ASPECTS REGARDING THE USE OF DYNAMIC MODELS IN THE STUDY OF VOLUMIC LOSSED BETWEEN PISTON AND BARREL AT BOREHOLE PUMPS

Stoian-Albulescu Ion[1], Dinu Florinel[2], Pana Ion[2], Dinu Paul-Octavian[3]

[1] SC Brent Exploration & Production, Blejoi DN1, km 5, **România**;
[2] Universitatea Petrol-Gaze din Ploiești, Bd. București 39, Ploiești, **România**;
[3] S.C. DEPOGAZ, Str. Ghe. Gr. Cantacuzino 184, Ploiești, **România.**
email: si@brent-mgt.ro, flgdinu@yahoo.com, ion.pana@upg-ploiesti.ro, dinup89@yahoo.com.

ABSTRACT

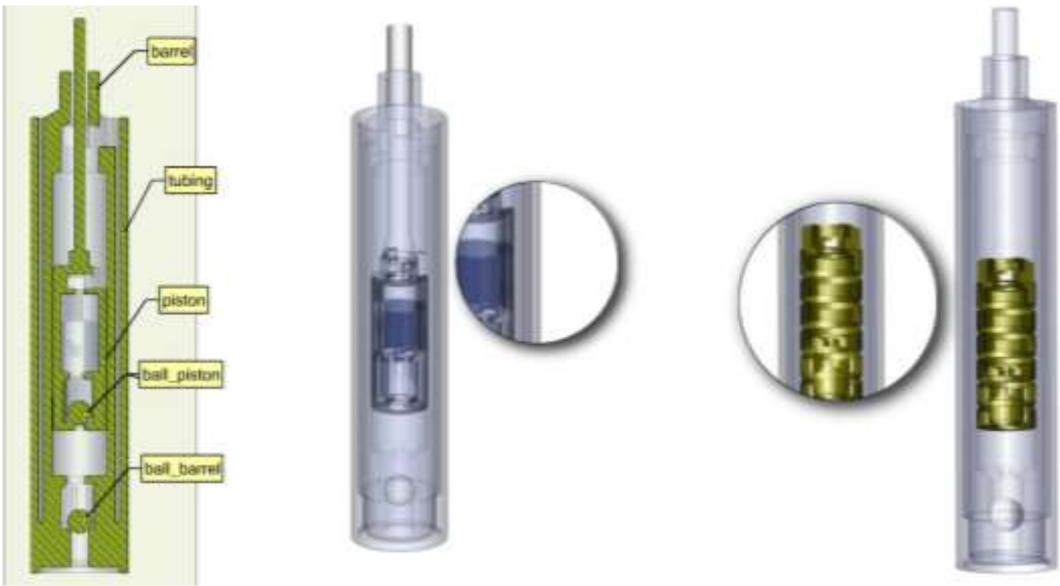
The article presents a dynamic model of the sealing system between the piston and the barrel, at a borehole pump used for the oil extraction. The model is made in the LMS Amesim program and allows a precise modeling of the geometric, cinematic and dynamic elements of the piston. Two constructive types of piston: smooth and with ditches designs are analyzed.

Key words: sucker rod pump, flow losses, LMS Amesim dynamic model.

INTRODUCTION

The sucker rod pump system is still the most widely used oil extraction system [10, 11]. Studies and research are being carried out continuously in this field, with important results in improving the operating performance and increasing the durability of the extraction system. Some good description of a **S**ucker **R**od **P**umping **S**ystem (SRPS) model are presented in [7, 13]. An image with the downhole pump is also in Figure 1. The domain is a multidisciplinary one involving the participation of mechanics, automation, chemists, drilling and extraction engineers etc. Torres and Schntman in [16] have shown that the satisfactory oil well operations with sucker-rod pumps are attributed to the techniques and methods which are able to control the performance of the well. They developed a mathematical dynamic model of a real sucker-rod pump based on the identification techniques by using input-output measured data. The developed model is able to deal satisfactorily with uncertainties and variations, such as for instance, the pumped fluid composition. Weicheng et al. in [17] have created an improved SRPS model, considering the impact of the fluid flowing into the pump on SRPS dynamic behaviors. Two oil wells are served to compare the difference between the current SRPS model and the improved one. The results indicate that the current SRPS model is unsatisfactory in calculating the pump filling, and this difference will increase with the reduction of the oil well deliverability; the influence of fluid flowing into the pump on the pump load cannot be ignored when the fluid velocity is high.

