CONCEPT, CIRCUIT DIAGRAM AND ALGORITHM FOR CONTROLLING MULTI-POSITION PNEUMATIC ACTUATOR WITH ADAPTIVE POSITIONING MODE

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The article is devoted to the development of the concept of adaptive control of a pneumatic actuator with the learning mode, a pneumatic circuit diagram and a mathematical model of a relay-operated multi-position pneumatic actuator. With the resulting pneumatic circuit there is no need in using throttle braking any more; the mode of operation of the pneumatic engine can be controlled only by changing the structure of switching connections. Thus, the most favorable uniformly decreasing braking mode can be created, even with significant inertial loads. The described control algorithm based on the obtained pneumatic circuit enables effective implementation of a learning program of the pneumatic actuator.

Keywords: multi-position pneumatic actuator, adaptive positioning mode, learning mode

1. Introduction and problem statement

Compressibility of air has long been preventing the implementation of free programming of the positioning points of the working element (WE) of the pneumatic actuator (PA). Therefore, at the initial stages of development of multi-position PA, the problem was solved by manufacturing actuators with a rigid positioning program. As an industrial design solution which was used commercially in loader cranes and other control systems, multi-piston actuators are worth mentioning [11].

The advance of computer technology resulted in microprocessor controls becoming cheaper and having significantly faster response. This has made using such a control in PA systems economically feasible. The positive effect of the transition to such control is the possibility to use much more complex control algorithms in real time. In view of this, discretely controlled PAs are most interesting, since they are more suitable for microprocessor control than analog tracking PAs. One of the first attempts to use a digital computer for discrete

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control of PA using standard two-position valves was made in [4]. However, sufficiently effective control algorithms could not be used because of the low level of development of computer technology in that time (1977-78). Nguen H. T. [10] moved much further in this direction since he could make advantage of much more advanced computer technologies. The author suggested using a two-stage positioning mode: free movement without any influence of the feedback to the positioning point for major positioning errors and actuating relay feedback for minor positioning errors.

Fruhaut F. and Juker K. R. [6] were the first to notice that when the computer-aided control is used, then a previous learning mode for the relay PA can be created. The inaccuracy of a stop of the piston when the brake valve actuates can only be compensated by returning the piston to its original position and adjusting the switching moment (coordinates) of the valve, taking into account the sign and magnitude of the positioning error from the previous trial. As a result of earlier learning in a series of trials, the computer selects the optimal valve control mode.

Linnet J. A. and Smith M. C. [8, 9] suggested using a two-stage positioning strategy, when at the second stage the switching line has the form of a nonlinear function. The latter corresponds to a parabola in the phase plane, which coincides with the piston trajectory during radical braking.

In the following years, significant progress in the development of pneumatic controls was largely determined by the commitment to combine pneumatic equipment with microprocessor controls. The nomenclature of leading manufacturers (such as Festo, SMC, Camozzi, etc.) included long-stroke low-friction pneumatic cylinders that provide high positioning accuracy, as well as pneumatic cylinders with precision pneumatic mechanical retainers [12]. The nomenclature of manufacturers of pneumatic equipment included programmable logic controllers (PLCs), digital-to-analog and analog-to-digital converters; incremental optical position sensors with high-resolution digital output are also widely used, as well as small-size reed switches to monitor the piston position. In [1, 2, 5, 7], the prospects of pneumatic actuators with digital control with a PLC are investigated and the possibility of achieving high accuracy in creation of a trajectory of the working element of PA with the analog control (accuracy up to ± 90 μm) is shown.

However, it should be noted that these results were achieved when PA operated in conditions of low inertial loads at relatively low speeds of movement of the working element of PA.

2. Purpose and objective of the study

The purpose of the study is to develop the concept of adaptive control with a learning mode, pneumatic circuit diagram and mathematical model of a relay-operated multi-position pneumatic actuator, as well as to study the regularities of its functioning. The advantage of such an actuator is the possibility to use it under significant inertial loads while simultaneously ensuring high performance.

3. Control concept

The fundamental difference of a relay-controlled system from an analog-control system is the method of formation of a control action depending on the feedback signal.

