Nafta-Gaz 2023, no. 9, pp. 596–603, DOI: 10.18668/NG.2023.09.05

Determination of the friction force between the draw rod and its guide in sucker rod well pumps and an analytical study of the stress deformation state of the valve assembly

Wyznaczanie siły tarcia pomiędzy cięgłem a prowadnicą cięgła w żerdziowych pompach wgłębnych oraz analiza odkształcenia naprężeniowego zespołu zaworowego

Sevda Y. Aliyeva, Sakit H. Abbasov

Azerbaijan State Oil and Industry University

ABSTRACT: The article is devoted to the determination of the friction force between the draw rod and the guide and to the analytical study of the stress deformation state of the valve assembly of the rod well pump. In sucker rod well pumps, a hollow cylindrical guide is used to ensure the same axis of the plunger as the cylinder during operation. The guide is attached to the upper end of the pump cylinder. The draw rod connecting the sucker rod and the plunger of the pump moves up and down in the internal cylindrical cavity of the guide in the corresponding movements of the balancer head. There must be a certain clearance between the draw rod and the guide to ensure free movement of the draw rod. Based on the calculation scheme for determining the friction force between the draw rod and the guide is given, and the necessary parameters are determined. According to the values obtained from the calculation, the graphs were built based on the dependences of the friction force between the draw rod and the guide on the angle φ, and on the path of the plunger when $\varphi = 30$. At the same time, according to the calculation scheme of the "ball-saddle" pair, the force acting on the ball, the stresses generated on the contact surfaces of the ball and the saddle, and other parameters were found. The friction and wear between the draw rod and the guide is also typical of the friction and wear between the polished rod and the wellhead forming structure. Because, in the latter case, as a result of the suspension point of the balancer head not having the same axis as the wellhead, the polished rod cannot move with the straight axis in wellhead valve.

Key words: valve assembly, torque, stock, integral constant, saddle, balancer head, contact voltage.

STRESZCZENIE: Artykuł zawiera opis metody wyznaczania siły tarcia pomiędzy cięgłem a prowadnicą cięgła dławikowej pompy wgłębnej oraz analizę odkształcenia naprężeniowego zespołu zaworowego pompy. W żerdziowych pompach wgłębnych stosuje się drążone cylindryczne prowadnice w celu zapewnienia współosiowości nurnika i cylindra pompy. Prowadnica ta jest przymocowana do górnej końcówki cylindra pompy. Cięgło stanowi połączenie żerdzi pompowej z nurnikiem pompy. Porusza się ono w górę i w dół w cylindrycznej prowadnicy, zgodnie z ruchem głowicy wyważającej. Pomiędzy cięgłem a prowadnicą należy zapewnić odpowiedni luz tak, aby zapewnić swobodny ruch cięgła. Wszelkie niezbędne parametry układu ustalono na podstawie schematu obliczeniowego siły tarcia występującego pomiędzy cięgłem a prowadnicą. Na podstawie wartości uzyskanych podczas obliczeń utworzono wykresy obrazujące zależności siły tarcia pomiędzy cięgłem a prowadnicą dla kąta φ oraz dla toru posuwu nurnika, gdy $\varphi = 30$. Jednocześnie, zgodnie ze schematem obliczeniowym pary „kula–gniazdo", wyznaczono siłę działającą na kulę, naprężenia powstające na powierzchniach styku kuli i gniazda oraz inne parametry. Tarcie i zużycie pomiędzy cięgłem a prowadnicą jest również typowe dla tarcia i zużycia występujących pomiędzy drążkiem polerowanym a prowadnicą w zagłowiczeniu odwiertu. W tym drugim przypadku ze względu na to, że punkt zawieszenia głowicy wyważającej nie znajduje się w osi głowicy odwiertu, drążek polerowany nie może się poruszać w osi zaworu głowicy odwiertu.

Słowa kluczowe: zespół zaworu, moment obrotowy, podstawa, stała całkowa, gniazdo, głowica wyważająca, napięcie międzystykowe.

Corresponding author: S.Y. Aliyeva, e-mail: *sevda.aliyeva.66@bk.ru*

Article contributed to the Editor: 12.05.2023. Approved for publication: 04.09.2023.