Therefore, the fluid flowing into the pump is necessary to be considered in order to improve the simulation accuracy. The problem of the damage to the rod string is frequently studied as in [6]. Causes of failures of sucker-rod pumping units, occurring on oil-and-gas production enterprises, include breakages and twist-offs of sucker-rods, which are exposed to static and dynamic loads [9]. The correct equipment selection is one of the directions of reliability growth of the rod strings. The dynamometer cards are commonly used to analyze down-hole working conditions of pumping systems in actual oil production [8, 19]. The centralizers can affect the polished rod load and the pumping system efficiency [20]. The influence of centralizers, stroke length, pumping speed and flooding period on SRPS efficiency and production has been studied and the results indicate that the SRPS efficiency and production rate decrease while the number of centralizer increases.



a b c

Figure 1. The main elements of a borehole pump: a. a section through the pump; b. a polished piston; c. a piston with ditches.

The problems of modernization of the existing sucker rod pump, the basic technical problems and new technical solutions that have been successfully applied in the innovative design of SRPS *BeeOilPump* are presented in [5]. The method for using finite element analysis to analyze and study the velocity and liquid pressure distribution of a sucker rod pump is proposed in [13]. The results indicate that this method can simulate the fluid in sucker rod pump reliably. Dinu in [1] presents the results of experimental researches of the pressure distribution along the extraction pump plunger during its running. The results are based on research [4] that involved many experimental determinations made in Petroleum and Gas University of Ploiesti. It ascertains that the law of pressure variation depends, mainly, on the existing radial gap, elastic radial deformation of the cylinder and on the type of the pumped fluid. All these elements influence the volumetric efficiency of the extraction pump. Also Dinu in [2, 3] reveals that the hydraulic yield of the borehole pumps is determined by the leakage between the piston and the cylinder and the two valves, fixed and mobile. The leakages through the ball valves occur during their sealing and as a result of the ball and seat wear. The specialty literature presents theories and results of the research on the stroke yield, the borehole pump admission coefficient influenced, as a rule, by the presence of gas in the liquid sucked up by the pump, the liquid leakage through the space between the piston and the cylinder of the borehole pumps, both for the new pumps and for the highly worn ones, without insisting on these leakages yield and without identifying the liquid leakage through valves at the piston up stroke and down stroke. For this reason, the paper presents the calculation program for the liquid leakage efficiency through the ball valves of the production wells, for which a computer model has been elaborated. The ball movement of the extraction pump valve in a viscous environment and limited space is well presented in [15]. It was experimentally determined the resistance to valve ball movement in the liquid.

CALCULATION OF DEBIT LOSSES

FLOW RELATIONSHIPS

Next, we propose to evaluate the debit losses using models developed in LMS Amesim. This program is offered free of charge by Siemens for students and university teachers https://www.plm.automation.siemens.com/global/en/index.html. The LMS Amesim program includes the facilities of a multidisciplinary software product, combined with the support of a knowledge base on engineering issues. Models close to real-world systems with good computer precision can be made, as shown below. The main issue we are looking at is the assessment of the liquid flow that is lost through the cylinder - piston gap in different technological and geometric conditions. In the paper [10] several calculation relations are determined, based on numerous experimental determinations made on static (fixed pump pistons) or dynamic (the piston of the pump is displaced at a constant speed). These relationships are valuable because they express the actual flow of extraction pumps and can be used as a comparison with the theoretical models. For the static case, the proposed relationship (1) is:

$$Q_{cs} = 563275.5\,\frac{D^{1.9944}\delta^{2.3374}P}{\mu\cdot L} \qquad (1)$$

and for the dynamic case the relationship is:

$$Q_{cd} = 370359.5\,\frac{D^{1.28}\delta^{2.08}P}{\mu\cdot L} + 0.53897\cdot D\cdot\delta\cdot V \qquad (2)$$

In relationships (1) and (2) the units used for the variables are: $Q_c\ [cm^3/s]$; $,L\ [cm]$; $P\ [atm]$; $\mu\ [cP]$; $V\ [cm^3/s]$. Using the above relationships provides an error [10] (relationship 3) of between 30-35%, which is closest to experimental data compared with that of the other researchers.

$$E_r = \frac{100}{n}\sum_{i=1}^{n}\frac{|Q_{ri}-Q_{ci}|}{Q_{ri}} \qquad (3)$$

The extraction pumps are made at a specific nominal diameter: of the barrel on the schedule (-40, -20, 0, + 20, + 40 mils) and are fitted with pistons for the schedule (-7, -5, -3, -2, -1, 0, +1, + 2, + 3, +5, +7 mils). The piston and barrel wear increases volumic losses and decreases the volume output. In terms of lubrication conditions, we should have bigger gaps. This ensures greater use of the pump and prevents the piston from catching the abrasive and corrosive fluids. So the extraction pump gap (piston – barrel) depends on the technological conditions and the physical & chemical properties of the fluid extracted.

