Combined pneumatic drive of the rotary-dividing mechanism of the revolver head

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Abstract. Currently, the use of a positional combined drive in metal-cutting equipment must have a high-precision ability to process new materials, the design of workpieces and parts; ensure the technical and environmental safety of personnel, etc. All these requirements must be met by CNC machines to produce competitive products in the market criteria. Metal cutting equipment due to its high cost has led to the need to use a single power source for multiple work movements. To increase the productivity of the machine, it is necessary to use a combined two-motor positional pneumatic drive. This increases the total production time. Reduce the main time (by automating two movements: rapid supply of the workpiece to the processing center, and reduce the auxiliary time (automating the installation of the workpiece and removing the part, reducing the path of movement of the tool), reduce the time for equipment changeover (using digital display and software control). The section "research results" presents the results of mathematical modeling based on a computer model implemented in the SimInTech software package. The graphs of pressure dependences in the piston and rod cavities of the pneumatic cylinder, speed and displacement of the pneumatic cylinder are obtained.

1. Introduction

The relevance of research
Multi-tool turrets are mainly equipped with an electric drive for performing pivoting and pivoting movements. But there are ways where it is not only possible to use, but also a pneumatic drive. For this, the actuator uses the principle of double positioning: pre-positioning, second precision positioning;

The novelty is as follows:
1. Variable drive structure during movement. (modular principle of variation of submodels in one working cycle, algorithms of block diagrams of the research model)
2. The complex dynamics of the entire pneumatic-mechanical system, because mutual influence of drives: pneumatic cylinders PC1, PC2 and pneumatic accumulator PA [1-5].
3. Double positioning
I pre-positioning $\Delta \phi \leq \left[ 8 - 12 \right]$;
II precision positioning $\Delta \phi \leq \left[ 5 - 8 \right]$
$\Delta \phi_1 \downarrow$ due to double-sided stop;
$\Delta \phi_1 \rightarrow \Delta \phi_2$
Reaching the goal: $T_c \rightarrow \min$

$$T_c = T_{rel} + T_r + T_{clam} + T_{clam}$$

(1)
$T_c$

$T_{rel}$ - Full cycle time of the drive, s;

$T_r$ - The time of the release actuator; s;

$T_{clam}$ - The time of rotation of the drive, s;

- Drive clamping time, s;

$T_{loc}$ - Drive locking time, s;

**Figure 1.** Pneumatic kinematic diagram of a combined positional pneumatic actuator

PL – pneumatic lock, PV1-PV6 – pneumatic valve, TH1 – throttle, PVA – pressure valve, AC – accumulator, PC1 – PC2 – Pneumatic cylinder, PLC – programmable logic controller, MPS – Mechanopneumatic sensor, YA1 – YA7 - Electromagnet. PM1 – Pressure meter, S – silencer, CV1 – CV2 – Check Valve;
Figure 2. Design diagram of a positional pneumatic actuator

Mechanical subsystem:

Equation of motion of a linear motion positional pneumatic actuator

\[ m_{pr} \ddot{L} = (P_1 \cdot S_1 - P_2 \cdot S_2) - F_{V.f} - F_{d.f} \cdot \text{sign} \left( \frac{dL}{dt} \right) - \alpha \cdot F_{l.f} ; \]  

\[ I_{pc1} \ddot{\varphi} = S_3 \cdot (P_3 - P_4) - M_{f,d} - M_Y + \alpha \cdot M_f ; \]

- \( m_{pr} \) - moving weight of the piston, kg;
- \( S_1, S_2 \) - effective area of the discharge and exhaust cavities, \( m^2 \);
- \( P_1, P_2 \) - air pressure, respectively, in the discharge and exhaust cavities of the cylinder PC 1, Pa;
- \( P_3, P_4 \) - air pressure, respectively, in the discharge and exhaust cavities of the cylinder PC 2, Pa;

\[ m_{pr} = m_{pc1} + m_v + m_{pis} ; \]

- \( m_v \) - rotating mass, kg;
\( m_{pis} \)- the mass of the piston, kg;

\( m_{pc1} \)- workpiece weight, kg;

\( L \)- stroke of the PC1 piston (mechanical closure coupling), m;

\[ F_c = F_{v.f} + F_{d.f} \] (5)

\( F_{v.f} \)- viscous friction force, N;

\( F_{d.f} \)- dry friction force, N;

\( F_{l.f} \)- Locking force, N;

\[ F_{l.f} = P_2 \cdot S_2 \cdot N; \] (6)

