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INVESTIGATION OF THE DURABILITY PARAMETERS AND JUSTIFICATION OF THE EFFECTIVENESS OF THE NEW MODEL OF THE VALVE ASSEMBLY OF THE ROD DEPTH PUMP

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Abstract. *The scientific article is devoted to the study of the criterion parameters of the durability of the rod depth pump and the determination of the dependencies of the effective operation of the valve on the structural and material characteristics of the ball-saddle pair. The purpose of the study is to substantiate the possibility of using modern structural materials, new structures with elastic-damping properties of the valve while increasing its tightness. To solve the problems of valve wear and increase the maintenance period, the choice of a synthetic polymer for the outer surface of the locking device from the group of organic synthetic polymers (7B-14MA TS 38-105-1082-86) with a ceramic filler made of zirconium dioxide (ZrO_2) is justified. This development will provide damping and an increase in the width of the contact belt of the conditional meridian of the ball for effective redistribution of the shock load. The proposed ceramic filler made of zirconium dioxide will reduce the wear on the surface of the locking device. Mathematical simulation and bench tests have confirmed the validity of the choice of method and materials for the developed combined ball design, which consists of a steel core gummed with a rubber compound with a zirconium dioxide filler. The use of a combined rubberized ball with filler increases the fatigue strength of the valve seat by more than 30%.*

Keywords: rod depth pump, valve pair, wear, resource durability, combined materials.

1. Introduction

The oil and gas industry plays an important role in the development of the economy of any dynamically developing country. According to experts, the stock of wells at a late stage of development (with a flow rate of less than 40 m^3/day) currently accounts for more than 50% of the total number of operating wells. This is due to a decrease in light oil reserves due to its intensive production and depletion of existing fields [1-2].

As oil fields are developed, the operating conditions of pumping complexes deteriorate, and accordingly, the requirements for their reliability and durability become stricter. Consequently, companies face the challenge of developing and applying modern energy-efficient extractive pumping systems with increased durability. Today, approximately 70% of existing oil wells in the world are operated by downhole rod pumps (DRP). Despite the high failure rate of rod installations, the massive scale of their operation and extensive functionality confirm the relevance of research and development work on the development of new structural and technological solutions in order to increase the service life of rod depth pumps [3-5].

The scientific and technical problem lies in the fact that the pumping unit is put into operation at the set factory (ideal) parameters. Since the beginning of operation of oil-producing pumps, their technical characteristics change and over time do not correspond to the nominal values of the manufacturer. The wear of the main pump parts, as a result of the aggressive action of the medium and dynamic loads, generally reduces the service life of mining plants. Preliminary studies have allowed us to form a group of parts operating with a high degree of loading (plunger-cylinder) and increased intensity of cyclic operation (valve assembly). As a result, there is a need to develop fundamentally new design and technological solutions to improve performance and resource durability. To solve the tasks set, a systematic research approach and digital methods (CAD/CAM/CAE) should be used to substantiate the optimal pump characteristics with a high confidence probability.

2. Materials and Methods

The analysis of the factors and causes of the failure of rod depth pumps for 2017-2024 under various operating conditions allowed us to form the main types of defects of the RDP (Figure 1) [6-12]. It can be seen from the diagram (Figure 1) that the main factors reducing the durability and reliability of RDP are the presence of mechanical impurities, asphalt-resin-paraffin deposits (ARPD) and corrosive environments in the extracted petroleum products. These figures are 27, 24 and 19%, respectively.

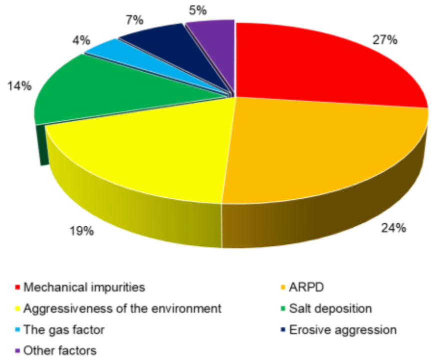


Fig.1. Factors determining the causes of wear of elements of rod depth pumps

It has been established that the main components of the RDP, which are subject to intense wear, include a fixed cylinder, a plunger, a rod, a valve assembly (suction and discharge) and a locking support (for plug-in pumps). The increased intensity (Figure 1) of abrasive-mechanical and corrosive wear of the coupled pump elements leads to a change in the design geometry, a decrease in the physical and mechanical properties of the working surface and a deviation from the design trajectory of the axis of movement of the rod and plunger of the pump. The combination of these parameters leads to an increase in the cyclically changing dynamic shock load, which causes failures (stopping and jamming) of the extraction pumps. The dynamics of the distribution of failures across the structural elements of the RDP is shown in the diagram (Figure 2) [5, 12-14].