STATIC CASE MODEL

The determinations in the LMS Amesim program were made on the model shown in Figure 2, a. The model includes: two pressure sources corresponding to the petroleum deposit (outlet) and the exit of the pump (inlet); the properties of the petroleum; the gap between barrel and piston, Figure 2, b. The relationships of the flow rate of volumetric losses used in the program LMS Amesim, for a concentric respectively eccentric geometry gap are respectively (4), (5), where the eccentricity e can be calculated with relationship (6). We have obtained the volumetric flow losses Figure 2, c (that is an information provided by the program) and volumetric efficiency (that is an information obtained with post processing facility of the program) Figure 2, d. The comparison between the values offered by the LMS program and the values obtained by the experiment (relationship 2) is made in the last three columns of the Table 1.

$$Q_{cs,LMS} = \frac{\pi \cdot D \cdot \delta^3}{12\mu} \cdot \frac{p_a - p_b}{L} \frac{\rho(p_{mid})}{\rho(0)} \quad (4)$$

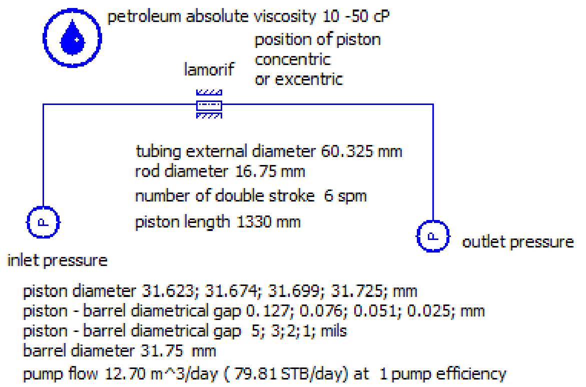
$$Q_{cs,LMS} = \frac{\pi \cdot D \cdot \delta^3}{12\mu} \cdot \frac{p_a - p_b}{L} \frac{\rho(p_{mid})}{\rho(0)} (1 + 1.5\varepsilon^2) \quad (5)$$

$$e = \varepsilon \frac{D-d}{2} \quad (6)$$

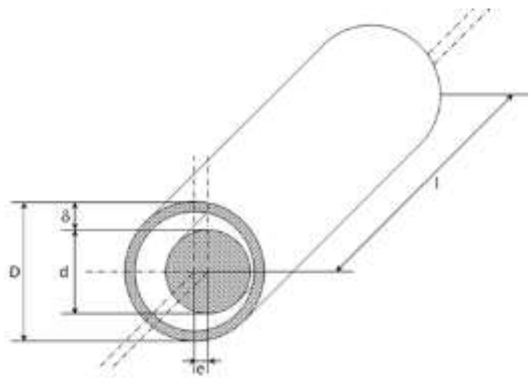
The values approaching the experimental values are those in the column 7, for an eccentricity ratio ε = 0.5. It can be noticed that for ε = 0, the loss values (obtained with LMS Amesim) are lower (column 8). In conclusion, the values obtained using the LMS Amesim program correspond to the experimental values as an order of magnitude, but in condition of existence of an eccentricity barrel - piston.

Table 1. Comparison between the experimental results and computer simulation model in LMS Amesim, for the static case.

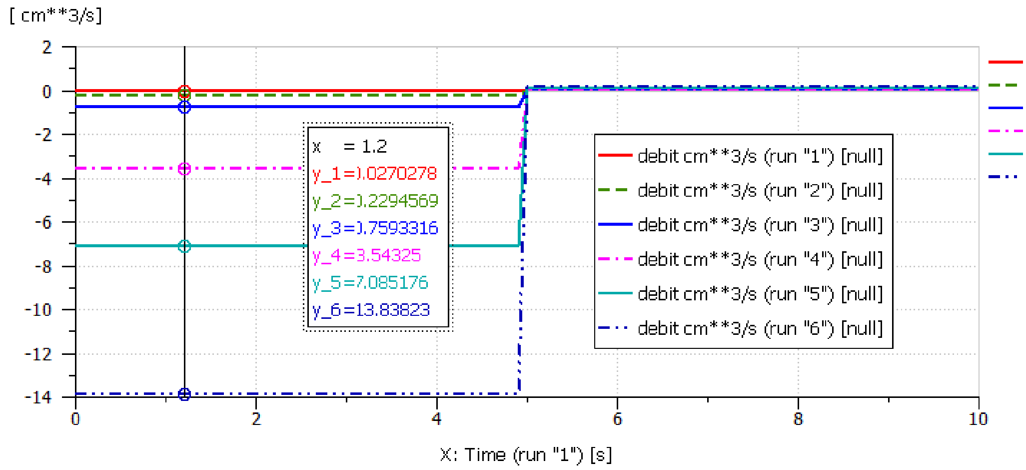
Piston diameter, *D*	**Radial gap, δ**	**Pressure difference, *P***	**Dynamic viscosity, μ**	**Piston length, *l***	**Flow losses, Q_{cs}**	**Flow losses, $Q_{cs,LMS}$, $\varepsilon = 0.5$**	**Flow losses, $Q_{cs,LMS}$, $\varepsilon = 0$**
cm	*cm*	*atm*	*cP*	*cm*	cm^3/s	cm^3/s	cm^3/s
1	**2**	**3**	**4**	**5**	**6**	**7**	**8**
3.1725	0.00127	150	10	133	0.10	0.03	0.02
3.1699	0.00254	150	10	133	0.54	0.23	0.17
3.1674	0.00381	150	10	133	1.37	0.76	0.55
3.1623	0.00635	150	10	133	4.55	3.54	2.57
3.1590	0.00800	150	10	133	7.80	7.08	5.15
3.1550	0.01000	150	10	133	13.11	13.83	10.07



