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ABSTRACT

of the dissertation for the degree of Doctor of Sciences

DEVELOPMENT AND SYSTEMATIC ANALYSIS OF THE NEW CONSTRUCTIVE SOLUTION OF BEAMLESS SUCKER-ROD PUMPING UNIT

Specialty: 3313.02 - Machines, equipment and processes

Field of science: - Technical sciences

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GENERAL DESCRIPTION OF THE WORK

The urgency of the problem. As a result of gaining independence by our republic, the "New Oil Strategy" was launched, based on the far-sighted policy of our National Leader Heydar Aliyev in connection with our national interests. On September 20, 1994, in the Gulistan Palace in Baku, the "Contract of the Century" was signed on the joint development of oil in the deep-water reservoirs of the Azeri, Chirag and Guneshli fields in the Azerbaijani sector of the Caspian Sea[1]. It included 13 leading oil companies from 8 countries. At present, the "New Oil Strategy" and the doctrine are being successfully implemented with the support of the President of the Republic of Azerbaijan Ilham Aliyev. Since the beginning of the implementation of the "Contract of the Century", the Azerbaijani economy has undergone a turnaround. It is no coincidence that oil production and exports form the basis of the Azerbaijani economy and provide 90% of state budget revenues.

Even in the former Soviet Union, the design and improvement of equipment used in oil production in our country has always been in the spotlight. It is no coincidence that in the former Soviet Union, the main leaders in the design and manufacture of borehole pumps used for oil production were the research institutes of our republic[2]. It is the borehole sucker rod pumps developed by these institutes, the pinch mechanisms and gearboxes used in them, have long been considered the standard.

Balancer sucker rod pumps make up 70% of the mechanical transmissions used in the petroleum industry. The main disadvantages of the widely used balancers of rockers are characterized by large losses of power, the appearance of additional dynamic forces due to

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[1] Мир-Юсиф Бабаев. Краткая история азербайджанской нефти. (2-ое издание, переработанное и дополненное). Баку, 2009, 279 с.

[2] Архипов К.И., Попов В.И., Попов И.В.Справочник по станкам-качалкам. Альметьевск, 2000, 195 с.

the fact that the laws of motion of the rods suspension point is differ significantly from the ideal laws of motion, the imperfect design of the converting mechanism and gear reducer (less transmission number, more metal capacity), high metal consumption, the impossibility of using high-speed electric motors, the need for a heavy foundation due to its large mass, the use of several belts due to the heavy load on the belt drive and rapid failure due to uneven distribution of the load between them are the main disadvantages of the mechanical transmission of borehole pumps.

At the present stage of the development of petroleum science, an urgent problem is, first of all, the creation of much more perfect beamless pumping units for mechanical transmission of sucker rod pumps, which provides energy savings, a reduction in overall dimensions, an increase in reliability, which reduces friction losses in kinematic pairs, approximating the laws of movement of the rod suspension point, its speed and acceleration to theoretical laws.

Dissertation work “Development, research and improvement of transmission mechanisms of machines and assemblies” State registration number 0106 Az0910, “Development of a new constructive solution for mechanical drive of sucker rod pumps” State registration number 0108 Az1052 and “Organization of production of a new constructive solution for the mechanical transmission of sucker rod pumps” State registration number 0114 Az1018 was introduced in accordance with the target scientific and technical program and is part of the research work of the "Machine design” department.

Purpose of work: The purpose of the dissertation is to develop and research an innovative mechanical transmission (a new constructive unbalanced clamping machine), the converter mechanism of which consists of a rope-block system, the determination of the kinematic parameters of its converting and transmission mechanism by an analytical method, balancing the cyclically changing load on a new design solution beamless sucker rod pumping unit, determination of the number of swings of the rod suspension point depending on the actual productivity of the well, development of a design that saves energy through the effective use of counterweights, reduces the weight of the mechanical system and greatly simplifies its design and solutions to important scientific problems for assessing its technical condition.

Methods of research. Scientific and theoretical results were compared with operating experience indicators. The research was carried out using methods of systematization of kinematic research, mathematical and statistical, fundamental and applied knowledge.

The main scientific theses for the defense:

- systematization and analysis of information on the calculation, design and construction of existing sucker rod pumping units;
- the development of new constructive solution of beamless sucker rod pumping unit converter mechanism of which consists of a rope-block system which ensures the regularity of movement of the rod suspension point, its speed and acceleration closer to the harmonic law, saves energy through efficient use of reverse loads, reduces the weight of the mechanical system and greatly simplifies its design;
- analytical kinematic study of the converting mechanism of the new constructive solution of beamless sucker rod pumping unit, analysis of its kinematic parameters and establishment of the relationship between these parameters;
- assessment of the effect of the kinematic scheme of the transforming mechanism of the new constructive solution of beamless sucker rod pumping unit on the displacement of the rod suspension point, the regularity of its speed and acceleration;
- ensuring equal loading of the electric motor of the rocking machine, as well as balancing the cyclic variable load on the new constructive solution of beamless sucker rod pumping unit, provided that there is no negative torque on the drive shaft of the reducer;
- investigation of the dynamic forces arising in the tensile element of the new constructive solution of beamless sucker rod pumping unit;
- determination of elastic and constructive displacements arising in the structural elements of the new constructive solution of beamless sucker rod pumping unit in both static and dynamic loading modes and

assessment of the impact of these displacements on the actual value of the path of the pump plunger;

- formation of the scientific basis for the development of new multi-stage cylindrical gear reducers and a detailed calculation of that reducer, taking into account both analytical and experimental study and friction of the coefficient characterizing the internal driving force of oil for double sliding bearing;

The validity and correctness of the main scientific provisions: The provisions, results and recommendations of the dissertation are based on numerous numerical calculations, mathematical and statistical methods, comparison of the proposed new constructive solution of beamless sucker rod pumping unit with the existing constructions and it is provided with a large number of cited reliable official sources, which include the construction, design and calculation of sucker rod pumps.

Scientific novelty of the research: The author for the first time developed the design of a new constructive solution of beamless sucker rod pumping unit with and proposed a scientific basis for its research. For the first time, the kinematic parameters of its converting mechanism were determined analytically and a connection was made between them, the scientific basis of the cyclic variable load balancing method was developed, the dynamic forces created in the tension element of the new constructive solution of beamless sucker rod pumping unit were studied and elastic deformations in its structural elements under the influence of dynamic forces were studied and the actual value of the path of the plunger was determined depending on the absolute values of these elastic deformations, rod suspension point reciprocating speed was determined depending on the actual productivity of the well, a design that allows to reduce the weight of the mechanical system design has been developed and its technical condition has been assessed, which will greatly simplify its construction

Practical significance of the work: The proposed new constructive solution of beamless sucker rod pumping unit have a great practical importance for the oil industry. The scientific provisions of the dissertation can be used as an important practical material for research institutions engaged in the construction and design of sucker rod pumps.

A laboratory design of the new constructive solution of beamless sucker rod pumping unit which converting mechanism consisting from rope-block system, a new designed two-line three-stage reducer, a belt drive and an electric motor has been successfully tested. The application of the new constructive solution of beamless sucker rod pumping unit with allows to save an average of 50% on electricity.

Approbation of the work. Materials of the dissertation and its separate fragments were reported and discussed at international and republican scientific and technical conferences and approved:

- International Congress "Mechanics and Tribology of Transport Systems-2003", Russia (Rostov-on-Don), 2003;
- International scientific-technical conference on modern problems of machinery and instrumentation. Baku, 2005;
- II international scientific congress innovations in engineering. Varna, Bulgaria, 2016
- International Scientific and Technical Conference on Intellectual Technologies in Mechanical Engineering. Baku, 2016;
- 2nd International Scientific and Technical Conference on "Problems of Metallurgy and Materials Science". Baku, 2017.
- Innovative trends in ensuring regional development: Realities and modern challenges. Materials of the republican scientific conference. Mingachevir, 2020.

The main provisions of the dissertation are reflected in 31 scientific articles, 1 methodical guarantee, 2 copyright certificates (patents) of Azerbaijan and Eurasia.

Volume and structure of the thesis. The dissertation consists of an introduction, 8 chapters, main conclusions and recommendations, and sources used. The total volume of the work consists of 318 pages of computer-generated text, including 69 figures, 14 graphs and 16 tables.

In the introduction the history of oil production in the Republic, mechanical transmissions of well pumps at the present stage, relevance of the topic, purpose of the work, current state of the researched scientific problem, main scientific provisions, scientific novelty of the work and its practical significance were determined. The connection, structure and scope of the work with the state scientific-technical programs and scientific-research works are shown.

The first chapter of the dissertation is devoted to the history of the development of the existing sucker rod pumps which used in oil extraction and a comparative analysis of the design of their transmission and converter mechanisms and a brief review of the literature on the calculation, design and production of sucker rod pumps.

As a result of the research, it was found that the overall size and metal capacity of balanced sucker rod pumping units are much higher than those of beamless sucker rod pumping units. The cost of electricity is much higher because the structures used to save power are not more efficient. The longevity of the reducer and the electric motor is reduced due to the negative torque on the output shaft of the reducer of pumping unit. Maintenance work is very difficult due to the impossibility of freeing the area around the well during installation and repair of wells. The fact that the reciprocating speed of the sucker rod pump does not depend on the geological reserves of the well leads to idle stops on the twisting machine, a decrease in its daily productivity and economic efficiency. In addition, the non-use of more advanced reducers on the sucker rod pump significantly reduces the efficiency and reliability of the transmission as a whole. All these provisions make it necessary to create more advanced new designs of sucker rod pumps.

The second chapter is devoted to the analysis of the innovative mechanical transmission consisting of the rope-block system, i.e. the new constructive solution of sucker rod pump and kinematic analysis of it is transforming mechanism.

Recently, along with balanced sucker rod pumps, new beamless sucker rod pumps have been widely used[3]. The design of the newly designed sucker rod should be simple, economically viable, and compact. Taking into account the influence of the above-mentioned factors, at the Department of "Machine Design" of the Azerbaijan Technical University was developed a new constructive solution of beamless sucker rod pump [27] (figure 1).

As a result of the research, it was determined that the proposed new design of beamless sucker rod pump have the following advantages compared to existing oil production equipment:

- It use 1.5-1.7 times less electricity than regular balanced pumping units;
- Since the gearbox does not have negative torque on the output shaft, the gearbox and the engine have a long service life;
- There is no need for a solid and tall foundation;
- It is less sensitive to uneven foundation;
- The absence of a heavy balancer and a balancer had (horse had) reduces the weight of the equipment and greatly simplifies its design;
- Since the change of movement, speed and acceleration of the rod suspension point at proposed pumping unit closer to the harmonica, it practically doesn't arise any additional dynamic forces and vibrations, which significantly increases the service life of the column rods;
- It has small overall dimensions;
- During the installation and repair of the well, it possible to free up space around the well and adjust the movement trajectory of the rod suspension relative to the vertical line;
- The ability to fully fold the front and rear racks makes it easy to transport the pumping unit to the installation site.

