Evaluation of the wear and corrosion resistance of coated parallel gate valve

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Abstract

This paper aims to experimentally evaluate the wear and corrosion resistance of the coated active parts surfaces of a cut off valve. This paper presents the electrochemical corrosion, hardness and wear tests of the gate valve coated surfaces with a plastic material, in order to evaluate the seat and gate valve couple behaviour in an aggressive medium containing crude oil with sand particles. Uncoated and coated surfaces were compared and evaluated by the coefficient of friction, wear rates and corrosion resistance. A comparison process between the previous results of the CFD analysis for erosion wear and the test results, at the covered gate valve, is also presented.

1. Introduction

Industrial valves are special devices fixed on pipes, tanks and installations designed for transport, stoking of distribution of fluids with the purpose of closing, opening, distribution of fluids flow or controlling or adjusting flow parameters of fluids [1]. Due to exceptional technical and economical consequences for safety working the adjusting and closing elements of valves must fulfil the operating demands and wear and corrosion behaviour must be known. Gate valves, widely used in the production and transportation of oil, operate in harsh conditions. In particular, crude oil extracted from the ground contains formation water, dissolved gases and sand. Because of this, seat rings and gates are not only exposed due to severe wear but are also frequently attached. Cut off valves are operated when is necessary, normally twice a year, at high pressures, in an aggressive medium, and between seat and gate appear adhesive and corrosion wear, which could increase friction force at a value higher than was calculated at the valve design, and to make the valve inoperable. To prevent this phenomenon, it

is recommended to coat the friction contact surface with another material with excellent resistance to corrosion and wear.

From this, Badicioiu et al. conducted an experimental study to determine a welding process necessary for weld overlay using hard metals with good resistance to wear and corrosion on the surface of the valve seat to increase the durability of the valve [2]. But, the geometric shape or metallographic structure of the valve can change due to excessive heating. Also, a thin coating cannot be done with this method.

Through an experimental study to increase the wear resistance of the surface of the valve parts using HVOF spraying, Ripeanu et al. confirmed the influence of the surface roughness on the adhesion resistance and proposed a method to increase the adhesion resistance [3]. Caltaru et al. conducted a study to deposit with WC10Co4Cr material with an average powder size of $22 - 62$ μm on the surfaces of the gate and the valve seat using HVOF spray equipment [4], and these studies were also reported in the literature [5-7]. However, this method of thermal spraying, as well as the weld overlay method, is very likely to cause metallographic structural changes and microcracks due to the high heating temperature and the rapid cooling rate.

From this, Frunzaverde et al. conducted an experimental study to prevent cavitation erosion of the hydraulic turbine by using an elastomer coating [8]. However, the elastomer is not suitable as a coating material for the valve because of its weak resistance to heat and ultraviolet rays.

Everslik, it is a polymeric lubricant lake, made from thermosetting resins such as phenolic resin or epoxy resin, and has excellent corrosion resistance in all environments. Especially, this lubricant has good friction behaviour, because different solid lubricants such as graphite, MoS_2, PTFE (polytetrafluoroethylene) are scattered in it.

Arunachalam et al. conducted a study to coat electronic circuit boards with a lubricant made of epoxy resin and graphene, which are thermosetting resins [9]. Of course, because the object of the application is an electronic circuit board, it may be different from the valve, but its characteristics as a lubricant were well shown.

For the petroleum industry, in the equipment design, is recommended to make a FEM simulation (ex: standard DNVGL-RP-O501) followed by an experimental test to validate simulation results. From this, in literature [10], the wear analysis results of gate valve coated with Everslik which were obtained using COMSOL Multiphysics 5.4 were presented in detail (Table 1).

Table 1. The analysis results according to the open state of the gate valve coated with Everslik with a thickness of 25 μm [10]

Valve open state	Liquid velocity, m/s	Liquid pressure, MPa	Mass loss $\times 10^{-7}$, kg/m^2	
			Finnie model	Oka model
State 1-4	122	6.22	10.2	22.6
State 2-4	67.5	1.79	7.59	4.38

As can be seen in Table 1, through the simulation, was revealed that the lubricating film has a great influence on the reduction of the erosion rate of the valve, and the effect is more pronounced as the passage area of the fluid is small. However, since these results of the erosion analysis are only the theoretical analysis, the accuracy of the analysis results must necessarily be verified by real experiments. From this, in this paper, the results of the comparison between the values of the wear test with the help of the CSM tester and the values of the erosion analysis presented in the literature [10] are presented.

ASME B16.34 impose the maximum thickness of the layer to be 25 μm [11]. The coefficient of friction of the surface plays a very important role in reducing wear. Of course, the hardness of the surface also affects the wear resistance.

