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RESEARCH THROUGH ANALYSIS AND SIMULATION OF THE 70 MPa HYDROSTATIC PUMP WITH PISTONS

BY

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Abstract. The paper relates to a analysis and simulation of the optimized version for the 70 MPa hydrostatic pump with pistons, for modular clamping devices, intended for machining or mounting processes.

Keywords: analysis; simulation; hydrostatic.

1. Introduction

The research of 70 MPa hydraulic power unit (Chiriță *et al.*, 2012, p. 4) as components of modular technological clamping devices has been challenging. The research through analysis and simulation of the optimized version for the 70 MPa hydrostatic pump with pistons (Chiriță *et al.*, 2014, p. 5) was performed with Amesim (Amesim Platform, 2013; Amesim Libraries, 2014; Downey, 2014; Șerbănescu *et al.*, 2009).

The hydraulic scheme of the optimized version of the 70 MPa power unit with hydrostatic pump with pistons, technical features of the system, physical model and simplification assumptions, the models of the components (hydrostatic pumps, spring loaded hydraulic check valve with saturation, hydraulic pressure-relief valve, hydraulic control valve and dynamic rotary

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mechanical node) there are presented in a previous article (Hanganu and Dumitraş, 2017).

The main topics of the project are: the verification through analysis and simulation of the hydraulic scheme and proposals for energy optimization with research influence of nominal diameter of pipes (Dn 4 to 14 mm) for total efficiency of the system (Alboteanu, 2017; Ansorge and Sonar, 2009; Backé, 2000; Cakaj, 2010; Chung, 2004; Huhtala *et al*., 1995; Kunes, 2012; Matache *et al*., 2013; Skjong, 2014; Theurillat, 2011).

2. Fluid Parameters

Technical parameters of the fluid is on the Table 1:

Table 1
Technical Parameters of the Fluid

Technical parameters	Unit	Value
density	kg/m^3	850
bulk modulus	MPa	1.700
slope of bulk modulus [MPa] in function of pressure [MPa] (in percentage)	-	0
absolute viscosity	cP	51
absolute viscosity of air/gas	cP	0.02
saturation pressure (for dissolved air/gas)	MPa	100
air/gas content	%	0.1
temperature	degC	40
temperature T_1	degC	40
kinematic viscosity at (p_{atm},T_1)	cSt	56.47
temperature T_2	degC	100
kinematic viscosity at (p_{atm},T_2)	cSt	9.71
coefficient for temperature viscosity characteristic	-	0.8
polytropic index for air/gas/vapor content	-	1.4
high saturated vapor pressure	MPa	-0.05
low saturated vapor pressure	MPa	-0.06
absolute viscosity of vapor	cP	0.02
effective molecular mass of vapor	-	200
air/gas density at atmospheric pressure 0 degC	kg/m^3	1.2
saturation pressure (for dissolved air/gas)	MPa	0

The characteristic of bulk modulus (10%) in function of the pressure is presented at Fig. 1.

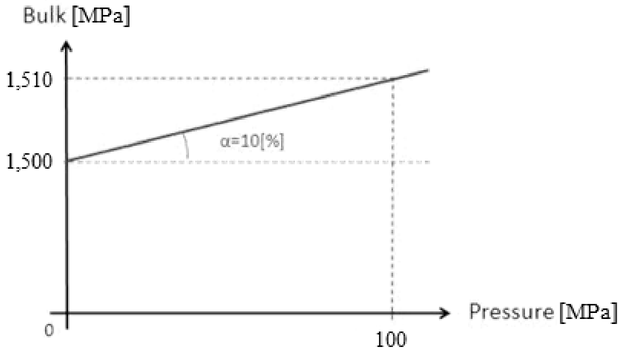

Fig. 1 – Bulk modulus [MPa] in function of pressure [MPa] (in percentage).

