Simulation of pneumatic actuator position system for long stroke mounting movements

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Abstract. The creation of new equipment and technical means for the implementation of promising processes and technologies is largely determined by the level of development of mechanical engineering. The most important requirements for technological equipment are an increase in work processes, flexibility and adaptability, automation, implementation of complex algorithms of executive movements. The mechanisms of executive movements ensure the operation of transporting, fixing and other target mechanisms, this largely determines the quality and efficiency of processing processes. Positional actuators have existing technical solutions. However, to a greater extent they relate to linear movements realized by rod cylinders, this does not allow adapting the results of existing research to the development of long-stroke positional movements. In the course of this work, an assessment was made of the existing types of drives and a displacement and position measurement sensors. The proposed positional pneumatic scheme allows solving the problem of positioning long-stroke displacements, its mathematical and computer models were built, and dynamic analysis and conclusions were made.

1. Assessment of the state of the issue
In order to achieve the linear movement of the output link of the technological equipment, pneumatic, electromagnetic and hydraulic drives are usually used. The latter two types of engine are more widespread. An electromagnetic drive, on the other hand, is clean and reliable to operate, but often requires a mechanical transmission, the combination of which can be quite expensive.

Pneumatic units have a number of advantages: they are fast, cheap, have outstanding power per unit weight, and are easy to maintain. A big problem when using a pneumatic cylinder is the presence of piston friction and non-linear characteristics of the compressed gas flow. Therefore, with long-stroke movements, there is a deterioration in the dynamics and positioning of the drive [1-3].

The use of existing sensors for measuring displacement and position is not always possible in industries where special attention is paid to fire and explosion safety. The use of electrical elements is not permitted there. The use of sensors with long travel distances can affect their cost or require feedback, which can also increase the cost of the drive.

2. Development of a pneumatic positioning scheme
Positional pneumatic actuator consists of: air preparation device (APD); rodless pneumatic cylinder (PC1); short-running pneumatic cylinders (PC2, PC3); tandem pneumatic cylinder (PC4); nozzle apparatus (N) and rails; throttle (TH1, TH2), which are used to regulate the speed of movement of the cylinder piston; two-position two-way valves with electro-pneumatic control (PV1, PV2, PV4); three-position five-line distributor (PV3), mufflers (S1-S5).
3. Description of the principle of operation
The operator sets the required stop coordinate in the PLC, rapid traverse occurs. The pressure enters the nozzle N, then the pressure sensor PS. When the drive reaches the previously set coordinate, electromagnets Y1, Y2 are turned on. The cylinder rod PC4 switches the PV3 valve, the drive travel slows down. When the drive reaches the coordinate set by the operator through the N nozzle through the aligned hole, the pressure is supplied to the control line of the PV4 distributor and the PC4 tandem cylinder, and it switches the PV4, PV3 valves and blocks the inlet and outlet cavities, and activates the blocking brake.

4. Formation of a generalized mathematical model
The mathematical model of the pneumatic drive demonstrates a system of differential equations describing the movement of the working body and the change in pressure in the cavities of the actuator, the mathematical model of the pneumatic drive includes the following equations:
1. The equation of motion of the actuator of the pneumatic cylinder.
2. Equations of pressure change in the discharge cavity.
3. Equations of pressure change in the exhaust cavity.
The design diagram of the pneumatic drive is shown in Figure 2.

**Figure 1.** Diagram of pneumatic positioning
When designing, one of the main conditions is to confirm the functionality of the developed drive, as well as to analyze the processes occurring in it when positioning the pneumatic cylinder.

When forming the mathematical model of the drive, the following assumptions were made:

- the pressure of the compressed air source is constant over time;
- the thermodynamic process of gas behavior in the pneumatic system is assumed to be adiabatic;
- in the description of pneumatic devices, the ideal gas model is used, since the pressure in the pneumatic system is below 10 bar;
- leaks are not taken into account;
- the force of viscous friction is proportional to the speed;
- the coefficients of expenses are taken as averaged;
- the mass of the moved parts is assumed constant;
- force \( F_c \) at the output link of the pneumatic drive is constant;
- relay control of pneumatic valves;
- the time of forming the control signal from the displacement sensor is not taken into account [4-5].

