Analysis of the Wear Influence on the Dynamic Performance of Electrohydraulic Servovalves

Aristia-Ioana POPOVICI
Technical University of Civil Engineering of Bucharest, Bucharest, Romania, arys_jo@yahoo.com

Abstract: - This study conducts an analysis of the impact of specific wear on the servovalves' quality indicators. The performance of servovalves is closely linked to geometric changes resulting from component wear. Based on Finnie's models, the value of erosive wear was determined, and the variation in radial clearance due to coupling wear was quantified. Numerical simulation was performed using MATLAB-Simulink to analyze how the variation in clearance over time affects the servovalves' performance. The results obtained for the two servovalves studied reveal a different, yet significant influence of clearance changes on quality indicators and, consequently, affect the optimal functioning of servovalves.

Keywords: - servovalve, abrasive-erosive wear, radial clearance, quality indicators, modeling-simulation.

1. INTRODUCTION

Servovalves (SV) are critical components in control and precision systems, with applications spanning a diverse array of fields. These include general-purpose servo controls, tracking systems, and flight control systems for high-speed aircraft and missiles, as well as the construction of industrial and humanoid robots. Additionally, servovalves are extensively utilized in the development of the most modern and efficient construction machinery, owing to their reliability and high performance [1-8].

Servovalves facilitate continuous information transfer by converting electronic signals into hydraulic signals. This process involves transforming an electric control signal, such as current or voltage, into output parameters, such as flow rate and pressure. The feedback mechanism between input and output consists of mechanical, hydraulic, and electrical components.

A servovalve typically consists of a main stage, referred to as the second stage, which includes a spool for flow regulation, and a pilot stage, known as the first stage, which serves as a hydraulic amplification system. These two-stage servovalves, featuring both preamplifier and hydraulic amplifier components, are the most prevalent for flow control, as illustrated in Figure 1 [9].

The first stage of the servovalve functions as an electromechanical converter, incorporating a torque motor with components delineated in Figure 1. The primary component is the movable armature, which is integrated with the flapper and the pre-amplifier's feedback spring. This feedback spring is characterized by its truncated-cone shape and a spherical head that contacts the spool's circular groove.

The hydraulic preamplifier employs a nozzle-flapper configuration that provides variable local hydraulic resistance depending on the flapper's position. The spool-type hydraulic amplifier functions as a mechano-hydraulic converter, converting displacement (input variable) into fluid flow (output variable). Depending on the position of the main spool, the pressure in channeled from pressure port **P** to control port **A,** while the oil returns from control port **B** to the tank **T.** Alternatively, the pressure is directed from port **P to B,** while the oil returns from **A to T**. Figure 1 illustrates the following components: **1** represents the torque motor, **2** denotes the nozzle-flapper, **3** indicates the feedback spring, **4** corresponds to the sleeve, **5** signifies the spool, **6** refers to the flexure tube, and **7** identifies the pilot bridge.

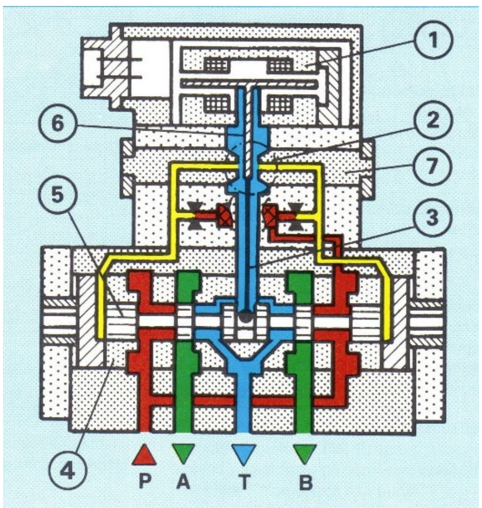


Figure 1. Principle diagram of a force-feedback servo system [9]

To ensure the correct and efficient functioning of the spool–sleeve assembly, which serves as the primary coupling of the hydraulic amplifier, it is imperative that it be manufactured with exceptionally precise dimensional, form, and positional tolerances, along with very low surface roughness. The typical radial clearance between the spool and the sleeve is within the range of 1 to 5 µm. The cylindricity tolerance is specified as 2 (µm) for the sleeve and 1.2 (µm) for the spool, while the surface roughness is $R_a = 0.05 \div 0.1$ [µm] [10].

