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MODELING FOR ANALYSIS AND SIMULATION OF 70 MPa HYDROSTATIC PUMP WITH PISTONS

BY

ADRIAN-CONSTANTIN HANGANU[*] and CĂTĂLIN GABRIEL DUMITRAȘ

"Gheorghe Asachi" Technical University of Iaşi,
Faculty of Machine Manufacturing and Industrial Management,

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Abstract. The paper relates modeling for 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: model; 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).

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; Matache *et al*., 2013).

[*]Corresponding author; *e-mail*: adrian.hanganu@tuiasi.ro

2. The Optimized Version of the 70 MPa Power Unit with Hydrostatic Pump with Pistons

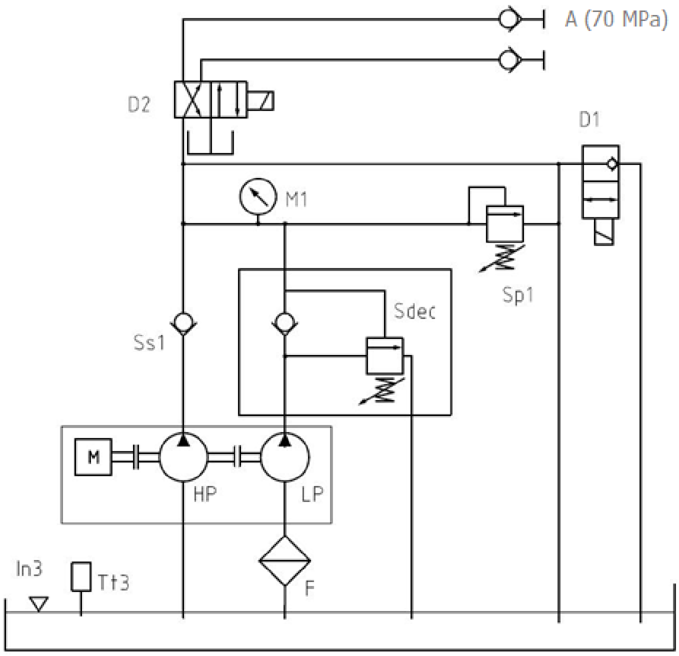

Fig. 1 – The hydraulic scheme of the optimized version of the 70 MPa power unit with hydrostatic pump with pistons.
M – AC motor; HP – high pressure hydraulic pump (70 MPa); LP – low pressure hydraulic pump (20 MPa); Ss1 – check valve; Sdec, Sp1 – pressure relief valves; D1, D2 – directional control valves; A – hydraulic rapid coupler; M1 – hydraulic pressure gauge; F – filter; In3, Tt3 – sensors.

Technical features of the system is on the Table 1:

Table 1
Technical Features of the System

Technical features	Unit	Value
maximum installed power	kW	1.5
number of revolutions of the AC motor	min^{-1}	1.415
maximum LP (20 MPa) flow	l/min	5.6
minimum HP (70 MPa) flow	l/min	0.6
number of HP (70 MPa) pistons	no	3
diameter of HP (70 MPa) pistons	mm	8
power stroke of HP (70 MPa) pistons	mm	3

3. Physical Model and Simplification Assumptions

The idea of the system is presented on Fig. 2.

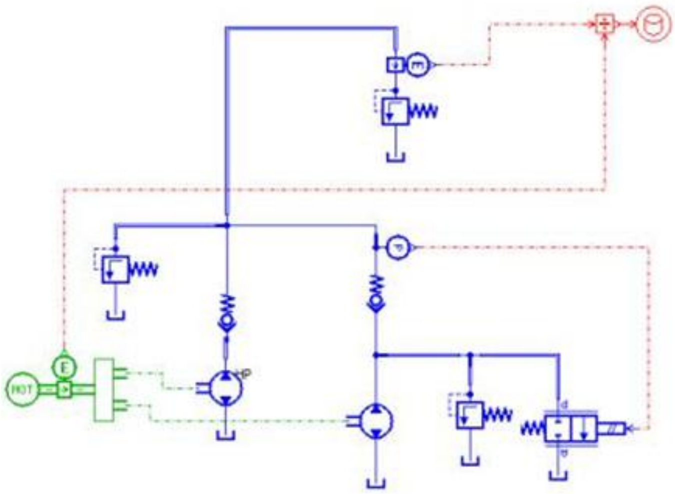

Fig. 2 – Schema of the pumps layout.

