INFLUENCE OF THE PRESSURE AND TEMPERATURE ON VOLUME LOSSES BETWEEN PISTON AND CYLINDER AT OIL EXTRACTION PUMPS

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ABSTRACT

The pressure and temperature influence oil density, viscosity, and modulus of elasticity. Their variation influences the flow through the piston-barrel interstitial to oil extraction pumps and the liquid flow that is lost through this gap. A model with 8 different liquid pressure zones was created in the LMS Amesim program to study fluid leakage through interstitial. The model is used in dynamic situations. The flow values of lost liquid are indicated and compared to the model with a single zone of liquid.

Key words: sucker rod pump, flow losses, viscosity, bulk, modulus of elasticity.

INTRODUCTION

The determination of the flow losses between piston and cylinder can be accomplished after many calculus relationships, according to the classical methods indicated in the literature. Based on experimental research, Petre proposed the computation relation [11]:

$$Q_{cs} = 563275.5 \, \frac{D^{1.9944} \delta^{2.3374} P}{\mu \cdot l} \tag{1}$$

for the static case, and for the dynamic case the relationship is:

$$Q_{cd} = 370359.5 \, \frac{D^{1.28} \delta^{2.08} P}{\mu \cdot l} + 0.53897 \cdot D \cdot \delta \cdot V \tag{2}$$

In the relationships (1) and (2) the units used for the variables are: Q_c $[cm^3/s]$; D, d, δ, l $[cm]$; P $[atm]$; μ $[cP]$; V $[cm/s]$. Piverdian [12] has established a more general relationship for the leakage flow of the eccentrically located piston:

$$Q_{cd} = \pi \cdot D \cdot \delta \cdot \left[\frac{(1+1.5c^2)}{12\mu} \cdot \frac{\delta^2 \cdot P}{l} - \frac{v_p}{2} \right] \tag{3}$$

Research made by the authors on the test stand of the extraction pumps within the production field Baicoi [6], led to the establishment of new relationships for the determination of liquid losses through the annular space between the piston and the pump cylinder under static conditions (4-5). Thus, for a 1¼ inch piston the relationship has the form:

$$Q_{cs} = 276891{,}53 \cdot \delta^{1{,}800861} \tag{4}$$

when the lost liquid flow is expressed in $\frac{cm^3}{5\,min}$, and δ is in cm. For a 1 ¾ inch piston the calculation relationship is:

$$Q_{cs} = 423304{,}14 \cdot \delta^{1{,}806535} \tag{5}$$

where the lost liquid flow is expressed in $\frac{cm^3}{5\,min}$. The relationships (4, 5) are valid for the following working conditions: the used fluid - diesel fuel and the working pressure 150 bars. Using of experimental researches (the echometer measurements), carried out in wells, a new empirical computation relation valid for the dynamic conditions was established:

$$Q_{cd} = 3.44 \cdot 10^5 \, \frac{D^{1.35}\delta^{2.04}P}{\mu \cdot l} + 0.366 \cdot D \cdot \delta \cdot V \tag{6}$$

The dynamic oil viscosity depends on the temperature and pressure. In the case under consideration, there is a large pressure difference between the two sides of the piston and therefore the viscosity can change. The fluid viscosity has been measured and indicated in many engineering works. It depends on the pressure and the type of fluid. The effect is generally neglected but becomes important at high pressures. The best-known model is the relationship:

$$\mu = \mu_0 \cdot e^{\alpha p} \tag{7}$$

where μ_0 and α depend on the nature of fluid Barus, [3]. The coefficient α has large variations between 10^{-8} and $1\,Pa^{-1}$. The formula is often used by engineers, sometimes in combination with temperature t dependence:

$$\mu = A\,\rho^{1/2} \cdot e^{(p+\rho^2 r)\frac{s}{t}} \tag{8}$$

where A, r, s are liquid constants. Mathematically, pressure-dependent viscosity makes the Navier-Stokes equation system much more complicated, Paloka [10]. The oil viscosity assessment is important for various oil refinery operations and operations. Rahuma has used some empirical models to estimate the oil viscosity of some reservoirs in Libya [10]. The purpose of the paper was to determine how different correlations Chew-Connally, Beggs-Robinson, Labedi indicate the oil viscosity, based on measured reservoir data: temperature, oil API gravity, gas-oil ratio, and saturation pressure. The results obtained on these models were compared with laboratory tests and showed the following differences: Chew-Connally (26.19 %); Beggs-Robinson (21.19%); Labedi (36.14%). Dinu indicates that the operating time and the volumetric efficiency of the extraction pumps are directly influenced by the gap used to the pumps mounting, the quality of the piston-cylinder jointing and the character of the extracted fluid [5, 6]. In some cases, the pump flow rate decreases considerably and therefore it is necessary to estimate the liquid losses prior to introducing the extraction pump into the well for mounting of the pump with the appropriate clearance. The normal wear of the piston and cylinder increases the initial radial gap and the fluid loss.