The aim of the paper was to validate the results of FEM simulation obtained by the authors by experimental results. In this paper, it is proposed to improve the corrosion resistance and wear of the gate surface using Everslik type plastic material coating. In addition, the author presents the hardness and wear tests performed to assess the effect of surface hardness and coefficient of friction on wear. Tests are also presented to assess the relationship between the wear rate and the load.

2. Materials and methods

2.1 Materials for electrochemical corrosion test

For the samples, the gate material (AISI 4130) was used and as solid lake lubricants, Everslik 1201 and Everslik 1301 were used. The samples were coated to a thickness of 25 μm in accordance with ASME B16.34 requirements for non-metallic coating. The samples were mechanically processed in the form of a disk having $\varnothing\ 16_{-0.1}$ mm, and a thickness of 2 mm. The sample coated with Everslik 1201 and Everslik 1301 is shown in Figure 1.

Figure 1. Coated sample

Figure 2. Potentiostat Voltalab 10

Figure 3. Corrosion cell

The electrolyte used for the electrochemical corrosion test is formation water. Figures 2 and 3 present a corrosion cell and potentiostat Voltalab 10. The formation water, which is mineral water extracted together with crude oil or natural gas, is composed of most chlorides with a small amount of complex salt solutions. For the corrosion test, formation water containing 10 % sand with a grain size of 0.077 mm was used. Corrosion cell works with a saturated calomel reference electrode, graphite counter electrodes, and specimen holder which expose 1 cm^2 of the specimen to the 1000 cm^3 test solution. The electrochemical corrosion test was performed according to the ASTM G5 standard [12].

2.2 Materials for hardness test

The materials used for the hardness test are the same as those for the corrosion test. The hardness test was performed using a Vickers hardness tester of the type EMCOTEST. For the hardness test, at each sample (coated sample and uncoated sample) hardness was measured in 3 − 5 points and surface hardness was determined as an average value. Vickers hardness test was performed according to the ASTM E92 standard [13].

2.3 Materials and methods for wear test

The samples used for the wear test were processed to the dimensions Ø 30 × 5 mm. For the wear test, a CSM tribometer (ball-on-disk tribometer) was used. The friction couple were ball on disk or cube on disk.

The parameters and conditions for the wear test by the CSM tester have been set as follows: temperature: 20 °C; relative humidity: 44 %; test medium: air or formation water; radius of rotation of the ball on the surface of the disk: 5 to 12 mm; rotational velocity: 0.1 m/s; friction length: 50 to 100 m; normal load: 0.5 to 2 N; disk material: AISI 4130; lubricants: Everslik 1201 and Everslik 1301; ball material: 100Cr6; ball diameter: 6 mm; cube material: AISI 4130 coated or uncoated with Everslik; cube dimensions: 4 × 4 × 4 mm.

The wear test was performed on the uncoated and coated samples, respectively, in air and formation water conditions. Wear tests depending on the type of sample, and the medium, are shown in Figure 4.

After the test, the tracks of balls and the wear track widths of disks corresponding to each sample were measured using an electron microscope.
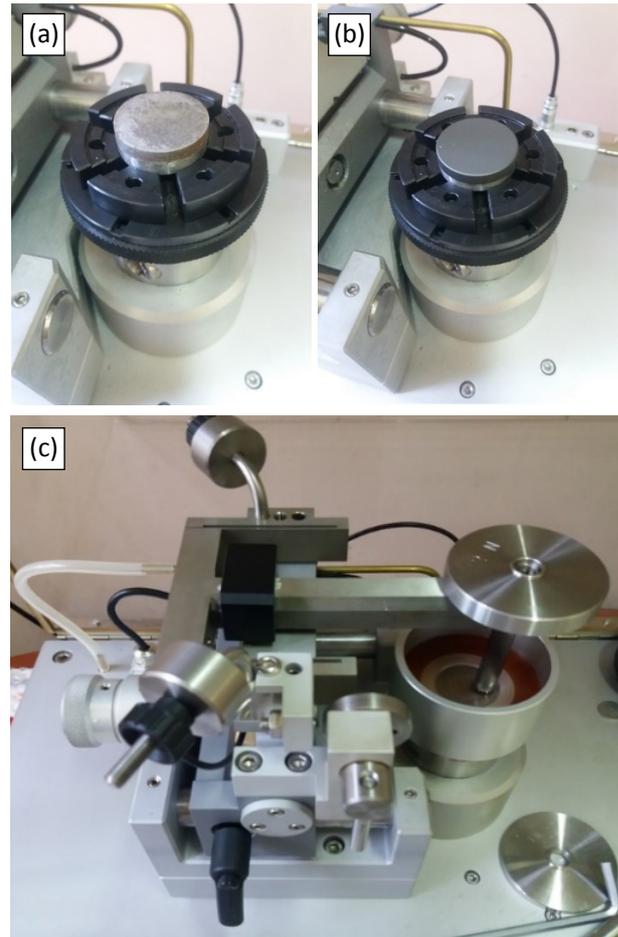