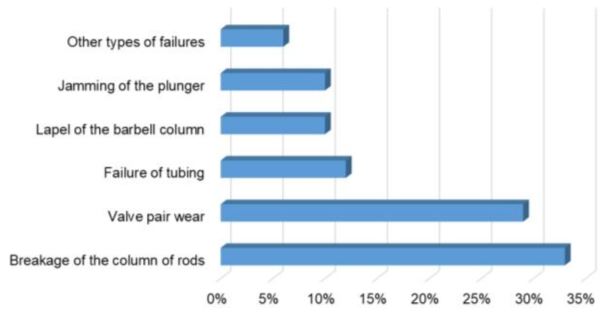


Fig.2. Distribution of the failure rate by the structural elements of the RDP

Valve sticking and ulcerative corrosion of the ball due to the presence of carbon dioxide in the extracted liquid significantly reduce the tightness and performance of the pump. In addition, the wear of the saddle in the ball-saddle pair (about 25%) is also a critical factor, since even minor damage leads to leaks and inefficient operation of the equipment. To solve the scientific and practical problem of determining the dependencies of RDP failures on changes in its dynamometric cyclogram during the wear of highly loaded couplings, an in-depth study of the nodes using 3D modeling is required. It is possible to increase the durability of the valve pair by using new manufacturing technologies and a multicomponent material with damping properties. To date, insufficient attention has been paid to the technical and operational significance of the valve mechanism. Operating under large alternating loads and fluid pressure, the valve mechanism (Figure 3) generates the volume of the intake oil column, and the condition and performance characteristics of the valve affect the volume of leaks between the coupled elements.

The valve assembly is being investigated as a single ball-seat system. Due to the extreme operating conditions of the ball-saddle system, strict requirements are placed on its reliability and durability. Quartz sand is known to be the main component of mechanical impurities. On the Mohs scale, the hardness of quartz sand particles corresponds to 7 degrees out of 10, which on the Rockwell scale is 60÷70 HRC [15]. Mechanical impurities during the operation of pumping equipment enter the contact zone of the ball and the valve seat, which leads to premature wear and destruction of the hardened surface of the structural elements. The presence of ARPD in the extracted liquid contributes to the build-up, pressing and coking of deposits, reducing the nominal diameter of the bore and the technological gap δ. The effect of alternating loads, cyclically repeated impacts of the ball on the seat contributes to the formation of microcracks and depressurization of the ball seat. Also, complicated operating conditions lead to additional sliding resistance forces of the mating surfaces, giving a singularity to the wear process. All of the above factors lead to rapid wear of the valve assembly, and as a result, to a decrease in the maintenance period and, subsequently, to the failure of the RDP [4]. Based on the research results of domestic and foreign scientists, an understanding of the process of ball deformation and seat discoloration due to friction and internal stresses during shock loads of the ball and seat has been formed.

Due to the configuration of the working surfaces of the ball and the seat, when examining wear, the mating surfaces should include a part of the edge (small area) of the seat with the conditional meridian of the contacting surface of the ball. A sphere inscribed in a cone forms a conditional meridian along the tangent line (Figure 4). With linear sealing, any discrepancy with the projected geometric shapes increases the size of the gap between the contacting elements, which leads to increased leaks of the pumped reservoir fluid [5, 15, 16]. Thus, the distribution of the current cyclic loads is concentrated on a small unit of contact area per unit time.

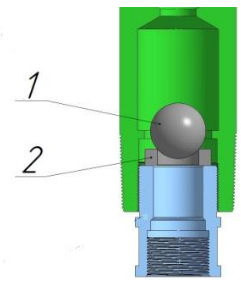
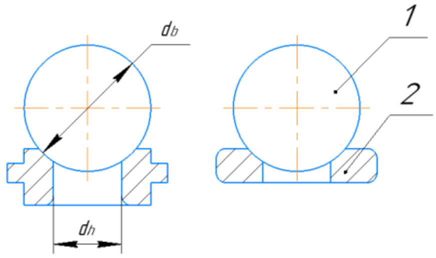


Fig.3. 3D modeling of the valve assembly: 1 – ball, 2 – valve seat

Fig.4. The contact field of the ball-seat valve pair elements

Dolov T.R., Ivanovsky V.N. and other researchers based on a mathematical model and bench tests proved that the ratio of hardness of the ball and the valve seat should be in the range of 1.01÷1.05 [5]. Thus, researchers prioritize the hardness of the ball-seat surfaces, which does not fully describe the effectiveness of the valve opening and closing process. According to GOST 31835-2012, the ball-seat valve pair of the pump is manufactured in two versions (Figure 5). The optimal value of the ratio of the diameter of the seat hole d_h to the diameter of the ball d_b is 0.865. This ratio plays an important role in optimizing the operation of the valve assembly, ensuring effective interaction between the seat and the ball, minimal hydraulic resistance of the fluid and increasing its operability [5, 17].