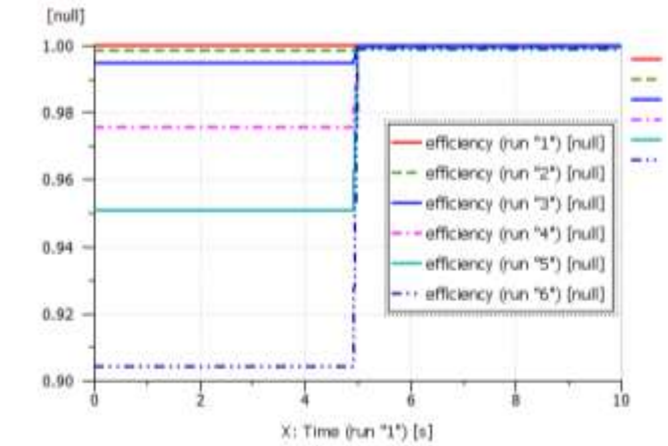
a



b



c



d

Figure 2. Static case results: a. Sucker rod pump model in LMS Amesim, static position of piston; b. Model sketch; c. Flow losses, piston – barrel in eccentric position; $\varepsilon = 0.5$*; d. Volumetrical efficiency of the sucker rod pump; run 1 diametrical gap 0.200 mm; run 2 diametrical gap 0.160 mm; run 3 diametrical gap 0.127 mm; run 4 diametrical gap 0.076 mm; run 5 diametrical gap 0.051 mm; run 6 diametrical gap 0.025 mm.*

DYNAMIC CASE MODEL

An advanced model of the piston pump is used in Figure 3, a. It is possible to model the piston with its construction details. Thus we can insert: the two chambers of the piston (1) and (3) with the corresponding liquid volumes (4) and (5). The room volumes are variable and take into account the piston movement. Between the two chambers is the piston (2) at which the gap between the piston and the barrel is formed. The properties of the oil are introduced with the block (7) and the block (8) is used for the gravity acceleration. The lost volume of oil is measured with the transducer (13). The volumetric flow of the pump is measured with the instruments (9). The element (11) introduces the stroke efficiency and element (12) splits the cinematic signals (piston velocity and displacement). The model allows the piston rod diameter to be specified, to impose the force at the piston rod and kinematic elements. In the case represented, the speed of the piston, corresponding to the crank shaft mechanism on the surface, is introduced, a mechanism completed with the walking beam. The speed of the piston is calculated with the relation [10]:

$$V_A = \frac{\omega a}{f^2}\left[\frac{pr\left(b^2+l^2-f^2\right)sin(\varphi+\varphi 0)}{\sqrt{4l^2b^2-(b^2+l^2-f^2)^2}} + \frac{\left(f^2+r^2-p^2\right)}{2}\right] \tag{7}$$

where the geometric elements of the beam unit are shown in Figure 3, b. A file (10) with these values is used into the model. For the set of values in Figure 3, c, the variation of the loss rate is shown in Figure 3, d. It is noted that during the 5 seconds of the lifting stroke, the flow stabilizes around 20 l / min. This flow is only for the lifting stroke. At the descent race, the liquid runs backwards with very low flow rates. Running the application for the five sets of values shown in Table 2 the volumetric efficiencies are obtained (the last column of the Table 2).

The efficiency calculated with relationship (8) reports the flow losses rate at the average unit flow rate Q_n (relationship 9). We considered: a stroke length at surface of 2.1844 m (86 inch); pump diameter 31.75 mm (1 ¼ inch); pump speed 6 spm; pump efficiency 1; stroke efficiency E_s =0.85; $Q_p = \frac{\pi 0.03175^2}{4} 2.1844 \cdot 6 \cdot 0.85 = 12.70\ \frac{m^3}{day}$.