In [8], Leeuwen indicated the following relationship for dynamic viscosity with very good results for small clearances:

$$\mu = \mu_0 e^{\left(ln\frac{\mu_0}{\mu_R}\right)\left[\left(1+\frac{p}{p_R}\right)^z-1\right]} \tag{9}$$

where z is:

$$z = (\alpha \cdot p_R) ln^{-1}\left(\frac{\mu_0}{\mu_R}\right) \tag{10}$$

Also Al-Rawahi using the data from Omani Fahud oil field has developed correlations with an acceptable accuracy range [2]. Schmelzer shows that for most of the liquids viscosity increases with pressure. But there are special situations in which the viscosity drops to the increase of the pressure exemplified in the paper [14]. Stoian – Albulescu described some experiments to determine the oil viscosity in the paper [17]. The author used a mixture of 50% water and 50% oil. This situation is often encountered in oil wells with water production. Liu established an accurate model for the viscosities of liquids at high pressure. The model was applied to correlate the viscosities of 30 compounds at temperatures from 65 K to 460 K and pressures from 0.1 MPa to 253.1 MPa [9]. The calculated values from the established model agree well with the experimental data in all cases, with the average absolute relative deviations lower than 3.7%.

Also, density depends on pressure and temperature. Since at the borehole pump we can consider the temperature approximately constant, we will focus our attention only on the influence of the pressure using Dowson relationship [7]:

$$\rho = \rho_0 \frac{p_{R2}+1.34p}{p_{R2}+p} \tag{11}$$

Alomair has established very accurate models for the relationship between oil density and temperature (for a wide range of temperatures from 20 to 160°C) but also for the relationship between density and viscosity [1]. 30 samples of heavy oil with API densities ranging from 11.7 to 18.8 were used:

$$\rho = a + b \cdot API_{60°\mathrm{F}} + c \tag{12}$$

where $a = 1.0722498845, b = -0.00652625, c = -0.0006639;$

$$\ln\mu = a + \frac{b}{T^2} + c \cdot \rho^2 \cdot ln\rho \tag{13}$$

The density and viscosity are for the dead oil (oil with no free gas in the solution) and depend on the temperature interval $20 - 100$°C $a = 10.76,\ b = 275.3,\ c = 107.8$ and over 100°C $a = 7.93, b = 309.6, c = 61.51$. The average absolute error of the relations is: for density 0.61 %; for dynamic viscosity 8 %.

The elasticity of the pumped liquid also plays an important role in the appreciation of the flow losses between the piston and the cylinder. Because the extracted oil is a mixture of oil, water and natural gas, the compressibility coefficient for the pumped liquid mixture β is determined by the relationship:

$$\beta = \frac{1}{K} = i \cdot \beta_a + (1-i)\beta_t \tag{14}$$

For the determination of the compressibility coefficients for water β_a and petroleum β_t, two correlations of these parameters were determined according to the temperature, for a pressure range up to 150 bar, as follows [6] where t is the temperature expressed in °C:

$$\beta_a = a_1 + b_1 \cdot t + c_1 \cdot t^2 + d_1 \cdot t^{2.5} + e_1 \cdot t^3 \tag{15}$$

$$\beta_t = a_2 + b_2 \cdot t + c_2 \cdot t^2 + d_2 \cdot t^{2.5} + e_2 \cdot t^3 \tag{16}$$