a) b)

Fig.5. Ball-seat valve pair: 1 – ball; 2 – seat; a – valve with a collar (VC); b –valve with a cylindrical seat (V)

The main materials used to make the valve pair are: for the ball – 95Cr18 steel (high carbon stainless steel with a hardness of 65 HRC) and for the seat – 30Cr13 and 95Cr18 (stainless steel with a hardness of 45 HRC). A selection of hard alloys such as tungsten carbide, titanium carbide and cobalt alloys provides high wear resistance. The use of stellites – alloys of cobalt, nickel and chromium, ceramets improve the performance and reliability of valves. These innovations help to reduce leaks and extend the service life of pumps [17, 18].

The problems of improving the efficiency of RDP were solved by domestic and foreign scientists. The main disadvantages of the known designs are an increase in the cost of the pumping unit due to the inclusion of additional elements in the design, for example, spring mechanisms, and a decrease in the reliability of equipment due to an increase in the number of parts, especially small ones.

It is proposed to increase the efficiency and service life of the RDP by increasing the wear resistance of the contacting elements of the parts, without changing the integrity of its structure. The operability of the valve assembly design, based on standard requirements, depends on maintaining the criteria parameters [5, 15, 18]:

- the mass of the ball should be less than the mass of the saddle (reducing the weight of the ball by reducing the diameter or replacing the material);
- the strength characteristics of the saddle must exceed the parameters of the ball to prevent the saddle from crumpling from the action of cyclic impacts of the ball;
- the hardness of the ball must be higher than the hardness of the saddle in order for the ball to retain its original shape and condition.

Therefore, the main focus of reducing operating costs is the introduction of innovative ways to reduce wear on the valve assembly. Since the operability of the valve assembly during operation is affected not only by the physico-mechanical properties of the mating surfaces, but also by the kinematic features (moment, forces, mass and velocity, gap overlap) of the locking element, it is theoretically assumed that in order to ensure high durability of the valve mechanism, it is necessary to ensure uniform redistribution in conditions of small contact areas a cyclically varying shock load at the moment the valve is closed.

The scientific and technical problem is proposed to be solved by creating an elastic valve closure effect through the use of materials with damping properties. To reduce valve wear and increase the maintenance period, it is proposed to use a coating of a material from a group of organic synthetic polymers with a ceramic filler made of zirconium dioxide (ZrO_2) for the outer surface of the locking device [19-21]. A reasonable choice of synthetic material applied to the valve ball will solve the problem of damping and increasing the width of the contact belt of the conditional meridian of the ball to effectively redistribute the shock load, and the use of a ceramic filler made of zirconium dioxide will increase the wear of the surface of the locking device.

The analysis of defective certificates and repair works of Mangystaumunaigas JSC (Kazakhstan) shows that the main wear is observed on the valve seat, where a harder ball creates an annular spot, which gradually increases due to deepening. The valve seat is made of 40Cr steel (GOST 4543-71) with normalization up to 45÷53 HRC. The locking ball is made of high-strength specialized steel type BCr15 (bearing structural steel) (GOST 801-2022) with a hardness of 60÷65 HRC. Therefore, it is necessary to conduct a theoretical study to substantiate the optimal kinematic parameters of a valve with different materials of the shut-off ball.

To evaluate the stress-strain state (SSS) of the valve pair elements during the seating of the closure member and to predict its operational durability, numerical modeling was performed using the Finite Element Method (FEM). The calculations were carried out in the COMPASS 3D computer-aided design environment using the integrated structural and fatigue analysis module APM FEM (Research and Development Center "APM"). The problem was solved in a three-dimensional quasi-static formulation (linear static analysis), where dynamic factors were converted into equivalent static loads. The structural configuration and boundary conditions of the numerical model were defined as follows:

– Seat Constraints: The lower and external cylindrical faces of the valve seat were rigidly fixed in all directions (full displacement constraint: $U_x = U_y = U_z = 0$), simulating its press-fit installation into the pump housing.

– Ball Degrees of Freedom: To ensure numerical stability of the contact interface and eliminate any unconstrained rigid-body motion (loss of stability), the ball was restricted from moving along the horizontal axes X and Y ($U_x = U_y$). Displacement along the vertical working axis Z (the valve's axis of symmetry) was left free to absorb operational loads.

– Contact Interaction: A non-linear, one-way "Sliding" interaction type was specified at the mating surfaces between the ball sphere and the seat cone. This contact condition allows mutual sliding and microdisplacements of the surfaces without structural penetration of the bodies, thereby capturing the true distribution of contact pressure.