$$E_{vn} = \frac{V_{cd,LMS} \cdot N}{Q_n} \tag{8}$$

$$Q_n = \frac{\pi D^2}{4} S \cdot N \cdot E_s \tag{9}$$

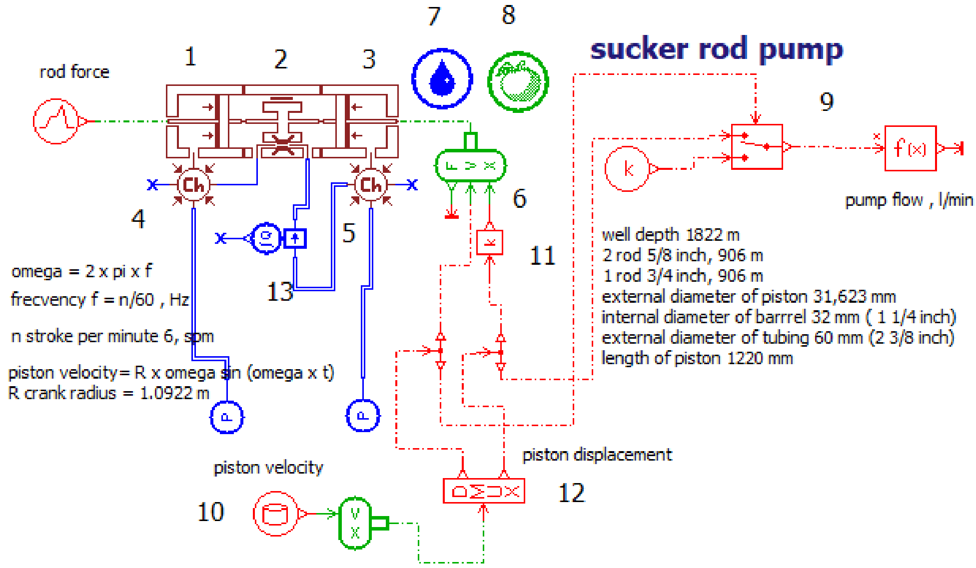
The equations of the model used by LMS Amesim are:

$$V_{cd,LMS} = \int_0^T Q_{cd} dt \tag{10}$$

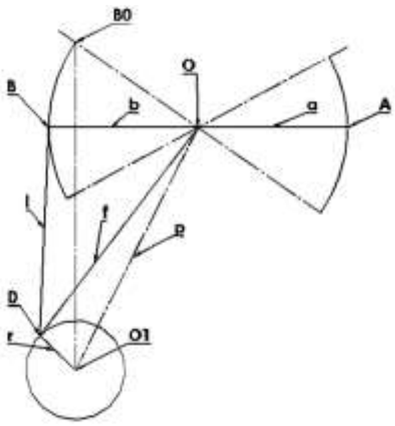
$$Q_{cd} = \frac{\pi \cdot D \cdot \delta^3}{12\mu} \cdot \frac{p_a - p_b}{l} \frac{\rho(p_{mid})}{\rho(0)} \left[1 + 1.5\left(\frac{\varepsilon}{\delta}\right)^2\right] + \frac{v^+ + v^-}{2}\ \pi \cdot D \cdot \delta \tag{11}$$

$$F^- = \frac{\pi\ (D-2\delta)}{2} \cdot (p_a - p_b) \cdot \delta + \mu \cdot l \cdot (v^+ - v^-) \cdot \frac{\pi\ (D-2\delta)}{\delta} \tag{12}$$

$$F^+ = \frac{\pi\ D}{2} \cdot (p_a - p_b) \cdot \delta - \mu \cdot l \cdot (v^+ - v^-) \cdot \frac{\pi \cdot D}{\delta} \tag{13}$$



a

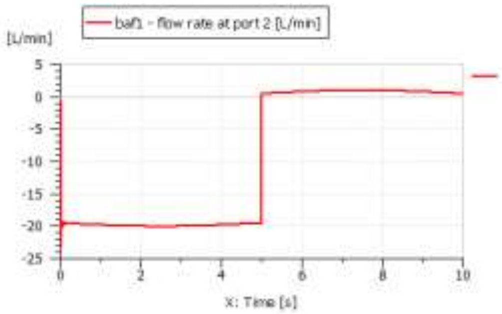


$$\varphi = \widehat{DO_1B_0}\ ;\ \varphi 0 = \widehat{OO_1B_0}$$

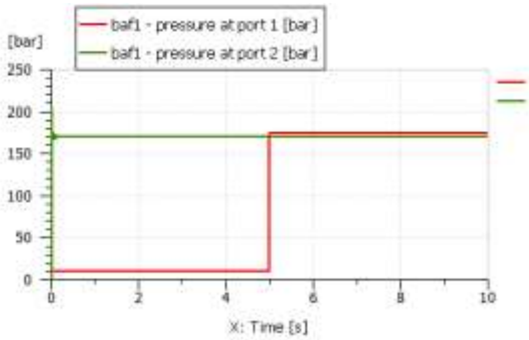
b

Title	Value	Unit
index of hydraulic fluid	0	
external piston diameter	**31.75**	mm
clearance on diameter	**0.508**	mm
eccentricity between piston and sleeve	**0.254**	mm
contact length		
variable contact length	no	
contact length	**1220**	mm

c



d



e

Figure 3. Flow losses for a dynamic model of a smooth piston: a. model achieved in LMS Amesim; b. the sketch of beam pumping unit with the geometrical elements used in the calculus of piston velocity; c. geometrical characteristics of the piston; d. variation of flow losses Q_{cd}*; e. distribution of the pressure on both sides of the piston.*

Table 2. Results of computer simulation (model in LMS Amesim), dynamic case, smooth piston.

