Область застосування пневмоприводів розширена убік збільшення інерційних навантажень за рахунок гальмування шляхом зміни структури комутаційних зв'язків. Доведено перевагу такого способу над традиційним способом дросельного гальмування. Також визначені способи зниження непродуктивних енерговитрат у пневмоприводах шляхом структурного й параметричного синтезу. Виділена область раціонального використання запропонованих схем пневмоприводів

Ключові слова: дискретні пневмоприводи, структура комутаційних зв'язків, зниження непродуктивних енерговитрат

Область применения пневмоприводов расширена в сторону увеличения инерционных нагрузок за счет торможения путем изменения структуры коммутационных связей. Доказано преимущество такого способа над традиционным способом дроссельного торможения. Также определены способы снижения непроизводительных энергозатрат в пневмоприводах путем структурного и параметрического синтеза. Выделена область рационального использования предложенных схем пневмоприводов

Ключевые слова: дискретные пневмоприводы, структура коммутационных связей, снижение непроизводительных энергозатрат

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THE SYNTHESIS OF STRUCTURE AND PARAMETERS OF ENERGY EFFICIENT PNEUMATIC ACTUATOR

G. Krytikov
Doctor of Technical Sciences, Professor
Department of Hydraulic and Pneumatic Machines**
E-mail: gkrutikov@gmail.com

M. Strizhak
PhD, Associate Professor*
E-mail: marjanastryzhak@gmail.com

V. Strizhak
PhD, Associate Professor*
E-mail: stryzhak.vsevolod@gmail.com

*Department of Lifting-and-shifting, construction, road-making, land reclamation machines and equipments**
**National Technical University «Kharkiv Polytechnic Institute»
Kyrpychova str., 2, Kharkiv, Ukraine, 61002

1. Introduction

Relevance and need for studies, directed toward development and introduction of energy-saving circuits for pneumatic actuators, is predetermined, one the one hand, by the fact that compressed air is one of the most expensive energy carriers. One kilojoule of energy, realized with the help of pneumatic motors, costs 5÷7 times more than that with the aid of electric motors [1]. From the other hand, this is due to the fact that the scale of application of compressed air in the industrially developed countries constantly grows and, at present, production of compressed air accounts already for 15 % of the country's total energy balance [1].

The schemes and algorithms, proposed and substantiated in present work, to control a pneumatic actuator provide for a cardinal reduction in unproductive power costs with a simultaneous improvement of other operating characteristics of the drive. Studies presented in the article are directly related to solving fundamental problems on the utmost increase in the energy effectiveness of contemporary production.

2. Literature review and problem statement

In contemporary basic research into the field of energy saving in pneumatic systems, the processes of braking are not connected as a rule with the energy saving [2–4]. Thus, article [2] mentions ten methods to reduce expenditures in the pneumatic systems, starting from eliminating leakages to the maximum minimization of the overall dimensions of pneumatic actuators. Paper [4] proposes to minimize operating pressure and dimensions of cylinder, as well as recycle exhaust air, but there are no references on the choice of energy-saving methods for braking and positioning of the operating units of pneumatic actuator.

Authors of article [5] were among the first who paid attention to the fact that using compressed air in pneumatic actuators makes it possible to realize a non-dissipative braking method in pneumatic actuators (to accumulate braking energy in the form of potential energy of compression, turning a brake cavity into compressor during the return stroke). The same approach is applied in an earlier paper by Japanese researchers [6]. A realization of such compression-driving regime, although undoubtedly decreasing unproductive consumption of compressed air, is inferior, however, to the methods of braking, proposed in this article as it requires the existence of mechanical switches of position of operating unit. A more exotic variant for the recovery of compressed air at braking is presented in article [7], in which authors propose to use recuperation of the compressed air to produce electrical energy.