A combination of external forces simulating the final phase of valve closure was simultaneously applied to the numerical model:

1 Resultant differential pressure (ΔP = 7.6 MPa): Applied normally to the upper geometric segment of the ball bounded by the actual circular seating contact line located below the equator. This consolidated parameter accounts for the combination of downhole and operational factors acting at the moment of valve closure: the wellhead discharge pressure, the hydrostatic pressure of the fluid column at the pump setting depth, the hydrodynamic water hammer pressure generated by the instantaneous stoppage of the flow upon valve closure, and the static reservoir pressure. Simultaneously, the structural elements of the valve seat were subjected to their respective downhole pressures: the upper face of the seat was exposed to the combined downward pressure of 20.78 MPa, while the inner cylindrical surface of the orifice (d_h=35 mm) experienced the outward radial reservoir pressure of 14.04 MPa.

2 Total equivalent vertical force (F_{tot}): Applied as a body force to the center of mass of the closure member, directed vertically downward along the valve axis of symmetry. This parameter combines the dead weight of the component and the dynamic inertial force generated during its seating deceleration. Depending on the design variant and material density, the total force is set to 6.03 N for the solid steel ball and 3.55 N for the rubberized ball with filler.

To assess the operational reliability of the ball-seat pair under cyclic loading, a fatigue life calculation was conducted within the APM FEM module:

– Loading Regime: An asymmetric loading cycle was defined to match the pulsating nature of the oil pump's operation (stress ratio R=0, where the lower limit of the cycle corresponds to the open valve state and the upper limit corresponds to the moment of hard seating under pressure).

– Fatigue Base: The calculation was performed for a standard base number of loading cycles $N = 2.5 \cdot 10^6$, which defines the limit of finite life for heavily loaded components operating under contact and crushing stresses.

– Fatigue Strength Parameters: The physical and mechanical parameters of the fatigue curves (including the endurance limits under symmetrical bending σ_{-1} and torsion τ_{-1}) for the investigated steel grades (Steel BCr15, Steel 40Cr and Steel 20) were retrieved automatically from the integrated standard reference database of the APM FEM software. This eliminated subjective interpolation errors associated with experimental Wöhler curves.

The physical and mechanical properties of the materials used are shown in Table 1.

Table 1. Physical and Mechanical Properties of Materials

Structural Element	Material / Regulatory Document	Elastic Modulus E, MPa	Poisson's Ratio, ν	Density ρ, kg/m³	Yield Strength, σ_y, MPa	Ultimate Tensile Strength, σ_{UTS}, MPa
Seat	Steel 40Cr / GOST 4543-71	$2.0 \cdot 10^5$	0.30	7820	~800	~1000
Metal ball	Steel BCr15 / GOST 801-2022	$2.1 \cdot 10^5$	0.30	7812	1100 – 1200	1400
Core	Steel 20 / GOST 1050-2013	$2.1 \cdot 10^5$	0.29	7860	250	410
Coating 1	Combined material: rubber compound 7B-14MA and PCRK-3 (20%) / TS 38-105-1082-86 and GOST 21907-76	30 ÷ 50*	0.45	~1621	–	10 ÷ 15*
Coating 2	Combined material: polyurethane brands Adiprene L167 and PCRK-3 (10%) / TS 38.103-137-78 and GOST 21907-76	500 ÷ 2000*	0.45	~1368	~10 – 40	~15 – 55*

** Note to Table 1: Values for the rubber compound are specified conditionally within a linear-elastic framework. The actual values of the elastic modulus E and ultimate tensile strength σ_{UTS} are subject to verification against the physical testing certificate of the specific rubber batch.*

The geometry was discretized using an adaptive method with a mesh composed of four-node rigid tetrahedral elements:

– the baseline mesh size for both the ball and the seat bodies was set to 2.0 mm;

– local mesh refinement (densification) was implemented in the immediate area of mechanical contact (on the conical chamfer of the seat and the mating zone of the sphere). The finite element size in the contact zone was reduced to 1.0 mm to ensure accurate approximation of the Hertzian contact stress gradient.

When interpreting the results of the numerical experiment, the fundamental limitations of the built-in APM FEM solver were taken into account:

– Linear Elasticity of Media: Material behavior is described by Hooke's generalized law. The module does not account for physical non-linearity (plastic deformation of metals during local crushing) or the hyperelastic behavior of elastomers (models of Mooney-Rivlin, Ogden and others). Stresses exceeding the yield strength σ_y are treated as potential initiation zones for plastic deformation.

– Quasi-Static Load Application: Transient dynamic wave processes occurring in the metal during the microsecond duration of the impact were not modeled. The load was applied instantaneously as a steady-state static system of forces.