During the calculation of the pneumatic actuator, the initial data for the modeling were calculated and presented in Table 1.

![Design diagram of a pneumatic drive](image-url)
Table 1. Initial data for modeling

| Name                              | Identification | Mean. | Size |
|-----------------------------------|----------------|-------|------|
| Drag coefficient                  | ks1            | 65    | m    |
|                                  | ks2            | 100   |      |
|                                  | ks3            | 70    |      |
|                                  | ks4            | 70    |      |
| Flow path diameter                | d1             | 0.006 | m    |
|                                  | d2             | 0.006 |      |
|                                  | d3             | 0.001 |      |
|                                  | d4             | 0.0012|      |
| Compressor pressure               | pm             | 600,000| Pa  |
| Atmospheric pressure              | pa             | 101325| Pa  |
| Adiabatic coefficient            | to             | 1.4   |      |
| Coordinates given                | x01, x02      | 0.008 | m    |
| Mass of moving parts of PC       | m              | 6     | kg   |
| Strength of external resistance   | Fvn            | 20    | N    |
| Pipeline diameter                | dtr            | 0.005 | m    |
| Pneumatic cylinder 1 piston diameter | D          | 0.04  | m    |
| Pneumatic cylinder 2,3 piston diameter | Dt        | 0.04  | m    |
| Viscous friction coefficient     | kvt            | 400   | N s/ m|
| Piston stroke                    | L              | 0.4   | m    |
| Friction coefficient             | kt             | 0.4   |      |

1. The equation of motion of the actuator of the pneumatic cylinder

\[ m \frac{d^2 x}{dt^2} = S \cdot (p_1 - p_2) - F_{bn} \cdot \text{sign} \left( \frac{dx}{dt} \right) - k_{vt} \cdot \left( \frac{dx}{dt} \right) - F_t \cdot \text{sign} \left( \frac{dx}{dt} \right) \]  

(1)

2. Equation of motion of the actuator of the pneumatic brake cylinder

\[ m \frac{d^2 x_t}{dt^2} = S \cdot p_m - F_{bn T} \cdot \text{sign} \left( \frac{dx_t}{dt} \right) - k_{vt T} \cdot \left( \frac{dx_t}{dt} \right) - c_{pr T} \cdot (x_0 + x_t) \]  

(2)

where \( S \) is the effective area of the cavities, m²;
\( p_1, p_2 \) - air pressure, respectively, in the discharge and exhaust cavities of the cylinders, Pa;
\( \frac{dx}{dt} \) - speed of movement of moving masses, m / s;
\( F_{bn} \) - the force of external forces, N;
\( k_{vt} \) - coefficient of viscous friction;
\( m \) - mass of moving parts of the drive, kg.
\( F_t \) - braking force, N;

\[ F_t = \mu \cdot N \]

(3)

\[ N = \frac{P_m \cdot S_T}{n} \]

(4)

where \( \mu \) is the coefficient of friction;
\( N \) - normal force, N;
\( P_m \) - supply pressure, Pa;
\( S_T \) - the area of the piston cavity in the pneumatic brake, m²

3. Equation of motion of the valve actuator 1

\[ m_x \frac{d^2 x_1}{dt^2} = S_{p1} \cdot (p_3 - p_y) - c_{np \cdot p1} \cdot x_{z1} - F_{c1} \cdot \text{sign} \left( \frac{dx_1}{dt} \right) - F_{y1} \cdot \left( \frac{dx_1}{dt} \right) - P_y \]

(5)

4. Equation of motion of the valve actuator 2

\[ m_x \frac{d^2 x_2}{dt^2} = S_{p2} \cdot (p_m - p_1) - c_{np \cdot p2} \cdot x_{z2} - F_{c2} \cdot \text{sign} \left( \frac{dx_2}{dt} \right) - F_{y2} \cdot \left( \frac{dx_2}{dt} \right) - F_{em2} \]