Among the factors contributing to the degradation (wear) of servovalves, particularly those arising from characteristic friction and wear processes, are the following [11]:

- the presence of abrasive particles in hydraulic oil has negative effects on the performance of servovalves;
- the increase in the clearance between the spherical end of the reaction rod and the lateral surface of the circular channel within the spool has been estimated to range from 1 to 20 µm;
- an increase in the frictional force between the spool and the sleeve;
- an increase in the radial clearance between the spool and the sleeve.

The alteration in the clearance at the spool-sleeve interface is a direct consequence of abrasive erosion wear.

Previous studies [4, 6, 7, 11] have concentrated on the development and validation of a mathematical model for erosion wear in various regions of the spool-sleeve coupling. In particular, paper [4] proposes and experimentally validates a computational model for the erosive wear of the spool edge in the fluid input area.

The dynamic performance of servovalves is subject to degradation over time, primarily due to the wear of internal components. This wear affects critical parameters such as response time, stability, and frequency bandwidth, ultimately compromising system reliability and control accuracy.

This study is centered on the analysis of wear in electrohydraulic servovalves, with the objective of investigating how wear (affected by modification to the radial clearance of the spool-sleeve coupling) influences their dynamic characteristics.

This study introduces a novel computational model that incorporates clearance variation through erosion wear models, examining the impact of this modification on the dynamic behavior of electrohydraulic servovalves.

2. EROSION MODEL APPLIED TO THE SPOOL-SLEEVE COUPLING

The specific wear region of the hydraulic amplifier's spoole-sleeve coupling is characterized by abrasion, abrasive erosion, adhesion, and surface mechanical fatigue.

Abrasive wear is a multifaceted mechanical process that manifests in several forms: micro-chipping of metal surfaces by the asperities of the opposing surface or by the sharp edges of hard particles; fracture resulting from crack convergence; fatigue due to repeated plastic deformations; and the extraction of hard grains from the material [12].

Erosive wear is observed on friction surfaces subjected to the action of a moving fluid containing hard and fine solid particles. The severity of the erosive process is influenced by the velocity and direction of the fluid transporting the abrasive particles, as well as the mechanical properties of the surfaces undergoing abrasion. The impact of these particles is evident on the cylindrical surfaces of the spool and sleeve (Figure 2b), the active edges of the spool (Figure 2c), and the contour of the distribution windows of the sleeve, resulting in destructive effects.

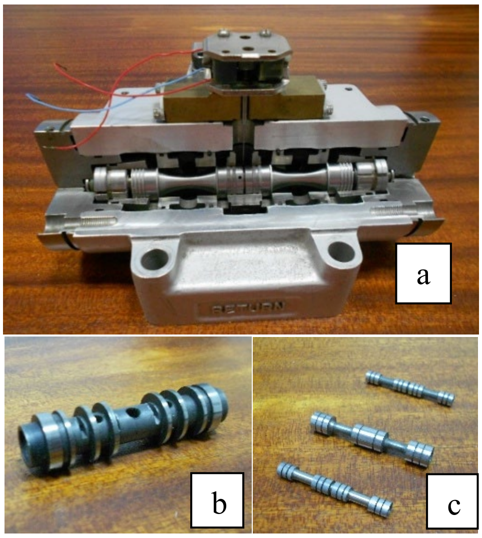


Figure 2. Electrohydraulic servovalve spool-sleeve coupling (a); the sleeve (b); the spool (c)

a) Quasi-static regime at the contact of the coupling parts

Wear processes result in alterations to the manufacturing clearance and the dimensions of flow control surfaces, which have predictable implications for the dynamic behavior of servovalves.

To examine the primary parameters influencing erosion wear under quasi-static conditions, the wear intensity (I_{er}) is employed. This is defined as the ratio of the mass of material removed by wear from the surface (m_{uz}) to the mass of abrasive particles that contributed to the wear (m_{ab}).