\( \alpha = 0 \) by \( L \leq L_{tz} \) and \( \alpha = 1 \) \( L \geq L_{tz} \);

\( L \)- moving the power pneumatic cylinder rod, m;

\( L_{tz} \)- the target coordinate of the stop, m;

\[ F_{v.f} = k_{v.f} \cdot V; \] (7)

\( k_{v.f} \)- coefficient of viscous friction, \( N \cdot s \cdot m^{-1} \);

\( V \)- the speed of movement of the stock, m/s;

\[ I_{rd} = I_{pc1} + I_{pc2} + I_c; \] (8)

\( I_c \)- moment of inertia of the control device, \( kg \cdot m^2 \);

\( I_{rd} \)- Reduced moment of inertia, \( kg \cdot m^2 \);

\( \alpha \)- a Boolean parameter

\( I_{pc1} \)- moment of inertia of the pneumatic cylinder 1, \( kg \cdot m^2 \);

\( I_{pc2} \)- moment of inertia of the pneumatic cylinder 2, \( kg \cdot m^2 \);

\( \varphi \)- angular displacement, rad/s;

\( R \)- Radius, m;

\[ I_c = I_{\Delta T} \cdot i^2 \] (9)

\( i = 1 / 3 \);

\( i \)- the ratio of gear set;

\( M_y \)- moment on the stop, N\( \cdot \)m;

\( M_{f.d} \)- moment of force of dry friction, N\( \cdot \)m;

\( M_f \)- locking torque;

\[ M_{f.d} = k_{v.f} \cdot \frac{d\varphi}{dt}; \] (10)

\[ M_f = F_0 \cdot R; \] (11)

\[ F_0 = F_f \cdot \tan \alpha; \] (12)
$F_0$ - drive stop power;

The kinematic equation of balance of mass flows on the pressure line of a pneumatic actuator

$$G_1 = G_{PC1} + G_{pis} + G_{1comp} + G_{PMS};$$

(13)

$$G_1 = \mu_1 \cdot f_{d1} \cdot p_p \cdot \left[ \frac{2k}{R \cdot T_p \cdot (k-1)} \right] \cdot \varphi (\sigma_1);$$

(14)

$$G_{IPC} = \frac{W_1 \cdot dp_1}{R \cdot T_p};$$

(15)

$$G_{pis1} = k_n \cdot (p_1 - p_2) \cdot \text{sign}(p_1 - p_2);$$

(16)

$$G_{1comp} = \frac{W_p}{E} \cdot \frac{p_1 \cdot gW_p}{R \cdot T_p};$$

(17)

$$G_{PMS} = P_m \cdot \frac{\pi \cdot d^2}{4};$$

(18)

$G_{PC1}$ - flow through the pressure line distributor, $kg \cdot s^{-1}$;

$G_{pis1}$ - flow rate in the Rodless cavity of the power pneumatic cylinder, $kg \cdot s^{-1}$;

$G_{1comp}$ - consumption caused by leaks in the piston, $kg \cdot s^{-1}$;

$k_n$ - the ratio of partecie;

$G_{PMS}$ - flow rate flowing through the pneumatic-mechanical sensor, $kg \cdot s^{-1}$;

$L$ - universal gas constant;

$K$ - ratio of specific heats (for air $k = 1.4$);

$E$ - air volume elasticity modulus, $Pa$;

$W_1$ - current volume of the piston cavity of the power pneumatic cylinder, $m^3$;

$W_p$ - current volume in the pressure line of the pneumatic actuator, $m^3$;

$T_p$ - air temperature in the discharge line, $K$;

$p_h$, $p_1$, $p_2$ - supply pressure, in the piston cavity of the power pneumatic cylinder, in the rod cavity of the power pneumatic cylinder, $PA$.

After converting the formulas, the equation will take the following form:

$$\frac{dp_2}{dt} = \frac{k_n \cdot (p_1 - p_2) \cdot \text{sign}(p_1 - p_2) - P_y \cdot \frac{\pi \cdot d^2}{4}}{k \cdot T_p} \cdot \frac{dp_2}{dt} = \frac{k_n \cdot (p_1 - p_2) \cdot \text{sign}(p_1 - p_2) - P_y \cdot \frac{\pi \cdot d^2}{4}}{k \cdot T_p};$$

(19)

Where $L_{11}$ is the ratio of the initial ("passive") volumes $W_n$ of the pneumatic actuator to the useful area of the pneumatic cylinder piston, respectively, $m$.

