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SAVING COMPRESSED AIR BY CHOOSING AN EFFICIENT PNEUMATIC DRIVE CONTROL CIRCUIT

The article presents the development of pneumatic circuits and control charts of pneumatic distribution valves, which make it possible to significantly expand the scope of application of power pneumatic tools to increase inertial loads and achieve a significant reduction in compressed air consumption compared to traditional throttle braking circuits with the simultaneous increase of the response speed of the pneumatic drive. A mathematical model of a pneumatic drive in a dimensionless form is obtained, and dynamic similarity criteria are highlighted. A methodology for determining the scope of efficient use of an energy-saving pneumatic drive circuits under specific operating conditions was created

Keywords: cost effectiveness, mathematical model, pneumatic drive, inertial load

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ЕКОНОМІЯ СТИСНЕНОГО ПОВІТРЯ ШЛЯХОМ ВИБОРУ ЕФЕКТИВНОЇ СХЕМИ КЕРУВАННЯ ПНЕВМАТИЧНИМ ПРИВОДОМ

У статті приведено розробку пневматичних схем та схем керування пневморозподільними клапанами, які дають змогу значно розширити сферу застосування силового пневмоприводу для збільшення інерційних навантажень та досягти суттєвого зниження витрат стисненого повітря порівняно з традиційними схемами дросельного гальмування. з одночасним збільшенням швидкості спрацювання пневмоприводу. Отримано математичну модель пневмоприводу в безрозмірному вигляді та виділено критерії динамічної подібності. Створено методику визначення області ефективного використання енергозберігаючих схем пневматичного приводу в конкретних умовах експлуатації.

Ключові слова: економічна ефективність, математична модель, пневмопривод, інерційне навантаження

Introduction and justification of relevance. Compressed air production in industrialized countries accounts for 10 to 15% of electricity from the total energy balance [9]. At the same time, compressed air is one of the most expensive energy carriers. One kilojoule of energy realized using a pneumatic drive is 5-7 times more expensive that realized using an electric drive [9]. Therefore, the task of energy saving should be solved at every phase, starting from the production and its transportation of this energy carrier up to the use as a means of automating production processes. In the latter case, the problem of designing an energy-efficient pneumatic drive (PD) should be solved in conjunction with other urgent problems: the expansion of the scope of application of power pneumatic tools to increase inertial loads, as well as the need to combine the pneumatic drive with modern microprocessor controls.

The most complete review of publications in this area with the addition of an extensive reference list is given in [9]. The authors focus in more detail on publications related to improving the energy efficiency of pneumatic motors, eliminating leaks in pneumatic pipelines, reducing pressure in the production of compressed air with subsequent restoration due to the multiplication effect, using recovery in the pipeline, and a rational controlling program for a group of compressors.

In [3], the idea of reducing the air pressure during its production with subsequent pressure restoration before the pneumatic motor using the multiplication effect based on the use of a booster valve is worked out in sufficient detail and justified. The effect of recovery of excess air mass into the pipeline is proposed to be used as the second source of energy efficiency improvement.

In [4], it is shown how, by replacing a 4-line servo valve with two 3-line servo valves, it is possible to introduce separate control over the air supply to the working cavity of the cylinder and the removal of air from the exhaust cavity of the cylinder, which leads to a more efficient use of compressed air energy.

The authors of [7] focus on the issues of optimizing compressed air production in order to minimize energy consumption. They investigate the energy efficiency of two methods of compressed air production: using one compressor with an adjustable speed of rotation of the drive motor and two or more compressors, which are switched on and off according to a specific program, depending on the consumption of compressed air by the system.

As the task of controlling the pneumatic drive basically includes braking and positioning the working body, solving this problem should provide opportunities to improve the energy efficiency of the pneumatic drive. Here, non-dissipative methods of quenching the kinetic energy of moving elements of the drive are of the greatest interest, i.e., methods based on the use of air compression force in the brake cavity of the cylinder as a braking force. The methods, when the cylinder cavities are connected to various switching objects without using throttles to accelerate, brake and lock (position) the piston, can be called braking by changing the structure of switching connections. As for the problem of high-precision positioning of the working body of the PD, English researchers Linnet and Smith first proposed the use of braking by switching switch connections using three-line pneumatic distribution valves [5, 6].

The authors of the works [1, 2] drew attention to an important feature of non-dissipative braking due to switching of switching connections - the use of braking energy in the form of potential energy of compressed air in the brake cavity either for recovery to the pipeline or for the implementation of the reverse course of the working body.

This article focuses on the further development of this area of research.

Purpose and objective of the study

The purpose of this study is to increase the energy efficiency of the pneumatic drive and expand the scope of application of power pneumatic tools for a significant increase in the inertial load due to the choice of a rational structure of switching connections for all phases of motion of the working element of the PD, as well as to determine the scope of efficient use of energy-saving pneumatic drive circuits.

To achieve this goal, the following tasks were identified and solved:

- justification of the most expedient structure of switching connections for all phases of motion of the working element of the PD, ensuring minimization of unproductive energy consumption;
- development of an energy-saving pneumatic drive circuits and its control algorithm;
- computer simulation based on the developed mathematical model in dimensional and dimensionless form in order to determine the scope of efficient use of the energy-saving pneumatic drive circuit.

