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EVALUATION OF THE ENERGY EFFICIENCY OF THE NEW MODEL OF SUCKER-ROD PUMPING UNIT

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Abstract: As is known, in world, oil production is carried out both in the sea and in oil fields on land. For the mechanized exploitation of oil fields on land, in most cases used different type of sucker-rod pumps. The main object of the study of the article is a new constructive solution of the pumping unit, the novelty of which is approved by the ownership certificates. The main factor characterizing the operation of pumping units is its quality indicators such as metal consumption, reliability, efficiency and energy consumption. When switching from a balanced sucker-rod pumping unit to an unbalanced the efficiency factor decreases from 0.834 to 0.65 and the operating power factor decreases from 0.605 to 0.312. In this case, power losses increase sharply not only in transmissions, but also in energy distribution networks. If the engine is not fully loaded or is not selected correctly, then these energy coefficients decrease even more. The selection of engine power is performed according to the worst conditions, i.e. the cyclic mode of operation of an unbalanced pumping unit. Despite this, it can be said that in almost all oil fields, the power of asynchronous motors is greatly exaggerated, which causes additional losses. In article proposed expressions for determining the optimal power of the electric motor were proposed, taking into account the efficiency factor of the balancing device during the balancing of the proposed new constructive solution of the beamless pumping unit with a combined, movable counter weights and rotary counter weights. This allows to prevent excess energy consumption and reduce energy costs. Which is very important in the era of globalization, when energy prices are rising sharply.

Keywords: *oil, pumping unit, efficiency, balancing, electric motor.*

Introduction.

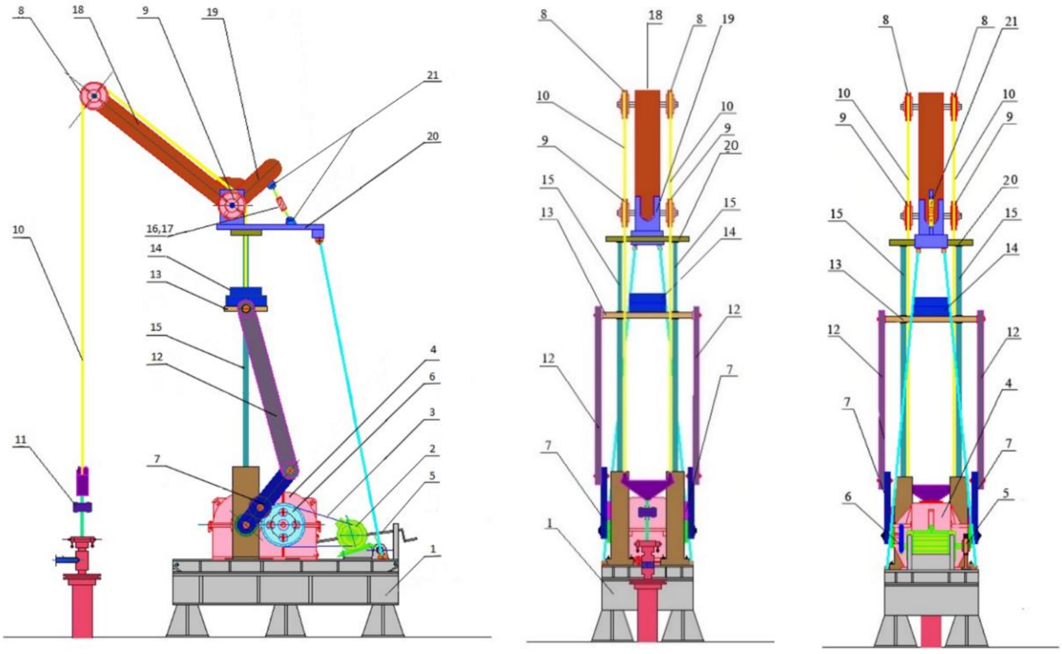
Oil extraction has important influence for the economic development of many countries, including the Middle East, North Africa, some countries of the America and the CIS. In this regard, the development of technologies in the oil industry is of particular importance. Today, most oil wells are onshore and are equipped with rocking machines, also known as sucker-rod pumping unit. These pumps are equipped with individual drives and are installed in wells, connected to the drive through a flexible mechanical connection called a rod column [1, 2, 3].