Figure 4. Wear test by CSM tester: (a) uncoated sample in air, (b) coated sample in air and (c) coated sample in formation water

2.4 Calculation of the amount of wear of the ball and the disk

The ball is worn as shown in Figure 5. Based on Figure 6, the volume of a ball lost per unit length and load can be calculated with the equations [14,15]:

$$WR_{\text{ball,volume}} = \frac{V_{\text{ball}}}{L_{\text{f}}F_{\text{n}}}, \qquad (1)$$

$$V_{\text{ball}} = \frac{\pi h}{6}\left(\frac{3}{4}d^2 + h^2\right) = \frac{1}{3}\pi h^2(3R - h), \qquad (2)$$

where $WR_{\text{ball,volume}}$ is the wear rate of the ball (mm^3/Nm), V_{ball} is the volume of the ball wear (mm^3), h is the height of the worn part (mm), d is the ball wear scar diameter (mm), R is the ball radius (mm), L_{f} is the friction length (m) and F_{n} is the applied load (N).

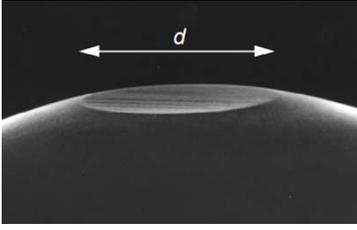The amount of disk wear can be determined using Figure 7.

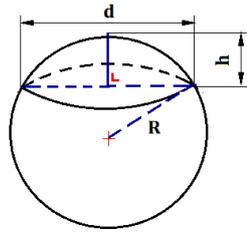**Figure 5.** Wear track of ball seen through an optical microscope [14]

Figure 6. Geometric shape of the ball

The wear rate of the disk, $WR_{disk,volume}$ is:

$$WR_{disk,volume} = \frac{V_{disk}}{L_f F_n}, \qquad (3)$$

$$V_{disk} = 2\pi R_{track} \cdot$$

$$\cdot \left[R^2 \arcsin\left(\frac{d_{track}}{2R}\right) - \frac{d_{track}}{4}\sqrt{4R^2 - d_{track}^2} \right], \quad (4)$$

where $WR_{disk,volume}$ is the wear rate of the disk (mm^3/Nm), V_{disk} is the volume of the disk wear (mm^3), d_{track} is the wear track width of the disk (mm) and R_{track} is the wear track radius of the disk (mm).

The disk gravimetric wear per unit area can be determined using Figure 8. This transformation was necessary to evaluate the FEM simulation results with experimental results.
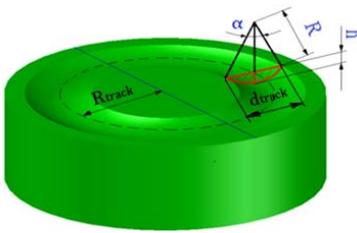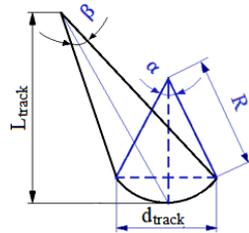

Figure 7. Geometric shape of the worn sample

Figure 8. Calculation scheme for worn surfaces

$$WR_{disk,mass} = \frac{V_{disk} \cdot \rho_{disk}}{A_{track}}, \qquad (5)$$

$$A_{track} = L_{track} R \arcsin\left(\frac{d_{track}}{2R}\right), \qquad (6)$$

where $WR_{disk,mass}$ is the gravimetric wear per unit area (kg/m^2), ρ_{disk} is the density of the disk (kg/m^3) and A_{track} is the total area of the worn part by the

rotational movement of the ball for the friction time (m^2).

3. Results and discussions

3.1 Electrochemical corrosion test results

To evaluate the corrosion resistance of the solid lubricant lake, the coated and uncoated samples were tested separately. The electrochemical corrosion test was performed for each sample. The test results are presented in Figures 9 and 10. Obtained electrochemical parameters are shown in Table 2. The test results, once again, confirmed that the coating significantly reduces the corrosion rate. The corrosion rate of the metal surface coated with Everslik is 8.61 µm/year, and the corrosion rate of the uncoated metal surface is 71.5 µm/year.
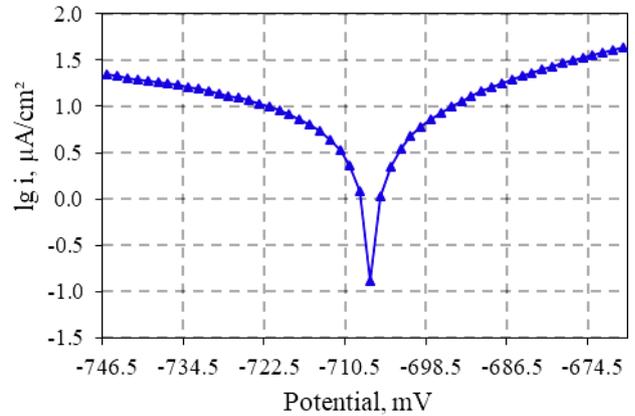