The current volume of air in the pressure line of the pneumatic actuator is determined by:

$$W_p = S \cdot L;$$

(20)

Where $L$ is the length of the pressure pipeline, $m$; $S$ is the cross-section area of the pipeline, $m^2$;

Equation of mass flow balance of the pneumatic drive drain line

$$G_2 = -G_{pc2} + G_y + G_{2comp} + \beta \cdot G_p + G_{tr};$$

(21)

$$G_2 = \mu_2 \cdot f_{d2} \cdot p_2 \cdot \left[ \frac{2k}{R \cdot T_2 \cdot (k-1)} \right] \cdot \varphi (\sigma_2);$$

(22)

$$G_{2pc} = \frac{W_{2pc} \cdot dp_2}{R \cdot T_2};$$

(23)

$$G_{y} = k_y \cdot p_2;$$

(24)

$$G_{2comp} = \frac{W_{2comp} \cdot dW_{2}}{R \cdot T_2};$$

(25)

$$G_{p_1} = \mu_{p1} \cdot f_{p1} \cdot p_1 \cdot \left[ \frac{2k}{R \cdot T_2 \cdot (k-1)} \right] \cdot \varphi (\sigma_2);$$

(26)

$$G_{tr2} = \mu_{tr2} \cdot f_{tr2} \cdot p_2 \cdot \left[ \frac{2k}{R \cdot T_2 \cdot (k-1)} \right] \cdot \varphi (\sigma_2);$$

(27)

Where $G_2$ - drain line flow rate $kg \cdot s^{-1}$; $G_{2pc}$ - flow rate in the rod cavity of the pneumatic cylinder $kg \cdot s^{-1}$; $G_y$ - flow rate of working air leakage into the atmosphere $kg \cdot s^{-1}$; $G_{2comp}$ - compressibility flow rate in the drain line $kg \cdot s^{-1}$; $G_{p_1}$ - flow through the pneumatic distributor $P_1$; $G_{2pc}$- flow rate via
pneumatic throttle, kg; \( \mu_2, \mu_p, \mu_{tr2} \) - flow coefficients for the rod cavity of the power pneumatic cylinder, pneumatic distributor P1, and pneumatic throttle DDR2, respectively; \( \beta \) - the Boolean parameter; (with \( \beta=1 \) when the distributor P1 is open during rapid feed, \( \beta=0 \) when the distributor P1 is closed during positioning);

\[
\frac{dp_2}{dt} = \frac{k_E}{k_2} \frac{3k-1}{2k} \frac{k}{2} \frac{1}{\sqrt{R}} \cdot \frac{1}{T_a} \cdot \varphi \left( \sigma_2 \right) + \frac{dp_2}{dt} \frac{k_p_2}{E_2} \frac{1}{s+L_{12}-L} \frac{1}{S_2} \frac{1}{T_a} \cdot \varphi \left( \sigma_1 \right) + \beta \cdot \mu_{tr2} \cdot f_{tr2} \sqrt{R} \frac{P_2}{p_2} \frac{k_1}{k_2} \frac{1}{T_2} \frac{1}{T_a} \frac{1}{S_2} \frac{1}{s+L_{12}-L} \frac{1}{T_a} \varphi \left( \sigma_1 \right) + \frac{dp_1}{dt} \frac{k_p_1}{E_1} \frac{1}{s+L_{12}-L} \frac{1}{S_2} \frac{1}{T_a} \varphi \left( \sigma_1 \right).
\]

Where s is the stroke of the pneumatic cylinder; \( L_{12} \) is the ratio of the initial ("passive") volume \( W_{dr} \) of the pneumatic drive to the useful area of the piston and rod cavities of the pneumatic cylinder, respectively, m.

Control algorithm
Entry conditions
\( t = 0; P_1 = P_2 = 0; P_1 = P_2 = P_0 = 0; \)

1) \( t > 0; P_2 = 0; \ (P_0 = P_0 = 0); \ P_1 = P_3 = P_0; \)

2) On diskret \( z=1 \) PLC \( \rightarrow xy \) to the Distributor \( P_2 \) Drain \( PC_2 \rightarrow P_4 \) pressure \( PC_1 \) consistent sequential, \( V \uparrow \ L = V t \); \( w \uparrow \ V \uparrow 4 \uparrow \left( \frac{d_k}{2} \right) ; \)

3) In diskret \( z, \) when combining slots in the PMD of pneumatic locks. The second positioning is mechanical closure.