The function of the converting mechanism of sucker rod pumping unit is to convert the rotational motion of the engine into the upstroke

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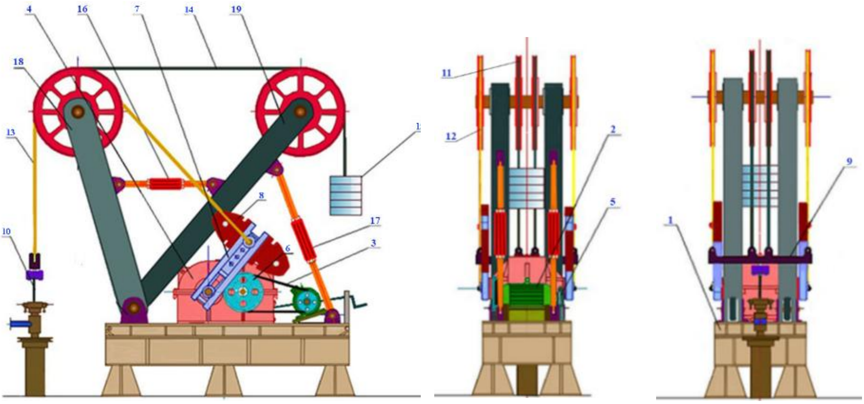
[3] Валовский В.М., Валовский К.В., Басос Г.Ю. и др. Эксплуатация скважин установками штанговых насосов на поздней стадии разработки нефтяных месторождений М.: Нефтяное хозяйство, 2016, 592 с.

and downstroke movement of the rod column. In this case, the normal operation of the deep well pump depends on the law of the movement of the rod suspension point[4]. The main factors influencing the regularity of the rod suspension are the constant angular velocity of the engine, the radius of the elbow, the ratio of the length of the crank with dimensionless parameters to the length of the rope (λ) and the ratio of relative eccentricity (ε). Taking into account the effect of all these kinematic parameters, the path of the suspension point of the rod of the new constructive solution of beamless sucker rod pump is determined by the following expressions in accordance with its speed and urgency:

$$S = r\left[1-\cos\varphi+\frac{\lambda}{4}\left(1-\cos 2\varphi\right)+\varepsilon\lambda\sin\varphi-\frac{\varepsilon^2\lambda^2}{2\left(1+\lambda\right)}+\frac{\lambda\varepsilon^2}{2}\right] \qquad (1)$$



a) *b)* *c)*

Figure 1. The new constructive solution of beamless sucker rod pump which converting mechanism consist from rope-block system: *a- side view; b- front view; c- rear view*

4 Аливердизаде К.С. Балансирные индивидуальные приводы глубиннонасосной установки / Баку: Азнефтеиздат, 1951, 215 с.

$$V = r \cdot \omega \left[\sin\varphi + \frac{\lambda}{2}\sin 2\varphi + \varepsilon\lambda\cos\varphi \right] \quad (2)$$

$$a = r \cdot \omega^2 \left[\cos\varphi + \lambda\cos 2\varphi - \varepsilon\lambda\sin\varphi \right] \quad (3)$$

The third chapter of the dissertation is devoted to the evaluation of the effect of the dexax distance on the kinematic parameters of innovative mechanical transmission. Central (axial) and mixed (dexaxial) converting mechanisms are mainly used as converting mechanisms in sucker rod pumping units. An additional geometric parameter, i.e. the location of the crank (dexax distance), has a significant effect on the kinematics of the mixed inverter.

The location of the crank on the proposed new constructive solution of beamless sucker rod pump can be different (Figure 2). As can be seen from Figure 2, in all three variants the path of the rod suspension point is the same, although the angle of coverage of the rope block is different ($S = R \cdot \cos\varphi$). These variants differ from each other in principle, when the rod moves up and down only with the change of the average speed of its rod suspension point in each half cycle.

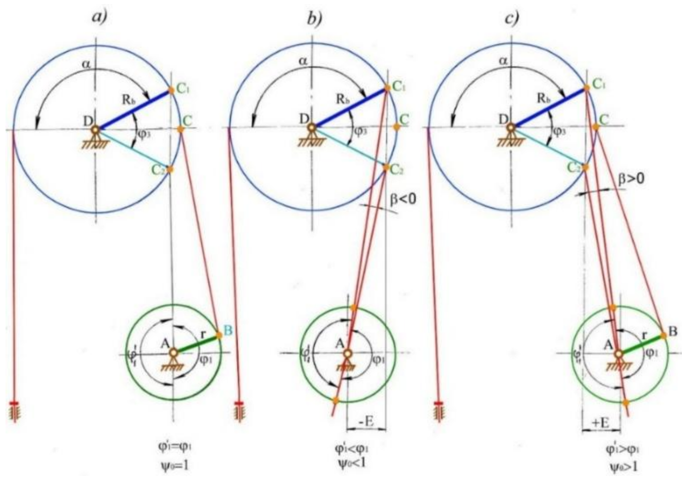


Figure 2. The diagrams of location of the crank on the proposed new constructive solution of beamless sucker rod pumping unit

In the ABCD axial (normal) mechanism shown in Figure 2.a, the crank is located in the center (point A) on a C_1C_2 straight line. In this variant, the time for the rod to move up and down is the same regardless of the direction of rotation of the crank.

In the deaxial (negative deaxial) mechanism shown in Figure 2 b, when the crank rotates clockwise, the bar moves upwards, sooner or later than when the crank rotates counterclockwise, and when the crank rotates counterclockwise, the rod rises the time it moves right is faster than the time it moves down.

In the deaxial (positive deaxial) mechanism shown in Figure 2 c, when the crank rotates clockwise (wellhead to the left), the time the rod moves upwards, sooner or vice versa than when it moves downwards, and the time the crank moves counterclockwise. during rotation, the time it takes for the rod to move upwards is later than when it moves downwards.

The constructions under consideration have advantages and disadvantages. It is known that when the rod suspension point moves upwards, it is loaded more than when it moves downwards. For this reason, the rod suspension point moves unevenly, and as a result, the forces of inertia are greater during the downward movement of the rod suspension than during the downward movement. Therefore, in order to reduce the forces of inertia during the upward movement of the rod suspension point, it is necessary to reduce the absolute value of acceleration during the upward movement. This requires reducing the average speed of the rod as the suspension point moves upwards. To perform this function, it is recommended to use a negative deaxial circuit when the crank rotates clockwise, and a positive deaxial circuit when the crank rotates counterclockwise.

It is not always helpful to rotate the crank counterclockwise. Therefore, during the upward movement of the rod suspension point, negative deaxial pumping units are used to reduce its acceleration. At present, more and more positive deaxial sucker rod pumping units are used in foreign countries.

The choice of sucker rod pumping units also depends on the properties of the extracted liquid. If the distance between the plunger and the cylinder is too long, the fluid will leak downwards, reducing the efficiency of the pump, as the time for the rod to move upwards is too long. Conversely, if a large negative deaxial pumping units is adopted unreasonably, the probability of the rod column hanging in the tubes increases as the lowering speed of the rod suspension point moves downwards.

The fourth chapter deals with the balancing of the cyclically variable load of the new constructive solution of beamless sucker rod pumping unit.

The main function of the balancing device on the pumping unit is to collect the potential energy during the downward movement of the rods column and to return it during the upward movement. In this case, the work done by the potential energy, together with the work done by the engine of the converter mechanism, is spent on good work during the upward movement of the rod suspension point[5].

Currently, the pumping unit operate under cyclically changing loads. Therefore, the work of the pinch transmission is irregular during one cycle, i.e. during the up and down movement of the rod suspension point. If the pumping unit is out of balance, the motor raises the rod column by exceeding the pressure of the fluid on the pump plunger as the rod suspension point moves upwards. In this part of the cycle, the engine is fully loaded. During the downward movement, the rods column descends due to its own gravity. In this case, the pressure of the fluid does not affect the rods because the plunger valve in the plunger is open. During this period, the engine does not run, on the contrary, it absorbs energy and operates in generator mode (Figure 3).

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[5] Агамалов Г.Б., Алиев З.З.,Романова Н.А., Ризванов Р.Р. Методика расчета веса устьевой уравновешивающей системы станка-качалки. Нефтегазовое дело (Электронный научный журнал) 2010,№2, с.1-11

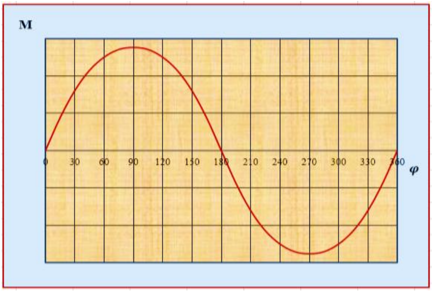

Figure 3. Graph of change of torque on an unbalanced sucker rod pumping unit

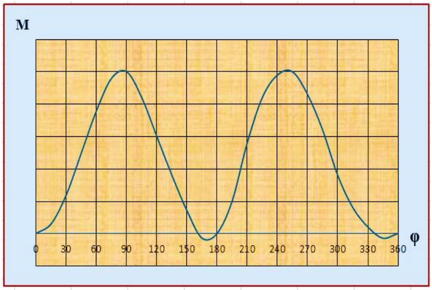

Figure 4. Graph of change torque on the output shaft of the reducer on a properly balanced sucker rod pumping unit

To overcome these shortcomings, the sucker rod pumping unit are usually balanced and the engine is evenly loaded. Figure 4 shows a graph of change torque on a properly balanced sucker rod pumping unit.

In the proposed design of new constructive solution of beamless sucker rod pumping unit are used combined balancing method. In this case, the following expressions are used to determine the weight of the moving counterweights and the distance of the rotary counterweights on the crank, provided that the work done by the transfer during the up and down movement of the rod suspension point is equal to each other[6]:

$$G_{cw} = G_{cw1} + x_{cw} = \left(G_{rod} + \frac{G_{fl}}{2}\right) - \frac{R}{r} G_r + \frac{M_{md}}{C} \qquad (4)$$

$$R = \frac{r}{G_r}\left(G_{rod} + \frac{G_{fl}}{2} - G_{cw} + \frac{M_{md}}{C}\right) \qquad (5)$$

there G_{rod}- is the weight of the rods in the fluid; G_{fl}- is the weight of fluid above the plunger; R- Distance from the crank rotation axle to

[6] Нефтегазопромысловое оборудование. Учебник для ВУЗов. М.: 2006, 720с. (Ивановский В.Н., Дарищев В.И., Каштанов В.С., Сабиров А.А., Пекин С.С., Мерициди И.А., Николаев Н.М.)

the gravity center of the rotor load; r- crank radius; x_{cw}- weight of an extra load; M_{md} - is the sum of the moments of the weights of the individual parts of the pumping unit relative to the support of the front pillars; C - is the distance from the support of the front pillars to the line of action of the force of the movable counterweight.

One of the important issues in the combined balancing method is to determine which portion of the balancing weight is the moving weight, and which portion is going to the rotor weight. In order to solve this problem first we have to define the weight of the rotor load rotor with the expression (4), then it is possible to determine which part of the weight is moving and which is rotor[7]:

$$\frac{RG_r}{r\left(G_{rod}+\frac{G_{fl}}{2}\right)}+\frac{G_{cw}\cdot C-M_{md}}{C\left(G_{rod}+\frac{G_{fl}}{2}\right)}=1 \quad (6)$$

Formula (6) shows that the first component of the sum relates the the rotor, and the second - to the balancing method.

The main part of the mechanical drive for sucker-rod pump is their cranks. Under generated in the flexible element force the cranks transmit the torque moment to the output shaft of ther gearbox. This moment has a substantial impact on the engine's required power. Changing of this forces characterizes the degree of uneven load of the engine. Usually, when determining this force, the frictional forces and inertial forces generated by the movement of the contrweights are ignored.

