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OLGA V. IVANOVA[1,2], ANDREI VICTOR SANDU[3,4,5*], VITALIY V. SAVINKIN[1,2],
IOAN GABRIEL SANDU[3], PETRICA VIZUREANU[3,5*]

INVESTIGATION OF ROD PUMP WEAR DURING CHANGES IN LOAD DYNAMICS AND THE DESIGN TRAJECTORY OF THE ROD

The article is the first to investigate the mechanism of wear of the oil production pump couplings as a result of a violation of the design trajectory of kinematic connections from a given axis of symmetry. The critical values of the acting moments and stresses in case of inefficient load redistribution across the cylinder at different positions of the worn plunger are substantiated and presented. Using the Cauchy parameter model, an indirect assessment of the effect of dynamic loads on the actual capacity of the pumping unit is given, taking into account the wear of kinematic connections. It was possible to improve the Lame equation by describing the law of reciprocating motion of a plunger with uneven wear, as the equation of longitudinal elastic vibrations of the rod, taking into account the distributed specific external load acting on the rods, distorting the design trajectory of the kinematic pairs. The process of interaction of the elements of the rod-plunger-cylinder system of an oil production pump with uneven axial moments is investigated. The authors proposed a new term for the deviation of the design trajectory ($\Delta\lambda$) of the rod and plunger relative to the axis of symmetry of the cylinder. The mathematical model of pump capacity (Q_a) and its resource life has been improved when changing the maximum allowable gap $\delta_t = f(i)$ in the plunger-cylinder system of a rod pump and leaks (q_t) due to violations of the design trajectory ($\Delta\lambda_\Sigma$) of kinematic pairs due to uneven distribution of dynamic loads. A dynamogram has been developed to determine the dependence of the load distribution on the movement of a worn plunger during basic contact stress cycles.

Keywords: Design trajectory; oil production pump; plunger wear; strength calculation; contact wear; mathematical model; pump cyclogram

1. Introduction

When developing low-flow wells, the operating conditions of pumping complexes deteriorate, and accordingly, the requirements for their reliability and durability are tightened. Consequently, mining companies are increasingly faced with the need to develop and apply modern, efficient mining pumping systems with increased durability. Their resource and efficiency parameters are laid down at the design stage with the formation of a forecast reliability map for operational wear.

The existing pump rods are manufactured using modern technologies from materials with high physical and mechanical properties. For example, pump cylinders are made of 38Cr2MoAlA Steel grade material (GOST 4543-2016), hardened by nitriding with an inner surface hardness of 45÷65 HRC (HV 4.5÷8.0 GPa) [1]. The total depth of the nitrided layer is not more than 0.2÷0.5 mm. However, there is an onset of maximum wear for 1/3 of the service life, followed by pump failure due to rod breakage or pump jamming. Consequently, unknown processes take place in the pump, which change the structure and geometry of its elements, reducing the efficiency of kinematic connections under constant external load of the rocking machine. The tension of the parts at different time intervals in different positions of the plunger also violates the logic of the movement of precision pairs and clearances in shaft and bore systems (plunger-cylinder).

The efficiency of the pump largely depends on the dynamic, power and power characteristics of the pump, the kinematics of its elements, the design features of the drive and the complexity of the well relief [2]. The hardness of the working surfaces

[1] NORTH KAZAKHSTAN UNIVERSITY, TRANSPORT AND MECHANICAL ENGINEERING DEPARTMENT, M. KOZYBAEV 86, PUSHKIN STREET., PETROPAVLOVSK, 150000, KAZAKHSTAN
[2] JOLDASBEKOV INSTITUTE OF MECHANICS AND ENGINEERING; 28, SHEVCHENKO STR., ALMATY, 050010, KAZAKHSTAN; ITF.NKZU@MAIL.RU (O.V.I.)
[3] GHEORGHE ASACHI TECHNICAL UNIVERSITY OF IAȘI, FACULTY OF MATERIALS SCIENCE AND ENGINEERING, BOULEVARD D. MANGERON, NO. 51, 700050 IASI, ROMANIA
[4] ROMANIAN INVENTORS FORUM, STR. P. MOVILA 3, 700089 IASI, ROMANIA
[5] ACADEMY OF ROMANIAN SCIENTISTS, 54 SPLAIUL INDEPENDENTEI ST., SECT. 5, 050094 BUCHAREST, ROMANIA

* Corresponding authors: andrei-victor.sandu@academic,tuiasi.ro; mailto:peviz2002@yahoo.com

is always taken as the basis for determining the contact strength limit and its base number of cycles before fracture. The experience of pump operation, the results of metallographic research and analysis of repair and restoration certificates of failed pumps allowed us to conclude that surface hardness in most cases is not the main cause of failure. When examining the section of the plunger, rod and their contact surfaces, the integrity of the coatings is observed, and the grain size and color range of pearlite inclusions in the phase structure vary from the focus of the load application to the base of the plunger. The rate and moment of fatigue stress occurrence largely depend on the metal base, its phase composition, and the dispersion of inclusions of martensitic and austenitic grains. Based on the conducted studies, it has been proved that cyclic durability depends more on the magnitude of the applied specific load acting for 1 s [2]. The wear of the interface surfaces not only changes the geometry of the parts, but also distorts the symmetry of the gaps and the design trajectory of the rod-plunger-cylinder cycloidal systems.

Examining the technological features of the extraction pump, it was found that with its high safety margin, there are frequent failures associated with a sharp decrease in the actual capacity (Q_a) of the pumping unit and the well productivity coefficient (K_{pr}).

Thus, there is a need to form and systematize the technical and operational parameters of the extraction pump and establish the dependence of changes in quality criteria that characterize the efficiency and durability of its operation with the inevitable wear of parts.

The relevance of the research is due to the need to improve the mathematical model for describing the causal relationship between the pump wear process and changes in the design trajectory of the rod and plunger relative to the axis of symmetry for effective design of rod pumps with high resource durability, with the possibility of effective redistribution of resistance forces along the contact surfaces of the plunger-cylinder system. The establishment and justification of optimal conditions for operating loads on worn surfaces with loss of mechanical properties of the wear surface when it deviates from the design trajectory relative to the axis of symmetry will ensure ultra-accurate prediction of the pump's resource during intensive operation. The relevance is confirmed by the scientific and practical problem of the lack of accurate methods for designing high-reliability rod pumps that ensure efficient production at low-flow wells.

The purpose of the research is to increase the operational efficiency and durability of a highly loaded rod extraction pump by justifying critical loads acting during intensive wear of the contact surface from a disrupted design trajectory.

Object of research: the process of dynamic interaction of the plunger-cylinder system and tubing, and load distribution.

The research methods of kinematic analysis and strength calculation were applied. The continuous process of cyclic wear of the plunger-cylinder coupling is investigated with a random deviation of the pump rod from the design trajectory. The methodology of systematization of cause-and-effect relationships has been adapted, and a method of load distribution in the time interval at different plunger positions has been developed. A mathematical apparatus has been effectively applied to establish the dependences of changes in geometric and strength characteristics on operational wear parameters when deviating from the design coordinates of the pump elements.

2. Substantiation of criteria for effective performance of a rod pump under the influence of dynamic forces on the rod-cylinder-plunger wear systems

The dynamic mode of operation of underground pumping equipment is a complex process consisting of static loads and dynamic components – inertia forces (F_{in}) and vibration loads (F_{vibr}). During the operation of the rocking machine (Fig. 1), when the polished rod moves down and up, the amount of force on the rod column and, consequently, on the pump plunger differs by 30-50%.

When moving the column down, the cylinder-plunger system does not perform useful work and idles; when moving up, the pump performs useful work on lifting the oil fluid [3]. When the rod column and plunger move downwards, resistance forces act on them. As a result of these resistance forces, the rod column bends and twists. This leads to additional rotation of the pump plunger, and when moving upward, the same phenomena occur, only in the opposite direction.

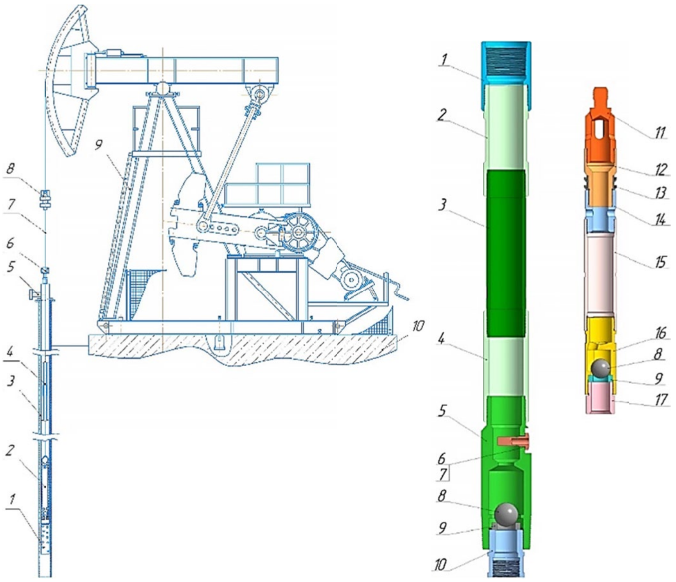
In general, the theoretical breakage frequency of the rod column can be expressed in terms of the dependence on the force acting on the column, the strength limits of the rod material. as well as the geometric parameters of the plunger and rod column [4]:

$$\varphi = b \cdot \left(\frac{D}{d}\right)^{3.0502k+0.1247} \cdot l^{2k+1}$$

$$\text{or } \varphi = 0.0018 \cdot \left(\frac{D}{d}\right)^{4.923} \cdot l^{4.146} \qquad (1)$$

where φ – frequency of rod column breaks, year^{-1}; D – diameter of the pump plunger, m; d – diameter of the rod column, m; l – length of the rod column, m; b – dimensionless coefficient depending on the properties of the rod material; k – the parameter characterizing the slope of the Wehler curve.

