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APPRECIATION OF WEAR ON ACTIVE SURFACES OF THE AUTOMATIC HYDRAULIC SYSTEMS IN TERMS OF CONTAMINATING PARTICLES

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Abstract: *Automatic Hydraulic Systems (AHS) are made of hydraulic or electrohydraulic elements. Electro-hydraulic servo valves (SV) are used in automatic systems to ensure high performance. Due to the requirements imposed by the systems, the SV must have a design and manufactured specifications that should ensure a minimum of wear overtime. Wear particles resulting from other components of the AHS are driven by the hydraulic oil and cause erosive wear of the SV. This paper presents the study of the abrasive erosion of SV by simulating the impact of the particles on their active surface and quantifying the erosion by using the classical models of erosive wear.*

Keywords: *Erosive wear, servo valves, modeling-simulation, analytical wear model, solid particle*

1. Introduction

The electro-hydraulic servo valves (SV) are the most complex components of AHS that provide information transfer from electronic area to hydraulic area of hydraulic systems.
In essence, the SVs are a key component for the control systems; servo valves convert electrical control signal (current, voltage) into output value (flow, pressure) of the system. Between input-output there are mechanical, hydraulic and electrical connections. The most common SVs are those with flow control with two amplification stages (preamplifier and hydraulic amplifier).
The spool-sleeve join (the main coupling of hydraulic amplifier) is made with very precise tolerance of form, dimensional and position, and with very low roughness for a proper operation and high performance of SV's.
Theoretical and experimental research has aimed to analyze the cause that generate SV components degradation (wear), especially those produces by friction and characteristic wear, among which progressive contamination of the hydraulic oil is the most important [1].
Spool and sleeve surfaces are subject to erosion in two cases:

1. Direct erosion on spool surface occurs when the spool is moving and particles from infested fluid are hitting spool's surfaces from SV's inlet to outlet (figure 1);

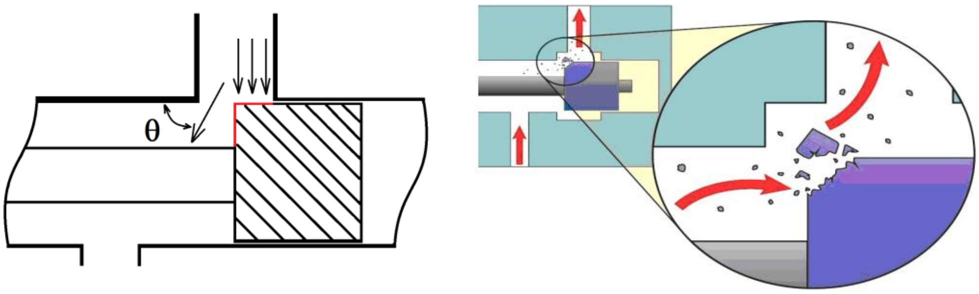


Fig. 1. Direct erosion on spool surface [2]

2. Erosion in annular clearance between spool and sleeve – the particles carried by fluid are entering the gap and causes surface erosion (figure 2).

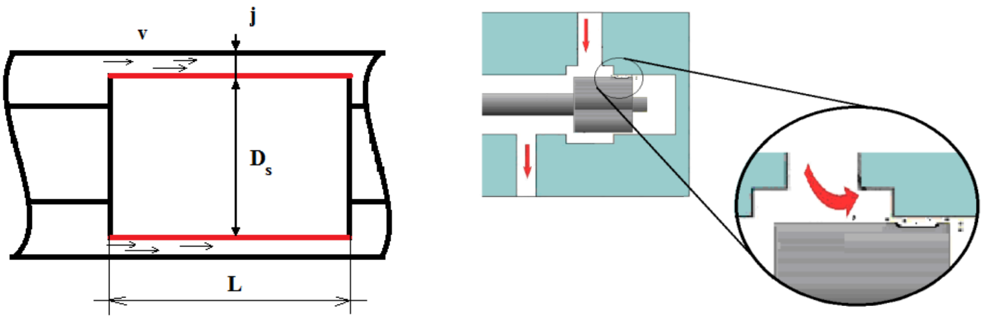


Fig. 2. Erosion in spool-sleeve clearance [2]