– Absence of Lubrication Film: The calculation assumes dry contact conditions between the surfaces. The damping and hydrodynamic effects of the fluid film (crude oil) directly within the contact gap were omitted, which provides a safety margin for structural strength.

3. Results and discussion

To maintain the lowest possible level of liquid leakage through the valve pair, it is necessary to ensure that the shut-off element fits on the valve seat in a timely manner. In the works of Pirverdyan A.M. and Greifer V.I., the dependence of the landing speed of the locking element on the valve seat was established. At the same time, the landing speed of the locking element must be at least 0.1 m/s. This dependency has the following form [15, 22]:

$$v = \frac{V_{l.e.} \cdot g \cdot (\rho_{l.e.} - \rho_l)}{6\pi \cdot r \cdot \mu}, \tag{1}$$

where v – the landing speed of the locking element on the valve seat, m/s; $V_{l.e.}$ – volume of the locking element, m^3; g – acceleration of free fall, g=9.81 m/s^2; $\rho_{l.e.}$ – material density of the locking element, kg/m^3; ρ_l – the density of the pumped liquid, kg/m^3; r – radius of the locking element, m; μ – dynamic viscosity of the liquid, mPa·s.

Substituting the expression for the volume of the ball into formula (1), we obtain:

$$v = \frac{2r^2 g \cdot (\rho_{l.e.} - \rho_l)}{9\mu}. \tag{2}$$

We accept oil with a density of 904 kg/m^3 and a dynamic viscosity of 39.4 mPa·s as a working medium (Kalamkas field, RK). The landing speed of the locking element in the form of a metal ball (Steel BCr15 GOST 801-2022, density 7812 kg/m^3) will be v_{st} = 0.24 m/s. When replacing the ball material, in order to ensure the optimal landing speed, it is necessary to change the mass of the locking element as little as possible (m_{st} = 0.511 kg). To achieve this goal, a multicomponent structure was developed, which is a spherical steel core, gummed with a combined shell. Taking into account the imposed restriction on the landing speed of the locking element, a dependence is obtained for calculating the minimum radius of the core r_c (3):

$$r_c > r \cdot \sqrt[3]{\frac{\frac{0,9\mu}{2r^2 g} + \rho_l - \rho_{sh}}{\rho_c - \rho_{sh}}}, \tag{3}$$

where ρ_c – core density (steel), kg/m^3; ρ_{sh} – shell density (rubber compound, polyurethane), kg/m^3.

Taking Steel 20 GOST 1050-2013 with a density of ρ_c=7860 kg/m^3 as the material for the ball core, for a shell made of a rubber compound, $\rho_{sh.rub}$=1370 kg/m^3 and for a shell made of polyurethane, $\rho_{sh.pol}$=1259 kg/m^3, we obtain the minimum radius of a core gummed with a rubber compound $r_{st.rub}$=0.0180 m and polyurethane $r_{st.pol}$=0.0182 m. To approximate the mass of a gummed ball to the mass of a solid ball, we take the core diameter to be 38 mm. Then, for a rubberized ball, the average density will be $\rho_{st.rub}$=4219 kg/m^3 ($m_{st.rub}$=0.276 kg), and for a ball with a polyurethane shell $\rho_{st.pol}$=4157 kg/m^3 ($m_{st.pol}$=0.272 kg), and, accordingly, the landing velocity of the locking element will be $v_{st.rub}$=0.115 m/s and $v_{st.pol}$=0.113 m/s. The obtained values of the landing velocity of rubberized and polyurethane balls with a metal core are slightly higher than the critical value (v>0.1 m/s). There are two possible solutions to increase the mass of the locking element. The first is to reduce the thickness of the gummed shell, which will lead to a decrease in the resource life of the coating. The second option is the introduction of strengthening additives into the elastomeric matrix, which will lead to an increase in density and an improvement in the strength characteristics of the shell.

A reasonable choice of a gumming compound consisting of a base (rubber compound or polyurethane) and a synthetic material with a zirconium dioxide filler applied to the metal core of the valve ball provides damping and an increase in the width of the contact belt of the conditional meridian of the ball for effective redistribution of shock load. The experiment used a rubber compound grade 7B-14MA according to Technical Specification TS 38-105-1082-86. To achieve maximum mechanical strength, aging resistance, and heat resistance, the maximum allowable mass fraction of the zirconium dioxide filler should not exceed 25 phr (parts per hundred rubber), i.e., no more than 20% (ρ_{ZrO_2}=6080 kg/m^3). Exceeding this threshold often leads to particle agglomeration (the filler clumping together) and deterioration of the physical and mechanical properties of the product. For polyurethane, this value should not exceed 10–15%.