(6)

5. Equation of motion of the valve actuator 3

\[ m_x \frac{d^2 x_3}{dt^2} = S_{p3} \cdot (p_2 - p_A) - c_{np \cdot p3} \cdot x_{z3} - F_{c3} \cdot \text{sign} \left( \frac{dx_3}{dt} \right) - F_{y3} \cdot \left( \frac{dx_3}{dt} \right) - F_{em3} \]

(7)
6. Equation of motion of the valve actuator 4

\[ m_2 p_4 \frac{dx_4}{dt^2} = S_p \cdot (p_b - p_t) - c_{hp} p_4 \cdot x_4 - F_c \cdot \text{sign} \left( \frac{dx_4}{dt} \right) - F_y \cdot \left( \frac{dx_4}{dt} \right) - P_y \]  

(8)

Where \( S_p \) is the effective area of the valve end of the distributor, m²;
\( P_b, P_t, P_y \) –Pressure at the inlet and outlet of the distributor, Pa;
\( F_c \) - forces of resistance to movement of the distributor spool, N;
\( F_y \) - thrust reaction forces, N;
\( c_{hp} \) - force of action of the electromagnet on the valve spool, N;
\( m_2 p \) - mass of the distributor spool, kg.

Next, we transform equations (13), (19), (20), we

perform the work of the drive

\[ dL = q \cdot dm, \]  

(10)

The specific energy of a gas is determined by its body content - enthalpy. Enthalpy is related to the specific and temperature of the gas in the pipeline by the dependence:

\[ q = l = c_p \cdot T_M \]  

(11)

Elementary mass of gas dm express it in terms of expense \( G_M \):

\[ dm = G_M \cdot dt \]  

(12)

Substituting (11) and (12) into equation (10), we find the amount of energy that enters the piston cavity

\[ dQ_M = c_p \cdot T_M \cdot G_M \cdot dt \]  

(13)

After the transformation, we obtain the equation for the change in the internal energy of the gas:

\[ dU_1 = d(c_v \cdot T_1 \cdot m_1) = c_v \cdot d(T_1 \cdot m_1) \]  

(14)

Where \( c_v \) – specific heat capacity of a gas at a constant volume;
\( T_1 \) – gas temperature in the discharge cavity;
\( m_1 \) – mass of gas in the discharge cavity.

The mass of gas is determined in the size and volume of the cavity \( \rho_1 V_1 \):

\[ m_1 = \rho_1 V_1 \]  

(15)

Next, we substitute (15) into formula (14) and obtain an expression for determining the change in the internal energy of the gas \( dU_1 \) in the cavity:

\[ dU_1 = d(c_v \cdot T_1 \cdot m_1) = c_v \cdot d(T_1 \cdot m_1) \]  

(16)

Next, we transform the equation (10)

According to the Mendeleev - Clapeyron equation \( \frac{p_1}{p_1'} = R \cdot T_1 \)

\[ T_1 \cdot \rho_1 = R \cdot T_1' \]  

(17)

Then, substituting (17) into formula (16), we transform the equation:

\[ dU = \frac{c_v \cdot d(V_1)}{R} \]  

(18)

After differentiation, we get the final expression

\[ dU = \frac{c_v \cdot p_1 \cdot dV_1}{R} + \frac{c_v \cdot V_1 \cdot dp_1}{R} \]  

(19)

Next, we get the work overcome by the gas:

\[ dL_1 = p_1 \cdot dV_1 \]  

(20)

Next, we transform equations (13), (19), (20), we obtain

\[ c_v \cdot T_M \cdot G_M \cdot dt = \frac{c_v}{R} \cdot p_1 \cdot dV_1 + \frac{c_v}{R} \cdot V_1 \cdot dp_1 + p_1 \cdot dV_1 \]

After that, we transform the expression and get:
\[ c_p \cdot T_M \cdot G_M \cdot dt = p_1 \cdot \left( \frac{c_v}{R + 1} \right) \cdot dV_1 + \frac{c_v}{R \cdot V_1} \cdot dp_1 \] (21)