The torque in the crankshaft of the mecanical drive of sucker-rod pump changes depending on the force in the rope arm to the crank and at the full period from the angles φ and β. Therefore, let's define the force in the rope arm to the crank during the downward and upward movement of the rods suspension point:

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[7] Чичеров Л.Г. Нефтепромысловые машины и механизмы / Л.Г.Чичеров. -М.: Недра, 1983, 312 с.

$$M_b = \left(X_{1up,dw} + X_{2up,dw}\right) r \sin\left(\varphi + \beta\right) - G_r R \sin\varphi \tag{7}$$

there

$$X_{1up} = \left(G_{rod} + G_{fl}\right) 4sin\frac{\alpha}{2} f \frac{d}{2}\frac{1}{D_b} + \frac{G_{fl}}{2} + \frac{R}{r} G_r - \frac{M_{md}}{C}$$

$$X_{2up} = \left(\frac{\left(G_{rod}^{in} + G_{fl}^{in}\right)}{g}\omega^2 r \cdot cos\varphi\right)\left(1 + 4sin\frac{\alpha}{2} f \frac{d}{2}\frac{1}{D_b}\right)$$

$$X_{1dow} = \frac{G_{fl}}{2} - \frac{R}{r} G_r + G_{rod} 4sin\frac{\alpha}{2} f \frac{d}{2}\frac{1}{D_b} + \frac{M_{md}}{C}$$

$$X_{2dow} = -\frac{G_{rod}^{in}}{g}\omega^2 r \cdot cos\varphi\left(1 + 4sin\frac{\alpha}{2} f \frac{d}{2}\frac{1}{D_b}\right)$$

As sown from this equation, the torque is the sum of two moments generated from static and dynamic forces, during the movement of the suspension point of the rods both upwards and downwards.

β	φ	r	R	Gr	X1	X2	M	Gəy
0	0	0,65	1	3940	11879,03	69,20367	0	7242,897
8,3	15	0,65	1	3940	11879,03	66,84561	2051,592	7242,897
15	30	0,65	1	3940	11879,03	59,93213	3517,378	7242,897
18	45	0,65	1	3940	11879,03	48,93438	4122,129	7242,897
20,3	60	0,65	1	3940	11879,03	34,60183	4221,009	7242,897
21	75	0,65	1	3940	11879,03	17,91123	3884,9	7242,897
20	90	0,65	1	3940	11879,03	4,24E-15	3315,713	7242,897
18,3	105	0,65	1	3940	11879,03	-17,9112	2638,098	7242,897
15,3	120	0,65	1	3940	11879,03	-34,6018	2003,209	7242,897
11,3	135	0,65	1	3940	11879,03	-48,9344	1480,509	7242,897
8	150	0,65	1	3940	11879,03	-59,9321	907,8822	7242,897
5	165	0,65	1	3940	11879,03	-66,8456	313,5095	7242,897
0	180	0,65	1	3940	-458,664	28,99254	-5,2E-13	7242,897
5	195	0,65	1	3940	-458,664	28,00465	1115,488	7242,897
8	210	0,65	1	3940	-458,664	25,10828	2143,5	7242,897
11,3	225	0,65	1	3940	-458,664	20,50082	3022,946	7242,897
15,3	240	0,65	1	3940	-458,664	14,49627	3691,399	7242,897
18,3	255	0,65	1	3940	-458,664	7,503822	4098,515	7242,897
20	270	0,65	1	3940	-458,664	5,33E-15	4220,152	7242,897
21	285	0,65	1	3940	-458,664	-7,50382	4050,887	7242,897
20,3	300	0,65	1	3940	-458,664	-14,4963	3608,596	7242,897
18	315	0,65	1	3940	-458,664	-20,5008	2927,399	7242,897
15	330	0,65	1	3940	-458,664	-25,1083	2051,386	7242,897
8,3	345	0,65	1	3940	-458,664	-28,0046	1056,654	7242,897
0	360	0,65	1	3940	-458,664	-28,9925	1,04E-12	7242,897

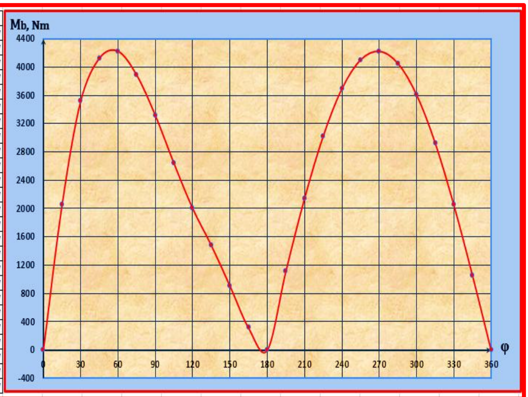


Figure 5. Graph of the torque dependence of the angle of rotation of the crank on the output shaft of the reducer of the new design solution beamless sucker rod pumping unit according to the technical parameters of the sucker rod pumping unit СКД 3-1,5-710

In order to analyze the obtained expressions, calculations were made for the new constructive solution of beamless sucker rod pumping unit in accordance with the technical parameters of the СКД3-1,5-710 sucker rod pumping unit and based on the results of the calculations, the dependence of the torque on the output shaft of the reducer on the angle of rotation of the crank on the values of the crank $r = 0{,}65m$ and the Distance from the crank rotation axle to the gravity center of the rotor load $R = 1{,}0m$. As a result of the calculations, it was determined that both the movable and rotary counter-weight have such a value ($G_{cw} = 7243N$, $G_r = 3940N$), at these values, the pumping unit is properly balanced as the rod suspension point moves both downwards and upwards, and the torque on the reducer output shaft reaches a minimum value (Figure 5).

Figure 6 shows the timing of the rotating torque in the output shaft of the gearbox, the moving and rotor weights according to the technical parameters of different models of the balanced new constructive solution of the mechanical drive for sucker-rod pump.

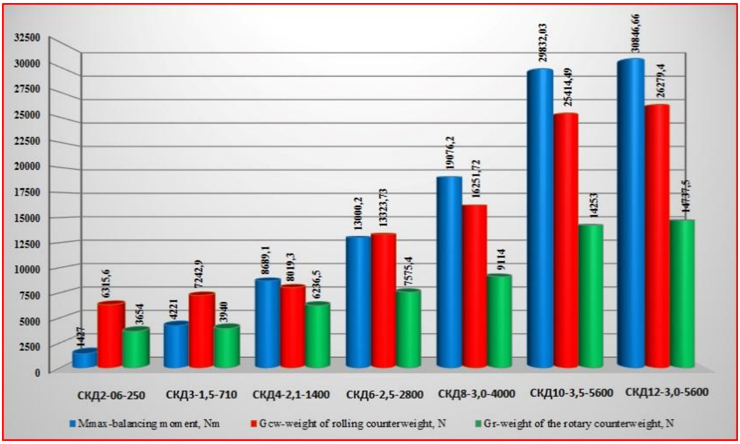


Figure 6. The graph to determine of the rotating torque in the output shaft of the gearbox, the moving and rotary counturweights according to the technical parameters of different models of the balanced new constructive solution of the mechanical drive for sucker-rod pump

Table 1 shows a comparative analysis of the new constructive solutions of sucker-rod pump unit with the normal series of pumping unit. As can be seen from the table in accordance to the technical specifications of various branded new constructive solutions of the mechanical drive for sucker-rod pump allows to obtain approximately 1.6 to 2.1 times the force, which will save about 45% of the electrical energy compared to the normal sucker-rod pump.

Table 1. Comparative analysis of normal range of current and a new designed mechanical drive of sucker rod pumps

Model of sucker-rod pump	**Load on the suspension point, kN**	**Stroke of suspsension point, m**	**Torque in the output shaft of the gearbox, Nm**		**Gain in force, time**	**Energy saving, %**
			Models			
			Existing	**Designed**		
СКД2-0,6-250	20	0,6	2500	1427	1,75	43
СКД3-1,5-710	30	1,5	7100	4221	1,68	41
СКД4-2,1-1400	40	2,1	14000	8689	1,61	38
СКД6-2,5-2800	60	2,5	28000	13000	2,15	54
СКД8-3,0-4000	80	3,0	40000	19076	2,10	52
СКД10-3,5-5600	100	3,5	56000	29832	1,88	47
СКД12-3,0-5600	120	3,0	56000	30847	1,82	45

To simulate the operation of the proposed beamless pumping unit was manufactured and tested it is working model. The model simulates the operation of a real oil pumping unit. This mini model of the pumping unit is intended for widespread use in various conditions. The model is made of plastic and metal on a scale of 1:10, 700 mm long and 850 mm high. Figure 7 shows a model of the proposed beamless pumping unit.

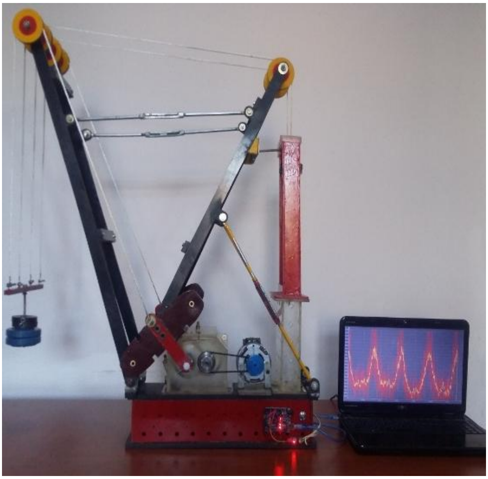
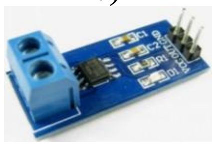


Figure 7. Working model of the proposed beamless pumping unit

a)



b)

Figure 8. Microcontroller Arduino UNO and ACS712 current sensor

At present, in the oil fields of most countries, the assessment of loading and balancing of pumping units is carried out, as a rule, with the maximum instantaneous values of active power during the upstroke of rod suspension, instantaneous values of the rotor rpm of the drive motor in one swing period, by using current clamps[8], which control the value of the acting current in the stator windings of an asynchronous electric motor.

To measure the amount of current in the stator windings of the asynchronous electric motor in the proposed model of beamless pumping unit used microcontroller Arduino UNO. To measure current with the Arduino UNO, we used the ACS712 sensor from Allegro Microsystems (measurement error no more than ± 1%, at temperatures from 25 to 150 ^{0}C). Pairing the ACS712 current sensor with the Arduino UNO helps to accurately measure the motor current. This

[8] Софьина Н.Н., Шишлянников Д.И., Корнилов К.А., Вагин Е.О. Способ контроля параметров работы и технического состояния штанговых скважинных насосных установок. Master`s journal № 1, 2016, с. 247-257.

sensor is based on the Hall effect and the measuring system has a built-in Hall effect device. The schematic diagram of connecting the ACS712 current sensor to the Arduino is shown in the figure 9.

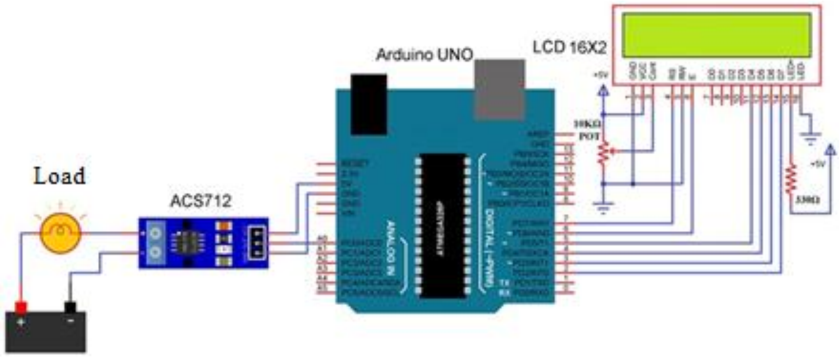


Figure 9. The schematic diagram of connecting the ACS712 current sensor to the Arduino

Regular recording of the current signals consumed by the electric motors of the mechanical drive of the beamless pumping unit allows obtaining relevant and reliable information about the operational parameters of the sucker rod pump, and based on the data obtained, it is most effective and reliable to control the loading of the sucker-rod pumping units, and to efficiently balance the sucker rod pumping unit.

The results of the experiments show that the development of control devices that monitor the technical condition of the pumping unit, which allows to estimate the value and nature of the current consumed by the electric motor of its converting mechanism, proves that it is important to assess the load capacity and balance of these pumping unit.

The fifth chapter deals with the determination of the elastic deformations of rods, tubes and rope in the static loading mode of innovative mechanical transmission, the assessment of the impact of total design and technological errors on its kinematic parameters and the reciprocating speed of the rod suspension point depending on well productivity.

