DYNAMIC PROCESSES MODELING IN MEASURING OF THE PEAK PRESSURE OF HYDROSYSTEMS OF TRACTORS USED IN AGRICULTURE

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ABSTRACT

This paper presents an analysis of the work performance of a tractor's hydraulic drive system, where a linear motor of reciprocating is used. A non-linear mathematical model describing the work of a given hydraulic system working in dynamic regime has been developed. Some numerical results for the transitional dynamic processes occurring in the model hydraulic drive system have also been presented. It is indicated, that using the developed mathematical model can help for the successful investigation of the impact of each of the hydraulic system's elements on the peak pressure at the end of the piston's work moves.

KEYWORDS: hydraulic system modeling, peak pressure

1. INTRODUCTION

One of the main problems related to the investigation of the hydraulic systems of recent tractors, used in agriculture, is the determination of their peak pressures. For this aim, in the test standards according to code 2 OECD (OECD STANDARD CODE) [1, 3] the existing requirements for the experimental achieving of the peak pressure values, respectively using of a double-acting cylinder whose parameters are: diameter of the cylinder's piston *D*=80 mm; diameter of the piston's rod *D*=30 mm and piston's move *l*=200mm; length of the connecting lines (hydraulic hose-pipes); hydraulic ports ensuring the connection between the tractor and hydro-cylinder *L*=2500 mm and the nominal inside bore/orifice *dpl*=10 mm.

The magnitude of the peak pressure's increase at the end of the piston's move provides information about the dynamic characteristics of the given hydraulic systems. To ensure a comparative analysis of these characteristics of hydraulic systems used in different models of tractors but belonging to a same class, it is necessary to clarify the impact of the key system's elements (pump, relief valve, manual controlled valve, and connecting lines) on the peak pressure's increase. The main purpose of this work is to present a mathematical model for the investigation of the dynamic processes in a given hydro-system used to drive a double-acting hydraulic cylinder, imitating the conditions of determining the peak pressure during different tractors have been tested. Using this model the processes achieved for different system's work regimes can be modified and on the base of the results a quality comparative analysis of the dynamic characteristics of a hydro-system, used in a given tractor, can be ensured.

2. MATHEMATICAL MODEL

An object of this research is a hydraulic drive system having an actuator double-acting hydraulic cylinder, whose principle scheme is given in fig. 1.

- hydraulic aggregate (imitating a tractor's hydro-system), including the following elements: a positive displacement pump 1; a relief valve 2; a hydraulic manual controlled valve 3, a filter 8 and a tank conatining the work fluid (hydro-grease) 8;
- actuator – a double-acting hydraulic cylinder having one-sided piston's rod 5;
- links (hose-pipes) connecting the hydraulic aggregate to the hydraulic cylinder 4 and 6.
- system's work performance in a dynamic work regime can be given by the following equations:

The mathematical model describing the system's work performance in a dynamic work regime can be given by the following equations:

2.1. Actuator's move equation

$$m\frac{dv}{dt}+k_f\,v=A_1\,p_{1m}-A_2\,p_{2m} \qquad (1)$$

where:

v – hydraulic cylinder piston's velocity;

p_{1m}, p_{2m} - pressure, respectively in the piston and rod cameras of the hydraulic cylinder;

m – mass of the moving parts (piston and rod) adducted to the piston of the hydraulic cylinder;

k_f - coefficient of friction;

A_1, A_2 – effective piston's areas from the side of the piston and rod cameras of the hydraulic cylinder.

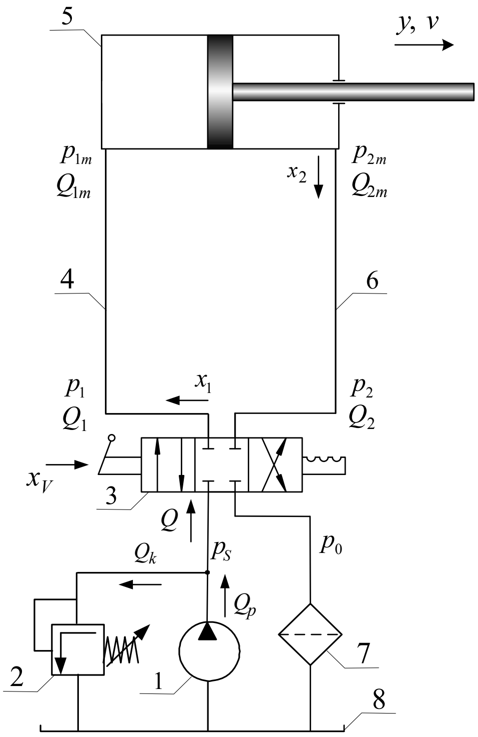

Fig. 1. A scheme of the investigated model hydraulic drive system
1-pump; 2-relief valve, 3-manual controlled valve, 4-high pressure pipe; 5-hydrocylinder, 6-returning pipe, 7-filter, 8-tank