The definition of the “air/gas content” is as follows: suppose a sample of hydraulic fluid is taken. Separate the air/gas including the dissolved air/gas from the liquid and store both of them at atmospheric pressure and 273 [degK].

$$\textit{air/gas content} = \frac{V_{air/gas}}{V_{air/gas} + V_{liq}} \tag{1}$$

where: V_{liq} – volume of the liquid at atmospheric pressure and 273 [degC], [m^3]; $V_{air/gas}$ – volume of the air or gas at atmospheric pressure and 273 [degC], [m^3].

Polytrophic index for air/gas/vapor content. The equation of state for a polytrophic fluid is:

$$p = K\rho^{(1+n^{-1})} \tag{2}$$

where: K is a constant and n is a quantity called the polytrophic index.

The kinematic viscosity [m^2/s] is equal to the absolute viscosity [Pa·s] divided by the density [kg/m^3].

Speed of sound c, [m/s]:

$$c = \sqrt{\frac{B}{\rho}} \tag{3}$$

where: B – bulk modulus, [Pa]; ρ – density, [kg/m^3].

Void content:

$$void\ content = \frac{V_{air,gas,vapor}}{V_{air,gas,vapor} + V_{liq}} \tag{4}$$

where $V_{air,gas,vapor}$: volume of air + gas + vapor, [m^3].

3. Simulation Model

Based on above physical model and mathematical equations was built a simulation model. Only standard simulation models available in the library elements of AMESim software were used.

The simulation analysis were done for the maximum pressure in the system. Pressure relief valves were set to: 20 MPa (LP) and 70 MPa (HP).

Nominal speed of electric motor was 1415 min^{-1}. In all the figures the following description is used to for different diameters *d* of the hydraulic pipes: run 1 (*d* = 6 mm), run 2 (*d* = 8 mm), run 3 (*d* = 10 mm), run 4 (*d* = 12 mm) and run 5 (*d* = 14 mm).

Analysis of the efficiency of the system has been carried out for the maximum pressure in the system.

4. Research Plan

During the simulation research were carried out the following simulations:

– verified the proper operation of the simulation model of hydraulic system;

– analyzed the hydraulic pipe diameter for the time of reach the maximum pressure in the system;

– the influence of the pressure set of relief valve on the efficiency of the system;

– an influence of the hydraulic pipe length on the HP chamber for the time to reach maximum working pressure;

– working pressure in function of the angular velocity of the electric motor;

– flow ratio through the relief valve according to the angular velocity of the motor.

5. Results

The results of the simulations can be seen in Figs. 2-13. For Fig. 8, it was considered Young's modulus for material $2.06 \cdot 10^5$ [MPa].

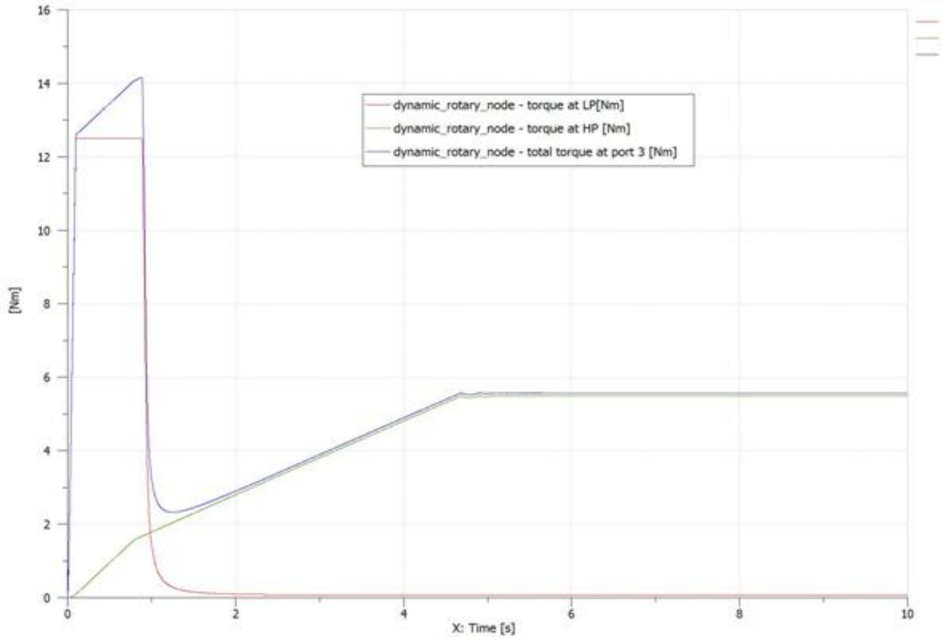

Fig. 2 – Torque on the motor shaft.