Deformations occur in the rod, pipes, rope and other structural elements of the pumping unit during the up and down movement of the rod hanging from the rope of the pumping unit[9]. The rods and pipes lowered into the well are subject to constant deformation due to their own weight. However, due to the changing static and dynamic loads acting on the pumping unit, the deformation of the rod and the pipe changes during the work process.

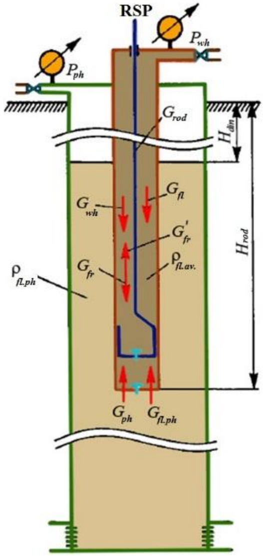


Figure 10. Diagram of forces acting on a rods column

In general, the force of gravity in the liquid of the rod column during the upward movement of the rod column (G_{rod}), due to the gravity of the fluid inside the pump-compressor pipes (G_{fl}), from the well-head pressure of the pump-compressor pipe column (G_{wh}) and friction of the rod column on the pump-compressor pipes (G_{fr}), forces due to the pressure in the area around the pipe (G_p) and the gravitational force of the fluid in the area around the pipe ($G_{fl.ph}$) (figure 10).

The direction of the first four forces coincides with the direction of the acceleration of release, and the direction of the last two forces is in the opposite direction.

When the rod is lifted upwards, the force due to the pressure difference acting on the suspension point causes the rod, tubes and

[9] Мищенко И.Т. Скважинная добыча нефти: Учебное пособие для вузов / И.Т.Мищенко. М.: ФГУП Изд-во «Нефть и газ» РГУ нефти и газа им. И.М.Губкина, 2003, 816 с.

rope to be subjected to tensile deformation. As a result of these deformations, the displacement of the plunger in the pump cylinder will begin when the plunger extends to the polished rod ($\Delta l_{rod} + \Delta l_b + \Delta l_k$) before moving upwards. Let's call this ***loss the of plunger travel*** and show it like:

$$\Delta l_{rod} + \Delta l_b + \Delta l_k = \lambda^* \tag{8}$$

then the actual travel of the plunger will be as follows:

$$S_{pl} = S_g - \lambda^* \tag{9}$$

there $\Delta l_{rod}, \Delta l_b, \Delta l_k$ – respectively are the elastic displacements of the rod, tubes and ropes.

The above-mentioned forces and the displacements caused by these forces affect the static operation of the device. In addition to these forces, the forces of inertia (G_{in}) and vibration (G_{vib}) also affect the dynamic operation of the device.

The force of inertia caused by the weight of the rod column and the liquid column affects the movement of the plug. Thus, during the upward movement of the rods column, the force of inertia is maximal at the upper dead center, and its direction of action is opposite to the force of gravity. As a result of this load, the rod column is compressed, which increases the travel of the plunger inside the cylinder. During the downward movement of the rod column, the direction of inertia coincides with the direction of gravity. Under the influence of this force, the rods column is stretched, which increases the travel of the plunger inside the cylinder. Then the total additional travel resulting from the elastic deformation of the rods column during one cycle of the plunger pump under the influence of inertia will be as follows:

$$\lambda_{in} = \Delta e_{rod}^{up} + \Delta e_{rd}^{dw} \tag{10}$$

Thus, in general, the true stroke of the plunger is

$$S_{pl} = S_g - \lambda^* + \lambda_{in} \tag{11}$$

there λ^* – elastic deformation of rods in static operation mode; λ_{in} - is the elastic deformation of the rods under the influence of inertial forces.

If we take into account the values of elastic deformations of rods under the influence of static and inertial forces in the expression (11), then:

$$S_{pl} = S_g \left[1 + \frac{n^2[cos\varphi + \lambda cos2\varphi - \varepsilon\lambda sin\varphi]H_{rod}}{450\left(2 + \frac{\varepsilon^2\lambda^2}{(1-\lambda^2)} + \frac{\varepsilon^4\lambda^4(\lambda^2+3)}{4(1-\lambda^2)^3}\right)EA_{rod}} \left(G_{şt}b' + \frac{G_{fl}}{2}\right) \right] - \lambda^*$$

If we perform the substitution in this formula

$$\delta^* = 1 + \frac{n^2[cos\varphi + \lambda cos2\varphi - \varepsilon\lambda sin\varphi]H_{rod}}{450\left(2 + \frac{\varepsilon^2\lambda^2}{(1-\lambda^2)} + \frac{\varepsilon^4\lambda^4(\lambda^2+3)}{4(1-\lambda^2)^3}\right)EA_{rod}} \left(G_{şt}b' + \frac{G_{fl}}{2}\right)$$

then we get the following expression to determine the real stroke of the plunger:

$$S_{pl} = S_g \cdot \delta^* - \lambda^* \qquad (12)$$

there δ^*- ***is the ods from real stroke of the plunger***.

To simplify the problems solved during the theoretical study of machines and mechanisms, kinematic pairs are considered as idealized mechanisms without gaps and errors. However, the experience of machine operation shows that in real mechanisms, the dimensions of all items differ from the dimensions adopted during design. Preparation, assembly, installation of these items, their wear, elastic and temperature deformations, etc. due to errors made during

It is known that the most important kinematic parameters of the converting mechanism of the pumping unit are the displacement, speed and acceleration of the rod suspension point. Therefore, the errors made during the preparation, assembly, installation, wear, elastic and temperature and other deformations of the elements of the converting mechanism of the pumping unit are taken as the difference between the positions of the point of the real and ideal converting mechanism in the same position of the crank.

Then, taking into account the effect of these errors, the actual displacement of the rod suspension point, its speed and acceleration are determined as follows:

$$S = r \cdot k_1 \left[1 - cos\varphi + \frac{\lambda}{4}\left(\frac{k_1}{k_2}\right)(1 - cos2\varphi) + \varepsilon\lambda\left(\frac{k_3}{k_2}\right) sin\varphi - \right.$$

$$-\frac{\varepsilon^2\lambda^2}{2(k_2+\lambda k_1)}k_1k_3^2+\frac{\lambda\varepsilon^2}{2}\frac{k_3^2}{k_1k_2} \quad (13)$$

$$V=r\cdot k_1\cdot\omega\left[sin\varphi+\frac{\lambda}{2}\left(\frac{k_1}{k_2}\right)sin2\varphi+\varepsilon\lambda\left(\frac{k_3}{k_2}\right)cos\varphi\right] \quad (14)$$

$$a=r\cdot k_1\cdot\omega^2\left[cos\varphi+\lambda\left(\frac{k_1}{k_2}\right)cos2\varphi-\varepsilon\lambda\left(\frac{k_3}{k_2}\right)sin\varphi\right] \quad (15)$$

there $k_1=1\pm\frac{\Delta_1}{r}$, $k_2=1\pm\frac{\Delta_2}{l}$ və $k_3=1\pm\frac{\Delta_3}{E}$ - are dimensionless quantities and coefficients that take into account the ratio of total structural and technological errors to the radius of the crank, the length of the rope and the length of the eccentric; Δ_1, Δ_2 and Δ_3- respectively, are the total design and technological errors of the pumping unit.

From the kinematic parameters, which characterize the sukre rod pumping unit main parameter, which is of practical importance, it is the acceleration of the rod suspension point. It was found that the dynamic forces in the rod column of oil wells depend on the value of the acceleration and the regularity of its change[10]. Therefore, in order to estimate the kinematic parameters that characterize the new constructive solution of beamless sucker rod pump, the deviation of the real maximum acceleration from the acceleration generated during the ideal harmonic motion is determined as follows:

$$\Omega=\frac{a_{Emax}}{a_{Ei}}=\frac{1}{cos\varphi}\left[cos\varphi+\lambda\left(\frac{k_1}{k_2}\right)cos2\varphi-\varepsilon\lambda\left(\frac{k_3}{k_2}\right)sin\varphi\right]$$

there a_{Ei}- maximum acceleration of the rod suspension point during ideal harmonic motion; Ω - is a kinematic parameter that takes into account the deviation of the real maximum acceleration from the maximum acceleration generated during the ideal harmonic motion.

To evaluate the effect of structural and technological errors on the kinematic parameter, this expression can be written as follows:

$$\Omega=\frac{1}{cos\varphi}\left[cos\varphi+\lambda\left(\frac{k_1}{k_2}\right)cos2\varphi-\varepsilon\lambda\left(\frac{k_3}{k_2}\right)sin\varphi\right]=$$
$$=1+\lambda\left(\frac{k_1}{k_2}\right)\frac{cos2\varphi}{cos\varphi}-\varepsilon\lambda\left(\frac{k_3}{k_2}\right)tg\varphi=1+\chi \quad (16)$$

[10] Молчанов А.Г. Машины и оборудование для добычи нефти и газа: Учебник для вузов. – М.: «Издательский дом Альянс», 2010, 588 с.

there $\chi = \lambda\left(\frac{k_1}{k_2}\right)\frac{cos2\varphi}{cos\varphi} - \varepsilon\lambda\left(\frac{k_3}{k_2}\right)tg\varphi$ - is the ***error coefficient*** that takes into account the effect of structural and technological errors on the kinematic parameter (Ω), taking into account the dimensional kinematic parameters (λ, ε).

As it is known, there is always a difference between the theoretical and actual productivity of an operating oil well[11]. This difference is directly related to the volume of fluid collected at the bottom of the well over a period of time. Experience of oil wells shows that the amount of fluid collected at the bottom of the well depends on the pressures in the reservoir and the bottom of the well, and the efficiency of sucker rod pumps depends on the diameter of the pump plunger, the actual flow point of the rod and the number of strokes.

When the relationship between the amount of fluid collected at the bottom of the well and the amount of fluid removed by the plunger pump is broken, the dynamic level of the fluid in the well and, accordingly, the actual productivity of the pumping unit changes. Decreased dynamic level occurs when the internal energy of the reservoirs is depleted during the long-term operation of the oil well, when the reservoir pressure drops and when the productivity of the plunger pump exceeds the productivity of the fluid collected at the bottom of the well.

A decrease in the dynamic level leads to an artificial decrease in the pressure drop and, consequently, the flow rate of the well. The proliferation of such processes necessitates the cessation of well operations and high-cost maintenance. Maintenance of the dynamic level within the set limits is carried out by automatically switching off the electric transmission of the pump when it falls below these limits, and by automatic start-up during the restoration of the previous set level. This results in idle downtime, a decrease in its daily productivity and economic efficiency of sucker rod pumping unit.

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[11] Мищенко И.Т. Скважинная добыча нефти: Учебное пособие для вузов / И.Т.Мищенко. М.: ФГУП Изд-во «Нефть и газ» РГУ нефти и газа им. И.М.Губкина, 2003, 816 с.

In order to prevent such cases, it is important to ensure the interaction between the two factors, ie the amount of fluid flowing to the bottom of the well and the amount of fluid discharged through the pumping unit. As it is known, one of the main factors influencing the amount of liquid extracted by the sucker rod pumping unit is the reciprocating speed per minute of the rod suspension point[12]. Therefore, the formation of a mathematical relationship between the reciprocating speed and the amount of fluid at the bottom of the well is of great practical importance.

Taking into account the influence of these factors, the optimal reciprocating speed of rod suspension points is determined as follows from the geological reserve (debit) in the well, the actual volume capacity of the sucker rod pump and the dynamic level in the well remains constant:

$$n = 480 \frac{kh_l}{D_{pl}^2 S_{pl(act)} \mu} \cdot \frac{\left(H_w - (H_{din} + H_{stat})\right)\rho_{fl} g}{\left(ln \frac{R_c}{r_c}\right)} \tag{17}$$

there k- effective conductivity of porous media, m^2; h_l- l effective layer thickness, m; R_c- is the radius of the feed contour of the well, m; r_c - the radius of a cylindrical well. m; H_w- depth of the well, m; H_{din}- the height of the dynamic level, m; H_{stat}- static level height, m; ρ_{fl}- density of fluid in the well, kg/m^3; g- release urgency, m/sec^2; D_{pl}- plunger diameter, m; μ - dynamic viscosity coefficient of fluid (fluid) in layers, $Pa \cdot sec$; $S_{pl(act)}$- is the actual stroke of the rod suspension point, m.