Hard particles that are entering the gap between the spool and sleeve can be found in three cases: moving freely through the gap, set on one of the friction surfaces, or making a “bridge” between the two surfaces.
Abrasive erosion is determined by speed, mass, shape, concentration, mechanical and metallurgical characteristics of contaminating particles. This is the most active form of wear for AHS, the most important source of involved surface damage. Small particles of micron size can be “crowded” in areas where the flow is achieved with lower speeds, in which they can plastically “weld” due to contact stress exceeding allowable pressures. Thus, overcrowding particles formed have more intense erosive effects than those of the separate particles.
Superficial mechanical fatigue is favored by cracks occurrence in the superficial surface layer, caused mainly by the formation and sheer of microjoints created at the asperity's peaks contact which is characteristic for adhesion wear. Fatigue wear particles is generated after following specific fatigue cycle: initiation, propagation and achieve critical crack length.
Effects caused by abrasive particles impact on target areas are determined by hydraulic oil characteristics of AHS and by mass contaminants characteristics. For high pressure and oil flow speeds abrasive-erosion effects are considerable.
Contaminating particles are described by size, number, concentration, nature, mechanical and metallographic characteristics of the material. It turned out that the most dangerous in terms of erosion and abrasion, are particles with sizes up to 5μm which enter in the gap between spool and sleeve [3]. Hard contaminating particles with sizes smaller than the radial clearance between spool and sleeve, usually in the range 1÷4 μm, are capable of temporary sticking to the sleeve [4].
The accumulation of contaminating particles in the small gaps, as that between the spool and sleeve, produces the uneven distribution of the radial load and the increase of the axial load necessary for the spool movement.
In this paper studies were carried out on the erosion wear of the SV coupling under different opening degrees, contaminated particle concentrations and considering the inlet and outlet pressures. This article focuses on the theoretical and simulation studies of the erosion and wear of SV spools and sleeves. Figure 3 shows the workflow chart of these studies.

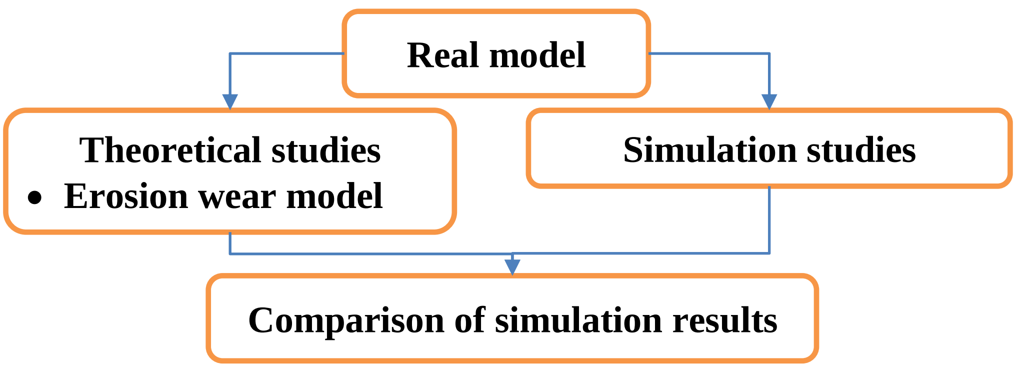


Fig. 3. Workflow chart

2. Tension state at hydraulic contact between spool and sleeve

Using SolidWorks Flow Simulation software fluid flow in spool-sleeve coupling was simulated. The software allows the definition of the working fluid (density, viscosity variation model, etc.). In this paper there was considered a contaminated hydraulic oil HLP 46. The flow of oil having solid contaminated particles inside the valve is a liquid-solid two-phase flow, in which the oil is the continuous phase, and the contaminated particles are the discrete phase.
For flow analyze, there were established boundary conditions and the flow space which was limited by inserting lids on the input and output areas of the fluid, as shown in figure 4 and as follows:

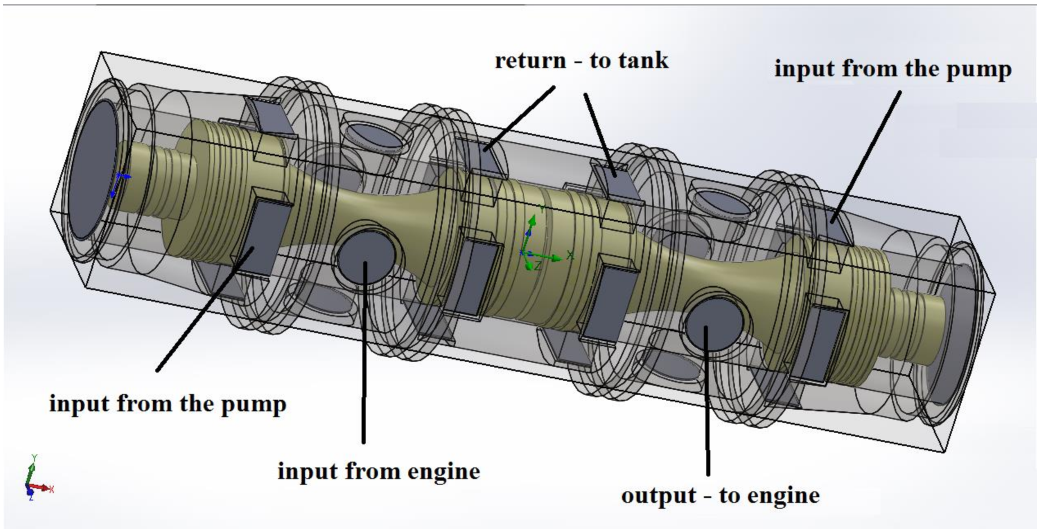
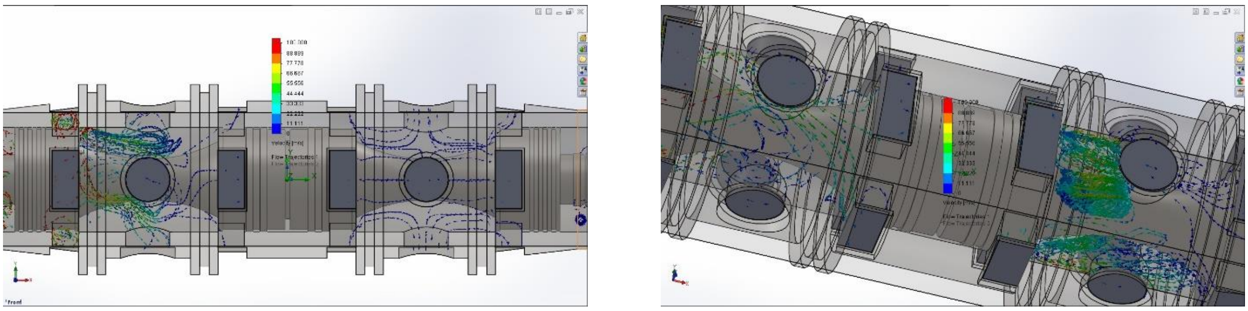


Fig. 4. Boundary condition for spool-sleeve coupling

1. Input from the pump – it was considered the input flow;
2. Output to engine – outlet pressure was imposed;
3. Input from the engine – it was imposed engine pressure;
4. Return (to tank) – atmospheric pressure;
5. Fluid-solid interface conditions – clearance in spool-sleeve coupling.

Simulations have been performed considering the spool opening of 0.5 mm and 0.1 mm.
After running the computing finite element applied for considered fluid flow were obtained results on flow path and velocity distribution (or pressure) inside the spool-sleeve coupling geometry. These results are shown in figures 5 a and b (both for contaminated oil at 1200 operation hours).



a) b)

Fig. 5. Flow path and speed flow variation for contaminated oil: a) spool movement of 0.1 mm; b) spool movement of 0.5 mm

3. Erosion analysis in terms of hydraulic oil contamination

3.1 Erosion wear rate obtained from fluid flow simulation

In the flow of liquid-solid fluids, the erosion of wall material by solid particles is a usual form of wear. The trajectory of contaminated particles in the flow field influences the flow field and the prediction of wear on the boundary surface.

During SV's operation, solid contaminating particles follow the fluid, and constantly hit the spool or sleeve, that causes the wear of the spool sleeve coupling.

