EMERG, Volume IX, Issue 3/2023 ISSN 2668-7003, ISSN-L 2457-5011

CHARACTERISTICS OF ABRASIVE EROSION WEAR SPECIFIC FOR PROGRESSIVE CAVITY PUMPS USED IN PETROLEUM INDUSTRY

CARACTERISTICI ALE UZĂRII EROZIV-ABRAZIVE SPECIFICĂ POMPELOR CU CAVITATE PROGRESIVĂ FOLOSITE ÎN INDUSTRIA PETROLIERĂ

Ammar MULLA[1], Aristia-Ioana POPOVICI[2]*, M. Iyad Al NABOULSI[1]*,
Niculae Napoleon ANTONESCU[1]

Abstract: *The progressive cavitation pumps (PCP), also called screw pumps, are frequently used in the petroleum industry for the transport of water, hydrocarbons and other viscous fluids as well as for the extraction of crude oil from wells, due to their continuously improved technological performance. The materials used for the construction of these pumps were described in [1][2][3]. The design and quality requirements are imposed by ISO standards, but the materials with the best tribological behavior are not indicated. In order to choose the couple of materials with minimal abrasive-erosive wear, studies are required on the abrasive effects of the stator and the rotor. In this paper it is analyzed the variation of the intensity of erosive wear depending on the material of the coupling surfaces and the impact speed of the particles that hit these surfaces. For this analysis were used four of the classic models (Finnie, Bitter, Hutchings and Sundararajan) with applicability in all abrasive-erosive environments.*

Keywords: progressive cavity pumps, stator-rotor couple, elastomers, drilling fluid, impact speed, erosion wear models.

Rezumat: *Pompele cu cavitate progresivă (PCP), denumite şi pompe cu şurub, sunt frecvent utilizate în industria petrolieră pentru transportul apei, a hidrocarburilor şi altor fluide vîscoase precum şi la extracţia ţiţeiului din sonde, datorită performaţelor tehnologice permanent îmbunătăţite. Materialele utilizate pentru construcţia acestor pompe sunt prezentate în lucrările [1][2][3]. Prin standarde ISO se impun cerinţele dimensionale şi de calitate, dar nu sunt indicate materialele cu cea mai bună comportare*

---

[1] Department of Mechanical Engineering, Petroleum-Gas University of Ploieşti, Romania, e-mail: nnantonescu@upg-ploiesti.ro, lyadnaboulsi@yahoo.com

[2] Department of Machines and Advanced Technologies in Construction, Technical University of Civil Engineering of Bucharest, Romania, e-mail: arys_jo@yahoo.com

tribologică. Pentru alegerea cuplului de materiale cu uzarea abraziv-erozivă minimă se impun studii privind efectele abrazive asupra statorului şi respectiv rotorului. În prezenta lucrare se analizează variaţia intensităţii de uzare erozivă în funcţie de materialul suprafeţelor cuplei şi de viteza de impact a particulelor ce lovesc aceste suprafeţe. Pentru această analiză s-au folosit patru dintre modelele clasice (Finnie, Bitter, Hutchings şi Sundararajan) cu aplicabilitate în toate mediile abraziv-erozive.

Cuvinte cheie: pompă cu cavitate progresivă, cupla stator-rotor, elastomeri, fluid de extracţie, viteză de impact, modelele uzării erozive

1. Introduction

Pumps with progressive cavity (PCP) are frequently used in the petroleum industry due to their continuously improved technological performance in relation to the abrasive work environment.

The main tribological coupling of the PCP, the rotor-stator coupling, is subject to the impact (elastic or plastic) of abrasive particles, carried by the fluid, under the specific conditions of the abrasion wear, which is manifested by specific forms: micro-cutting, plastic deformation grinding, abrasive erosion, coulting and scratching [4].

The wear of the rotor and the stator (of the sealing surfaces and the pumping chambers) is a tribological process influenced by several factors, including the impact speed of the abrasive particles carried by the fluid, the angle of incidence characteristic of the particle impact on the active surface of the pump elements, the characteristics of the erosive particles [5][6]. In abrasive environments erosive wear increases proportional to the speed square [7]. Increasing the pumping speed causes an increase in the flow rate and the impact rate of the particles with the target surface.