Sucker-rod pumping unit have played a key role in oil operations since the 20th century, and are considered the most reliable and easily accessible maintenance equipment, according to experts [4]. These pumping systems incorporate a four-bar mechanism that converts the rotary motion of the motor into vertical motion of the rods and consist of several independent assembly units [5]. Considering that most onshore fields are in the late production stage, characterized by a decrease in well productivity and a decrease in crude oil prices on world markets, increasing the energy efficiency of mechanized oil production methods becomes especially relevant. The efficiency and reliability of these pumping systems play an important role in determining the net return on oil production while minimizing energy costs.

Formulation of the problem.

In many countries, the exploitation of oil wells on land is carried out directly with the help of rod well pumps. Statistics show that about 2/3 of active wells are operated by rod well pumps. To lift the oil from inside the well, to ensure the operation of the well pumps with a rod that is lowered into the well, a device called sucker-rod pumping unit, which are ground equipment, is used. The rod well pump and sucker-rod pump is the most widespread mechanized complex used in the artificial extraction of oil, distinguished by its simple operation, high reliability, and satisfactory productivity. Undoubtedly, the sucker-rod pumping unit is the leading point of this mechanized complex. It is the number of trips of the suspension point of the sucker-rod pumping unit, the length of the travel path, the power of the electric motor of the mechanical transmission, the number of transmissions, and the structural features that determine the effective operation and economic efficiency of the well. For this reason, in order to ensure the operation of onshore wells, serial production of pumping units with the most diverse designs is organized in many countries of the world.

At present, beam and beamless sucker-rod pumping unit are used for the mechanization of rod well pumps. Although in practice more beam sucker-rod pumping unit are used, these machines differ in that the cost is high and the weight is heavy due to the high metal capacity. This factor creates obstacles to its installation in harsh and unfavorable climatic conditions, as it causes serious requirements to be placed on the foundation built for the installation of the device [6, 7].



a) b) c)

Figure 1. New constructive solution of beamless pumping unit consisting of the slider-crank mechanism and the rope-block system:
a - side view; b - front view; c - rear view

One of the main solutions to these problems is to use beamless sucker-rod pumping units, which dispense with the balancer and balancer head, which make up most of the pumping units metal capacity. At the same time, it should be specially noted that, in comparison with beam pumping units, the regularities of movement of the rods suspension point and the dynamic forces affecting its construction in beamless pumping units are significantly different in a positive sense [8, 9]. Taking into account the above mentioned, at the department of “Mechatronics and machine design” of the Azerbaijan Technical University has lower metal capacity, compact overall dimensions, the regularity of movement of the suspension point is closer to the harmonic law, high reliability and low energy consumption, the originality of the construction is approved by Eurasian Patent Organization [10] and the Intellectual Property Agency of the Republic of Azerbaijan [11] the construction of a new beam-less sucker-rod pumping unit consisting of a rope-block and crank-slide converter mechanism has been developed .

At Figure 1 shown the overview of the new constructive solution of the sucker-rod pumping unit [12]. The new constructive solution of the sucker-rod pumping unit consist from frame (1), three-phase short-circuited asynchronous motor (2), V-belt drive (3), rigidly connected two-flow three-stage reducer (4), on the drive shaft of which on one side mounted double-shaped brake (5) and on the other side V-belt pulley (6), and on the driven shaft of its installed two cranks (7); guide blocks (8, 9), ropes (10) connected to the rods suspension point (11). The converting mechanism, which consists of two slider-crank linkage (12), converts the rotational motion of the crank into the upstroke and downstroke movement of the rod suspension point. The mechanical transmission has counterweights whose weight can be adjusted (14), located on a movable cross beam (13), which is connected with hinge joint to the connecting-rod.