The compressibility coefficients for gas are:

$$\beta_g = \frac{1}{\kappa \cdot p_c} \tag{17}$$

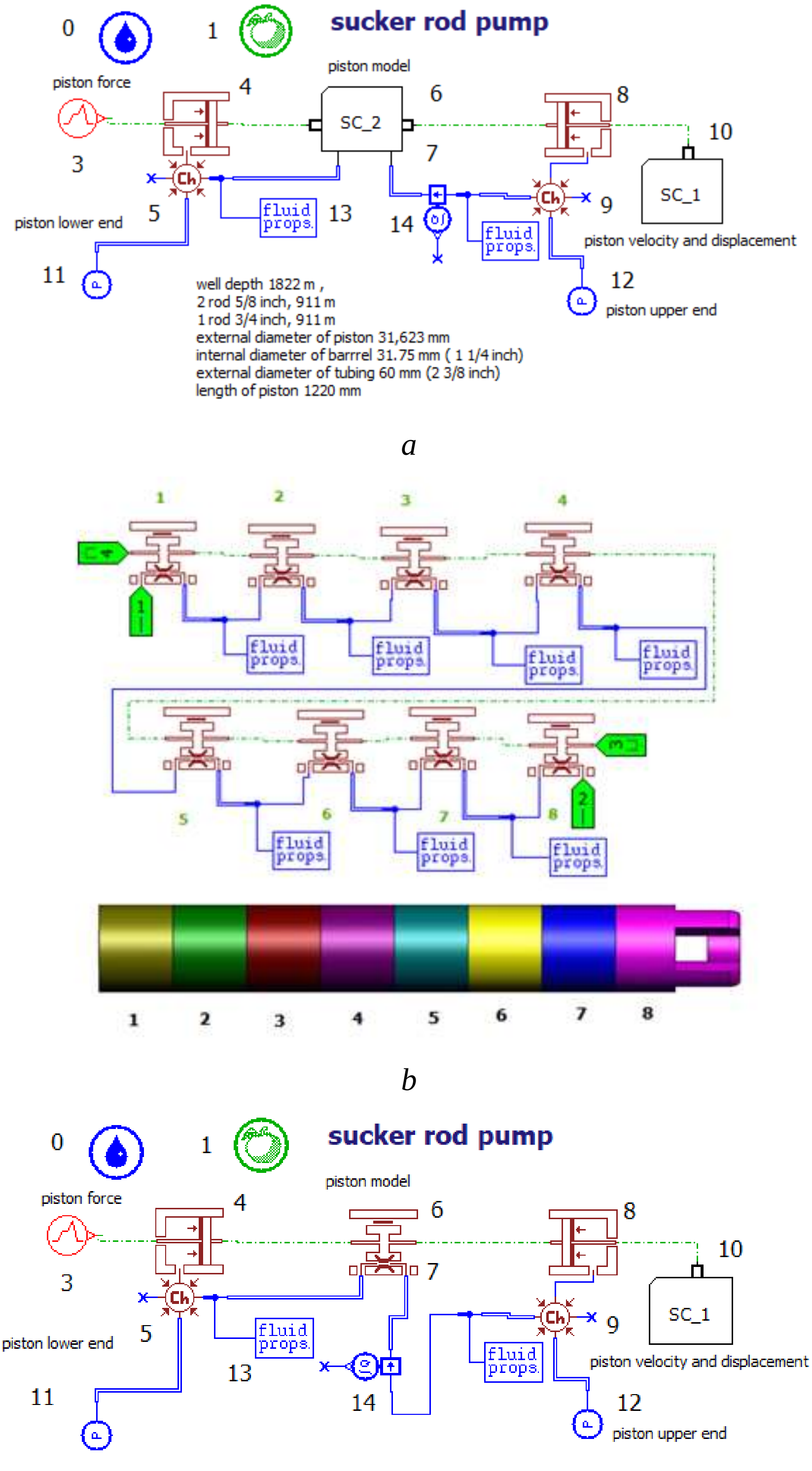
$$p_c = g \cdot (\rho - \rho_g) \cdot (H - h_d) \tag{18}$$

If the mixture has a gas-liquid ratio RGL, considering the above, it follows that the complex value of the compressibility coefficient β of the pumped fluid is:

$$\beta = \frac{[\beta_a \cdot i + \beta_t \cdot (1-i)] + \beta_g \cdot RGL \cdot \frac{p_a}{p_c}}{1 + RGL \cdot \frac{p_a}{p_c}} \tag{19}$$