It has been experimentally demonstrated that for great incidence angles the speed value is the one that influences and differentiates the processes of generating the wear particles [8].

In selecting and utilization of the rotor-stator coupling materials are considered the specific features of the PCP’s abrasive-erosive process. In this case was imposed the couple of hard-elastic materials with significant tribological advantages.

The control of erosive-abrasive wear is achieved by specific processes of coated layers on the active surface (of contact) of the rotor material and by using materials with predominantly elastic properties for the stator. For low abrasive environments are used hard chromed or nichel coated rotors;

however they are not recommended in high working wills with a percentage of sand greater than 10% [7].

For PC pumps working in aggressively abrasive environments, it is recommended to be used elastomers for stators manufacturing. The advantages of these materials refer to their high flexibility during exploitation, which facilitates the passage of larger abrasive-erosive particles through the sealing areas, without significant wear effects; also, larger hard particles can be embedded in the elastomer material, thus reducing their erosive effects.

The performance of elastomers depends on their mechanical properties, the hardness in particular; from tribological point of view the optimal efficiency is achieved for the hardness of the material from 50 to 65 shoreA. They are also dependent on the mechanical and physical properties of the elastomer.

2. Analytical models of abrasive-erosive environments

The infiltration of solid particles (e.g. sand) in the petroleum industry is inevitable and thus the erosive wear of the drilling equipment occurs.

For wear processes studies, especially abrasive erosion, and to be able to estimate material erosion, were developed mathematical models that consider as many of the influencing factors as possible.

In his paper, *Hunt* [9] studied and analyzed 28 mathematical models specific to erosive wear. Of these models, 4 are considered classical models and were developed before 1990 and successfully used in the analysis of erosive wear of mechanical equipment; these were proposed by Finnie (in 1958 and 1960), Bitter (in 1963), Hutchings (in 1974, 1979, 1981) and Sundarajan along with Shewmon (in 1983).

The use of estimation methods for describing the erosion mechanism inaugurates a new stage in the research field, a phase between 1990 – 1995 and continued to this day [10].

Most erosion prediction equations are empirical and are based on experimental data obtained for solid particles in continuous flow (in gas or solids) [11].

Among the models suitable for abrasive-erosive environments, four of the classical models were used in this analysis; Finnie, Bitter and Hutchings were presented in paper [2] of the authors, and the Sundarajan model is described below.

Sundarajan [12] and *Shewmon* set a valid model for the normal incidence angle; they proposed the following equation of the wear rate:

$$R_{ss} = \frac{6.5 \cdot 10^{-3} \rho_a^{0.25} v^{2.5} (1 - e_r^2)^{1.25}}{C_p T_t^{0.75} H_s^{0.25}}, \tag{1}$$

where e_r, the restitution coefficient is calculated with the equation:

$$e_r = \frac{1.36 \cdot H_s^{0.625}}{E_c^{0.5} \rho_a^{0.125} v^{0.25}}, \tag{2}$$

Then *Sundarajan* [13] developed the model considering the relationship between deformation and kinetic energy (differentiated for normal and oblique impact) and proposes an equation for cumulative erosive wear. The equations are as follows:

- Normal impact:

$$R_{sn} = \left[\frac{5.5 \cdot 10^{-2}}{(T_s - 436)^{0.75}}\right] \frac{2^n \cdot F_t \cdot v^2 \cdot sin(\alpha)^2 \, (1 - e_r^2)}{n \cdot C_p}, \tag{3}$$

- Oblique impact:

$$R_{s0} = \left[\frac{5.5 \cdot 10^{-2}}{(T_s - 436)^{0.75}}\right] \frac{(n+1) \cdot \frac{\mu}{\mu_{crt}} \cdot \left(2 - \frac{\mu}{\mu_{crt}}\right) v^2 \cdot cos(\alpha)^2}{2^{2-n} \cdot C_p \cdot n \cdot (1 + \lambda)}, \tag{4}$$

- Cumulative intensive erosive wear (***R***$\boldsymbol{R_c}$) is calculated with the equation:

$$R_c = R_{sn} + R_{s0}, \tag{5}$$

3. Specific parameters of the analysis models

The classical models are successfully used in the analysis of mechanical equipment erosion. These models consider as common parameters the materials of the target surfaces and of abrasive particles as well as their impact speeds.

