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Cooperation of the PTFE sealing ring with the steel ball of the valve subjected to durability test

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Abstract: The paper presents an quantitative assessment of the wear process of the sealing ring made of pure PTFE cooperating with steel ball of the industrial valve. The three variants of the sealing PTFE ring differing in shape were taken into consideration. In the first stage the numerical simulation by means of Finite Element Method was used to determine the contact pressure between the steel ball and PTFE seal ring. In the second stage the experimental test was carried out to check the wear rate of the PTFE ring. The DN50 PN10 industrial ball valve was subjected to 1000 of the closing/opening cycles. In the three variants of the sealing ring the friction torque necessary to open and close of valve was measured. The weight loss due to wear was determined which was the assessment of the wear degree in macro-scale. As a wear in a micro-scale the roughness parameter on the contact surface of the PTFE ring was taken as an index.

Keywords: durability; wearing; ball valve; PTFE; seal

1 Introduction

Properties of polytetrafluoroethylene (PTFE) like very good chemical stability, low friction coefficient, good thermal stability, make this material attractive for sliding applications. However, there are some disadvantages – low wear resistance and high viscoelastic deformation under load comparing to other polymeric material. These features of PTFE provoked research in direction of tribological features improvement. Graphite flakes [1] additive, basalt fiber [2] additive, glass fiber, carbon bronze metal oxides additives [3], modify the mechanical properties of PTFE. Basing on cited achievements one can summarize that additives generally improve wear resistance of PTFE. However some parameters can be deteriorated comparing to pure PTFE, e.g. friction coefficient, chemical resistance [4]. In sliding pair steel-PTFE composites crucial are the surface parameters of cooperating bodies [5]. Too high roughness parameter of steel surface as well as too low roughness can increase the wear rate of polymer [6]. Various handbooks [7, 8] give broad spectrum of physical, chemical and mechanical parameters of unfilled and filled with some additives PTFE. The a synthetic approach to the problem of friction was presented in [9]. The issue of ball and seat cooperation as a tribological pair in ball valve is included in the field of scientific interests of the authors. The tests of the prototypical ball valve in terms of durability and friction resistance were performed [10]. There was taken under consideration the influence of seat fatigue wear on functional parameters of the ball valve [11].

In order to limit the range of variables in practical application the typical industrial ball valve with floating ball was destined to durability tests. Crucial parameters of the valve serviceability result from the PTFE-steel friction pair condition. There were analyzed three different shapes of the seal cooperating with the ball. Contact pressure and intensity of wear in PTFE-steel pair determine the friction torque, wear rate, and tightness of the valve.

2 The research object

2.1 Construction of the ball valve

Based on references review the authors were taken to work towards to modernization of the geometry of the sphere-seat interface. A ball valve with floating ball was chosen as a typical solution for small diameters of pipelines. The tested ball valve was presented in Fig. 1. In the housing
(8) there is located a ball (10) controlled with the spindle (4). As the most important factor of valve functionality was tightness, interpreted as internal and external, the sealing rings (9) and the stuffing box (1) with fabric rings (2) are inevitable parts of each ball valve. The higher contact pressure between the ball and the sealing rings the higher tightness level. However higher contact pressure can cause operational torque increase. One should bear in mind that finding balance between torque and tightness is a key factor. Applied sealing material should have suitable parameters because the mechanical, chemical, and wear resistance is required. Leakage aspects will be discussed in separate article.

Figure 1: Construction of ball valve to be tested
1 – stuffing box, 2 – fabric ring, 3 – supporting ring, 4 – spindle, 5 – distance bushing, 6 – washer, 7 – screwed gland, 8 – housing with flange, 9 – PTFE seat, 10 – ball, 11 – locking sleeve with flange

2.2 The aim and scope of the work

The main aim of the work was the quantitative assessment of the wear process of the sealing ring made of pure PTFE cooperating with steel ball of the industrial valve. The three variants of the sealing PTFE ring differing in shape were taken into consideration. In the first stage the numerical simulation by means of Finite Element Method was used to determine the contact pressure between the steel ball and PTFE seal ring. In the second stage the experimental test was carried out to determine the friction resistance of the ball and assessment of the roughness parameter of the sealing ring surface before and after of 1000 opening and closing cycles the valve.