When the cylinder's piston achieves the rightmost (leftmost) work point $|y|=y_{max}$, it is necessary for the movement's physical restrictions to be added:

$$If\ \ y\geq y_{max}\ \ and\ \ \frac{dv}{dt}\geq 0\ \ then\ \begin{cases} y=y_{max}\\ v=0\\ \dfrac{dv}{dt}=0\end{cases}$$

$$If\ \ y\geq y_{max}\ \ and\ \ \frac{dv}{dt}<0\ \ then\ \begin{cases} y=y_{max}\\ v=0\end{cases}$$

$$If\ \ y\leq y_{min}\ \ and\ \ \frac{dv}{dt}\leq 0\ \ then\ \begin{cases} y=y_{min}\\ v=0\\ \dfrac{dv}{dt}=0\end{cases}$$

$$If\ \ y\leq y_{min}\ \ and\ \ \frac{dv}{dt}>0\ \ then\ \begin{cases} y=y_{min}\\ v=0\end{cases}$$

2.2. Equations describing the flow rates of the work fluid going through the hydraulic manual controlled valve 3

$$Q_1=\begin{cases}\mu_1\pi d\,x_v\sqrt{\dfrac{2(p_S-p_1)}{\rho}}, & 0\leq x_v\leq x_{max}\\[2ex] \mu_1\pi d\,x_v\sqrt{\dfrac{2(p_1-p_0)}{\rho}}, & x_{min}\leq x_v<0\end{cases} \qquad (2)$$

$$Q_2=\begin{cases}\mu_2\pi d\,x_v\sqrt{\dfrac{2(p_2-p_0)}{\rho}}, & 0\leq x_v\leq x_{max}\\[2ex] \mu_2\pi d\,x_v\sqrt{\dfrac{2(p_S-p_2)}{\rho}}, & x_{min}\leq x_v<0\end{cases} \qquad (3)$$

where:

Q_1, Q_2 – flow rates of the work fluid going through the hydraulic manual controlled valve;

μ_1, μ_2 – flow rate coefficients for the relevant passages through the hydraulic manual controlled valve;

ρ - working fluid's density;

d – diameter of the hydraulic manual controlled valve's plunger;

p_S, p_0 - input and output pressures of the hydraulic manual controlled valve;

p_1, p_2 - pressures at the begging and the end of the connecting pipes 4 and 7 (fig. 1).

2.3. Equations of the incoming and outgoing through the hydraulic cylinder flow rates, indicating the work fluid's contractility

$$Q_{1m}=A_1v+\frac{V_{01}+A_1\,y\,sign(v)}{B}\frac{dp_{1m}}{dt} \qquad (4)$$

$$Q_{2m}=A_2v-\frac{V_{02}-A_2\,y\,sign(v)}{B}\frac{dp_{2m}}{dt} \qquad (5)$$

where:

Q_{1M}, Q_{2M} - incoming and outgoing flow rate from (to) the hydraulic cylinder;

V_{01}, V_{02} - initial volumes of the rod and piston cameras of a hydraulic cylinder;

y - hydraulic cylinder piston's move, $v = \frac{dy}{dt}$;

B - bulk module of elasticity of hydro-grease.

2.4. Equation of the fluid's movement through the pipe under the high pressure 4 (Fig. 1), $0 \le x_1 \le L_1$

It is assumed that the pipe is long, where the work fluid's contractility and the pipe's contractility are indicated:

$$\begin{aligned} \frac{\partial p}{\partial x_1} &= -\frac{\rho}{A_{T1}}\frac{\partial Q}{\partial t} - \frac{f_1 \rho}{2\, d_1 A_{T1}^2}|Q|Q \\ \frac{\partial p}{dt} &= -\frac{a_1^2 \rho}{A_{T1}}\frac{\partial Q}{\partial x_1} \end{aligned} \tag{6}$$

where:

$p(x_1,t)$, $Q(x_1,t)$ – pressure and flow rate along the pipe's length ;

f_1 - a hydraulic friction coefficient;

d_1 - a pipe's internal diameter;

A_{T1} – pipe's cross-section area, $A_{T1} = \pi d_1^2 / 4$;

a_1 – noise distribution's velocity;

x_1 – a current coordinate along to the pipe's length.