Paper [8] proposed for improving energy characteristics of pneumatic actuator to use potential energy of compression in the brake cavity, by commutating brake cavity at a specific moment with the working cavity with the help of electro-pneumatic valve. This method of braking, however, leads to big problems in the control of such system, as it requires conducting synchronous control of both the basic distributor, switching it into braking mode and the electro-pneumatic valve in charge of commutation of the working and exhaust cavities of cylinder. Moreover, in the first case, directional control is used, and in the second – temporal. This is an essential loss in comparison with the circuit solutions, which would use directional control for only one distributor.

In addition, the application of pneumatic actuator in the industry is substantially limited by the magnitude of inertia loads for avoiding the developed oscillatory processes [9]. Circuits of pneumatic actuators with the programs of rational control for distributing valves are more promising since they make it possible to considerably expand the area of applying power pneumatics towards increasing the inertia loads.

3. The aim and tasks of the study

The purpose of present work is to substantiate a possibility of increasing the inertia loads with simultaneous attainment of the more effective use of the working capacity of compressed air, which will make it possible to substantially expand the area of application of power pneumatics.

To achieve the set aim, the following tasks were to be solved:

– to define basic components of unproductive power consumption in the operation of pneumatic actuator with throttle braking;

– to perform structural synthesis of energy-saving pneumatic actuator with braking of operating unit of the drive by changing the structure of commutation relations;

– to develop a mathematical model for energy-saving pneumatic actuator in the dimensionless form with the isolation of basic criteria in dynamic similarity;

– to run, based on the obtained mathematical model, a computer simulation of transient processes in the energy-saving pneumatic actuators and to isolate the areas of their rational utilization;

– to determine those parameters of pneumatic cylinder that provide for the unconditional expediency of using the circuit solutions proposed at the assigned load.

4. Materials and methods of research into the energy-saving circuits of pneumatic actuators

When constructing a mathematical model for pneumatic actuator, we used thermodynamic dependences of the body with variable mass in the form of equations of thermal balance for the cavities of pneumatic actuator. Equations were brought to the dimensionless form employing the procedure, which makes it possible to reduce the number of independent parameters to a minimum. In this case, the remaining independent dimensionless complexes were the criteria of dynamic similarity. This allowed us, during computer study, to apply a small number of calculations, to cover the entire area of pneumatic actuator existence. The equations were brought to the Cauchy form with their subsequent numerical integration by the stepwise Runge-Kutta method of fourth order of accuracy.

Experimental data were read out by the oscillographing method. To measure the pressure, we used the membrane strain pressure sensors DD, which in conjunction with the strain amplifier made it possible to convert pressure into the proportional electrical signal. The calibration of sensors was conducted on the model oil calibration press.

5. Basic results of research into structure and parameters of the energy-saving pneumatic actuator

In order to solve the set tasks, it is necessary to implement new radical and reliable means of braking and positioning, capable of not only ensuring the shockless operation of PA at large inertia load, but also of attaining effective energy saving. Such methods of braking are the methods, based on the change in the structure of commutation relations [10].

The proposed method of braking, which provides for the energy-saving mode, is assessed in comparison with the traditional method of throttle braking, which is commonly employed (Fig. 1).

A circuit in Fig. 1 is based on the use of brake valve (positions 2, 3, 4) with a discrete change in exhaust resistance, which is manufactured in series and is widely used in power pneumatics.

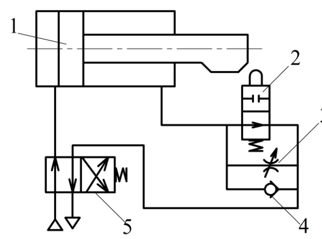

Fig. 1. Braking of PA OU with the help of discrete brake valve: 1 — pneumatic cylinder; 2, 5 — pneumatic distributors; 3 — adjustable throttle; 4 — non-return valve

Fig. 2 shows result of the numerical integration in dimensional form for PA, which uses a discrete brake valve. The examined PA consists of pneumatic cylinder (PC) with parameters: D_c=50 mm, d_s=16 mm, L=500 mm, effective to the area of feed and discharge pneumatic lines $f_1^e = f_2^e = 0{,}3 \cdot 10^{-4}$ m^2. Parameters of load P=100 N, m=50 kg.