2.3 Sealing variants

The most popular sealing material is PTFE. However pure PTFE has high level of thermal expansion and high creep. Another drawback is low mechanical strength. Taking under consideration mentioned weak features there are used various fillers that improve some of material properties. Three types of PTFE seals were designed for testing (Fig. 2). Such a sealing ring were produced on CNC lathe machine. Shape of sealing ring should provide good tightness, good running-in, and low friction force. In many cases manufacturers use the flat conical shape of sealing seat [12]. During running-in process shape of the surface changes into concave as a result of cooperation with the ball. Higher values of contact pressure at the beginning, decrease during the wear process. Another solution assumes concave shape of the sealing face from the beginning [13].

Figure 2: Types of seal construction were considered in tests with various geometry of contact area; a) concave shape – variant 1, b) convex shape – variant 2, c) concave shape with peripheral groove – variant 3

In this case contact pressure is less likely to decrease during running-in process. Hence expected durability should be longer. For operational reasons the tightness and possible low friction force is required.

3 FEM analysis of ball and seal contact zone

3.1 The numerical model

In order to assess the pressure contact distribution between the ball and sealing ring the Finite Element Method
was used. Based on the symmetry plane crossing the valve only one-fourth of the model was subjected to analysis.

In the Fig. 3 the axi-symmetrical model was presented. The whole assembly was discretized with planar element type PLANE 183 with higher order shape function.

The mesh density in the contacting region was increased. Model in this configuration consisted averagely of the 10000 nodes and 3500 elements. The contact between element was discretized with CONTA 172 and TARGET 169 element using frictional behavior with friction coefficient 0.05. The material property of the ball was modeled with Young moduli 206000 MPa and poison ratio 0.3 (structural steel). For PTFE ring the biaxial isotropic model describing with relationship between stress and strain was used in the form presented in the Fig. 4. The Poisson ratio for PTFE was 0.42. The boundary conditions were shown in the Fig. 5. The upper edge of the ring was fixed to simulate the embedded sealing in the housing element of the valve. To simulate the thrust force the lower edge of the ball was loaded with pressure equals to 8 bar (air pressure during valve operation). The analysis was carried out in ANSYS 15.0 software using static structural issue.

3.2 Numerical results

Figures from 6 to 8 present the result in the form of the contact pressure along with contacting line between the ball and seal. In case of seal variant 1 (Fig. 6) the contact pressure distribution was not uniform. The concentration of the pressure value appeared in vicinity of the edge of seal in upper part of contact area and equals 4.71 MPa. The average value of the contact pressure was 2.61 MPa.

In seal variant 2 (see Fig. 7) one can be notice that convex shape caused that the pressure distribution was more concentrated. This is typical pressure distribution for Hertz issue. The maximum value in this case was 15.8 MPa and average was 8.4 MPa. In this variant the higher pressure value can provides the bigger tightness, but could increase wear rate. Thus, the durability of seal variant 2 may
be shorter comparing with seal variant 1. In seal variant 3 (see Fig. 8) pressure distribution was similar to variant 1 with stress concentration appeared in the vicinity of edges. The maximum stress equals to 6.58 MPa and average value was 3.65 MPa.

Figure 7: The contact stress distribution between the ball and seal variant 2

Figure 8: The contact stress distribution between the ball and seal variant 3

In this the wear could be slower in comparison to variant 2. Groove made in this seal caused reduction of contact area, and hence increase of pressure value.

4 Experimental tests

In order to assess the wear rate of the particular variants of the seals the experimental test was performed.

4.1 Test stand

Figure 9 shows the test rig that main element is the tested valve placed on the frame in horizontal position. Two flanges of the valve were blinded, whereas one of them was drilled and joined with the hose supplying the air pressure from the reservoir. The valve’s spindle was connected to the rod of the driving gear by means of the mechanical coupling. The end of the spindle was equipped with the torque measuring unit which enabled the recording of the torque value during the opening and closing the valve. The number of cycles were registered too.