The SolidWorks [5] program allows visualization of contaminating particle trajectories, and it can calculate erosion wear rate with the following equation:

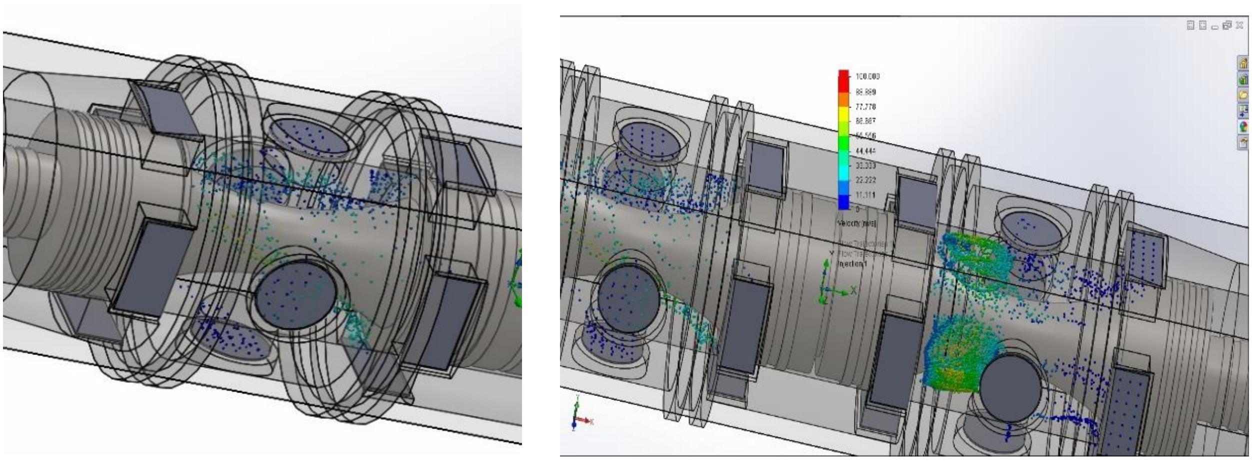
$$R_{erosion} = \sum_{P=1}^{N_p} K \frac{m_p C(d_p) f(\theta) V^{b(V)}}{dS} \quad (1)$$

where: $\mathbf{R_{erosion}}$ – the erosion wear rate of the SV's coupling, [kg/m^{2}s]; $\mathbf{N_p}$ – the number of solid contaminating particles; $\mathbf{m_p}$ – mass of particles carried by fluid in a certain time [kg/s]; **V** – difference between the particle's velocity and the wall speed, [m/s]; **dS** – a finite element surface of solid contamination particles impacting the wall of the coupling, [m^2]; **K** – erosion coefficient; **C**(d_p) – coefficient that defines erosion dependence of the particle size; **b(V)** – coefficient that defines erosion dependency of the particles relative speed; **f(θ)** – coefficient that defines erosion dependency of the impact angle; **θ** – the impact angle between the contaminating particles and the wall of the SV's coupling.

Erosion dependency coefficients can be tabularly defined by angle, speed, etc.

Equation (1) defines the physical significance of the wear quality of the spool sleeve in relation to the erosion rate in erosion wear.

Regarding the contamination particles, their trajectories were highlighted for the input area (figure 6.a) and output area (return to tank) (figure 6.b) to and from the SV. There were considered 200 particles of 2 μm size and the particle concentration of 3.33·10-5 kg/s.