However, this relationship is based on data from an existing cliff and does not allow us to establish a cause-and-effect relationship. In the proposed relationship, the main emphasis is placed on the rod material and the coefficient of complexity of the well in terms of its angle of inclination. This does not fully reflect the true operating modes, the kinematics of the pumping unit, and the cyclically unstable dynamic loads associated with structural malfunctions of the mechanism at different time intervals. The dynamics of the mechanical system is also not taken into account, and the coupled parts are subject to external oscillatory (ω) effects of the system, especially with varying speed and amplitude at the time of formation of mechanical deposits and paraffination of the friction surface.



a) b)

Fig. 1. Diagram of a rod pumping unit with a well depth of $H \leq 3000$ m: a) 1 – protective device in the form of a gas or sand filter; 2 – downhole plunger pump of plug-in or non-plug-in types; 3 – tubing; 4 – column of pumping rods; 5 – T-joint; 6 – gland seal; 7 – polished rod; 8 – pipe suspension of wellhead fittings; 9 – rocking machine; 10 – foundation; б) 1 – the clutch; 2 – upper extension cable; 3 – cylinder; 4 – extension cable bottom; 5 – valve housing; 6 – washer; 7 – the screw is knocked down; 8 – ball; 9 – valve seat; 10 – the nipple; 11 – cage; 12 – housing; 13 – dirt collector; 14 – cage adapter; 15 – plunger; 16 – valve housing; 17 – seat holder

Analysis of research by domestic and foreign scientists such as Neil Robert Hall, John F. Mabry, Nielsen Jr. William D., Agamirzoeva D.I., Adonin A.N. [5], Virnovsky A.S., Ivanovsky V.N., Ismagilov F.G., Urazakov K.R., Yurchuk A.M. and other researchers have shown that the determining criteria for the operability and efficiency of a rod pumping unit are resource durability (T, day), overhaul life ($OL \geq 360 \div 500$ days), the operating coefficient ($K_{op} = 0.94 \div 0.98$), as well as pump performance indicators: Q_a – the actual capacity of the pumping unit and K_{pr} – the productivity coefficient of the well (Q_a = 2.5÷110 m^3/day, K_{pr} – m^3/(day·MPa)) [6-8].

TABLE 1

Pump performance indicators

Name of the parameter	The physical meaning	Pump efficiency criteria
Operating coefficient, K_o	The ratio of the time worked by the pump (T_{sp}) to the number of calendar days/period (T_{cal})	$K_o = \frac{T_{work}}{T_{cal}}$
Overhaul life, OL	The ratio of the total time worked (Tot) per year to the number of repairs (P) over the same period	$OL = \frac{T_{work}}{P}$
Productivity coefficient K_{pr}	The amount of oil (Q) that can be extracted from a well when creating a pressure drop at its bottom of 0.1 MPa (Δp)	$K_{pr} = \frac{Q}{\Delta p}$

Ensuring the durability, efficiency, strength and quality of rod depth pumps (RDP) is characterized by the reliability and trouble-free operation of all structural elements in difficult operating conditions.

Reliability depends on the complex parameters of the RDP's operability, which are determined by such design properties as reliability, durability, maintainability, safety, and a complex indicator – the availability coefficient [9-11].

The reliability indicators include the following parameters, namely, the average operating time to failure is 365 days; the average operating time to failure is 274 days. Durability indicators include the following indicators, namely, the average service life is 5 years; the average pre-repair life is 730 days; the average inter-repair life is 548 days. The maintainability indicators include the following indicators, namely, the average recovery time is 8 hours; the average recovery time is 8 people · h.

The retention rate is the average shelf life of the pump – 2 years. A comprehensive reliability indicator is the availability coefficient, K_{AV} [11-13]:

$$K_{AV} = \frac{T_f}{T_f + T_r} = 0.972 \qquad (2)$$

where: T_f – average operating time per failure, h; T_r – object recovery time in case of sudden failures, h.

According to the regulatory documents, "during the full service life of the rod pump (5 years), 2 major repairs and 3 routine repairs are provided". The average duration of repairs: current – 8 hours, capital – 96 hours [12].

The condition for ensuring a given pump performance is its capacity (Q) equal to the inflow of petroleum fluid. The theoretical performance of the pump delivery its operation is determined by the following relationship [5]:

$$Q = 1440 \cdot \frac{\pi \cdot d_{pl}^2}{4} \cdot n \cdot S_{pl} \cdot \eta \qquad (3)$$

where: Q – theoretical pump capacity, m^3/day; 1440 – number of minutes in a day; d_{pl} – diameter of the pump plunger, m; n – number of double strokes (number of swings) per minute, min^{-1}; S_{pl} – plunger stroke length, m; η – pumping unit conveyance factor, η = 0.6÷0.8.

Thus, the set pump performance depends on the following parameters: plunger diameter (d_{pl}); plunger stroke length (S_{pl}); fluid extraction rate (v).

Maintaining the nominal values of the listed parameters is ensured by strict requirements for the wear resistance of the pump elements. The wear process of pump elements is multiparameter and depends on both structural and technological parameters (diameters (d), geometry change Δ, motion kinematics (n, S, m), technology quality criteria (Ra, HB, δ, porosity %) and manufacturing material (E), as well as dynamic loads (μ), the cyclical distribution of the acting moments of forces ($M(F_i)$) and the vibrational (ω) characteristics of the system.

According to the elementary theory of A.N. Adonin, the kinematic parameters of the rocking machine affect the performance of the pumping unit, taking into account the elongation of the pumping rods λ_r and pipes λ_p from the weight of the liquid column. In this model, the Cauchy parameter m provides an indirect assessment of the effect of dynamic loads on the actual supply of the pumping unit [5]:

$$Q_a = 1440 \cdot F_{pl} \cdot n \cdot \left[S_A \cdot \left(1 + m \cdot \frac{\mu^2}{2} \right) - \left(\lambda_r + \lambda_p \right) \right] \tag{4}$$

where: Q_a – actual pump capacity, m^3/day (Q_a = 2.5÷110 m^3/day); F_{pl} – the cross-sectional area of the plunger (or pump cylinder), m^2; n – number of double strokes (number of swings) per minute, min^{-1} (up to 12÷15 min^{-1}); S_A – stroke length of the balancer head (polished rod), m; μ – the Cauchy parameter, μ = 0.35÷0.45; m – the kinematic coefficient, which takes into account the kinematic parameters of the rocking machine:

$$m = \frac{1 + \frac{r}{L_{cr}}}{\sqrt{1 - \left(\frac{r}{k} \right)^2}} \tag{5}$$

where: r – crank radius, m; L_{cr} – connecting rod length, m; k – the length of the rear shoulder of the balancer, m; λ_r and λ_p – elongation of pumping rods and pipes from the weight of the liquid column, that is, from the action of static forces, m:

$$\eta_1 = \lambda_r + \lambda_p = \frac{P_l \cdot L}{E} \left[\sum \left(\frac{1}{f_{i(r)}} + \frac{1}{f_{i(p)}} \right) \right] \tag{6}$$

where: η_1 – conveyance factor, which takes into account the elastic elongation of pumping rods and pipes from the action of static forces; P_l – the weight of the liquid column above the pump plunger, N; L – depth of descent of the column of rods, m; E – modulus of elasticity of the material, Pa; f_r – cross-sectional area of the pumping rods, m^2; f_p – cross-sectional area of pumping pipes (in metal), m^2.

The equality of pump delivery obtained by A.N. Adonin requires improvement, it partially takes into account the dynamic properties of individual system elements, focusing on the performance characteristics of the rocking machine and the natural deformation of rods and tubing at a given grade of material. However, this equation does not provide a clear description of the wear processes during narrowing of pipe diameters when asphalt-resin-paraffin deposits adhere, changes in inertial forces and masses of parts with asphalt-resin-paraffin deposits are not taken into account, and there is no description of the complex processes of rod movement, that is, how the deviation of the rod from its design trajectory affects the dynamics with total wear of the rod and cylinder under the action of complex resistance.

In the theory of delivery change, Yurchuk A.M. added a term that takes into account the gain of the move due to inertial forces, but there is no dynamics of the system elements and the influence of the kinematic characteristics of the system under study changed during wear is not taken into account. The delivery parameter according to the theory of Yurchuk A.M. is determined by the dependence [8]:

$$Q_a = 1440 \cdot F_{pl} \cdot n \cdot \left[S_A - \left(\lambda_r + \lambda_p \right) + \frac{225 \cdot L^2 \cdot n^2 \cdot S_A}{10^{12}} \right] \tag{7}$$

where L – depth of pump descent, m.

When studying the processes affecting pump performance, Virnovsky A.S. obtained a fairly capacious model of pump delivery, taking into account the elongation of the pumping rods λ_r and pipes λ_p from the weight of the liquid column. In this model, the influence of dynamic loads is estimated through the Cauchy parameter m and through the coefficient β, which takes into account the resistance to rod movement in a viscous medium. The mathematical description of the pump performance according to Virnovsky A.S. is determined by the following relationship:

$$Q_a = 1440 \cdot F_{pl} \cdot n \cdot \left[\frac{S_A}{\left(\cos^2 \mu + sh^2 \beta \right)^{1/2}} - \lambda \right] \tag{8}$$

where: λ – the sum of static deformations of pumping rods and pipes, m; β – a coefficient that takes into account the force of resistance when moving rods in a viscous liquid:

$$\beta = \frac{b \cdot L_r}{a} \tag{9}$$

where: b – the friction constant is assumed to be in the range 0.2¸1.0 s^{-1}; L_r – depth of descent of the column of rods, m; a – speed of sound propagation in rods, m/s.

3. Investigation of pump efficiency due to changes in geometric and strength characteristics of rods under variable loads

When the column of rods moves down, an axial force (P_c) occurs, which is caused by several factors, including the friction of the plunger against the cylinder and the resistance in the discharge valve to the flow of liquid. This force creates stress in the column of rods, which can lead to various deformations, including longitudinal bending and compression of the lower part of the column.

Compression of the rod column under the action of axial force can significantly affect the efficiency of the pump. The deformation resulting from the action of this force leads to a decrease in the stroke length of the plunger. This, in turn, reduces the volume of liquid lifted per pump cycle, which negatively affects the overall delivery of the system.

In addition, the longitudinal bending of the lower part of the column can cause additional stresses on the joints between the rods and the plunger, which increases the risk of fatigue damage and rod breaks. Over time, if such deformations are not controlled, this can lead to significant damage to both the rod equipment and the pump itself.