Then the parameters of the studied samples of the locking elements will be as follows:

– for a rubberized ball with filler: the maximum permissible mass fraction of the filler with zirconium dioxide is 20%; the average density of $\rho_{st.rub.fil}$=4355 kg/m^3, the mass of the ball is $m_{st.rub.fil}$=0.285 kg;
– for a polyurethane ball with filler: the maximum permissible mass fraction of a filler with zirconium dioxide is 10%; the average density of $\rho_{st.pol.fil}$=4218 kg/m^3, the mass of the ball $m_{st.pol.fil}$=0.276 kg.

The average density of the shell of the ball for different variants is calculated according to the following dependence (4):

$$\frac{1}{\rho_{sh}}=\frac{\omega_{base}}{\rho_{base}}+\frac{\omega_{ZrO_2}}{\rho_{ZrO_2}}, \tag{4}$$

where ρ_{sh} – the density of the shell with filler, kg/m^3; ρ_{base} – the density of the base material, kg/m^3; ρ_{ZrO_2} – the density of zirconium dioxide, kg/m^3; ω_{base} – mass fraction of the base material in the shell; ω_{ZrO_2} – the mass fraction of zirconium dioxide in the shell material.

The total density of the ball and its mass are determined by well-known mathematical relationships.

The landing speeds of the locking elements will be $v_{st.rub.fil}$=0.135 m/s and $v_{st.pol.fil}$=0.123 m/s, respectively. The obtained value of the landing velocity of the locking element made of a combined rubberized ball with filler more fully satisfies the condition v>0.1 m/s.

To calculate the hydraulic losses in the valve assembly connections, it is necessary to determine the pressure loss in the valve assembly during suction (Δp, Pa) [22]:

$$\Delta p=\frac{v_{\max}^2\cdot\rho_{ld}}{2\cdot\xi_{val}^2}, \tag{5}$$

where v_{max} – the maximum speed of product movement in the valve seat opening, m/s; ρ_{ld} – the density of the degassed liquid, kg/m^3; ξ_{val} – the valve flow rate, determined by special graphs (Figure 6) depending on the Reynolds number in the valve (in the seat opening).

Maximum speed of product movement in the valve:

$$v_{\max}=\frac{4\cdot q_{val}}{d_{val}^2}, \tag{6}$$

where q_{val} – product consumption through the valve, m^3/s; d_{val} – valve seat hole diameter, m.

The Reynolds number in the valve seat opening is determined by the following relationship:

$$Re_{val}=\frac{v_{\max}\cdot d_{val}}{v_l}, \tag{7}$$

where v_l – kinematic viscosity of a liquid, m^2/s.

To perform the calculations, we take the following initial data: valve seat hole diameter d_h=35 mm; maximum pump pressure p_{exit} =6.3 MPa; volumetric pump flow q_{val}=20.9 m^3/day, q_{val}=2.42·10^{-4} m^3/s; kinematic viscosity of oil v_l=4.36·10^{-5} m^2/s; oil density ρ_l=904 kg/m^3.

The maximum velocity of the working fluid in the pump is calculated by formula (6) – $v_{\max}$=0.79 m/s. We determine the Reynolds number by the formula (7) – Re_{val}=634.17. The resulting value of the Reynolds number is less than the critical value ($Re_{cr}\approx$ 2100÷2300). The value of the Reynolds number depends on the specific type of flow through the cross-section of the passage channel when the ball is flowing. Small values of the Reynolds number correspond to a situation where the viscosity forces (v_l=4.36·10^{-5} m^2/s) dampen turbulence, making the flow laminar. The flow coefficient of the valve ξ_{val} with a value of the Reynolds number in the range 600<Re< 3·10^4 will be equal to ξ_{val}=0.4 (Figure 6, curve 1) [22].

Using formula (5), we calculate the pressure loss in the valve Δp=0.002 MPa during suction, taking into account that the density of the degassed liquid ρ_{ld}=990.4 kg/m^3. Then the maximum pressure in the pump cylinder will be: $p = p_{exit} + \Delta p.$ Then p=6.302 MPa.

The coefficient of hydraulic resistance λ along the length of the flow, which takes into account the hydraulic conditions of the fluid flow, the viscosity of the fluid and the relative roughness Δ/d of the walls (Δ is the absolute roughness) is a function of these parameters, that is, λ=f (Re, Δ/d).