ACHIEVEMENT OF THE MODEL

The model used (made in LMS Amesim program) is based on the construction of the joint between the cylinder and the piston, Figure 1, a. The following elements from the program library are used: 0. the extracted oil properties; 1. the gravitational acceleration, the piston movement is considered taking into account its weight; 3. piston force; 4. the bottom chamber of the pump; 5. the dead space at the bottom chamber of the pump; 6. the gap between cylinder and piston (see Figure 1, b); 7. link channels; 8. the upper chamber of the pump; 9. the dead space at the top of the pump; 10. the piston speed input model and the displacement of piston calculation; 11. the reservoir pressure (10 bar); 12. the well pressure (170 bar); 13. the characteristics of the liquid; 14. the transducer expressing the amount of volumic lost fluid between the piston and the cylinder. The joint of the piston-barrel was divided into 8 zones in order to highlight the pressure variation throughout the piston and the modification of the fluid characteristics according to pressure and temperature. The temperature at the pump was assumed constant and equal to 63.66 °C. For the liquid properties, a file, with the structure from Figure 2 was used. The values of the liquid density (11, 12), compressibility coefficient (modulus of elasticity) (14-19) and viscosity (9, 10, 13) are inserted into this file. The values are extrapolated by the program to cover the range of functional points. The piston is in motion and the forces acting on it are taken into account.



a

b

c

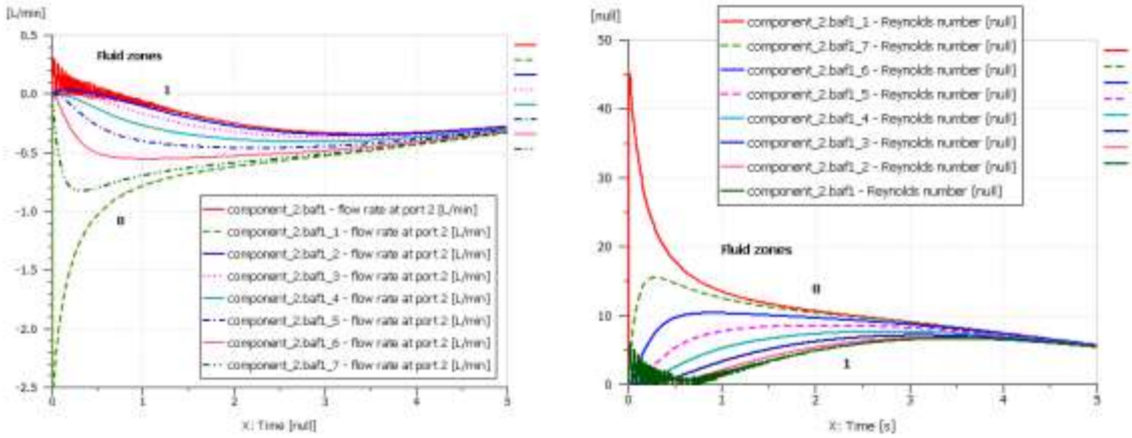
Figure 1. Model achievement in the dynamic case: a. LMS Amesim model; b. piston sub model; c. one zone piston –cylinder gap. Piston diameter 31.75 mm; piston – cylinder gap 0.127 mm; piston length 1220 mm.

#The construction of the file for the properties of #the liquid , # means comment # the construction mode, there are defined 3 cases #(1,2 or 3) 1 #Then the number of temperature values. This #must be greater than zero. #If it is one, it is assumed that density and bulk modulus do not vary with temperature. 3 #Now the ref. density in kg/m**3 and pressure in #bar gauge. 846.0 10 #Next the first temperature in Celsius. 10 #Then the number of pressure-bulk modulus pairs #N (greater than 0) followed by N pressure-bulk #modulus pairs. # The pressure values must form a #strictly monotonically increasing sequence. #The pressure is in bar gauge and the bulk modulus #also in bar. 4 #The first bulk modulus table. 20 15344.0 80 15507.0 140 15699.0 200 15831.0 #We repeat for the second temperature. ……………………………………. #Now the third and in this case last temperature. ……………………………………………. #The definition of bulk modulus and density is #now complete. Next we define the absolute #viscosity in mode 1.	#In mode 1 the absolute viscosity is defined from #tables of absolute viscosities. #Mode for viscosity. 1 #Then the number of temperature values. This must be greater than zero. # If it is one, it is assumed that absolute viscosity does not vary with temperature. 3 #The first temperature. 50 #Then the number of pressure gauge-absolute viscosity pairs M (greater than 0) followed by M # pressure gauge-absolute viscosity pairs. The pressure values must form a strictly # monotonically increasing sequence. The pressure is in bar gauge and the absolute viscosity in cP. 9 #The first absolute viscosity table. 10 26.1 20 26.38 40 26.96 60 27.55 80 28.15 100 28.75 120 29.37 140 29.99 170 30.95 #The second temperature. ………………………………………… #The third temperature. …………………………..