Problem Statement

In the study of energy indicators, the efficiency of compressed air (exergy) was used as the consumed energy, i.e., the maximum useful work that can be obtained from a thermodynamic system as a result of its reversible transition to a thermal and mechanical state of equilibrium with the environment. If the compressed air in the compressor receiver and refrigerator comes into thermal equilibrium with the environment, then the specific operability can be determined by the expression [8]:

$$l_{\text{p}} = R \cdot T_{\text{м}} \cdot \ln \frac{p_{\text{м}}}{p_{\text{a}}} = U + \frac{R \cdot T_{\text{м}}}{p_{\text{м}}} \cdot \left(p_{\text{м}} - p_{\text{a}}\right), \tag{1}$$

where $\frac{R \cdot T_{\text{м}}}{p_{\text{м}}} \cdot \left(p_{\text{м}} - p_{\text{a}}\right)$ – specific work of ejection (air blast), which is commonly called transit efficiency; U - operation of isothermal expansion (compression).

Work defined by the area $p_a\, p_{\text{м}}\, a b$ (Fig. 1) is $V_2\left(p_{\text{м}} - p_{\text{a}}\right) = \frac{R \cdot T_{\text{м}}}{p_{\text{м}}} \cdot \left(p_{\text{м}} - p_{\text{a}}\right)$ is characteristic of the compressed air flow and is carried out not by changing the state of the gas in the flow, but by the energy that is transmitted through this gas from an external source (compressor), i.e., compressed air acts here as a kinematic link connecting the compressor displacement body with the working element of the pneumatic motor.

Work defined by the area abc is the work of the gas at rest and is associated with the potential energy of compressed air U which can be realized by isothermal expansion of the gas up to reaching complete equilibrium with the environment

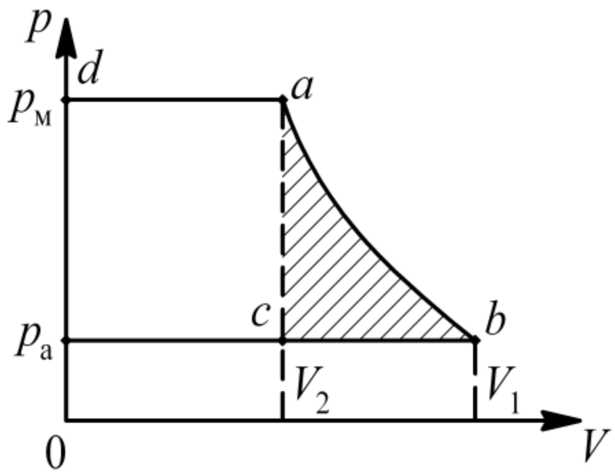


Fig. 1 – Image of specific efficiency in $p-V$ coordinates

In pneumatic motors with a full filling of the working volume, only the transit efficiency of compressed air is used, and air with unrealized potential energy is simply released from the exhaust cavity of the pneumatic motor into the atmosphere. This disadvantage affects all pneumatic motors in pneumatic drives with traditional throttle braking (Fig. 2).

In addition to low energy efficiency, another drawback of pneumatic drives with traditional throttle braking is due to the fact that the scope of operation of such drives is limited by a rather low inertial load. The transition process in Fig. 3, obtained as a result of computer simulation for PD with a piston diameter $D = 50$ mm, piston stroke length $L = 0{,}5$ m and compressed air pressure in the main pipeline $p_м = 0{,}6$ MPa, shows that already at mass load $m = 30$ the transition process during throttle braking is accompanied by a developed oscillatory process and a long response time.

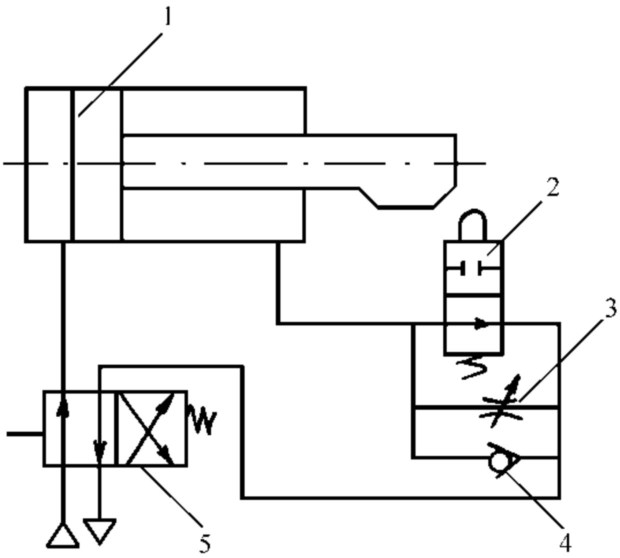


Fig. 2 – Throttle braking circuit with a discrete brake valve

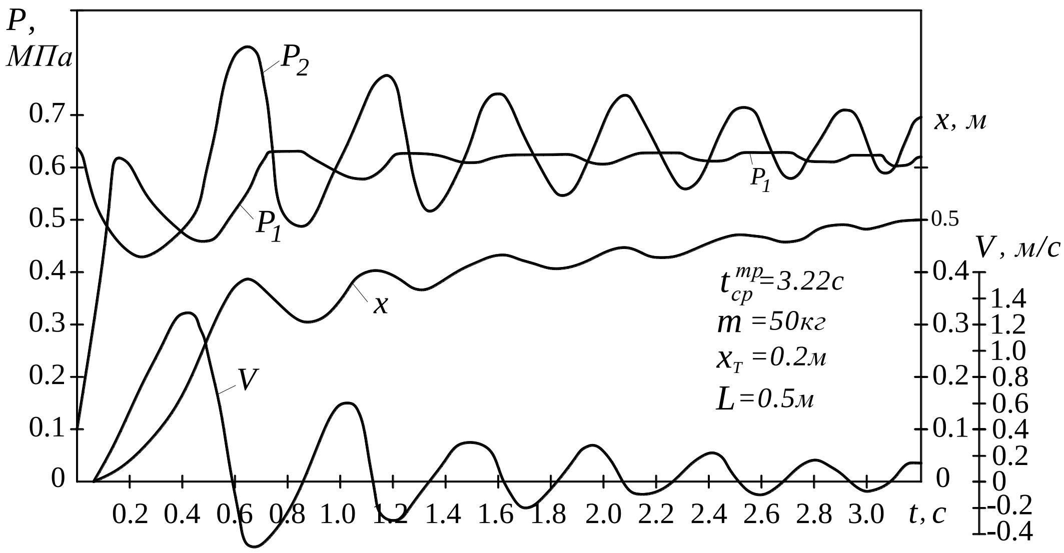


Fig. 3 – Transition process in PD with a discrete charge brake valve

Braking of the motion of the working body by changing the structure of switching connections