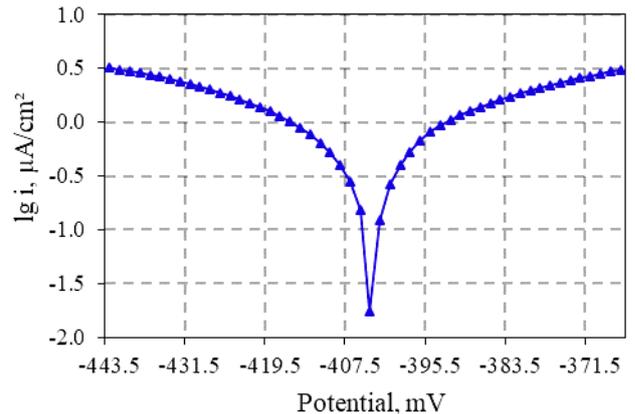**Figure 9.** Tafel's curve for the uncoated sample

Figure 10. Tafel's curve for the coated sample

Table 2. Electrochemical corrosion test results

Sample	Density, kg/m^3	Corrosion potential, mV	Corrosion current, µA/cm^2	Polarization resistance, kΩcm²	βa, mV	βc, mV	Corrosion rate, µm/year
Uncoated sample	7850	− 707.4	6.1528	1.36	42.6	− 67.8	71.5
Coated sample	7850	− 404.6	0.7404	11.86	58.4	− 57.4	8.61

3.2 Hardness test results

Vickers hardness measurement results are presented in Figure 11 and Table 3.
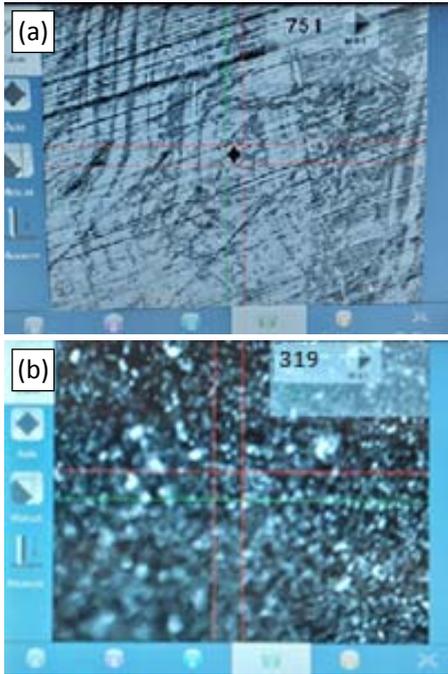

Figure 11. Images of hardness measurement: (a) uncoated sample and (b) coated sample

Table 3. Hardness measurement results for each sample

Sample	Load, N	Vickers hardness HV			
		1	2	3	Average
Uncoated sample	0.98	751	764	754	757.3
Coated sample	0.98	319	436	411	388.7

The test result showed that the surface hardness of the uncoated sample is nearly twice that of the sample coated with Everslik.

3.3 Wear test results

Using a CSM tribometer, according to the test conditions set out above, the wear tests were performed separately in air and in formation water for the coated and uncoated samples. The results obtained, according to each sample, are presented in Figures 12 and 13 and Table 4.

The wear values, according to the test conditions of the samples, calculated based on the experimental data measured using an electron microscope, are shown in Table 5 and Figures 14 and 15.
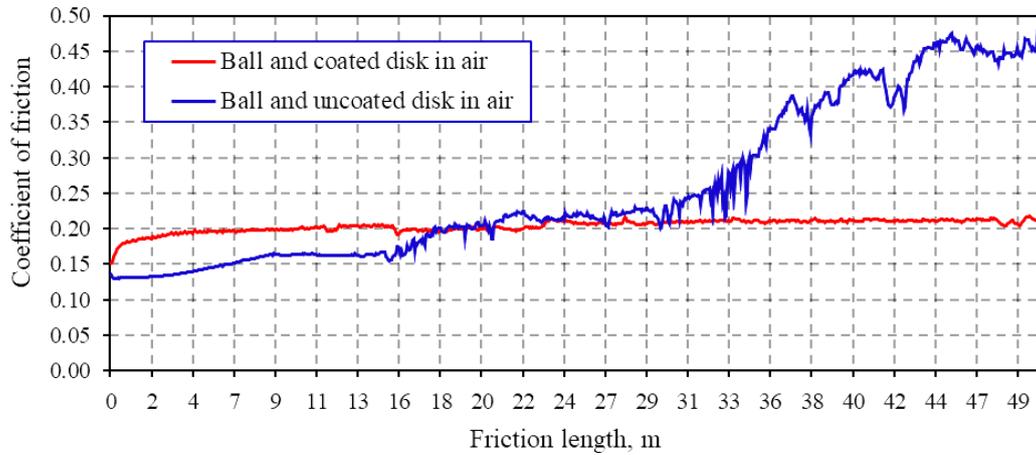