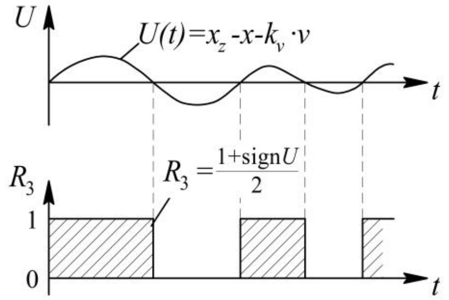


Fig. 1. Formation of discrete control signals relative to the circuit in Fig. 3

Like analog tracking systems, pneumatic relay systems are equipped with position and speed sensors and have a feedback loop. However, instead of an analog servo valve that responds to the sign and value of the mismatch, they have a relay pneumatic control valve and respond only to the sign of the mismatch function (Fig. 1). The analog mismatch signal is converted into homogeneous control pulses (Fig. 1) by using a polarized relay. In case of the relay control, the mismatch function $U = x_z - x - k_v v,$ which is represented in the phase plane as an inclined switching line, appears to be most rational (Fig. 2).

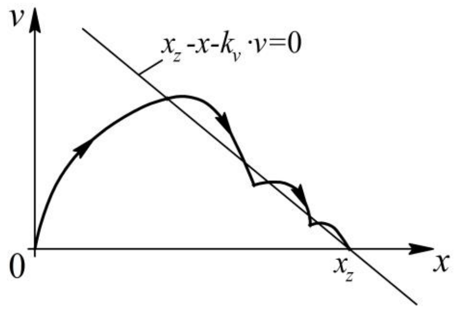


Fig. 2. Movement of WE of PA along the switching line in the phase plane

The process of the switching line tracking is easy to implement technically, because this algorithm is identical to the use of proportional-differential feedback with a relay element in the direct circuit. If the mismatch function has the form $U = x_z - x - k_v v,$ and the relay element (discrete pneumatic control valve) responds only to the sign of the mismatch function, this ensures the movement of the WE along the switching line defined by the equation $x_z - x - k_v v = 0$ with a greater or less deviation from it (Fig. 2).

However, the need for the constant switching line tracking (Fig. 2) leads to a fairly long actuation process and prevents the creation of a learning mode in order to achieve maximum response.

In this regard, the three-stage positioning mode (Fig. 4) seems to be the most rational. In this mode, at the first two stages: drive and brake until the first stop, the WE of the actuator remains open (trajectories 0-1-2 (Fig. 4). The third (final) stage is the switching line tracking (trajectory 2-3-4), when the relay feedback is activated and the reverse control valve switches depending on the sign of the mismatch function U (Fig. 4).

In this case, the testing (learning) process means the adjustment of the coordinates of the beginning of braking x. Then, a randomly selected coordinate of the beginning of braking x_b^0 as a result of a series of trials is adjusted in accordance with a simple recurrent formula

$$x_b^i = x_b^{i-1} + \frac{x_v^{i-1} - x_v}{r}, \tag{1}$$

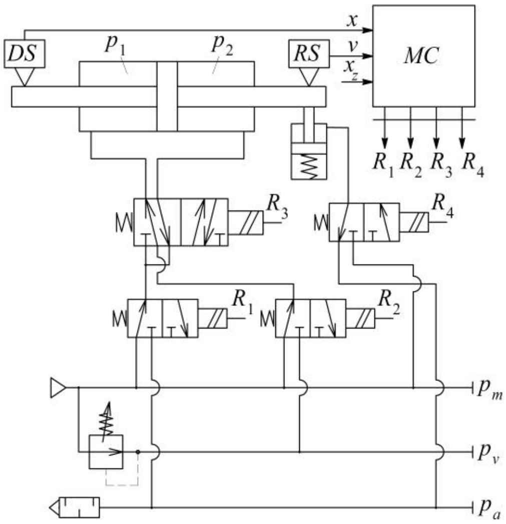
where x_b^{i-1}, x_v^{i-1} are coordinates of the beginning of braking and the first stop in the previous trial, respectively; x_z is the specified positioning coordinate; x_b^i is the coordinate the beginning of braking in the next trial.