The guide blocks are surrounded by a flexible rope, one end of which is connected to the movable beam, and the other end is connected to the rods suspension point. In addition, the mechanical drive has a guide system consisting of two vertically located cylindrical tubes (15) and the movable beam. The mechanical transmission has articulated front (18) and rear (19) arms, which can be adjusted by using (16,17) screw tensioners, a fixed cross beam (20) rigidly connected to the guide tubes. The screw tensioner (21) is connected to the frame of the construction, as well as to the joints with the front and rear arms.

The converting mechanism is used on the pumping unit to ensure the upstroke and downstroke movement of the rods column. Currently, the four-link hinged slider-crank linkage mechanism is used in the converting mechanisms of the pinches. It is known that the purpose of kinematic analysis of each hinged mechanism is to determine the displacement, velocity and acceleration of its corresponding points [13, 14, 15].

The operation of sucker-rod pumping units meeting the set requirements under the given conditions is characterized by its quality indicators. The main quality indicators of modern sucker-rod pumps include the efficiency of the mechanical system that makes up it, as well as the energy consumption of its engine. These two quality indicators, which are closely related to each other, are the main factors that determine both productivity and economic efficiency of the sucker-rod pump [16].

At present, when determining the useful work coefficient of the mechanical transmission of the rod well pump, expressions built on the “sequential” scheme are used. At this time, it is assumed that the transfer of energy is carried out in one direction. The transmitted power is determined based on the power of the driving engine. In this case, the total losses are defined as the sum of the energy losses in each element of the transmission. The total useful work coefficient is determined as the product of the useful work coefficients of the separate pairs that make up the transmission [17].

As is known, the useful work coefficient of the mechanical transmission of the rod well pump depends on the ratio of the good work done during the cyclic movement of the rod column to the total work spent by the engine of the transmission. Let's consider the process of transferring energy from individual blocks to each other according to the adopted structural scheme [18, 19].

Some work must be done to raise the rods column between the actuator and the balancing device of a pumping unit equipped with a balanced device. During the upward movement of the rods column, this energy is transferred from the balancing device to the actuator device, and during its downward movement from the actuator device to the balancing device [20, 21].

Successful competition in the market economy is possible only if the costs incurred in the production of the product are minimal. For the case studied, it is related to the cost of electricity during oil production through sucker-rod pumping unit. Recently, technological operations requiring additional energy costs have been applied due to the decrease in the flow rate of the wells, which leads to an increase in the cost of each ton of extracted oil. In this regard, one of the important issues is finding ways to apply energy saving techniques and technologies.

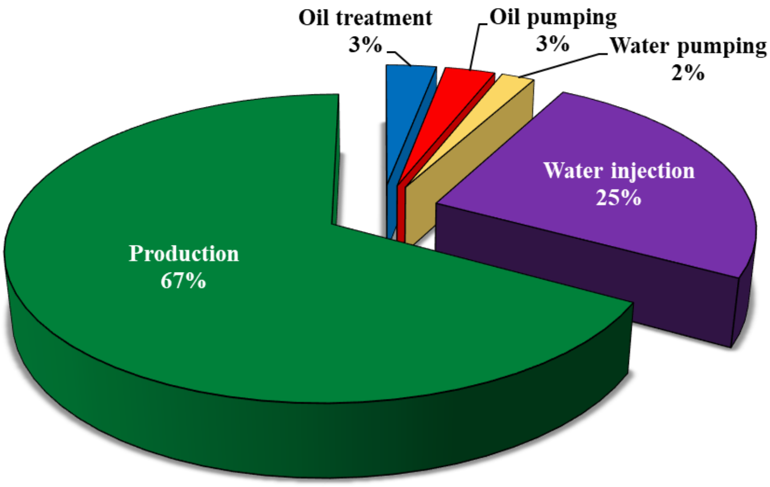


Figure 2. The structure of electricity costs for the whole complex of technological processes of oil production

According to statistical studies, a large part of the structure of electricity costs is spent on oil production (Figure 2). These costs are specific for each operated field, and are mainly determined by the dynamic level of wells producing oil and the structure of the pumping equipment fleet being operated.