Introduction

The development of the oil industry requires a further increase in oil production. Oil well operation equipment, including rod well pumps, play an exceptional role in the performance of this task. For nearly 100 years, oil wells, especially with low-production (up to 5 m^3/day), medium-production (up to 100 m^3/day) and high-production (up to 500 m^3/day) capacities, widely use sucker rod well pumps. During this period, sucker rod well pumps have undergone a huge development and their designs have been improved in accordance with the requirements of operating conditions. It should be noted that, although the modern condition of rod well pumps is considered satisfactory, due to increasingly difficult operating conditions, there are a number of problems in their operation. These problems include the low resistance of the cylinder and plunger pair to wear due to friction, the riveting of the plunger inside the cylinder, the frequent failure of the individual elements of the pump, the sensitivity to the effect of the physical and mechanical composition of the extracted oil, and the low practical productivity, etc. These problems have always occurred during the existence of pumps and continue to exist today. Therefore, solving the mentioned problems is of special importance in oil extraction.

Setting the problem

At present, sucker rod well pumps are widely used to raise oil from wells with a depth of 3000–3500 m to the surface of the earth. As the depth increases, sucker rod well pumps face greater difficulties in the operation process. Therefore, especially in the operation of deep wells, breaking of pump elements is often observed. On average, the period between repairs of the pump is 30–35 days. Considering the scale of the underground repair of wells related to the replacement of the pump at "Azneft" Operating Union, it can be emphasised once again how important the solution of the previously mentioned problems is for the oil extraction industry.

It should be noted that usually at the last stage of operation and from the first day of operation of a number of wells, sucker rod well pumps are used as an alternative. Taking this into account, sucker rod well pumps will preserve their leading role unless a new operating method and device is created. In connection with what has been said, ensuring the effective operation of rod well pumps by improving their constructions remains a necessary issue of the day.

Many scientists and academicians have had assistance in solving the theoretical and technical problems of the operation of wells with a sucker rod pump device, as well as in the development of rod well pumps.

Thanks to the achievements obtained as a result of the relevant research works, the process of structural improvement of the sucker rod well pumps has been continued, which allows them to fulfill the task in front of them in increasingly difficult operating conditions.

Problem solving

As has been mentioned above, failure of the pump elements in the operation process as a result of the wearing of the pump elements, riveting of the plunger inside the cylinder, deficiencies in the valve joints, and obstacles created by the well factors still do not allow the useful work coefficient of the rod well pump unit to be improved.

Considering what has been said, it is necessary to ensure its longevity and high practical productivity at the stage of design and construction of sucker rod well pumps. For this purpose, the rationality of the construction should be ensured by choosing corrosion-resistant materials using the modern capabilities of technology and equipment. Along with the technical conditions, the development and application of auxiliary structures that directly serve to improve the performance of the pump should be carried out. To ensure this, the following measures should be taken:

- not allowing abrasive particles to come into contact with friction surfaces during operation, which cause the pump elements to be worn and the plunger to be riveted inside the cylinder;
- stable maintenance and reduction of the normal pressure force affecting the rubbing surfaces;
- not allowing dynamic shocks to occur in the pump elements;
- reducing volume losses in the pump;
- ensuring the stable operation of the valve assembly, etc.

Currently, sucker rod pumps used in the oil extraction industry are mainly equipped with ball valves. The main reason for this is that ball valves have several advantages. Despite working under high pressure, the mentioned valves provide hermeticity for a long time and are resistant to corrosion.

The valve assembly of the sucker rod pump, whether suction or injection valve, mainly consists of a body, a ball, and a saddle. The efficiency of the rod well pump is directly related to the normal operation of the valve assembly (Kerimov and Abbasov, 2016).

In the process of operation, especially in the "ball-saddle" pair of pumps operating in deep wells, changes in the shape of the saddle chamfer are observed due to deformation – the saddle loses its own weight and the liquid leaks back as a result of one-sided wearing. These issues remain the main problems of the valve assembly. The reason for the negative cases that we have mentioned can be such factors as: the valve sits on the saddle with an impact during operation; the ball moves irregularly inside the body and touches the saddle and the body with impacts; the ball does not sit properly on the saddle for various reasons.

Determination of the friction force between the draw rod and its guide in sucker rod pumps and the analytical study of the stress deformation state of the valve assembly of the pump, to increase the performance of sucker rod pumps and ensure their efficient operation in complex well conditions.

Discussion problem solving

Let's determine the friction force between the draw rod and its guide in sucker rod pumps.

Let's assume that depending on the position of the pump in the well at the point where the thrust rod is connected to the drilling string, the draw force of the rod forms a certain angle φ with its direction of movement. At this time, the horizontal sum of the draw force will be $P_x = P\sin\varphi$ (Figure 1).