Figure 12. Friction coefficients vs. friction length for uncoated and coated samples tested in air
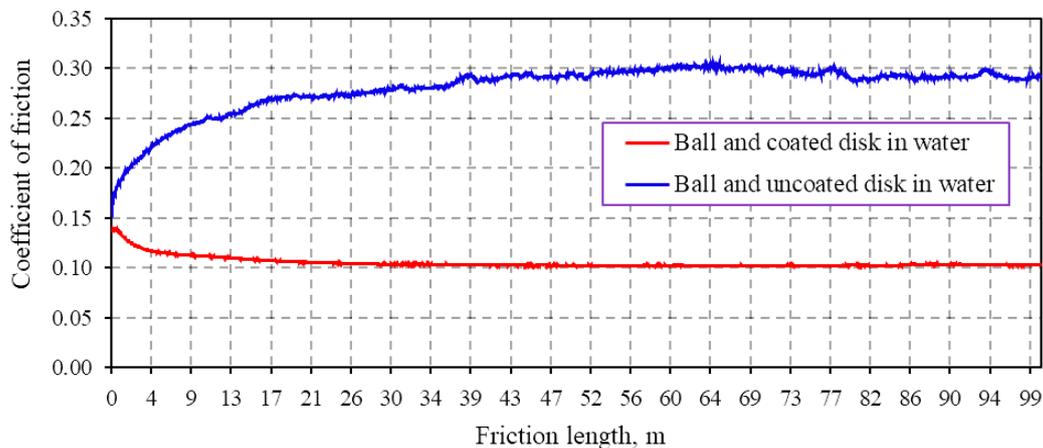

Figure 13. Friction coefficients vs. friction length for uncoated and coated samples tested in formation water

Table 4. Results of wear measurement

Test condition	Load, N	Friction length, m	Coefficient of friction			Ball wear scar, μm	Disk wear track, μm
			Minimum	Maximum	Average		
Ball and uncoated disk in air	2	50	0.1297	0.4759	0.2588	144	96
Ball and uncoated disk in water	2	100	0.1501	0.3050	0.2807	124	64
Ball and coated disk in air	2	50	0.1497	0.2176	0.2039	152	72
Ball and coated disk in water	2	100	0.1012	0.1242	0.1060	160	48

Table 5. Results of the calculation of the amount of wear

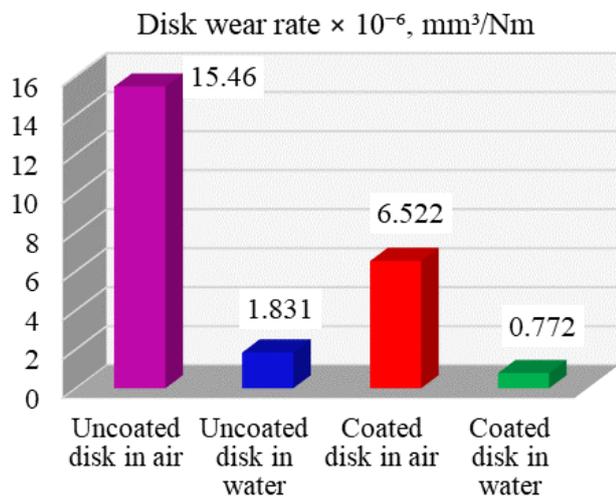| Test condition	Ball wear rate × 10^{-8}, mm^3/Nm	Disk wear rate × 10^{-6}, mm^3/Nm	Disk gravimetric wear per unit area × 10^{-7}, kg/m^2
Ball and uncoated disk in air	7.034	15.46	50.5085
Ball and uncoated disk in water	1.934	1.831	8.9791
Ball and coated disk in air	8.736	6.522	3.7712
Ball and coated disk in water	5.361	0.772	0.6704