Solution of the problem

The main energy in the electrical supply systems of the pumping units is consumed in the engine of the transmission mechanism. Asynchronous motors with a short-circuited rotor with a high starting torque with a maximum voltage of 380V, a power of 1.5...55 kW are used in the transmission of the sucker-rod pumps [22]. The synchronous rotation frequency of the electric motors used in the transmissions of these looms is 500, 750, 1000 and 1500 min^{-1}, the ratio of the starting torque to the nominal torque $M_i / M_n = 1.8...1.9$ and the ratio of the maximum torque to the nominal torque $M_{\max} / M_n = 2.1...2.8$ are within the limit.

Asynchronous motors work with a small operating rate. Because the required starting torque of the electric motor of the looms is 5...6 times higher than the torque in the fixed mode. Engine loading does not exceed 40%. Usually, these engines have small power factor $cos\varphi$ and efficiency.

The value of the power coefficient ($cos\varphi$) for asynchronous motors is 0.55...0.64, and it depends on both the shape coefficient of the load curve K_f and the power of the motor.

The operating mode of electric motors is long-term under a cyclically changing load. Therefore, to determine the power of an electric motor, it is enough to study one of its working cycles. In this case, one of the methods of equivalent torque, equivalent power, or equivalent current is used.

The equivalent or effective power P_e of the electric motor is determined by the following expression

$$P_{eff} = \sqrt{\frac{P_1^2 t_1 + P_2^2 t_2 + ... + P_n^2 t_n}{t_1 + t_2 + ... + t_n}} \quad (1)$$

where t_1, t_2 – interval times; P_1, P_2 - is the average value of the power in the interval.

According to the expression proposed by B.M. Plyush, the effective power is determined by the following expression

$$P_{eff} = \left(k_1 + k_2 G_{liquid} S\right) \frac{n}{\eta_{mex}} \quad (2)$$

where G_{liquid} - is the weight of the liquid column on the plunger; S - travel of the rods suspension point; n - the number of swings per minute of the rods suspension point; η_{mex} - efficiency factor of the transmission mechanism of the sucker-rod pumping unit; k_1 - is a constructive coefficient, depends on the type of pumping unit; k_2 -is the calculation coefficient.

The effective power of the engine is also determined by another expression:

$$P_{eff} = 1{,}7 \cdot k_0 \cdot k_a \cdot d^2 \cdot H \cdot S \cdot n \cdot 10^{-7} + P_s \quad (3)$$

where k_0 - the relative form factor of the torque loading curve on the motor shaft; $k_0 = \frac{k_f}{1{,}11}$; k_a - the correction factor that takes into account the deformation of rods and pipes; H - pump-setting depth; P_s - constant losses on the pumping unit.

Calculating the losses in the electric motor of the pumping unit is complicated because the load changes periodically in each oscillation cycle. Therefore, within this cycle, all the parameters of the electric motor, as well as the efficiency factor and the power factor, change. The graph of the change of the active power consumed by the electric motor during one cycle is shown in figure 3 [23].

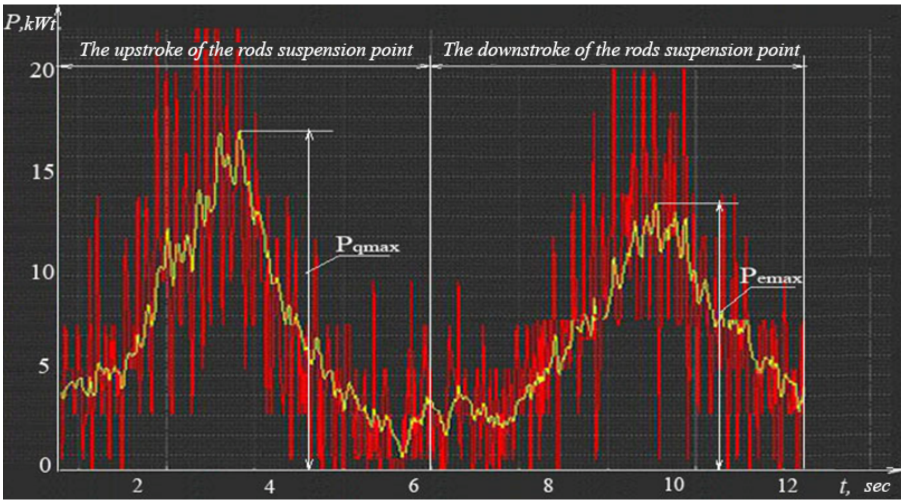


Figure 3. The graph of the change of the active power consumed by the electric motor of the pumping unit during one cycle

