On the efficiency of sealing the fountain valve lock joint

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Abstract

The straight-flow valves are designed to shut off and fully open the flow in the oil and gas industry.Valves are widespread equipment in the oil and gas industry, and maintaining the tightness effect in them remains an ever-present issue.The scientific research work is devoted to ensuring the sealing efficiency in the improved valve assemblies. It was found that the relative pressure created in the improved valve assembly is the most important parameter in the formation of metal-metal sealing. The relative pressure created in the new sealing assembly satisfies the sealing condition.

Keywords: valve, seal, oil and gas, relative pressure, tightness.

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1. Introduction

The straight-flow valve occupies one of the most important places in the well equipment complex(Figure 1). The durability and reliable operation of the valve determines the uninterrupted operation of the well armature as a whole. The valve has several important nodes and junctions. It is a very urgent issue to study the durability, tightness and other properties of the shut-off node (shield-saddle pair), which is one of its most productive nodes [1-11].

Experience shows that the reliable operation of a fountain valve depends on the reliability of its valves, and ultimately on the operability, reliability and durability of the valve-valve shut-off assembly.

By ensuring the reliability and durability of the operability of the shut-off assembly at the required level, the reliability of the valve and, as a result, the fountain valve as a whole is increased.

The purpose of the work is to determine the factors influencing the tightness of the shut-off unit.

Due to the friction that occurs on the contact surfaces of the stationary and pressure-moving parts of the valve, its parts and nodes fail due to wear. If the relative pressure is not evenly distributed in the affected parts, the scale of external wear factors affecting the working surfaces increases many times, leading to rapid failure.

Abrasive particles in the working solution have a serious impact on the longevity of the valve. Depending on the abrasives in the solution, the working life of the valves can vary from several weeks to several years.

Analysis of the above shows that the low life of the parts occurs mainly as a result of mechanical wear on the working surfaces. The criterion of mechanical wear is friction. Any scientific research work carried out in this direction can be considered relevant.

At present, research work aimed at improving the performance of valves is carried out in 3 directions:

-study of the influence of the physical and mechanical properties of the contact surfaces on pairwise friction and wear, taking into account the influence of the environment;

-by applying polymer materials to the contact process between the metal surfaces of the valve body parts and ensuring an even distribution of relative pressure in the contact areas;

-by providing lubricating fluid and solid lubricants to the contact surfaces.

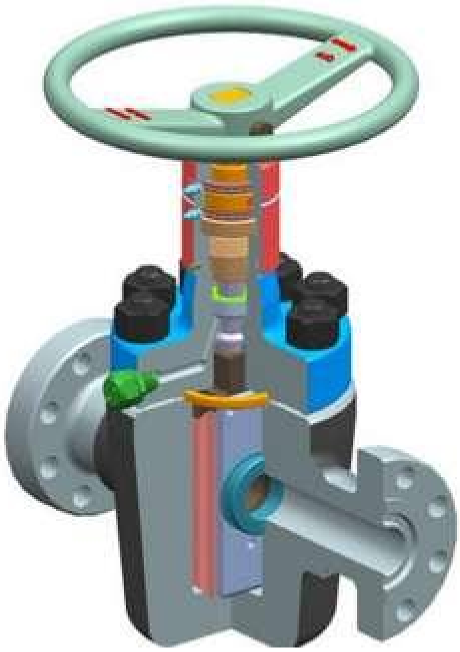

Figure 1. The straight-flow valve.

2. Methodological part

The shut-off assembly consists of six main parts in the version shown in Figure 2: shield 1, saddle ring 2 made of wear-resistant alloy, saddle 3, saddle seals 4 and 5, as well as a shield (guide) 6. Seals are placed on both the inlet and outlet sides of the saddle between the saddle and the body and between the saddle ring and the saddle (on the inlet side).

The tightness of the shut-off assembly is ensured by the indicated seals and the metal-to-metal contact surface of the shield and the saddle ring. This guarantee is based on a number of factors [2-13]:

relative deformation caused by the pressure of the environment on the contact surface; roughness of the rubbing surfaces; presence of lubricating material on the contact surface; unevenness of the contacting surfaces of the shield and the saddle.

The pressure acting from the inlet side (when the valve is closed) tends to bend the shield mounted on the annular support (on the saddle ring). This deflection should not exceed the experimentally determined limit. This deflection depends on the strength of the shield material and its thickness. The shield design plays a role here [1,13-18].

In order to ensure high reliability and operational properties and condition of the fountain armature, the design of the entire armature and its connecting devices must meet the following requirements: the seal must be created between the "metal-to-metal" contact surfaces; the sealed joints of the new metal-to-non-metal construction must be filled with lubricating oil; the areas where the medium (dry product) comes into contact with the metal surfaces must be as small as possible; the fountain armature must be remotely controlled.

