Study of non-linear synergy control laws on the experimental stand of pneumatic actuators

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Abstract. The purpose of this work is to synthesize the laws for controlling the position of the piston of a pneumatic cylinder using the synergetic method of analytical design of aggregated controllers (ACAR), which takes into account the nonlinear dynamics of the system under consideration.

The procedure for the synthesis of synergetic control is based on the introduction of a sequence of invariant manifolds, proceeding from the technological problem of positioning the pneumatic cylinder rod at the required position, and the subsequent step-by-step dynamic decomposition of the original dynamic system.

As a result, the synthesized nonlinear two-channel regulator moves the pneumatic actuator rod to a predetermined position while simultaneously stabilizing the pressure in the drain chamber. Simulation of the system with a regulator showed that the control law obtained in analytical form moves the pneumatic actuator rod to a given position while simultaneously stabilizing the pressure in the drain chamber.

The paper deals with practical research that was carried out at the educational and experimental stand of pneumatic drives for vertical and horizontal movement of the Camozzi company.

1. Introduction

The technological relevance of the task of controlling the pneumatic system is associated with the complexity of precise and smooth braking of the pneumatic drive piston in a given coordinate of its trajectory. This is justified by the fact that the working medium of the system is compressed air, which has the physical property of strong compressibility, as well as significant nonlinearity of the thermodynamic processes occurring in the system.

In the last century, the most used in the control of pneumatic systems were industrial regulators that implement typical linear control laws by adjusting the P, I, D components of the regulator. Linear control laws were distinguished by a simple algorithm for the functioning of regulators, a wide range of their use for most applied problems, and economically low costs for implementing control laws. In this case, problems arose associated with the complexity of tuning the differential component of the regulator, an increase in the bandwidth and, as a result, the appearance of high-frequency interference, which affected the operation of the system and the appearance of errors [1, 2].

In work [3], studies of the synthesis of linear typical laws of regulation in a pneumatic servo drive were carried out in sufficient detail and clearly, where the disadvantages of the developed PI-controller are given, which are associated with the appearance of overshoot and the occurrence of low-frequency self-oscillations.

The active development of microprocessor technology in recent decades has led to the improvement of the algorithms for tuning PID regulators. At the moment, there are industrial regulators with automatic
tuning of parameters using adaptive algorithms, neural networks, fuzzy logic methods, as well as genetic algorithms [4].

Various modifications of PID controllers have appeared, for example, a controller that performs the task of tracking control based on Petri nets [5] or a controller with two degrees of freedom, which ensures the independence of the solution of two control problems [6]. In addition, additional functions of the controller have been added, namely: alarm functions, control of the rupture of the control loop, going beyond the boundaries of the dynamic range [7]. However, mostly fuzzy logic algorithms are used in systems that are difficult to formalize and mathematical description [8].

In this paper, it is proposed to consider the synthesis of nonlinear synergistic laws for controlling the position of the pneumatic actuator piston by changing the pressures in the filling and exhaust chambers. The synthesis of control laws is carried out using the method of analytical design of aggregated controllers (ACAR) of synergetic control theory [9-11].

2. Pneumatic system dynamics and mathematical model

Figure 1 shows a closed control loop of a pneumatic system including a pneumatic actuator (4), electro-pneumatic valves (3), a programmable logic controller (2) and a displacement sensor (1). The programmable logic controller issues discrete commands in the form of control voltages to the electro-pneumatic valves, which, in turn, change the cross-sectional area of the hole, while controlling the incoming and outgoing air flows.

A pneumatic valve (PR) with electromagnetic proportional control converts the input electrical signal - voltage $U_i$ on the electromagnet of the i-th PR into the bore area $U_{if}$. Thus, the cross-sectional areas $U_{1f}$ and $U_{2f}$ PR are related to the control voltage by the following relationship [10]:

$$U_{if} = g \cdot U_i,$$

where $g$ is the voltage gain $1.2 \cdot 10^{-6}$ m$^2$/V.