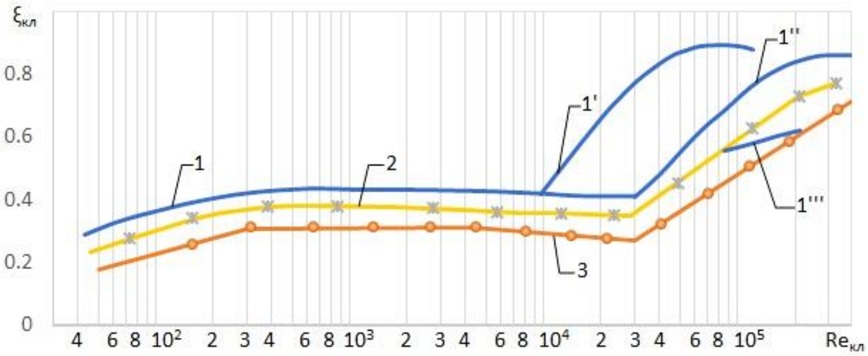


Fig.6. Dependence of the valve flow rate on the Reynolds number: 1 – with one ball and with windows: 1' – d_{val}=14 mm; 1'' – d_{val}=5 mm; 1''' – d_{val} =30 mm; 2 – with one ball and a stack; 3 – with two balls

For the studied operating conditions of the valve pair in laminar operation (Re=634.17), the hydraulic resistance coefficient λ along the flow length is a function of the Reynolds number – λ=f (Re) and, according to the method of T.M. Bashta, it is proposed to determine the dependence (8):

$$\lambda = \frac{75}{\mathrm{Re}}. \tag{8}$$

Then, at Re < 2320, the coefficient of hydraulic resistance will be equal to λ=0.118.

Provided that the initial velocity of the shut-off element is v_0=0 m/s and the movement is considered to be equidistant, the opening time of the valve t_x (s) will determine the dependence (9):

$$t_x = \sqrt{\frac{2m_P \cdot h_{\max}}{\Delta p \cdot F_{sav}}}, \tag{9}$$

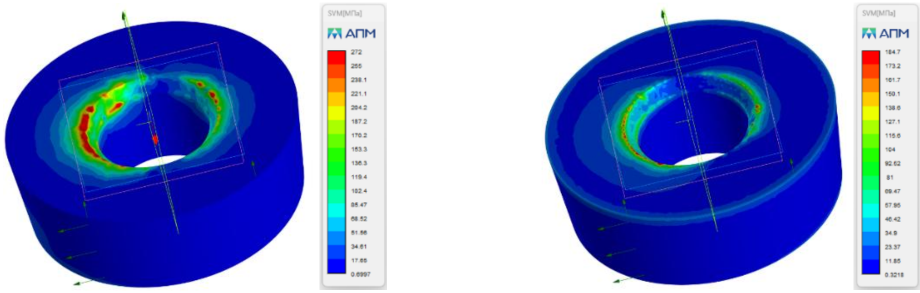
where m_P – weight of the locking element with attached parts, kg; h_{max} – maximum stroke of the closing element, m; F_{sav} – cross-sectional area of the valve passage, m^2.

From the valve design, the maximum stroke of the closing element is 23 mm. For a steel ball, the opening time t_{st} and acceleration a_{st} of the valve movement is t_{st}=0.12 s and a_{st}=2 m/s^2. Accordingly, for a rubberized ball with filler, the opening time and acceleration of the valve movement are $t_{st.rub.fil}$=0.09 s and $a_{st.rub.fil}$ =1.5 m/s^2. According to a theoretical study, it is advisable to replace the steel shut-off element of the valve with a new ball design, which is a metal ball gummed with a composite rubber compound with a zirconium dioxide filler. The effectiveness of the combined valve must be checked by simulation. The simulation problem was solved using the numerical method in the APM FEM software module. This software module is an APM Studio module adapted for COMPASS 3D from APM WinMachine. The geometric 3D model was created in the COMPASS-3D environment, where materials and their properties were also specified (Figure 7a). The finite element model is based on 10-node tetrahedra and contains 86077 nodes and 50112 finite elements. To simulate the stress-strain state of the valve pair and perform calculations, a computational model was built (Figure 7b).

a) b)

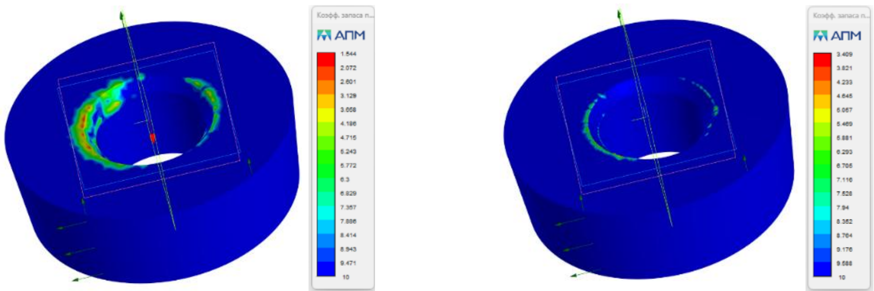
Fig.7. Valve pair: a) a model of a pump valve pair made in the Compass 3D software environment; b) calculated CAM of the valve assembly in the APM FEM environment