3. Main part and discussions

Currently, fountain fittings designed for pressures of 21, 35, 70 MPa are equipped with ZMS type single-layer plate, shield valves. The tightness of the plug is ensured by creating the necessary pressure on the sealing surfaces of the shield and saddle. In the considered design, the initial pressure is created by non-metallic seals. In disc (plate) spring plugs, the initial initial pressure is created by plate springs.

Reliable sealing of the sealing and non-sealing gaps between the parts of the plug and other components is restored by applying sealing lubricant Az-162 to the striker valve in the middle part of the valve body (for valves other than ZM-652).

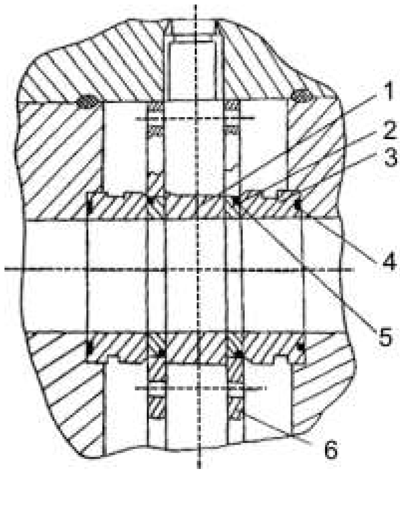

Figure 2. Straight-through valve. 1 – shield; 2 – saddle ring; 3 – saddle; 4 and 5 – saddle seals; 6 – shield (guide).

The tightness of the body and the cover is achieved by tightening the pins on the flange with nuts.

The adjustment of the axial clearance of the through holes of the shield and the body is achieved with the help of adjusting plugs, the correct positioning of the shield is controlled by a nut,

To facilitate control of the valve, the travel nut is completed on ball support pads. Since the spindle and travel nut are removed from the groove zone, their working conditions are improved, the sealing of the spindle and the stem is ensured by the use of sleeves made of AMQ material.

To ensure the operability of the shut-off valve, it is necessary to calculate the strength of its parts and study the roughness of the contact surfaces of the shield and the saddle ring.

The calculation of the strength, stiffness and type of the parts of pneumatic actuator valves with different passage diameters and working pressures, which are manually and remotely controlled, is carried out in order to ensure their operability.

The strength of the parts is related to their hardness, as well as wear resistance, therefore, ensuring operability according to these criteria allows ensuring the required reliability of the valve.

The coefficients, general formulas and details used in the calculation of the strength of the shield and other parts were taken from the relevant literature on the strength of the parts and from the current standards, technical conditions and other normative technical documents, as well as from the relevant test results conducted at AzINMAS.

All parts of the valve operating under high mechanical stress are calculated for strength, and their safety factor for strength is determined.

This article provides an analysis of the details of the shut-off assembly of manually operated valves.

The valve shield is usually made of 38XMA steel according to the current standard (GOST5632-72). The yield strength of the material in the heat-treated state is assumed to be $\sigma_{\text{yield st}}$ = 880 MPa. In order to increase the wear resistance of the part, the working surface of the shield is coated with PQ-12H-02 powder (TS 48-19-389-84). The calculation of the shield deflection is carried out within the given conditions. The medium at the inlet of the valve compresses the shield towards the outlet side with a working pressure of $P_{\text{worc pres}}$ = 35 MPa. In this case, the compressive force $\theta_{comp\,for} = \frac{\pi D_{o.d.}^2}{4} P_w + \theta_{r.f.}$ (1)

where $\theta_{comp\,for}$ – compressive force created by the working pressure; $D_{o.d.}$ – outer diameter of the saddle seal; $P_{worc\,pres}$ – working pressure; $\theta_{r.f.}$ – residual force of the saddle seal located on the inlet side.

This force is much smaller than the compressive force, its numerical value is determined experimentally in laboratory conditions. It is possible to calculate the experimental value of $\theta_{r.f.}$ In this case, the residual force of the seal for a valve with a nominal passage D = 80 mm does not exceed 0.01 MH.

(1) düsturuna daxil olan parametrlərin işarələri də və mm keçidli siyirtmələr üçün həmin parametrlərin ədədi qiymətləri müəyyən edilmişdir. The signs of the parameters included in formula (1) are also determined for valves with $D_ş = 65$mm and $D_ş = 80$ mm. The numerical values of these parameters have been determined.