The mathematical model of a complex dynamic system, to which the EPS belongs, represents the relationship between the variables that characterize the behavior of the system, based on the equation of motion of the mechanical part of the device, as well as from the equations characterizing the thermodynamic and gas-dynamic processes occurring in the cylinder cavities [12-14]:

$$\begin{align*}
    x_1(t) &= x_2; \\
    \dot{x}_2(t) &= a_{21} \cdot x_3 - a_{22} \cdot x_4 - a_{23} \cdot x_2 - a_{24}; \\
    \dot{x}_3(t) &= \left[ a_{31} \cdot (x_1 + l_{01})^{-1} \cdot U_{1f} \cdot \sqrt{p_m^2 - x_3^2} \right] - \frac{k x_3 x_2}{(x_1 + l_{01})}; \\
    \dot{x}_4(t) &= - \left[ a_{41} \cdot (a_{42} - x_1)^{-1} \cdot U_{2f} \cdot \left( \frac{x_4}{p_a} \right)^{k-1} \frac{2k}{2k} \cdot \sqrt{x_4^2 - p_m^2} \right] + \frac{k x_4 x_2}{(a_{42} - x_1)}. 
\end{align*}$$

Figure 1. Closed loop control pneumatic system
The following coefficients are introduced into the mathematical model (2) for a simplified representation and further analysis: 

\[ a_{21} = \frac{s_1}{M}; \quad a_{22} = \frac{s_2}{M}; \quad a_{23} = \frac{k_{\text{BP}}}{M}; \quad a_{24} = \frac{p_u(s_1-s_2)}{M}; \quad a_{31} = \frac{k_f p_a}{S_{\text{ch}^2}}. \]

In the model, the following parameters are designated as: \( x_1 \) – coordinate of the piston movement \( l \) (m); \( x_2 \) – speed of movement of moving masses \( V \)(m/s); \( x_3 \) – pressure in the filling chamber \( p_1 \) (Pa); \( x_4 \) – pressure in the exhaust chamber \( p_2 \) (Pa); \( M \) – is the mass of the moving part of the piston and rod (0,5 kg); \( S_1 \) and \( S_2 \) – the effective areas of the piston and rod cavities of the pneumatic cylinder \((8 \cdot 10^{-4} \text{ m}^2 \text{ and } 6 \cdot 10^{-4} \text{ m}^2)\); \( k_{\text{BP}} \) – viscous friction coefficient \((100 \text{ N} \cdot \text{s/m})\); \( T_m \) – absolute gas temperature \((293 \text{ K})\); \( k \) – is the adiabatic index for air \((1,4)\); \( R \) – universal gas constant \((\text{(287 J) } / \text{ (kg} \cdot \text{K})\)); \( l_{01} \) and \( l_{02} \) – initial and final coordinates of the piston position \((0.002 \text{ m})\); \( \xi \) – is the total resistance coefficient of the chokes included in the line \((30)\); \( p_{\text{in}} \) – pressure level at the inlet of the line \((5 \cdot 10^5 \text{ Pa})\); \( p_a \) – atmospheric pressure \((10^5 \text{ Pa})\).

3. Formulation of the control problem and methods of its implementation

One of the initial stages in the synthesis of control laws by the ACAR method is the selection of control objectives - invariants that are set based on the technological problem, taking into account the physical essence of the dynamics of the processes occurring in the system under study [15]. Since the considered electro-pneumatic system (EPS) is described by the equations of the dynamics of the piston motion and the equations of thermodynamic equilibrium, it is necessary to specify technological and thermodynamic invariants that will be the final goal of the state of the system.

According to the control task, pneumatic drives are divided into positional and tracking, thus the ultimate control goal - technological invariants can be: positioning the piston coordinate \( \ell \) to a given position, as well as changing the piston speed \( V \) according to a given time law.