Considering the effect of the force P_x let's find the friction force between the draw rod and guide from the approximate differential equation of the inclined axis (Kerimov, 1999; Vahidov et al., 2008).

$$y'' = \frac{M}{EJ} \tag{1}$$

Here:
M – is the bending moment,
E – modulus of elasticity of the rod material,
J – the inertia moment of the rod, $J = \pi d^4/4$,
d – is the diameter of the draw rod.

Let's assume that the bending moment created by the force P_y presses the lower part of the rod to the guide and rests on it in a flat position. At the same time, the force P_x bends the rod, separates it from the guide at a distance c in its upper part and compresses it at point B (Figure 1). Let's take the point O in the section where the rod is separated from the guide as the coordinate origin and direct the coordinate axes as shown in Figure 1.

The reaction force R_A generated at point A in the section where the rod is separated from the draw is based on the moment equation written considering point B:

$$R_A = P_x \frac{l}{c} \tag{2}$$

and considering that the angle α is small, the reaction force R_B created by the stock at point B will be as follows:

$$R_B = P_y\left(1 + \frac{b}{c}\right) \tag{3}$$

Let's note that the distance between the force P_x and point B is small, and it was not included in the equation (Karimov and Abbasov, 2016; Aliyeva et al., 2021).

Deflection moment generated at point c of the rod at an arbitrary distance x from the coordinate origin:

$$M_B = R_A x = P_x \frac{l}{c} x \tag{4}$$

If we substitute expression (4) in (1), we get the following:

$$y'' = \frac{P_y}{EJc} x \tag{5}$$

If we integrate expression (5) twice:

$$y' = \frac{P_y l}{2EJc} x^2 + C_1 \tag{6}$$

then we obtain:

$$y = \frac{P_y l}{6EJc} x^3 + C_1 x + C_2 \tag{7}$$

Here:
y' – is the turning angle of the rod,
y – deflection of the rod axis,
C_1, C_2 – are integral constants;
$y' = 0$ when $x = 0$ based on the conditions $y = 0$, we get $C_1 = 0$ and $C_2 = 0$ from expressions (3) and (4).

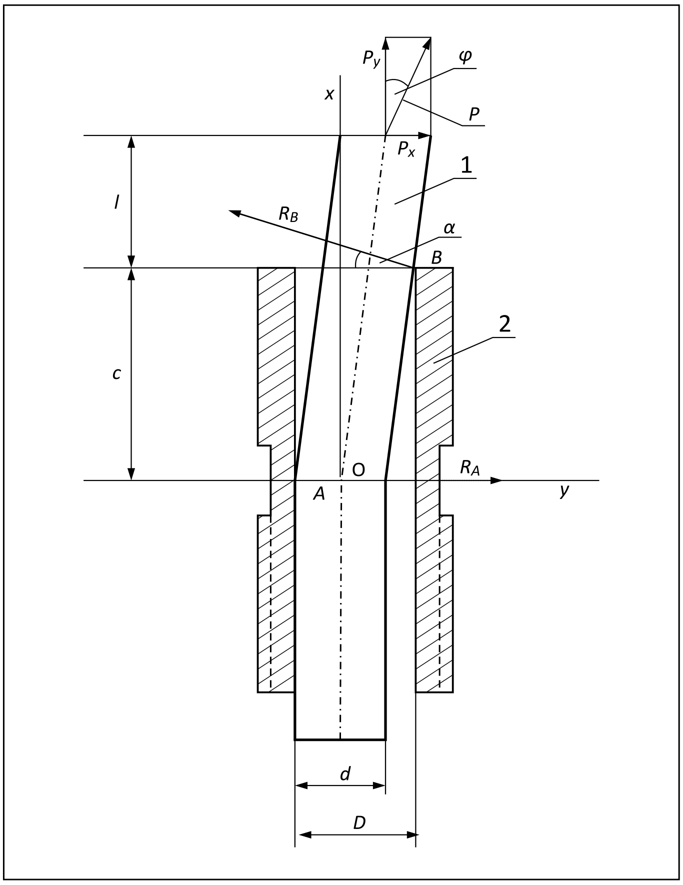

Figure 1. Scheme of determining the friction force between the draw rod and the guide: 1 – draw rod; 2 – guide

Rysunek 1. Schemat wyznaczania siły tarcia pomiędzy cięgłem a prowadnicą: 1 – cięgło; 2 – prowadnica

If to substitute the values of C_1 and C_2 in equation (5), we obtain:

$$y = \frac{P_y l}{6EJc} x^3 \tag{8}$$

According to the figure when $x = C$:

$$y = D - d$$

and as a result, expression (8) has the following form:

$$D - d = \frac{P_y l}{6EJ} c^2 \tag{9}$$

From expression (9):

$$C = \sqrt{\frac{6EJ(D-d)}{P_y l}} \tag{10}$$

Here:
D – is the inner diameter of the guide,
d – the diameter of the draw rod.