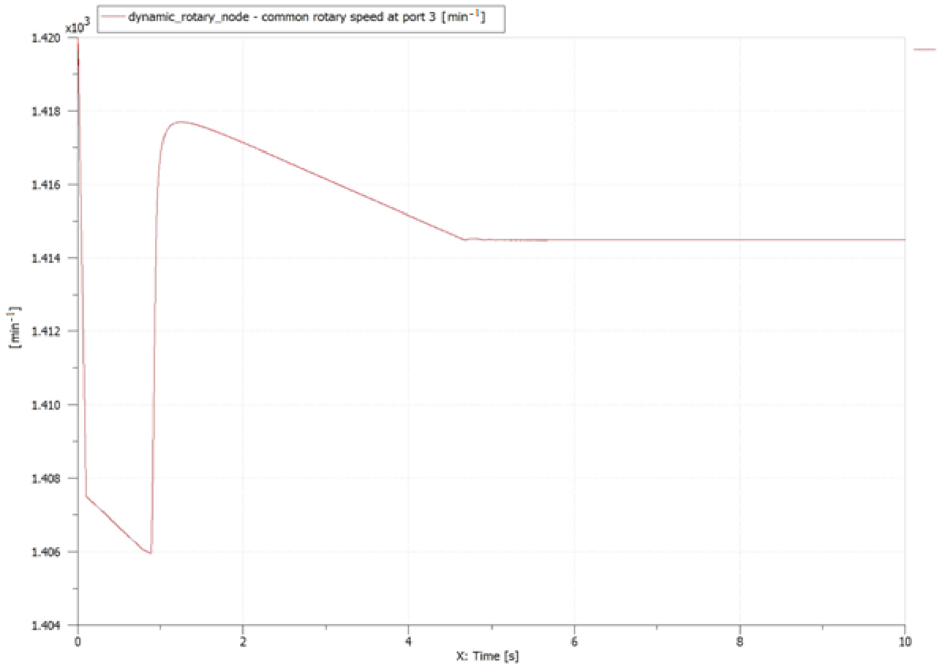

Fig. 3 – Angular velocity of electric motor.