Even in ideally balanced pumping unit, the engine load schedule is uneven. For this reason, both efficiency factor and $cos\varphi$ of the asynchronous motor decrease in the direction opposite to the value corresponding to the constant load. At this time, the engine's efficiency factor drops below the nominal value even when the nominal and mean square powers (P_n and P_{kv}) are equal.

When the engine works under a periodically changing load, its efficiency factor and power factor depend on the shape coefficient K_f of the loading curve. If the balancing of the pumping unit is not enough, then the shape factor of the load curve K_f increases, which causes an additional decrease in the engine efficiency. Incomplete loading of the engine due to heating, i.e., the case of $P_{kv} < P_n$, further reduces its energetic characteristics.

According to the load characteristics of the given electric motor, the equivalent powers efficiency factor η_e and power coefficient $cos\varphi_e$ are determined. Then, based on these, the cyclic values of these coefficients are determined depending on the shape factor K_f of the loading curve.

The useful work coefficient of the electric motor of the transfer of the pumping unit in variable cyclic loading is determined by the following expression:

$$\eta_{md} = \frac{\eta_{en}}{\eta_{en} + (1 - \eta_{en})K_f} \tag{4}$$

there η_{en} - Energy conversion efficiency, corresponding to the mean square electric power of the engine; is the shape factor of the loading curve.

The shape factor K_f of the loading curve is equal to the ratio of the mean square power P_{msq} to the mean power P_m during one cycle

$$K_f = \frac{P_{msq}}{P_m} = \frac{\sqrt{\frac{1}{T}\int_0^T P^2 dt}}{\frac{1}{T}\int_0^T P dt} \tag{5}$$

where P - is the power generated by the engine during time t; T - is the time of change of the load in one cycle, or the time of one complete cycle of the elbow of the pendulum machine.

If the balancing of the sucker-rod pumps is insufficient $K_f \geq 4$ and if it is underloaded $(K_{load} \leq 0.3)$ it causes a sharp increase in the loss of electrical energy. Therefore, one of the important issues during the operation of the sucker-rod pumping unit is to ensure the balancing of the engine load schedule and the correct selection of its power.

The graphs of the change of the energy conversion efficiency (η) and the power factor $(cos\varphi)$ on the motor shaft are called the operating characteristics of the pumping unit engine. The average values of the parameters during a cycle are called cyclic energy conversion efficiency (η_{ts}) and cyclic power factor $cos\varphi_{ts}$.

The operating power factor in cyclic loading is determined by the following expression:

$$\cos\varphi_{ts} = \cos\varphi_{en}\left(\frac{\eta_{en}}{K_f} - \eta_{en} + 1\right) \tag{6}$$

where $\cos\varphi_{en}$ - is the coefficient corresponding to the root mean square power at constant loading.

When switching from a balanced sucker-rod pumping unit to an unbalanced when the engine is fully utilized due to heat, the energy conversion efficiency decreases from 0.834 to 0.65 and $\cos\varphi_{ts}$ decreases from 0.605 to 0.312. In this case, power losses increase sharply not only in transmissions, but also in energy distribution networks. If the engine is not fully loaded or is not selected correctly, then these energy coefficients decrease even more.

The selection of engine power is performed according to the worst conditions, i.e. the cyclic mode of operation of an unbalanced pumping unit. For this case, the shape coefficient of the loading graph corresponds to $K_f = 3.94$.

Despite this, it can be said that in almost all oil fields, the power of asynchronous motors is greatly exaggerated, which causes additional losses.

