

Analytical study of the dynamics of a pumping unit, consisting of a fixed pump and a flow rate regulator

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Abstract. A mathematical model of a pumping unit, consisting of a fixed pump and an automatic flow rate regulator, intended for use as part of a mathematical model of a volume hydraulic drive that requires a constant supply pressure for its operation is described in article. As the carried out studies showed (their results are given in this article), the proposed model describes the physical processes during the operation of the pump unit quite well. In addition, based on these studies, recommendations are given on the need to include a hydropneumatic accumulator in the structure of the unit, which will significantly improve its dynamic characteristics.

Introduction

One of the main characteristics of transport machines is the average speed, which is limited and determined by the traffic conditions and technical characteristics of the machine. Limitations, depending on traffic conditions include limitations on thrust force, on the danger of uncontrolled movement and on smoothness of movement [1]. An increase in the specific power of modern machines and the correct choice of traction motors made it possible to remove a number of limitations on the thrust force [2]. Methods to reduce the probability of occurrence of uncontrolled movement are the subject of [3]. In this context, the problem of improving the cushioning systems of transport machines for an increasing the motion speed on rough road is being important.

Recently, hydropneumatic cushioning systems are widely used in transport, which allow obtaining progressive suspension characteristics and changing the position of the coachbuilder in space. Determining the characteristics of the elements included in the hydropneumatic cushioning system is impossible without the use of simulation mathematical simulation.

Mathematical models of the movement of transport machines for studying smoothness of movement are widely presented in the literature [4–14], they are used to obtain the characteristics of elastic and damping elements of the suspension. In this paper, a mathematical model of the pumping unit of the coachbuilder position change system is presented. The quality of work and the system operation speed influence on the possibility of its use when the machine is moving, which is typical for controlled active cushioning systems [15–31].

The paper proposes a mathematical model of a pumping unit, consisting of a fixed pump and an automatic flow rate regulator, intended for use as part of a mathematical model of a volume hydraulic drive that requires a constant supply pressure for its operation.



Methods and instruments of research

Based on the analysis of the current pumping units used in practice [32–35], which provide a constant supply pressure at the inlet to the hydraulic system, it is proposed to use a unit including a fixed axial-piston pump and an automatic flow rate regulator when developing this mathematical model. The corresponding design model is shown in Fig.1.