Clearance	Eccentricity	Piston diameter	Average flow	Lost volume*	Average Lost liquid flow	Efficiency
mm	*mm*	*mm*	*l/min*	*l*	*l/min*	-
1	**2**	**3**	**4**	**5**	**6**	**7**
0.508	0.254	31.750	88.20	1.66	9.96	0.89
0.533	0.266	31.725	88.06	1.93	11.58	0.87
0.558	0.279	31.699	87.91	2.21	13.26	0.85
0.584	0.292	31.674	87.78	2.52	15.12	0.83
0.609	0.304	31.650	87.64	2.85	17.10	0.80

The leakage flow rate is proportional with the cube of the clearance and the square of eccentricity and is then very sensitive to these 2 parameters, relation (11). The friction forces on the piston and on the envelop are calculated with relations (12, 13). It is worth pointing out that in this model the flow through the piston cylinder interstitial is modeled taking into account the velocity of piston, Figure 4 and the friction forces on the piston – barrel gap. For this reason, at small gaps LMS Amesim does not find a flow solution (1-7 mils), in agreement with the actual situation where the piston displacement is made with difficulty.

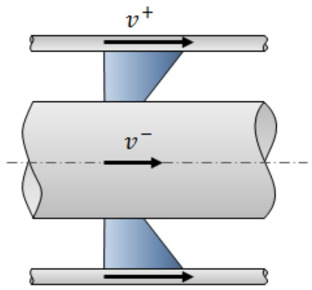


Figure 4. Scheme in which are figured the velocities for the barrel and the piston, velocities used in the relationships (11-13).

DYNAMIC CASE, PISTON WITH DITCHES

Since several pistons are realized with ditches, Figure 1, c in Figure 5, a, the model of this type of piston was made. The piston model is included under the *SC_2* sub module and it can be seen in Figure 5, b. Each hydraulic chamber *Ch* corresponds to a piston channel. Between the two chambers is the piston area through which a sealing gap is achieved. As with the smooth piston model of Figure 3 is introduced by the submodule of Figure 5, c the velocity of the piston calculated over time with the relationship (7). Because the sealing areas are much shorter (140 mm) and the pressure difference on each sealing area is smaller (Figure 5, d), the laminar flow rates are obtained. Thus, we have a solution to the flow problem at smaller interstices, with the lost volume of liquid indicated by the Table 3. In these conditions the volumetric efficiencies are better, compared with Table 2.