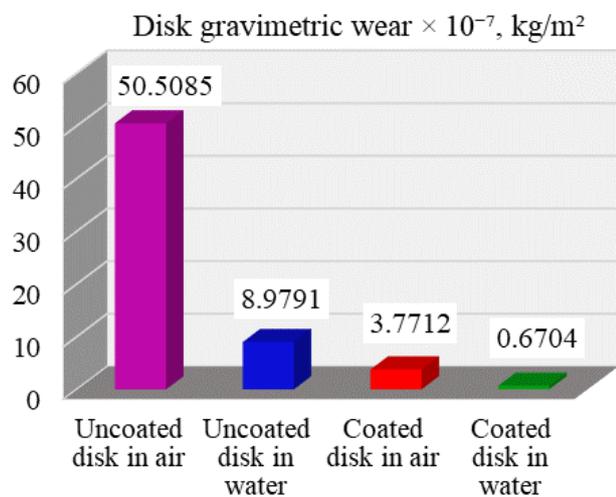**Figure 14.** Disk wear rate according to test conditions

Figure 15. Disk gravimetric wear according to test conditions

By the wear test with the CSM tribometer, it was confirmed that the wear of the coated disk with a lower coefficient of friction, is much lower than that of the uncoated disk with a higher hardness.

The measured friction coefficients correspond to the friction surface between the ball and the disk. The real friction at a parallel gate valve does not occur between the sphere and the plane, but between the two planes. Therefore, it was necessary to establish the coefficient of friction for a surface friction couple, between the cube and the disk couple, by assuming the real working conditions. The coefficients of friction between the cube and the disk vs. friction length according to the test conditions are presented in Figure 16 and Table 6.

As can be seen in Figure 16, the coefficient of friction between the coated disk and the coated cube in formation water is the most stable and smallest. The coefficient of friction in this case is 0.085.

The influence of the load on the wear characteristics of the coated samples was observed, applying loads by 0.5 from 0.5 to 2 N. The results of the measurement according to different loads are presented in Figure 17 and Table 7.

As can be seen in Table 7, there was a slight difference between the maximum and minimum values, but there was no significant difference in the coefficients of friction of the coated samples depending on the change in load when viewed as an average value.

Next, the wear amounts, calculated using the data in Table 7 and the Equations (1), (3) and (5) are shown in Figure 18 and Table 8.

Table 6. Friction coefficients between cube and disk depending on test conditions

Test condition	Load, N	Friction length, m	Sliding speed, m/s	Coefficient of friction		
				Minimum	Maximum	Average
Coated disk and uncoated cube in water	2	100	0.1	0.112	0.185	0.149
Coated disk and coated cube in air	2	100	0.1	0.134	0.303	0.234
Coated disk and coated cube in water	2	100	0.1	0.077	0.122	0.085

Table 7. Results of wear measurement depending on the load

Load, N	Time, s	Coefficient of friction			Ball wear scar, µm	Disk wear track, µm
		Minimum	Maximum	Average		
0.5	1000	0.1509	0.2803	0.2498	100	80
1	1000	0.1526	0.2619	0.2456	188	88
2	1000	0.1378	0.3122	0.2487	160	80

As can be seen from Figure 18, the volume of sample lost per unit load and length decreased with increasing load. In other words, the increase in the volume loss per unit load tends to decrease with increasing load. Of course, as the external load increases, the volume lost also increases. However, because the change in volume loss is too small compared to the amount of change in load, as a result, the volume loss per unit load decreases as the load increases.
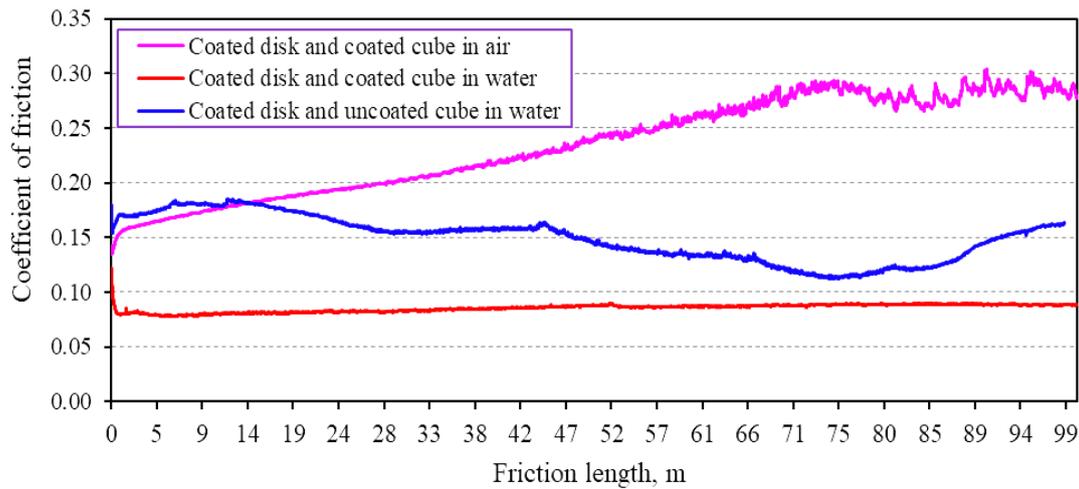