Figure 2. The structure of the file for liquid properties.

RESULTS

The use of the model allows to highlight the volume flow through the piston-barrel gap, Figure 3, a. The flows are represented from the zone 1 (lower piston end) to the zone 8 (at the top of the piston) Figure 1, b. It is represented the upward stroke. It is noted that initially the flow rates are different (the model takes into account the compressibility of the liquid and viscous friction forces), after which the flow values become equal to the end of the lifting stroke and the movement gains a stabilized aspect. The Reynolds number which is proportional with the velocity is Figured in 3, b. The fluid zones have different velocities. Upon the upward stroke, the fluid from zone 8 moves at the highest speed, which gradually decreases along the ascending stroke. The realized model involves defining the properties of the liquid: the viscosity, the modulus of elasticity and density in accordance with the values obtained by an experiment in relation to temperature and pressure, the file from Figure 2. During the displacement of the piston, the liquid (in the dynamic model) has variable characteristics, Figure 4, which modifies the flow conditions. The viscous friction forces are also included in the model. The liquid in other areas behaves similarly but with lower speeds. Under these conditions, the flow rates are different in the transient regime and have a stabilization trend that respects the continuity equation after a period of time. Due to these modifications, the only way to appreciate the effectiveness of the sealing system (the gap between piston and barrel) is to measure the volume of fluid flowing through the gap, Figure 5 (this information is given by the transducer 14 Figure 1, a). The lost liquid flow depends on eccentricity, Figure 5, a and the diametral gap Figure 5, b.



a b

Figure 3. Flow variation on each zone of the piston

If the diametral gap is higher, the aspect of the velocities has changed (expressed as Reynolds number variation), see the differences between the Figure 6 (diametral gap 0.508 mm) and Figure 3, b (diametral gap 0.127 mm).

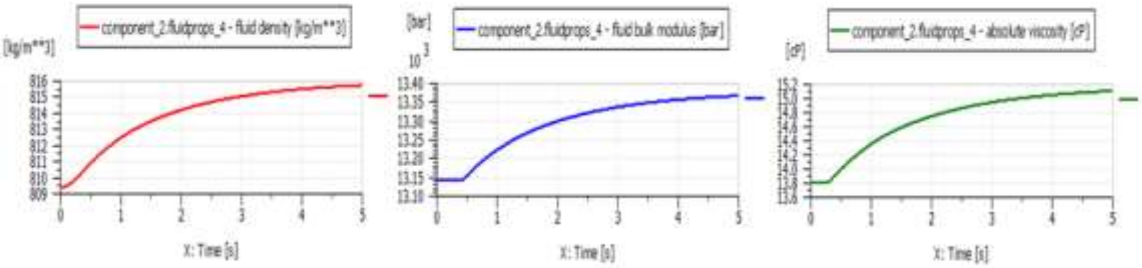
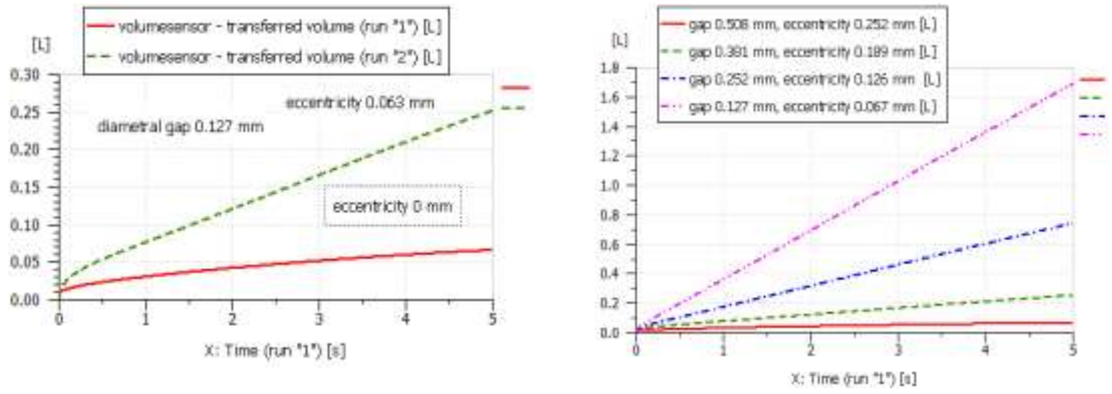


Figure 4. Variation of the liquid properties during the simulation – exemplified on the zone 5, see Figure 1,b.



a b

Figure 5. Variation of the volume of liquid lost: a. at differed eccentricity; b. at different gaps, eccentricity is ½ of the gap.