For spherical erosive particles, the formula [12, 13] for calculating erosive wear intensity is expressed as follows:

$$I_{er} = \frac{m_{uz}}{m_{ab}} = \frac{3}{4\pi} \cdot \frac{\rho_m}{\rho_{ab}} \cdot \frac{1}{n_{ab}} \cdot \frac{V}{r^3} \;\longrightarrow\; I_{er} = \frac{V_{ech}}{r^3} \quad (1)$$

where:

$$V_{ech} = \frac{3}{4\pi} \cdot \frac{\rho_m}{\rho_{ab}} \cdot \frac{V1}{n_{ab}} \quad (2)$$

for: V – the volume of worn material;
ρ_m – the density of the worn surface material;
r – abrasive particle size;
ρ_{ab} – density of the abrasive particle material;
n_{ab} – the number of abrasive particles striking the target surface.

If V_{ech} remains constant and the abrasive material is granular, with $r\epsilon(0, +\infty)$, the wear intensity I_{er} is significantly elevated for $r \rightarrow 0$ (very fine particles) and diminishes as particle size increases $r \rightarrow \infty$ (see Figure 3). I^*_{er} is the calculated erosive wear intensity based on a given value of r^*.

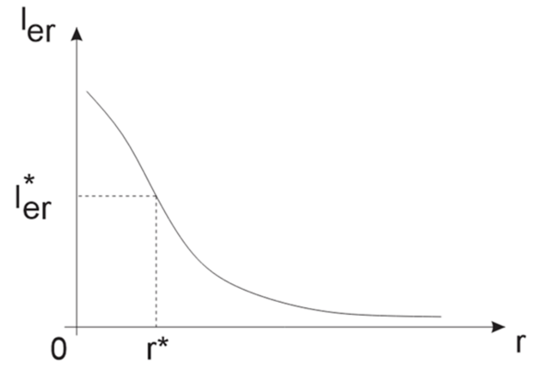


Figure 3. The influence of particle size on erosive wear intensity

Reference [13] examines the fundamental aspects of erosion and evaluates the impact of erosive wear parameters on the process's intensity.

b) Dynamic behavior of abrasive particles on the friction surface of the coupling

Theoretical and experimental investigations on erosion are based on specific mathematical models, among which the Finnie model, the Sundararajan model, the Bitter model, and the Hutchings model are the most frequently employed and offer the most precise description of the erosive mechanism [14-17].

In this study, we employed the Finnie model to simulate the erosion process [13]. The derivation of the calculation relationships within these models was based on the equation of motion for a solid abrasive particle with mass m, which impacts the target surface at velocity v and an incidence angle α.

According to this mathematical model, the intensity of erosive wear I_{er} is calculated using specialized formulas. These formulas consider two variants, which are distinguished by the percentage of particles with an abrasive effect.

The **initial Finnie model** [15] is characterized by the following structural relationships:

$$R_e = \frac{p_a \cdot \rho_m \cdot v^2 \cdot (1 - c_r^2)}{0.9272 \cdot HB \cdot \psi \cdot K_F} \cdot f(\alpha) \quad (3)$$

where:

$$f(\alpha) = \begin{cases} \left(\sin 2\alpha - \frac{6}{K_F}\sin^2\alpha\right), tg\alpha \le \frac{K_F}{6} \\ \left(\frac{K_F}{6}\cos^2\alpha\right), tg\alpha \ge \frac{K_F}{6} \end{cases}$$

where: R_e – dimensionless erosive wear intensity;
p_a – the percentage of abrasive particles with microcutting effects [kg/m^3];
ρ_m – target material density [kg/m^3];
v – particle impact speed [m/s];
α – the angle of incidence [degrees];
c_r – restitution coefficient;
HB – static hardness [N/m^2];
Ψ – the ratio of the contact length (L [μm]) and cutting depth (δ [μm]) of the impact area;
K_F – the ratio of the horizontal component (F_0 [N]) and the vertical component (F_v [N]) of the characteristic impact force.

For the **second Finnie model**, the proportion of abrasive particles with micro-chipping effects [16] is reassessed by adjusting the factor p_a from 50% to 10%.

