages, troubleshooting of heating exchanger problems by operators etc. Fouling should be analysed as it has a significant impact on maintenance issues. Up to 30 % decrease of maintenance costs can be achieved annually by applying advanced reliability results and determining heat exchanger failure causes. These analyses include the investigation of failure causes, prediction of future probabilities of failures, cleaning planning and scheduling and the calculation of reliability and maintainability.
Keywords: heat exchangers, modelling and optimisation, design of network, oil refinery, heat recovery systems.

Introduction: A heat exchanger network is an important part of many processing and power generating plants. In most cases the HEN synthesis and design are assuming steady- state and non-variable operating conditions. In practice they can change and disturbances may occur. Operational maintenance, availability and cost are some of the most important factors of HENs. All heat exchangers must be able to provide a specified heat transfer while maintaining a pressure drop across the exchanger. The propensity for fouling must be evaluated to assess the requirements for periodic cleaning. Fouling affects nearly every plant relying on heat exchangers for its operation. It is the accumulation of undesired solid material at the fluid or solid interface. Fouling introduces various costs,e.g. increased capital expenditure, and increased maintenance, loss of production, quality control problems, cleaning costs, additional hardware, and energy losses [1].Fouling of individual heat exchangers has been the subject of intensive research in recent decades. However, HEN fouling has even more issues. Maintenance considerations in the scheduling of continuous and batch plants have recently received increasing attention. Muller-Steinhagen [2] proposed an integrated approach for developing alternative fouling mitigation strategies based on both experimental and modelling work. Georgiadis et al. considered the short term cleaning scheduling in special classes of HENs [3]. A wide variety of approaches are not limited to mathematical models and practical methodologies. The lifetime of heat exchangers varies with the application. A common practice to fouling mitigation is the implementation of Cleaning-In-Place (CIP) operations. Special heat exchangers exists which eliminate cleaning scheduling and maintenance activities by a self-cleaning mechanism. There are advanced materials, such as the hyper-duplex stainless steel, that designed and developed to increase operating performance and to extend service life in severely corrosive heat exchanger applications. Reliability estimation is a useful tool to improve HEN design subject to uncertainties in the operating conditions. The efficiency of the technique has been proved by Tellez et al. [4]. The analysis of the design constraints has been performed for different possible variations in the operating conditions. Failure analysis contains Fault Tree Analysis (FTA). An FTA of a coolant supply to Heat Exchanger has been described as an example by Lazor in the reliability handbook of Ireson et al. [5] Although several approaches and methodologies have been studied in the field of heat exchanger and HEN fouling, Reliability, Availability and Maintenance (RAM) issues of HENs should be further studied, especially for optimisation purposes. Scheduled and unexpected shutdowns should be differentiated. Maintenance times should be optimised. The characteristics of units should be considered. Another important issue is the mean time between maintenance and the reduction of efficiency to certain levels. Present paper focuses on the possibilities of advanced software tools that support this field. The paper introduces a methodology for effective modelling and optimisation of HEN maintenance and reliability.

Problem statement: Market pressures drive management to achieve higher and higher levels of availability and reliability. However, the motivation to improve reliability is more complex than simply to reduce maintenance costs. There is a need to control reliability for best economic performance. In fact, least cost maintenance is not recommended when a plant strives to achieve high reliability. Maintenance costs are relatively low when compared with loss opportunity costs. HEN reliability issues can be effectively handled with RAMS software. The reliability issues of HENs deserve much more attention. Most designers mainly focus on HEN capital and operation cost optimisation only. The reliability issues of HENs should be properly analysed and improved by relevant software tools. All relevant features affecting availability, reliability and maintenance should be considered while modelling and optimising HENs. One of these factors is the interaction between heat exchangers in the HEN. The main task of optimisation is the appropriate scheduling of cleaning interventions of the individual exchangers in the HEN. It can be based on a priori knowledge of the time behaviour of the thermal resistance of fouling [6]. Further tasks to be optimised are the operating costs of the HEN. The estimation of current and future failure probabilities are required to make the decision for equipment replacements performed at the right time to eliminate unnecessary shutdowns cased by unexpected faults. The simple cases, including series and parallel HENs, and the complex arrangements (block HENs) should be differentiated as their fouling factors are different. The detrimental effect of fouling can be reduced by adopting appropriate measures in HEN design [7]. Important is the choice of local parameters and the HEN structure [8]. Enhancing heat exchanger reliability has many advantages. Cost effectiveness can be increased; plant assets management can be improved.