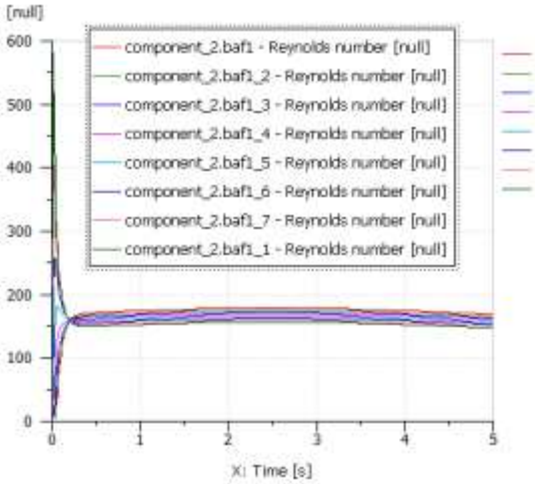
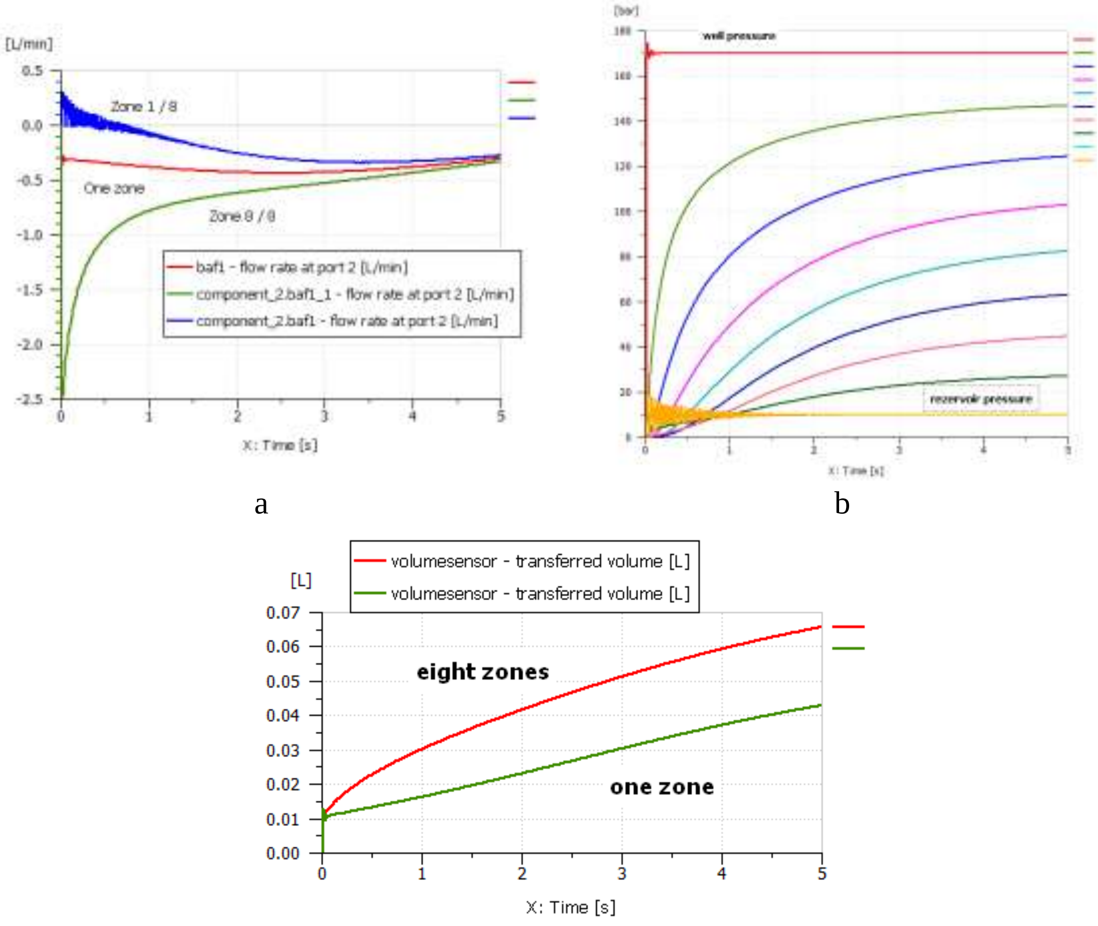


Figure 6. Reynolds number for a diametral gap of 0.5 mm.