If, as a result of testing at the i^{th} step (trial) $\left| x_z - x_v^i \right| < \varepsilon,$ then the optimal coordinate of the beginning of braking x_b^* for this positioning coordinate x_z is remembered. The resulting positioning mode will be optimal, because the WE will be positioned with the specified accuracy only with a single switch of the brake valve.

The transition to the third positioning stage in the operational mode is possible only in case the operating conditions change so that the positioning coordinate goes beyond the specified accuracy as a result of braking. The third stage will not only be a “safety” one, i.e., ensuring unconditional reproduction of the given positioning coordinate under varying operational conditions, but also ensure the change of the coordinate of the beginning of braking x_b^*. The latter property can be absolutely reasonably considered an implementation of the *adaptive mode*.

4. Pneumatic circuit diagram and mathematical model

The pneumatic circuit diagram of PA is developed using the standard 5/2 and 3/2 relay control valves and provides constant and adjustable pressure drops on the piston during braking and the switching line tracking. This ensures a non-fluctuating uniformly decreasing braking mode of the WE even under large inertial loads (Fig. 3).



Phase of motion	R_1	R_2	R_3	R_4
Initial condition (fixation)	0	0	0	0
Drive to the right	1	0	0	1
Brake to the right	0	1	0	1
Switching line tracking option A	0	1	$(1+signU)/2$	1
Switching line tracking option B	1	1	$(1-signU)/2$	1
Brake to the left	1	0	1	1
Brake to the left	0	1	1	1

Fig. 3. Pneumatic circuit diagram and control algorithm for implementing a three-stage positioning process

The mathematical model of the pneumatic actuator in Fig. 3 (2) is obtained on the basis of thermodynamic dependences for a body of variable mass [3]. The equations of energy (heat) balance are written as a dependence of the speed of pressure change in the cylinder cavities on the parameters of the pneumatic system. On the basis of the differential gas state equation, the dependences of the rate of change of the absolute temperature in the same cavities on the parameters of the pneumatic system are obtained.

In this case, the experimental nature of the flow rate in the switching paths of the actuator was taken into account when using the circuit in Fig. 3, as well as the transition from filling to emptying the cylinder cavities, a discrete change in the capacity of the pipelines connected to left and right cylinder cavities. It was assumed that the heat exchange of the cylinder cavities with the environment should not be taken into account because of a significant speed of the process.

$$\begin{cases} \dfrac{dp_1}{dt} = \dfrac{k}{W_0 + F\,x}\left[\sqrt{kR}\cdot s_1 z_1 \varphi\left(I_1\right) - p_1 F\,\dfrac{dx}{dt}\right]; \\ \dfrac{dT_1}{dt} = \dfrac{T_1 F_1}{W_0 + F\,x}\cdot\dfrac{dx}{dt} + \dfrac{T_1}{p_1}\cdot\dfrac{dp_1}{dt} + \dfrac{s_a z_1 \sqrt{kR}\cdot\varphi\left(I_1\right)}{F\,x + W_0}; \\ \dfrac{dp_2}{dt} = \dfrac{k}{L + x_{02} - x}\left[\dfrac{\sqrt{kR}}{F}\cdot s_2 z_2 \varphi\left(I_2\right) - p_2\,\dfrac{dx}{dt}\right]; \\ \dfrac{dT_2}{dt} = \dfrac{T_2}{x_{02} + L - x}\cdot\dfrac{dx}{dt} + \dfrac{T_2}{p_2}\cdot\dfrac{dp_2}{dt} + \dfrac{s_b z_2 \sqrt{kR}\cdot\varphi\left(I_2\right)}{F\left(L + x_{02} - x\right)}; \\ \dfrac{dx}{dt} = v; \\ \dfrac{dv}{dt} = \dfrac{1}{m}\left(p_1 F - p_2 F - P_{fr}\right), \end{cases} \tag{2}$$

where $\varphi\left(I\right)$ is the flow function written with reference to both supercritical and subcritical gas flow modes.