We get the value of c by substituting in (2) and (3):

$$R_A = \sqrt{\frac{P_x l^3}{6EJ(D-d)}} \tag{11}$$

$$R_B = P_x + \sqrt{\frac{P_x^3 l^3}{6EJ(D-d)}} \tag{12}$$

As a result, there is a frictional force between the draw rod and the guide:

$$F_S = (R_A + R_B) f = \left(P_x + 2\sqrt{\frac{P_x^3 l^3}{6EJ(D-d)}} \right) f \tag{13}$$

Here, f is the coefficient of friction between the materials of the draw rod and the guide.

In expression (13), the force P_x depends on the angle φ between the direction of movement of the draw rod and the draw force. As for the l – distance, its maximum length is equal to the plunger travel.

Figure 2 shows the dependence of the friction force on φ in the case where a sucker rod pump with a standard diameter of 32 mm works at a depth of 1500 m, and the dependence of the draw rod on the travel path is shown in Figure 3. It is assumed that $E = 2.1 \cdot 10^5$ MPa, $J = 0.515$ cm^4, $D = 20$ mm, $d = 18$ mm for the considered pump. As it can be seen from the picture, the friction force in both cases increases with the growth of the angle φ and the distance l and can take any values (Karimov and Abbasov, 2016; Rahimova and Mansurova, 2022).

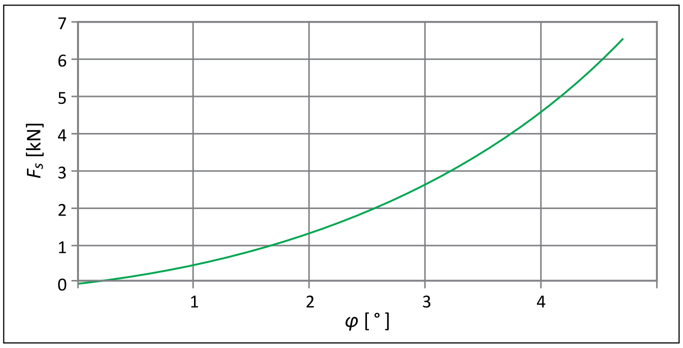

Figure 2. Dependence of friction force between draw rod and guide at φ angle

Rysunek 2. Zależność siły tarcia pomiędzy cięgłem a prowadnicą przy kącie φ

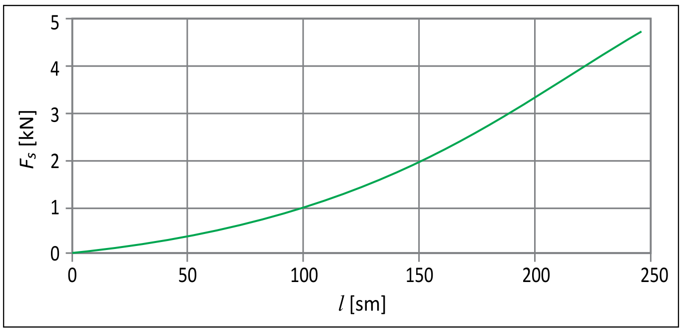

Figure 3. The graph of the dependence of the friction force between the draw rod and the guide on the travel path of the plunger when $\varphi = 3°$

Rysunek 3. Wykres zależności siły tarcia między cięgłem a prowadnicą na ścieżce posuwu nurnika, gdy $\varphi = 3°$

Experimental part

It should be noted that in several wells, including wells with a curved profile, the thrust force applied to the draw rod usually does not coincide with its direction of movement. In such cases, the contact pressure on the contact surface of the draw rod with the guide increases. An increase in the pressure on the contact surfaces leads to an increase in the friction force and, as a result, an acceleration of the linear wear of the rubbing surfaces. Cases of breakage of the draw rod are often observed. In wells observations made on failed draw rod mainly show that they are worn away and broken.

In the research conducted on the H007 sucker rod pump with a conventional diameter of 43 mm, operating in the well No. 3117 belonging to the Oil and Gas Extraction Department, the draw rod of the pump was worn within 1 month at 1.5 m from the walkway and showed a 50% decrease in its weight. This situation is observed in other pumps as well. Such amount of wear is of course caused by the normal pressure force between the draw rod and its guide. For this reason, there is always wear because of friction between them.