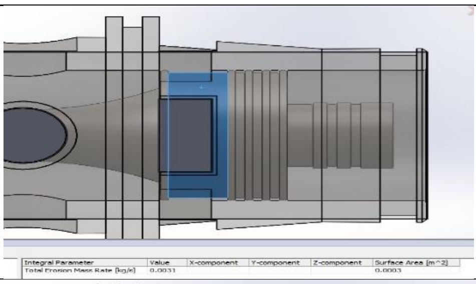
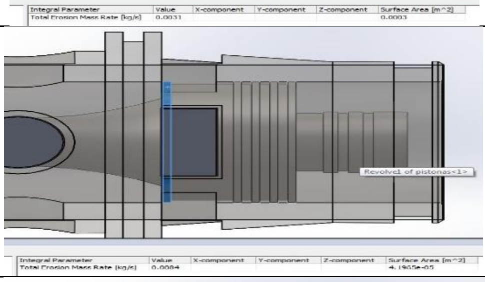
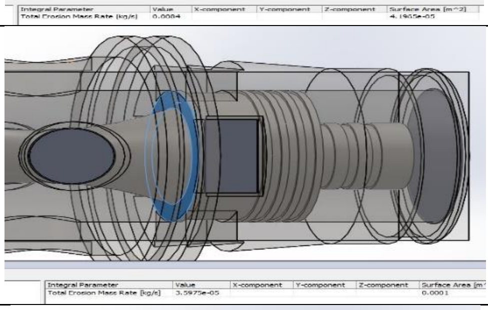
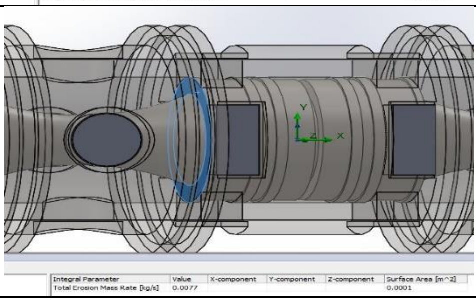


a) b)

Fig. 6. The trajectory for the imput area: a) spool displacement of 0.1 mm; b) spool dispalacement of 0.5 mm

In table 1 are presented the values of wear rate for different areas of the servovalve spool, for two different spool opening (0.1 mm and 0.5 mm).

Table 1: Wear rate values obtained from fluid flow simulation

The area subject to erosion	**Ware rate values** [kg/m^{2}s]	
	Spool opening of 0.1 mm	**Spool opening of 0.5 mm**
	**0.0017**	**0.0031**
	**0.0025**	**0.0084**
	**7.3713·10^{-6}**	**3.5975·10^{-5}**
	**0.00622**	**0.0077**

3.2 Wear rate obtained with the classical models for erosive wear

3.2.1 Classical models for erosive wear

The main form of failure of SV coupling elements is erosion failure. Finnie [6] has conducted theoretical and experimental studies on the erosion and wear mechanism of plastic impact of the contaminating particles and proposed a micro-cutting theory of erosion and wear, from which an empirical formula for its erosion was derived, and the micro-cutting theory was verified through experiments. The disadvantage of his theory is that when particles are eroded at large angles, there is a large error between the equation of wear and the experimental results. Bitter [7] proposed a deformation wear theory based on micro-cutting wear and deformation wear mechanisms. According to this theory, that material wear is the result of a combination of micro-cutting and plastic deformation wear, with cutting wear dominating when solid particles hit the surface of an object at a small angle, and deformation wear dominating when particles hit the surface of an object at a large angle.

1) Finnie's models
First Finnie's model
For the calculation of the wear rate (defined as the ratio between the mass of the worn material and the mass of the eroded material), Finnie [6] considers sharp abrasive particles, their recoil effects and micro-cutting effects and that the material is plastically deformed following the impact of the abrasive particle.
The calculation relationships proposed by Finnie have the following structure:

$$R_u = \frac{p_a \cdot \rho_m \cdot v^2 \cdot (1-c_r^2)}{0.9272 \cdot H_S \cdot \psi \cdot K_F}\left(\sin 2\alpha - \frac{6}{K_F}\sin^2\alpha\right), \quad \text{for} \quad tg\alpha \le \frac{K_F}{6} \qquad (1)$$

$$R_u = \frac{p_a \cdot \rho_m \cdot v^2 \cdot (1-c_r^2)}{0.9272 \cdot H_S \cdot \psi \cdot K_F}\left(\frac{K_F}{6}\cos^2\alpha\right), \quad \text{for} \quad tg\alpha \ge \frac{K_F}{6} \qquad (2)$$

where: $\boldsymbol{R_u}$ – dimensionless wear rate; $\boldsymbol{p_a}$ – the percentage of abrasive particles with micro-cutting effects [kg/m³]; $\boldsymbol{\rho_m}$ – target material density [kg/m³]; $\boldsymbol{v}$ – particle impact speed [m/s]; $\boldsymbol{\alpha}$ – the angle of incidence [degrees]; $\boldsymbol{c_r}$ – restitution coefficient; $\boldsymbol{H_s}$ – static hardness [N/m²]; $\boldsymbol{\Psi}$ – the ratio of the contact length ($\boldsymbol{L}$ [μm]) and cutting depth ($\boldsymbol{\delta}$ [μm]) of impact area; $\boldsymbol{K_F}$ – the ratio of the horizontal component ($\boldsymbol{F_o}$ [N]) and the vertical component ($\boldsymbol{F_o}$ [N]) of the characteristic impact force.