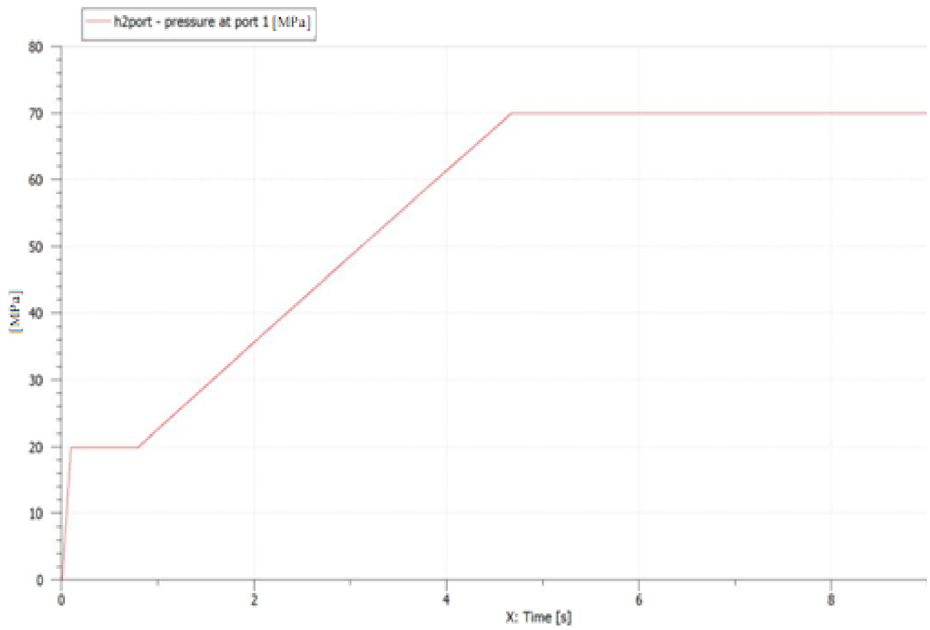

Fig. 4 – Function of pressure in hydraulic system.

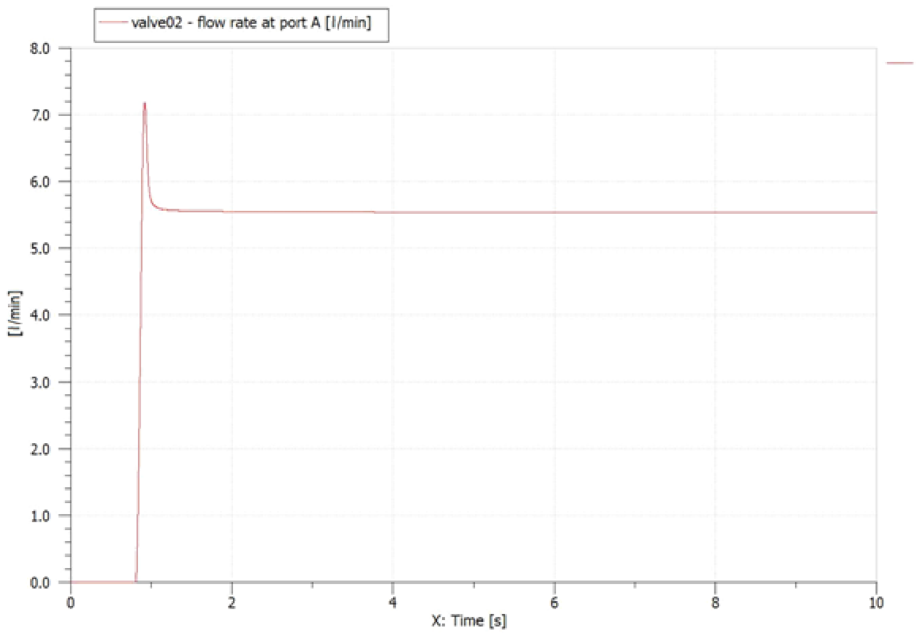

Fig. 5 – Function of flow through the relief valve of low-pressure (LP) pump.