The same case can be applied to the suction valve lifting rod in tubular pumps because of friction with the tip of the plunger.

Thus, the frictional forces arising due to operating conditions in sucker rod pumps, on the one hand, complicate its work by increasing the load on the rod, and at the same time, it causes the pump elements to wear out and fail due to untimely breaks (Vahidov et al., 2008).

Analytical study of the stress deformation state of the valve assembly of the sucker rod pump.

To analyse the operation of the ball and saddle pair, let's consider the following problem (Karimov and Abbasov, 2016).

Let's assume that the valve (ball) 1 sits on the saddle 2, and at the same time it is affected by a compressive force P.

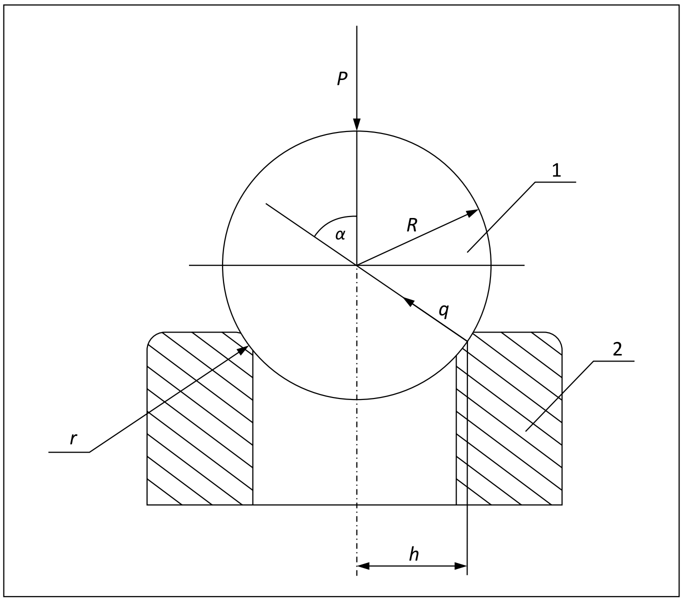

Figure 4. Calculation scheme of the "Ball-saddle" pair: 1 – sphere; 2 – saddle

Rysunek 4. Schemat obliczeń pary „kula–gniazdo": 1 – kula; 2 – gniazdo

The axial compressive force P acting on the sphere will consist of the following sums:

$$P = (H_m + H_d)\pi R^2 \tag{14}$$

Here:

H_m – is the column pressure of the liquid on the sphere inside the pump-compressor pipeline,

H_d – additional dynamic pressure caused by the time delay of the sphere sitting on the saddle,

R – is the radius of the sphere.

The pressure of the liquid column on the sphere consists of the pressure created by the liquid column, the height of which is equal to the depth of discharge of the pump into the well.

As for the dynamic pressure, it can be determined from N.Y. Zhukovsky's formula (Aliyev and Aliyeva, 2017).

$$H_d = v_0 \rho \sqrt{\frac{k}{\rho}} \tag{15}$$

Here:

v_0 – is the velocity of liquid falling in the pump-compressor tube,

ρ – liquid density,

k – the elasticity modulus of the liquid.

Velocity of the plunger:

$$v_p = \frac{\pi n s}{60} \sin\delta$$

if to consider a condition of flow continuity, then:

$$v_0 (f_b - f_{st}) = v_p f_p$$

using this expression, for the velocity of the liquid in the pipe we obtain the following equation:

$$v_p = \frac{f_p}{f_b - f_{st}} \cdot \frac{\pi n s}{60} \sin\delta$$

If to substitute the last expression in (15), then the following is obtained:

$$H_d = \frac{f_p}{f_b - f_{st}} \cdot \frac{\pi n s}{60} \cdot \rho \sqrt{\frac{k}{\rho}} \sin\delta \tag{16}$$

Here, accepting the plunger as a whole f_p is its cross-sectional; f_b – cross-section of the internal cavity of the tubing; f_{st} – cross-sectional area of the rod; n – the number of cycles per minute of the balance head; s – movement of the balancer stroke; δ – is the delay angle ($\delta = 8° – 20°$).