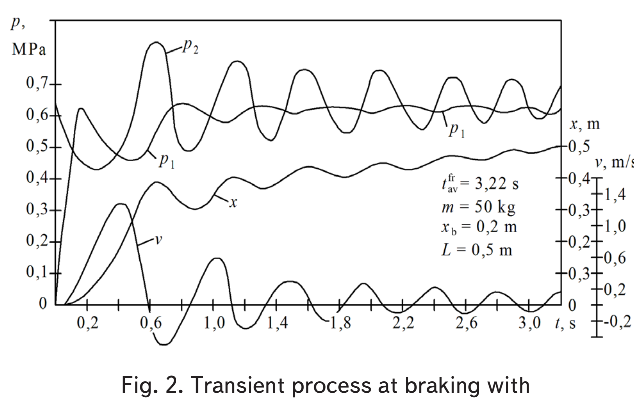

Fig. 2. Transient process at braking with the help of discrete brake valve

At such load, the realization of throttle braking leads to the occurrence of a developed oscillating process, longer response time, which makes it impossible to operate PA, especially when solving a problem on the precise positioning of operating unit.

The second essential drawback is the unproductive consumption of compressed air. At the PA parameters, which correspond to the transient process in Fig. 2, mass amount of compressed air per one cycle M=0,0085 kg.

In the research into energy characteristics, we employed as the consumed energy the working capacity (exergy) of compressed air. That is, the maximum useful work, which could be obtained from a thermodynamic system as a result of its reversible transition into the state of mechanical and thermal equilibrium with the environment [11]. It is demonstrated that, if in a receiver and refrigerator of compressor the compressed air reaches thermal equilibrium with the environment, then specific working capacity can be determined by expression [12].

$$l_c \cong RT_p \ln\frac{p_p}{p_a} = U + \frac{RT_p}{p_a}\left(p_p - p_a\right), \quad (1)$$

where $RT_p\left(p_p - p_a\right)/p_a$ is the specific work of ejecting (forcing), which is conventionally designated as transit working capacity; U is the work of isothermal expansion (compression) or potential energy of compressed air; T_M, r_m are the parameters of compressed air in PA supply main line; R is the gas constant; p_a is the atmospheric pressure.

An analysis of all components in power consumption revealed that the basic sources of losses in the working capacity of compressed air are:

1) losses due the incompleteness of air expansion in the working cavity and because of mismatch between actual expansion indicator and the ideal (isothermal);

2) losses in the dead space;

3) losses for throttling;

4) additional losses to fix OU;

5) losses, related to the work on ejecting compressed air from the exhaust cavity.

The first source of losses in connection to the actuators with throttle braking is because they are related to the drives with complete filling of the working volume. In such drives only transit working capacity of compressed air (1) is employed – the compressed air appears to be a simple kinematic component, which connects the body of displacement of compressor with the PC piston. Potential energy of compression U (1) is not used practically. Furthermore, in the actuators with throttle braking, kinetic energy of movable elements during braking is converted into heat and is lost irreversibly. That is why it is expedient to use the non-dissipative method for damping the kinetic energy of movable elements, which makes it possible to utilize recovery into the power line of braking energy.

When designing PA with braking by changing the structure of commutation relations, there appears the possibility to ensure the most rational, from the point of energy saving and maximum performance speed, commutation relations to each phase of OU motion, namely:

– piston fixation in late stroke is achieved with the help of the minimally required drop in the piston pressure, coordinated with the built-in load. In this case, lower pressure corresponds to atmospheric p_a, and air with the higher pressure p_v is selected from the output of reduction valve;

– in the acceleration phase, the working cavity, commutated during fixation with atmosphere, is connected to the main feed line (pressure p_p), while the exhaust cavity, commutated during fixation with the source of reduced pressure, is connected to the atmosphere. High operation speed is ensured due to this. In this case, the unproductive work of air ejection from the exhaust cavity becomes minimum;

– in the braking phase, working cavity through a preliminarily closed non-return valve should be connected to the source of reduced pressure (the output of reduction valve), while the exhaust cavity through a preliminarily closed non-return valve is connected to the main feed line. This enables a better realization of the working capacity of compressed air as a result of using potential energy of compression (expansion) U (1), as well as the braking energy recuperation into main feed line. In this case, a constant drop in the piston pressure is ensured at breaking (equally slowed braking mode with the possibility to change the magnitude of deceleration).