At constant pressure and constant volume, the specific heat capacities of the gas are interconnected by dependencies:

\[ c_v + R = c_p \] (22)

Substituting equation (18) into (17) we get:

\[ c_p \cdot T_M \cdot G_M \cdot dt = \frac{c_p}{R \cdot p_1} \cdot dV_1 + \frac{c_v}{R \cdot V_1} \cdot dp_1 \] (23)

To simplify the expression, the adiabatic exponent for air, we multiply the entire equation by R and divide by, as a result we get:

\[ \frac{c_p}{c_v} = k = 1.4 \]

\[ k \cdot R \cdot T_M \cdot G_M \cdot dt = k \cdot p_1 \cdot dV_1 + V_1 \cdot dp_1 \] (23)

Let us find expressions for the components of this equation. The volume of the discharge cavity \( V_1 \) consists of the working (variable) volume of the working cavity \( V_{1p} \) of the pneumatic cylinder and the initial (constant) volume \( V_{01} \) of the pneumatic drive:

\[ V_1 = V_{1p} + V_{01} \] (24)

The working volume of the discharge cavity \( V_{1p} \) is expressed through the area of the piston \( S_1 \) in the discharge cavity and the coordinate \( x \):

\[ V_{1p} = S_1 \cdot x \] (25)

The initial volume \( V_{01} \) includes the structural volume of the discharge cavity in the extreme position of the piston and the volume of the supply line, which consists of the volumes of the pipeline and the connected pneumatic equipment.

The initial volume \( V_{01} \) of the supply line can be written, as is customary in some works [6-8], as follows:

\[ V_{01} = S_1 \cdot x_{01} \] (26)

where \( x_{01} \) is the reduced initial coordinate of the piston position.

Finally, substituting (25) and (26) into equation (24), we find the volume of the injection cavity \( V_1 \):

\[ V_1 = S_1 (x + x_{01}) \] (27)

Gas flow in pneumatic industrial drives. Coming into the discharge cavity depends on the nature of the process.

In pneumatic industrial drives, the gas flow is close to the isothermal process [7-9]. Therefore, the gas flow rate is determined from the equation which we write in the following form, bearing in mind that \( \sigma = \frac{p}{p_m} \)

\[ G_m = K_1 \cdot \sqrt{p_m^2 - p_1^2}, \] (28)

where

\[ K_1 = \frac{f_1}{\sqrt{R \cdot T_M} \cdot \xi_1} \] (29)

Substituting the values and from (27) and (28) into the original equation, after transforming the equation (9), we obtain the equation for the change in pressure in the pressure cavity and emptying the pneumatic cylinder (30):

\[ \frac{dp_1}{dt} = k \cdot f_1 \cdot \frac{\sqrt{R \cdot T_M}}{\sqrt{\xi_1}} \cdot \sqrt{p_m^2 - p_1^2} \cdot \sqrt{\frac{p_2^2}{p_1}} \cdot \left( \frac{p_1}{p_2} \right)^{k-1} \cdot \sqrt{p_1^2 - p_3^2} \cdot \frac{k \cdot p_1}{x + x_{01}} \cdot dx \] (30)

where is the adiabatic index; \( k \)

\( R \) - gas constant, \( 1 / \text{kg} \cdot \text{K} \);
7. Equation of pressure in the exhaust cavity.

\[-dQ_2 = dU_2 + dL_2; \quad (31)\]

Similarly, we derive the equation in the drain cavity, the amount of energy leaving the exhaust cavity:

\[dQ_2 = c_p \cdot T_2 \cdot dm_2 = c_p \cdot T_2 \cdot G_2 \cdot dm_2 \quad (32)\]

After the transformation, we obtain the equation for the change in the internal energy of the gas:

\[dU_2 = d(c_v \cdot T_2 \cdot m_2) = c_v \cdot d(T_2 \cdot \rho_1 \cdot V_2) \quad (33)\]