Component lifetime can be extended, and leakages can be detected prior to shutdown. Improved reliability results in reduced maintenance costs and a better economic performance.

RAM of Heat Exchanger Networks: Reliability is the probability that the HEN will perform satisfactorily for at least a given period of time when used under certain conditions [5]. The availability of a Heat Exchanger Network represents the capability to manage heat and power streams continuously in a usual and regular way. The cause of failures varies, including mechanical failures, corrosion, fouling, and sealing problems. Maintenance has an important role in plant design. Optimum maintenance planning is a key factor in modern HENs. Maintenance covers the activities undertaken to keep the HEN operational (or restore it to operational condition when a failure occurs). A measure of ease and speed a system can be restored to operational status after a failure occurs can be expressed as a percentage. It is called maintainability, i.e. the probability of performing a successful repair action within a given time.Analysing failures is an important way to determine availability and reliability issues of a system, including component failures, service failures, mechanical failures, control system failures (or malfunction), the failures to detect faults, changeover failures, lack of manpower, operator errors, and instrument failures.The most important failure characteristics can be expressed by mean times. The widely used is the Mean Time Between Failures (MTBF), i.e. the reciprocal of failure rate. Further types of mean times are the Mean Time Before Maintenance Actions, the Mean Time Between Repairs, and the Mean Time To Failure.Two types of fouling-induced effects should be identified in HENs [7]:
(i)Changes in outlet temperature of process streams caused by the thermal resistance of fouling in an exchanger; (ii) Changes in inlet temperature of process streams caused by the thermal resistance of fouling in other exchangers (“antecedent exchangers” serving the same process streams). Some problems to be handled are the interaction between heat exchangers within the HEN and the optimisation of online cleaning.

Heat Exchanger Network system analyses: HEN analysis can be approached from various angles.

HEN design analysis: HEN design depends significantly on the types of the heat exchangers used. Several factors should be considered when selecting the heat exchanger type [9]. Determining the list of system components is one of the first steps in availability and reliability analysis. The calculations require different kind of data of each component that build up the system. The component table could be useful to describe these data, because the majority of parts have different characteristics. The next step is to build up a system tree (either as a drawn tree structure or as a table).

Failure analysis: Failure Analysis (FA) or Root Cause Analysis (RCA) is a detailed examination of failed items to determine the root cause of failure and to improve product reliability. It helps with developing tests focused on problematic failure modes and with selecting better materials and/or designs and processes, and with implementing appropriate design changes to make products or processes more robust [10]. FA is strongly supported by reliability software tools.

RAM analysis: Availability analysis identifies the items that can affect system operation. It can be extended by considering the combination of maintainability and reliability data. System Reliability Analysis with Reliability Block Diagrams (RBDs) can be used for testing systems via component reliability and overall system reliability. Weak points can be identified. Different designs can be compared. Maintainability analysis provides data for the optimisation of maintenance and repair actions. These analyses can be performed in all major RAMS software packages.