For all the analyzed models, the equation of the wear rate includes certain parameters that considers the material of the abrasive particles and / or of the target surface. In Table 1 are specified the properties of the stator and rotor materials. Were studied 5 polyamides [1] for stator and 2 steels (chrome steel and non-coated steel) for rotor.

The values of the parameters involved in the equations of the analyzed models are specified in Table 2.

Table 1. **Properties of the analyzed materials**

Material / **Properties**	**Rotor (steel)**		**Stator (elastomers)**				
	AISI 4130 (462)	**AISI 4130 (352)**	**RA6**	**PC**	**SKN**	**Z13**	**PA6.6**
Hardness, ***HB*** [N/m^2]	$800\cdot10^6$	$650\cdot10^6$	--	--	--	--	--
Hardness, ***HS*** [shoreA]	--	--	77	55	75	80	85
Young's Module, ***E*** [N/m^2]	$2.1\cdot10^{11}$		$4.5\cdot10^6$	$4.5\cdot10^6$	$5.6\cdot10^6$	$6\cdot10^6$	$3.45\cdot10^6$
Material Density, ρ_m [kg/m^3]	7850	7750	1140	1350	950	1000	1150
Melting Temperature, T_m [K]	1705		353	320	306	330	469
Specific Heat, C_p [J/kg K]	477		1180	1120	1940	1470	1590

Table 2. **Parameters in the calculating equations**

Model	**Parameters**	
	Specific to the models	**Common to the models**
Finnie	p_a – the percentage of abrasive particles (sand) with micro-cutting effects for concentration of 2.4x10^3 [mg/l] ● 50% for the first model: p_a = 1.2 [kg/m^3]; ● 10% for the second model: p_a = 0.24 [kg/m^3]; c_r – restitution coefficient: 0.5; Ψ – the ration of the contact lenght and the cutting depth of the impact area: 2 for the stator and 10 for the rotor;	ρ_m - density of the target material (table 1) [kg/m^3]; HB - surface hardness (table 1) [N/m^2] E - Young's modulus (table 1) [N/m^2] α - the angle of incidence 10° v - particle impact speed, between 0.5÷5 [m/s]; μ - friction coefficient between the particle and the material: 0.3.
Hutchings	c_r – restitution coefficient: 0.5 ; Γ – rate of removed volume depending on the particle volume and the critical stress: 10;	

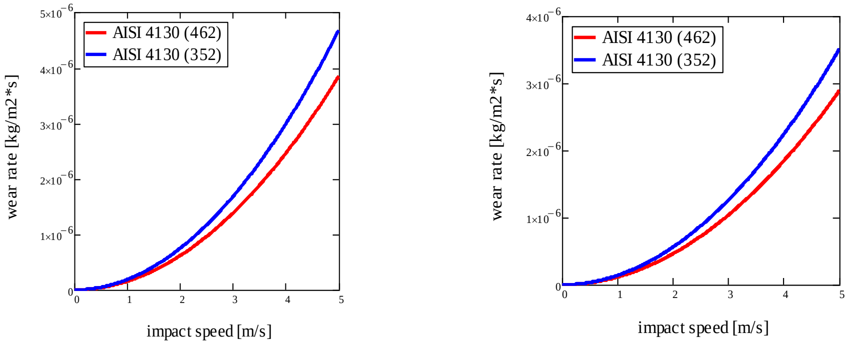
Table 2 (continued)