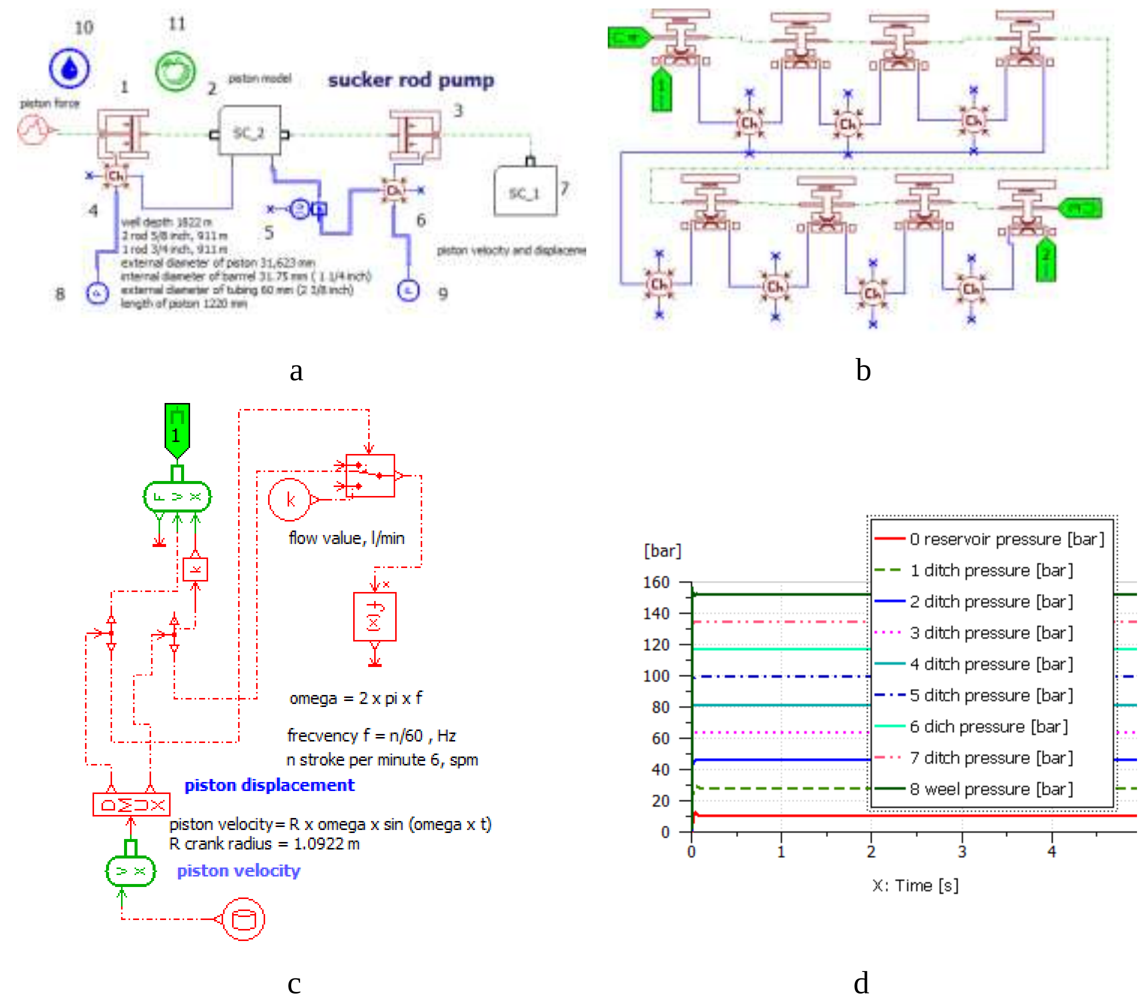


Figure 5. Piston with ditches: a. LMS Amesim model; b. sub model of piston with diches; c. sub model of piston velocity; d. pressure distribution through the piston length, into the hydraulic chambers.

Table 3. Results of the computer simulation (model in LMS Amesim) dynamic case piston with ditches.

Clearance	**Eccentricity**	**Piston diameter**	**Average flow**	**Lost volume***	**Average Lost liquid flow**	**Efficiency**
mm	*mm*	*mm*	*l/min*	*l*	*l/min*	-
1	**2**	**3**	**4**	**5**	**6**	**7**
0.508	0.254	31.750	88.20	1.11	6.66	0.92
0.533	0.266	31.725	88.06	1.26	7.56	0.91
0.558	0.279	31.699	87.91	1.43	8.58	0.90
0.584	0.292	31.674	87.78	1.62	9.72	0.89
0.609	0.304	31.650	87.64	1.82	10.92	0.88

CONCLUSIONS

- The models presented offer good consistency with the experimental data for the static case.
- High precision models are used which include forces acting on the depth pump including friction forces and velocities.
- The models are dynamic including piston displacement.
- The efficiency is calculated using the lost volume of liquid and the average flow rate of the pump.
- The pump piston is modeled without ditches and with ditches, and the influence of the lost volumes is evaluated. Efficiency is better when we use ditches.
- It is possible to ensure a laminar flow to the piston with ditches, which gives also better lubrication conditions.

Nomenclature

E_r – calculus error;

E_s – stroke efficiency;

Q_{cs} – static flow losses;

Q_{cd} – dynamic flow losses;

Q_{ri} – measured flow losses;

Q_{ci} – calculated flow losses;

P – pressure difference $p_a - p_b$;

p_a – input pressure;

p_b – output pressure;

v^- – velocity of the piston;

v^+ – velocity of the barrel;

D – barrel diameter of the sucker rod pump;

L – piston length;

N – sucker rod pump speed;

S – stroke length at surface;

V_A – piston velocity, at the head of the horse;

V – piston velocity;

V_{cd} – lost liquid volume;

T –upward stroke duration;

d – diameter of the piston;

e – eccentricity;

i – index of an experiment;

n – total number of experiments;

δ – radial gap between working barrel and piston;

ε – eccentricity ratio;

μ – dynamic viscosity of the oil;

$\rho(p_{mid})$ – density of petroleum at average pressure $p_{mid} = \frac{p_a+p_b}{2}$;

$\rho(0)$ – density of petroleum at normal pressure;

ω – angular velocity;

LMS –index calculated with LMS Amesim.