Figure 16. Friction coefficients vs. friction length between the cube and the disk according to the test conditions
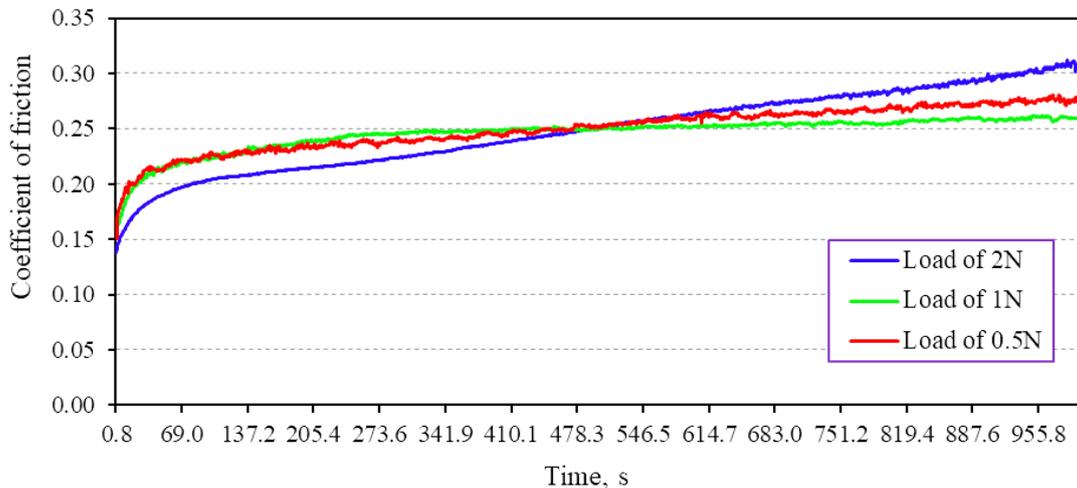

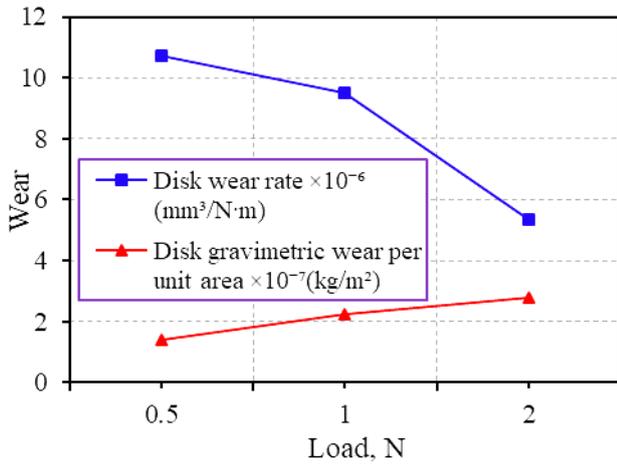**Figure 17.** Coefficients of friction of the coated sample tested in air

Figure 18. Coated disk wear depending on load

Table 8. Results of calculation of the amount of wear depending on the load change

Load, N	Ball wear rate × 10⁻⁸, mm³/Nm	Disk wear rate × 10⁻⁶, mm³/Nm	Disk gravimetric wear per unit area × 10⁻⁷, kg/m²
0.5	3.2728	10.7239	1.3967
1	20.4467	9.5158	2.2534
2	5.3629	5.3619	2.7935

However, in the case of the disk gravimetric wear, as the load increases, the volume loss of the disk becomes larger than the area of the part worn by the ball rotation, so disk gravimetric wear per unit area increases with increasing load.

3.4 Comparison of theoretical results with those obtained by the CSM tester

The wear test results by the CSM tester and the CFD analysis results (Table 1) were compared. For comparison, the correlation between applied load and gravimetric wear per unit area was calculated and it is shown in Figure 19.

That is, the correlation between load change and wear mass per unit area is (× 10⁻⁷ kg/m²):

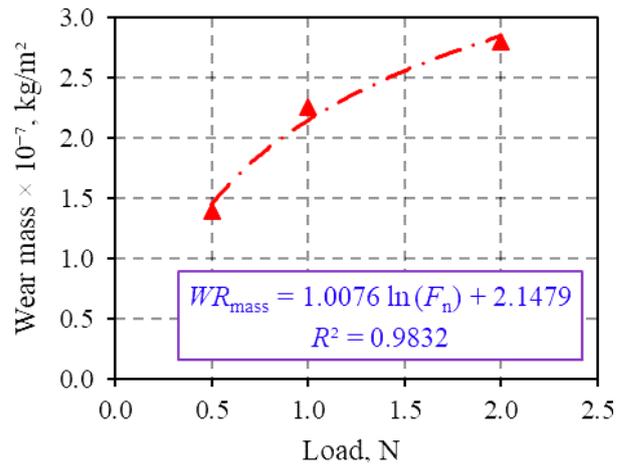$$VR_{mass} = 1.0076 \ln(F_n) + 2.1479. \tag{7}$$