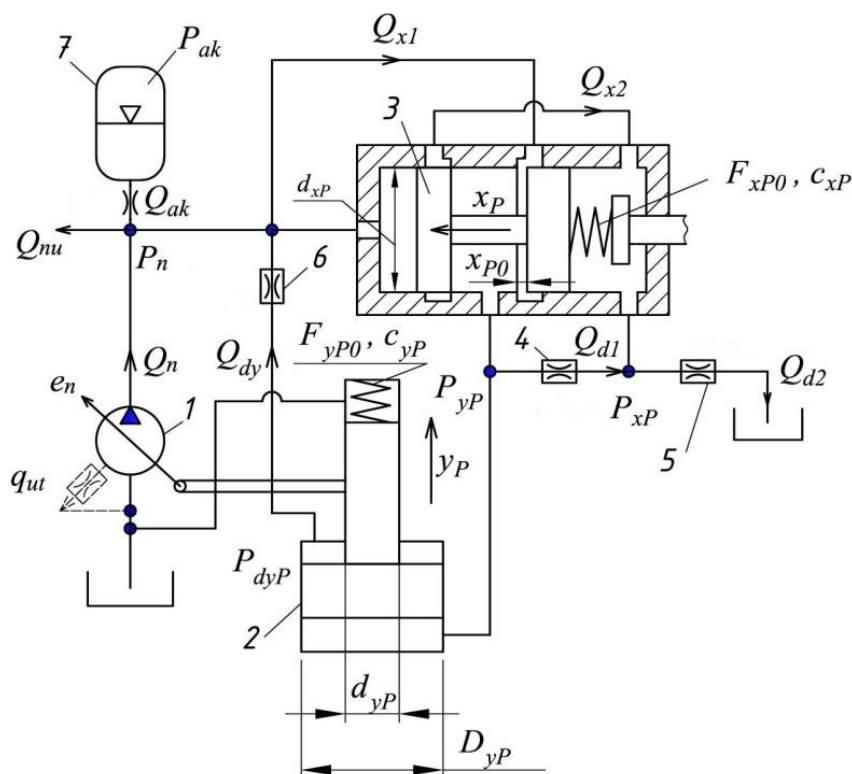


Fig. 1. Design model of the pumping unit: 1 — fixed axial-piston pump; 2 — differential piston of the flow rate regulator; 3 — spool throttle control valve; 4, 5, and 6 — tuned throttles; 7 — hydropneumatic accumulator

The mathematical description of the adopted version of the pumping unit includes:

- 1) equation characterizing the change in manometer pressure p_n at the pump outlet:

$$\dot{p}_n = \frac{B_g}{W_n} \cdot \left(Q_n - Q_{nu} - Q_{ak} - Q_{x1} + Q_{dy} + \dot{x}_p \cdot \frac{\pi d_{xp}^2}{4} \right)$$

- 2) equation characterizing the change in manometer gas pressure p_{ak} in the hydropneumatic accumulator:

$$\dot{p}_{ak} = \frac{(p_{zap} + p_{at}) \cdot n \cdot W_g^n \max \cdot Q_{ak}}{(W_g \max - W_{ak})^{n+1}}$$

- 3) equation characterizing the piston movement y_p in regulator:

$$m_{y_p} \cdot \ddot{y}_p = p_{y_p} \cdot \frac{\pi D_{y_p}^2}{4} - p_{dy_p} \cdot \frac{\pi}{4} \left(D_{y_p}^2 - d_{y_p}^2 \right) - \left(F_{y_p 0} + c_{y_p} \cdot y_p + k_{y_p} \cdot \dot{y}_p \right)$$

4) equation characterizing the change in manometer pressure p_{yp} in the piston cavity of the regulator:

$$\dot{p}_{yp} = \frac{B_g}{W_{yp}} \cdot \left(Q_{x1} - Q_{x2} - Q_{d1} - \dot{y}_p \cdot \frac{\pi D_{yp}^2}{4} \right)$$

5) equation characterizing the change in manometer pressure p_{dyp} in the rod cavity of the regulator:

$$\dot{p}_{dyp} = \frac{B_g}{W_{dyp}} \cdot \left(\dot{y}_p \cdot \frac{\pi}{4} (D_{yp}^2 - d_{yp}^2) - Q_{dy} \right)$$

6) equation characterizing the movement of the spool x_p of the regulator:

$$m_{xp} \cdot \ddot{x}_p = (p_{xp} - p_n) \cdot \frac{\pi d_{xp}^2}{4} + F_{xp0} - (c_{xp} \cdot x_p + k_{xp} \cdot \dot{x}_p)$$

7) equation characterizing the change in manometer pressure p_{xp} in the end cavity of the regulator spool:

$$\dot{p}_{xp} = \frac{B_g}{W_{xp}} \cdot \left(Q_{x2} + Q_{d1} - Q_{d2} - \dot{x}_p \cdot \frac{\pi d_{xp}^2}{4} \right)$$

In these equations, in addition to the notations used in Fig. 1, the following notations are agreed:

p_{zap} —manometer pressure of gas charging of the hydropneumatic accumulator;

p_{at} —atmosphere pressure, $p_{at} = 0,1$ MPa;

$W_{g \max}$ —constructive volume of hydropneumatic accumulator;

n — gas polytropic coefficient, $n=1,4$;

W_{ak} —volume of fluid in the hydropneumatic accumulator at present sampling time:

$$W_{ak} = W_{ak0} + \int_0^t Q_{ak} \cdot dt$$

W_{ak0} — initial volume of fluid in the hydropneumatic accumulator;

B_g —fluid modulus of inelastic buckling, $B_g = 1000$ MPa;

$W_n, W_{yp}, W_{dyp}, W_{xp}$ — fluid volumes reduced to the corresponding node points;

m_{yp} — reduced mass of the regulator piston;

k_{yp} —coefficient of viscous friction of the regulator piston;

m_{xp} —regulator spool mass;

k_{xp} —coefficient of the spool viscous friction of the regulator.