2. Discussion
Simulation program for transient processes of positional pneumatic mechanical drive in the simintech computer simulation system

When creating pneumatic actuators with positional control, it is required to ensure that the pneumatic actuator stops at any point along the stroke length. To ensure braking and stopping in these cases, a special pneumatic brake is used.

The mathematical model of a 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. Thus, the mathematical model of the pneumatic drive includes the following equations.

1. 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 [6-11].
Figure 3. Computer model of a pneumatic cylinder created in the Simintech program

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. The positioning process of a dynamic system, including mechanical and pneumatic subsystems, must be simulated, because the process is complicated by the non-stationarity of the compressed gas flow, the instability of the real effects on the pneumatic subsystem. Therefore, when forming the mathematical model of the drive (the design diagram is shown in Figure 2), 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 counted; the force of viscous friction is proportional to the speed; the cost coefficients 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 formation of the control signal from the displacement sensor is not taken into account [12-16].
Figure 4. Program for implementing mathematical modeling

```
input p1,p2,v1,x1;
output p1,p2,v1,x1,Fy;
init p1=600000,p2=1000000,v1=0,x1=0;
//starting conditions
k=1.4; // the ratio of specific heats for air//
p1a=600000; // the inlet pressure into the cylinder, Pa//
pa=100000; // atmospheric pressure, Pa //
R=287; // gas constant //
dtr=0.008; // pipeline diameter, m //
f1=3.14*(dtr^2)/4; // cross-section area of the supply line //
f2=3.14*(dtr^2)/4; //cross-section area of the diverting line //
 zlib=65; // resistance coefficient of the supply line //
 z2=100; // the resistance ratio of the exhaust line //
 Tm=293; // gas temperature in the pipeline, K //
 L=8; //cylinder stroke, m //
 Ltr=0.1; // pipeline length, m //
 W=(3.14*(dtr^2))/4; // the volume of the pipeline //
 D=0.05; // air cylinder piston diameter //
 Dst=0.04; // diameter of the pneumatic cylinder rod //
 Mpr=5; // weight of moving parts of the pneumatic cylinder //
cpr=50000; // spring constant //
 S1=(3.14*(D^2))/4; // air cylinder piston area //
 S2=3.14*((D^2)-(Dst^2))/4; // air cylinder rod area //
 x01=0.08; //the given coordinate //
 x02=0.08; //the given coordinate //
 Fv=1200; // the force of external resistance //
 kv=300; // coefficient of viscous friction of the piston //
begin
    if (x1<0 then Fy=(cpr*(x1-1))
    else if (L>x1) or (x1>0) then Fy=0
    else if x1=0 then Fy=cpr*x1
end
p1'=k*v1*sqrt(R*Tm)/(S1*(x1+x01)*sqrt(z1))*sqrt(abs(p11*p1-p1^2))-(k*abs(v1))/(x1+x01);
p2'=k*v2*sqrt(R*Tm)/(S2*(1-x1)*x02)*sqrt(abs(p22*p2-p2^2))*((p2/pa)^((k-1)/(2*k)))*((k*p2*v1)/(1-x1+x02));
v1'=(S1*p1)-(S2*p2)+Fv*sign(x1')-kv*v1'-Fy/Mpr;
x1'=x1;
```
The results of computer simulation

**Figure 5.** Graph of pressure changes in the pressure cavity of the PC

**Figure 6.** Graph of pressure changes in the exhaust cavity of the PC
Figure 7. Graph of changes in movement from time to time

Figure 8. Graph of speed changes from time
Combining the obtained dependencies, we obtain a system of equations describing the process of positioning the output link of the pneumatic actuator. The system of differential equations can be solved by various numerical methods (Euler, Runge-Kutta, etc.) under given initial conditions, drive parameters, and control actions that functionally depend on the coordinate of the output link. The model was studied in the SimIntech program using numerical methods.

References
[1] Tamarkin M.A., Tishchenko E.E., Gordiyenko A.V., Grebenkin R.V. Reliability control of final vibration strengthening of part processing in steel balls medium. Vestnik of Don State Technical University. 2017;17(3):38-45. (In Russ.) https://doi.org/10.23947/1992-5980-2017-17-3-38-45.