The APM FEM software module performs calculations for strength and fatigue at $2.5 \cdot 10^6$ loading cycles. The simulation was carried out for two cases: the first, the interaction of the valve seat with a steel ball, and the second, the interaction of the seat and a combined rubberized locking element with a zirconium dioxide filler. The parameter to be determined in the study of the strength characteristics of the working elements of the pump structure is equivalent stresses (σ_e), determined by the fourth theory of strength in dynamic calculation. Based on the simulation results, maps of the equivalent stress distribution according to Mises (Figure 8), yield strength reserve coefficients (Figure 9) and fatigue strength reserve coefficients (Figure 10) were obtained.



a) b)

Fig.8. Distribution of equivalent Mises stresses in the valve seat: a – when interacting with a steel ball (σ_{max}=272 MPa); b – when interacting with a rubberized ball with filler (σ_{max}=185 MPa)

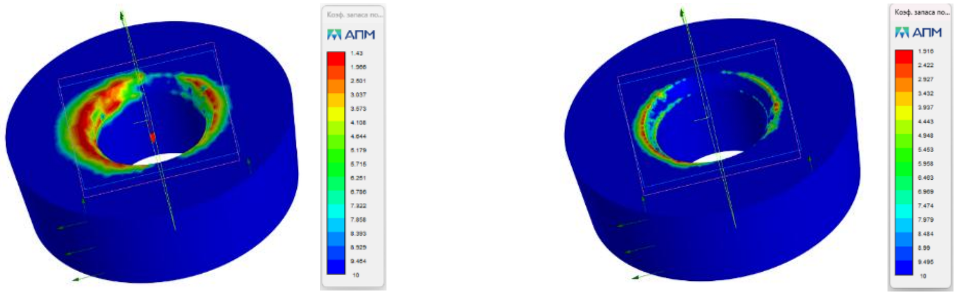


a) b)

Fig.9. Distribution of the reserve coefficient according to the yield strength of the valve seat: a – when interacting with a steel ball (n_{min}=1.55); b – when interacting with a rubberized ball with filler (n_{min}=3.41)

The calculations were performed for a shut-off element lined with a combined rubber compound containing a ceramic filler of zirconium dioxide (the mass fraction of the filler is 20%).

An analysis of the results obtained from a study of the design of a ball gummed with a combined rubber compound with a ceramic filler made of zirconium dioxide shows a 30% reduction in stresses in the valve seat and an increase in the coefficient of reserve for the fatigue strength of the valve seat surface in the most stressed areas from n_{min}=1.43 to n_{min}=1.9.



a) b)

Fig.10. Distribution of the reserve coefficient for the fatigue strength of the valve seat: a – when interacting with a steel ball (n_{min}=1.43); b – when interacting with a rubberized ball with filler (n_{min}=1.9)

4. Conclusion

Reducing the rate of change in the regulatory clearance value and reducing the leakage rate during pump operation is achieved by using an effective method to increase the wear resistance of the contacting surfaces of the ball-seat valve pair, which ensures uniform redistribution of the cyclically varying shock load per unit time of valve closure. To meet the speed requirement (v>0.1 m/s) for the landing of the locking element, a composite ball structure consisting of a steel core and a combined rubber shell on the outside has been developed. A ceramic material based on zirconium dioxide (ZrO_2) was chosen as the filler for the rubber compound. The elastic effect of closing the valve is also created through the use of materials (combined rubber compound) with damping properties. The additive material was determined – PCRK-3 powder with a content of 2.5-5% ZrO_2 (72 HRC) –15Cr17Ni12V3F35ZrO_2.

The validity of the choice of the valve assembly improvement method is confirmed by the method of 3D simulation modeling. The use of a combined ball with a filler based on a rubber compound and zirconium dioxide increases the fatigue strength of the valve seat by more than 20%, which leads to an increase in the life of the valve assembly with the same number of loading cycles.

Conflict of interest statement

The authors declare that they have no conflict of interest in relation to this research, whether financial, personal, authorship or otherwise, that could affect the research and its results presented in this paper.

CRediT author statement

Ivanova O.V.: Conceptualization, Methodology, Validation, Investigation, Software, Data Curation, Writing - Review & Editing; **Savinkin V.V.**: Methodology, Validation, Investigation, Data Curation, Writing Review & Editing, Writing - Original Draft. The final manuscript was read and approved by all authors.

Statement on the use of Artificial Intelligence.

During the preparation of this manuscript, artificial intelligence tools were used solely for language editing and grammatical improvement. No AI tools were used to generate scientific content, analysis, results, or conclusions.

Data Availability Statement

The data are available upon reasonable request from the authors.

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