A diversity scheme, which reflects this algorithm, is represented in Fig. 3.

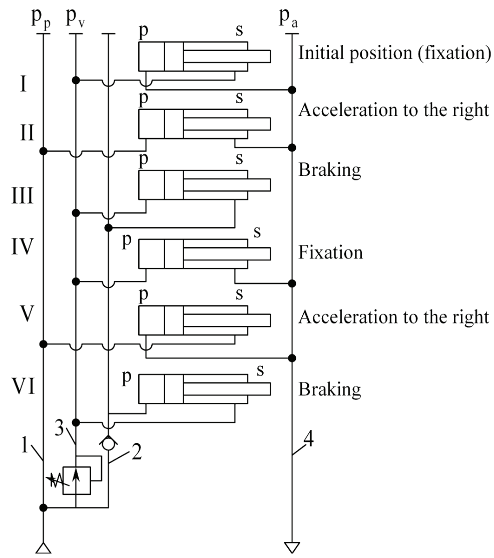

Fig. 3. Diversity scheme of the energy-saving structure of pneumatic actuator

The hardware realization of this scheme, based on 3/2 pneumatic distributors, is shown in Fig. 4, and the appropriate phases of distributor starts are described in Table 1. When using a five-line 3-position pneumatic distributor (at present, such distributors are available commercially in the product lines from most manufacturers), the scheme takes a more compact form (Fig. 5). The appropriate phases of distributor starts are described in Table 2.

In order to determine objectively the region of the most rational utilization of the proposed methods of braking, it is necessary to develop a mathematical model (MM) in the dimensionless form based on the formalized method for differential equation normalization.

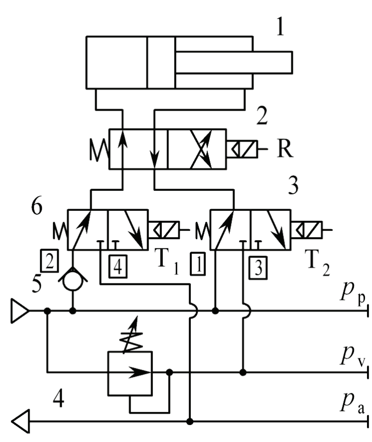

Fig. 4. Energy-saving circuit of pneumatic actuator based on 3/2 pneumatic distributors

Table 1

Phases of control valves motion in pneumatic distributors that correspond to the circuit of pneumatic actuator in Fig. 4

Phases of motion		T_1	T_2	R
Motion to the right	Initial position	1	1	0
	Acceleration	1	0	1
	Braking	0	1	1
	Fixation	1	1	1
Motion to the left	Initial position	1	1	1
	Acceleration	1	0	0
	Braking	0	1	0
	Fixation	1	1	0

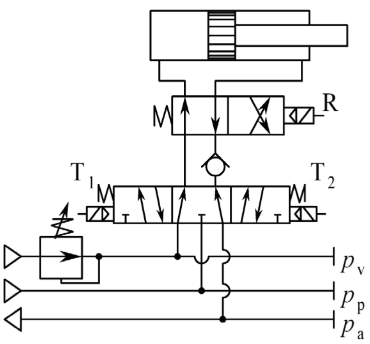

Fig. 5. Energy-saving circuit of pneumatic actuator based on 5/3 pneumatic distributor

Table 2

Phases of control valves motion in pneumatic distributors that correspond to the circuit of pneumatic actuator in Fig. 5

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

Underlying this method is the principle of maximum simplification in the structure of differential equations for MM when the number of independent variables is reduced to the minimum number of criteria of dynamic similarity [13, 14].