The calculation formulas can be stated as follows:

$$R_e = \left(\frac{p_a}{2K}\right)\frac{\rho_m \cdot v^2 \cdot (1 - c_r^2)}{0.9272 \cdot HB} \cdot f(\alpha) \quad (4)$$

where:

$$f(\alpha) = \begin{cases} \left(\sin 2\alpha - \frac{8}{K_F}\sin^2\alpha\right), tg\alpha \le \frac{K_F}{8} \\ \left(\frac{K_F}{8}\cos^2\alpha\right), tg\alpha \ge \frac{K_F}{8} \end{cases}$$

The relationships underscore the significant influence of both the direction and velocity of the jet carrying abrasive particles, as well as the mechanical properties of the surface materials, on the severity of abrasive erosion.

To evaluate the impact of wear on the dynamic performance of servovalves, the clearance in the spool-sleeve coupling is quantified according to the intensity of specific erosive wear.

By applying the specified relations, the erosive wear intensity I_{er} is calculated for specific values of the influencing parameters. Subsequently, these values are utilized to determine the volume of worn material, V, in accordance with relation (1). The variation of the erosive wear intensity I_{er} is then graphically represented (figure 4 a – first model and b – second model) for a speed of $v = 0.5$ [m/s], as a function of the angle of incidence, with values between $\alpha = 0° \div 90°$, and for three different values of the coefficient of friction $\mu = 0.1;\ 0.5;\ 1$. The other parameters in the Finnie equations are constants (as specified in [13]), with the following values: $p_{a1} = 10^{-2}$ [kg/m^3], $p_{a2} = 2\cdot10^{-3}$ [kg/m^3], $\rho_m = 7800$ [kg/m^3], $c_r = 0.5$, $HB = 570\cdot10^6$ [N/m^2], $\Psi = 2$, $K_F = 0.2$.

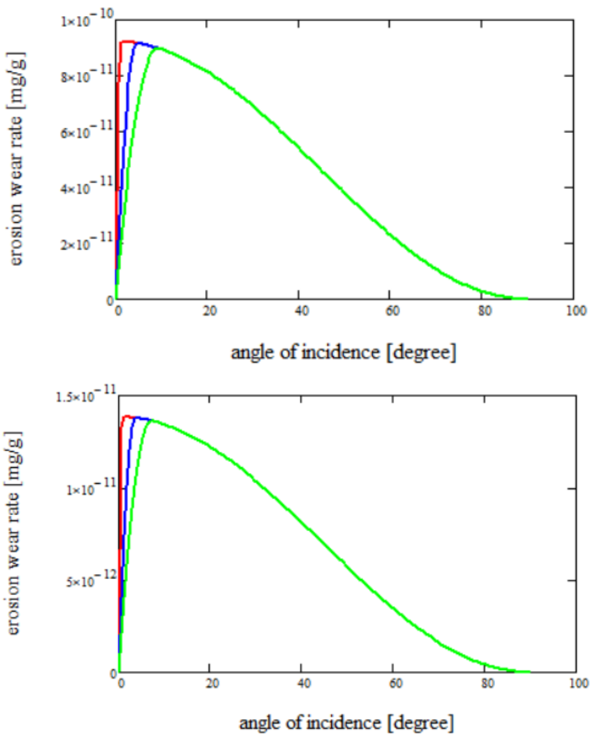


Figure 4. Variation in the intensity of erosive wear: a) first Finnie model; b) second Finnie model

The experimental findings from study [4] indicate that as the spool's edge undergoes wear, new wear particles are generated (thereby increasing the proportion of particles transported by the oil), and this process contributes to the development of a groove on the spool’s surface.

Thus, the wear volume is assumed to be uniformly distributed along the circumference of the cylinder and the length of the spool, resulting in a ring-shaped distribution of the worn material. The thickness of this ring is determined by the variation in clearance (ΔJ). In the cases analyzed in Figure 4a, which reference the model validated in [13], a radial clearance value corresponding to 3000 hours of operation, specifically ΔJ=7.46 µm, was obtained. This value was employed to simulate the impact of wear on the dynamic behavior of the servovalve.