$$\varphi\left(I\right) = \frac{1 + sign(I - 0,528)}{2}\cdot\sqrt{\frac{2}{k-1}\cdot\left(I^{\frac{2}{k}} - I^{\frac{k+1}{2}}\right)} + 0,579\cdot\frac{1 - sign\left(I - 0,528\right)}{2}, \tag{3}$$

p_1, p_2, T_1, T_2 are pressure and temperature in the cylinder cavities; k is adiabatic index; F is the area of the piston; x is the coordinate of the piston; L is full stroke of the piston; m is mass of moving parts; P_{fr} is friction force; $x_{01} = \dfrac{W_{01}}{F}$, $x_{02} = \dfrac{W_{02}}{F}$ is initial coordinate of the piston on the left and right; I is the ratio of pressure at the ends of the feed and exhaust pipelines:

$I_1 = \left(\dfrac{p_1}{p_L}\right)^{sign\left(p_L - p_1\right)}$; $I_2 = \left(\dfrac{p_2}{p_p}\right)^{sign\left(p_p - p_2\right)}$ are ratio of pressure at the ends of the switching paths for the left and right cylinder cavities, respectively.

Pressures p_p and p_L in the switching objects for the right and left cylinder cavities are variable and depend on the state of the solenoids. R_1, R_2, R_3. Whereas, R is considered as a Boolean variable (R=1 – the solenoid is powered; R= 0 – no current):

$$\begin{cases} p_L = \left[(1-R_1)R_3 + (1-R_2)(1-R_3)\right] p_m + R_2(1-R_3)\,p_v + R_1 R_3 \cdot p_a; \\ p_p = \left[(1-R_2)R_3 + (1-R_1)(1-R_3)\right] p_m + (1-R_2)\,p_v + R_1(1-R_3)\,p_a. \end{cases} \quad (4)$$

Logical algebraic functions s_1, s_2, s_a, s_b discretely change the structure of the right hand sides of the first four equations (2) when switching from filling the cylinder cavity to emptying and vice versa:

$$\begin{cases} s_1 = \dfrac{1+sign(p_L - p_1)}{2} p_L \sqrt{T_m} - \dfrac{1+sign(p_1 - p_L)}{2} p_1 \sqrt{T_1}; \\ s_a = -\dfrac{1+sign(p_L - p_1)}{2} \dfrac{T_1^2 p_L}{\sqrt{T_m}\, p_1} + \dfrac{1+sign(p_1 - p_L)}{2} \sqrt{T_1} T_1; \\ s_2 = -\dfrac{1+sign(p_p - p_2)}{2} p_p \sqrt{T_m} + \dfrac{1+sign(p_2 - p_p)}{2} p_2 \sqrt{T_2}; \\ s_b = -\dfrac{1+sign(p_p - p_2)}{2} \dfrac{T_2^2 p_p}{\sqrt{T_m}\, p_2} + \dfrac{1+sign(p_2 - p_p)}{2} \sqrt{T_2} T_2. \end{cases} \quad (5)$$

Adjustments z_1 and z_2 simulate discrete changes in the flow capacity of main pipelines depending on the state of the pneumatic control valves.

The control algorithm in relation to the circuit diagram in Fig. 3 when a three-stage positioning mode is implemented has the form shown in Table 1.

Table 1

PA control algorithm

No.	Phase	The switch condition of the valves
1	Determination of the initial conditions x_0, x_z, k_v	
2	I. Drive to the right	If $x_z > x_0$, then $x_b^0 = \dfrac{x_z - x_0}{2} + x_0$ – choosing the initial coordinate of "braking". If $x < x_b^0$, then $R_1 = R_4 = 1$; $R_2 = R_3 = 0$. Otherwise, transfer to phase II.
3	II. Brake	If $x > x_b^i$ and $\lvert v \rvert > \varepsilon_v$, then $R_1 = R_3 = 0$; $R_2 = R_4 = 1$. If $x > x_b^i$ and $\lvert v \rvert \le \varepsilon_v$, then $x_v^i = x$. If $\lvert x_v^i - x_z \rvert > \varepsilon$, then $x_b^{i+1} = x_b^i - \dfrac{x_v^i - x_z}{r}$ (determining a new coordinates of the beginning of braking), transfer to phase III. Otherwise, the transition to phase IV.