Figure 9: Test stand

4.2 Test procedure

In the first step the seals were mounted in the valve housing. After that the valve was subjected to pressurization of 8 bar, the thousands of opening/closing cycles were performed. The degree of wear was assessed in two aspects, macroscale and microscale. The assessing wear in microscale consisted of measuring roughness (parameter $Ra$) before and after the test, whereas macroscale included in measuring the weight of the seals before and after the test. Generally the wear process changes the micro geometry of both friction couple (seal and ball) however, in this paper the changes of the sealing material only were taken under consideration. Tylor Hobson Talysurf Model 120 stylus profilometer was used for measurement of roughness. The weight of the specimens was measured by means of electronic scale with the accuracy of 1 mg. Test whole procedure was the following:

- Measurements of roughness parameter of the PTFE seal before tests.
- Weight of sealing rings before tests.
- Valve assembly.
• Torque of operation measurements (close/open) under pressure of 8 bar.
• Performing 1000 cycles of closure/opening.
• Torque of operation measurements (close/open) under pressure of 8 bar.
• Measurements of roughness parameter of the PTFE seal after tests.
• Weight of sealing rings after tests.

4.3 Tests results

4.3.1 Torque measurement

In Fig. 10 the comparison of friction torque values for seal variant 1 is presented. Green line shows first 6 cycles from 1000, whereas red line shows last 6 cycles from 1000.

At the beginning of the test the value of the friction torque equaled 35.6 Nm was characteristic for opening and 46 Nm for closure operation. Much lower value was obtained after 1000 cycles. In final cycles the friction torque at opening was equal to 16.2 Nm and to 18.8 Nm during closure operation.

The results of seal variant 2 is presented in Fig. 11.

At the beginning of the test the friction torque during opening was equal to 35.8 Nm and during closure was equal to 24 Nm. After 1000 cycles the friction torque during opening was nearly the same at initial cycles, whereas during closure the torque decreases to value of 10 Nm.

The results of seal variant 3 is presented in Fig. 12.

At the beginning of the test the friction torque value during opening was equal to 13 Nm and to 13.8 for closure. A little lower value of torque necessary to close/open a valve after 1000 cycles indicates that running-in process caused decreasing of contact pressure between ball and the seal. In this case it was measured 9 Nm during opening and 15.1 Nm at closure. Not so much noticeable change in torque value proves that seal variant 3 was less susceptible to wear process comparing with variant 1.

4.3.2 Roughness measurement

The roughness parameters of PTFE seal were measured before and after tests of valve operation. In the case of original seal (before tests) measurement was performed in one place on the seal perimeter, because of uniformity of machined material. After 1000 cycles measurement of seal surface roughness was performed in place corresponding to conventionally called “equator” and “pole” of the ball. Roughness was measured in direction perpendicular with respect to ball motion. In Fig. 13 and 14 there are pre-
sented the exemplary profilogram charts of original and worn seal, respectively.

The measured roughness parameters of tested seals were listed in the Table ?? Data presented in the Table 1 indicates that seal made from pure PTFE wear relatively fast, so the process of running-in is efficient in short period of time. The roughness parameters on the level of 0.01 µm are typical for polished surface. In Fig. 15 the views of the seal variant 1 before and after the tests are presented.

4.3.3 Weight measurement

Before and after the durability tests the weight of PTFE seals was measured. The weight loss can be a parameter indicating the degree of wear. In the Table ?? the measurements results are presented.

| Description | Ra before, µm | Ra after, µm |
|-------------|---------------|--------------|
| Variant 1 p | 7.9940        | 0.2797       |
| Variant 1 e | 4.3000        | 0.1437       |
| Variant 2 p | 3.6978        | 0.2228       |
| Variant 2 e | 10.8940       | 0.1213       |
| Variant 3 p | 0.8471        |              |
| Variant 3 e | 0.8471        |              |

Table 2: Weight of seal before and after tests.