The ball sits on a spherical chamfer with a very small radius of convexity (r = 1–2 mm) in the saddle. Therefore, the contact area of the ball and the saddle can be considered as a straight line with a length of $2\pi h$. h is the distance from the axis of the saddle to the point of contact of the ball and the chamfer. As we mentioned, since the contact area of the ball and the chamfer of the saddle is very small, during their deformation, mutual pressure forces are also applied to small areas. Therefore, the tension generated here should be considered as local. Considering the contact surfaces of the ball and the chamfer of the saddle as the compression of cylinders with parallel axes of $2\pi h$ length, we can find the normal stress on their contact surface according to Hertz's formula:

$$\sigma_{\max} = 0.418 \sqrt{q \frac{E(R+r)}{R \cdot r}} \tag{17}$$

Here:

E – is the modulus of elasticity of the ball and saddle material,

q – the intensity of the load per unit length of the contact line,

R – the radius of the ball,

r – is the radius of the spherical chamfer of the saddle.

In expression (17), it is assumed that the modulus of elasticity and Poisson's ratio of the material of the ball and the saddle are the same.

To find the force q per unit length of the contact surface, let's write the equilibrium condition of the ball (Figure 4):

$$P = q2\pi h\cos\alpha$$

From here:

$$q = \frac{P}{2\pi h\cos\alpha} \tag{18}$$

If to substitute expression (18) in (17), we will get:

$$\sigma_{\max} = 0.418\sqrt{\frac{P}{2\pi h}\cdot\frac{E(R+r)}{Rr\cos\alpha}} \tag{19}$$

For the normal operation of the "ball-saddle" pair during exploitation:

$$\sigma_{\max} \le [\sigma]_{con} \tag{20}$$

must be met conditionally.

Here $[\sigma]_{con}$ is the stress that can be accepted in the contact area of the ball and the saddle.

According to condition (20):

$$0.418\sqrt{\frac{P}{2\pi h}\cdot\frac{E(R+r)}{Rr\cos\alpha}} \le [\sigma]_{con}$$

and from here we get for the acceptable value of the compressive force:

$$[P] = \frac{2\pi hRr\cos\alpha}{0.175E(R+r)}[\sigma]^2_{con} \tag{21}$$

Using this expression, it is possible to analyse the stress deformation state of the saddle and the changes in shape observed in it.

If we write expression (14) in the form:

$$P = (H_m + H_d)\pi R^2 = \pi R^2 L\rho \tag{22}$$

and equate it with the expression (21), it is possible to determine the maximum depth to which the pump should be lowered into the well for the normal operation of the "ball-saddle" pair, i.e.:

$$L = \frac{2\pi hr\cos\alpha[\sigma]^2_{con}}{0.175E(R+r)\rho R} \tag{23}$$

Here:

L – is the discharge of the pump into the well, considering the dynamic pressure depth,

ρ – is the density of the produced fluid.

The comparison of the maximum values of the depth of discharge into the well according to the instructions of the sucker rod pumps and the discharge depth determined from the expression (22) is given in Table 1.

Table 1. Comparison of the discharge depth of the pump according to the instruction and determined by the expression (23)

Tabela 1. Porównanie głębokości tłoczenia pompy zgodnie z instrukcją i określonej wzorem (23)

Number	Conventional diameter of the pump	Discharge depth of the pump into the well according to:	
		instructions	calculation
	[mm]	[m]	[m]
1	28	2500	1810
2	32	2500	1710
3	38	2000	1390
4	43	1500	1230
5	55	1200	1081

As it can be seen from Table 1, the recommended discharge depth of the sucker rod pumps according to the manual is greater than the maximum discharge depth obtained by the expression (22). In this case, the actual compressive force acting on the valve is greater than the releasable compressive force, i.e. $P > [P]$.

Let's note that when calculating the actual compressive force, the excess pressure at the wellhead is not considered. At the time of calculation, the modulus of elasticity of the fluid k = 21 000 kg/cm^2; $\rho = 10^{-6}$ [(kg · s^2)/cm^4]; the stroke of the polished rod is S = 300 mm, the number of revolutions of the wheel arm is assumed to be n = 10 revolutions/min. Valve and saddle parameters were selected according to (Aliyev, 2023).

Under such conditions, the chamfer of the saddle must be deformed or disintegrated. Since the hardness of the material of the ball HRC56-62 is higher than the hardness of the material of the saddle HRC40-45, the saddle is mainly deformed. Observations made on the valves of the pumps lifted from the well show that the following types of deformations prevail in the face of the saddle.

It is inevitable that the facet of the saddle will be deformed or fall apart. Since the hardness of the material of the ball (HRC56-62) is greater than the hardness of the material of the saddle (HRC40-45), the saddle is mainly deformed. Observations made on the valves of the pumps lifted from the well show that the following types of deformations prevail in the chamfer of the saddle.