Second Finnie's model
For the second model, Finnie reconsider the percentage of the abrasive particles with micro-cutting effects by reducing the p_a factor from 50% to 10%. Thus, the calculation relations become:

$$R_u = \left(\frac{p_a}{2K}\right)\frac{\rho_m \cdot v^2 \cdot (1-c_r^2)}{0.9272 \cdot H_S}\left(\sin 2\alpha - \frac{8}{K_F}\sin^2\alpha\right), \text{ for } tg\alpha \le \frac{K_F}{8} \qquad (3)$$

$$R_u = \left(\frac{p_a}{16}\right)\frac{\rho_m \cdot v^2 \cdot (1-c_r^2)}{0.9272 \cdot H_S}(\cos^2\alpha), \text{ for } tg\alpha \ge \frac{K_F}{8} \qquad (4)$$

2) Bitter's model
Bitter considered that the abrasive erosion mechanism has two components that occur simultaneously: the micro-cutting component, which occurs at small incidence angles, and the repeated deformation component of the target surface. In the case of micro-cutting, Bitter considers two cases related to the moment when micro-cutting ceases, the tangential component of the velocity has a certain value when the particle leaves the surface and the case when this speed becomes zero during the micro-cutting process [7].
The total erosive wear rate is the sum of the two component and is differentiated by the value of the impact angle at which the particle leaves the surface:

$$R_u = R_{u,DW} + R_{u,C_1}, \quad \text{for} \quad \alpha \le \alpha_0 \qquad (5)$$

$$R_u = R_{u,DW} + R_{u,C_2}, \quad \text{for} \quad \alpha > \alpha_0 \qquad (6)$$

where α_0 - corresponds to the angle of impact at which the tangential component of speed is zero, which occurs when the particle leaves the incident surface.

For repeated surface deformation component, Bitter suggest the following relation for calculating the rate of wear:

$$R_{u,DW} = \frac{\rho_m (U_i \sin\alpha - U_{el})^2}{2\varepsilon_D} \qquad (7)$$

When the tangential component of the particle speed is not zero, the wear rate is defined with the following equation:

$$R_{u,C_1} = \frac{2\rho_m C_1 (U_i \sin\alpha - U_{el})^2}{(U_i \sin\alpha)^{0.5}} \cdot \left(U_i \cos\alpha - \frac{\varepsilon_C C_1 (U_i \sin\alpha - U_{el})^2}{(U_i \sin\alpha)^{0.5}}\right) \quad (8)$$

and for the second case, when the speed is zero, the following relation is used:

$$R_{u,C2} = \frac{\rho_m \left(U_i^2 \cos\alpha - C_2 (U_i \sin\alpha - U_{el})^{1.5}\right)}{2\varepsilon_C} \quad (9)$$

For the previous equation, the terms have the following meaning: $\boldsymbol{R_{MC_1}}$, $\boldsymbol{R_{MC_2}}$ and $\boldsymbol{R_{M,DW}}$ – dimensionless erosion wear rate; ***v – impact speed components:*** $\boldsymbol{U_i}$ - impact speed of the particle [m/s] and $\boldsymbol{U_{el}}$ - the impact speed that is reached at the elastic limit of the material [m/s]; $\boldsymbol{\varepsilon_D}$ - the specific energy of deformation [J/m]; $\boldsymbol{\rho_a}$- density of the erosive particle material [kg/m^3]; $\boldsymbol{\rho_m}$ – density of the target material [kg/m^3]; $\boldsymbol{E_e}$ - modulus of elasticity on impact [N/m^2]; $\boldsymbol{\varepsilon_c}$ - the specific wear energy for microcutting [J/m]; $\boldsymbol{C_1}$ and $\boldsymbol{C_2}$ - constants with specific relations; $\boldsymbol{H_s}$ – static hardness [N/m^2]; $\boldsymbol{\alpha}$ – the angle of incidence [degrees].