4	III. Movement to the positioning coordinate along the switching line (option A)	$U = x_z - r - k_v \cdot v$ and $R_3 = \frac{1+signU}{2}$, i.e. $R_3 = 1$ at $x < x_z - k_v \cdot v$ and $R_3 = 0$ at $x > x_z - k_v \cdot v$. If $\lvert v \rvert < \varepsilon_v$, then phase IV.
5	IV. Clamping the working element	$R_1 = R_2 = R_3 = R_4 = 0.$

Here, x_0 is the initial coordinate of WE; x_z is the specified coordinate of WE; x_b^i is the coordinate of the beginning of braking in the given trial; x_b^{i+1} is the coordinate of the beginning of braking in the next trial; x_v^i is the coordinate the first stop of the piston in this trial; ε_v is the minimum speed (defined zero); ε is specified positioning error; i is the number of trial; ε_v is the number which determines the convergence of the iterative series (1).

Movement of WE in the phase plane in relation to the control circuit and algorithm in Fig. 3 is shown in Fig. 4. When this strategy is used, the phase plane is divided into four sub-domains, and the switching connections should be

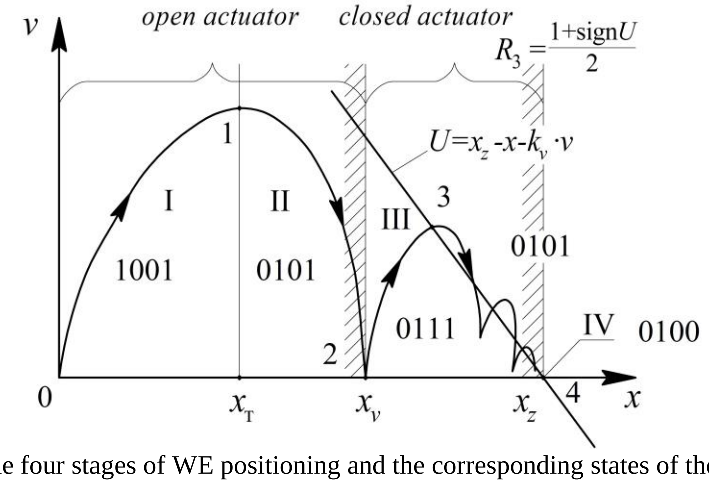


Fig. 4. The four stages of WE positioning and the corresponding states of the solenoids

switched using control valves to move from one subdomain to another. Meanwhile, in sub-domains I and II, the pneumatic actuator remains open and only after the first stop of WE (coordinate x_v) the relay feedback is activated and the switching line is tracked $x_z - x - k_v \cdot v = 0$.

Modern pressure reducing valves from leading manufacturers have a large margin of stability due to the presence of a sub-diaphragm damping chamber and a feedback throttle, and this allows maintaining a stable pressure level in the working cavity over a significant section of the braking distance of the piston.

One of the most important advantages of the pneumatic control circuit in Fig. 3 is that throttle braking is not necessary any more and the operation mode of the pneumatic motor can be controlled only by changing the structure of switching connections. Thus, the most favorable uniformly decreasing braking mode can be created, even with significant inertial loads.

5. Features of functioning of a multi-position PA. Influence of settings

Fig. 5 shows the transition process obtained by numerical integration using the Runge-Kutta method of the 4th order of accuracy of the system of differential equations (2) with the PA control algorithm given above (Table 1). The software was created by the authors independently in the Turbo-Basic environment. Cylinder diameter 50 mm; mass of moving parts $m = 80$ kg; specified positioning coordinate $x_z = 0{,}8$ m at full stroke length $L = 1{,}0$ m; supply pressure $p = 0{,}5$ MPa; pressure setting of the reduction valve $p_v = 0{,}45$ MPa.

The smooth uniformly decreasing braking mode even in the presence of a significant inertia load is worth noting ($m = 80$ kg). This algorithm facilitates WE clamping in the operational mode during the two-stage positioning process.