Bitter	c_r – restitution coefficient: 0.5; α_0 – the angle at which the speed is zero: 5º; - the specific energy for wear: ε_D – elastic deformation: $4.7 \cdot 10^{10}$ [J/m^3]; ε_C – micro-cutting: $2.2 \cdot 10^{10}$ [J/m^3]	
Sundararajan	n – material coefficient: 0.1 F_t – coefficient of the impact: 0.025 λ – particle shape coefficient: 2.5 C_p – specific heat (table 1) [J/kg K]; T_m – melting temperature of the target material (table 1) [K]; ρ_{ab} – density of the particle material (sand): 1600 kg/m^3;	

4. Wear analysis

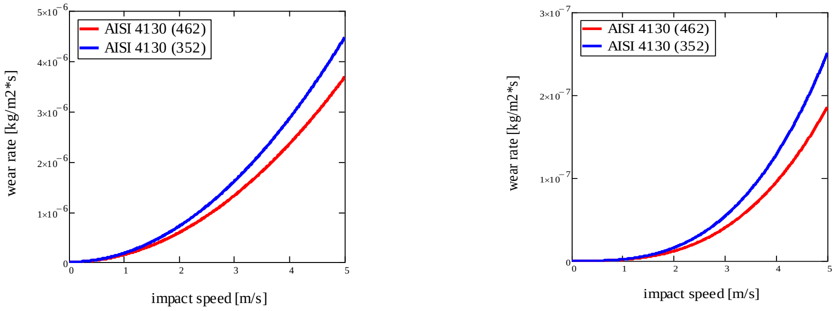
In the petroleum industry there is a compromise for the operation of the PC pumps at certain speeds to ensure the pumping pressure and at the same time to achieve the maximum wear life [7].

In this paper we will analyze the influence of the impact speed of the abrasive particles carried by the pumped fluid (for the values in table 2) and the target surface material (from table 1), in order to be able to estimate the optimal operating speed so that the wear of the PCP components is minimal and to keep the dosage adequate for the extracted liquid. The results are presented in figures 1 to 3 for rotor materials and figures 4 to 6 for stator materials, for each model analyzed, given the parameters specified in tables 1 and 2.



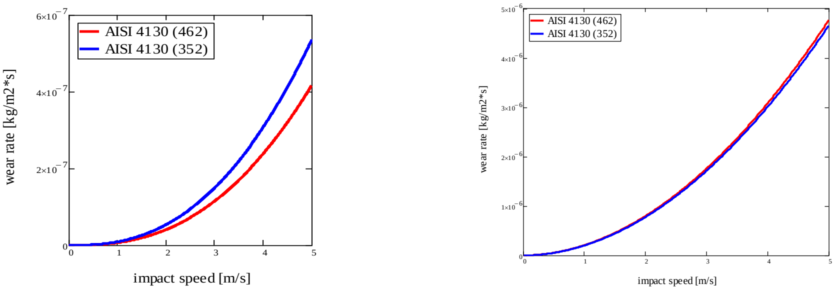
(a) (b)

Figure 1. Wear rate variation depending on the impact speed for the analyzed materials of the rotor, with (a) first Finnie model and (b) the second Finnie model.



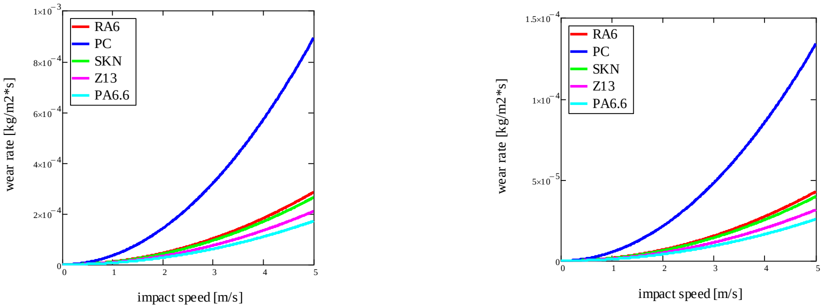
(a) (b)

Figure 2. Wear rate variation depending on the impact speed for the analyzed materials of the rotor, with (a) first Hutchings model and (b) the second Hutchings model.