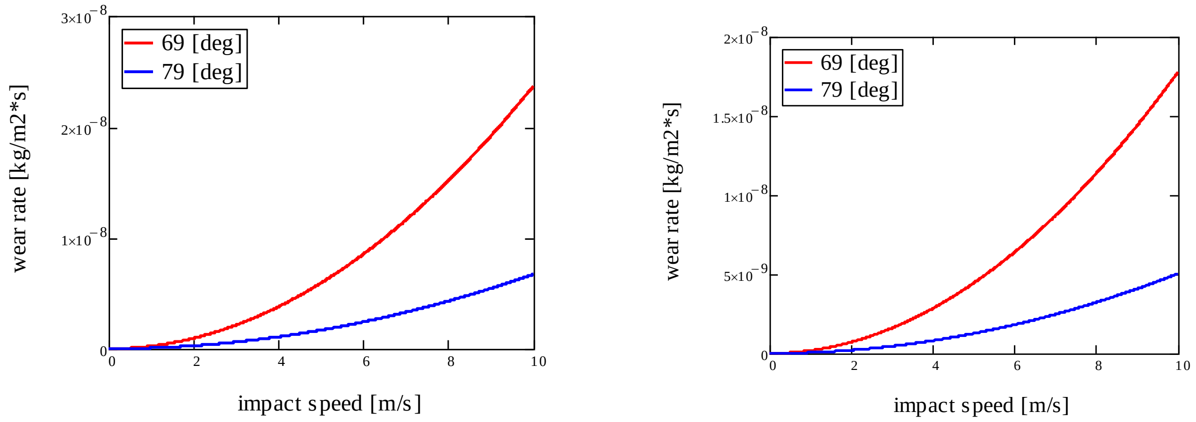
3.2.2 Classical models for erosive wear

For the studied erosion wear models is considered the material 38 MoCrAl 9 with the properties specified in table 2. It is also considered the influence of the material of the abrasive particle through the density parameters (ρ_{ab}), considering the concentration of contaminating particles.
For Finnie models, the influence of the particle characteristics is considered by the pa factor (percentage of abrasive particles) calculated according to the particle concentration.
In table 4 are listed the values of the parameters involved in the calculation equations of the analyzed models.

Table 2: The parameters of the calculating equations

<table>
<tr><th rowspan="2">Model</th><th colspan="2">Parameters</th></tr>
<tr><th>Specific to the model</th><th>Common to the models</th></tr>
<tr><td>Finnie</td><td>p_a - the percentage of abrasive particles with micro-cutting effects
• 50% for the first model: $\boldsymbol{p_a}$=10^{-2} kg/m^3;
• 10% for the second model $\boldsymbol{p_a}$=$2\cdot10^{-3}$ kg/m^3;
$\boldsymbol{\Psi}$ – the ration of the contact length and the cutting depth of the impact area: 10 for SV’s material;</td><td rowspan="2">$\boldsymbol{\rho_m}$ - density of the target material: 7800 kg/m^3;
$\boldsymbol{\rho_{ab}}$ - density of the target material: 4000 kg/m^3;
HB - surface hardness: $570\cdot10^6$ [N/m^2]
E - Young’s modulus: $2.1\cdot10^{11}$ [N/m^2]
$\boldsymbol{\alpha}$ - incidence angle: 69° and 79°
v - particle impact speed: 0.1÷10 m/s;
μ - friction coefficient between the particle and the material: 0.1.
$\boldsymbol{c_r}$ - restitution coefficient: 0.5;</td></tr>
<tr><td>Bitter</td><td>$\boldsymbol{\alpha_0}$ - the angle at which the speed is zero: 0º;
$\boldsymbol{\varepsilon_D}$ - the specific energy for wear: $\boldsymbol{\varepsilon_D}$ – elastic deformation: $4.7\cdot10^{10}$ J/m^3; $\boldsymbol{\varepsilon_C}$ - micro-cutting: $2.2\cdot10^{10}$ J/m^3</td></tr>
</table>

Considering the equations and the specific parameters, the variation of the erosive wear rate was analyzed for each model presented, depending on the impact speed that is a common parameter for both models. The results obtained are listed in figures 7 and 8.