Basic similarity criteria:

$$\beta = \frac{mL}{t_b^2 F_1 p_p},$$

β is the criterion of inertness (dimensionless mass), numerically equal to the ratio of inertial force at basic acceleration L/t_b^2 to the maximum force, developed by piston; $\chi = P/p_p F_1$ is the parameter of static load; $\Omega = f_2^e / f_1^e$ is the relative effective area of exhaust circuit. Here $t_b = \frac{F_1 L}{f_1^e a_p}$ is the basic unit of time, numerically equal to the magnitude of filling the working volume of pneumatic cylinder (PC) by compressed air that moves at the speed of sound $a_p = \sqrt{kRT_p}$ through the opening, equal to the effective area f_1^e of feed tract, L is the full piston stroke; F_1 is the piston area from the side of rodless cavity.

A mathematical model in the dimensionless form, constructed on the equations of thermal balance with the application of these principles takes the form [13]:

$$\begin{cases} \dfrac{d\sigma_1}{d\tau} = \dfrac{k}{\xi_{01}+\xi}\left[s_1 \cdot z_1 \cdot \mathrm{sign}(\sigma_{p1}-\sigma_1)\cdot\varphi(I_1) - \sigma_1 \cdot \dfrac{d\xi}{d\tau}\right]; \\ \dfrac{d\theta_1}{d\tau} = \dfrac{\theta_1}{\sigma_1}\cdot\dfrac{d\sigma_1}{d\tau} + \dfrac{\theta_1}{\xi_{01}+\xi}\cdot\dfrac{d\xi}{d\tau} - \mathrm{sign}(\sigma_{p1}-\sigma_1)\cdot s_1' \cdot z_1 \cdot \dfrac{\varphi(I_1)}{\xi_{01}+\xi}; \\ \dfrac{d\sigma_2}{d\tau} = -\dfrac{k}{\xi_{02}+1-\xi}\left[s_2 \cdot z_2 \cdot \mathrm{sign}(\sigma_2-\sigma_{p2})\cdot\dfrac{\varphi(I_2)}{\Pi_{21}^F} - \sigma_2 \cdot \dfrac{d\xi}{d\tau}\right]; \\ \dfrac{d\theta_2}{d\tau} = \dfrac{\theta_2}{\sigma_2}\cdot\dfrac{d\sigma_2}{d\tau} - \dfrac{\theta_2}{\xi_{02}+1-\xi}\cdot\dfrac{d\xi}{d\tau} - \mathrm{sign}(\sigma_2-\sigma_{p2})\cdot\dfrac{s_2' \cdot z_2 \cdot \varphi(I_2)}{\Pi_{21}^F\cdot(\xi_{02}+1-\xi)}; \\ \dfrac{d\xi}{d\tau} = \dot{\xi}; \\ \dfrac{d\dot{\xi}}{d\tau} = \dfrac{1}{\beta}\left(\sigma_1 - \sigma_2 \cdot \Pi_{21}^F - \chi\right). \end{cases} \quad (2)$$

In the process of numerical integration of differential equations, in addition to determining the parameters of internal transient processes (dimensionless pressures and temperatures in the PC cavities $\sigma_1 = p_1/p_p$, $\sigma_2 = p_2/p_p$, $\theta_1 = T_1/T_p$, $\theta_2 = T_2/T_p$) and the variables of state $\xi = \frac{x}{L}$, $\dot{\xi} = \frac{\dot{x}}{L} t_b$ (displacement and OU), we determine relative mass amount of compressed air, consumed in the process of one operation $\bar{M}$ and averaged over the efficiency cycle η_{cp}:

$$\bar{M} = \int_0^\tau G\, d\tau \Big/ \left(F_1 L \rho_p\right) = \int_0^\tau \varphi(I_1)\, d\tau, \quad (3)$$

where G is the mass flow rate, which enters PC; $F_1 L \rho_p$ is the mass amount of compressed air, required to fill the working volume; $\varphi(I)$ is the expenditure function:

$$\varphi(I) = \frac{1+\mathrm{sign}(I-0{,}528)}{2}\sqrt{\frac{2}{k-1}\left(I^{2/k} - I^{(k+1)/2}\right)} + $$
$$+0{,}579\frac{1-\mathrm{sign}(I-0{,}528)}{2}. \quad (4)$$