The transition to pneumatic drives with braking by changing the structure of switching connections provides a large number of circuits and options for switching the working and exhaust cavities of the pneumatic motor in the mode of acceleration, braking and locking the working element (WE) of the pneumatic drive. According to the principle of energy saving, preference should be given to the control circuit and algorithm that meets the following conditions:

- when the piston is locked, it must be held by means of the minimum required differential pressure on the piston, and the lower pressure must correspond to atmospheric pressure, and air with a higher pressure must be taken from the outlet of the pressure reducing valve (pressure $p_к$);
- in the acceleration phase, the working cavity that was previously switched with the atmosphere is combined with the supply line (pressure $p_м$), and the exhaust cavity is combined with the atmosphere. Low back pressure in the exhaust cavity in the initial phase of motion ensures the fastest acceleration of the piston;
- in the braking phase, the working cavity must be connected to the low-pressure source through a pre-closed check valve ($p_к$), and the exhaust cavity is connected to the main supply line through a pre-closed check valve ($p_м$). This ensures a more complete use of the efficiency of compressed air in the working cavity due to the realization of the potential energy of compressed air, as well as the recovery of braking energy to the supply pipeline from the exhaust (brake) cavity and an approximately constant pressure drop on the piston;
- in the final phase of braking $p_м - p_к$ (uniformly decelerating braking mode). After the piston stops, the locking mode begins, in which the working cavity is further connected with the low-pressure source ($p_к$), and the exhaust cavity is connected with the atmosphere.

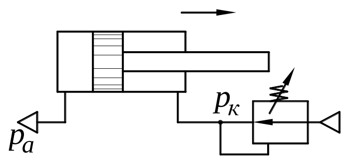
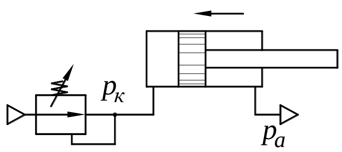
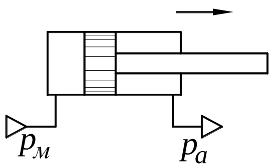
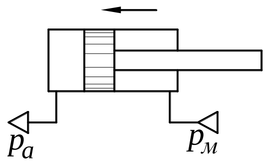
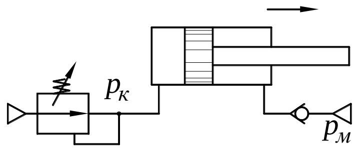
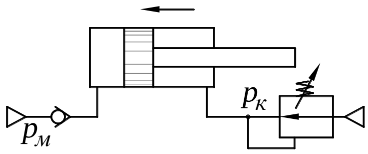
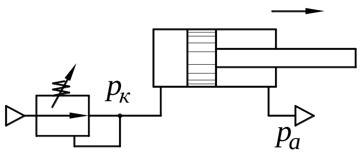
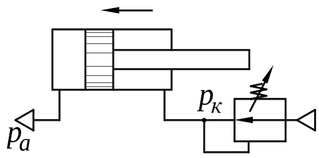
The desired switching connections are shown in Table 1.

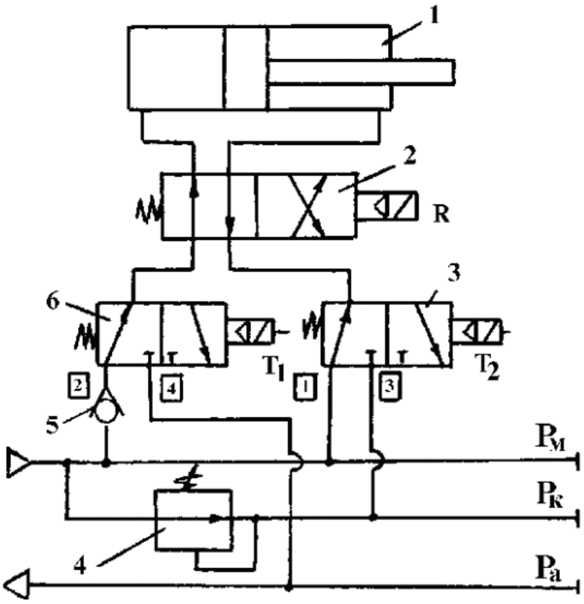
A pneumatic circuit and control chart of distribution valves based on 3/2 valves is shown in Fig. 4. A smaller pneumatic circuit can be obtained using a 5/3 distribution valves (Fig. 5). The circuits are made with pressure reducing valves without a discharge valve, which close the supply channel of the working cavity of the cylinder as long as the pressure in the working cavity is higher than the pressure of the gearbox setting ($p_к$).

The coordination of the start of braking is controlled using miniature remote position sensors with a Reed or inductive electric output, installed directly on the cylinder liner [11].