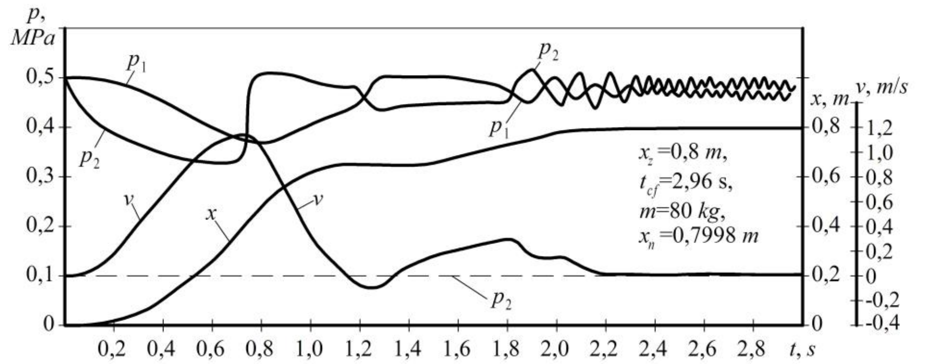
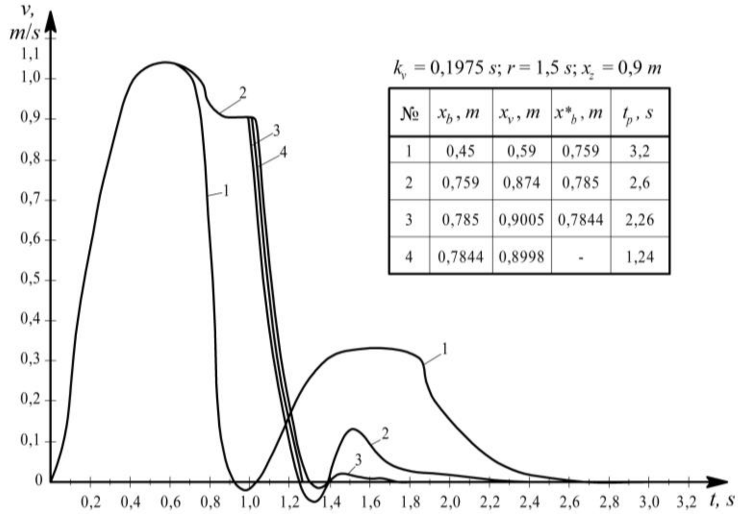


Fig. 5. Positioning WE of PA in the three-stage pre-test mode (option A)

The described control algorithm based on the pneumatic circuit diagram in Fig. 3 provides for effective implementation of a learning program. Fig. 6 shows the graphs of changes in the WE speed during learning, when as a result of four trials, the controller automatically determines the optimal braking coordinate x_b^*.

Number r that determines the convergence of the iteration series (1) is 1.5 and may not change with other combinations of positioning coordinates.

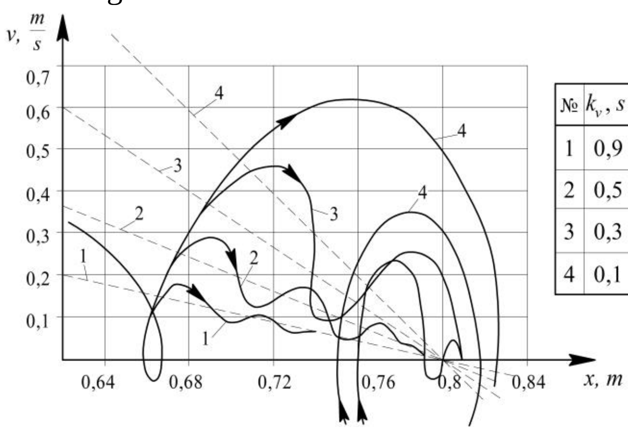
The adjustable parameters are the speed feedback gain ratio k_v and the pressure setting of the pressure reducing valve p_v.