The proposed expression can provide an important basis for ensuring the relationship between the amount of fluid flowing to the bottom of the well and the amount of fluid discharged through the pumping unit, maintaining the dynamic level of the fluid column in the well, eliminating idle time on the pumping unit, increasing its daily productivity and economic efficiency.

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[12] Кудинов В. И.: Основы нефтегазопромыслового дела. «Институт компьютерных исследований», Москва, 2004, 720 с.

The sixth chapter is devoted to the assessment of the quality of the transmission mechanism of the new constructive solution of pumping unit, i.e. the study of the efficiency of double sliding bearings of the new three-stage two-line reducer[13] used in the transmission of pumping unit and checking the accuracy of the obtained results is devoted to the improvement of the test device, test planning and the detailed calculation of reducer of the mechanical transmission.

According to design and economic considerations, the "double sliding friction pads" play the role of support for the rotating gear blocks in the new design reducer, not the rolling friction pads. These pads are mounted on the rotating shaft and under the block gears, which rotate freely around its axis.

Based on the analysis of possible kinematic schemes of reducers of this design, it was determined that if the number of gears of the reducer is one, the directions of rotation of the drive and driven shafts coincide with the directions of rotation of the package gears mounted on sliding friction pads. In this case, the resistance in the double sliding friction pads becomes a beneficial driving force, which becomes a useful factor that characterizes the internal driving force of the oil. This not only increases the reliability of the mechanism, but also significantly reduces the energy requirements for the transmission of new equipment.

This ratio is defined as the ratio of the force exerted on the input shaft to the displacement of the oil layers in the interstitial spaces of the sliding bearings, the partial contact of the contact surfaces and the repulsive forces associated with the viscosity of the oil:

$$\psi_{sb}^{*} = \frac{P_f}{P_1} = \frac{\pi\mu_0\omega_1^2}{2P_1}\left(\frac{l_1 d_1^2}{\psi_1}\left(\frac{(1+u_1^2)}{u_1^3}\right)^2 + \frac{l_2 d_2^2}{\psi_2}\left(\frac{(1+u_2^2)}{u_2^2}\right)^2\right) \quad (18)$$

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[13] Наджафов А.М. Поисковое конструирование механическкго привода штанговых насосов. Безбалансирный станок-качалка с зубчаторычажным преобразующим механизмом и пакетным редуктором на двух валах. Palmarium Academic Publishing, Sarbrücken, Germany, 2013, 135 с.

there μ_0- apparent viscosity; ω_1- angular velocity of the drive shaft of the reducer; u_1 and u_2- transmission numbers of the first and second stages of the reducer, respectivelyi; ψ_1 and ψ_2 – respectively, the relative distances between the shaft and the filling in the first and second stages; d_1 , d_2, l_1 and l_2- respectively, are the diameters and lengths of the pads on the drive and carrying shafts of the reducer.

If the power in the drive shaft of the new multi-stage reducer is P_1, and the power in the outlet shaft without the sliding bearings is P_2, then the power in the outlet shaft of the reducer will be as follows, taking into account the positive effect of the internal resistance of the oil in the double sliding bearings:

$$P_2^* = P_2 - \psi_{sb}^* P_2 = P_2(1 - \psi_{sb}^*)$$

If we take into account the friction loss when the gears are engaged (ψ_{ge}) and the friction loss in the roller bearings (ψ_{rb}), then the efficiency factor of the current mechanical system:

$$\eta^* = \frac{P_2^*}{P_1} = \frac{P_2(1 - \psi_{sb}^*)}{P_1} \quad \Longrightarrow \frac{\eta^*}{(1 - \psi_{sb}^*)} = \frac{P_2}{P_1} = 1 - \psi_{ge} - \psi_{rb}$$

from there

$$\eta^* = (1 - \psi_{ge} - \psi_{rb})(1 - \psi_{sb}^*)$$

This expression allows us to estimate the performance of the new three-stage reducer, as well as the energy loss in the transmission.

It is known that

$$1 - \psi_{ge} - \psi_{rb} = \eta_{ge} \cdot \eta_{rb}$$

$$(1 - \psi_{sb}^*) = \eta_{sb}$$

Then the efficiency factor of the new three-stage reducer:

$$\eta^* = \eta_{ge}^3 \eta_{rb}^2 \eta_{sb} \tag{19}$$

To quantify the internal resistance of the oil in the sliding bearings, we assume that the new constructive solution of reducer uses industrial oil И-30, and for this oil the dynamic viscosity at 60^0C is $\mu_0 = 0{,}014\, Pa \cdot sec$. The power of the electric motor of the converting mechanism of the pumping unit is $P_M = 7{,}5kVt$, the rotational speed of its shaft is $n_M = 750\, min^{-1}$, so the angular velocity of the drive shaft of the reducer:

$$\omega_1 = \frac{\omega_M}{u_q} = \frac{\pi \cdot n_M}{30 \cdot u_q} = \frac{3{,}14 \cdot 750}{30 \cdot 1{,}6} = 49{,}06\ sec^{-1}$$

there $u_b = 1{,}6$ - is the transmission number of the belt drive applied in the converting mechanism of the pumping unit.

Assuming that the efficiency factor of the belt drive is $\eta_b = 0{,}96$, then the force on the gear shaft of the reducer:

$$P_1 = P_M \cdot \eta_b = 7{,}5 \cdot 0{,}96 = 7{,}2\ kVt$$

The new constructor assumes the diameter of the gear shaft under the bearing $d_1 = 50mm$, its length $l_1 = 50mm$, relative distance $\psi_1 = 0{,}002$, diameter of the drive shaft under the bearing $d_2 = 50mm$, its length $l_2 = 100mm$ and relative distance $\psi_2 = 0{,}003$. The coefficient of the total internal resistance of the oil in the sliding bearings is as follows:

$$\psi_{sb}^* = \frac{P_f}{P_1} = \frac{\pi \mu_0 \omega_1^2}{2P_1}\left(\frac{l_1 d_1^2}{\psi_1}\left(\frac{(1+u_1^2)}{u_1^3}\right)^2 + \frac{l_2 d_2^2}{\psi_2}\left(\frac{(1+u_2^2)}{u_2^2}\right)^2\right) =$$

$$= \frac{3{,}14 \cdot 0{,}014 \cdot 49{,}06^2}{2 \cdot 7{,}2 \cdot 10^3}\left(\frac{0{,}05 \cdot 0{,}05^2}{0{,}002}\left(\frac{(1+4^2)}{4^3}\right)^2 + \frac{0{,}1 \cdot 0{,}05^2}{0{,}003}\left(\frac{(1+4^2)}{4^2}\right)^2\right) =$$

$$= 0{,}007348(0{,}00441 + 0{,}0941) = 0{,}000724$$

If we take into account efficiency factor of gear drive $\eta_{ge} = 0{,}98$, efficiency factor of a pair of rolling bearing $\eta_{rb} = 0{,}99$, then the efficiency factor of new three-stage reducer:

$$\eta^* = \eta_{ge}^3 \eta_{rb}^2 (1 - \psi_{sb}^*) = 0{,}98^3 \cdot 0{,}99^2 (1 - 0{,}000724) = 0{,}9218$$

efficiency factor of reducer with classic construction:

$$\eta = \eta_{ge}^3 \eta_{rb}^4 = 0{,}98^3 \cdot 0{,}99^4 = 0{,}9041$$

Comparison of the efficiency of the new three-stage reducer with the efficiency of the classic three-stage reducer as a percentage:

$$\Delta\eta = \frac{\eta^* - \eta}{\eta} 100\% = \frac{0{,}9218 - 0{,}9041}{0{,}9041} 100\% = 1{,}96\ \%$$

As can be seen, the new design has a positive effect on the movement of the shafts and, consequently, its efficiency, because the direction of rotation of the shafts is in the same direction as the direction of rotation of the sliding friction bearing. According to the results of the calculations, the efficiency of the new three-speed gearboxes is about 2% higher than that of conventional gearboxes.

Gear transmissions refer to clutch transmissions made by direct contact of a pair of gears. When transmitting torque, the normal force (F_n) on the coupling and the frictional force (F_s) due to sliding are affected. Under the influence of these forces, the tooth becomes in a complex state of tension. The main factors that affect the performance of a gear are the surface stress (σ_H) and the bending stress (σ_F), which vary with time. Variable stresses cause the teeth to fall out tirelessly[14]. One of the types of collapse as a result of fatigue is the breaking of teeth under the influence of bending stresses and the wear of working surfaces under the influence of surface stress. The strength calculation of closed gear transmissions is performed according to the maximum surface stress (σ_{max}) created by the coupling of two teeth meeting at the pole point[15]. This stress is used as an important performance criterion for gear transmission.

In the current calculation method of gear transmission, the value of the maximum contact stress is determined according to the Hers formula[16;17]. In this calculation, the solution of the contact problem is based on the theory of elasticity. The solution of the problem is based on some limited assumptions, for example, it is assumed that the contact area is very small, the coefficient of friction between the meeting surfaces is zero.

As mentioned, normal force (F_n) and frictional force (F_s) as a result of sliding are affected when transmitting torque[18]. Friction is caused

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[14] Decker K.H. Maschinenelemente. Carl Hanser Verlag. Munchen,Wien, 2000, 706p.

[15] Иванов М. Н. Детали машин: учеб. для вузов / М. Н. Иванов, В.А.Финогенов. М.: Высшая школа, 2008, 408 с.

[16] DIN 3990: Grundlagen für die Tragfähigkeitsberechnung von Gerad-und Schrägstirnrädern. Beuth-Verlag, 1990. 20 p.

[17] Damir Jelaska. Gears and gear drives.Croatia. John Wiley & Sons Ltd, 2012, 444p.

[18] Матлин М. М., Мозгунова А. И., Лебский С. Л., Шандыбина И. М. Основы расчета деталей и узлов транспортных машин. Волгоград, 2010, 251 с.

by the teeth of the gears sliding on top of each other. Although the sliding friction is zero at the pole point, it has a maximum value at the base and top of the tooth. Therefore, the friction force causes a loss of force in the grip and erosion of the teeth.

For this reason, the calculation of the strength of the gears for the contact stress is more complete, taking into account only the friction. Therefore, the current calculation assumes that the coefficient of friction between the meeting surfaces is not equal to zero, it has a certain value, and the area of contact and the shape of the boundaries have a certain value[19].

To determine the contact stress by taking into account the friction in the field of contact, the following expression was used to determine the contact stress between arbitrarily shaped objects from the problem of the effect of a rigid stamp on the elastic plane, imagining the gears as two elastic bodies close to half-plane:

$$\sigma_{Hmax}^{*}(t_0) = -\frac{4\mu^* sin(\pi\alpha^*) cos(\pi\alpha^*)}{\chi^* + 1} f'(t_0) + \frac{P_0 cos(\pi\alpha^*)}{\pi(t_0 - a)^{\frac{1}{2}+\alpha^*}(b - t_0)^{\frac{1}{2}-\alpha^*}} +$$

$$+\frac{4\mu^* cos^2(\pi\alpha^*)}{\pi(\chi^* + 1)(t_0 - a)^{\frac{1}{2}+\alpha^*}(b - t_0)^{\frac{1}{2}-\alpha^*}} \int_a^b \frac{(t - a)^{\frac{1}{2}+\alpha^*}(b - t)^{\frac{1}{2}-\alpha^*}}{t - t_0} f'(t) dt$$

In this case, from the specified solution of the contact problem, the condition of strength in the field of contact of two objects is as follows:

$$\sigma_{Hmax}^{*}(t_0) = \left[\sqrt{2}\left(\frac{1}{4} + \frac{{\alpha^*}^2}{8} + \frac{2\alpha^*}{3\pi} + \frac{4\left(\alpha^* + 2{\alpha^*}^3\right)}{45\pi}\right) cos^2(\pi\alpha^*) + \right.$$

$$\left. + \frac{1}{\sqrt{2}} cos(\pi\alpha^*)\right] \sigma_H \leq [\sigma]_H$$

there σ_H – Hers is the formula.