The boundary conditions are:

$$\begin{aligned} x_1 = 0 &\;\rightarrow\; p(0,t) = p_1(t),\; Q(0,t) = Q_1(t) \\ x_1 = L_1 &\;\rightarrow\; p(L_1,t) = p_{1m}(t),\; Q(L_1,t) = Q_{1m}(t) \end{aligned} \tag{7}$$

2.5. Equation the fluid's movement through the pipe 6 (Fig. 1), $0 \le x_2 \le L_2$:

$$\begin{aligned} \frac{\partial p}{\partial x_2} &= -\frac{\rho}{A_{T2}}\frac{\partial Q}{\partial t} - \frac{f_2 \rho}{2\, d_2 A_{T2}^2}|Q|Q \\ \frac{\partial p}{dt} &= -\frac{a_2^2 \rho}{A_{T2}}\frac{\partial Q}{\partial x_2} \end{aligned} \tag{8}$$

where:

$p(x_2,t)$, $Q(x_2,t)$ – pressure and flow rate along the pipe's length;

f_2 - hydraulic friction coefficient;

d_2 - pipe's internal diameter;

A_{T2} – pipe's cross-section area, $A_{T2} = \pi d_2^2 / 4$;

a_2 – noise distribution's velocity;

x_2 – current coordinate along to the pipe's length.

The boundary conditions are:

$$\begin{aligned} x_2 = 0 &\;\rightarrow\; p(0,t) = p_{2m}(t),\; Q(0,t) = Q_{2m}(t) \\ x_2 = L_2 &\;\rightarrow\; p(L_2,t) = p_2(t),\; Q(L_2,t) = Q_2(t) \end{aligned} \tag{9}$$

For finding a solution of the mathematical model for the fluid's movement through the pipes – equations (6)-(9), a high accuracy numerical method consisting of semi-discretization [1] (not described in this work), has been used.

2.6. Equation of the incoming fluid (flow rate) transferred from the hydro-aggregate to the system

According to the equation of continuity for the incoming fluid (flow rate) transferred from the hydro-aggregate to the system, as per the symbols used in fig. 1, the following equation can be given:

$$Q = Q_p - Q_k \tag{10}$$

where: Q - incoming fluid (flow rate) transferred from the hydro-aggregate to the system;

Q_p - pump's flow rate, $Q_p = Q_{tp} - k_y p_s$;

k_y - a coefficient of the pump's volume losses;

Q_k - flow rate through the relief valve:

$$Q_\kappa = k_\kappa (p_s - p_{S0});$$

p_{S0} - relief valve opening pressure;

k_k - a coefficient of the relief valve's characteristic:

$$k_k = \frac{Q_{p\,max}}{p_{s\,max} - p_{S0}};$$

$p_{S\,max}$ - maximal input pressure;

$Q_{p\,max}$ - pump's flow rate when the pressure is $p_{S\,max}$.

Figure 2 shows the approximated work characteristic of the relief valve (pos. 2 in fig. 1). The relief valve is closed and its input pressure p_s is lower than the opening pressure for the same valve, $0 \le p_{S0} \le p_{s0}$. For values of p_s in the range of $p_{S0} < p_S \le p_{S\max}$ the value of the flow rate going through the relief valve can be estimated by using the equation - $Q_\kappa = k_\kappa (p_S - p_{S0})$.