The power consumed by the electric motor, that is, the power of the mechanical transmission of the rod well pump as a whole, depends on the efficiency factor in cyclic loading:

$$P_\Sigma = \frac{P_{eff}}{\eta_{ts}} \tag{7}$$

The efficiency factor of the electric motor of the transfer of the sucker-rod pumping unit in variable cyclic loading is determined by the following expression:

$$\eta_{ts} = \frac{P_m}{P_m + \Delta P} \tag{8}$$

where P_m - the mean power on the engine shaft during one cycle during; ΔP - is the average value of power loss in the engine during one cycle.

As mentioned, the energy efficiency of the electrical transmission is evaluated by its main characteristics, the energy efficiency and the power factor. However, the expressions proposed for their calculation are true when the work process does not change depending on time.

If the load changes significantly depending on time t, then the concepts of energy loss estimation in time period t should be used:

$$A = \int_0^t P(t)dt \tag{9}$$

and

$$\Delta A = \int_0^t \Delta P(t)dt \tag{10}$$

Cyclic energy efficiency is used as the main energy indicator during the cyclic operation of the mechanical transmission of the rod well pump in the time period t_{ts}

$$\eta_{ts} = \frac{A_{ts}}{A_{ts} + \Delta A_{ts}} = \frac{\int_0^{t_{ts}} P(t)dt}{\int_0^{t_{ts}} P(t)dt + \int_0^{t_{ts}} \Delta P(t)dt} \tag{11}$$

there A_{ts} and ΔA_{ts} are the usable power and power loss during one cycle, respectively.

Cyclic power factor

$$\cos\varphi_{ts} = \frac{A_{ak}}{A_{tot}} = \frac{\int_0^{t_{ts}} P_{ak}(t)dt}{\int_0^{t_{ts}} P_{tot}(t)dt} \tag{12}$$

there A_{ak} and A_{tot} - active and total energy spent during one cycle; P_{ak} and P_{tot} are the active and total power consumed during one cycle.

The power consumed by the engine of the pumping unit consists of the good work spent on lifting the liquid and the work spent on overcoming the power losses in the equipment.

The power consumed by the device during calculation with the specified method is determined by the following expression

$$P_M = P_{XG} + \Delta P_{KL} + \Delta P_{SM} + \Delta P_{SH} + \Delta P_{SNS} + \Delta P_{MD} + \Delta P_{RED} + \Delta P_{BL} + \Delta P_{EM} + \Delta P_{IS} \tag{13}$$

there P_{XG} - the usable power used to lift the liquid from the well; ΔP_{KL} - power loss in pump valves; ΔP_{SM} - power used to overcome mechanical friction on the rods; ΔP_{SH} - the power used to overcome the hydrodynamic friction on the rods; ΔP_{SNS} - the force used to overcome the friction of the plunger in the cylinder of the pump; ΔP_{MD} - power loss in the elements of the sucker-rod pumping unit; ΔP_{RED} - power loss in the reducer; ΔP_{BL} - power loss in belt transmission; ΔP_{EM} - power loss in the electric motor; ΔP_{IS} - is the power loss in the control system.

However, it is difficult to use the refined method in practice. Because it is necessary to know many parameters about the work of the pump suspended in the well, which cannot be measured and can only be calculated with certain errors and assumptions.

As we mentioned, the engines of the sucker-rod pumps work in a very complex variable mode. Power selection for variable loads is more complicated than power selection for steady operation. In addition, sucker-rod pumps differ from other mechanisms in that each type of pump requires not one, but several engines, which differ from each other based on the price of power, and in some cases, even the frequency of rotation. This is explained by the fact that each sucker-rod pumping unit has its own field of application.

The power of the motor that drives the well pump unit depends on many factors, first of all the good work done to lift the fluid from the formations per unit time.