When a mathematical model for computer simulation of the operation processes in these pneumatic drives was developed, the task was not only to obtain a transition process

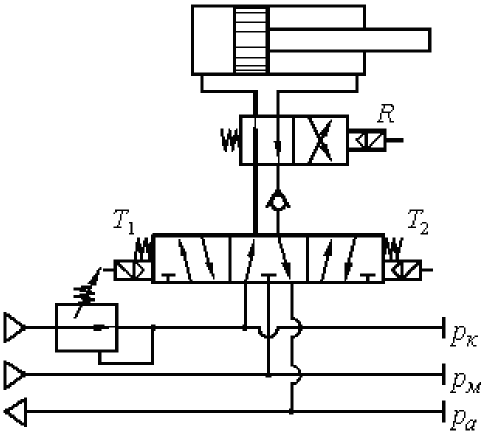
Table 1 - Switching connections for different phases of motion

Moving to the right		Moving to the left	
Locking the left position		Locking the left position	
Acceleration		Acceleration	
Braking		Braking	
Locking the right position		Locking the left position	



Phases of motion		T_1	T_2	R
Moving to the right	Initial state	1	1	0
	Acceleration	1	0	1
	Braking	0	1	1
	Lock	1	1	1
Moving to the left	Initial state	1	1	1
	Acceleration	1	0	0
	Braking	0	1	0
	Lock	1	1	0

Fig. 4 – Energy-saving pneumatic drive circuit using 3/2 distribution valves



Situation		R	T_1	T_2
Th the right	Acceleration	0	1	0
	Braking	0	0	1
	Lock	0	0	0
To the left	Acceleration	1	1	0
	Braking	1	0	1
	Lock	1	0	0

Fig. 5 – Energy-saving pneumatic drive structure using 5/3 distribution valves

in an energy-saving pneumatic drive in order to justify its advantage over drives with throttle braking, but also to determine the scope of efficient use of such drives. For such purposes, a mathematical model in dimensional form cannot be used because of multiple independent parameters that determine the nature of transients. American researchers Tsai and Cassidy in their work [10] were among the first to propose normalization of differential equations describing gas-mechanical processes in a pneumatic gearbox, based on the principle of minimizing dimensionless complexes. To achieve this purpose, they proposed using dimensionless time as an independent variable.

Based on these principles, we present a nonlinear mathematical model of the drive in dimensionless form, reducing it to the Cauchy Form (2).

$$\begin{cases} \dfrac{d\sigma_1}{d\tau} = \dfrac{k}{\xi_{01}+\xi}\left[s_1 z_1 \varphi(I_1) - \sigma_1 \dfrac{d\xi}{d\tau}\right]; \\ \dfrac{d\theta_1}{d\tau} = \dfrac{\theta_1}{\sigma_1}\dfrac{d\sigma_1}{d\tau} + \dfrac{\theta_1}{\xi_{01}+\xi}\dfrac{d\xi}{d\tau} - s_1' z_1 \dfrac{\varphi(I_1)}{\xi_{01}+\xi}; \\ \dfrac{d\sigma_2}{d\tau} = \dfrac{k}{\xi_{02}+1-\xi}\left[s_2 z_2 \dfrac{\varphi(I_2)}{\Pi_{21}^F} - \sigma_2 \dfrac{d\xi}{d\tau}\right]; \\ \dfrac{d\theta_2}{d\tau} = \dfrac{\theta_2}{\sigma_2}\dfrac{d\sigma_2}{d\tau} + \dfrac{\theta_2}{\xi_{02}+1-\xi}\dfrac{d\xi}{d\tau} - s_2' z_2 \dfrac{\varphi(I_2)}{\Pi_{21}^F(\xi_{02}+1-\xi)}; \\ \dfrac{d\xi}{d\tau} = \dot{\xi}; \\ \dfrac{d\dot{\xi}}{d\tau} = \dfrac{1}{\beta}\left(\sigma_1 - \sigma_2 \Pi_{21}^F - \chi\right), \end{cases} \tag{2}$$

where $\sigma_1 = p_1/p_м$, $\sigma_2 = p_2/p_м$, $\theta_1 = T_1/T_м$, $\theta_2 = T_2/T_м$ - dimensionless pressures and temperatures in the working and exhaust cavities of the cylinder, respectively $(p_м, T_м$ - parameters of air in the supply line); $\xi = x/L$, $\dot{\xi} = \dfrac{v}{L} t_б \cdot \xi$ – respectively, dimensionless coordinates and velocity of the piston; $t_б = \dfrac{F_1 L}{f_1^e \sqrt{kRT_м}}$ - base unit of time, numerically equal to the time of filling the working volume of the cylinder with air moving at the speed of sound $a_м = \sqrt{kRT_м}$ (L - full stroke of the piston; F_1 - area of the piston) through an opening equal to the effective area of the intake pneumatic line $f_1^e = \mu f_1$; $\tau = t/t_б$ - dimensionless time; $\Pi_{21}^F = F_2/F_1$ – the ratio of piston areas in the rod cavity and in the rodless cavity; ξ_{01}, ξ_{02} — dimensionless initial boundaries of the piston on the side of the working and exhaust cavities.

The main criteria for dynamic similarity of a pneumatic drive include:

$\beta = \dfrac{m L}{t_б^2 F_1 p_м}$ - inertia criterion (reduced mass) which is numerically equal to the ratio of the inertia force at basic acceleration $L/t_б^2$ to the maximum (indicator) force developed by the piston; $\chi = \dfrac{P}{F_1 p_м}$ - relative static load.

Dimensionless parameters Π_{21}^F, ξ_{01}, ξ_{02} are considered to be secondary similarity criteria, because they either hardly change in real pneumatic drives or have little effect on the transition process.

The system (2) is the basic platform for describing all pneumatic drive circuits with braking by changing the structure of switching connections.