In formula (8) λ – the deformation of pumping rods and pipes consists of several components that determine the deformation process from the action of external factors and emerging internal force factors:

$$\lambda = \lambda_r + \lambda_p + \lambda_{com} + \lambda_{ben} \tag{10}$$

where: λ_r – deformation of the rods caused by a pressure drop above and below the plunger space when the rods move upward, m; λ_p – deformation of pipes during downward movement of rods, m; λ_{com} – the deformation resulting from compression of the lower part of the column of rods, m:

$$\lambda_{com} = \frac{P_c \cdot L}{E_r \cdot f_r} \tag{11}$$

where: L – depth of pump descent, m; P_c – force causing longitudinal bending and compression of the lower part of the rod column, N; E_r – modulus of elasticity of the rod material, Pa; λ_{ben} – the deformation resulting from the longitudinal bending of the lower part of the column of rods, m [13]:

$$\lambda_{ben} = \frac{P_c \cdot R_s^2 \cdot L_{com}}{2 \cdot I \cdot \left[\sqrt{1 + \frac{P_c \cdot R_s^2}{E_r \cdot I}} + I \right]^2} \tag{12}$$

where: $L_{com} = P_c / q_r$ – length of the compressed part of the column of rods, m; q_r – weight of one linear meter of rod length in liquid, kg/m; I – moment of inertia of the cross section of the rods, m^4; R_s – the radius of the spiral (helical line) along which the compressed part of the column of rods is bent, m:

$$R_s = \frac{D_p - d_r}{2} \tag{13}$$

where: D_p – inner diameter of pipes, m; d_r – rods diameter, m.

In known techniques, the deformation (elongation) of the column of pumping rods/tubing is determined by the following ratio:

$$\lambda = \frac{P_l \cdot L}{E \cdot f} \tag{14}$$

where P_l – the weight of the liquid above the pump plunger and in the tubing column, N:

$$P_l = \rho_l \cdot g \cdot L \cdot F_{pl} \tag{15}$$

where: L – depth of descent of the column of pumping rods/tubing, m; ρ_l – liquid density in kg/m^3; g – acceleration of free fall, $g = 9.81$ m/s^2; E – modulus of elasticity of the first kind of column of pumping rods/tubing, Pa; f – $f_{r.\,av}$ (average cross-sectional area of rods of 1, 2, 3 steps) or f_p (cross-sectional area of tubing), m^2:

$$f_{r.av} = \frac{1}{\frac{\varepsilon_1}{f_{r1}} + \frac{\varepsilon_2}{f_{r2}} + \frac{\varepsilon_3}{f_{r3}}} \tag{16}$$

where: f_{r1}, f_{r2}, f_{r3} – cross-sectional area of rods of 1, 2, 3 steps, m^2; ε – part of the column of rods of a given diameter in hundredths.

Expression (14) does not fully reflect the actual geometric changes of the rod during operation of the pump under load. During operation, cyclic alternating loads arising from bending, stretching, torsion, compression and friction act on the column of pumping rods. All these types of loads act on the pumping rods when they are working in the well constantly and in a complex. Therefore, formulas (10) and (14), for example, do not take into account the effect of complex resistance that occurs in the structure under the dynamic action of not only compressive and bending loads, but also torques. The influence of the weight of asphalt-resin-paraffin deposits on the load was also not taken into account.

Therefore, the model of the mathematical description of the physical process of rod extension requires improvement by introducing new parameters: $\sum_{i=1}^{n} G_i$ – the total weight of the plunger, rods and fluid located above the pump plunger and in the tubing column acting on the suspension point and θ_{ARPD} – a coefficient that takes into account the weight from the adhesion of asphalt-resin-paraffin deposits:

$$\lambda = \frac{\sum_{i=1}^{n} G_i \cdot \theta_{ARPD} \cdot L}{E \cdot f} \tag{17}$$

The torsional stiffness condition of the pump rod column is characterized by the twist angle (φ) and is defined by the following expression:

$$\varphi = \frac{M_{tor} \cdot l}{GI_p} \le [\varphi] \text{ or } \varphi = \varphi_0 + \int_0^z \frac{M_{tor}}{GI_p} dz \tag{18}$$

where: M_{tor} – the torque in the cross-section under study, N·m; l – the length of the section in question, m; GI_p – torsional stiffness, N·m^2; G – modulus of elasticity of the second kind (shear module), Pa; I_p – polar moment of inertia of the rod section, m^4; $[\varphi]$ – permissible twisting angle, rad.

The strength resistance of the pump link to complex internal force factors is characterized by the relative angle of twisting and simultaneous deflection f. Accordingly, the maximum permissible values of these quantities are limited by the conditions $\varphi \le [\varphi]$ and $f \le [f]$, respectively.

However, the wear process is multifaceted and difficult to study, and even more so in mathematical description, due to the operation of pump elements in different environments with several physical processes (erosion, corrosion, mechanical and vibration effects, as well as potential differences of heterogeneous materials). These parameters are not taken into account in the classical Virnovsky A.S. equation and significantly reduce the confidence probability of failure prediction, increasing the standard deviation from reliable values. The Virnovsky equation (14) determines the performance under ideal operating conditions of the pump, since the standard parameters (ρ_l, F_{pl}, f) and reference coefficients (E, μ, β) are accepted. During operation, the moving working elements of the contacting surfaces (rod, tubing, plunger-cylinder, ball-seat, etc.) inevitably wear out and change not only geometric shapes, but also distort the kinematics of the trajectory relative to the design axis of symmetry. The deviation of the rod trajectory from the design axis under the influence of external loads leads to an inefficient distribution of forces over the working section, causing fatigue stresses in the structure of the pump elements. The dynamics of moving elements under cyclic loading generates vibrations, leading to an increase in the zones and areas of concentration of internal stresses and, as a result, the occurrence of failures.

The importance of studying the design deviation of the trajectory of structural elements from the design axis of symmetry and the original coordinates is confirmed by the breadth of the scope of the results. This phenomenon is fairly applied in the field of research and strength calculations for determining the resource of railway transport units, road construction equipment and oil drilling rigs [14]. Changing the directional load vector affects the redistribution of critical stresses in the structure, which differ from those planned during the design of parts. The main disadvantage of classical methods of calculating tension and wear resistance is the need to perform calculations in ideal conditions when the calculated object of study meets the manufacturer's standards. Under real-world operating conditions, the interface surfaces wear out intensively. Wear leads to deterioration of the structural parameters of the contact surface (fatigue wear) and changes in the spatial shape, geometry of the structure and surface roughness [14]. Their random distribution makes it difficult to predict the resource durability of highly loaded machines. The change in the gearing efficiency of the pinion wheel during wear of the contact teeth is described in sufficient detail [15]. Due to a slight deviation of the contact spot from the design axis of symmetry during tooth wear, the map of the stress and critical load redistribution spectrum changes significantly; these processes are described in sufficient detail in [15].

Thus, an important scientific task is to study the cyclically varying load (μ) (vibrations (ω_0)) and the effect of the deviation of the trajectory ($\Delta\lambda$) of the movement of parts from the design axis of symmetry on the fatigue (σ_s) degradation of the phase structure of the material, leading to surface wear and, as a result, to failures (N^r) of the pump.

Therefore, the new mathematical description of the pump supply in the first approximation, taking into account the influence of the total load from the weight of the plunger, rods and liquid located above the pump plunger and in the tubing column, and the weight of asphalt-resin-paraffin deposits during operation, will take the following form:

$$Q_a = 1440 \cdot \frac{\pi d_{pl}^2}{4} \cdot n \cdot \left[\frac{S_A}{(\cos^2 \mu + sh^2 \beta)^{1/2}} - \frac{\sum_{i-1}^{n} G_i \cdot \theta_{ARPD} \cdot L}{E \cdot f} - \overline{\lambda} \right] \quad (19)$$

In expression (19), the term $\overline{\lambda}$ characterizes the deformation of pumping rods and pipes resulting from compression of the lower part of the rod column and from the longitudinal bending of the lower part of the rod column, proposed by Virnovsky A.S. An important component in this model will be the proposed parameters measured directly by standard measuring instruments during major or routine repairs, allowing accurate prediction of the service life the pump.

Another important characteristic of the rod-plunger pump design and technology system is the natural oscillation frequencies (ω_0). Dynamic frequency studies are due to the fact that if the external periodic load coincides with the natural frequencies of the system, a resonance phenomenon occurs, which causes vibration, a multiple increase in stresses, leading to a violation of the design trajectory of the pump links, creating additional resistances that contribute to the destruction of the entire pump structure as a whole. To avoid resonance in operating conditions, or, conversely, to use its effect, it is necessary to determine its frequency range at the design stage of the pump design.

Over time, the geometry and physical and mechanical properties of the material change in the working parts of the pump. The causes may be different: static, dynamic, and temperature effects; corrosion, fatigue aging, mechanical wear, and so on. The transverse vibrations of the products are of the greatest interest, which researchers generally represent in the form of a conditional rod with a cantilever attachment.

However, the issue of the influence of vibration vibrations of the pump elements on the change in the forces of resistance to movement of the rod-plunger system and the efficiency of the pump supply with steady wear of the working surfaces has not been studied. In matters of reliability and wear resistance of the system for a given period of operation, gaps have been identified in knowledge about the formation of fatigue stresses in the phase structure of the martensitic class of steels, at the stage of the beginning of the destruction of interatomic bonds leading to pores and microcracks. The assessment of the mechanical properties of the pump material for the probability of resonances in the operating frequency range of external influences makes it possible to make changes to the design at the design stage. This will help to avoid or significantly reduce the likelihood of resonances, failures and increase the service life of pumps during operation. In order to implement technological solutions to increase resource durability, it is necessary to substantiate the main criteria for failures of the rod depth pump.

4. Development of a model for changing pump performance due to deviation of the design trajectory (Δλ) of movement from the axis of symmetry of the cylinder

Currently, despite the high reliability of the equipment for oil production from wells with a high degree of production, the most common failures of RDP are associated with breaks of rods and polished rod, lapels of rods and plunger, jamming of the plunger in the cylinder and the destruction of the valve seat.

The researchers have established a dependence characterizing the frequency of failures of the RDP on the pump supply coefficient, which in turn depends on the size of the operating gap. The greater the deviation from the nominal clearance value, the lower the feed rate, and therefore the higher the risk of an emergency. Serious problems for pump operation are caused by plunger impacts in the cylinder, especially at the end of its stroke when moving downwards. Shock loads lead to the destruction of the valve seat and, as a result, to the flaps and jamming of the plunger. The wear of the main plunger-cylinder interfaces forms an irregular trajectory of movement of the structural elements during the working stroke [15]. The plunger under the pressure of liquid forces due to the increased gap changes its axial position in the cylinder with the angles skewed to the cylinder surface. At the first stage, during its reciprocating motion, the walls are bullied and riveted. As the gap increases, the angle of its inclination changes, increasing the moment, which leads to increased vibrations, breakage or jamming of the rod. Therefore, important criteria for the reliability of the pump will be the frequency characteristics of vibration (ω_0), a change in the intensity (σ_{-1}) of the phase structure in a given time interval, and a deviation of the trajectory ($\Delta\lambda$) of the movement of parts from the design axis of symmetry.