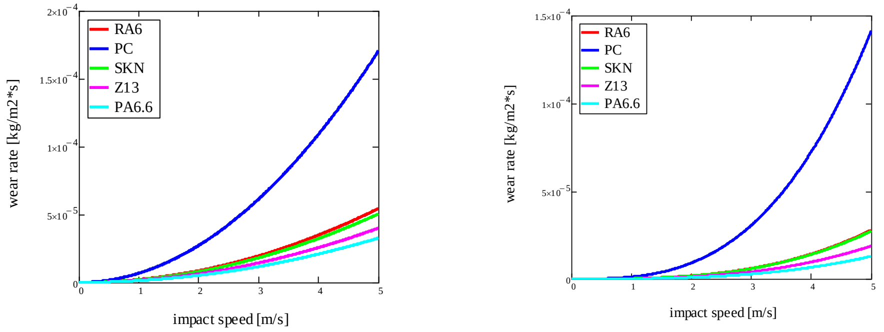
(a) (b)

Figure 3. Wear rate variation depending on the impact speed for the analyzed materials of the rotor, for (a) Bitter model and (b) Sundararajan model.



(a) (b)

Figure 4. Wear rate variation depending on the impact speed for the analyzed materials of the stator, with (a) first Finnie model and (b) the second Finnie model.



(a) (b)

Figure 5. Wear rate variation depending on the impact speed for the analyzed materials of the stator, with (a) first Hutchings model and (b) the second Hutchings model

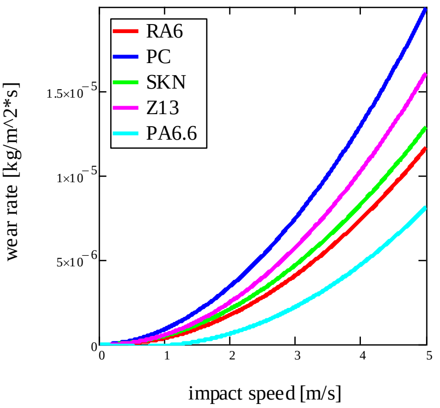


Figure 6. Wear rate variation depending on the impact speed for the analyzed materials of the stator, for Sundararajan model

The results obtained in graphs 1 to 6 are presented in Table 3 for rotor materials and in Table 4 for stator materials. The tables include the values of the erosive wear intensity measured for three of the particle impact speeds, for v = 1; 2; 4 [m/s] based on the recommendations of [14], for each analyzed model. The results for Bitter model for the stator materials are not displayed in the table due to their unreliable values - the Bitter model cannot be applied to elastomeric materials.

Table 3. **The values of the erosion wear rate for the rotor materials**

Model	**Material**	**Erosion wear rate depending on three considered values of the impact speed [kg/m^2·s]**		
		$v_1 = 1$	**$v_2 = 2$**	**$v_3 = 4$**
Finnie 1	Rotor 1	$1.54\cdot10^{-7}$	$6.10\cdot10^{-7}$	$24.63\cdot10^{-7}$
	Rotor 2	$1.87\cdot10^{-7}$	$7.48\cdot10^{-7}$	$29.93\cdot10^{-7}$
Finnie 2	Rotor 1	$1.15\cdot10^{-7}$	$4.62\cdot10^{-7}$	$18.5\cdot10^{-7}$
	Rotor 2	$1.40\cdot10^{-7}$	$5.61\cdot10^{-7}$	$22.4\cdot10^{-7}$
Hutchings 1	Rotor 1	$1.47\cdot10^{-7}$	$5.89\cdot10^{-7}$	$23.55\cdot10^{-7}$
	Rotor 2	$1.79\cdot10^{-7}$	$7.15\cdot10^{-7}$	$28.62\cdot10^{-7}$
Hutchings 2	Rotor 1	$0.01\cdot10^{-7}$	$0.11\cdot10^{-7}$	$0.95\cdot10^{-7}$
	Rotor 2	$0.02\cdot10^{-7}$	$0.16\cdot10^{-7}$	$1.28\cdot10^{-7}$
Bitter	Rotor 1	$0.06\cdot10^{-7}$	$0.40\cdot10^{-7}$	$2.37\cdot10^{-7}$
	Rotor 2	$0.09\cdot10^{-7}$	$0.53\cdot10^{-7}$	$3.06\cdot10^{-7}$
Sundararajan	Rotor 1	$2.07\cdot10^{-7}$	$7.96\cdot10^{-7}$	$30.02\cdot10^{-7}$
	Rotor 2	$2.01\cdot10^{-7}$	$7.75\cdot10^{-7}$	$30.75\cdot10^{-7}$