Comprehensive software for modelling: The significance of the latest approaches of modelling the behaviour of the heat transfer surface by using the fouling resistance factor increased recently. As a result of intensive research, some of the classical theories are either being validated or modified to suit current design and optimisation techniques. Various types of modern software packages are used for modelling the different issues of HENs. Fouling mitigation strategies can be effectively modelled by Fuzzy- Logic Expert Systems (FLES) and Computational Fluid Dynamics (CFD) software. General MILP solvers can be used for optimisation of processes. However, another kind of software can be used for modelling and optimising the reliability related issues of HENs: Reliability, Availability, Maintainability and Safety (RAMS) software. The proper choice of RAMS software depends on many factors (e.g. the place of application, the amount of provided data, calculation precision). Availability can be treated together with reliability and life cycle cost modelling as they are influenced by each other.Some examples of RAMS packages are BlockSim and Weibull++ (ReliaSoft [11]), Reliability Studio (Relex Software Corp. [12]), Item ToolKit (ITEM Software [13]).Most of them can be tested as a trial version for free. A more detailed description was introduced by Sikos and Klemeš [14]. There are specific CAD approaches as well, e.g. process design software tools applied for capacity increase and advance of distillation units, performing predictions of fouling layer structure or improving design [15]. Prototype software 'FiltraDynaSim' for the analysis of the fouling layer properties in microfiltration was introduced by Tung et al. [16] It can be used both to design a membrane filtration system and to predict or monitor its performance during plant operation. Nordman and Berntsson presented a graphical method to determine heaters and coolers for retrofit in HENs [17]. They used advanced composite curves to describe the potential amount of heat that could be saved. They can serve as a monitoring tool to identify heat recovery targets and show the order of units to be targeted for rearrangement. Bertolini et al. determined an algorithm that allows a methodical, automatic approach for management of failure data in oil refineries [18]. It seems to be a significant potential to make substantial improvements in task organisation and decision-making processes. The aim of the Risk-Based Inspection and Maintenance (RBI&M) method proposed is to solve two important problems of reliability of refineries: (i) The personnel available for the analysis of critical items and events is limited and it is not able to assess in detail all the events occurred; (ii) Once defined the critical level of an item or event, reliability department has to decide the best maintenance actions and work orders to carry out. Kim et al. introduced a systematic methodology for designing wastewater and heat exchange networks [19]. Based on cost estimation, networks for water and heat exchange were optimised simultaneously. Their proposal is a useful guideline for wastewater and heat exchanger network design with greater cost efficiency and environmental performance. The method employed a specific strategy to address mixed integer non-linear programming

(MINLP) formulations. The mathematical formulation was based on water-pinch analysis and an expanded transhipment model. Bulasara et al. [20] addressed a revamp study of the crude distillation unit (CDU) heat exchanger network (HEN) of a typical refinery with and without the consideration of the free hot streams available in the delayed coking unit (DCU). Two sub-cases of revamp study have been considered: (a) Installation of new heat exchangers for the entire network; (b) Reutilisation of existing heat exchangers. They demonstrated the relevance of the consideration of DCU free hot streams in CDU revamp and retrofit design choices.

A suggested methodology: The proposed methodology suggests a combination of specific HEN optimisation tools with reliability software packages to improve the effectiveness of reliability optimisation in heat exchanger networks. A structure to conduct reliability, availability and maintainability analyses is presented as well. The reliability analysis of HENs can be performed manually or using reliability software packages. The main advantages of using RAMS software in HEN reliability modelling and optimisation are:(i.) Prediction of replacement times can eliminate costs via failure analysis. Performing Weibull analysis on the failure data yields an estimation of MTBF and provides parameters that can be used to estimate future probability of failure.Based on Weibull parameters, it can be determined if exchangers in the HEN will probably survive until the next scheduled outage. If not, what is more cost effective: to replace now or to wait for the next scheduled outage? The answer if it is worth to wait until the next outage is delivered.
(ii.) Estimation of current probability failure with failure probability calculator.
(iii.) Realization of the establishment of new reliability culture at the plant.
(iv.) Improvement of system reliability.
(v.) Reduction and optimisation of maintenance costs.
(vi.) Reduction of loss opportunity costs.
(vii.) Optimisation of the balance between maintenance costs and loss opportunity costs via cost benefit analysis.
(viii.) Control over equipment reliability.
(ix.) Cohesive modelling modules. System tree, FMEA table, fault tree table, subdiagrams/mirrored blocks/multiblocks, LCC, Weibull tree, throughtput analysis, downtime distributions, parts table, prediction data, FMEA worksheet, maintainability data, maintenance policies, RBD, fault tree diagram, event tree diagram, Weibull graph.
(x.) Simultaneous analyses. This feature can be used to handle Availability, Reliability and Maintenance together.
(xi.) Combining series and parallel subsystems. HENs can be analysed by calculating the reliabilities of individual series and parallel sections and combining them in the appropriate manner.
(xii.) A wide variety of distributions. The standard definition of maintainability $M(t) = 1 - e-\mu t$ can be easily expanded to different distributions. In case of the Weibull distribution, for example, MTTR should be modified from $1/\mu$ to

$$MTTR_{weibull}\quad \eta.r\left(\frac{1}{\beta}+1\right)$$

with the scale (η) and shape (β) Weibull parameters. RAMS software packages allow total system approaches in analyses of individual system components. This contributes to the optimisation of design targets for plants to improve equipment selection, replacement, maintenance, as well as to increase system reliability.