Thermodynamic invariants should reflect the "internal" balance of forces of the system, which will ensure the fulfillment of the final - technological control task. Similar forces in the system under study are: the pressure of compressed air in the filling chamber \( p_1 \), which is essentially the "working" pressure that affects the movement of the piston, as well as the pressure in the exhaust chamber \( p_2 \), a change in which affects the braking of the piston, and therefore its speed movement. Thus, a systemic control model is traced, in which the equilibrium of the system state variables will correspond to certain control objectives.

According to the ACAR method, the number of assigned objective functions must correspond to the number of control channels. The control in the electro-pneumatic system is carried out through two channels:

- control \( U_{1f} \) of the incoming mass air flow, forming the pressure \( p_1 \) in the filling chamber, is carried out by changing the sectional area \( f_1 \) of the PR1 valve;
- control \( U_{2f} \) of the mass flow of air leaving the chamber, which is physically reflected in the form of pressure \( p_2 \), is carried out by changing the cross-sectional area \( f_2 \) of the valve PR2.

In EPS, the created pressure \( p_1 \) in the filling chamber is responsible for the starting and further movement of the piston, and the increase in pressure \( p_2 \) in the exhaust chamber decelerates the piston.

In back pressure control, the piston is braked by increasing the pressure \( p_2 \) by forcing an air flow in the exhaust chamber of the pneumatic cylinder, which is carried out by connecting the pneumatic valve to a source with increased pressure.

Thus, the control task is that it is necessary to synthesize such control actions \( U_{1f} \) and \( U_{2f} \) providing certain pressure ratios \( p_1 \) and \( p_2 \), at which the piston of the pneumatic cylinder makes a smooth stop at a given position \( \ell \).
When the drive stops completely, the pressures in the inlet and outlet chambers will be equal to each other. Thus, during the movement of the stem, the pressure is equalized. It is possible to reflect this alignment by the corresponding invariant manifolds during the synergistic synthesis procedure.

According to the formulated technological problem of positioning the pneumatic cylinder rod at the required position, we introduce the first invariant corresponding to the control goal:

$$x_1 = x_1^*,$$

where $x_1$ is the current stem position, and $x_1^*$ is the required value.

The second invariant of the system is the condition:

$$x_4 = x_3,$$

at which the pressures in the inlet and outlet chambers are equal to each other.

### 4. Synthesis of nonlinear synergistic laws of backpressure control

At the first stage of the synthesis, invariant (4) can be taken into account in the introduced set of invariant manifolds:

$$\psi_1 = x_4 - x_3 = 0,$$

$$\psi_2 = x_3 - \varphi_1(x_1, x_2) = 0$$

In this case, the set of manifolds (5) must satisfy the solution of the system of functional equations:

$$\begin{aligned}
T_1 \dot{\varphi}_1(t) + \psi_1 &= 0, \\
T_2 \dot{\varphi}_2(t) + \psi_2 &= 0,
\end{aligned}$$

When the representing point of the system falls into the neighborhood of the intersection of manifolds (5) in the closed system, dynamic decomposition will occur [10] and the behavior of the system will be described by a reduced system of second-order differential equations:

$$\begin{aligned}
\dot{x}_1(t) &= x_2, \\
\dot{x}_2(t) &= (a_{21} - a_{22}) \cdot \varphi_1(x_1, x_2) - a_{23} \cdot x_2 - a_{24},
\end{aligned}$$

where $\varphi_1(x_1, x_2)$ is the so-called internal control of the decomposed system.

To find the internal control $\varphi_1$, a manifold is introduced, the structure of which corresponds to the invariant (3):

$$\psi_3 = x_2 - k(x_1 - x_1^*) = 0.$$  

For asymptotically stable motion of the system on the attractor to the control goal $x_1 = x_1^*$ invariant manifold (8) must satisfy the solution of the functional equation:

$$T_3 \dot{\varphi}_3(t) + \psi_3 = 0.$$  

The solution of the functional equation (9), taking into account the invariant manifold (8) and the model of the decomposed system (7), determines the structure of the internal control:

$$\varphi_1 = \frac{T_3(a_{23}x_2 + a_{24} + kx_2) - x_2 + k(x_1 - x_1^*)}{T_3(a_{21} - a_{22})}.$$  