Table 4. **The values of the erosion wear rate for the stator materials**

Model	**Material**	**Erosion wear rate depending on three considered values of the impact speed [kg/m^2·s]**		
		$v_1 = 1$	**$v_2 = 2$**	**$v_3 = 4$**
Finnie 1	Stator 1	$11.39\cdot10^{-6}$	$45.60\cdot10^{-6}$	$182.3\cdot10^{-6}$
	Stator 2	$35.66\cdot10^{-6}$	$140\cdot10^{-6}$	$570.5\cdot10^{-6}$
	Stator 3	$10.57\cdot10^{-6}$	$42.3\cdot10^{-6}$	$169.1\cdot10^{-6}$
	Stator 4	$8.39\cdot10^{-6}$	$33.5\cdot10^{-6}$	$134.28\cdot10^{-6}$
	Stator 5	$6.84\cdot10^{-6}$	$27.4\cdot10^{-6}$	$109.5\cdot10^{-6}$
Finnie 2	Stator 1	$1.71\cdot10^{-6}$	$6.83\cdot10^{-6}$	$27.34\cdot10^{-6}$
	Stator 2	$5.35\cdot10^{-6}$	$21.4\cdot10^{-6}$	$85.58\cdot10^{-6}$
	Stator 3	$1.58\cdot10^{-6}$	$6.34\cdot10^{-6}$	$25.37\cdot10^{-6}$
	Stator 4	$1.26\cdot10^{-6}$	$5.03\cdot10^{-6}$	$20.14\cdot10^{-6}$
	Stator 5	$1.02\cdot10^{-6}$	$4.11\cdot10^{-6}$	$16.43\cdot10^{-6}$
Hutchings 1	Stator 1	$2.17\cdot10^{-6}$	$8.80\cdot10^{-6}$	$3.48\cdot10^{-6}$
	Stator 2	$6.82\cdot10^{-6}$	$27.27\cdot10^{-6}$	$10.91\cdot10^{-6}$
	Stator 3	$2.02\cdot10^{-6}$	$8.71\cdot10^{-6}$	$3.23\cdot10^{-6}$
	Stator 4	$1.60\cdot10^{-6}$	$6.42\cdot10^{-6}$	$2.56\cdot10^{-6}$
	Stator 5	$1.31\cdot10^{-6}$	$5.24\cdot10^{-6}$	$2.09\cdot10^{-6}$
Hutchings 2	Stator 1	$0.22\cdot10^{-6}$	$1.77\cdot10^{-6}$	$72.37\cdot10^{-6}$
	Stator 2	$1.13\cdot10^{-6}$	$9.05\cdot10^{-6}$	$14.22\cdot10^{-6}$
	Stator 3	$0.22\cdot10^{-6}$	$1.74\cdot10^{-6}$	$13.93\cdot10^{-6}$
	Stator 4	$0.15\cdot10^{-6}$	$1.19\cdot10^{-6}$	$9.59\cdot10^{-6}$
	Stator 5	$0.10\cdot10^{-6}$	$0.82\cdot10^{-6}$	$7.39\cdot10^{-6}$

Table 4 (continued)