Taking into account the goal and prospects for further use, the mathematical model developed in the MATLAB Simulink software package based on the mathematical description of the pumping unit above is formed as a block «Pumpy» (Fig. 2). The input signals for this block are the pump shaft speed $n_n = nn$ and the flow rate consumed by the connected hydraulic system from the pumping unit $Q_{nu} = Qnu$, and output signals — are the pressure $p_n = pn$ created by the pump, and the resulting moment of resistance to rotation of the pump shaft $M_n = Mn$.

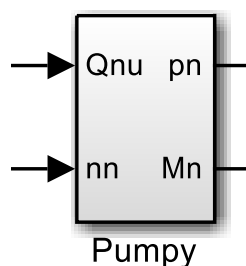


Fig.2. Block «Pumpy»

The composition of the block «Pumpy» is shown in Fig. 3.

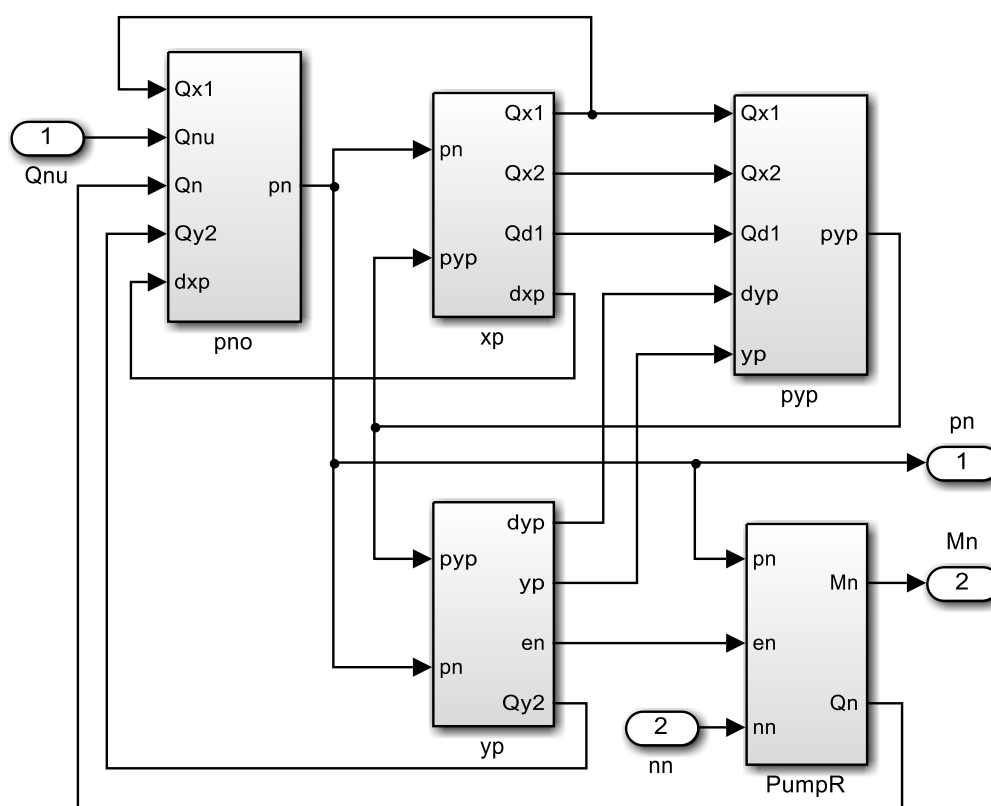


Fig.3. The composition of the block «Pumpy»: pno— block specified pressure p_n ; xp— block specified movement x_p ; pyp— block specified pressure p_{y_p} ; yp— block specified movement y_p and regulation parameter of pump e_n ; PumpR— block specified the flow rate of variable pump Q_n and the shaft torque M_n