It has been established that pump operation is considered normal if the pump capacity coefficient is within the range of η_{pc} = 0.6÷0.8 [5,16]. The feed rate of the downhole pump rod and its actual delivery are influenced by constant and variable factors that have been systematized (Fig. 2).

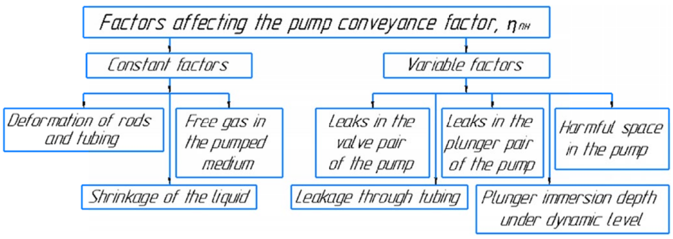


Fig. 2. Factors negatively affecting pumping unit conveyance factor

The maximum permissible condition of the pump occurs with mechanical corrosion damage to the cylinder and plunger of the pump; complete jamming of the plunger (inability to disassemble); when the residual thickness of the nitrided (hardened) layer of the cylinder is less than 50 microns (about 10%), that is, restoration of the cylinder is impractical [17,18]. With an increase in the gap (δ) of more than 0.2 mm, the operation of the pump becomes ineffective, since liquid leaks will amount to more than 25% [19,20].

There are minimum and maximum clearance values in the plunger-cylinder pair (TABLE 2).

During the operational wear of the pump elements, the values of the nominal clearance change relative to violations of the fit and clearance fields. The change in clearance is associated with wear during deflection of the kinematic pump links. Inefficient load redistribution during deflection of the kinematic links of the pump occurs due to an increase in frictional resistance forces when the contact pressure on the contact spot changes. An increase in contact pressure over a limited area is associated with

TABLE 2

RDP fit groups

Fit group	Minimum clearance, mm	The maximum clearance, taking into account the tolerance for the manufacture of the cylinder and plunger, mm
1	0.0	0.063
2	0.025	0.088
3	0.050	0.113
4	0.075	0.138
5	0.100	0.163

an increase in bending and torsion moments when the design trajectory ($\Delta\lambda$) of the rod and plunger movement deviates relative to the axis of symmetry of the cylinder. Previously, researchers did not take into account the deviation of the design trajectory ($\Delta\lambda$) of the rod movement in the strength calculation for wear. The wear rate (J) is also not taken into account when the load is dynamically applied to the contact area of the plunger and cylinder with increased wear (i). Therefore, it is proposed to introduce a new parameter, such as the deviation of the design trajectory ($\Delta\lambda$) of the worn rod from the design axis of symmetry.

This means that when calculating the bending moments of the rod-plunger-cylinder system, it is necessary to take into account the deflection of the rod (f) when the design trajectory ($\Delta\lambda$) of movement deviates from the axis of symmetry of the cylinder:

$$f = \frac{M_x \cdot d_r^2}{10E \cdot I_x} \tag{20}$$

where: M_x – the moment of bending along the axis of the rod, N·m; d_r – rod cross-sectional diameter, m; I_x – moment of inertia of the cross section of the rod, m^4.

The condition for limiting the material properties of the rod under study under the action of a load under conditions of complex resistance will be a limit on the maximum allowable deflection value $f \le [f]$. The specified permissible conditions additionally have limitations on the design geometric parameters of the cylinder and tubing. The physical meaning of the processes taking place under this condition is as follows:

– if the specified condition $f \le [f]$ is violated, contact-mechanical friction of the limited area of the links occurs in the limiting values of the geometric parameters, leading to intensive wear;
– if the condition $f \le [f]$ is violated beyond the limit value of the geometric parameters, the shock-dynamic loads on the design cross-sectional area are exceeded, which leads to a fracture and breakage of the rod.

The deviation of the trajectory of the rod ($\Delta\lambda_\Sigma$) from the design axis of motion for the entire system will take on a new form:

$$\Delta\lambda_\Sigma = \frac{M_x \cdot d_r^2}{10E \cdot I_x} + \delta_w \tag{21}$$

where δ_w – the value of the gap value, taking into account wear, мм.

Thus, the value of $\Delta\lambda_\Sigma$ makes it possible to significantly refine the correction of the calculation of the acting moments of forces and contact pressure in case of deviation of the design axis of symmetry as a result of actual wear of the contact surfaces of the rod-plunger-cylinder system.

During the operation of the pump, due to the inevitable operational wear of the contacting surfaces, leaks of the pumped medium (q) occur through the annular gap of the plunger pair, which are determined by the Hagen-Poiseuille formula (22) for the laminar flow regime [20]:

$$q = \frac{\pi \cdot d_{pl} \cdot \Delta p}{12 \cdot \mu \cdot l} \delta^3 \tag{22}$$

where: d_{pl} – plunger diameter, m; Δp – pressure drop at the plunger ends, Pa; δ – the gap between the cylinder and the plunger of the rod pump, m; μ – dynamic viscosity of the liquid, Pa·s; l – plunger length, m.

To determine leaks through the gap of the plunger pair, we use a relationship that more fully reflects the actual situation of pumping equipment, taking into account the eccentricity of the plunger in the cylinder and the volume of liquid:

$$q = 4.7 \cdot \left(1 + 0.3 \cdot \varepsilon^2\right) \frac{\pi \cdot d_{pl}}{v^{1/7}} \left(\frac{\delta^3 \cdot \Delta p}{\rho \cdot l}\right)^{4/7} - \pi \cdot d_{pl} \frac{v_0}{2} \delta \tag{23}$$

where: ε – relative eccentricity, equal to the ratio of the absolute eccentricity to the gap of the plunger pair, in fractions of units; v_0 – the average speed of the pump plunger, m/s; v – kinematic viscosity of a liquid, m^2/s.

As can be seen from formula (23), the amount of leakage in the plunger pair will be determined and depend on the size of the gap, and increase in proportion to the increase in the gap to the third degree, which is inextricably linked to its wear.

To describe the current gap value as a function of time, the equality was proposed in [5,20]:

$$\delta = A_c \left(2\pi \cdot S_{pl} \cdot t\right)^{\alpha} \left[\left(\frac{l}{S_{pl}}\right)^{\alpha} + m\right] \tag{24}$$

where: A_c – the proportionality coefficient, depending on the operating conditions, the properties of the cylinder material; n – number of swings per minute, min^{-1}; S_{pl} – plunger stroke length, m; t – time, s; l – plunger length, m; $m = A_{pl}/A_c$ – the ratio of the proportionality coefficient of the plunger and cylinder, $m = 0.3$; α – an indicator that characterizes the wear rate of the cylinder and plunger, respectively.

This equality is inaccurate and does not take into account the effect of rod deflection on the change in wear value when its trajectory deviates from the axis of symmetry under the influence of load. Also, the dynamic component (ω_0) of frequency fluctuations, which affect the increase in wear values and the formation of fatigue damage during vibration and shock loads, is not reflected:

$$u(t) = A_0 \cdot \sin(\omega_0 \cdot t + \varphi) \tag{25}$$

where: ω_0 – natural oscillation frequency, s^{-1}; A_0 – the amplitude, m; t – link movement time, min.

This expression (25) describes the oscillation of all points of the system according to the sinusoidal law. Analyzing the above dependencies, an improved model of the formation of actual wear of parts (δ_t) as a function of time in the plunger-cylinder system is proposed, taking into account the violation of motion kinematics during the operation of an oil production pump:

$$\delta_t = \left(\Delta\lambda_\Sigma - \frac{M_x \cdot d_r^2}{10E \cdot I_x}\right) + A_c \cdot \left(\frac{2 \cdot n \cdot S_{pl} \cdot t}{A_0 \cdot \sin\left(\omega_0 \cdot t + \varphi\right)}\right)^{\alpha} \cdot \left[\left(\frac{l}{S_{pl}}\right)^{\alpha} + \frac{A_{pl}}{A_c}\right] \tag{26}$$

Therefore, an improved mathematical model describing the pump supply process (Q_a) when the gap $\delta_t = f(i)$ changes in the plunger-cylinder system of a rod pump, taking into account leaks (q_t) and violations of the kinematics of motion ($\Delta\lambda_\Sigma$) during operation under dynamic operation mode, will take the following form:

$$\begin{cases} Q_a = 1440 \cdot \dfrac{\pi d_{pl}^2}{4} \cdot n \cdot \left[\dfrac{S_A}{(\cos^2 \mu + sh^2 \beta)^{1/2}} - \dfrac{\sum_{i-1}^{n} G_i \cdot \theta_{ARPD} \cdot L}{E \cdot f} - \overline{\lambda}\right] \cdot \left(1 - \dfrac{q_t}{Q_{Th}}\right), \\ \delta_t = \left(\Delta\lambda_\Sigma - \dfrac{M_x \cdot d_r^2}{10E \cdot I_x}\right) + A_c \cdot \left(\dfrac{2 \cdot n \cdot S_{pl} \cdot t}{A_0 \cdot \sin\left(\omega_0 \cdot t + \phi\right)}\right)^{\alpha} \cdot \left[\left(\dfrac{l}{S_{pl}}\right)^{\alpha} + \dfrac{A_{pl}}{A_c}\right], \\ q_t = 4.7 \cdot \left(1 + 0.3 \cdot \varepsilon^2\right) \dfrac{\pi \cdot d_{pl}}{v^{1/7}} \left(\dfrac{\delta_t^3 \cdot \Delta p}{\rho \cdot l}\right)^{4/7} - \pi \cdot d_{pl} \dfrac{v_0}{2} \delta_t, \\ \Delta\lambda_\Sigma = \dfrac{M_x \cdot d_r^2}{10E \cdot I_x} + \delta_w \end{cases} \tag{27}$$

The system of equations (27) shows that the fundamental and more significant parameter that more fully describes the wear process of pump elements is the size of the operational gap between the friction surfaces, which is a function of linear wear (i).

The algorithm of the methodology for determining the maximum allowable wear can be represented as follows (Fig. 3).