[19] Мусхелишвили Н.И. Некоторые основные задачи математической теории упругости. М., Наука, 1966, 708 с.

It should be noted that for $\alpha^* = 0$ ($f = 0$), or $t_0 = 0$, the exact solution of the problem exceeds the known expression of Hers by 6%, i.e.:

$$\sigma^*_{Hmax}(t_0) = \frac{3\sqrt{2}}{4}\sigma_H \approx 1{,}06\sigma_H \tag{22}$$

It is known that ГОСТ 21354-87 includes a factor for calculating the contact stress of the gear transmission[20]. This ratio takes into account the effect of the lubricant. This ratio takes into account the effect of the lubricant. Since the effect of oil on the bearing capacity of the gear is not sufficiently studied, this ratio is assumed to be equal to the unit in the existing calculations. That is, the ideal case is accepted without taking into account the effect of lubrication and friction. As is well known, one of the main tasks of lubrication is to reduce friction between contact surfaces. Therefore, the coefficient taking into account the effect of oil should also be closely related to the coefficient of friction and should be taken as a criterion for the coefficient taking into account the effect of oil.

To quantify the effect of the lubricating material on the gear capacity, the value of the maximum contact stress is compared with the allowable contact stress. Analysis of the calculations shows that $\alpha^* \approx f$. Considering this case and dividing $cos(\pi\alpha^*)$ by the Taylor series, we obtain the following analytical expression for the coefficient taking into account the effect of oil:

$$Z_L = 12{,}575(f - 0{,}73264)(f - 0{,}46778)(0{,}24607 + 0{,}94055 + f^2) \tag{23}$$

As can be seen from the last expression, the coefficient that takes into account the effect of oil is closely related to the coefficient of friction. Therefore, МС-20, МК-22, Тп-46, Тп-30, Тп-22, И-40А, И-30А and И-20А oils were used in a high-sensitivity two-wheel drive device that mimics the operation of a gear transmission. Based on the results of experimental studies, the analytical dependence of some

[20] ГОСТ 21354-87 (СТ СЭВ 5744-86). Передачи зубчатые цилиндрические эвольвентные внешнего зацепления М.: Изд-во стандартов, 1987, 110 с.

control of the coefficient of friction between the meeting surfaces on the factors was determined and the following expression was proposed for the coefficient taking into account the effect of oil, taking into account in (23):

$$Z_L = 1{,}0605 + 0{,}177 \cdot 10^4 \frac{(1-0{,}065V_\Sigma)R_a}{\sigma_H^{0{,}223}\nu^{0{,}3}V_s^{0{,}334}\rho_g}\left(1-10{,}45\cdot 10^2 \frac{(1-0{,}065V_\Sigma)R_a}{\sigma_H^{0{,}223}\nu^{0{,}3}V_s^{0{,}334}\rho_g}\right) \quad (24)$$

As mentioned, the design calculation of closed gear transmission is currently performed according to the allowable contact stress[21]. According to the current standard, the maximum contact stress between the teeth of the gear transmission is determined according to the Hers formula. Therefore, taking into account the effect of oil, this expression will be as follows:

$$\sigma_H = Z_H Z_M Z_\varepsilon \sqrt{\frac{T_2 K_{H\alpha} K_{H\beta} K_{HV}(u \pm 1)^3}{2a_W^3 u^2 \psi_{ba} Z_L^2}} \le [\sigma_H] = \frac{\sigma_{Hlimb} Z_N}{S_H} Z_R Z_v Z_X Z_w$$

As mentioned, the design calculation of the closed gear transmission is performed according to the allowable contact stress. Studies have shown that normal contact stress has a partial effect on the types of failure of gears.

From this point of view, the contact stress generated in the active contact zone of the teeth should be taken as the stress that causes damage to the gear transmissions. As the contact zone moves away from the pole of the tooth, the contact voltage (τ_{max}) increases, intensifying the decay process. As a result, the voltage touching the largest surface $(\tau_{max,n})$ can increase more intensively and exceed the maximum stress (τ_{max}). The direction of interaction of these tensions also creates real conditions for the formation of fatigue cracks. As noted, the value of these stresses depends on the friction force (F_s) caused by the normal force (F_n) acting on the coupling, which in turn depends directly on the coefficient of friction (f).

[21] Воробьев Ю.Б., Ковергин А.Д. Проектирование зубчатых передач на долговечность с учетом трения. Вестник ТГТУ, Том 10, №1Б, 2004, с.205-210.

Studies have shown that there is the following relationship between the shearing and the contact stress:

$$\tau_{max} = 0{,}347\sigma_H = 0{,}347 \cdot Z_H Z_M Z_\varepsilon \sqrt{\frac{T_2 K_{H\alpha} K_{H\beta} K_{HV} (u \pm 1)^3}{2 a_W^3 u^2 \psi_{ba} Z_L^2}} \leq [\tau_s] \quad (25)$$

In a complex stress situation, the applied (equivalent) stress is used as the calculated stress. In this case, the theory of strength, which corresponds to the stress state of the material under consideration, is used. For plastic materials:

$$\sigma_{ek} = \sqrt{\sigma_H^2 + 4\tau_{max}^2} = 1{,}217 \cdot Z_H Z_M Z_\varepsilon \sqrt{\frac{T_2 K_{H\alpha} K_{H\beta} K_{HV} (u \pm 1)^3}{2 a_W^3 u^2 \psi_{ba} Z_L^2}} \leq [\sigma_H]$$

In design calculations, the distance between the centers of the transmission or the diameter of the gear circumference is usually determined. Therefore, the following expression is proposed to determine the distance between the centers, taking into account the combined effect of shearing and contact stress:

$$a_w = K_a (u \pm 1) \sqrt[3]{\frac{T_2 \cdot K_{H\beta}}{[\sigma_H]^2 \cdot u^2 \cdot \psi_{ba} \cdot Z_L^2}} \quad (26)$$

The results of the study show that the effect of the coefficient of friction on the coefficient taking into account the effect of oil and the associated effect of gear transmission on the load capacity is significant. The proposed method allows to solve the problem in the opposite way, i.e. to know the kinematic and energy parameters of the gear transmission and to choose a more effective oil depending on the coefficient that takes into account the effect of oil.

The seventh chapter is devoted to the study of the dynamic forces arising in the transforming mechanism of innovative mechanical transmission.

Statistical studies show that 80% of the failures of machine tools are caused by intensive wear of surfaces under the influence of dynamic forces, fatigue collapse of details of mechanisms and load-bearing meralconstructions, formation of unbearable residual deformations, etc. occurs due to. This proves that it is impossible to create high-tech pliers without dynamic calculations.

Due to the specific gravity of the load acting on the rod suspension point and the metal construction of the sucker rod pumping unit, the presence of large loads affecting the mechanisms and the small movement speeds of the mechanisms, large mass of moving parts, repetitive-short-term operation, elastic suspension of the rod suspension from the rope and impact loading conditions are characteristic of pumping unit. These and other factors increase the role of dynamic calculations when designing pumping unit.

The dynamic calculation, taking into account the elasticity of the elements of the transforming mechanism of the pumping unit, significantly increases the accuracy of determining the actual loads. As the simplest dynamic calculation scheme of the transforming mechanism of the sucker rod pumping unit, a two-mass scheme connected to each other by an elastic element was adopted. With the help of such a simple scheme, the dynamic loads on the elastic elements of the transforming mechanism of the sucker rod pumping unit in some operating modes were determined.

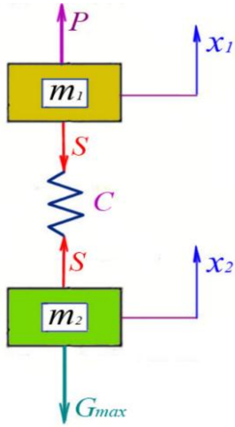

Figure 11. Calculation scheme of a two-mass system

For this purpose, as a calculation scheme, a system is considered in which two units of mass m_1 and m_2 are connected by an elastic element with a stiffness c (Fig. 11). Where m_1 - is the total mass of all rotating parts of the transforming mechanism; m_2- mass of load acting on the rod suspension point; P- driving force; G_{max}-maximum load acting on the rod suspension point; c - is the total hardness.

In the initial state, as the masses stand motionless, the tensile force on the elastic element is $S = G_{max}$.

Therefore, after calculating the equation of motion for each mass, the tensile force on the elastic element is determined as follows:

$$S = S_{st} + S_d = G_{max} + \frac{(P - G_{max})m_2}{m_1 + m_2}(1 - cos\lambda_k t) \qquad (27)$$

there S_{st}- static force ($S_{st} = G_{max}$); λ_k- circular frequency of special dances; S_d- is a dynamic force:

$$S_d = \frac{(P - G_{max})m_2}{m_1 + m_2}(1 - cos\lambda_k t)$$

Since $P > G_{max}$ is in working mode, the maximum force in the elastic contact corresponds to $cos\lambda_k t = -1$

$$S_{max} = G_{max} + \frac{2 \cdot (P - G_{max})m_2}{m_1 + m_2}$$

The minimum force in the elastic contact is instantaneous when $t = 0$ ($S_{min} = G_{max}$).

One of the main parameters of the sucker rod pumping unit is the load acting on the rod suspension point. This parameter determines the correct selection of the pump set depending on the operating conditions of the sucker rod pumping unit.

The load acting on the rod suspension point varies as it moves up or down and depends on many factors. As can be seen, both the static and dynamic forces affect the rod suspension point. Static forces are caused by the weight of the liquid column inside the rod column and the pump-compressor pipes and the force due to the friction of the rod column on the pump-compressor pipes, and the dynamic forces due to the force of inertia due to the weight of the rod column and the liquid column inside the pump-compressor pipes and the force due to the vibration of the rod column. The value of these forces is determined by the ratio of the elements of the pliers and the number of oscillations of the suspension point of the rod. The friction force is 2-5% of the static force. The value of these forces is determined by the ratio of the elements of the sucker rod pumping unit and the pump rate of the rod suspension point. The friction force is 2-5% of the static force.

Currently, there is no universal methodology for calculating extreme forces (maximum G_{max} and minimum G_{min}). Because it is impossible to take into account the effect of all these forces acting on the rods column. However, the results of tests performed in real wells and statistical processing of production data use the following expression as a more accurate expression for wells with normal operating mode[22]:

$$G_{up} = G_{rod} + G_{fl} + G_{in.up} + G_{vib.up} \quad (28)$$

$$G_{dw} = G_{rod} - (G_{in.dw} + G_{vib.dw}) \quad (29)$$

One of the most unfavorable factors for the normal operation of the pliers is the constant downward force of the rod suspension point, and its value varies by 30 ... 50% during the up and down movement of the rod suspension point, which leads to uneven loading of the electric motor.

As mentioned, the pumping unit are balanced to compensate for the difference in load during the up and down movement of the rod suspension point of the sucker rod pumping unit.

The purpose of balancing is to achieve equal loading of both the reducer and the motor during the up and down movement of the rod suspension point.