REFERENCES

[1]. Dinu F., Grigore V., Tudor I., *Distribuția presiunii în lungul pistonului pompei de extracție,* Buletinul Universității Petrol - Gaze din Ploiești, Volum LVI, Seria Tehnică, nr. 1/2004, p. 74-81, ISSN 1221-9371, Ploiești, 2004;

[2]. Dinu F., Grigore V., *Modelarea matematică a funcționării supapelor cu bilă care echipează pompele de extracție*, Contract nr. A.M.- 161/2000, S.N.P. „PETROM" S.A. București, 2000;

[3]. Dinu F., Grigore V., *Model matematic pentru determinarea randamentului supapei la cursa ascendentă și descendentă a pistonului pompei de extracție*, Buletinul Universității Petrol - Gaze din Ploiești, Seria Tehnică: Vol. LV, Nr. 3/2003, p. 13-18, Ploiești, 2003;

[4]. Dinu F., Grigore V., Tudor I., ş.a. *Model matematic pentru determinarea randamentului scurgerilor de lichid la nivelul pompelor de extracție cu piston*, Contract de cercetare S.C. Petroconsulting S.R.L, nr. A.M – 174/2001;

[5]. Ryzhov E.V., Ryzhov M.E., *Advanced sucker rod pump ¾ Bee Oil Pump increasing economic efficiency of oil production,* Comp. nanotechnology, 2016, Issue 2, 139–145;

[6]. Fitrianti H., Anwar H., *Analysis of Damage Rod String Components in Sucker Rod Pump In The Field Ss*, JEEE Vol. 7 No. 1, 2018;

[7]. Fozao K. F., Lissouck M., Lontsi F., Ngasa A., Mbanda N. *Hands-on model of sucker rod pumping facility for oil well production*, Journal of Petroleum and Gas Engineering Vol. 6(4), pp. 45-53, April, 2015;

[8]. Kun L., Gao X.W., Zhou H.B., Han Y., *Fault diagnosis for down-hole conditions of sucker rod pumping systems based on the FBH–SC method*, Pet. Sci12:135–147, 2015;

[9]. Novokshonov D.N., Baranov M.N., Božek P., Prajová V., Ivanova T.N., Korshunov A.I., *Constructing and calculating of multistage sucker rod string according to reduced stress.* Acta Montanistica Slovaca Volume 22, number 2, 107-115, 2017;

[10]. Petre N., Chiţu-Militaru P., *Extracția țițeiului prin pompaj cu prăjini*, Editura Tehnică, București, 1986;

[11]. Popescu C., Coloja M.P., *Extracția țițeiului şi gazelor asociate*, Editura Tehnică, București, 1993;

[12]. Popovici Al., *Cercetări asupra dinamicii instalațiilor de pompare pentru extracția petrolului*, Teză de doctorat, Institutul de Petrol și Gaze, Ploiești, 1976;

[13]. Shanpo J., Xiangzhen Y., Xiujuan Y., *Numerical Study of Fluid Flow in Sucker Rod Pump Using Finite Element Method*, Transport & Storage Civil Engineering, China University of Petroleum, Dongying 257061, China, print on line at https://www.ansys.com/-/media/ansys/corporate/resourcelibrary/conference-paper/2004-int-ansys-conf-142.pdf;

[14]. Stan Al. D., Tudor I., Stoianovici D., *Alunecarea exactă la pompele de adâncime*, Revista Mine, Petrol şi Gaze, nr. 3 şi 4, 1988;

[15]. Stoian Albulescu I., *Elemente de optimizare a unor operații de extracție*, Karta Graphic Ploiești, 2015;

[16]. Torres L.H.S., Schntman L., *Sucker-Rod Pumping System of Oil Wells: Modelling, Identication and Process Control*, 6th IFAC Conference on Management and Control of Production and Logistics, The International Federation of Automatic Control September 11-13, p. 260-265, 2013. Fortaleza, Brazil. 10.3182/20130911-3-BR-3021.00052, 978-3-902823-50-2/2013 © IFAC;

[17]. Weicheng L., Dong S., Sun X., *An Improved Sucker Rod Pumping System Model and Swabbing Parameters Optimized Design*, Hindawi Mathematical Problems in Engineering, Article ID 4746210, 15 pages, https://doi.org/10.1155/2018/4746210, 2018;

[18]. Zett C. *Studiul fenomenelor hidrodinamice la pompele de extracție în vederea măririi eficienței acestora,* Teză de doctorat, Universitatea „Petrol-Gaze" Ploieşti, 1998.

[19]. Zhang A., Gao X., *Fault diagnosis of sucker rod pumping systems based on Curvelet Transform and sparse multi-graph regularized extreme learning machine*, International Journal of Computational Intelligence Systems, Vol. 11, p. 428–437, 2018;

[20]. Zhen J.H., *The Sucker Rod Pump Parameters Optimization of Horizontal Wells with ASP Flooding Considering the Influence of Centralizers*. MATEC Web of Conferences 77 01026 (2016) DOI: 10.1051/matecconf/20167701026, ICMMR 2016;