- The chamfer of the saddle takes a wide mold-like shape and loses enough weight (Figure 5).This type of deformation is observed in most valves. Usually, such deformations are characteristic of cases where the pump has little mechanical mixtures in the liquid, and it works for a long time. It is felt that the saddle is not corroded, and the chamfer is processed as if deepened (absence of protrusion).
- In the process of use, a wide chamfer and an annular protrusion appear on the saddle. Such saddles are often found.

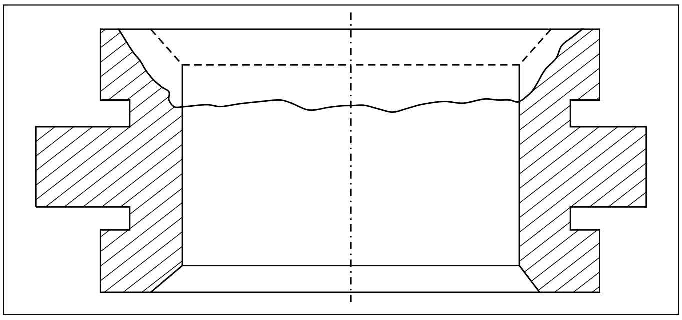

Figure. 5. A saddle with a wide chamfered shape

Rysunek 5. Gniazdo z szeroką fazą

As it can be seen from Figure 6a, the chamfer of the saddle has widened and received an annular protrusion. On the other hand, on the upper surface of the saddle, there are metal protrusions in the form of sharp pointed petals. These protrusions are at the same level (Figure 6b). The valve saddle takes such a condition in a corrosive environment and when the content of sand and gas in the liquid is high. The transition diameter of the saddle almost does not change. These deformations especially apply to saddles with low hardness.

through those places under high pressure and accelerates the disintegration process (Ragimova, 2013).

In addition to the above-mentioned, it can be shown that the valve opens a seat in the saddle and is subjected to other types of deformations.

The plastic deformation or collapse of the saddle in the valve assembly can be explained as follows. We mentioned above that the valve sits on the saddle mainly with a shock. According to the theory of increasing the hardness of metals by forging, it can be said that in the contact area between the ball and the saddle, since the strength of the material of the saddle is less than that of the ball, two cones (sliding cones) are formed in the deformed part of the saddle from molecules that do not participate in the deformation. In the process of forging, these cones come closer to each other and compress the metal particles.

When the vertices of the cones touch each other, they either collapse or slide on their origin. At this time, they push other molecules in the direction perpendicular to the compressive forces to the less resistant part, i.e., the upper and inner space of the saddle.

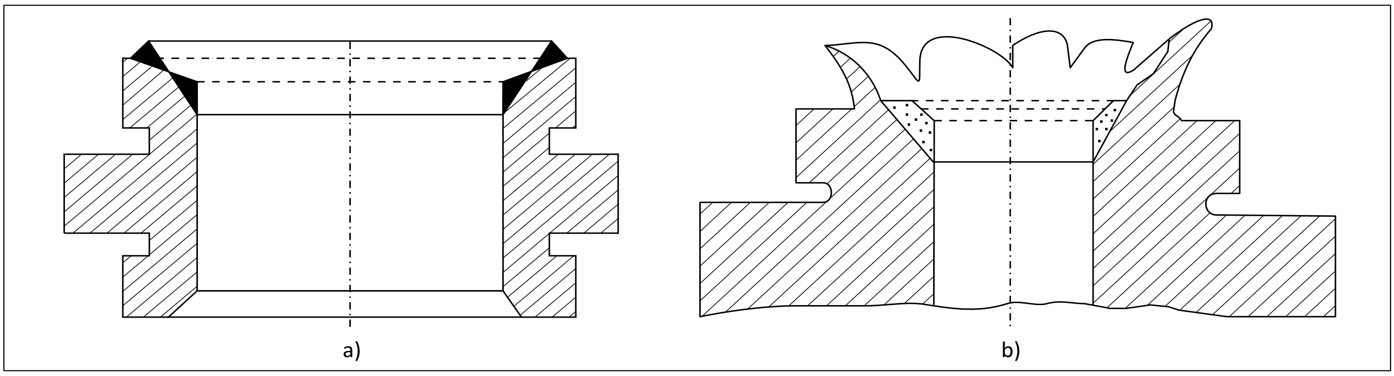

Figure 6. A saddle with a deformed chamfer: a) a saddle with an enlarged chamfer as a result of deformation, b) protrusions formed in the facade as a result of deformation

Rysunek 6. Gniazdo ze zdeformowaną fazą: a) gniazdo z powiększoną fazą w wyniku odkształcenia, b) wybrzuszenia powstałe w obudowie w wyniku deformacji

- The chamfer of the saddle is irregularly deformed during use. These types of deformations have a local character in the places where the ball sits on the saddle when the percentage of sand in the liquid is high. Disintegration of these saddles can be observed in several places, and sometimes the chamfer has changed its shape. There is also some wear in the transition area of the saddle. Such a deformation is usually observed when the hardness of the valve seat is HRC45-50. As it is known, the local disintegration of the material of the saddle is caused by the impact of the valve on it.