Model	**Material**	**Erosion wear rate depending on three considered values of the impact speed [kg/m$^2\cdot$s]**		
		$v_1 = 1$	**$v_2 = 2$**	**$v_3 = 4$**
Sundararajan	Stator 1	$0.36\cdot10^{-6}$	$1.72\cdot10^{-6}$	$7.39\cdot10^{-6}$
	Stator 2	$0.88\cdot10^{-6}$	$3.42\cdot10^{-6}$	$12.97\cdot10^{-6}$
	Stator 3	$0.49\cdot10^{-6}$	$2.08\cdot10^{-6}$	$8.29\cdot10^{-6}$
	Stator 4	$0.55\cdot10^{-6}$	$2.48\cdot10^{-6}$	$10.27\cdot10^{-6}$
	Stator 5	$0.\ 70\cdot10^{-6}$	$0.63\cdot10^{-6}$	$4.72\cdot10^{-6}$

5. Conclusions

Using the mathematical models, we analyzed three different impact speeds of the particles transported by the fluid and the specific materials of the rotor-stator coupling. These parameters are important for optimizing the constructive form of the PCP.

By comparing the results for both Finnie models, there is a higher wear rate for the first model, a difference given by the fact that the percentage of abrasive particles, considered by the first model is 50%, compared to only 10% in the case of the second model (the models are designed based on this assumptions).

Because the two Hutchings models have different premises the values of the wear rate are different; for the second model it is found a higher value, which highlights the effect produced by micro-cutting (for the first model) compared to the wear generated by the plastic deformation of the target surface (for the second model).

By comparing the values in Table 3 and 4 between the Finnie, Hutchings, Bitter and Sundarajan models, stand out that the lowest values are given by the Hutchings 2 and Bitter model for rotor materials and Hutchings 2 and Sundarajan for stator materials, but close to those of the Finnie models. Bitter models are more appliable for steel materials than for elastomeric materials.

The influence of the material characteristics has a particular importance; according to the results of Figures 1 to 6 and Tables 3 and 4, a higher hardness can reduce the erosive wear rate by two to five times.

For Finnie models it is highlighted that with the increase in the percentage of particles, the wear rate will increase significantly for the rotor materials and has a lower increase for the stator materials.

For all the analyzed models figures 1 to 6 show an exponential increase of the wear rate with the increase of the speed; the variation is differentiated according to the material of the incident surface and by the hardness and their composition.

By reducing the working speed and thus the flow rate of the fluid to half of its value, according to the values in Tables 3 and 4, it is found that the wear rate is reduced by up to three to four times.

For both Finnie (Figure 1) models it stands out that hard chromed steel has a lower wear rate than non-coated steel; for stator materials (figure 4) it is found that the PC elastomer has the highest wear rate, followed by RA6, SKN, Z13 and PA6.6 has the best behavior. This ranking is also available for Hutchings models (figures 2 and 5); Bitter model is valid only for the rotor materials (figure 3.a) and for the Sundarajan model the ranking remains valid only for the stator materials (figure 6). According to figure 3.b and the values from table 3 (for the Sundarajan model) the hardness of the rotor materials does not have a major influence because the values of the wear rate are almost identical. For stator materials, the Bitter model cannot be applied; in this case the values are inconsistent with reality and are much different from those experimentally determined [1][2][3].

There is a higher wear rate of stator materials than for the rotor materials, which makes the stator the part of the coupling that wears out faster.

The results obtained with any of the four models used for the analysis of the erosive wear rate are close. By performing experimental determinations, as presented in paper [1][2][3] and by comparison of the results with those calculated based on erosion models it can be validated the models that best approximate the value of the erosive wear intensity for PCP.

REFERENCES

[1] *A. Mulla, N.N Antonescu, R.G. Ripeanu,. M.G. Petrescu, I.N. Ramadan*, „Research on wear behavior of rotor-stator materials for screw pumps”, EMERG, Volume IX, Issue 2/2023, ISSN 2668-7003, ISSN-L 2457-5011

[2] *M. I. Al Naboulsi, A.I. Popovici M. Ammar, N. N. Antonescu, M.G. Petrescu*, „Erosion Wear Analysis With Classical Models For The Couple Piston-Cylinder Materials Of Drilling Fluid Pumps,” Journal of the Balkan Tribological Association 28(5), pp. 626-648, 2022