Degradation of the friction surface during dynamic operation leads to a violation of the kinematics of the rod-plunger system, causing the rod to deviate from the design axis of the trajectory, creating a pendulum stroke and shock loads. This

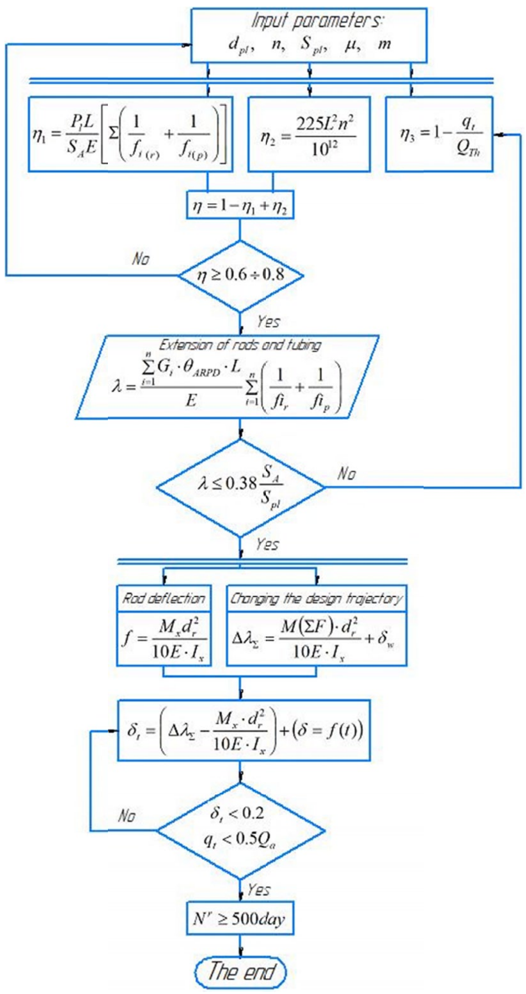


Fig. 3. Algorithm of the methodology for determining the maximum allowable wear

process of pump operation leads to an uneven distribution of friction forces and stress concentration, which accelerates mechanical and fatigue wear (wear) of the hardened surface layer of the part, increases leaks through gaps and, as a result, reduces pump delivery, which is characterized by its capacity.

As can be seen from the presented dependencies (27), the pump delivery and its resource life are determined by the duration of the wear process and are characterized by the size of the operating gap, which in turn will depend on the violation of the specified design trajectory relative to the axis of symmetry of the elements, the physico-mechanical properties of the hardened inner surface of the cylinder.

5. Discussion of the results of the study of load redistribution from changes in the design trajectory of the rod

The main cycle of the rod column is represented by a reciprocating motion. It can be described by the wave equation of longitudinally elastic vibrations of the rod, taking into account the distributed specific external load acting on the rods. The relationship that describes this movement is derived from the Lame equation. At the same time, the movement of the suspension point of the rod column and, accordingly, the movement of the plunger is carried out according to a harmonic law, while at the initial moment of time the polished rod is in the lower position. The specific external force (f) acting on the rod column consists of the gravity of the pumping rods in the liquid (f_g), the viscous friction force of the rods against the liquid (f_h) and the force of the boundary friction of the rods against the tubing walls (f_b): $f = f_g + f_h + f_b$ [21].

The conducted research allowed us to systematize the essence of the power balance of the rod-plunger pump system (28). To do this, we accept the following conditions: 1) the pump cylinder is rigidly fixed in the tubing column; 2) tubing is not compressed or stretched.

$$\begin{cases} G(t) = p_{wel.p.}S_{pl} + \rho_{mix} g S_{pl} L_r \cos\varphi - \\ \qquad \dfrac{1}{4A\nu}\left(1.65\dfrac{d_{pl}}{\delta} - 127\right)\dfrac{\partial u(L,t)}{\partial t} - p_{l.pl}S_{pl}, \\ f = \dfrac{(\rho_r - \rho_{mix})}{\rho_r} g\cos\varphi - \dfrac{16\mu P}{\rho_r(d_{tubing} - d_r)S_r}\cdot \\ \qquad \left(\dfrac{\partial u}{\partial t} + \upsilon\right) - \dfrac{f_{fr}\cdot\varphi}{8A\nu}\cdot\dfrac{\partial u}{\partial t} g \end{cases} \tag{28}$$

Equation (1) in system (28) represents the resulting time variable force $G(t)$ acting on the plunger. This force is applied to the lower end of the rod column and acts on the plunger during the entire pump cycle. The terms of this equation are as follows [22]:

– the strength of the wellhead pressure – $F_{wel.p.} = p_{wel.p.}S_{pl}$ (constant, acts on the plunger and is always directed downwards), N;
– gravity of the gas-liquid mixture – $G_{mix} = \rho_{mix} g S_{pl} L_r \cos\varphi$ (constant, acting on the plunger, pointing downwards), N;
– the friction force in the plunger pair – $F_{fr.pl} = -\dfrac{1}{4A\nu}\left(1.65\dfrac{d_{pl}}{\delta} - 127\right)\dfrac{\partial u(L,t)}{\partial t}$ (the variable acting on the plunger is directed in the direction opposite to the movement of the plunger), N;
– the pressure force of the liquid in the under-plunger space – $F_{l.pl} = -p_{l.pl}S_{pl}$ (variable, pointing upwards), N.

Equation (2) in system (28) represents the resultant force acting on the rod column. The terms of this equation are as follows:

– gravity of the column of pumping rods in the liquid – G_r, N;
– viscous friction force – F_μ (the variable acting on the rods is directed in the direction opposite to the direction of movement of the rods), N;
– dry friction force – F_f (the variable acting on the rods is directed in the direction opposite to the direction of movement of the rods), N.

The effective length of the plunger movement in the pump is affected by the inertia force. When the column of rods moves downwards, the forces of inertia coincide with gravity, which causes the column to stretch. This results in an additional stroke of the plunger in the cylinder exceeding the actual stroke of the polished rod. Such stretching can increase the wear of the plunger and cylinder, as well as reduce the tightness of the valves, which in turn affects pump performance. To take these factors into account, it is necessary to carry out accurate calculations in order to optimize the operation of the equipment and increase its reliability.

The load from the action of vibrational forces is determined from the equality (29) [3]:

$$F_{vib} = P_l \cdot \mu_\partial \sqrt{\frac{S}{\lambda} K} \tag{29}$$

where: P_l – the load from the weight of the liquid column above the pump plunger, determined taking into account the dynamic level, N; μ_∂ – dynamic coefficient; λ – the amount of joint deformation of pipes and rods (10), m; K – the coefficient determined from the condition:

$$K = \begin{cases} 1.01 & \text{if} \quad d_{pl} < 44 \\ 1.05 & \text{if} \quad d_{pl} = 44 \\ 1.03 & \text{if} \quad d_{pl} > 44 \end{cases}$$

The resistance force in the discharge valve is determined by the formula (30) [3]:

$$F_{fr.val} = \frac{2.62 \cdot 10^{-3} k \cdot \mu \cdot S \cdot n \cdot d_p^2 \cdot \left(d_p^2 - d_o^2\right)}{d_o \cdot \rho_{mix}} \tag{30}$$

where: n – number of swings per minute, min^{-1}; S – stroke length, m; k – number of valves; d_o – diameter of the valve seat opening, m; d_p – pump diameter, m.

Based on the conducted studies, the loading scheme of the rod-plunger RDP system is significantly changing and will take the form shown in Fig. 4.

Of the variety of forces acting on the plunger-cylinder system, it is the friction force that has the most weight in the overall force system. Taking into account all the friction forces that depend on the operating conditions and the type of friction (dry, boundary, mixed and liquid) is a difficult tribotechnical task.

The degree of curvature of the well, the multiple links of the rod column, and the aggressive environment change the kinematics of movement of the pump's structural elements, creating additional resistance forces during friction of the contact surfaces. The inevitable wear and sticking of the ARPD change the geometric shapes of the parts, and under the action of a load, the technological axis of movement of the rod changes, that is, bending and twisting of the rod column.

When the same load is applied, but with its direction changed, the gravity of the equipment causes deformation and bending of the cylinder in the filler column, as shown in Fig. 5a, a change in the position of the plunger in the cylinder and its misalignment (Fig. 5b). And the continued operating load from the rocking chair leads to an increase in the forces of dry and hydraulic friction, their uneven distribution over the surface, changing the trajectory of the system, which leads to breakage.

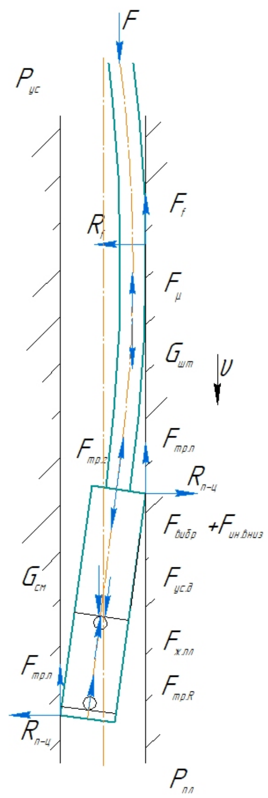
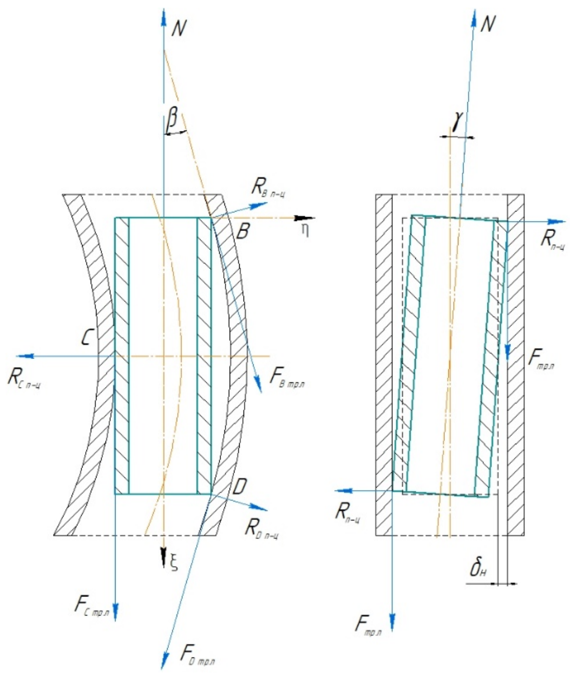


Fig. 4. Diagram of the forces acting on the pump elements during dynamic operation



a b

Fig. 5. Calculation scheme for determining the longitudinal friction force between the plunger-cylinder pair and the straightening moment: a – deformation and bending of the cylinder in the filler column; b – jamming of the plunger

With a significant viscosity (μ > 0.03 Pa·s) of an oil liquid, the forces of liquid friction can reach 15-20% of the total double amplitude of the loads. For example, during research in the field of tribotechnics, it was found that the friction force of rods against tubing for normal operating conditions reaches 2% of the total weight of the rods and, with increasing viscosity of the extracted medium, can exceed 1 kN. The friction force in the plunger-cylinder pair is much higher and can be on the order of 3 kN. This force will depend on the diameter of the plunger, the viscosity of the produced oil, the gap between the plunger and the cylinder, the amplitude and frequency of vibrations of the system, and the amount of mechanical impurities [23,24].