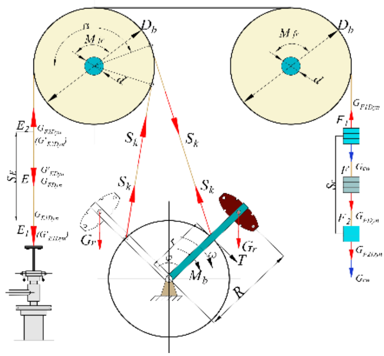

Figure 12. Scheme of determination of inertial forces on a new constructive solution of pumping unit

In order to determine the forces of inertia on a new constructive solution of beamless pumping unit, the scheme of action of these forces is taken at point E, i.e. at the point where the rods column is connected to the transverse beam of the pumping unit (Figure 12). Let G_{Ein} denote the force of inertia at point E during the upward movement of the rod suspension point, and G'_{Ein} during the downward movement.

$$G_{Ein} = ma_E = \frac{G_{rod}+G_{fl}}{g} a_E = \frac{a_E}{g}\left(G_{rod} + G_{fl}\right) \quad (30)$$

[22] Мищенко И.Т. Скважинная добыча нефти: Учебное пособие для вузов / И.Т.Мищенко. М.: ФГУП Изд-во «Нефть и газ» РГУ нефти и газа им. И.М.Губкина, 2003, 816 с.

$$G'_{Ein} = m'a_E = \frac{a_E}{g} G_{rod} \tag{31}$$

burada a_E-acceleration of the rod suspension point; g- acceleration of gravity.

Then the inertial force generated during the upward movement of the rod suspension point in accordance with the value of the acceleration of the rod suspension point for the new constructive solution of pumping unit is as follows:

$$G_{in} = \frac{a_E}{g}\left(G_{rod} + G_{fl}\right) = \frac{r \cdot \omega^2 [cos\varphi + \lambda cos2\varphi - \varepsilon\lambda sin\varphi]}{g}\left(G_{rod} + G_{fl}\right)$$

The force of inertia created during the downward movement of the rod suspension point ($G_{fl} = 0$):

$$G_{in} = \frac{a_E}{g} G_{rod} = \frac{r \cdot \omega^2 [cos\varphi + \lambda cos2\varphi - \varepsilon\lambda sin\varphi]}{g} G_{rod}$$

As a result of calculations for the new constructive solution of pumping unit, it was determined that the maximum displacement of the rod suspension point at different values of the coefficient, taking into account the ratio of relative eccentricity (ε=0,5; ε=1,0; ε=1,5 və ε=2,0) and crank length to rope length (λ=0,1; λ=0,25 və λ=0,4), varies in the range $S_g = (2{,}05 \dots 3{,}31)r$. The maximum acceleration of the rod suspension point varies in the range $a_{max} = (1{,}41 \dots 1{,}52)r\omega^2$ according to the maximum value of the relative length coefficient (for pumping units for СКД2-0,6-250, СКД4-2,1-1400, СКД6-2,5-2800, СКД8-3-4000, СКД10-3,5-5600 and СКД12-3-5600 relative length coefficient λ = 0.4 is accepted). Therefore, assuming $S_g \approx 2{,}5r$ and $a_{max} = 1{,}47r\omega^2$, the inertia force of the rod suspension point is determined as follows:

during upward movement

$$G_{in} = \frac{S_g n^2}{1530}\left(G_{rod} + G_{fl}\right) \tag{32}$$

during downward movement

$$G_{in} = \frac{S_g n^2}{1530} G_{rod} \tag{33}$$

there $\frac{a_E}{g} = \psi_d = \frac{S_g n^2}{1530}$ – dynamic factor.

As can be seen from (32) and (33), both the dynamic factor and the force of inertia increase in proportion to the square of the rod's suspension point and the number of its oscillations.

As noted, at the new constructive solution of beamless sucker rod pumping unit used combined metod of balancing. That is, balancing loads are placed both on moving and rotary counterweights.

When rotary counterweights are placed on the crank, the inertial forces of these loads are absorbed only by the bearings of the reducer output shaft, and these forces are not transmitted to other parts of the rocking machine at a constant rotation speed of the crank. Therefore, in determining the inertia forces in this case, only the forces of inertia created by movable counterweight were considered.

As can be seen from Figure 12, during the up and down movement of the rod suspension point, the point F is affected by a static force equal to the weight of the movable counterweight (G_{cw}) and directed in the vertical direction. Therefore, the mass acting on point F during up and down motion will be as follows. As shown in the figure, the maximum acceleration of point F will be equal to the acceleration of the rod suspension point. Given that $S_E = S_{cw}$, then the inertia force at point F during the up and down movement of the rod suspension point will be as follows:

$$G_{Fin} = ma_F = \frac{G_{cw}}{g} a_F = \frac{S_g n^2}{1530} G_{cw} \tag{34}$$

$$G'_{Fin} = m' a_F = \frac{G_{cw}}{g} a_F = \frac{S_g n^2}{1530} G_{cw} \tag{35}$$

So, $G_{Fin} = G'_{Fin}$

During the up and down movement of the rods column, the inertia force at point F varies depending on the rotation angle of the crank (φ). The characteristic feature of the change of inertia during the period of up and down movement of the rod suspension point is the decrease of G_{Fin} from the maximum value to zero and then its transition to the negative value range.

Although the value of the crank is $\varphi_{max} = 0^0$ we get $cos\varphi = 1$, and the inertia force is directed in the opposite direction of motion, i.e.

from the bottom to the top, and it has a negative sign during the upward movement:

$$G'_{Fin} = -\frac{S_g n^2}{1530} G_{cw}$$

The angle of rotation of the crank (at point F_2) at the beginning of the downward movement is $\varphi_{max} = 122^0 56'$ (for $\varepsilon = 0{,}5$), $\varphi_{max} = 119^0 04'$ (for $\varepsilon = 1{,}0$), $\varphi_{max} = 116^0 08'$ (for $\varepsilon = 1{,}5$), $\varphi_{max} = 113^0 50'$ (for $\varepsilon = 2{,}0$) the forces of inertia are negative. Then the inertia force at the beginning of the downward movement of the rod suspension point will be as follows:

$$G_{F2in} = -\frac{S_g n^2}{1530} G_{cw}$$

Then for the downward movement:

$$G_{Fin} = -\frac{S_g n^2}{1530} G_{cw}$$

Then the maximum force at points E and F will be the sum of the constituents of the static and inertial forces.

If we consider all the values of static and inertial forces at point E, then:

$$G_{Emax} = G_{Est} + G_{Ein} = \left(G_{rod} + G_{fl}\right)\left(1 + \frac{S_g n^2}{1530}\right) \tag{36}$$

$$G'_E = G'_{Est} + G'_{Ein} = G_{rod}\left(1 + \frac{S_g n^2}{1530}\right) \tag{37}$$

there $\left(1 + \frac{S_g n^2}{1530}\right) = k_d$ - dynamic factor.

Similarly, at point F:

$$G_{Fst} = G'_{Fst} = G_{cw}; \quad G_{Fin} = G'_{Fin} = -\frac{S_g n^2}{1530} G_{cw} \tag{38}$$

then

$$G_F = G_{Fst} + G_{Fin} = G_{cw} + \left(-\frac{S_g n^2}{1530} G_{cw}\right) = G_{cw}\left(1 - \frac{S_g n^2}{1530}\right) \tag{39}$$

The analytical expression for determining the dynamic load in addition to the effect of the instantaneous vibratory load applied to the lower end of the liquid column above the plunger during the upward movement of the rod column is defined as follows[23]:

$$G_{vib} = A_{rod}E_m \frac{v}{v_{so}}(1 + 0{,}3k)\psi_{pr} \tag{40}$$

there $A_{şt}$- the cross-sectional area of the rod; v_{so}- the sound speed in the rods column; E_m- modulus of elasticity of the material of the rods column; v- the speed of the rod suspension point at the moment of deformation during the upward movement; ψ_{pr}- is a factor that takes into account the ratio of the cross-sectional areas of the pump pipes and the rod:

$$\psi_{pr} = \frac{A_p}{A_p + A_{rod}} = \frac{\psi_p^2}{\psi_p^2 + 1}$$

$$k = \left(\frac{A_{pl} - A_{rod}}{A_p - A_{rod}}\right)\frac{G_{fl}}{G_{rod}} = \left(\frac{\psi_{pl}^2 - 1}{\psi_p^2 - 1}\right)\frac{G_{fl}}{G_{rod}}$$

there A_{pl}- the cross-sectional area of the plunger; $\psi_{pl} = \frac{D_{pl}}{D_{rod}}$; $\psi_b = \frac{D_p}{D_{rod}}$ - are dimensionless parameters that take into account the ratio of the diameter of the plunger and the pipe to the diameter of the rod.

As mentioned, as a result of calculations for the new constructive solution of beamless sucker rod pumping unit, it was determined that the maximum displacement of the rod suspension point at different values of the coefficient, taking into account the ratio of relative eccentricity (dexaxial) and crank length to rope length varies within $S_g = (2{,}05 \ldots 3{,}31)r$. The maximum value of the rod suspension point varies in the range $v_{max} = (1{,}05 \ldots 1{,}48)r\omega$ according to the maximum value of the relative length coefficient. Therefore, if we accept $S_g \approx 2{,}5r$ and $v_{max} = 1{,}27r\omega$ and make some simplifications, then to determine the vibration force we get the following expression:

[23] Вирновский А.С. Теория и практика глубиннонасосной добычи нефти: Избранные труды / А.С.Вирновский // Тр.ВНИИ-Вып.LVII. М.: Недра, 1971, 184 с

$$G_{vib} = \frac{S_g n}{1440}\left[1 + 0{,}3k' \frac{G_{fl}}{G_{rod}}\right]\psi_{pr}\frac{E_m}{v_{so}}D_{rod}^2 \tag{41}$$

Then the maximum force acting on the rod suspension point during the upward movement of the rod suspension point will be as follows:

$$G_{max} = \left(G_{rod} + G_{fl}\right)\left(1 + \frac{S_g n^2}{1530}\right) + \frac{S_g n}{1440}\left[1 + 0{,}3k' \frac{G_{fl}}{G_{rod}}\right]\psi_{pr}\frac{E_m}{v_{so}}D_{rod}^2 \tag{42}$$

The maximum force acting on the rod suspension point during the downward movement of the rod suspension point:

$$G_{min} = G_{rod}\left(1 - \frac{S_g n^2}{1530}\right) - \frac{S_g n}{1440}\left[1 + 0{,}3k' \frac{G_{fl}}{G_{rod}}\right]\psi_{pr}\frac{E_m}{v_{so}}D_{rod}^2 \tag{43}$$

The dynamic similarity parameter, ie the Cauchy parameter- $\varphi(\mu)$ is accepted as a criterion for estimating the operating mode of the pumping unit:

$$\varphi = \frac{\omega H}{v_{so}} \tag{44}$$

There ω - angular velocity of rotation of the crank, sec^{-1}; H- rod column length, m; v_{so}- the sound speed in the rods column, m/sec.

The operating modes of the pumping unit, i.e. its operation in static or dynamic mode, can be determined using the Cauchy parameter. It is known that for such a separation it is necessary to accept a certain (limit) value of the Cauchy parameter. For this we use the expression of the dynamic factor.

It is known that an unreasonable increase in the reciprocating speed of the rod suspension point can lead to an inertia acceleration greater than the acceleration of release (g). Therefore, when designing a pumping unit, the oil extraction rate is considered a critical velocity if the acceleration ratio is equal to the one.