According to the Mendeleev-Clapeyron equation

\[\frac{p_2}{T_2} = \frac{\rho_2}{\rho_2} = \frac{R \cdot T_2}{G_2} \quad (34)\]

Then, substituting (32) into formula (31), we transform the equation:

\[dU = \frac{c_v}{R} \cdot d \left( \frac{p_2 \cdot V_2}{R} \right) = \frac{c_v}{R \cdot p_2} \cdot dV_2 + \frac{c_v}{R \cdot V_2} \cdot dp_2 \quad (35)\]

Next, we get the work done by the gas in the exhaust cavity:

\[dL_2 = p_2 \cdot dV_2 \quad (36)\]

Next, we transform equations (32), (33), (34) into equation (31), we obtain

\[-k \cdot R \cdot T_2 \cdot G_2 \cdot dt = k \cdot p_2 \cdot dV_2 + V_2 \cdot dp_2; \quad (37)\]

The volume of gas in the exhaust cavity, taking into account the volume of the outlet pipeline and pneumatic equipment, is:

\[V_2 = (S + x_2 - x) \cdot F_2 \quad (38)\]

The outflow of gas from the exhaust cavity through the pipeline is close to the isothermal process. Therefore, the flow rate is determined from the simplified equation corresponding to the isothermal gas outflow:

\[G_2 = K_2 \cdot \sqrt{p_2^2 - p_a^2} \quad (39)\]

Where

\[K_2 = \frac{f_2}{\sqrt{R \cdot T_2 \cdot \xi_2}} \quad (40)\]

Substituting the values \(V_2\) and \(G_2\) from (38) and (39) into equation (37), we obtain the equation for the pressure change in the exhaust cavity:

\[-\frac{dp_2}{dt} = -\frac{k \cdot f_2 \cdot \sqrt{R \cdot T_M}}{s(L-x+x_0)\sqrt{\xi_2}} \cdot \left(\frac{p_2}{p_a}\right)^{k-1} \cdot \sqrt{p_2^2 - p_a^2} + \frac{k \cdot p_2}{L-x+x_0} \cdot \frac{dx}{dt} \quad (41)\]

where is the adiabatic index; \(R\) - gas constant, \(J / kg \cdot K\);
\(T_M, T\) - air temperature in the exhaust line, K;
\(P_2\) - pressure in the exhaust line, Pa;
\(P_a\) - atmospheric pressure, Pa;
L - maximum piston stroke, m;
f₂ - flow area of the drain pipeline, m²;
L - piston stroke, m;
\(x_{02}\) - the ratio of the initial ("passive") volume \(V_{02}\) of the pneumatic drive to the useful area of the rod cavity of the pneumatic cylinder, m;
\(\frac{dx}{dt}\) - piston movement speed, m/s;
\(\zeta_2\) - the coefficient of resistance of the exhaust line.

5. Equation of pressure change in the sensor.

\[
\frac{dp_3}{dt} = \frac{k_f^3 \cdot R \cdot T_M}{W_1 \cdot \sqrt{\zeta_3}} \cdot \sqrt{p_1^2 - p_a^2} - \frac{k_f^4 \cdot R \cdot T_M}{W_2 \cdot \sqrt{\zeta_4}} \cdot \left(\frac{p_3}{p_a}\right)^{\frac{k-1}{2k}} \cdot \sqrt{p_3^2 - p_a^2} \tag{42}
\]

where is the adiabatic index; \(R\) - gas constant, J/kg·K;
- \(T_w\) - air temperature in the discharge line, K;
- \(P_2\) - atmospheric pressure, Pa;
- \(P_1, P_3\) - pressure in the flowing part of the pipeline, Pa;
- \(W_1, W_2\) - the volume of the flow parts, m³/s;
- \(\zeta_3, \zeta_4\) - the coefficient of resistance of the supply and exhaust lines.