The low pressure pump (LP) is a gear pump with a pressure control valve (PCV) on the delivery in order to keep a constant pressure of 20 MPa. The high pressure pump (HP) in this model is fixed displacement hydrostatic pump. LP pump and HP pump are in series. Simplification assumptions:

– the absolute viscosity is assumed to be constant;

– the air/gas release phenomenon is considered and starts when the pressure drops below the "saturation pressure";

– the bulk modulus at pressures above the "saturation pressure" is assumed to be constant;

– below the "saturation pressure", the bulk modulus is also assumed to be constant and its value is equal to the bulk modulus above saturation pressure divided by 100;

– the liquid density is assumed to be independent of the temperature;

– the influence of leakage in the system has been omitted;

– the flow through the resistant elements is treated as a turbulent flow;

– the pressure in the system is treated as overpressure in reference to pressure in outlet line;

– stability of physical properties of the liquid: viscosity, density, compressibility and temperature have been assumed;

– pressure fluctuation of hydrostatic pump and motor has been omitted;

– the existence of air/gas bubbles at atmospheric pressure was assumed.

4. Model of the Hydrostatic Pumps

The model of fixed displacement hydrostatic pumps (high and low pressure) is presented on Fig. 3 (Kunes, 2012; Skjong, 2014; Theurillat, 2011).

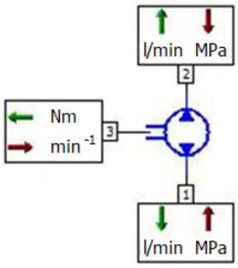

Fig. 3 – Model of fixed displacement hydrostatic pumps.

The theoretical flow rate of a hydrostatic machine, Q, [$l\cdot min^{-1}$], either pump or motor, is defined as:

$$Q=V\omega \tag{1}$$

where: V is the displaced volume, [l]; ω – the shaft speed, [min^{-1}].

The theoretical torque, either for a pump or for a motor, is defined as:

$$T=10^3 V\Delta p \tag{2}$$

where: T is the theoretical torque, [Nm]; Δp – the differential pressure at the hydraulic ports of the hydrostatic machine, [MPa].

The real quantities are then derived by the theoretical ones through efficiencies.

In a real displacement machine several energy losses occur, mainly divided in two categories:

– volumetric losses, determined by internal and external flow leakages through different gaps or clearances, flow losses due to the fluid compressibility;

– hydraulic mechanical losses, representing torque losses due to dry and viscous friction between moving elements.

Due to volumetric losses, the real flow rate delivered by a displacement pump is consequently less than the theoretical value generated by an ideal pump (with the same shaft speed).

Due to the hydraulic mechanical losses, the torque required by a displacement pump with a certain differential pressure is greater than the ideal value.

For the pump working mode, the real flow rate and torque are evaluated as follows:

$$Q_r = Q\eta_{vol} \tag{3}$$

where: Q_r is the volumetric flow rate, [l]: η_{vol} – the volumetric efficiency, [-];

$$T_r = T\eta_{hm} \tag{4}$$

where: T_r is the real torque, [Nm]; η_{hm} – the hydraulical-mechanical efficiency, [-].

5. Model of the Spring Loaded Hydraulic Check Valve with Saturation

The model of the spring loaded hydraulic check valve with saturation is presented on Fig. 4.

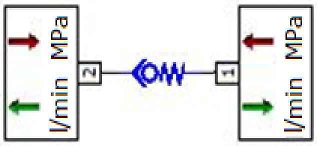

Fig. 4 – Model of the spring loaded hydraulic check valve with saturation.

The check valve is modeled using basically two equations: a force equality and the flow equation. The force equation calculates the fractional check valve spool position which is used in the flow equation for calculating the orifice opening. This spool position is between 0 and 1.