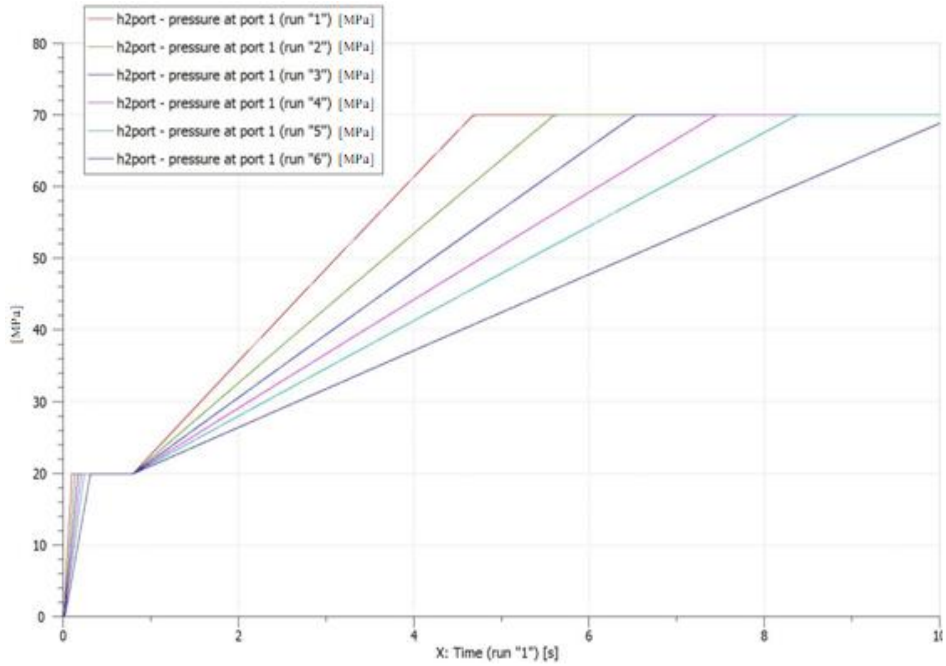

Fig. 6 – Function of the time of reach the maximum working pressure to the hydraulic pipe length in the HP chamber (length 1; 1.5; 2; 2.5; 3; 3.5 [m]).

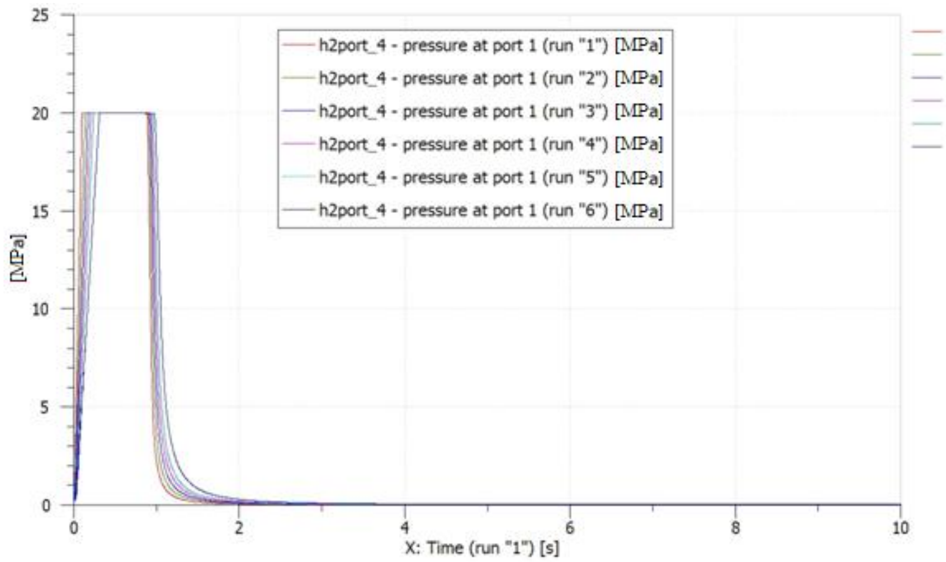

Fig. 7 – Pressure on the LP chamber.