The initial data (values of design parameters), that are needed for the analysis of the operation of the pumping unit were taken from design considerations, as well as on the basis of the conditions for obtaining the required static characteristics. In addition, the resulting dynamics of the transient response processes accompanying the operation of the pump unit were taken into account. The type of static characteristics obtained as a result of the calculation and taken as the basis is shown in Fig. 4 (abscissa axis — l/s; left ordinate axis — MPa; right ordinate axis is dimensionless).

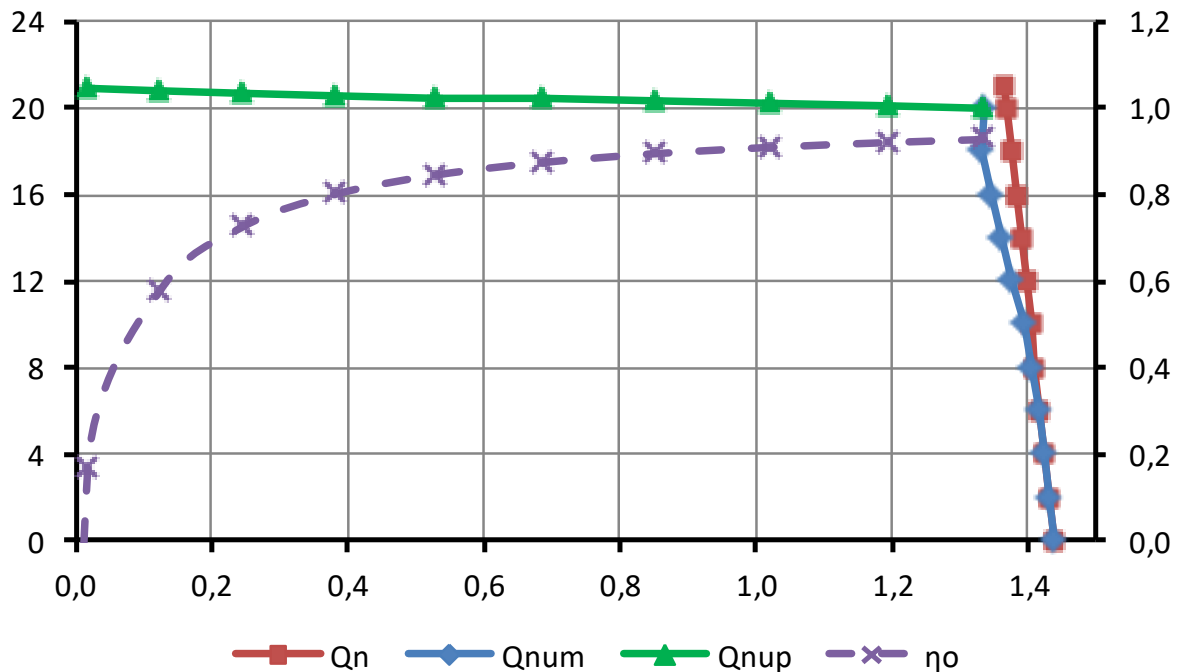


Fig.4. Static characteristics of pump unit: Q_n —pq pump curve; Q_{num} - pq pump curve with $p_n < 20$ MPa; Q_{nup} —pq pump curve with $20 < p_n < 21$ MPa; Efficiency —the change in volumetric efficiency of the pumping unit with $20 < p_n < 21$ MPa (right axis values)

It should be also noted that further studies of the developed mathematical model of the pumping unit under consideration suggest that the initial steady state of its moving elements corresponds to those shown in Fig. 1. This operation mode of the pumping unit occurs when the pump shaft rotational speed is constant and there is no fluid consumption of the hydraulic system connected to it. Therefore, before carrying out the planned studies, it is necessary to determine the appropriate initial conditions.