Ratio of pressures on supply and discharge line:

$$I_1 = \left(\frac{\sigma_1}{\sigma_{p1}}\right)^{\mathrm{sign}(\sigma_{p1}-\sigma_1)};$$

$$I_2 = \left(\frac{\sigma_2}{\sigma_{p2}}\right)^{\mathrm{sign}(\sigma_{p2}-\sigma_2)}. \quad (5)$$

Values of pressures in the objects of commutation for the left and right cavities of the cylinder σ_{p1} and σ_{p1} are calculated depending on the values of function of connecting the objects of commutation y_1 and y_2: $\sigma_{p1} = y_1/p_p$; $\sigma_{p2} = y_2/p_p$:

$$y_1=\bar{T}_1Rp_v+\bar{T}_2R\left(p_a\frac{1+\operatorname{sign}(p_1-p_a)}{2}+p_1\frac{1+\operatorname{sign}(p_a-p_1)}{2}\right)+$$

$$+\bar{T}_1T_2R\left(p_p\frac{1+\operatorname{sign}(p_1-p_p)}{2}+p_1\frac{1+\operatorname{sign}(p_p-p_1)}{2}\right)+T_1\bar{T}_2Rp_p;$$

$$y_2=\bar{R}\bar{T}_1p_v+\bar{T}_2R\left(p_a\frac{1+\operatorname{sign}(p_2-p_a)}{2}+p_2\frac{1+\operatorname{sign}(p_a-p_2)}{2}\right)+$$

$$+\bar{T}_1T_2R\left(p_p\frac{1+\operatorname{sign}(p_2-p_p)}{2}+p_2\frac{1+\operatorname{sign}(p_p-p_2)}{2}\right)+T_1\bar{T}_2Rp_p. \quad (6)$$

For PA with braking in late stroke, the displacement of specific mass from one point of space into another is a basic function. In order to estimate the degree of energy perfection of such PA, when computing efficiency (η_{cp}), we considered as useful work:

1) work to overcome the force of static resistance $P\cdot L$;

2) kinetic energy of the movable parts, averaged over cycle

$$\eta_{cp}=\frac{\frac{\beta}{2\tau'}\int_0^{\tau'}\dot{\xi}^2d\tau+\chi}{z_1\left(\ln 1/\sigma_a\right)\cdot\int_0^{\tau_c}\varphi\left(I_1\right)d\tau}, \quad (7)$$

where τ is the dimensionless time of motion of PC operating unit (OU) from one position into another; τ is the total response time; $\sigma_a=p_a/p_p$; z_1, z_2 are the corrections, which consider a discrete change in the throughput of the feeding and exhaust tracts at changing the structure of commutation relations.

Thus, the averaged value of PA efficiency can be calculated in the course of a general process of numerical integration of initial MM (1), extended due to the introduction of integrated parameters y and $\bar{M}$:

$$\begin{cases}\dfrac{d\bar{M}}{d\tau}=z_1\bar{R}\,\varphi\left(I_1\right)+\\ +R\left[z_1\varphi\left(I_1\right)\dfrac{1+\operatorname{sign}\left(\sigma_v-\sigma_1\right)}{2}-z_2\varphi\left(I_2\right)\dfrac{1+\operatorname{sign}\left(\sigma_2-1\right)}{2}\right];\\ \dfrac{dy}{d\tau}=\dfrac{\xi\beta}{2}.\end{cases} \quad (8)$$

After PA completed an operation cycle and we obtained, based on the numerical integration of system (2) and (8), values τ', y and $\bar{M}$, the PA efficiency, averaged over cycle, is calculated:

$$\eta_{cp}=\frac{y/\tau'+\chi}{\bar{M}\ln 1/\sigma_a}. \quad (9)$$

Fig. 6 shows a transient process in PA at OU braking by changing the structure of commutation relations with the energy-saving operation mode (Fig. 5) at the same parameters of PA as for the case of throttle braking. A difference is in the fact that the mass load is brought to m=200 kg. Feed pressure p_m=0.6 MPa, pressure of the reduction valve (RV) setting p_v=0.2 MPa.