It should be noted that in case of local damage on the chamfer of the saddle, the flow of fluid-mechanical mixture passes

Conclusions

In the first of the mentioned cases, that is, when the sliding cones destroy each other, the plastic of the saddle is deformed and undergoes a change in shape. In the second case, i.e., the sliding cones move along their origin, compressing the molecules and creating metal protrusions in the structure, because of which the saddle collapses.

The cases we have mentioned are most often observed when the ball hits one side of the saddle for any reason and sits on it.

Considering what has been said, it is necessary not to allow the compressive force to exceed its allowable value and to ensure that the ball sits in the saddle in the center and without impact. According to the calculation, saddles made of BK-6 and BK-15 solid alloys can be used in their valve assemblies when operating certain deep wells with pumps.

1. Breakage of the draw rod in sucker rod pumps was determined analytically because of its friction with the draw, and an analytical expression for the friction force is obtained.
2. Based on the theoretical and practical studies, the constructions can be developed, which serve to increase the performance of sucker rod pumps, considering the well factors, applied in production and giving positive technical and economic results:

- a draw that limits the settling of sand on the plunger in the pump-compressor pipeline and ensures the rectilinear movement of the plunger inside the cylinder. The design provides 37% increase in operating time and 0.525 m^3 daily productivity due to 18% wear intensity reduction compared to common pumps;
- a new type of plunger with different types of channels opened on its surface to reduce the backflow of liquid between the "plunger-cylinder" pair. The plunger allows fluid backflow to be reduced by 30% and corrosive intensity – by 20% in sucker rod pumps.

References

Aliyev A.M. 2023. Determination of kinematic parameters of the compressor unit applied to low-pressure petroleum gas. *Nafta-Gaz*, 79(4): 252–256. DOI: 10.18668/NG.2023.04.03.

Aliyev A.M., Aliyeva S.Y., 2017. Constructions of oilfield equipment and special facilities. *Azerbaijan State Oil and Industry University, Baku*.

Aliyeva S.Y., Demirova C.R., Abbasova S.A., 2021, Research of the problem of optimization and development of a calculation method for two-stage chain drives used in heavy industrial vehicles in conditions of economic efficiency. *EUREKA: Physics and Engineering*, 4: 105–112.

Karimov O.M., Abbasov E.I., 2016. Study of the influence of the plunger position inside the cylinder on the operation of the pump. *News of the Azerbaijan Engineering Academy*, 2: 56–61.

Kerimov O.M., 1999. Increasing the performance and operational parameters of sucker rod pumps. *Maarif, Baku*.

Kerimov O.M., Abbasov E.I., 2016. Determining the friction force between the guide and the draw rod in an insert-type downhole pump. Equipment and technologies for the oil and gas complex. *IRIOEOG*, 2: 82–87.

Ragimova M.S., 2013. Calculation of the thickness of seals for flange joints of oilfield equipment. *Oilfield Business*, 5: 36–38.

Rahimova M.S., Mansurova S.I., 2022, Methods for calculating the fatigue and strengths machine parts. *Nafta-Gaz*, 78(12): 888–892. DOI: 10.18668/NG.2022.12.06.

Vahidov M.A., Karimov O.M., Eyvazova Z.E., 2008. Oil and gas production technique. *Azernashr, Baku*, 1–458.

Sevda Yagub ALIYEVA, Ph.D.
Associate Professor at the Department of Mechanics
Azerbaijan State Oil and İndustry University
16/21 Azadliq Avenue, AZ1010, Baku, Azerbaijan
E-mail: *sevda.aliyeva.66@bk.ru*

Sakit Hasan ABBASOV, Ph.D.
Assistant Professor at the Department of Mechanics
Azerbaijan State Oil and İndustry University
16/21 Azadliq Ave, AZ1010 Baku, Azerbaijan
E-mail: *s.h.abbasov@mail.ru*