For wells with a curved trajectory, bending and intensive wear of the rod pump are possible, a new refined dependence for determining the friction force has been established, taking into account the radius of curvature of each well section R [25]:

$$F_{fr.R} = 25000 \cdot D + 7.2 \cdot 10^{-3} \frac{\Delta IE}{R\sqrt{R\delta}} \tag{31}$$

where: D – pump diameter, m; ΔI – the difference of the moments of inertia of the plunger and cylinder section, m^4; E – Young's module, Pa; R – radius of curvature of the borehole section, m; δ – the gap between the plunger and the cylinder, m.

To determine the total friction force generated in the plunger pair, it is necessary to take into account the hydrodynamic friction that occurs during the flow of liquid in the gap of the plunger pair:

$$F_{fr.hyd.} = S_{pl} \cdot \left(\frac{r_{pl}P}{2} + \frac{P \cdot \left(R_c^2 - r_{pl}^2\right)}{4 \cdot r \cdot \ln\left(\frac{r_{pl}}{R_c}\right)} \pm v_{pl} \frac{\mu}{r_{pl} \cdot \ln\left(\frac{r_{pl}}{R_c}\right)} \right) \tag{32}$$

where: S_{pl} – the surface area of the plunger, m^2; r_{pl} – наружный радиус плунжера, м; R_c – внутренний радиус цилиндра, м; P – pressure differential of the liquid on the plunger; μ – dynamic viscosity of the liquid, Pa·c; v_{pl} – plunger movement speed, m/s.

Applying formulas (31) and (32) to determine the friction forces occurring in the plunger pair, the resulting time variable force $G'(t)$ acting on the plunger (equation 1 in dependence (28)) can be written as follows:

$$G'(t) = p_{wel.p.}S_{pl} + \rho_{mix} g S_{pl} L_r \cos\varphi - \frac{1}{4A\nu}\left(1.65\frac{d_{pl}}{\delta} - 127\right)\frac{\partial u(L,t)}{\partial t} - p_{l.pl}S_{pl} + 7.2 \cdot 10^{-3} \frac{\Delta IE}{R\sqrt{R\delta}} + F_{fr.hyd.} \tag{33}$$

The presented equation (33) makes it possible to more accurately model the complications in the operation of a rod pumping unit. Two types of malfunctions in the operation of the rod pump were investigated, namely, leaks in the suction valve and changes in the clearance in the plunger-cylinder system. To perform the study, a rod-mounted plug-in pump with an upper lock support – 73-НВ1Б-44-12-18. The main characteristics of the pump are presented in TABLE 3, the characteristics of the rocking machine are presented in TABLE 4, the physico-mechanical and physico-chemical characteristics of the Uzen oil field (TABLE 5).

It is established that with an increase in the gap from the wear of the plunger pair, the force of viscous friction decreases, at the same time, with an increase in the diameter of the pump, it increases. With increasing wear of the gap, the trajectory of the rod changes and the plunger has an oscillatory motion at different points of movement and risks jamming, increasing the pressure and moment of bending stresses. Fig. 6 shows a diagram of the

TABLE 3

Pump characteristics 73-НВ1Б-44-12-18

Pump characteristics	**Pump code**	**Conditional size**	**Plunger diameter, mm**	**Plunger working stroke, mm**	**Plunger length, mm**	**Cylinder diameter, mm**	**Cylinder length, mm**	**Clearance for 2 groups of fits, mm**	**Pressure, m**	**Diameter of tubing, mm**
Plug-in with mechanical fastening and upper lock support (anchors)	73-НВ1Б-44-18-12-2-И	44	$44.35^{\circ}_{-0.013}$-$44.45^{\circ}_{-0.013}$	1800	1200	$44.45^{+0,05}$	3900	0.025-0.088	2000	73

И – wear-resistant pumps with a mechanical impurity content of more than 1.3 g/l;
Depth of pump descent L – 1600 m;
The column of pumping rods is two-stage:
– for the upper stage: rod diameter d_{r1} = 22 mm and the length is 41%, or l_{r1} = 656 m;
– for the lower step: rod diameter d_{r2} = 19 mm and the length is 59%, or l_{r2} = 944 m.

TABLE 4

Characteristics of the rocking machine

The cipher of the rocking machine	**Indicators**			
	rated load (on the wellhead rod), kN	**nominal stroke length of the wellhead rod, m**	**number of balance beam swings per minute at 1500 rpm**	**debit, m^3/day**
СКД8-3.5-5600	80	3.5 (actual 1.5)	15	30

TABLE 5

Physico-mechanical and physico-chemical characteristics of the Uzen oil field

Properties of oil	**Designation**	**Unit of measurement**	**Values**
Density	ρ_l	kg/m^3	859
Kinematic viscosity at 20°C	ν	m^2/s	65.18·10^{-4}
Dynamic viscosity at 50°C	μ	mPa·s	39.4
Solidification temperature	t	°C	+40
Mass fraction of water	ω_{water}	%	до 1
Water cut		%	83
Concentration of chloride salts	C_m	mg/l	50-300
Mass fraction of mechanical impurities	$\omega_{m.im}$	%	0.05
Mass fraction of sulfur	ω_S	%	0.18
Mass fraction of paraffin	ω_{par}	%	20.7
Mass fraction of selikagel resins	ω_{resin}	%	16.6
Mass fraction of asphaltenes	ω_{asph}	%	2.88
Gas content in oil	Γ_0	m^3/m^3	111.89
Thermobaric conditions:			
Reservoir temperature	t_r	°C	82.09
Reservoir pressure	p_r	MPa	22.75/14.04
Oil saturation pressure with gas	P_{sat}	MPa	5.2 до 14.87
Average depth of productive horizons	H_{dep}	m	1464.4

forces and moments acting on the plunger pair when the plunger moves down (oil rise). Conventionally, we divide the working stroke of the plunger into three parts of its position (1 – up to the plunger space, 2 – plunger, 3 – above the plunger space).

When the plunger moves up from the bottom dead center, a vacuum is created in the pump cylinder, as a result of which the suction valve opens and the liquid fills the cylinder space under the plunger. At the same time, the liquid above the plunger exerts pressure on the discharge valve, sealing the space, and moves upward with the plunger (Fig. 6).

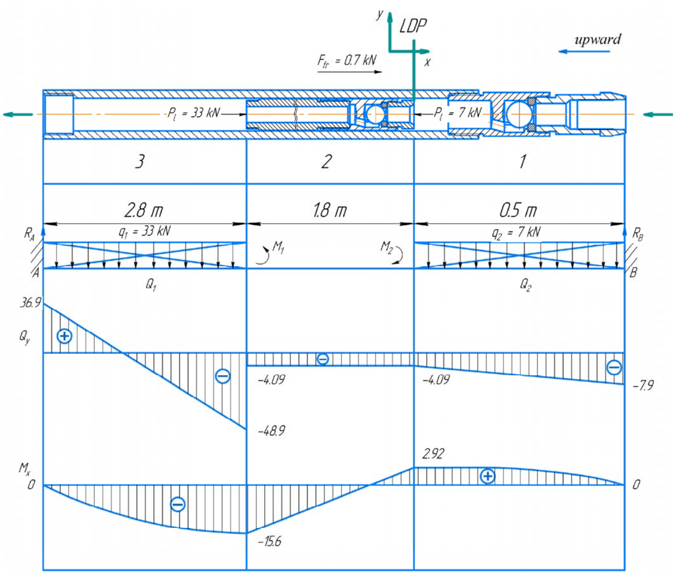


Fig. 6. Diagram of transverse forces and moments during upward movement of the plunger of the rod pump

During the downward stroke of the plunger, which begins when it reaches the upper dead center, the liquid filling the cylinder is compressed, the suction and discharge valves are closed and the liquid flows from under the plunger into the space above it. After a certain number of cycles, the tubing column is filled and the liquid begins to flow into the discharge line (Fig. 7).

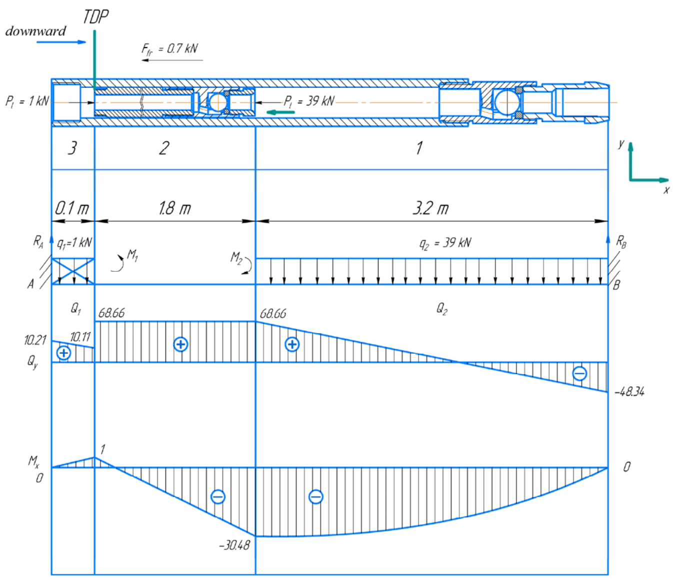


Fig. 7. Diagram of transverse forces and moments during downward movement of the plunger of the rod pump.