Expressions of control actions $U_{1f}$ and $U_{2f}$ are determined from the joint solution (5) and (6) taking into account the found internal control (10) and the mathematical model of the object (2). Thus, the control law for the flow area of the valve on PR1 has the form:

$$U_{1f} = \frac{T_2(\lambda_1 + \lambda_2 + \lambda_3) - x_3 + \lambda_5}{T_2 \lambda_3}, \quad U_{2f} = \frac{T_3(\lambda_6 U_{1f} + \lambda_4 + \lambda_7) - x_4 + x_3}{-T_1 \lambda_6},$$

where

$$\begin{aligned}
\lambda_1 &= -\frac{kx_2}{T_3(a_{22} + a_{21})}; \\
\lambda_2 &= \frac{(T_3 + T_4 a_{23} - 1)(a_{21}x_3 - a_{22}x_2 - a_{23}x_2 - a_{24})}{T_3(a_{22} + a_{21})}; \\
\lambda_3 &= \frac{a_{31} \sqrt{p_0^2 - x_3^2}}{x_1 + l_{01}}; \\
\lambda_4 &= \frac{kx_3x_2}{(x_1 + l_{01})}; \\
\lambda_5 &= \frac{T_3(kx_2 + a_{23}x_2 + a_{24}) + k(x_1 - x_1^*) - x_2}{T_3(a_{21} - a_{22})}.
\end{aligned}$$
5. Modeling a synergistic control system

Figures 2-3 show the results of modeling a system with the obtained nonlinear control laws (11). During the simulation, the following initial conditions of the system were set: \( x_1 = 0 \) m; \( x_2 = 0 \) m/s; \( x_3 = 10^5 \) Pa; \( x_4 = 10^5 \) Pa and the desired values of piston displacement \( x_1^* = 0.1 \) m and pressure in the exhaust chamber \( x_4 = x_3 = 10^5 \) Pa. Controller parameter values: \( T_1 = T_2 = T_3 = 0.001 \) s, \( k = -1 \). The position of the rod \( x_1 \) reaches the desired displacement value in a time of 4 seconds (Fig. 2), which indicates the adequacy of the synthesized control law, as well as its asymptotic stability. The graph of the change in the speed of the rod \( x_2 \) of the pneumatic cylinder shows that the speed, increasing to the maximum at the beginning of the movement of the pneumatic cylinder, decreases, taking on a zero value when the specified displacement is reached. The pressure in the filling chamber \( x_3 \) at the end of the movement cycle coincides with the pressure in the exhaust chamber \( x_4 \) (Fig. 3), which was set when forming the invariant (4).

![Figure 2](image1.png)

**Figure 2.** Graphs of changes in the phase variables of the EPS with coordinated control: 1) piston displacement \((X_1, m)\); 2) piston speed \((X_2, m/s)\)

![Figure 3](image2.png)

**Figure 3.** Graphs of changes in the phase variables of the EPS: 1) change in the pressure in the inlet chamber \((X_3, Pa)\); 2) change in pressure in the exhaust chamber \((X_4, Pa)\)

The analysis of the graphs of the transient processes presented in Figure 4 shows that when the back pressure is controlled, the opening area of the valve that admits the compressed air flow \( U_{1f} \) is larger, this leads to an increase in the working pressure \( X_3 \) (Figure 3). At the same time, due to a smooth decrease in the cross-sectional area of the exhaust valve \( U_{2f} \), where air is supplied with excess pressure, the pressures in the filling chambers \( X_3 \) and exhaust \( X_4 \) are stabilized to the value of the atmosphere (Figure 3).
Figure 4. Graphs of transient processes with coordinated control:
1) cross-sectional area PR1 \( U_{1f}, \ m^2 \); 2) the cross-sectional area of the PR2 \( U_{2f}, \ m^2 \)

Figure 5 shows the voltage graphs for PR1 and PR2. The value of the control voltage \( U_1 \) equal to \( \approx 10 \ V \) corresponds to the full opening of the PR1 port and its connection to the compressed air supply line, this happens in a fraction of a second, then the port is completely closed. A positive voltage value \( U_2 \) corresponds to the connection of the PR2 port with a source of increased pressure, which can be either a power source or a special pneumatic device. In this case, the exhaust chamber is filled with compressed gas to a certain limit.