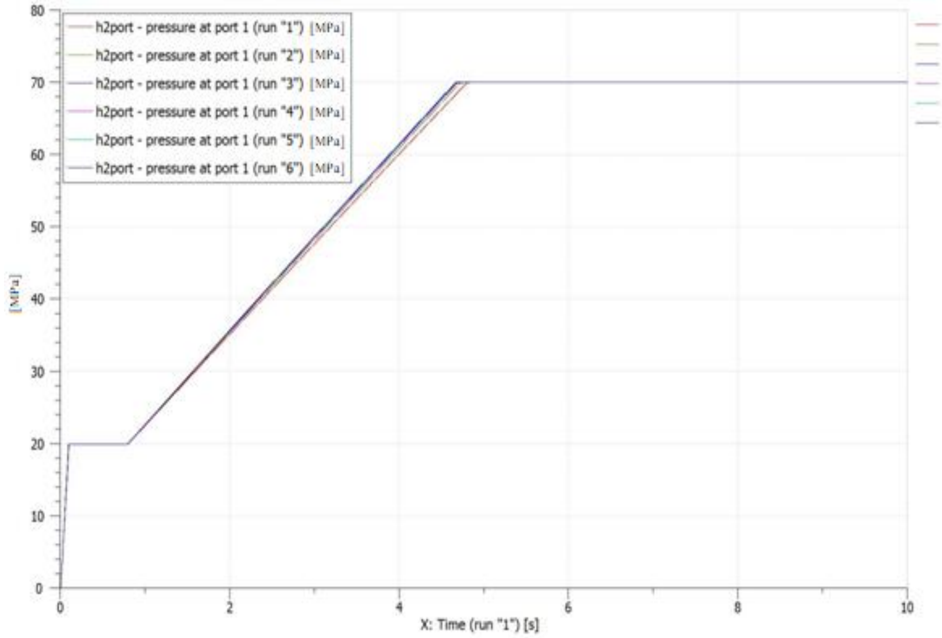

Fig. 8 – Pressure in the function of wall thickness of the hydraulic pipe - pipe stiffness (diameter of the pipe from 4 to 14 [mm]).

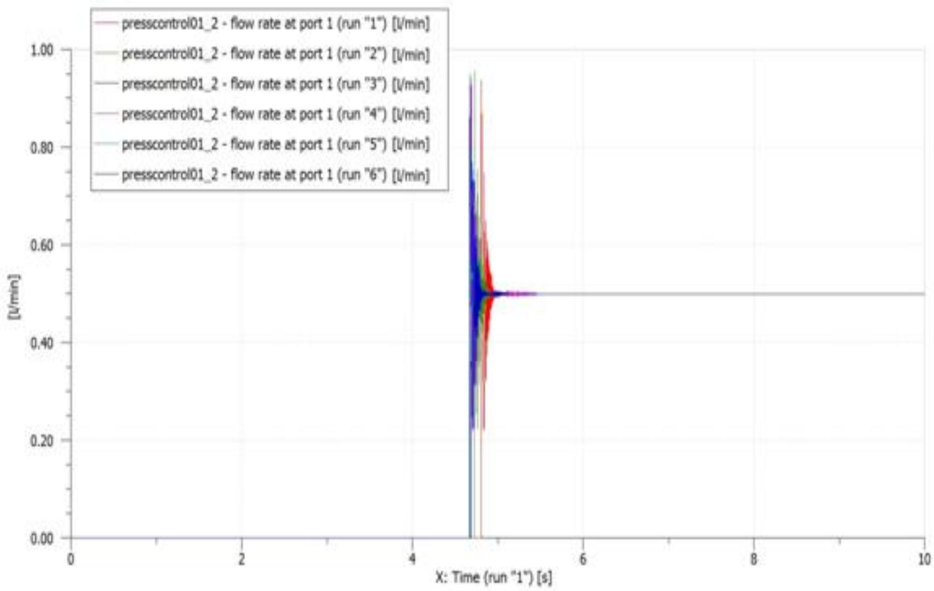

Fig. 9 – The flow ratio of HP pump according to the stiffness of the pipe.