For this purpose, the simulation of the process of turning on the engine driving of the pump was completed.

The calculation results are shown in Fig. 5 (along the abscissa axis — time in seconds).

In the simulation, it was assumed that the pump shaft rotational speed increases linearly from zero to the adopted $n_n = 1920$ rpm in 1 second (Fig. 5a), and the design volume of the hydropneumatic accumulator $W_{g\max} = 2$ l.

The graphs show that approximately 6 seconds after turning on the pump drive, all transient responses associated with this are completed. The resulting steady-state values of the variable values were used as initial conditions for further mathematical simulation the operating mode of the pump unit under study. These initial conditions are the following variable values: $p_n = p_{ak} = p_{dy_p} = 20,902$ MPa, $p_{y_p} = 20,1$ MPa, $p_{x_p} = 0,0034$ MPa, $W_{ak} = 1$ l, $x_p = 0,0084$ mm, $y_p = 28,025$ mm, with $e_n = 0,066$.

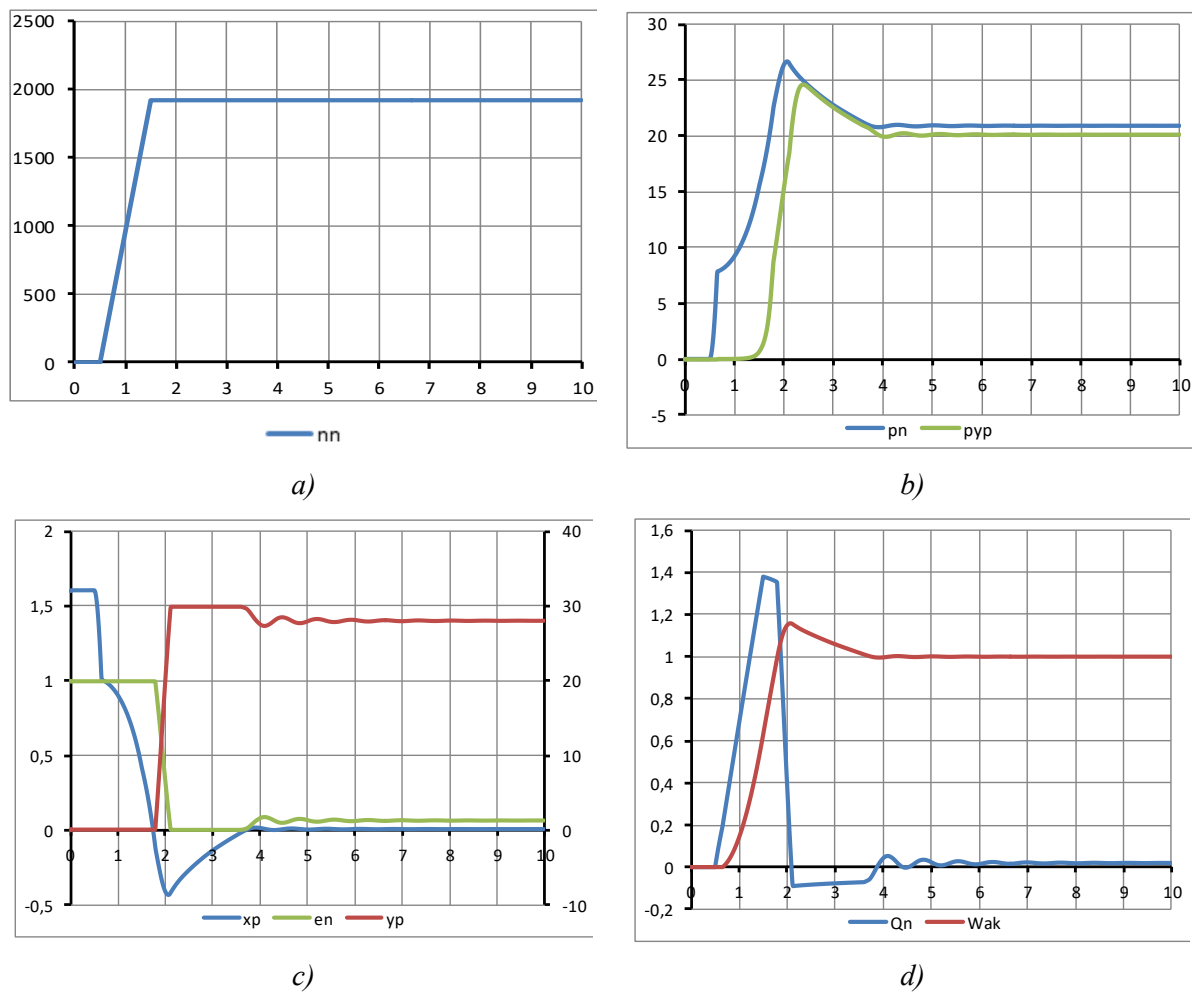


Fig. 5. The results of simulation of the turning on the engine driven of the pump: *a*— the initial frequency change law of the pump shaft speed n_n (rpm); *b*— pressure changes p_n and p_{yp} (MPa); *c*— movement of the spool x_p (mm) and piston y_p (mm on the right axis) of the regulator and the corresponding change in the regulation parameter e_n ; *d*— changes in the pump flow Q_n (l / s) and the volume W_{ak} (l) of fluid in the hydropneumatic accumulator

Research Results

In the study of the dynamic processes accompanying the operation of the pump unit, the behavior of the change in the amount of fluid flow consumed from it, shown in Fig. 6 (the abscissa axis shows time in seconds; the ordinate axis l / s), is taken as the initial external disturbance. Fig. 7 and Fig. 8 show the results of mathematical simulation of the pumping unit in question. For comparison, the figures show the results for a pump installation without a hydropneumatic accumulator (Fig. 7a and Fig. 8a) and with its presence (Fig. 7b and Fig. 8b).