When the plunger is moved to the lowest position, the weight of the liquid column from above presses on the plunger, which leads to a maximum voltage in section 3, equal to Q_y = 48.9 kN. Fig. 6 shows the maximum load of 344 MPa at the same point. The plot (Fig. 7) shows the maximum torque M_x = 30.48 kN/cm^2 in this area. Thus, there is a dangerous phenomenon of uneven load redistribution at different time intervals, equal to the passage of the plunger from the conditional section 2 to section 3. At the moment when the plunger begins to move upward, a load Q_y = 48.9 kN is created in section 2 (Fig. 6). When the plunger is further moved at a constant pressure of the rocking rod, the plunger moves to section 3 and forms a load Q_y = 68.66 kN (Fig. 7), which is several times higher than the initial one. The increasing natural wear of the couplings during the dynamics of motion forms a deviation of the plunger from the design axis of symmetry ($\Delta\lambda_\Sigma$) and in the most loaded sections (3) there is an increase in the moment of forces and jamming or breaking of the rod under the condition $Q_y > E \cdot \Delta\lambda_\Sigma$. Where, when moving the plunger in the same cylinder with the same load from the rocking rod, inefficient operation of friction forces is observed. The largest displacement of 0.055 mm leads to an increase in the gap in the plunger pair, which, due to the deviation of the plunger from the design axis of symmetry ($\Delta\lambda_\Sigma$), leads to a decrease in the supply of the borehole pump, scuffing of the plunger pair and jamming of the pump. The installed displacement can be a corrective condition in the design of the pump for the selection of clearances and fit. To reduce the amount of leakage through the plunger pair and accurately predict the resource, the application of a new mathematical model will help [20,24,26,27].

Based on the simulation results, a graph was constructed of the dependence of the displacement of the plunger and the total force acting on it on time (Fig. 8) and a dynamogram when the plunger is jammed in the cylinder.

The simulation process was performed with a change in the size of the gap δ_t in the plunger pair (Fig. 9). Using the method of constructing a wellhead dynamogram, mathematical modeling was performed (Fig. 10) of the plunger operating conditions in case of leaks in the valve pair and changes in the gap in the plunger-cylinder system. The analysis shows that due to leaks in the valve pair, the pressure in the under-plunger space does not decrease so quickly, which leads to an increase in the load on the plunger pair.

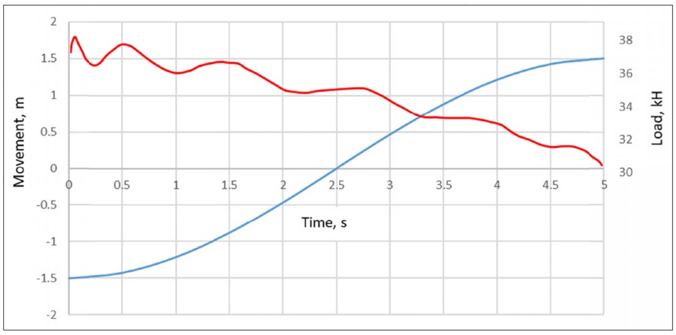


Fig. 8. The dependence of the plunger movement on the loads acting on it at each moment of time: 1 – the movement of the plunger; 2 – the load on the plunger

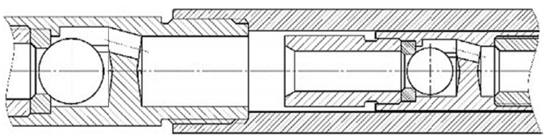


Fig. 9. Valve assembly

The structural elements of the valve assembly (Fig. 9) are studied as a single ball-seat system. During the operation of the pump, due to the difference in oil pressure at the inlet and outlet, as well as due to the landing speed of the locking element, shock loads act, deforming the landing surfaces of the valve seat and forming wear, distorting its design geometry. Mechanical impurities containing quartz sand wear out the surface, increasing the gap, and the presence of ARPD contributes to the build-up of layers, compaction and coking of deposits, reducing the nominal diameter and technological gap δ. The quartz sand particles present in the extracted liquid are quite solid. On the Mohs scale, the hardness of quartz sand corresponds to 7 degrees out of 10. This corresponds to a hardness of 60-70 HRC units on the Rockwell scale. Mechanical impurities move during the operation of pumping equipment and destroy the hardened surface.

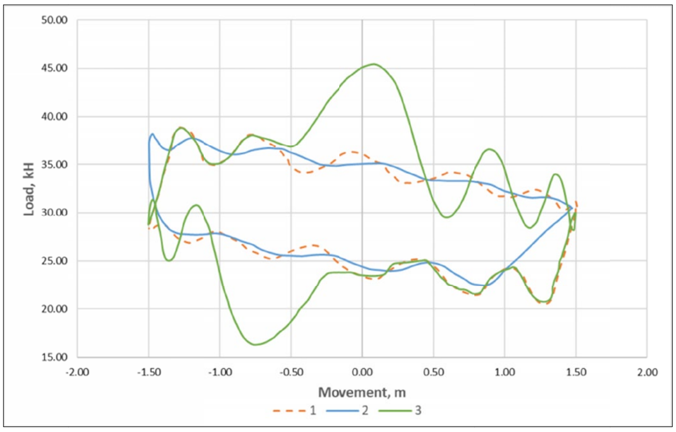


Fig. 10. Dynamogram of the load distribution from the movement of a worn plunger: 1 – theoretical (without complications), 2 – in case of leaks in the suction valve, 3 – change in the size of the gap in the plunger-cylinder system.

Of the greatest interest is the research analysis of the data array of changes in the size of the gap in the plunger-cylinder dynamogram system at different positions of the piston, such as displacement. The dynamogram shows the course of the plunger wear process under the influence of different loads in different positions of movement with the same pressure force. It is possible to interpret unambiguously only by understanding the pattern of parameter changes with a reliable dependence of the change in the variable gap in the plunger-cylinder system when the design trajectory of the rod and plunger deviate from the axis of wear symmetry. The effect of wear on the load distribution (Q) in the plunger-cylinder system, depending on the change in the displacement position (h) of the plunger when deviating from the design trajectory, can be justified by establishing equations using the least squares method, the reliability of which will be shown by the coefficients and indices of correlation and determination. Another important step will be to evaluate the significance of the parameters of the regression equation using Fisher's F-test, Student's t-test and the Darbin-Watson criterion.

TABLE 6 shows the auxiliary values of the cubic regression equation.

TABLE 6

Table of auxiliary quantities of the cubic regression equation $\hat{Q} = ah^3 + bh^2 + ch + d$

i	h_i	Q_i	h_i^2	h_i^3	h_i^4	h_i^5	h_i^6	h_iQ_i	$h_i^2Q_i$	$h_i^3Q_i$
1	0.043931159	45.33954727	0.0019	0.0001	0	0	0	1.9918	0.0875	0.0038
2	0.089300725	45.43275632	0.0080	0.0007	0.0001	0	0	4.0572	0.3623	0.0324
3	0.132149758	45.15312916	0.0175	0.0023	0.0003	0	0	5.9670	0.7885	0.1042
4	0.177519324	44.54727031	0.0315	0.0056	0.0010	0.0002	0	7.9080	1.4038	0.2492
5	0.230450483	43.52197071	0.0531	0.0122	0.0028	0.0006	0.0001	10.0297	2.3113	0.5326
6	0.275820048	41.89081225	0.0761	0.0210	0.0058	0.0016	0.0004	11.5543	3.1869	0.8790
7	0.311107488	40.25965379	0.0968	0.0301	0.0094	0.0029	0.0009	12.5251	3.8966	1.2123
$i_{n+1}\cdots$	…	…	…	…	…	…	…	…	…	…
$i_{n+2}\cdots$	…	…	…	…	…	…	…	…	…	…
48	1.455428744	28.93475366	2.1183	3.083	4.4871	6.5306	9.5049	42.1125	61.2917	89.2057
49	1.465510870	28.37549933	2.1477	3.1475	4.6127	6.76	9.9068	41.5846	60.9427	89.3122
50	1.483154589	28.18908123	2.1997	3.2626	4.8389	7.1768	10.6443	41.8088	62.0089	91.9687
51	1.488195652	28.88814913	2.2147	3.2959	4.9050	7.2996	10.8633	42.9912	63.9793	95.2138
52	1.495757246	29.96005326	2.2373	3.3464	5.0055	7.4870	11.1987	44.8130	67.0293	100.2596
Σ	**47.1927**	**1751.984**	**52.7664**	**64.3723**	**82.3284**	**108.46**	**145.8242**	**1511.2545**	**1661.9161**	**2010.5916**

Let's find the coefficients a, b, c and d of the cubic regression equation $\hat{Q} = ah^3 + bh^2 + ch + d$ from the system of equations:

$$\begin{cases} a\sum h_i^3 + b\sum h_i^2 + c\sum h_i + nd = \sum Q_i, \\ a\sum h_i^4 + b\sum h_i^3 + c\sum h_i^2 + d\sum h_i = \sum h_i Q_i, \\ a\sum h_i^5 + b\sum h_i^4 + c\sum h_i^3 + d\sum h_i^2 = \sum h_i^2 Q_i, \\ a\sum h_i^6 + b\sum h_i^5 + c\sum h_i^4 + d\sum h_i^3 = \sum h_i^3 Q_i; \end{cases} \tag{34}$$

$$\begin{cases} 64.372a + 52.766b + 47.192c + 52d = 1751.98, \\ 82.328a + 64.372b + 52.766c + 47.192d = 1511.25, \\ 108.46a + 82.328b + 64.372c + 52.766d = 1661.91, \\ 145.824a + 108.46b + 82.328c + 64.372d = 2010.59 \end{cases}$$

$$\Delta = \begin{vmatrix} 64.372 & 52.766 & 47.192 & 52 \\ 82.328 & 64.372 & 52.766 & 47.192 \\ 108.46 & 82.328 & 64.372 & 52.766 \\ 145.824 & 108.46 & 82.328 & 64.372 \end{vmatrix} = 151.0452 \tag{35}$$

$$\Delta a = \begin{vmatrix} 1751.98 & 52.766 & 47.192 & 52 \\ 1511.25 & 64.372 & 52.766 & 47.192 \\ 1661.91 & 82.328 & 64.372 & 52.766 \\ 2010.59 & 108.46 & 82.328 & 64.372 \end{vmatrix} =$$

$$= -3313.6883 \Rightarrow a = \frac{\Delta a}{\Delta} \approx -21.9384 \tag{36}$$

$$\Delta b = \begin{vmatrix} 64.372 & 1751.98 & 47.192 & 52 \\ 82.328 & 1511.25 & 52.766 & 47.192 \\ 108.46 & 1661.91 & 64.372 & 52.766 \\ 145.824 & 2010.59 & 82.328 & 64.372 \end{vmatrix} =$$

$$= 9564.3344 \Rightarrow b = \frac{\Delta b}{\Delta} \approx 63.321 \tag{37}$$

$$\Delta c = \begin{vmatrix} 64.372 & 52.766 & 1751.98 & 52 \\ 82.328 & 64.372 & 1511.25 & 47.192 \\ 108.46 & 82.328 & 1661.91 & 52.766 \\ 145.824 & 108.46 & 2010.59 & 64.372 \end{vmatrix} =$$

$$= -9091.1073 \Rightarrow c = \frac{\Delta c}{\Delta} \approx -60.188 \tag{38}$$

$$\Delta d = \begin{vmatrix} 64.372 & 52.766 & 47.192 & 1751.98 \\ 82.328 & 64.372 & 52.766 & 1511.25 \\ 108.46 & 82.328 & 64.372 & 1661.91 \\ 145.82 & 108.46 & 82.328 & 2010.59 \end{vmatrix} =$$

$$= 7736.4788 \Rightarrow d = \frac{\Delta d}{\Delta} \approx 51.2196 \tag{39}$$

Then the equation under study for the dependence of the load distribution in the plunger-cylinder system on the movement of the plunger in different parts of the cylinder when deviating from the design trajectory of the rod movement will take the form (40):

$$\hat{Q} = -21.9384h^3 + 63.321h^2 - 60.188h + 51.2196 \tag{40}$$

Therefore, the scattering diagram and the graph of the regression equation are constructed as follows (Fig. 11).