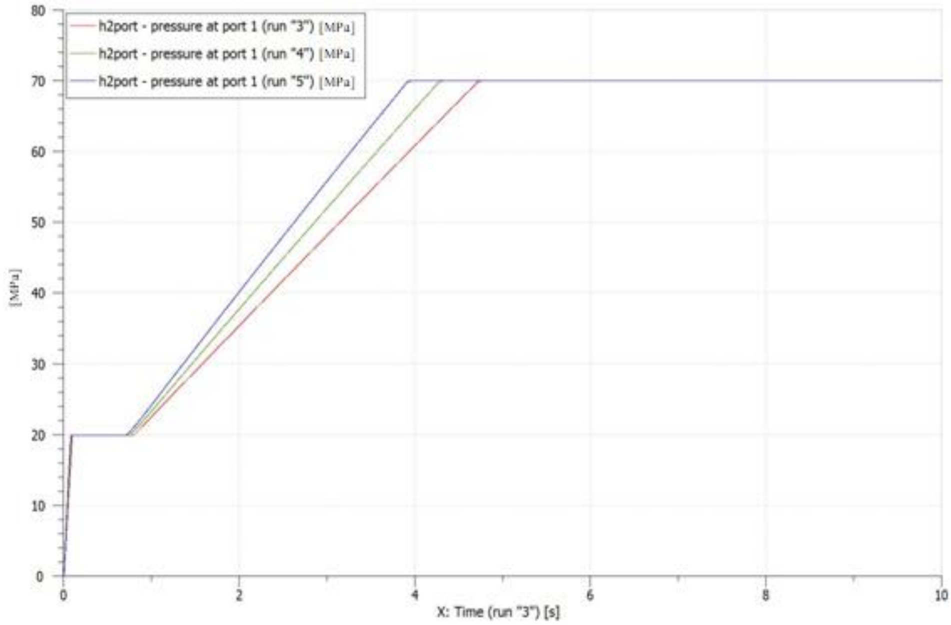

Fig. 10 – Working pressure in function of the angular velocity of the electric motor (1.415; 1.600; 1.800 [min^{-1}]).

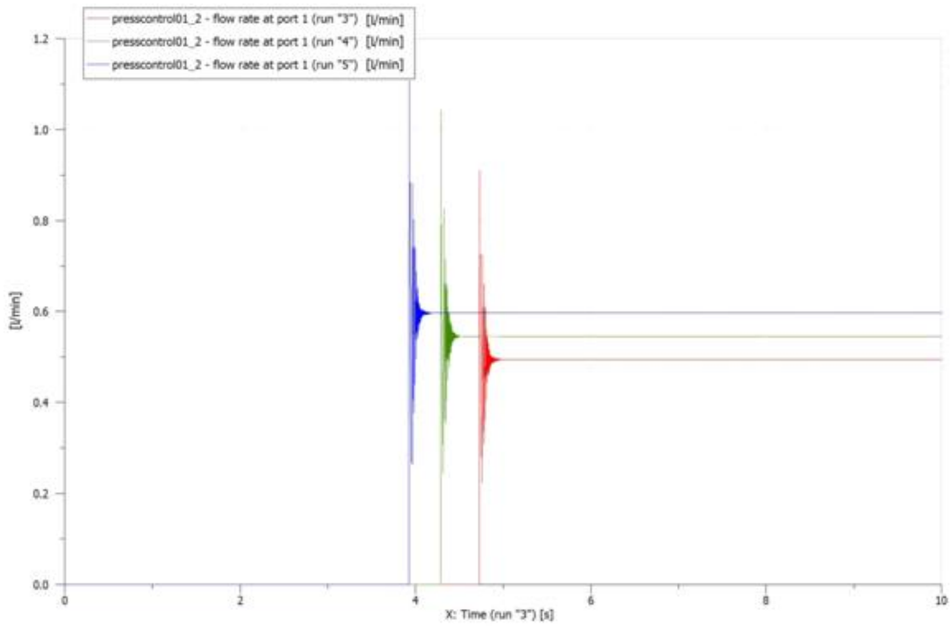

Fig. 11 – Flow ratio through the relief valve according to the angular velocity of the motor.