To identify a specific circuit, logical-algebraic modules are introduced I_1, I_2, s_1, s_2, s_1', s_2', z_1, z_2, which describe the nature of switching connections for this circuit at different phases of piston motion. The pressure ratios on the supply and discharge lines are determined taking into account changes in the flow direction:

$$I_1 = \left(\frac{p_1}{y_1}\right)^{sign(y_1 - p_1)}; \quad I_2 = \left(\frac{p_2}{y_2}\right)^{sign(y_2 - p_2)},$$

where y_1 and y_2 - function of connecting the switching objects for the left and right cylinder cavities, respectively. For pneumatic drive with the control circuits and algorithm in Fig. 5:

$$\begin{aligned}
y_1 = {} & \overline{T}_1 \overline{R}\left[\frac{1+sign(p_k - p_1)}{2} p_k + \frac{1+sign(p_1 - p_k)}{2} p_1\right] + T_1 \overline{T}_2 \overline{R}\, p_м + \\
& + \overline{T}_2 R\left[\frac{1+sign(p_1 - p_a)}{2} p_a + \frac{1+sign(p_a - p_1)}{2} p_1\right] + \\
& + R T_2 \overline{T}_1\left[\frac{1+sign(p_1 - p_м)}{2} p_м + \frac{1+sign(p_м - p_1)}{2} p_1\right];
\end{aligned}$$

$$\begin{aligned}
y_2 = {} & \overline{T}_1 R\left[\frac{1+sign(p_k - p_2)}{2} p_k + \frac{1+sign(p_2 - p_k)}{2} p_2\right] + T_1 \overline{T}_2 R\, p_м + \\
& + \overline{T}_2 \overline{R}\left[\frac{1+sign(p_2 - p_a)}{2} p_a + \frac{1+sign(p_a - p_2)}{2} p_2\right] + \\
& + \overline{R} T_2 \overline{T}_1\left[\frac{1+sign(p_2 - p_м)}{2} p_м + \frac{1+sign(p_м - p_2)}{2} p_2\right].
\end{aligned}$$

Boolean variables T_1, T_2, R take the value 1 when current is applied to the corresponding electromagnets, and the variables $\overline{T}_1$, $\overline{T}_2$, $\overline{R}$ take the value 1 when the current is disconnected from the corresponding electromagnets.

Dimensionless pressure parameters of switching objects for the left and right cylinder chambers $\sigma_{м1} = y_1/p_м$, $\sigma_{м2} = y_2/p_м$.

Spending function:

$$\varphi(I_1)=\frac{1+sign(I-0{,}528)}{2}\sqrt{\frac{2}{k-1}\left(I^{2/k}-I^{(k+1)/k}\right)}+0{,}579\frac{1-sign(I-0{,}528)}{2}, \qquad (3)$$

where k is an adiabatic indicator.

Functions s_1, s_2, s_1', s_2' allow us to perform a discrete change in the structure of the first six equations of the system (2) due to the transition from filling the cylinder cavities to emptying and vice versa:

$$s_1=\frac{1+sign(\sigma_{\text{м}1}-\sigma_1)}{2}\sigma_{\text{м}1}-\frac{1+sign(\sigma_1-\sigma_{\text{м}1})}{2}\sigma_1\sqrt{\theta_1};$$

$$s_2=-\frac{1+sign(\sigma_{\text{м}2}-\sigma_2)}{2}\sigma_{\text{м}2}+\frac{1+sign(\sigma_2-\sigma_{\text{м}2})}{2}\sigma_2\sqrt{\theta_2};$$

$$s_1'=-\frac{1+sign(\sigma_{\text{м}1}-\sigma_1)}{2}\frac{\theta_1^2}{\sigma_1}+\frac{1+sign(\sigma_1-\sigma_{\text{м}1})}{2}\sigma_1\sqrt{\theta_1};$$

$$s_2'=-\frac{1+sign(\sigma_{\text{м}2}-\sigma_2)}{2}\frac{\theta_2^2}{\sigma_2}+\frac{1+sign(\sigma_2-\sigma_{\text{м}2})}{2}\sigma_2\sqrt{\theta_2}.$$

During numerical integration of differential equations in dimensionless form, in addition to determining the parameters of internal transients (dimensionless pressures and temperatures in the cavities of the pneumatic cylinder σ_1, σ_2, θ_1, θ_2) and state variables ξ, $\dot{\xi}$ (motion and speed of the working element) the relative mass amount of compressed air consumed during a single operation was determined $\overline{\text{M}}$ and averaged per cycle efficiency $\eta_{\text{ср}}$:

$$\overline{\text{M}}=\int_0^{\tau_c}Gd\tau/F_1\,L\,\rho_{\text{м}}=\int_0^{\tau_c}\varphi(I_1)d\tau, \qquad (4)$$

where G - mass flow rate supplied to the pneumatic cylinder; $\varphi(I_1)$ - cost function; τ_c - dimensionless response time; $F_1\,L\,\rho_{\text{м}}$ - mass amount of compressed air required to fill the working volume with compressed air to its parameters in the supply line ($p_{\text{м}}$, $T_{\text{м}}$), $\rho_{\text{м}}=\frac{RT_{\text{м}}}{p_{\text{м}}}$ - compressed air density.