To evaluate the significance of the regression and correlation parameters, first we find the Q average:

$$\bar{Q} = \frac{1}{n}\sum Q_i = \frac{1751.98}{52} = 33.692$$

Next, we will compile a table of auxiliary quantities, where (TABLE 7)

$$\varepsilon_i = Q_i - \widehat{Q}_i, \Delta\varepsilon_i = \varepsilon_i - \varepsilon_{i-1}, A_i = \left|\frac{Q_i - \widehat{Q}_i}{Q_i}\right|$$

TABLE 7

Formation and calculation of auxiliary quantities

i	h_i	Q_i	$\widehat{Q}_i$	$Q_i-\overline{Q}_i$	$(Q_i-\overline{Q}_i)^2$	ε_i	ε_i^2	A_i	$\Delta\varepsilon_i$	$(\Delta\varepsilon_i)^2$
1	0.043931159	45.33954727	48.6958	11.6475	135.6653	−3.3563	11.2646	0.0740	—	—
2	0.089300725	45.43275632	46.3341	11.7408	137.8453	−0.9014	0.8125	0.0198	2.4549	6.0266
3	0.132149758	45.15312916	44.3210	11.4611	131.3575	0.8322	0.6925	0.0184	1.7335	3.0051
4	0.177519324	44.54727031	42.4078	10.8553	117.8369	2.1395	4.5773	0.0480	1.3073	1.7091
5	0.230450483	43.52197071	40.4436	9.8300	96.6283	3.0784	9.4765	0.0707	0.9389	0.8816
6	0.275820048	41.89081225	38.9755	8.1988	67.2205	2.9153	8.4992	0.0696	−0.1630	0.0266
7	0.311107488	40.25965379	37.9628	6.5677	43.1341	2.2969	5.2756	0.0571	−0.6185	0.3825
$i_{n+1}\cdots$	…	…	…	…	…	…	…	…	…	…
$i_{n+2}\cdots$	…	…	…	…	…	…	…	…	…	…
48	1.455428744	28.93475366	30.1155	−4.7572	22.6314	−1.1808	1.3942	0.0408	−0.9116	0.8310
49	1.465510870	28.37549933	29.9581	−5.3165	28.2652	−1.5826	2.5046	0.0558	−0.4018	0.1615
50	1.483154589	28.18908123	29.6663	−5.5029	30.2821	−1.4772	2.1822	0.0524	0.1054	0.0111
51	1.488195652	28.88814913	29.5791	−4.8039	23.0770	−0.6909	0.4774	0.0239	0.7863	0.6183
52	1.495757246	29.96005326	29.4449	−3.7319	13.9274	0.5152	0.2654	0.0172	1.2061	1.4546
Σ	—	—	—	—	**1189.3376**	—	**315.5445**	**3.3723**	—	**47.0999**

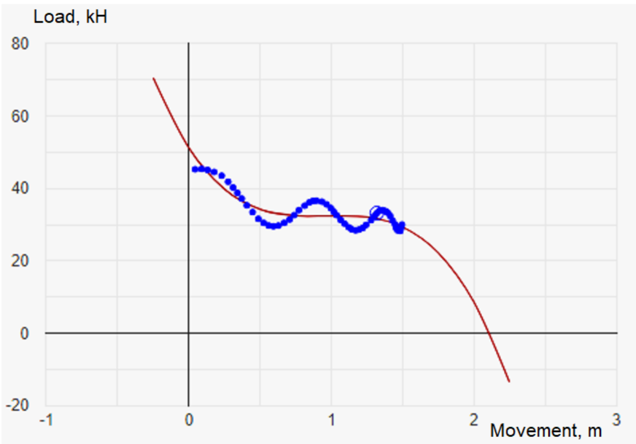


Fig. 11. Graph of the dependence of the load distribution in the plunger-cylinder system on the movement of the plunger in different parts of the cylinder when deviating from the design trajectory

Based on the data obtained, we will determine the correlation index:

$$R=\sqrt{1-\frac{\sum\left(Q_i-\widehat{Q}_i\right)^2}{\sum\left(Q_i-\overline{Q}_i\right)^2}}=\sqrt{1-\frac{315.5445}{1189.3376}}\approx 0.8571 \qquad (41)$$

The index of determination is equal to $R^2=0.8571^2\approx 0.7347$. The average approximation error is defined as:

$$\overline{A}_i=\frac{1}{n}\sum\left|\frac{Q_i-\widehat{Q}_i}{Q_i}\right|\cdot 100\%=\frac{3.3723}{52}\cdot 100\%\approx 6.4852\%$$

Let's define the Fischer criterion-F. Critical (tabular) $F_{tabl}=F(a,k_1,k_2)=F(0.05,3,48)\approx 2.7981$. The actual criterion:

$$F_{fakt}=\frac{R^2}{1-R^2}\cdot\frac{k_2}{k_1}=\frac{0.7347}{1-0.7347}\cdot\frac{48}{3}\approx 44.3065$$

Since $k_1=m=3$, $k_2=n-m-1=52-3-1=48$ and $a=0.05$, where m is the number of parameters for the variables of the regression equation.

Next, we define the Darbin-Watson Criteria:

$$d=\frac{\sum\left(\varepsilon_i-\varepsilon_{i-1}\right)^2}{\sum\varepsilon_i^2}=\frac{47.0999}{315.5445}\approx 0.1493$$

As the gap in the cylinder-plunger pair decreases, the friction force between the plunger and the cylinder increases dramatically. In this regard, the movement of the plunger in the cylinder will be uneven, which will be characterized by deceleration and acceleration processes. This is shown on the dynamogram (Fig. 10) as a sharp load jump in the middle of the plunger's upward movement and a similar increase in forces acting on the plunger and, as a result, on the polished rod, in the middle of the plunger's downward movement. The same pattern is recorded in the curve of the load distribution graph in the plunger-cylinder system from the movement of the worn plunger in different parts of the cylinder when deviating from the design trajectory (Fig. 11). Consequently, at different time intervals, with the same external load, the pump parts in a certain position have different critical stress values and wear intensity.

With an abrupt increase in load, an intensive process of wear of the cylinder-plunger system and the valve pair occurs. It is at this point that failure is most likely. Thus, the studied dynamics of the plunger movement in synchronous operation of the valve showed that an uneven stress redistribution occurs due to a slight change in the design trajectory of the worn plunger, the boundary conditions of wear and durability change, and consequently the efficiency of the pump and the energy efficiency of the ground rocking machine.

6. Conclusions

For the first time, the mechanism of wear of oil production pump couplings has been investigated as a result of a violation of the design trajectory of kinematic connections from a given axis of symmetry. The critical values of the acting moments and stresses in case of inefficient load redistribution across the cylinder at different positions of the worn plunger are substantiated and presented.

Using the Cauchy m parameter model, an estimate of the effect of dynamic loads on the actual supply of the pumping unit is given fairly accurately, taking into account the wear of kinematic connections. The Lame equation has been improved by describing the law of reciprocating motion of a plunger with uneven wear as the equation of longitudinally elastic vibrations of a rod, taking into account the distributed specific external load acting on the rods, which distorts the design trajectory of the kinematic pairs. The process of interaction of elements of the rod-plunger-cylinder system of an oil production pump with uneven axial moments is investigated. The authors proposed a new term for the deviation of the design trajectory ($\Delta\lambda$) of the rod and plunger relative to the axis of symmetry of the cylinder.

To implement technological solutions to increase the service life of the rod pump, the criteria and conditions of failure ($\delta_t > 0.2$ mm, $q_t > 0.5Q_a$ m^3/day) are justified, reducing the efficiency of its operation when exposed to dynamic resistance forces on the wear elements of the plunger-cylinder system.

The mathematical model of pump performance (Q_a) and its resource life has been improved when changing the maximum allowable gap $\delta_t = f(i)$ in the plunger-cylinder system of a rod pump and leaks (q_t) due to violations of the kinematics of motion ($\Delta\lambda_\Sigma$) from dynamic loads, physico-mechanical properties of the hardened inner surface of the cylinder, which It allows for a significant adjustment of the calculation of the effective moments of forces and contact pressure in case of deviation of the design axis of symmetry as a result of actual wear of the contact surfaces of the rod-plunger-cylinder system.

The durability of the pump is due to the complex process of moving the plunger and the dynamics of load redistribution along the contact surface of the plunger-cylinder and ball-saddle systems when the design trajectory of the kinematic connections deviates from a given axis of symmetry.

For the first time, the dependence of the load distribution in the plunger-cylinder system on the movement of a worn plunger in different parts of the cylinder when it deviates from the design trajectory of the pump rod has been established.

The developed dynamogram for determining the dependence of the load distribution on the movement of a worn plunger during basic contact stress cycles makes it possible to predict the pump life depending on the wear of the parts. This dynamogram makes it possible to clearly establish the range of optimal pump performance values under different loading conditions, ensuring control of the efficiency of load redistribution along the contact surfaces of kinematic coupling pairs.

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