If

$$\frac{a_{max}}{g} = \frac{S_g n^2}{1530} = 1$$

so, the critical number of oscillations of the suspension point of the rod:

$$n_{co} = \sqrt{\frac{1530}{S_g}} = 39{,}12\sqrt{\frac{1}{S_g}} \tag{45}$$

If we assume that the actual oil extraction rate is equal to 50-75% of the critical speed, then the maximum reciprocating speed of the rod suspension point will be as follows

$$n_{max} = 19{,}56 \ldots 29{,}34\sqrt{\frac{1}{S_g}} \tag{46}$$

Since $n^2 = \frac{(19{,}56^2 \ldots 29{,}34^2)}{S_g}$ and $S_g n^2 = (19{,}56^2 \ldots 29{,}34^2)$ the maximum dynamics factor:

$$\psi_d = \frac{S_g n^2}{1530} = \frac{(19{,}56^2 \ldots 29{,}34^2)}{1530} = 0{,}25 \ldots 0{,}56 \tag{47}$$

If $\psi_d < 0{,}4$, then the operating mode of the sucker rod pumping unit is static, if $\psi_d > 0{,}4$, then the operating mode of the sucker rod pumping unit is dynamic. The results of the calculations show that the limit value of the Cauchy parameter should be taken to be approximately $\psi_d = 0{,}4$ for the most suitable operating conditions on the pumping unit during the operation of the wells.

The eighth chapter is devoted to the evaluation of the economic efficiency of new constructive solution of beamless sucker rod pumping unit.

Successful competition in a market economy is possible only by obtaining production at minimal cost. In our case, this is due to the cost of electricity during the production of oil through pumping unit. Saving electricity during oil extraction is one of the most effective ways to optimize resources. Thus, the most promising solution for oil production is to save electricity during production and increase the time between repairs of wells.

General methodological issues of energy efficiency in oil production are the analysis of the structure of production costs, as well as methods of transportation and determination of energy efficiency indicators. Figure 13 shows the structure of electricity costs for the whole complex of technological processes of oil production.

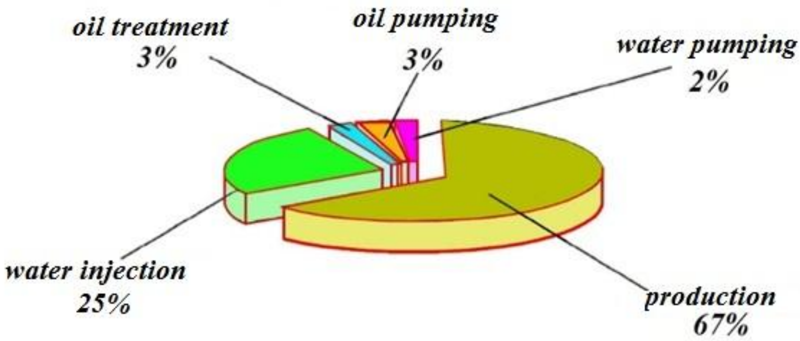

Figure 13. The structure of electricity costs for the whole complex of technological processes of oil production

General methodological issues of energy efficiency in oil production are the analysis of the structure of production costs, as well as methods of transportation and determination of energy efficiency indicators[24]. At Figure 13 shown the structure of electricity costs for the whole complex of technological processes of oil production.

As can be seen from the figure, in the structure of electricity costs, most of the money is spent on oil production (extraction). These costs are specific to each field and are determined primarily by the dynamic level of the oil wells and the structure of the operating pumping equipment fleet.

As a result of the calculations, it was determined that the full cost of the new constructive solution of beamless oil pumping unit:

$$Э_f = C + Э_d = 18642 + 6000 = 24642\ azn$$

[24] Расчет экономической эффективности внедрения новой техники на предприятиях легкой промышленности / Баторова С.З., Алексеева Р.Д. / Издательство ВСГТУ. г. Улан-Удэ, 2006, 21 с.

there $C = 18642\, azn$ - cost of production of one exemplar new constructive solution of beamless pumping unit; $Э_d = 6000\, azn$ – is the cost of designing of pumpin unit.

Payment period of the new constructive solution of beamless pumping unit, taking into account the economic benefits obtained only from electricity:

$$T = \frac{Э_f}{Ф} = \frac{24642}{1340{,}28} = 18{,}4\, year$$

Daily production of pumping unit:

$$Q_d = \frac{\pi d^2}{4} \cdot S_{st} \cdot n \cdot k_{day} \cdot t =$$

$$= \frac{3{,}14 \cdot 0{,}038^2}{4} \cdot 1{,}5 \cdot 10 \cdot 24 \cdot 60 = 24{,}5 \frac{m^3}{day}$$

there D=38 mm – plunger diameter; $S_{st} = 1{,}5\, m$ – plunger pump stroke; n=10 min^{-1} – reciprocating speed of the pumping unit; $k_{day} = 24$ – hours per day; $t = 60$ - minutes per hour.

Taking into account the losses during production, $Q_d = 20\, m^3/day$ can be accepted.

Then the actual volume of oil that the pumping unit will produce in one month:

$$Q = Q_d \cdot \frac{T_\varphi^1}{24} = 20 \cdot \frac{620{,}5}{24} = 517{,}08\, m^3/month$$

there T_φ^1=620,5 hour - is the actual monthly operating time of the pumping unit.

If we take into account that the cost of one barrel (159 liters) of oil produced on the world market is about \$ 70 (119 azn), then the cost of oil produced by the pumping unit during the actual operating time in one month:

$$C = \frac{Q \cdot 10^3}{159} \cdot 119 = \frac{517{,}08 \cdot 10^3}{159} \cdot 119 = 386997\, azn$$

Annual electricity savings are defined as follows ($kVt \cdot hour/year$):

$$\Delta W = (P_{se} - P_{em})T_{\varphi} = (5{,}5 - 3{,}5)7446 = 14892\ kVt \cdot \frac{hour}{year}$$

there $P_{se} = 5{,}5\ kVt$ - the power of the electric motor of the existing oil pumping unit; $P_{em} = 3{,}5\ kVt$ - the power of the electric motor of the proposed new constructive solution of beamless oil pumping unit; T_{φ}- is the annual actual operating time of the pumping unit, $hour/year$.

$$T_{\varphi} = T_{ma}(1 - K_{re}) = 8760(1 - 0{,}15) = 7446\frac{hour}{year}$$

there $T_{ma} = 8760\ hour$ – machine time of the pumping unit; K_{re} - is the coefficient taking into account the time of repair, $K_{re} = 0{,}15$ is accepted according to the calendar time of maintenance of the new constructive solution of oil pumping unit.

Then, taking into account the economic benefit obtained only from electricity, the annual economic benefit obtained as a result of the application of a new constructive pumping unit:

$$\Phi = \Delta W \cdot Ц_{el} = 14892 \cdot 0{,}09 = 1340{,}28\frac{azn}{year}$$

It was noted that the State Oil Company of the Republic of Azerbaijan has more than a thousand oil wells operated by the “Absheronneft” OGPD alone. Therefore, if we assume the number of new constructive solution of beamless oil pumping unit to be operated in these wells is 1000, then the annual economic benefit will be $1340280\ azn$.

PRACTICAL RECOMMENDATIONS

1. It was developed, prepared and tested laboratory sample of new constructive solution of beamless sucker-rod pumping unit, which converting mechanism consisting from crank-block-rope, and its novelty was approved by the Eurasian patent [27].
2. Based on the research and calculations, it was determined that along with the reduction of metal capacity and reliability of new constructive

solution of beamless sucker-rod pumping unit, it is possible to save up to 45% of electricity through the regulation of reverse counter weights during their operation, which is important for economic development [28].

3. The kinematic analysis of new constructive solution of beamless sucker-rod pumping unit was performed to assess the relative length coefficient and the effect of the relative eccentricity on the travel of the rods suspension point, its speed and acceleration. For the first time, the maximum values of speed and acceleration for deaxial mechanisms and the angles of rotation of the crank according to these values were determined analytically. It was found that the relative length coefficient and the relative eccentricity have a significant effect on the travel of the rods suspension point and phase angles [29].

4. For the first time, method of balancing of new constructive solution of beamless sucker-rod pumping unit was developed and analytical expressions were proposed to calculate the torque on the output shaft of the gearbox. As a result of calculations made for the new constructive solution of beamless sucker-rod pumping unit in accordance with the technical parameters of the CKD3-1,5-710 pumping unit, it was determined that both the moving and rotary counterweights have such a value that where during the rod suspension point up and down stroke the pumping unit is properly balanced and the torque on the output shaft of the reducer reaches the minimum value [15; 28].

5. According to the comparative analysis of the calculations made in accordance with the technical parameters of the existing pumping unit, it was determined that the value of torque on the output shaft of the new constructive solution of beamless sucker-rod pumping unit′s reducer is on average 1.85 times less than the value of torque on the output shaft of existing pumping unit′s reducer, which saves about 45% electricity and the new constructive solution of beamless pumping unit have a more perfect design [28].

6. It has been determined that the main factor influencing the interaction between the amount of fluid flowing to the bottom of the

well and the amount of fluid extracted through the pumping unit, maintaining the dynamic level of the liquid column in the well, eliminating idle stops on the pumping unit, increasing its daily productivity and economic efficiency is the reciprocating speed of the rods suspension point. For the first time, an analytical expression has been proposed to determine the optimal reciprocating speed of the rods suspension point depending on the wellbore geological reserve (debit) and the actual volume of the borehole well pump and the condition that the dynamic level remains constant [25].

7. It was found that the loss of plunger flow in static mode depends not only on the design of the rod, pipes, rope and the diameter of the pump, but also on a lot of technological parameters, such as wellhead and pipe pressures, pump deflection depth and dynamic level [24].

8. Based on the analysis of possible kinematic schemes of the new constructive solution of beamless sucker-rod pumping unit, it was determined that the directions of rotation of the drive and driven shafts coincide with the directions of rotation of the gear blocks mounted on the sliding bearing. In this case, the resistance in the double sliding bearings becomes a beneficial driving force, which becomes a useful factor that characterizes the internal driving force of the oil. For the first time, analytical expressions were obtained to determine the coefficient characterizing the total internal resistance force of the oil in the sliding bearing of a new constructiv solution of reducer [8].

9. The design of the device was developed for experimental study of the coefficient characterizing the internal resistance force of the oil in the sliding bearings of the new construtive solution of redcuer and its laboratory sample was prepared and tested and its novelty was confirmed by the patent of the Republic of Azerbaijan [4; 11]. It has been found that since the rotation direction of sliding bearings in the new constrive solution of reducer in the same direction as the direction of rotation of the shafts, the mutual displacement of the oil layers in the gaps between the bearings, the partial contact of the contact surfaces have a positive effect [5].

10. Based on the results of experiments conducted on the test rig, it was determined that the coefficient of friction in double sliding bearings is significantly affected by the load, rotational speed of the main and auxiliary shafts and the diametrical gap between the auxiliary shaft and the stuffing, and it is important to expect their limit values to ensure that the cushion button operates continuously for the required service life [31].

11. The results of the study show that the effect of the coefficient of friction on the coefficient taking into account the effect of oil and the associated effect of gear transmission on the load capacity is significant. The proposed method, i.e. the method of calculating gear transmission taking into account friction, allows not only to assess the latent effect of friction when designing gear transmissions, but also to achieve significant economic benefits in the production and operation of reducer [22].

12. It was determined that the total technological and design errors made during the manufacture and installation of pumping unit affect it is the main kinematic parameters, i.e. the displacement, speed and acceleration of the rods suspension point is approximately (3 ... 9%). Therefore, in order to study the effect of the acceleration of the rod suspension point, which is the main parameter of experimental importance, on the dynamic forces on the rod column, its deviation from the acceleration during ideal harmonic motion was determined and the error coefficient and dimensional kinematic parameter were evaluated [21].

13. Studies have shown that the change in the value of dynamic force and its mode of operation during a cycle of operation of a new constructive solution of beamless sucker-rod pumping unit depends not only on the angle of rotation of the crnak, but also depends on the ratio of the length of the crank to the length of the rope (λ) and relative eccentricity coefficient (ε) [30].

The main points of the dissertation are presented in the following articles:

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[3, 15, 19, 21, 25, 29, 31] work was done alone.
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