3. SIMULATION OF SERVOVALVE OPERATION IN DYNAMIC MODE CONSIDERING THE IMPACT OF WEAR

The dynamic performance of electrohydraulic servovalves, considering the impact of increased clearance in the spool-sleeve coupling, has been evaluated through operational simulations and quantified using step response quality indicators [13].

The simulation scheme of the servovalve operation, as depicted in Figure 5, incorporates the transfer function (equation 5) derived from the linearization of the servovalve’s mathematical modeling [1, 18-20].

$$H_{SV}(s) = \frac{K_a}{\frac{s^2}{\omega_s^2} + 2\xi\frac{s}{\omega_s} + 1} \tag{5}$$

where: ω_s – the natural frequency;
ξ – the damping factor;
K_a – the amplification factor;

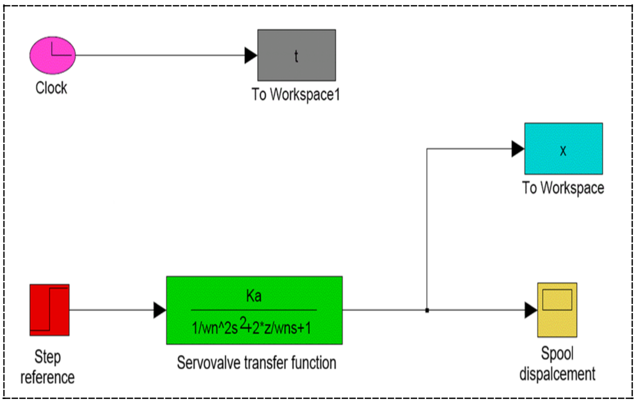


Figure 5. Simulation diagram

Table 1. Technical data of servovalves

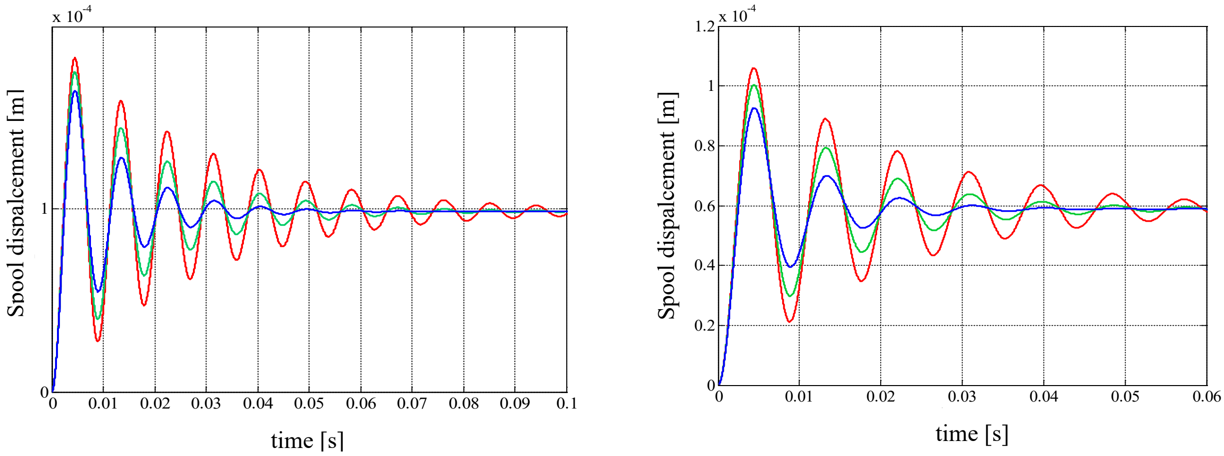
Parameters	Servovalve 1 4WSE2EM10	Servovalve 2 4WSE2EM16
nominal flow rate, Q_n (l/min)	45	100
nominal diameter, D_N (mm)	10	16
supply pressure, p_N (N/m^2)	$300\cdot10^5$	$300\cdot10^5$
nominal pressure drop, Δp_N (N/m^2)	$70\cdot10^5$	$70\cdot10^5$

The simulation program was developed using the MATLAB-Simulink environment and was applied to two servovalves, as characterized in Table 1 [9], to evaluate the impact of clearance on impulse time.