- **Seal variant 1**: 82.085 g before, 81.275 g after, Δ = 0.987%
- **Seal variant 2**: 81.530 g before, 81.020 g after, Δ = 0.626%
- **Seal variant 3**: 82.260 g before, 81.730 g after, Δ = 0.644%

5 Summary of the work

Measured values of friction torque prove that the biggest drop in torque value was in case of standard typical seal variant 1 (decrease about 50 %). In case of seal variant 2 the reduction of torque during closing was approximately 45%. The seal variant 3 after 1000 cycles of opening and closing caused torque drop during opening of 36%.

Comparing the roughness parameters during wear process a considerable reduction of roughness parameter $Ra$ can be noticed. The roughness measurement in equator direction indicated the higher intensity of wear comparing with a pole zone. Also the seal located on the outlet side was exposed to more intensive wear as the pressure generates higher load on this side comparing with the inlet side. In the seal of variant 1 a 94% drop of $Ra$ parameter was measured. In the seal of variant 2 a 93% drop of roughness value was measured, and a 94% reduction in the seal of variant 3. Smaller value of $Ra$ parameter after test was
measured in the vicinity of pole zone. This phenomenon can be explained by possible higher value of contact pressure in this region. However higher linear speed and longer path of friction is in equator zone.

Force resulting from air pressure (0.8 MPa) acting on ball gives 1055 N. In seal of variant 1 contact area was equal to 565.5 mm². Calculated contact pressure was equal to 1.87 MPa. Numerical model gave value of 2.61 MPa (average pressure on the length of ball-seal contact). In variant 2 the contact area was much smaller as two surfaces were convex. Calculated value subjected to deformation was equal to 148 mm² that gave calculated contact pressure of 7.12 MPa. Numerical model gave the value of 8.4 MPa. In variant 3 calculated contact area was equal to 506 mm². Calculated contact pressure was equal to 2.08 MPa. Numerical model gave average value of 3.65 MPa. Both calculated and numerically obtained values of contact pressure were in correlation with leakage value. Although the tightness measurement results were not included in the article. The smallest contact area of seal variant 2 caused the highest contact pressure and, therefore, the possibly of the highest damage risk.

The weight of seal in each variant was measured. It was noticeable weight loss in each case, however the biggest weight reduction was noticed in the seal of variant 1 (about 0.98%), much smaller weight loss was measured in seal of variant 3 (about 0.64%) and the smallest difference of weight was in case of seal variant 2 (0.62%). Taking under consideration the convex shape of seal variant 2, even small wear (small weight loss) can produce high drop of contact pressure and of the torque. High contact pressure in the seal of variant 2 results in increased creep comparing with seal of variant 1 and 3, what can be drawn from properties handbook [7]. The highest wear degree was expected in the seal of variant 2, and hence the most noticeable drop of torque value was preliminary expected. Also the highest weight loss was expected in the seal of variant 2 due to wear, as the highest contact pressure occur in this design, however the test provided opposite results.

In basic friction theory the contact area of sliding bodies was negligible. However in this case, there was no direct correlation between contact pressure and friction torque. As it was noticed, the smaller contact area, the smaller torque. This indicates a much more complex mechanism of friction taking into account phenomena such as adhesion and deformation.

6 Conclusions

Previous studies may be assumed as follows.

Influence of seal shape on torque. Shape of the ball seals affects the torque and wear intensity. Experiment indicates that the smaller the contact area, the lower the torque value.

Influence of seal shape on wear rate. The biggest weight loss was observed in variant 1, smaller in variant 3 and the smallest in variant 2.

Taking under consideration all results obtained during investigation, the most interesting seal shape could be variant 3, because it doesn’t generate as high contact pressure as variant 2, but guarantees much lower wear rate than variant 1. However the variant 2 provided the most surprising results in several respects. Further research requires to explain the mechanisms of cooperation and wear. The obtained results will be the basis for finding more optimal shape of the contact area of cooperating parts.

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