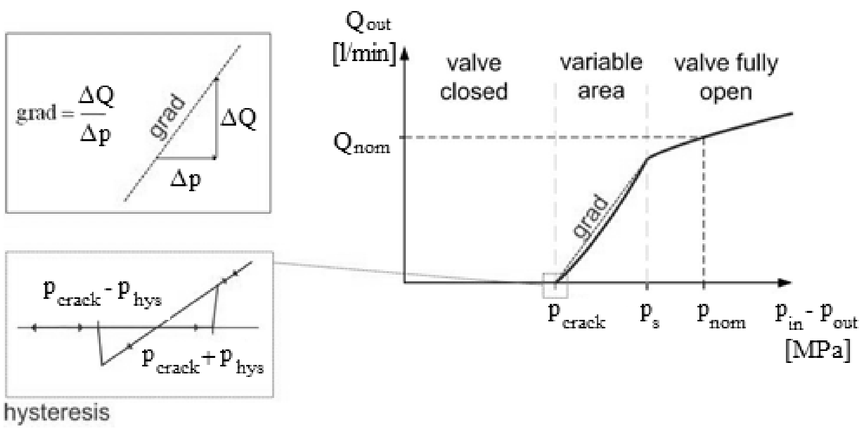

Fig. 5 – Characteristic of a spring loaded check valve with saturation.

There are three parts in the valve behavior:

$$\Delta p = p_{in} - p_{out} \tag{5}$$

where: p_{in} is the input pressure at the valve, [MPa]; p_{out} – the output pressure from the valve, [MPa].

– the first part: Δp in the interval [0, p_{crack}]; the valve is close; the fractional spool position is always 0;

– the second part: for Δp in the interval [p_{crack}, p_s]; the valve is close at p_{crack} and just fully open at p_s.

The fractional spool position x_v must be 0 when $\Delta p = p_{crack}$ and must be 1 when the valve is just fully open: $\Delta p = p_s$ (Fig. 5). The spool position is linear with the pressure between 0 and 1, so the fractional spool position is expressed as:

$$x_v = \frac{(p_{in} - p_{out}) - p_{crack}}{k_{spring}} \tag{6}$$

where:

$$k_{spring} = p_s - p_{crack} \tag{7}$$

The flow is calculated using:

$$A = x_v a_{max} \tag{8}$$

where a_{max} is the area corresponding to the given flow pressure drop pair (Q_{nom} and p_{nom}).

The value of a_{max} is calculated using the utility function *eqarea* (Amesim Platform, 2013).

The flow Q calculation is made by the utility function *orif3f* (Amesim Platform, 2013).

This utility is a double precision which computes an estimate of the flow rate through a hydraulic orifice and the corresponding flow coefficient and flow number from the following inputs:

– port pressures p_1 and p_2;

– cross sectional area A;

– hydraulic diameter d;

– maximum flow coefficient C_{qm};

– flow number at which transfer from laminar to turbulent characteristics occurs λ_{crit}.

Referring to the orifice ports as 1 and 2, the pressure drop is $\Delta p = p_2 - p_1$. The density ρ and dynamic viscosity η are evaluated at mean pressure $(p_1+p_2)/2$. The flow number is:

$$\lambda = \frac{d}{\eta}\sqrt{\frac{2|\Delta p|}{\rho}} \tag{9}$$

The flow coefficient used varies with the flow number λ according to the formula:

$$C_q = C_{qm}\ tanh(\frac{2\lambda}{\lambda_{crit}}) \tag{10}$$

This expression means that C_q approaches C_{qm} asymptotically as Δp increase. Usually, C_{qm} = 0.7.