CONCLUSIONS

Using the dependence of the temperature and pressure of liquid characteristics (Figure 2) and separating the space between the piston and the cylinder in several zones (Figure 1, a) highlights the difference between the loss rates, Figure 7, a. It is noted that in a model with 8 zones, during the transient regime we have different moving speeds, smaller as we move away from the upper end of the piston. The one-zone liquid model (Figure 1, c) shows an approximately constant velocity in the gap. The losses evidenced by an 8-zone model are greater 0.065 l compared to one zone model 0.042 l (54% extra). In the model with one zone we have at the ends of the gap a pressure difference from 170 to 10 bars. In the 8 zone model the pressure difference is time dependent and the pressure drop across the gap is better appreciated, Figure 7, c. In conclusion, a discretised model is better because it allows an appropriate selection of the average values used in the calculation.



a b

c

Figure 7. Differences between one zone and eight zones model: a. flow variation; b. distribution of the pressure; c. volume losses;

Nomenclature

$a_i, b_i -$ polynomial coefficients.

$a_1 = 4.5999514e - 9; b_1 = -1.77667e - 11; \quad c_1 = -2.753963e - 13; d_1 = 7.44996e - 14; \; e_1 = -3.43195e - 15; a_2 = 4.59956e - 9; b_2 = 2.37465e - 11; c_2 = 1.16335e - 12; d_2 = 2.46986e - 14; e_2 = -1.31588e - 15.$

$c -$ relative eccentricity (the distance between the center of the cylinder and the piston section / the difference between the cylinder and the piston section); for coaxial positioning, c = 0, and for the complete plunging of the piston on the cylinder,

$z -$ coordinate across the lubricant film, m

$H -$ the depth of the well, m

$K -$ bulk modulus of elasticity, Pa

$RGL -$ gas liquid ratio, $RGL = \frac{V_g}{V_l}$

$V -$ piston velocity, m/s

$V_0 -$ initial volume of the elementary liquid, m^3

$V_1 -$ final volume, m^3

$V_g -$ gas volume, m^3

$V_l -$ liquid volume, m^3

$c = 1.$

$c_e -$ experimental coefficient, Pa^{-1}

$dp -$ differential change in pressure on the liquid, *Pa*

$dV -$ differential change in volume of the liquid, m^3

$h_d -$ dynamic depth, m

$i -$ water fraction, -

$p_0 -$ initial pressure, *Pa*

$p_1 -$ final pressure, *Pa*

$p_R -$reference fluid pressure according to Roelands [14] $p_R = 1.98 \cdot 10^8$, Pa

$p_{R2} -$reference fluid pressure according to Dowson [7] $p_{R2} = 5.9 \cdot 10^8$, Pa

$p_a -$ atmospheric pressure, *Pa*

$p_c -$ calculus pressure at the depth pump level, Pa

$t -$ liquid temperature, ℃

$\alpha -$ pressure–viscosity coefficient, Pa^{-1}, $\alpha = 2.18 \cdot 10^{-8} Pa^{-1}$

$\beta -$ compressibility coefficient of pumped liquid, Pa^{-1}

$\kappa -$ adiabatic exponent, -

$\mu -$ dynamic viscosity of petroleum at pressure p_c, $Pa \cdot s$

$\beta_a -$ compressibility coefficient of water, Pa^{-1}

$\beta_g -$ compressibility coefficient of gas, Pa^{-1}

$\beta_t -$ compressibility coefficient of petroleum, Pa^{-1}

$\rho -$ liquid density, $\frac{kg}{m^3}$

$\mu_0 -$ dynamic viscosity of petroleum at atmospheric pressure, $Pa \cdot s$

$\mu_R -$ reference dynamic viscosity $\mu_R = 6.315 \cdot 10^{-5}$, $Pa \cdot s$

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