Therefore, indicating the relief valve's characteristic, the incoming fluid (flow rate) transferred from the hydro-aggregate to the system can be found by using the following equation:

$$Q = \begin{cases} Q_{tp} - k_y p_S, & 0 \le p_S \le p_{S0} \\ Q_{tp} - k_y p_S - k_\kappa (p_S - p_{S0}), & p_{S0} < p_S \le p_{S\,max} \end{cases}$$

For the aims of the numerical research, it is necessary to transform the given system of equations to become in dimensionless type by implementing the following dimensionless magnitudes:

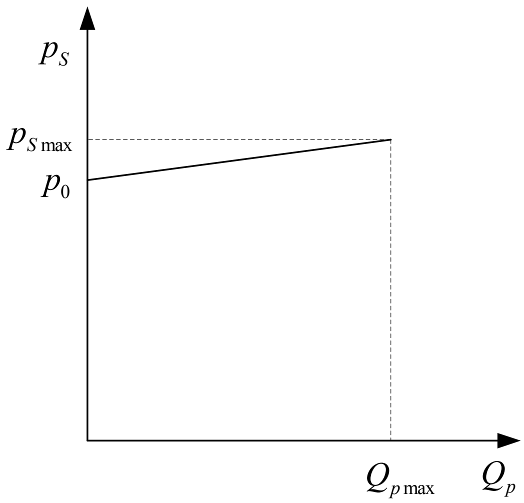

Fig. 2 Relief valve’s characteristic.

$$\overline{y} = \frac{y}{y_{max}};\ \overline{x}_V = \frac{x_V}{x_{V\,max}};\ \overline{v} = \frac{d\overline{y}}{dt};$$

$$\overline{p}_1 = \frac{p_1}{p_{S\,max}};\ \overline{p}_2 = \frac{p_2}{p_{S\,max}};\ \overline{p}_{1m} = \frac{p_{1m}}{p_{S\,max}};$$

$$\overline{p}_{2m} = \frac{p_{2m}}{p_{S\,max}};\ q_1 = \frac{Q_1}{Q_{pt}};\ q_2 = \frac{Q_2}{Q_{pt}};\ q_{1m} = \frac{Q_{1m}}{Q_{pt}};$$

$$q_{2m} = \frac{Q_{2m}}{Q_{pt}};$$

where: y_{max} - is the maximal piston’s move; $p_{S\,max}$ - pump’s maximal input pressure;

Q_{pt} - pump’s theoretical flow rate;

$x_{V\,max}$ - maximal move of the manual controlled valve’s plunger.

On the basis of the hydraulic system’s mathematical model presented in this work an analog model of the same system has been developed. Its scheme can be seen in fig. 3.

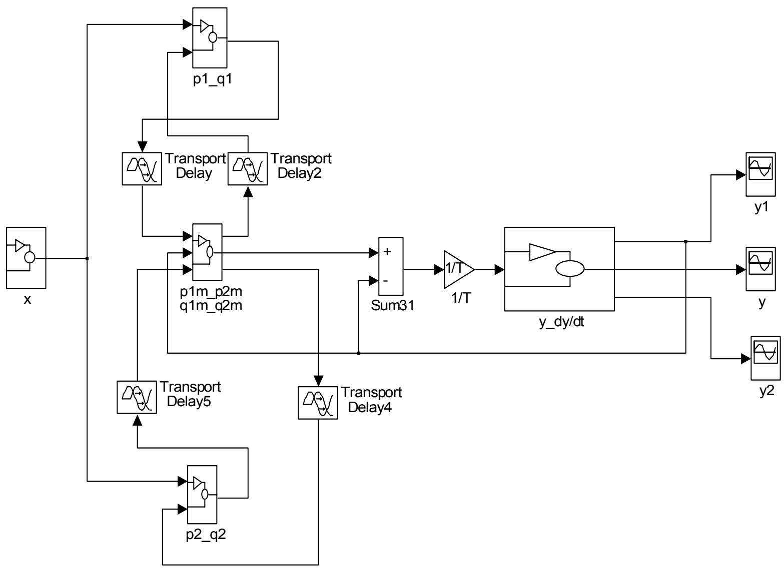

Fig. 3. A scheme of the system’s analog model.

Table 1. Values of some of the significant parameters used in this research.

d_1	10 mm	d	30 mm	L_1	2.5 m	a_2	330 m/s
Q_{po}	40 l/min	P_{S0}	16 MPa	L_2	2.5 m	μ_1	0.62
D	80 mm	p_{Smax}	17 Mpa	d_v	16 mm	μ_2	0.62
B	$2.1\text{x}10^9$Pa	m	3 kg	a_1	330 m/s	ρ	880kg/m^3

3. RESULTS AND DISCUSSION

A solution of this problem can be found by using two different computing schemes in respect to the fluid’s move through the pipes, connecting the hydraulic manual controlled valve to the hydraulic cylinder (pos. 4 and 7 in fig. 1) – with distributed and allocated parameters.