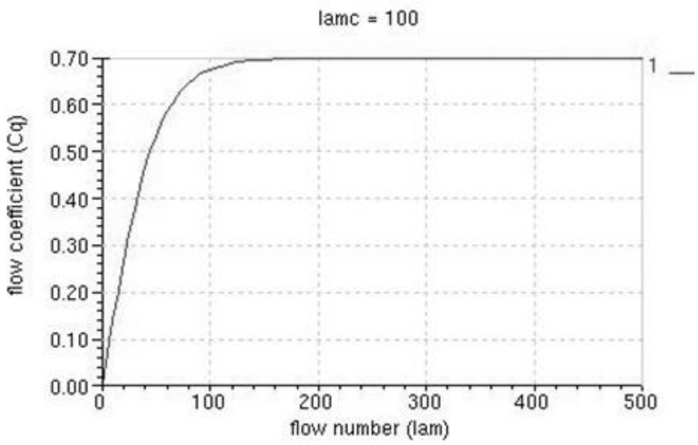

Fig. 6 – The flow coefficient function.

The mean fluid velocity is:

$$flv = C_q\sqrt{\frac{2|\Delta p|}{\rho}} \tag{11}$$

The flow rate is then:

$$Q = C_q A\frac{\rho}{\rho(0)}\sqrt{\frac{2|\Delta p|}{\rho}}sign(\Delta p) \tag{12}$$

The computed orifice flow rate is assumed to apply at the mean of p_1 and p_2, and is corrected to 0 pressure gauge.

Since no flow forces are incorporated in the model and the opening of the valve is proportional to the spool position the resulting flow rate/pressure drop is slightly curved. This curvature is very small for "normal" parameters settings as can be seen in the diagram.

The third part: for $d_p > p_s$: the valve is open. The flow is calculated using $A = a_{max}$ where a_{max} is the area corresponding to the given flow pressure drop pair (q_{nom} and p_{nom}).

Calculation of p_s is the intersection of a curve for a_{max} as a cross sectional area, and the line corresponding to the parameter "grad".

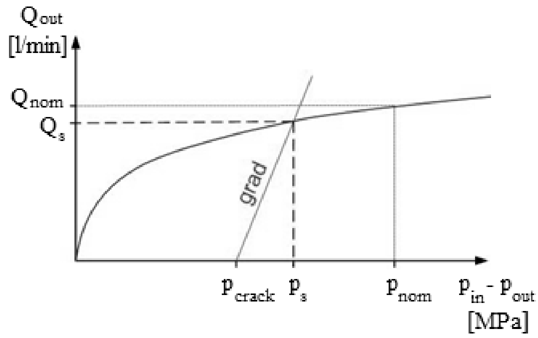

Fig. 7 – The p_s parameter setting.

The equation that represents the intersection of the two curves is:

$$Q_s = (\Delta p - p_{crack})grad \tag{13}$$

$$Q_s = C_q A \frac{\rho}{\rho(0)} \sqrt{\frac{2 \cdot 10^5 |\Delta p - p_{crack}|}{\rho}} \tag{14}$$

where $\Delta p = p_{in} - p_{out}$, [MPa]

6. Model of the Hydraulic Pressure-Relief Valve

The role of the relief valve is to limit the upstream pressure within a hydraulic circuit and thus protect hydraulic components from over pressure. This component is also known as pressure limiting valve, maximum-pressure valve or safety valve.

The valve is initially closed. When the pressure drop across the valve exceeds the relief valve cracking pressure (typically a spring force), the valve opens and let the fluid flow across so that the pressure drop gets regulated to the cracking pressure.

The flow rate/pressure drop characteristic is linear during the valve regulation. A functional hysteresis can be specified to the model in order to take into account dry friction effects. The valve dynamics can be set to static (for real-time purposes), first-order or second-order lag.

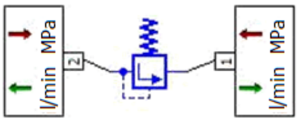

Fig. 8 – Model of hydraulic pressure-relief valve.

Relief valve cracking pressure: pressure at which the valve starts opening. This parameter is compared to a pressure difference and will not be converted when units are changed from absolute to relative pressures and vice versa.

Relief valve flow rate pressure gradient: slope of linear characteristic flow rate/pressure drop.