Compressed air energy consumed during a single actuating (E) is defined according to expression (1):

$$E=RT_{\text{м}}\cdot\ln\frac{1}{\sigma_a}\int_0^{t_{cp}}Gdt=f_1^e\,p_{\text{м}}\sqrt{k\,RT_{\text{м}}}\ln\frac{1}{\sigma_a}\int_0^{t_{cp}}\varphi(I_1)dt. \qquad (5)$$

Since that PD with large mass loads and braking of the working element at the end of the stroke are studied, then as a useful work in the calculation $\eta_{\text{ср}}$, in addition to work to overcome the static resistance force $P\cdot L$, the average kinetic energy of moving parts per cycle was taken into account

$$\eta_{\text{ср}}=\frac{\dfrac{m}{2\cdot t_{cp}'}\displaystyle\int_0^{t_{cp}'}V^2dt+P\cdot L}{f_1^e\cdot p_{\text{м}}\sqrt{k\,RT_{\text{м}}}\ln\dfrac{1}{\sigma_a}\displaystyle\int_0^{t_{cp}}\varphi(I_1)dt}; \qquad (6)$$

Or in dimensionless form:

$$\eta_{\text{cp}} = \frac{\dfrac{\beta}{2\cdot\tau'_{cp}}\displaystyle\int_0^{\tau_{cp}}\dot{\xi}^2 d\tau + \chi}{z_1 \ln\dfrac{1}{\sigma_a}\displaystyle\int_0^{\tau_{cp}}\varphi(I_1)d\tau}, \tag{7}$$

where τ'_{cp} - dimensionless time of motion of the working element of the PD from one position to another; τ_{cp} - full response time; $\sigma_{\text{a}} = p_a / p_{\text{м}}$; z_1, z_2 - coefficients of a discrete change in the throughput of the supply and exhaust lines when the structure of switching connections is changed.

Thus, the average efficiency of PD can be calculated during the general numerical integration of the original mathematical model, expanded by introducing new equations (6) and new integrated parameters Y and $\overline{\text{M}}$ into the system (2).

$$\begin{cases} \dfrac{d\overline{\text{M}}}{d\tau} = z_1 \bar{R}\, T_1 \varphi(I_1) + \bar{R}\, T_2 \left[z_1 \varphi(I_1)\dfrac{1+sign(\sigma_k - \sigma_1)}{2} - z_2\, \varphi(I_2)\dfrac{1+sign(\sigma_2 - 1)}{2}\right]; \\ \dfrac{dY}{d\tau} = \dfrac{\xi^2 \beta}{2}. \end{cases} \tag{8}$$

After the pneumatic drive completed the actuation cycle and values τ'_{cp}, Y and $\overline{\text{M}}$ were obtained based on numerical integration, the averaged efficiency of PD over the cycle was calculated:

$$\eta_{\text{cp}} = \frac{Y/\tau'_{cp} + \chi}{\overline{\text{M}}\cdot \ln 1/\sigma_a}. \tag{9}$$

Fig. 6 shows the transition process in PD with braking of the working elements by changing the structure of switching connections with an energy-saving mode of operation at the same drive parameters as with throttle braking (Fig. 3). The difference is that the mass load is brought up to 100 kg. Supply pressure $p_{\text{м}} = 0{,}6$ MPa, pressure setting $p_{\text{м}} = 0{,}2$ MPa. The very form of the transition process changed dramatically and it approached the optimal one, when the speed change is close to the cycloidal law.

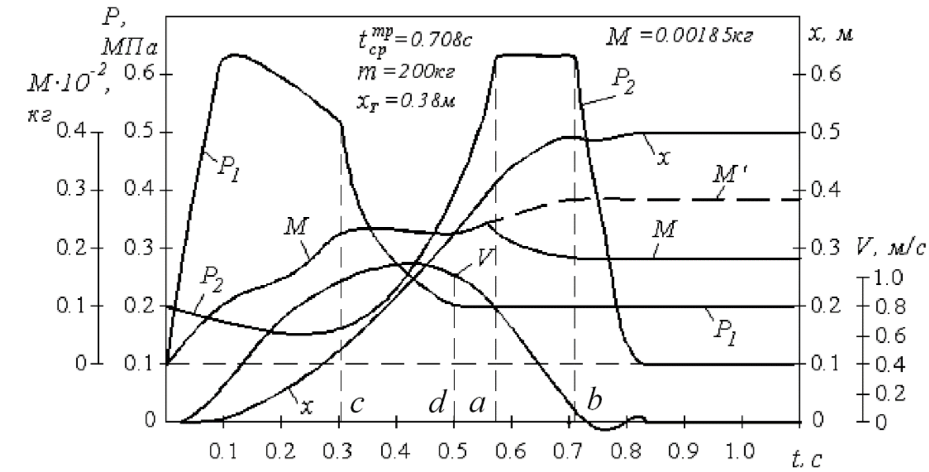


Fig. 6 – Transition process in a pneumatic drive operating according to the circuit in Fig. 5

When braking with a controlled and regulated brake pressure pulse was used, the following significant changes occurred:

- the response speed due to the reduction of deadwork of pushing air out of the exhaust cavity and optimizing the form of the transition process has significantly increased (the response time has been reduced by 2.5 to 3 times compared to the throttle braking option);
- the transition process became unfluctuating with a uniformly decelerating braking mode, and the amount of negative acceleration can be controlled by adjusting the pressure of the pressure reducing valve;
- at medium and high inertial loads, the effective energy saving mode is implemented (Fig. 7, 8);
- for a period of time c – d (fig. 6) the work of compressed air in the left cavity of the cylinder is carried out due to the potential energy of expansion (compression) U (1);
- for a period of time *a* – *b* compressed air from the brake cavity is recovered into the pipeline.

Curve M' displays compressed air consumption without considering recovery. Curve M corresponds to the compressed air flow rate, considering recovery.

Determination of the scope of efficient use of a pneumatic drive with energy-saving control circuits

Transition process in PD with an energy-saving circuit in dimensionless form (Fig. 7) was obtained at $\beta = 2{,}5$ and $\chi = 0{,}1$, and its defining feature is the ability to estimate the reduction in compressed air consumption immediately as a percentage compared to throttle braking (full filling of the working volume). In this case $\overline{M} \cdot 100\% = 22\%$ from full filling of the working volume.