4. RESULTS

The simulation was performed for clearance values (as calculated in [13]) of 3 μm (representing the manufacturing clearance corresponding to 0 hours of operation), 4.7 μm (an intermediate value corresponding to 1500 hours of operation), and 7.46 μm (corresponding to 3000 hours of operation). These simulations were performed with a constant lubricant viscosity of 0.0291 [Pa·s] (for HLP32 oil) and an input voltage of U = 1.85V, as referenced in [13]. The results are graphically depicted in Figure 6 a (SV1) and b (SV2).

Table 2 presents the quality indicators of the step response, while Figure 7 illustrates the variation in the duration of the transient regime as a function of clearance for the two servovalves. The overadjustment (σ) and the duration of the transitory regime (t_t) are particularly significant in relation to the problem under analysis.



a) **b)**

Figure 6. The impulse response obtained for the variable clearance j_1 = 3; j_2 = 4.7; j_3 = 7.46 (μm), a) servovalve 1, b) servovalve 2

Table 2. Quality indicators according to clearance variation

Quality index	Servovalve 1			Servovalve 2		
	j = 3	j = 4.7	j = 7.46	j = 3	j = 4.7	j = 7.46
The overadjustment (σ) [%]	0.6673	0.7732	0.8494	0.5737	0.7035	0.8003
Duration of transitory regime (t_t) [ms]	**33.5**	**52.4**	**82.4**	**24.1**	**37.8**	**59.4**
Growth time (t_c) [ms]	2.36	2.28	2.24	2.39	2.3	2.23
Degree of depreciation (δ)	0.5541	0.4021	0.278	0.669	0.5051	0.356

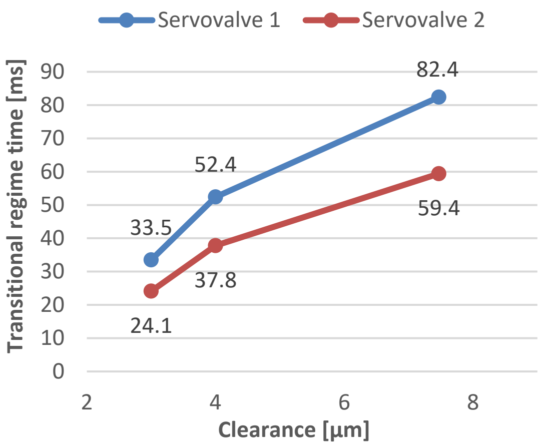


Figure 7. Variation of the duration of the transitory regime, depending on the variation of the clearance, for the two servovalves (SV 1 and SV 2).

5. CONCLUSIONS

The analysis of data concerning the variation of clearance within the spool-sleeve pair, caused by erosive wear, and its impact on the dynamic behavior of servovalves reveals the following:

a) the parameters that exert the most significant influence on the intensity of erosive wear are velocity, angle of incidence, and the characteristics of the target surface materials.
b) the examination of quality indicators of the impulse response, contingent upon variations in the clearance, reveals that:
   - the values of the overadjustment (σ) and the duration of the transitory regime (t_t) are

affected by variations of the clearance as follows:

- the overadjustment (σ) and the duration of the transitory regime (t_t) increase as the clearance increases, for a constant value of viscosity;
- increasing the flow rate of the servovalve leads to a reduction in overadjustment (σ) and a shorter duration of the transitory regime (t_t);

▪ servovalves characterized by lower flow rates exhibit heightened sensitivity to variations in the clearance between the spool and the sleeve compared to those with higher flow rates. This phenomenon can be attributed to the presence of internal leaks caused by clearance, which constitute a significant proportion of the overall flow. Consequently, the precision of servovalves is contingent upon the flow rate, particularly in light of clearance alterations induced by erosive wear.

c) the calculation relationships for the intensity of erosive wear, as an expression of the fatigue process, facilitate the determination of the number of cycles and the time required for the removal of a material layer. This layer's thickness is equated with the variation in radial clearance (ΔJ). This understanding enables further investigation into the influence of radial clearance on the performance of servovalves.

By analyzing the correlation between wear progression and performance degradation, it is possible to formulate maintenance strategies and propose enhancements in their design, thereby ensuring the longevity and operational efficiency of servovalves.