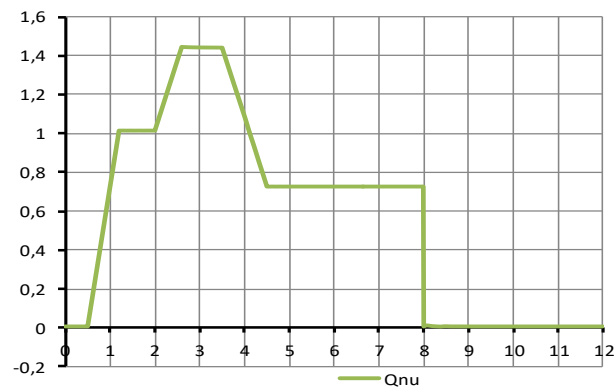


Fig. 6. The behavior of the change in flow rate Q_{nu} adopted in the simulation

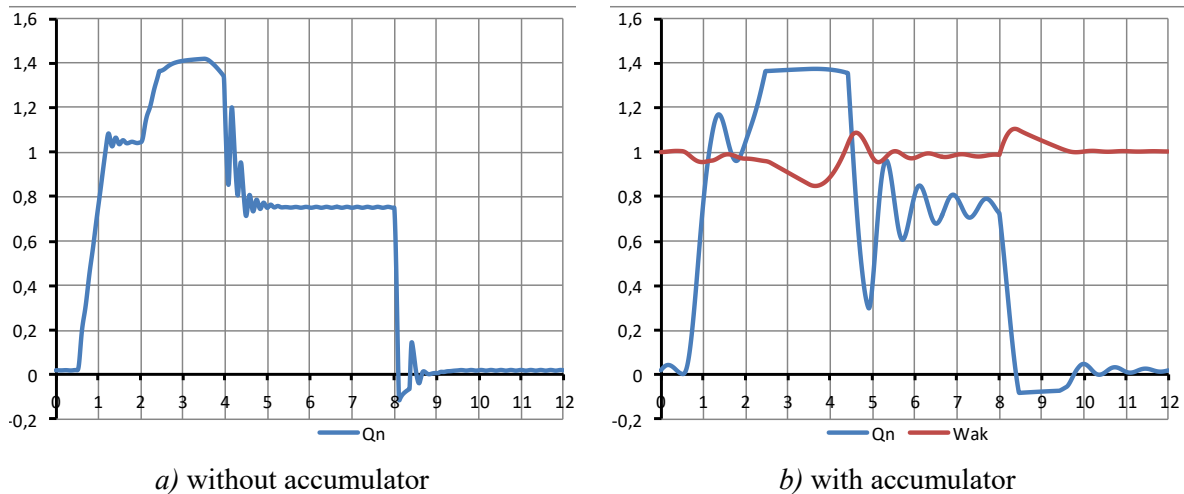


Fig. 7. The behavior of the change in the flow rate Q_n (l / s) of the pump and the volume of fluid W_{ak} (l) in the accumulator

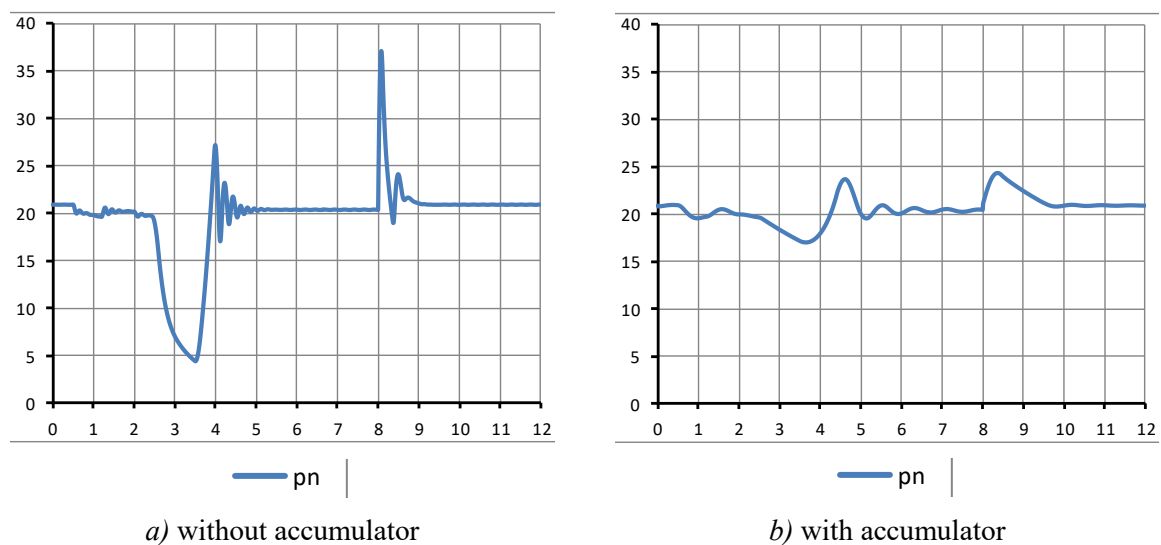


Fig.8. The behavior of the pressure p_n (MPa) change at the pump outlet

Conclusion

An analysis of the results of mathematical simulation of the pumping unit operation in question shows that:

- 1) the developed mathematical model adequately reflects the physical processes that accompany the operation of the pumping unit in question under simulated modes;
- 2) this mathematical model can be recommended for simulation the operating modes of the pumping unit in question with a connected to it hydraulic system, which requires a significantly changing flow rate of pressure fluid for its operation;
- 3) the use of a hydropneumatic accumulator at the outlet of the pump unit in question significantly improves the quality of its operation in the conditions of implementation of transient response dynamic processes. That is, the use of a hydropneumatic accumulator at the output of the pump installation should be recognized as necessary.

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