The shield is pressed against the saddle ring with the output sides by a force $\theta_{comp\ for}$. In this case, the shield is considered as a circular plate loaded with a uniformly distributed load in the center axis zone and freely supported along the contour. The maximum deflection of the shield is determined by the plate theory:

$$W_c = \frac{P_{worc\ pres} D_0}{1024D} + \frac{5+\mu}{1+\mu} \quad (2)$$

Where W_c – is the deflection of the shield in the central axis zone; D – is the diameter of the shield plate cylinder; μ – is the Poisson's ratio for the shield material; D – is the cylindrical stiffness of the plate:

$$D = \frac{Eh^3}{12(1-\mu)} \quad (3)$$

here E – is the longitudinal modulus of elasticity of the shield material; h – is the thickness of the shield.

According to the thick plate theory (3), to which the shield is partially related by the ratio of its geometric dimensions, its maximum deflection (at the center) must be less than the allowable deflection [W].

The removable lining of the shield is determined by ensuring the thickness of the joint details of the stopper:

$$[W] = \frac{2W_c'}{D_{o.d.}} \sqrt{\frac{8Q_{SIX}}{\pi q_t}} \quad (4)$$

Here $D_{o.d.}$ – is the outer diameter of the contact surface (saddle) of the shield and the saddle on the outlet side.

q_t – is the relative contact stress of the sealing elements for resistance to sliding and is determined experimentally: $q = 260$ MPa is assumed.

Maximum value of the circumference and radial stresses at the center of the shield (along the central axis)

$$\sigma_r = \sigma_p = \frac{3(3+\mu) P_{worc\ pres} D_0}{32h^2} \sqrt{\frac{8Q_{SIX}}{\pi q_t}} \quad (5)$$

Here σ_r – is the circumferential (around the circumference) bending stress at the center of the shield.

Equivalent stress, taking into account the pressure of the environment acting normal (perpendicular) to the surface of the shield.

$$\sigma_{ekv} = \sigma_p + P_{worc\ pres} \quad (6)$$

The safety factor for the yield strength of the material when bending stress is applied in the center axis zone of the shield

$$n_c = \frac{\sigma_{yield}}{\sigma_{ekv}} = 2{,}5 \qquad (7)$$

The safety factor of the shield strength ensures its rigidity and does not change the deformation under pressure.

The study of the wear resistance of the shield-saddle pair shows that the reliability and operability of the joint depends on a number of factors (chemical composition of the environment, abrasive particles, ambient temperature, etc.). In a normal environment, the wear resistance of the pair depends on mechanical friction. The dependence of the wear of the shield on the cycles of reciprocating motion of the rubbing pair under such friction conditions is shown in Figure 3.

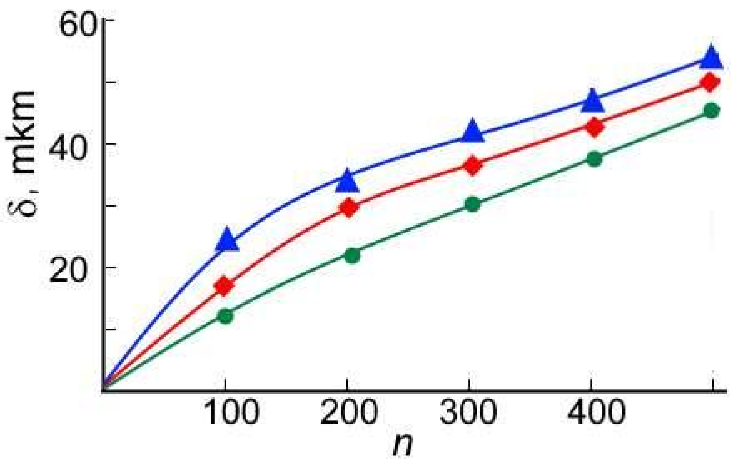

Figure 3. Dependence of shield wear on the cycles of reciprocating motion of the rubbing pair.

As a result of the study, it was determined that the wear of the shield saddle pair with a hardness of HRC=55-66 on the Rockwell scale is 0.040...0.050 mm after 500 cycles of back-and-forth movement intended for the valve.

4. Conclusions

It has been established that the dependence of the shield wear resistance on the cycles of mechanical friction of the reciprocating motion of the rubbing pair, as well as other factors (chemical composition of the environment, abrasive particles, ambient temperature, etc.) significantly affects the reliability and operability of the pair connection. In particular, the wear of the shield-saddle pair with a hardness of HRC = 55-66 on the Rockwell scale is 0.040 ... 0.050 mm after 500 cycles of reciprocating motion.

Conflıct of ınterest

The authors of this work declare that they have no conflicts of interest.

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