This work can be defined simply as follows:

during the upward movement of the rods column

$$A_{up} = (G_{rod} + G_{liquid})S \tag{13}$$

during the downward movement of the rods column

$$A_{down} = -G_{rod} S \tag{14}$$

effective work (useful work) on the double movemen

$$A = A_{up} + A_{down} = (G_{rod} + G_{liquid})S - G_{rod} S = G_{liquid} S \tag{15}$$

The weight of the liquid column on the plunger of a well pump is proportional to the area of its plunger (the square of its diameter). Therefore, the power used to lift the liquid will be as follows

$$N = Hd^2 nS \tag{16}$$

If both the efficiency factor of the device (η_Σ) and the coefficient (η_{en}), which takes into account the load characteristic of the electric motor, are known, then its power can be accurately determined in the following way

$$N_{or} = \frac{1}{\eta_\Sigma \eta_{en}} Hd^2 nS \tag{17}$$

The quantity Hd^2nS actually represents the hydraulic or positive force, i.e. the lifting of a liquid of volume Q to a height H.

The useful load can be defined by the following expression

$$N_{us} = 0{,}114 \cdot 10^{-3} QH \tag{18}$$

where Q - flowrate of well, m^3/day ; H - is the lifting height of the liquid, m .

When choosing an electric motor based on the maximum value of the tangential force, it will work in the half-load mode at small values of the efficiency factor and the power factor $(cos\varphi)$. The operating experience of lathes shows that if the electric motor is selected based on the average value of the tangential force, then the motor will overheat. Because it will be overloaded most of the time. During the incomplete charging period, it will not have time to cool down enough. Currently, various methods and expressions are used to determine the required power of the motor of the pumping units.

In order to rationally use the power of the engine and to create a normal thermal regime for it, the engine is usually selected based on the mean square tangential force:

$$N_{kv} = \frac{\omega r}{\eta} T_{kv} \tag{19}$$

where ω - is the angular velocity of the elbow; r - elbow radius; η - efficiency factor of the depth pump; T_{kv} - is the root mean square tangential force.

The operating experience of sucker-rod pumping units shows that the determination of engine power by the expression (20) is convenient for wells with a relatively small depth. D.V. Efremov's expression can be used to determine the power of the engine for wells with a large depth

$$N = 0{,}0409 \cdot \pi \cdot D_{pl}^2 \cdot S \cdot n \cdot \rho \cdot g \cdot H \cdot \left(\frac{1 - \eta_n \cdot \eta_{md}}{\eta_n \cdot \eta_{md}} + \eta_0 \right) \cdot k \tag{20}$$

Where D_{pl} - diameter of the plunger; S - travel of the rods suspension point; n - the number of departures; ρ - the density of the extracted liquid; H - liquid lifting height; η_n - efficiency factor of the depth pump; η_{md} - efficiency factor of the pumping unit; η_0 - pump efficiency; k - is a coefficient that takes into account the degree of balancing of the sucker-rod pumping unit.

In connection with the application of electric motors with a rotation frequency of 1500 min^{-1} in oil fields, an empirical statement was proposed to determine the required power of the electric motor for the transfer of pumping unit:

$$N = \left[\frac{0{,}02 n d_n^2 S}{(k - \varphi)\sqrt[3]{0{,}3 + \eta_0}} + \frac{0{,}1Q}{0{,}5 + \eta_0} \right] H \cdot \psi \tag{21}$$

where n - the number of movements of the balancer per minute; d_n - pump diameter; η_0 - pump efficiency; k - coefficient that takes into account the type of pumping unit; φ - the length function of the rods column; Q - flowrate of well, m^3/day; H - liquid lifting height; ψ - is the nominal shift function of the selected electric motor.

If we accept the combined balancing method of the new pumping unit that we have studied, then the work done during one cycle, taking into account the efficiency factor of the balancing device:

during the upward movement of the rods column

$$A_{up} = \left(G_{rod} + G_{liquid}\right) \cdot S_g - G_{cw} \cdot S_{cw} \cdot \eta_{bl} - G_r \cdot S_r \tag{22}$$

during the downward movement of the rods column

$$A_{down} = -G_{rod} \cdot S_g + \frac{G_{ey}}{\eta_{bl}} \cdot S_{cw} + G_r \cdot S_r \tag{24}$$