[3] *M.I. Al Naboulsi, N.N. Antonescu, A. Dinita, M. Moroasnu*, „Tribological characterization of some elastomers used at progressive cavity and piston pumps”, ICMMEN Matec Web of Conference, 318, 01016, 2020

[4] *A. Tudor*, „Frecarea şi uzarea materialelor” (Friction and materials wear) – Editura Bren, Bucureşti, 2002

[5] *I.M. Hutchings*, „A model for the erosion of metals by spherical particles at normal incidence”, Wear 70, pp 269-281, 1981

[6] *A.I. Popovici*, „Contribuţii la studiul efectelor uzării asupra performanţelor sistemelor hidraulice de reglare automată” (Contribution to study of wear effects on the performance of the automatic controlled hydraulic systems), PhD Thesis, Bucharest, 2012

[7] *M. Delpassand*, „Progressing Cavity (PC) Pump Design Optimization for Abrasive Applications”, 10.2118/37455-MS, (1997).

[8] *A. Tudor, M. Vlase*, „Uzarea materialelor” (Material wear), Ed. Bren, Bucharest, 2010

[9] *T.M. Hunt*, „Handbook of wear debris analysis and particle detection in liquids”, Elsevier Applied Science, London, 1993

[10] *Y.F. Wang, Z.G. Yang*, „Finite element model of erosion wear on ductile and brittle materials”, Wear 165, pp 871-878, 2008

[11] *A. Akar*, „Estimating erosion in oil and gas pipe line due to sand presence”, Master’s Degree Thesis, ISRN: BTH-AMT-EX2011/D-16-SE, Sweden, 2011

[12] G. *Sundararajan*, „A comprehensive model for the solid particle erosion on ductile materials”, Wear 149, pp 111-127, 1991

[13] *G. Sundararajan*, „The depth of palstic deformation beneath eroded surfaces: the influence of impact angle and velocity, particle shape and material properties”, Wear 149, 1991, pp 129-153

[14] L.V. Whittaker, „Evaluation and analysis of wear in progressive cavity pumps”, PhD Thesis, Great Britain, september 2003

Authors' biographies

Ammar Fa. MULLA holds a bachelor's degree in Mechanical Engineering from the Mechanical and Electrical Faculty of Petroleum-Gas University of Ploiesti, specializing in Hydrocarbon Transport and Storage Equipment, Master's degree in Risk Management and Reliability Engineering of Oil and Petrochemical Equipment. Working as a 3D modeling engineer on piping projects at Petrostar SA Ploiesti.

Email: ammar_mulla18@yahoo.com

Aristia-Ioana POPOVICI holds a PhD degree in Mechanical Engineering and is a university lecturer at the Faculty of Mechanical Engineering and Robotics in Construction - University of Civil Construction of Bucharest. She is a member of the Romanian Tribological Association. Her research is focused on erosion wear characteristics and 3D modeling of mechanical equipment’s.

Email: arys_jo@yahoo.com

M. Iyad Al NABOULSI holds a bachelor's degree in Mechanical Engineering from the Faculty of Mechanical and Electrical Engineering of University of Damascus Syria and an Executive MBA in Management and Human Resource Development (MHR). Experienced professional with more than two decades of expertise in the Oil & Gas sector, with a focus on engineering, project management, operations, contracts, and project development. Proficient in maintaining adherence to safety standards, facilitating process enhancements, and attaining corporate goals.

Email: lyadnaboulsi@yahoo.com

Niculae Napoleon ANTONESCU holds a PhD in Mechanical Engineering. He is a full university professor, PhD adviser and he introduced the tribology course for the first time in Romania. He was elected vice-president of the Romanian Tribological Association, he is member of Technical Sciences Academy of Romania and the Academy of Mining Sciences in Ukraine. A corollary of the appreciation of his didactic, scientific, managerial activity is represented by the large number of orders, medals, diplomas, distinctions, titles that he received and last but not least, he was always with the students whom he understood and helped.

Email: nnantonescu@upg-ploiesti.ro