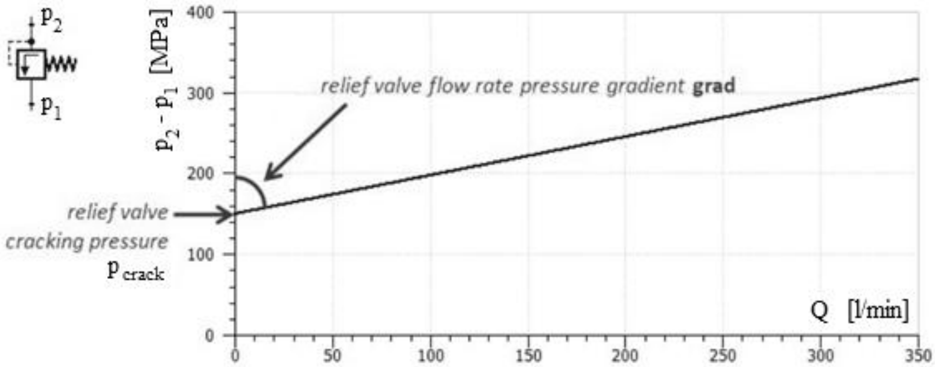

Fig. 9 – Characteristic of flow rate of relief valve.

The net pressure opening the valve is:

$$\Delta p = (p_2 - p_1) - p_{\text{crack}} \tag{15}$$

where: p_{crack} is the equivalent pressure of the spring preload, [MPa].

7. Model of Hydraulic Control Valve

Hydraulic control valve is a simple submodel of a 2 port 2 position proportional valve with pressure input at each hydraulic port and flow rate computed and output at both those ports (Fig. 10).

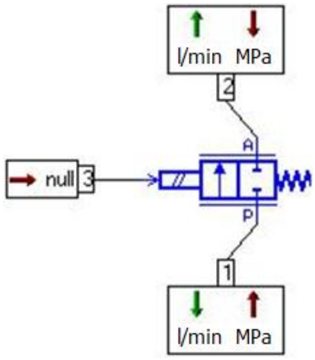

Fig. 10 – Model of hydraulic control valve (2 position 2 port).

To define the flow characteristic, it have to supply:

– either a value of flow rate at maximum opening and the corresponding pressure drop;

– or directly the orifice cross sectional area and maximum flow coefficient.

The flow rate through the valve is determined using the orifice flow rate with an area defined by:

$$\mathrm{A} = xA_{\max} \tag{16}$$

8. Model of Dynamic Rotary Mechanical Node

Hydraulic control valve is a simple submodel of a 2 port 2 position proportional valve with pressure input at each hydraulic port and flow rate computed and output at both those ports (Fig. 10).

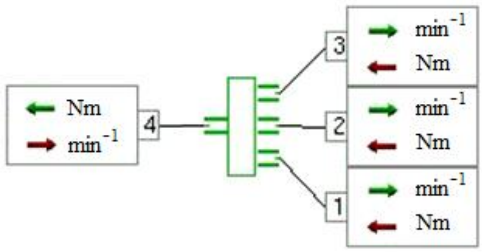

Fig. 11 – Model of a dynamic rotary mechanical node.

The rotary in [min^{-1}] which is an input on the left port is copied without modification to all the right hand ports. Torques in [Nm] are input to all right hand ports and are summed to give a total torque output on the left port.

9. Conclusions

The major goal of this thesis work was to develop a model for analysis and simulation of the optimized version for the 70 MPa hydrostatic pump with pistons. Within this article the structure of the hydraulic scheme on functional and component level was proposed. In order to modernize the hydraulic scheme it was necessary to design in detail the electro-hydraulic section. To be able to control the power unit, it was necessary to derive mathematical models of individual components of the control system. These are expressed in form of the block diagrams and the function transfer of the regulated system. We tried to keep the mathematical expressions rather simple by making some smaller assumptions. Future work should focus on validating the models for analysis and simulation.

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MODELAREA PENTRU ANALIZA ȘI SIMULAREA UNEI POMPE HIDROSTATICE CU PISTOANE DE 70 MPa

(Rezumat)

Lucrarea prezintă modelarea propusă pentru analiza şi simularea versiunii optimizate a pompei hidrostatice cu pistoane de 70 MPa, pentru dispozitive de prindere modulare, destinate proceselor de prelucrare sau de montare.