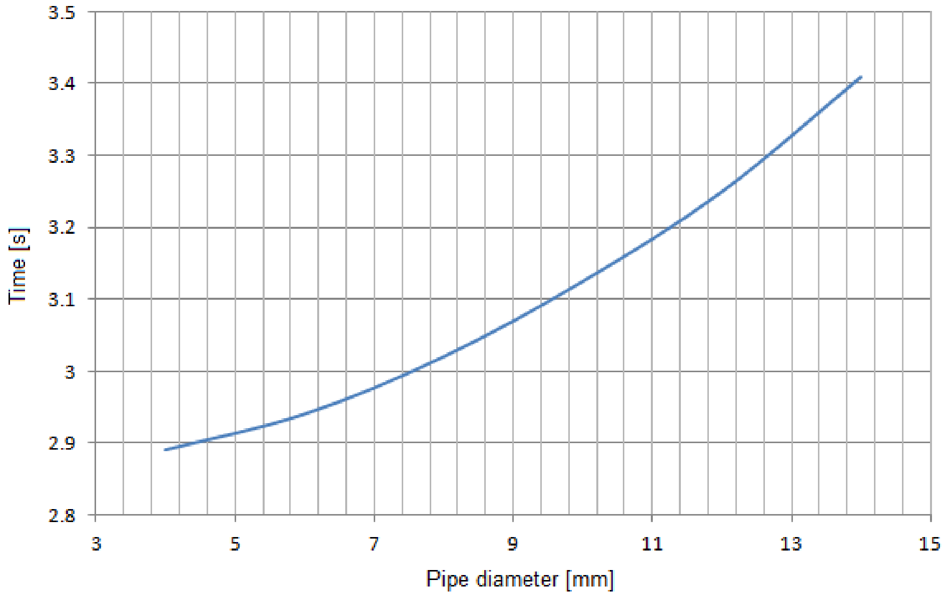

Fig. 12 – Function of the opening of the relief valve
according to the pipe diameter.

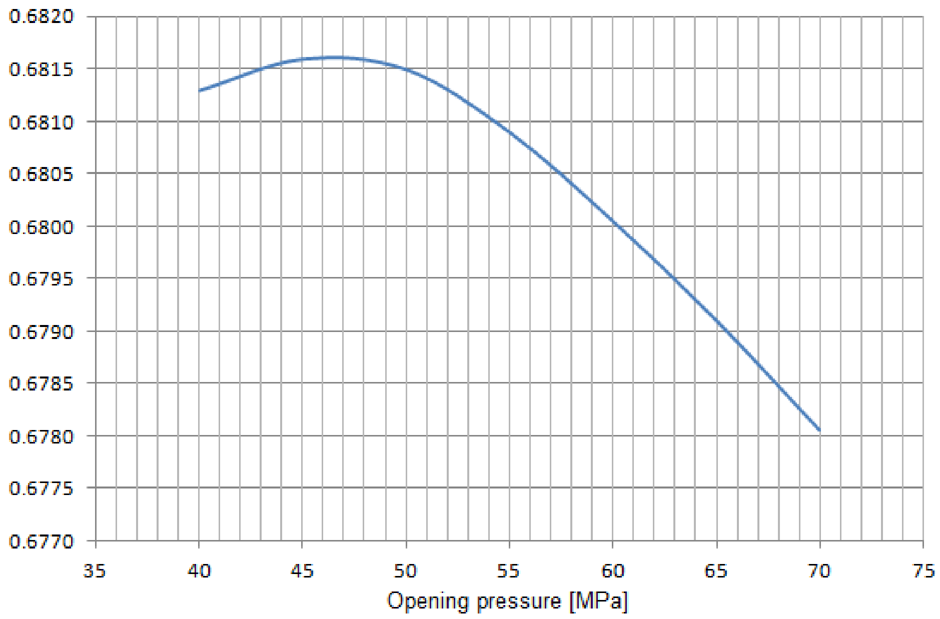

Fig. 13 – Efficiency of the system according to opening
pressure of the relief valve.

6. Conclusions

When using the Amesim platform, we faced the difficulty of converting units of measurement. The research shows that the simulation model is working properly. Due to the lack of the actual characteristics of hydraulic components (pumps, valves, pressure relief valves, etc.), presented results of simulations are only approximate (based on standard components selected from the Amesim Libraries). In order to achieve better results it is needed to put to the simulation model real characteristic of the elements (*e.g*. obtained from experimental tests).

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CERCETARE PRIN ANALIZĂ ȘI SIMULARE A UNEI POMPE HIDROSTATICE CU PISTOANE DE 70 MPa

(Rezumat)

Articolul se referă la o analiză și simulare a versiunii optimizate pentru pompa hidrostatică cu pistoane, de 70 MPa, pentru dispozitive de prindere modulare, destinate proceselor de prelucrare sau de montare.