The analysis presented herein requires validation through additional experimental research. Moreover, the scope of the research may be expanded by investigating the effects of additional parameters, such as variations in oil viscosity, and by developing computational models for erosive erosion that incorporate these factors.

REFERENCES

[1] Pătruţ, P.; Nicolae, I., *Hydraulic Drives and Automation – Theory, Apparatus, Automatic Systems and Industrial Applications,* "Nausicaa" Publishing House, Bucharest 1998

[2] Plummer, A., Electrohydraulic servovalves – past, present and future, *10th International Fluid Power Conference (IFK2016),* pp. 405–424, 2016

[3] Tamburrano, P.; Plummer, A.R.; Distaso, E.; Amirante, R., A Review of Electro-Hydraulic Servovalve Research and Development, 2019, *International Journal of Fluid Power*, volume 20, issue 1, pp. 53-98 DOI: 10.13052/ijfp1439-9776.2013

[4] Liu, X.; Liao, H.; Li, D.; Hou, P.; Lu, B.; Ma, X.; Jiao, Z., Tribological modeling and erosion wear compensation in valve-controlled hydraulic motor systems, 2025, *Wear*, Volumes 578–579, DOI: 10.1016/j.wear.2025.206215

[5] Liu, X.; Ji, H.; Min, W.; Zheng, Z.; Wang, J., Erosion behavior and influence of solid particles in hydraulic spool valve without notches, 2020, *Engineering Failure Analysis*, Volume 108, DOI: 10.1016/j.engfailanal.2019.104262.

[6] Liu, X., et al, Experimental study on the correlation between erosion microstructure and performance degradation of hydraulic servo spool valve, 2023, *Surf. Topogr.: Metrology and Properties,* Volume 11, No. 1, DOI: 10.1088/2051-672X/acb228

[7] Yaobao, Y.; Jiayang, Y.; Shengrong, G., Numerical study of solid particle erosion in hydraulic spool valves, 2017, *Wear*, Volumes 392–393, Pages 174-189, DOI: 10.1016/j.wear.2017.09.021.

[8] Wang, Y.; Yin, Y., Performance reliability of jet pipe servo valve under random vibration environment, 2019, *Mechatronics*, Volume 64, DOI: 10.1016/ j.mechatronics.2019.10228

[9] Catalog Rexroth, https://www.boschrexroth.com/en/ro/ products/

[10] Ionescu, I.; Mareş, C., *Electrohydraulic servo valves – conception, physical bases, manufacturing, trials, trends,* Lux Libris Publishing House, Braşov, 1996

[11] Borello, L.; Vedova, M.D.; Jacazio, G.; Sorli, M., A prognostic model for electrohydraulic servovalves, *Annual Conference of the Prognostics and Health Management Society*, San Diego, USA, 2009

[12] Tudor, A., *Friction and Wear of Materials,* Bren Publishing House, Bucharest, 2002

[13] Popovici, A.I., *Contributions to the study of the effects of wear on the performance of automatic control hydraulic systems*, PhD Thesis, Bucharest, 2012

[14] Sundararajan, G., A comprehensive model for the solid particle erosion on ductile materials, 1991, *Wear*, Vol. 149, Issue 1-2, pp. 111-127

[15] Finnie, I., Erosion of surface by solid particles, 1960, *Wear*, Vol. 3, Issue 2, pp. 87-103.

[16] I. Finnie, *Some observations on the erosion of ductile metals*, Wear, **19** (1), 81-90 (1972).

[17] Bingley, M.S., O'Flynn, D.J., *Examination and comparison of various erosive wear models*, Wear, volume 258 (2005), issues 1-4, pp. 511-525

[18] Catană, I., *Automatic electrohydraulic systems, Lithographing, Ecological University,* Bucharest, 1995

[19] Călinoiu, C., Vasiliu, D., Vasiliu, N., Catană, I., *Modelling, Simulation and Experimental Identification of Hydraulic Servomechanisms,* Technical Publishing House, Bucharest, 1998

[20] Catană, I., Vasiliu, D., Vasiliu, N., *Electrohydraulic servomechanisms – construction, operation, modeling, simulation and computer-aided design,* Bucharest, 1995