For the scheme with the allocated parameters, if the impact of the connecting pipes on the cylinder’s work is not indicated, the equations (6), (7), (8) and (9) shouldn’t exist in the mathematical model’s structure, as in this case it will be valid that: $p_{1m}=p_1$; $p_{2m}=p_2$; $Q_{1m}=Q_1$; $Q_{2m}=Q_2$ .The initial research indicate that for the simulation of the dynamical processes by using this computing scheme the results do not give any qualitative information about the actual processes of the peak pressure’s increase. That is why for the existing research it has been used a computing scheme whose parameters are distributed. For finding a solution of the mathematical model for the fluid’s movement through the pipes – equations (6)-(9), a high accuracy numerical method consisting of semi-discretization [2] (not described in this work), has been used.

The values of some of the important parameters used to ensure the simulation process are given in table 1.

Figure 4 shows a graph of the transitional processes existing in the given hydraulic drive system, when the cylinder’s piston has been moved to its right direction (a forward move).

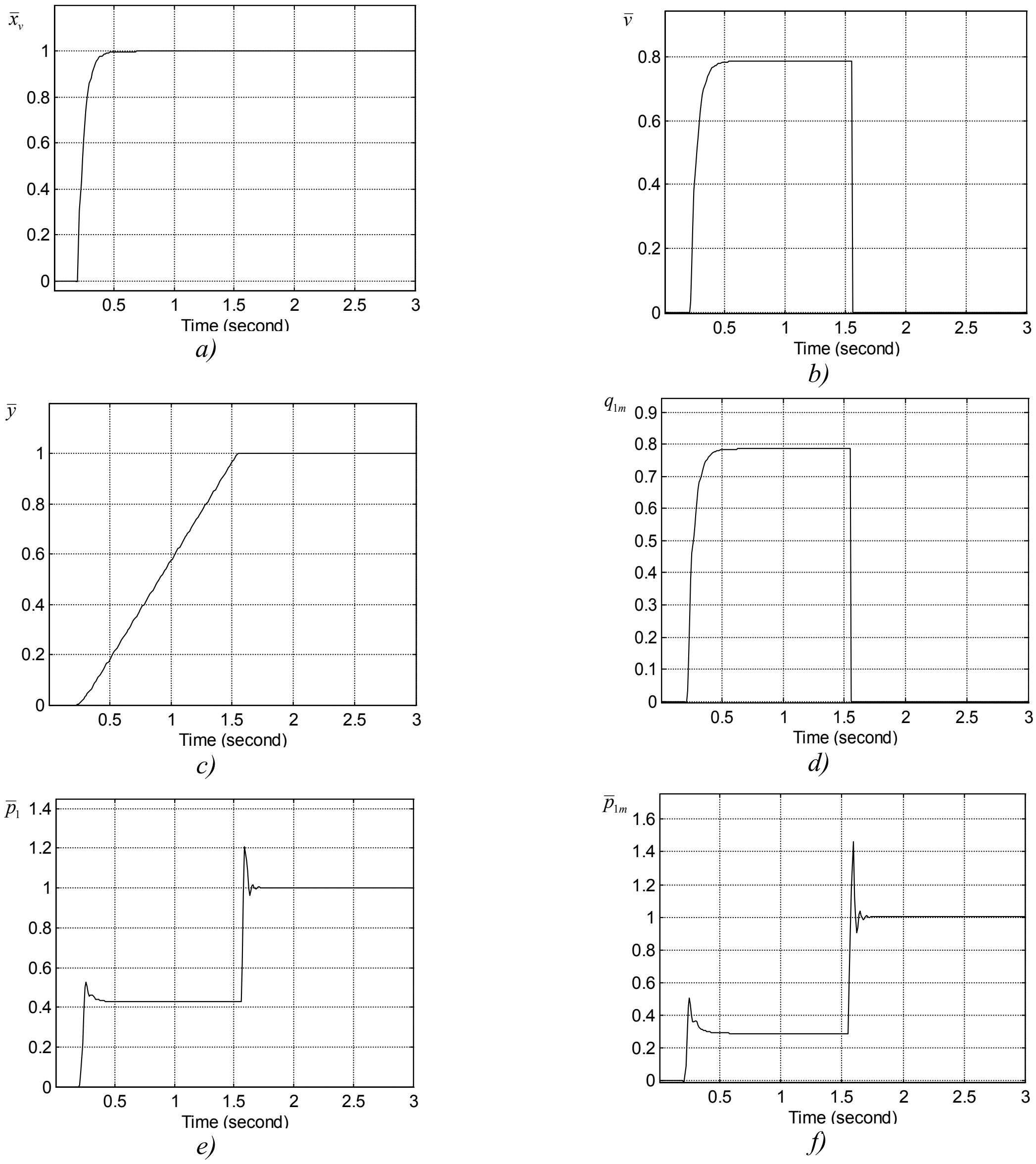

Fig. 4. System’s transitional processes.

For example in fig. 5 it can be seen the experimentally received initial characteristics of the transitional processes existing at the end of each of the hydraulic cylinder piston’s moves, leading to a peak pressure’s increase.

The accomplished comparative analysis indicates that the solution of the developed mathematical model, consisting of distributed parameters in terms of the fluid’s movement through the pipes, ensure analogical qualitative results compared to the results found experimentally. The graphs presented in figures 4 and 5 indicate that the values of the theoeritaclly and experimentally found peak pressures at the end of the piston’s move have similar character. The difference between the magnitudes of these peak pressure values can be explained by the existing difference between the flow rate (Q=40 l/s) used for the numerical simulation and the flow rate (Q=10 l/s) of the experimentally ensured transional process.