Figure 5. Graphs of voltages on PR with coordinated control: 1) voltage on PR1 \( (U_1, V) \); 2) voltage across PR2 \( (U_2, V) \)

This back pressure control is often used in pneumatic automation due to the possibility of smooth movement of the stem in real experimental conditions.

6. Experimental study of the obtained control laws on the bench of pneumatic drives of the Camozzi Company

The obtained control laws were empirically tested on the educational and experimental stand of the EPS, a photo of which is shown in Figure 6. Below are the main functional elements of the EPS stand:
- Pneumatic drives of horizontal and vertical movements Camozzi QCT2A032A200 series (1);
- LRWA2-36-2A00 series proportional pneumatic distributors of incoming and outgoing compressed air flows Camozzi (2);
- Programmable logic controller PLC Aries 150;
- MPS-128TSTP0 series magnetic piston position sensor.
This ERS stand is used for educational purposes to study the horizontal and vertical positioning of pneumatic drives, with proportional distributors of compressed air flows that provide braking and stopping of the drive at a given coordinate.

Figure 6. Educational and experimental stand for horizontal and vertical positioning of pneumatic drives

To graphically represent the electrical and pneumatic relationships between the functional elements included in the EPS, a schematic pneumatic diagram of the stand was implemented in the environment of the computer-aided design package "Compass", shown in Figure 7.

Figure 7. Schematic pneumatic diagram of the stand with horizontal positioning of the pneumatic drive
Designing, debugging and loading a program into a PLC is carried out in the CoDeSys industrial automation software development environment in one of five IEC 61131-3 programming languages. In this case, the graphical language of functional block diagrams FBD (Function Block Diagram) was chosen as the programming language, which allows for an accessible and visual representation of the processing and transmission of the control signal from the PLC to the PR, in the form of chains from certain elements-program components (POU).

Figure 8 shows the simultaneous digital tracing of two control signals - voltage U1 on PR1 (green graph) and voltage U2 on PR2 (red graph). The abscissa axes represent the cycles of the controller, while in the emulation mode in the CoDeSys software debugging environment, the PLC cycle is 55 milliseconds. Thus, 500 PLC cycles in terms of the operating time of the controller is 27.5 seconds. The ordinate axes represent the values of the output voltages - the unit is Volt.

As seen from Fig. 8, the controller begins to generate a control signal voltage U1 at about 140 cycle, while the voltage signal U2 at about 150 cycle, which corresponds to a time of 8.25 seconds. This natural time delay of milliseconds is due to the fact that first the PR1 is triggered to the compressed air inlet into the filling chamber, and then the valve on the PR2 opens.

At the same time, the presence of initial controller cycles in which there are no values of control signals is due to the delay time, which includes: the operating system delay when the programming environment is connected to the controller, physical delays in the propagation of signals to the PLC inputs, as well as the delay in reproducing digital tracing in the controller emulation mode.
Judging by the voltage graph U2, (Fig. 5), the time during which the PR2 valve will completely close is 13.75 seconds, while, according to the graph obtained in the course of mathematical modeling in the Maple package, the time for the complete closing of PR2 is 6-8 seconds (Fig. 5). This is due to the mathematical idealization of obtaining analytical data, as well as the lack of accounting, when modeling processes, the time for receipt, processing and issuance of a control signal by the controller.

7. Conclusion
In general, comparing the graphs of control actions - voltages U1 and U2, obtained using the synergistic ACAR method and the graphs of signals of discrete outputs of the controller in Fig. 8, which are control actions on the valves PR1 and PR2, one can definitely state the identity of these graphs, which indicates the high accuracy of the analytically synthesized nonlinear synergistic control laws.

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