The reliability program: When making the decision which tube bundles to replace, reliability groups should look at the reliability engineering principles and applying their results. The heat exchanger reliability could be supported by specification of the scope of the reliability analysis, which should contain:

•The total number of failures
•MTBF
•Fluid velocity
•Years of service
•The current age of heat exchangers.

From them further requirements are recommended to investigate operation classes, date of installation, date of last replacement, tube age factor table, remaining life factor table, production criticality factor, service conditions, bundle replacement costs, and other considerations (e.g. corrosion mechanism).

Data requirement: Numerous data required to perform reliability and maintainability calculations. They can be classified in three groups. First, general characteristics needed, including the general description of the plant, HEN usage, the period of continuous HEN operation, system structure, HEN arrangement. Data collection time interval need to be settled for further experiments. Features to be considered include but not limited to the list of equipments, units, flow streams of the HEN, the heat duty, the heat transfer area, heat transfer coefficients, shell and tube passes, the number of tubes, and design ΔT. The main groups of candidate variables are: operational (tube/shell inlet temperatures, flowrates, and integral flows), maintenance (peak efficiency), scheduling (crude/condensate in blend), and predicates (shell/tube side outlet temperatures). Availability and reliability calculations are in the second group. They require failure history, durations of failures and shutdowns. Three classes of heat exchanger operation modes should be differentiated both for forced (unplanned) shutdowns (e.g. tube leakage) and scheduled maintenance actions. Equipments of Class A require total unit shutdown. In Class B, production cutback is required by a certain percentage. Class C equipments have no impact on system output (run-to-fail). The optimisation of maintenance actions need variables depending on the system to be analysed. In a crude oil plant, for example, the ones that affect HE fouling required, including operational variables (inlet and outlet variables, flow rates of the crude, fluid temperatures, wall temperature of tubes, tube side and shell side flow rates), maintenance variables (e.g., peak efficiency), scheduling variables (crude/condensate in blend), as well as crude property variables.Analyses can be more easily understood if a figure of the HEN is provided.Process data are required as well, including inlet and outlet, mass flowrate (both for shell side and tube side), the exchanged heat, and the heat transfer surface.

System reliability analysis with RAMS software: System reliability analysis can be divided into parts. Firstly, the required reliability approach should be decided. The steps for conducting reliability analysis of HENs are:
(i)Data collection: gather specific bundle failure history.
(ii)Conduct Weibull analyses on the failure data.
(iii)Failure probability calculations to estimate current probability of failure. During this step, the various operation classes of heat exchangers should be differentiated.
Both the maintenance activities that ensue after a failure and the failure mechanisms at work affect the way of conducting the Weibull analyses of heat exchangers. For tube failures caused by process corrosion, failures are expected to reveal a wearout pattern, represented by $\beta > 1$. If the Weibull analysis results in $\beta < 1$, the heat exchanger is failing due to lack of quality or some other process that occurred during manufacturing. To control heat exchanger reliability, the probability of failure should be accurately predicted. Estimating tube bundle reliability should account for the age of each individual tube that failed. Each tube lifetime is estimated from the installation date of the bundle to the failure date, at which time the failed tube is plugged.

Conclusions: The proposed software modelling and optimisation methodology provides a way to reduce future losses in HENs and reach these aims through optimum maintenance. Several new approaches have been reviewed. The proposed methodology focuses on HEN maintenance through the influence of availability and reliability rather than the optimisation of cleaning schedules only. It has been shown that the failure analysis is capable to predict heat exchanger bundle replacement times, leading to significant savings. All major failure types are taken into account, including heat exchanger breakdowns, fouling, and leakages. Units of HENs are classified through operating classes and the most specific maintenance characteristics of fouling. This can reduce or even eliminate the need for total unit shutdowns required for cleaning. Reliability culture of plants or subsystems can be developed via total system approach, considering antecedent exchangers, HEN structure, local parameters, spares, and cost. The case study demonstrated that the RAMS software approach is capable to find the weak points in HEN maintenance, and emphasize the small corrections that could improve these issues towards optimality.