a) **b)**

Fig. 5. Variation of wear rate depending on the impact speed:

a) Finnie's first model; b) Finnie's second model

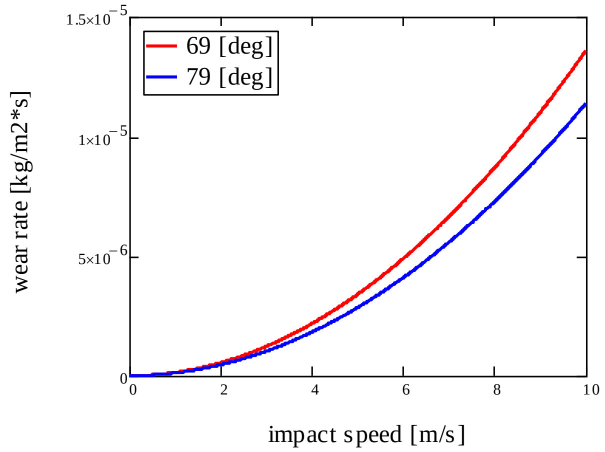


Fig. 6. Variation of wear rate depending on the impact speed for Bitter's model

Table 3 includes the values of the erosive wear intensity measured for two considered values of the impact speed. The impact angles are 69° and 79° and are specific to the spool opening considered in the earlier flow analysis.

Table 3: The values of the erosion wear rate for two impact angles

Model		**Wear rate depending on two considered values of the impact speed [kg/m²·s]**	
		$v_1 = 2$	**$v_2 = 10$**
Finnie 1	**69°**	$9.57 \cdot 10^{-10}$	$2.37 \cdot 10^{-8}$
	79°	$2.66 \cdot 10^{-10}$	$6.69 \cdot 10^{-9}$
Finnie 2	**69°**	$7.11 \cdot 10^{-10}$	$1.77 \cdot 10^{-8}$
	79°	$1.99 \cdot 10^{-10}$	$5.04 \cdot 10^{-9}$
Bitter	**69°**	$4.54 \cdot 10^{-7}$	$1.36 \cdot 10^{-5}$
	79°	$5.48 \cdot 10^{-7}$	$1.14 \cdot 10^{-5}$

4. Conclusion

Due to the extremely high pressure which develops in the flow space, SV elements are elastically deformed until the lubricant film is broken and there is direct contact between the spool and the sleeve. Finite element analysis of flow and strain generated by the contaminated hydraulic oil in the clearance between spool and sleeve, highlighted the areas most exposed to erosive wear.
The study highlights the occurrence of high strains, randomly located (according to the relative position of the pattern elements - 0.5 respectively 0.1 mm) on the contact area with the pressure fluid, but not as a direct result of the fact that their areas are hardened (by heat treatment) so it can be considered that the maximum strain does not lead to plastic deformation but to an elastic deformation of superficial layer, which may lead in time to fatigue. If it is considered the particles impact, this process is accelerated.
The results confirmed the prediction of erosion models, which the intensity of erosive wear is greatly reduced in the coupling clearance. The maximum erosion rate increases with the opening degree and reaches a maximum value when the opening degree is 0.5 mm.
There is a certain amount of sleeve erosion near the corners (table 1); for the spool, the erosion wear rate of the sharp edge of the control surface of the slide valve sink groove is significantly greater than that of the sharp edge of the control surface of the convex shoulder.
The erosion rate is influenced by the contaminated solid particles of the oil, with the increase in the contaminated solid particles parameter, the erosion rate changes more obviously. The erosion rate is less affected by the change in opening degree and increases with the increase in opening degree.
By comparing the values of the two mathematical models from table 3 it was found that the minimum value of the wear intensity results for Finnie's models and the maximum value is in Bitter case. These differences are due to the different parameters the models consider and as Bitter specifies two components of the erosive wear.
The presented studies must be confirmed through experimental research to show which model has the most exact calculation equation.
In the presented analysis were considered the two of the most influential parameters: opening degree (corresponding to the flow rate of the valve) and oil contamination with solid particles.
It is necessary for further study on the erosion wear of SV's coupling considering more parameters, like temperature and operating pressure. These studies can establish a mathematical model that predicts with great accuracy the wear phenomenon of the SV coupling parts.
This experimental study will be used to analyze the SV's performances by simulating their motion to specified parameters.

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