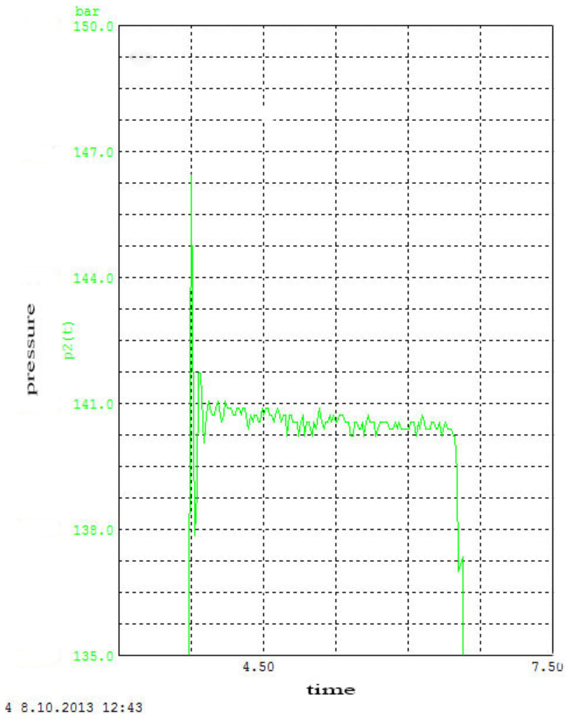

Fig. 5. Experimentaly achieved characteristics of the transitional processes existing at the end of each of the piston's moves for a given experimental hydro-cylinder.

Figures 4e and 4f show that there is a significant difference in the values of the peak pressure at and begging and the end of the pipe 4 (fig. 1). This is due to the impact of the high elasticity of the connecting lines – hydro-hoses 4 and 6, on the sound distribution's velocity, which is used in equations (6) and (8).

The received initial results (theoretical and experimental) clearly show that the developed mathematical model provieds information about some specific features of the scheme used for the measuring of the given hydraulic system's peak pressure. This can be used to illustrate the hydraulic system's dynamic processes (fig. 1) on supposition that first they have been modeled and simulated. However, its main function is to value the impact of the different system's elements on the system's type and the magnitude of its peak pressure's increase at the end points of the hydraulic drive actuator's moves.

4. CONCLUSION

A mathematical model which can be used for the simulation of the dynamic processes related to the change of the peak pressure of different hydraulic systems, used in tractors, has been developed. In this model the characteristics of the system's key elements – pump, relief valve, manual controlled valve and their connecting elements, are indicated. It makes possible the investigation of the impact of these elements on the existing trasitional processes at the end of the piston's moves, which determine the magnitude of the hydraulic system's pressure. For the better coincidence of the numerical simulations and the experimentally found transitional processes it is necessary to identify the given system by the elaboration of an identification procedure for making a comparison between the two different types of processes.

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