Limitation of differential equations based on the principle of minimizing dimensionless complexes allowed us to generalize the results of computer simulation in the form of a single graph in the plane of similarity criteria β and χ (Fig. 8).

Determination of the scope of efficient use of energy-saving circuits (Fig. 5, 6) is based on the selection of boundaries in the space of dynamic similarity criteria β and χ (Fig. 8).

The area of the most efficient use of energy-saving pneumatic drive circuits lies within the limits of $2 \le \beta \le 5$ and $\chi \le 0{,}3$, where the consumption of compressed air can be reduced by 2 to 10 times. At the same time, the area of effective braking is significantly expanded up to $\beta = 5$, which greatly overlaps the scope of use of PD with traditional throttle braking ($\beta < 0{,}5$).

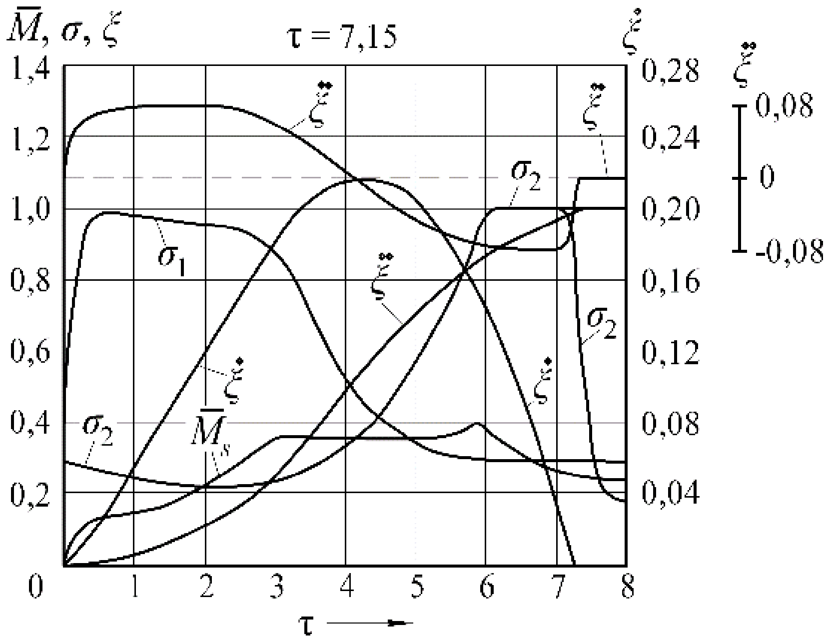


Fig. 7 – Transition process in dimensionless form

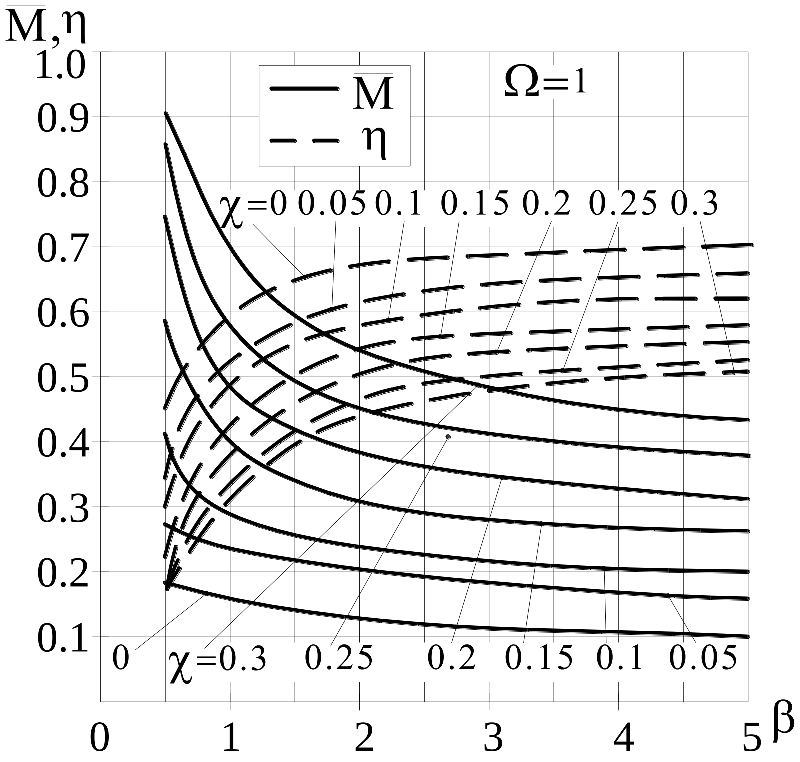


Fig. 8 – Dependence of the relative mass of air consumed $\overline{M}$ and focused efficiency η on similarity criteria β and χ

Calculating the values of similarity criteria based on the initial design data and load conditions of the PD β and χ you can use the graph in Fig. 8 to determine the savings of compressed air with one actuation of the PD knowing the service life of the pneumatic motor (for a pneumatic cylinder $2\cdot10^6$ compressed air savings can be calculated for the entire period of operation of the pneumatic system. When the price 1 $м^3$ is known, it is possible to compare operating costs (the cost of compressed air consumed) and one-time costs (the cost of hardware implementation of the corresponding pneumatic circuits) to conduct a cost analysis and make an informed decision on the feasibility of switching to an energy-saving PD circuit.

In an experimental study of the proposed braking methods, a long-stroke cylinder by SMC (Japan) was used ($D = 50$ mm; $L = 1000$ mm; $d = 20$ mm), installed on the lathe bed. A trolley with loads was used as a loading device, which could move along the guides of the bed. The trolley could be loaded with flat weights with a weight of up to 20 kg up to 400 kg (the weight of the trolley is 10 kg).