№	x_b, m	x_v, m	x^*_b, m	t_p, s
1	0,45	0,59	0,759	3,2
2	0,759	0,874	0,785	2,6
3	0,785	0,9005	0,7844	2,26
4	0,7844	0,8998	-	1,24

Fig. 6. Determining the optimal braking coordinate as a result of learning

The effect of the speed feedback coefficient k_v on the switching line tracking at the third stage of the positioning process is illustrated by the family of phase trajectories in Fig. 7.



№	k_v, s	t, s
1	0,9	3,38
2	0,5	2,65
3	0,3	1,85
4	0,1	∞

Fig. 7. Phase trajectories of WE in the area of the positioning point at different speed feedback coefficients

The parameters of the PA and the operating conditions are the same as for the transition process in Fig. 5. The larger k_v is, the higher the damping properties of the system are and the closer is the phase trajectory of WE to the switching line. At large values k_v ($k_v = 0{,}9$ c), a so-called “sliding mode” occurs. This mode is characterized by the transformation of the complex dynamic model of PA into the equation of the switching line $x_z - x - k_v \cdot v = 0$.

The disadvantage of this mode is a fairly large number of switchings of the control valve (Fig. 5). However, the reliability and stability of adherence to the switching line makes this mode still the most preferred.

A significant influence on the stability of the switching line is provided by the pressure drop on the piston during the switching line tracking which is determined by the pressure setting of the pressure reducing valve p_v (Fig. 8).

$m = 80\ kg;\ x_b = 0{,}4\ m;\ p_m = 0{,}5\ MPa$

№	p_v, MPa	The time of positioning t_k, s	The actual posit. coordinate x_v, m
1	0,25	3,5	0,79986
2	0,35	2,6	0,79982
3	0,45	3,04	0,79988

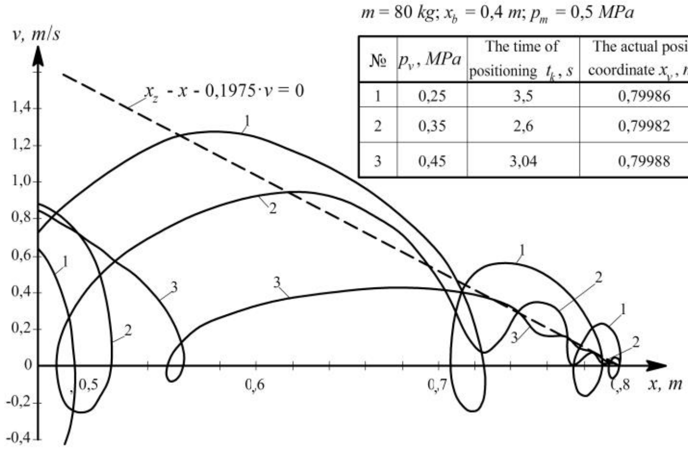


Fig. 8. Positioning in the phase plane at different settings of the pressure reducing valve

As can be seen from the series of phase trajectories shown in Fig. 8, the pressure increase p_v, i.e. the decrease in the pressure drop on WE, leads to stabilization of the positioning process (curve 3). More or less the same effect can be achieved by adding to the switching function U a term proportional to the pressure drop on WE, i.e. by introducing an additional pressure drop feedback $U = x_z - x - k_v \cdot v - k_p \cdot \Delta p.$

6. Conclusions

The proposed control concept and structure of the pneumatic actuator allows for the learning mode to optimize the positioning mode both in terms of accuracy and speed.

The three-phase positioning process during testing allows for maximum response speed during the implementation of the first two phases (drive, brake) due to the absence of feedback effects, and during the switching line tracking at the third phase due to the implementation of a “sliding” mode it provides rigid tracking of the switching line. In the operational mode, only a two-phase positioning mode is used, and the third phase is a kind of “safety” and an adjustment mechanism under variable operating conditions.

It is shown that the positioning mode can be adjusted only by two parameters: the speed feedback gain ratio (k_v) and the pressure setting of the pressure reducing valve (p_v).

The difference between the proposed drive is the ability to operate in conditions of a large inertial load, still providing high performance.

Further research carried out by the authors is aimed at improving the drive control algorithm to overcome the error introduced by the relatively low speed of standard relay pneumatic equipment.

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