[2] Pershin V.A., Khinikadze T.A. Technique of functional unification of adaptive hydraulic drive module capable of load stabilization on the working body of mobile machines. Vestnik of Don State Technical University. 2018;18(3):318-325. https://doi.org/10.23947/1992-5980-2018-18-3-318-325

[3] Sirotenko A.N., Partko S.A., Saed B.A. Dependence of energy-speed characteristics of pneumatic drive on initial parameters of additional volume under counterpressure braking. Vestnik of Don State Technical University. 2017;17(4):69-76. (In Russ.) https://doi.org/10.23947/1992-5980-2017-4-69-76

[4] Sirotenko A.N., Partko S.A., Salloum W. Effect of recuperative volume parameters on dynamic characteristics of pneumatic drive underbraking. Vestnik of Don State Technical University. 2018;18(4):379-384. https://doi.org/10.23947/1992-5980-2018-4-379-384

[5] Sidorenko V.S., Grishchenko V.I., Rakulenko S.V., Poleshkin M.S., Dymochkin D.D. Study on oil pilot circuit of adaptive hydraulic drive of tool advance in mobile drilling machine. Vestnik of Don State Technical University. 2019;19(1):13-23. https://doi.org/10.23947/1992-5980-2019-1-13-23

[6] Rybak A.T., Tsybriy I.K., Nosachev S.V., Zenin A.R. Theoretical background of hydraulic drive control system analysis for testing piston hydraulic cylinders. Vestnik of Don State Technical University. 2019;19(3):242-249. https://doi.org/10.23947/1992-5980-2019-3-242-249
[7] Dao The An. Modeling of positioning processes of a high-speed pneumatic robot drive / Dao The An, V. S. Sidorenko, D. D. Dymochkin // Fundamental research. - 2015. - Issue # 7. (part 2). pp. 285-292. https://fundamental-research.ru/ru/article/view?id=38687

[8] Dao The An. Research of positioning accuracy of an automated pneumatic drive with an external brake device / Dao The An, V. S. Sidorenko, D. D. Dymochkin // Bulletin of DSTU. - 2015. - Issue No. 4(83). Pp. 46-53. https://www.vestnik-donstu.ru/jour/article/view/41

[9] Obukhova E. N., Grishchenko V. I., Dolgov G. A. Formalization of a dynamic model of a pneumatic actuator with a variable structure. 14th international scientific-technical conference "dynamic of technical systems", DTS. 2018;226(51): 02022. https://elibrary.ru/item.asp?id=38626650

[10] Sirotenko A.N., Partko S.A., Voinash S.A., Research of pneumodrive with energy recovery into additional volume. Lecture notes in mechanical engineering. 2020;9(1): 1325-1333. https://link.springer.com/chapter/10.1007%2F978-3-030-22063-1_140

[11] Grishchenko V.I., Tumakov A.A., Poleshkin M.S., Kilina M.S., Dymochkin D.D. (2019) Modeling of automatic leveling system steep mobile machine with a hydraulic tilt sensor. Omsk Scientific Bulletin. - № 2 (164). P. 11-17.

[12] Vardhan, Alok & Dasgupta, Kaustubh & Mishra, Santosh. (2018). Dynamic analysis of a closed-circuit hydraulic drive system used in the rotary head of blasthole drilling machine using MATLAB–Simulink environment. Proceedings of the Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering. 233. 095965181880887. 10.1177/0959651818808870.

[13] Mikhail Lemeshko, Mikhail Molev, and Igor Golovin Hydraulic technological machines with adaptive drive structure/ (ICMTMTE 2018) Sevastopol, Russia, September 10-14, 2018 MATEC Web of Conferences, Volume 224, 02087 (2018) DOI: https://doi.org/10.1051/matecconf/2018224020875.

[14] Ozersky A.I. Modelirovanie dinamicheskikh rezhimov raboty gidroprivodnyh sistem s teplovymi i elektricheskimi istochnikami `energii [Modelling of dynamic operating modes hydromedriven systems with thermal and electric energy sources]. Izvestiya vuzov.Severo-Kavkazskij region. Tehnicheskie nauki, 2013, no. 5, pp. 37 - 43.

[15] Nekrashevich, K.Ya. Matematicheskaya model' gidrosistemy, realizovannoy s primeneniem kombinirovannogo printsipa adaptatsii k nagruzke. [Mathematical model of hydraulic system designed with using of combined principle of adaptation to load.] Mechanics of Machines, Mechanisms and Materials, 2014, no. 1 (26), pp. 21– 31.

[16] Gorin, A. & Tokmakov, N. & Kyznetsov, I. (2020). Substantiation of Parameters of Machine with Volumetric Hydraulic Drive for Formation of Wells in Ground. 10.1007/978-3-030-22063-1_139.