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ОПТИМИЗАЦИЯ НАДЕЖНОСТИ, ДОСТУПНОСТИ И ОБСЛУЖИВАНИЯ ТЕПЛООБМЕННЫХ СЕТЕЙ

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АННОТАЦИЯ

Предлагается новая методология использования комплексных современных коммерческих программных средств для моделирования и оптимизации надежности сети теплообменников (СТО). Идея, лежащая в основе этого предложения, заключается в том, что применение комбинации специальных программных пакетов для оптимизации и обеспечения надежности СТО имеет ряд преимуществ по сравнению с широко используемым подходом. Существует множество функций, которые необходимо учитывать, чтобы выбрать правильный программный инструмент. Конструкция СТО оказывает значительное влияние на проблемы надежности, и это следует учитывать. Существует множество связанных с этим проблем и особенностей - надежность, тип сварки, увеличение максимального механического сопротивления, влияние на производственные затраты, сокращение упущенных возможностей, вызванных отключениями теплообменника, устранение неполадок с теплообменником операторами и т.д. Загрязнение должно быть проанализировано, поскольку оно оказывает значительное влияние на проблемы технического обслуживания. Ежегодное снижение затрат на техническое обслуживание до 30 % может быть достигнуто за счет применения передовых результатов по надежности и определения причин отказа теплообменника. Эти анализы включают в себя исследование причин сбоев, прогнозирование будущих вероятностей сбоев, планирование и планирование очистки, а также расчет надежности и ремонтопригодности.

Ключевые слова: теплообменники, моделирование и оптимизация, проектиро¬вание сети, нефтеперерабатывающий завод, системы рекуперации тепла.

İSTİLİK DƏYİŞDİRİCİ ŞƏBƏKƏNİN ETİBARLILIĞI, MÖVCUDLUĞU VƏ TEXNİKİ XİDMƏTİNİN OPTİMALLAŞDIRILMASI

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XÜLASƏ

İstilik mübadiləsi şəbəkələrinin (İMŞ) etibarlılığının modelləşdirilməsi və optimallaşdırılması üçün kompleksli müasir kommersiya proqram vasitələrindən istifadə edilməsinin yeni metodologiyası təklif olunur. Bu təklifin əsasını təşkil edən ideya ondan ibarətdir ki, İMŞ etibarlılığının optimallaşdırılması və təmin edilməsi üçün xüsusi proqram paketlərinin kombinasiyasının tətbiqi geniş istifadə olunan yanaşma ilə müqayisədə bir sıra üstünlüklərə malikdir. Doğru proqram alətini seçmək üçün nəzərə alınması lazım olan bir çox xüsusiyyətlər var. İMŞ-in dizaynı etibarlılıq problemlərinə əhəmiyyətli təsir göstərir və bu nəzərə alınmalıdır. Etibarlılıq, qaynaq növü, maksimum mexaniki müqavimətin artırılması, istehsal məsrəflərinə təsir göstərilməsi, istilik mübadiləçisinin kəsilmələri nəticəsində əldən çıxmış imkanların azaldılması, istilik mübadilə operatorları ilə bağlı nasazlıqların aradan qaldırılması və s.kimi çirklənmə təhlil olunmalıdır, çünki bu, texniki xidmət problemlərinə əhəmiyyətli dərəcədə təsir

göstərir. 30% - ə qədər texniki xidmət xərclərinin illik azalması etibarlılığa dair qabaqcıl nəticələrin tətbiqi və istilik mübadiləçisinin uğursuzluğunun səbəblərini müəyyənləşdirməklə əldə edilə bilər. Bu analizlər uğursuzluqların səbəblərinin araşdırılması, gələcək qəza ehtimallarının proqnozlaşdırılması, təmizləmənin planlaşdırılması və planlaşdırılması, eləcə də etibarlılıq və təmir işlərinin hesablanması daxildir.
Açar sözlər: istilik mübadiləsi aparatları, modelləşdirmə və optimallaşdırma, şəbəkənin layihələndirilməsi, neft emalı, istilik rekuperasiyası sistemləri.

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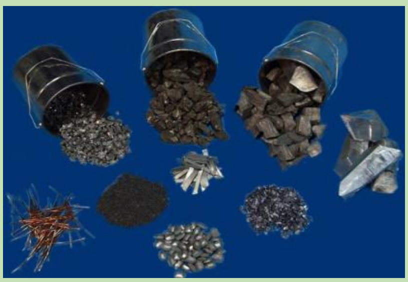







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