Figure 19. Correlation between applied load and gravimetric wear per unit area

On the other hand, the force acting on the worn surface can be calculated with the equation:

$$F_n = P_{valve} \cdot S_{wear}, \tag{8}$$

where F_n is the external load (N), P_{valve} is the internal pressure of the valve depending on the open state of the gate (MPa), S_{wear} is the worn surface area of the disk (mm²).

Worn surface area can be determined using Figure 7. The worn surface area is calculated with the equation:

$$S_{wear} = 4\pi R_{track} R \arcsin\left(\frac{d_{track}}{2R}\right). \tag{9}$$

Mass losses according to the open state of the valve calculated using Equations (7) and (8) are shown in Table 9 and Figure 20.

Through comparison, it was confirmed that the smaller the passage area of the fluid, the smaller the difference between the test value and the theoretical value. Of course, the theoretical value corresponds to erosion wear and the test value corresponds to friction wear, so differences in the results are inevitable. Nevertheless, the analysis values using the Finnie model were very similar to the test results than the Oka model. When the

Table 9. Comparison between experimental and theoretical values

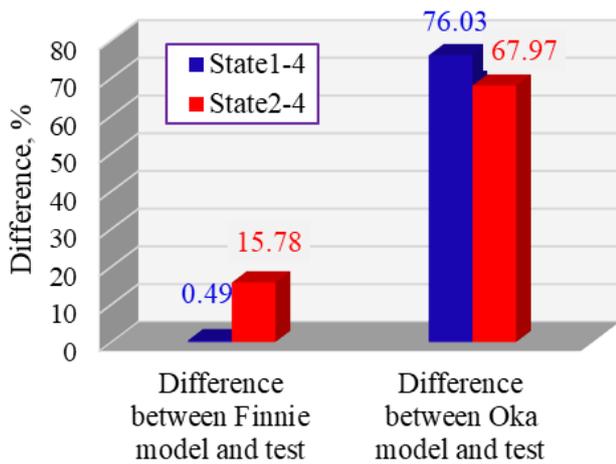| Valve open state	Pressure, MPa	The difference between the experimental value and theoretical value through the Finnie FEM model simulation			The difference between the experimental value and the theoretical value through the Oka FEM model simulation		
		Experimental	Theory	Difference, %	Experimental	Theory	Difference, %
		× 10⁻⁷, kg/m²			× 10⁻⁷, kg/m²		
State 1-4	6.22	10.15	10.2	0.49	10.15	22.6	76.03
State 2-4	1.79	8.89	7.59	15.78	8.89	4.38	67.97

Figure 20. Comparison between theoretical and experimental values

valve was opened by a quarter or a half, the differences between the experimental results and the Finnie model was 0.49 % and 15.78 %, respectively. However, in the case of the Oka model, the differences were very large, of 76 % (state 1-4) and 67 % (state 2-4), respectively.

4. Conclusions

Everslik lubricant lake significantly reduces the corrosion rate of the surface. The corrosion rate of the metal surface coated with Everslik is 8.605 μm/year, and the corrosion rate of the uncovered metal surface is 71.5 μm/year.

As a result of the Vickers hardness test, the average hardness of the surface coated with Everslik is 388.7 HV, and the average hardness of the surface of the gate material without coating is 757.3 HV.

The coefficient of friction of the metallic surface covered with Everslik is smaller than that of the uncovered metallic surface. As a result of which the coefficient of friction between the coated cube and the coated disk was measured assuming the real working conditions, the coefficient of friction of their contact surfaces in the formation water is 0.085.

The wear test with the CSM tribometer confirmed that the mass loss of the coated disk with a lower coefficient of friction is much lower than that of the uncoated disk with a higher hardness. In other words, in order to reduce wear losses, it is more important and more effective to reduce the coefficient of friction than to increase the surface hardness of the material.

The difference between the wear test result and the theoretical analysis result is relatively small, and in particular, the analysis value using the Finnie model is closer to the test value. For the

Finnie model, the differences are 0.49 % (state 1-4) and 15.78 % (state 2-4), respectively, and for the Oka model, the differences are 76 % (state 1-4) and 67 % (state 2-4), respectively.

This simulation method, for which the accuracy of the analysis was confirmed by comparing the results, can be applied not only to gate valves but also to other valves and all objects (products made of metal) for the purpose of erosion research.

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