Since the electric motor operates in a fixed operating mode and in the nominal sliding range, its average power can be determined as the ratio of the work done by the motor to time. Then, taking into account the efficiency factor of engine, we can determine the forces expended during the up and down movement of the rods suspension point as follows

$$\left.\begin{aligned} N_{up} &= \frac{\left(G_{rod} + G_{liquid}\right) \cdot \eta_{en} \cdot S_g - G_{cw} \cdot S_{cw} \cdot \eta_{bl} - G_r \cdot S_r}{\eta_{en} \cdot t_{up}} \\ N_{down} &= \frac{-G_{rod} \cdot S_g + \dfrac{G_{cw}}{\eta_{bl}} + G_r \cdot S_r}{\eta_{en} \cdot t_{down}} \end{aligned}\right\} \tag{25}$$

In order to determine the power of the electric motor of the balanced pumping unit $\left(N_{up} = N_{down}\right)$, let us add the right and left sides of the expression (25), then

$$N \cdot \eta_{en} \cdot \left(t_{up} + t_{down}\right) = \left(G_{rod} + G_{liquid}\right)\eta_{en} \cdot S_g + G_{rod} \cdot S_g - G_{cw} \cdot S_{ey}\left(\eta_{bl} + \frac{1}{\eta_{bl}}\right) - 2G_r S_r$$

Taking into account that $T = t_{up} + t_{down}$ and efficiency factor, after a series of simplifications, we can determine the required power of the electric motor of the transmission with the following expression

$$N = \frac{1}{\eta_{mex} \cdot \eta_{en} \cdot T}\left[\left(G_{rod} + G_{liquid}\right)\eta_{en} S_g + G_{rod} S_g - G_{cw} S_{cw}\left(\eta_{bl} + \frac{1}{\eta_{bl}}\right) - 2G_r S_r\right] \tag{26}$$

When we use balancing with a movable counterweight according to the proposed expression

$$N = \frac{1}{\eta_{mex} \cdot \eta_{en} \cdot T}\left[\left(G_{rod} + G_{liquid}\right)\eta_{en} S_g + G_{rod} S_g - G_{cw} S_{cw}\left(\eta_{bl} + \frac{1}{\eta_{bl}}\right)\right] \tag{27}$$

And during balancing with rotary counterweight, it will be as follows

$$N = \frac{1}{\eta_{mex} \cdot \eta_{en} \cdot T}\left[\left(G_{rod} + G_{liquid}\right)\eta_{en} S_g + G_{rod} S_g - 2G_r S_r\right] \tag{28}$$

where the S_g - travel of the suspension point of the bar; S_{cw}- travel of the movable counterweight; S_r travel of the rotor counterweight travel; η_{bl} - efficiency factor balancing device; η_{mex}- is the efficiency factor of the transmission mechanism of the pumping unit.

Results and conclusions.

As a result of the research carried out in the article, the following main conclusions and recommendations can be highlighted:

- one of the main shortcomings of mechanical transmissions of rod well pumps currently used in oil production is related to their high energy efficiency, and the main solutions to the problem is the design of new energy-saving beamless pumping units.

- the value of the equivalent or effective power of the motor of the pumping units depends on the influence of many factors, and even in an ideally balanced device, it is impossible to completely equalize the change in the power consumed by the electric motor during such a cycle.

- when switching from a balanced sucker-rod pumping unit to an unbalanced when the engine is fully utilized due to heat, the efficiency factor decreases from 0.834 to 0.65 and $cos\varphi_{ts}$ decreases from 0.605 to 0.312. In this case, power losses increase sharply not only in transmissions, but also in energy distribution networks. If the engine is not fully loaded or is not selected correctly, then these energy coefficients decrease even more. The selection of engine power is performed according to the worst conditions, i.e. the cyclic mode of operation of an unbalanced pumping unit. For this case, the shape coefficient of the loading graph corresponds to $K_f = 3.94$. Despite this, it can be said that in almost all oil fields, the power of asynchronous motors is greatly exaggerated, which causes additional losses.

- proposed expressions for determining the optimal power of the electric motor were proposed, taking into account the efficiency factor of the balancing device during the balancing of the proposed new constructive solution of the beamless pumping unit with a combined, movable counter weights and rotary counter weights. This allows to prevent excess energy consumption and reduce energy costs. Which is very important in the era of globalization, when energy prices are rising.

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