\(f_3, f_4\) - pipeline section area, m²;

We obtain the system of equations of the mathematical model of this pneumatic system:

\[
\begin{align*}
m \frac{dx}{dt} &= S \cdot (p_1 - p_2) - F_{in} \cdot sign\left(\frac{dx}{dt}\right) - F_{w} \cdot sign\left(\frac{dx}{dt}\right)
m \frac{dx_1}{dt^2} &= S_p \cdot (p_3 - p_y) - c_{np} \cdot x_1 - F_{c1} \cdot sign\left(\frac{dx_1}{dt}\right) - F_{y1} \cdot \frac{dx_1}{dt} - P_y \\
m \frac{dx_2}{dt^2} &= S_p \cdot (p_m - p_2) - c_{np2} \cdot x_2 - F_{c2} \cdot sign\left(\frac{dx_2}{dt}\right) - F_{y2} \cdot \frac{dx_2}{dt} - F_{w2} \\
m \frac{dx_3}{dt^2} &= S_p \cdot (p_m - p_1) - c_{np3} \cdot x_3 - F_{c3} \cdot sign\left(\frac{dx_3}{dt}\right) - F_{y3} \cdot \frac{dx_3}{dt} - F_{w3} \\
m \frac{dx_4}{dt^2} &= S_p \cdot (p_m - p_3) - c_{np4} \cdot x_4 - F_{c4} \cdot sign\left(\frac{dx_4}{dt}\right) - F_{y4} \cdot \frac{dx_4}{dt} - P_y \\
\end{align*}
\tag{43}
\]

The system of differential equations can be solved by various numerical methods (Euler, Runge-Kutta, etc.) for given initial conditions, drive parameters and control actions that functionally depend on the coordinate of the output link [10-13]. The study of the model was carried out in the SimIntech program by numerical methods.

5. Computer model

The computer model is shown in Figure 3.
Figure 3. Block diagram of the mathematical model

input p1, p2, p3, V, x, f4, ds;
output p1, p2, p3, V, x, d;
init p1=100000, p2=100000, p3=100000, V=0, x=0;
x0=0.31; //The target coordinate
R=287; //Universal gas constant
Tm=290; //Ambient temperature
ks1=65; //Drag coefficient
ks2=100; //Drag coefficient
ks3=70; //Drag coefficient
ks4=70; //Drag coefficient
d3=0.001; //Diameter of the passage section
d5=0.006; //Diameter of the passage section
pm=6*10^5; //Compressor pressure
pa=1.01325*10^5; //The pressure of the atmosphere
k=1.4; //The ratio of specific heats
x01=0.008; //The given coordinate
x02=0.008; //The given coordinate
m=6; //Weight of moving parts of the pneumatic drive
Fv=20; //The force of external resistance
dtr1=0.005; //Pipeline diameter
dtr2=0.005; //Pipeline diameter
d=0.84; //Cylinder piston diameter
dt=0.04; //Brake piston diameter
kvt=320; //Diameter of the brake device piston
Figure 4. Computer simulation of automatic positioning pneumatic actuator system (1)

Figure 5. Computer simulation of automatic positioning pneumatic actuator system (2)
The results obtained confirm the functionality of the proposed drive and make it possible to evaluate the processes occurring during the positioning of the pneumatic cylinder piston. This drive model makes it possible to conduct a computational experiment for various positional cycles. The operator in the program sets positioning coordinates. The computer model can be modernized and corrected in order to introduce other control actions in other parameters after the experiment.

6. Discussion
The generalized mathematical model made it possible to analyze the functionality and characteristics of the developed positional actuator, to highlight and determine the assumptions necessary for further design. The developed computer model made it possible to evaluate the transient processes occurring in the drive. As a result of the computational experiment, the efficiency of the proposed technical solution was confirmed and the functionality of the positional pneumatic drive was determined.
Oscillographic analysis of the positioning process has established that the dynamic and control characteristics of the proposed drive are significantly improved when using a pneumatic brake. The drive run-out is significantly reduced, while providing a reliable fixation of the mechanism, which maintains accuracy under subsequent external influences [14-15]. The speed of movement of the pneumatic cylinder has a decisive influence on the processes of deceleration, stopping and positioning of the control object [16-17]. So for a better stop, a deceleration was added at the penultimate coordinate.

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