Fig. 9, 10 show waveforms of the transition process obtained when the drive is loaded with a mass load of 30 kg (medium load) and 190 kg (heavy load), respectively. The shape of the transition process on the waveforms indicates that in the braking mode a constant pressure drop on the piston $p_2 - p_1$ and uniformly decelerating braking mode at the final phase of braking can be maintained.

Moreover, as the load increases, the braking mode becomes more stable (Fig. 10), which suggests the possibility of effective use of the proposed method of controlling the pneumatic drive in conditions of inertial loads that are many times higher than the capabilities of the drive with throttle braking.

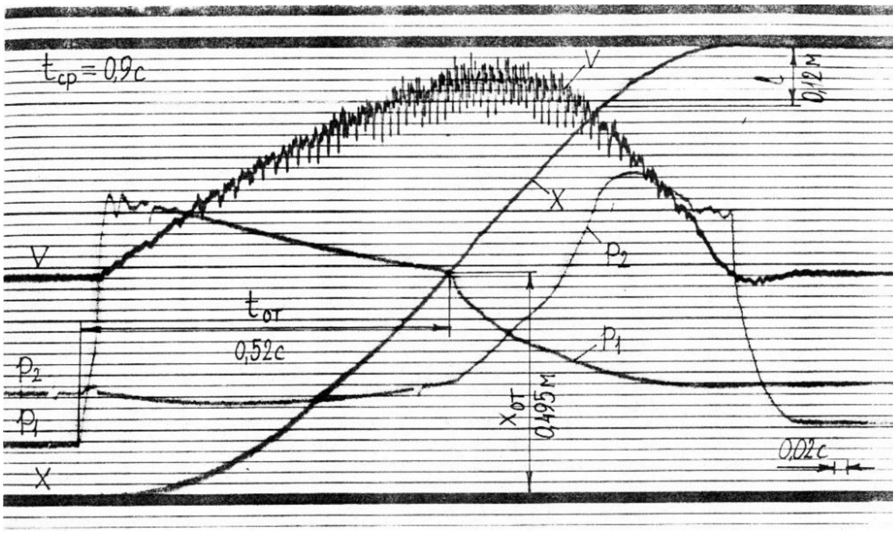


Fig. 9 – Waveform of the transition process for circuit No. 3 when the rod is extended
($p_М = 0{,}45$ MPa; $p_k = 0{,}1$ MPa; $x_{01} = x_{02} = 0{,}08$ m; $П_{21}^F = 0{,}84$; $m = 30$ kg; $L = 1000$ mm)

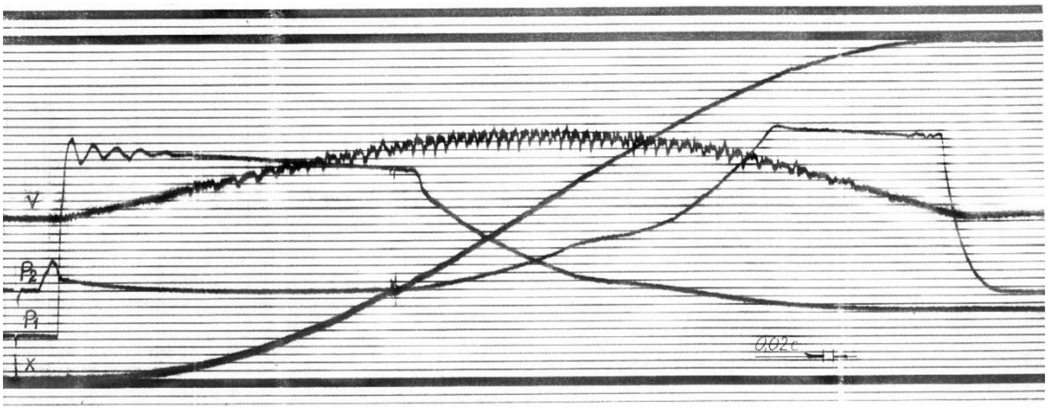


Fig. 10 – Waveform of the transition process for circuit No. 3 when the rod is extended
($p_М = 0{,}45$ MPa; $p_k = 0{,}1$ MPa; $x_{01} = x_{02} = 0{,}08$ m; $П_{21}^F = 0{,}84$; $m = 190$ kg; $L = 1000$ mm)

The waveforms also show areas that indicate the effective use of the potential expansion energy of compressed air in the working cavity of the pneumatic cylinder (curve $p_1(t)$), recovery of compressed air from the brake cavity into the pipeline (curve $p_2(t)$), as well as a low level of back pressure in the exhaust cavity until braking begins. This confirms a significant improvement in the energy characteristics of PD.

Conclusions. Pneumatic circuits and control charts of distribution valves are proposed, which make it possible to significantly expand the scope of application of power pneumatic tools to increase inertial loads and achieve a significant reduction in compressed air consumption compared to traditional throttle braking circuits with the simultaneous increase of the response speed of the pneumatic drive.

Energy and dynamic analysis allows us to establish with a high degree of confidence that the developed pneumatic drive circuit provides high response time over the entire range of inertial and static loads with smooth impact-free operation with an equal-speed and adjustable braking mode.

The drive provides optimal switching connections for each phase of motion, thereby achieving:

- sufficiently complete use of the potential energy of air expansion in the working cavity;
- reduction of deadwork of pushing compressed air out of the exhaust cavity;
- significant reduction in compressed air consumption for locking the piston in the final position;
- recovery of potential compressed air energy from the brake cavity into the pipeline.

Using a dimensionless form of a mathematical model of a pneumatic drive with the identification of dynamic similarity criteria, it was possible to create a methodology for determining the scope of efficient use of an energy-saving PD circuit under specific operating conditions.

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