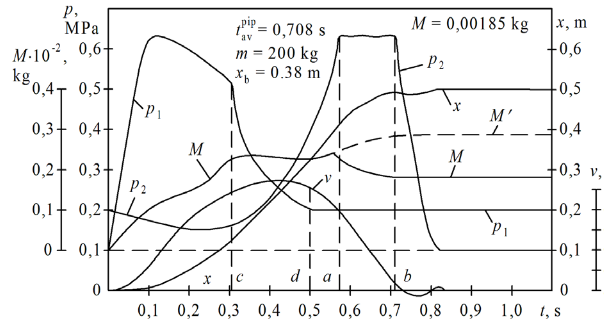

Fig. 6. Transient process in the pneumatic actuator that operates according to circuit in Fig. 5

There occurred a cardinal change in the very form of the transient process in comparison with the case of throttle braking (Fig. 2), and it approached the optimum form when a change in the OU speed is close to the cycloidal law. In this case:

– operation speed sharply increased due to the optimization of transient process (response time was reduced by 2.5 times in comparison with the variant of throttle braking);

– the transient process became non-oscillating with the equally-retarded braking mode; moreover, the magnitude of deceleration can be regulated by the pressure of RV setting;

– at the average and large inertia load, the regime of effective energy saving is realized, as a result of which consumption of compressed air under given conditions of functioning were lowered to 0,00185 kg per one operation ($\bar{M}=0,217$), that is, they were reduced by 4.6 times in comparison with throttle braking;

– in the time interval c–d (Fig. 6), the work of compressed air in the left PC cavity is carried out through the work of expansion U (1);

– compressed air recuperation from the brake cavity into the feed line is conducted in the time interval a–b. Curve M′ reflects consumption of the compressed air without regard to recuperation. Curve M corresponds to the consumption of compressed air, considering the recuperation.

6. Discussion of results: the area for rational utilization of energy-saving circuits and their parametric synthesis

Determining the area for rational utilization of energy-saving circuits (Fig. 4, 5) is based on the isolation of boundaries in the space of criteria of dynamic similarity β and χ (Fig. 7). The region of the most rational utilization of energy-saving circuits is located within the range $2\le\beta\le5$ and $\chi\le0,3$, where it is possible to decrease the consumption by 4|10 times in comparison with traditional pneumatic actuators with throttle braking and with the full filling of working volume. In this case, the region of effective braking substantially expands up to β=5, that considerably overlaps the area of applying PA with traditional throttle braking (β<0.3).

A parametric synthesis of energy-saving PA includes the following stages:

– the charts on a change in the dimensionless amount of compressed air, consumed by PA, are built in the space of basic criteria of dynamic similarity β and χ (Fig. 7);

– the region of existence of energy-saving PA (a b c d) is isolated in the plane of basic criteria of dynamic similarity $\beta-\chi$, where $\bar{M}\in\left(0,25\div0,1\right)$ (Fig. 8);

– based on determining the boundary values β and χ, passing from the dimensionless form to the dimensional, we define the region of permissible values of the piston diameter, which, at the assigned load, makes the use of this PA circuit unconditionally expedient (10).

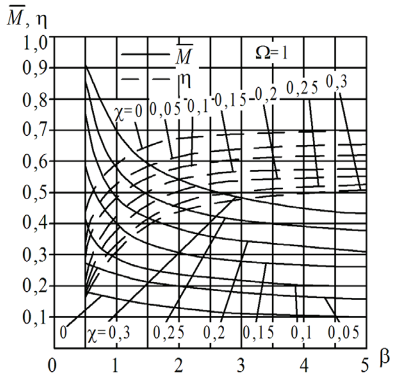

Fig. 7. Dependence of relative mass of consumed air $\overline{M}$ and the averaged efficiency η on the criteria β and χ for the energy-saving circuits

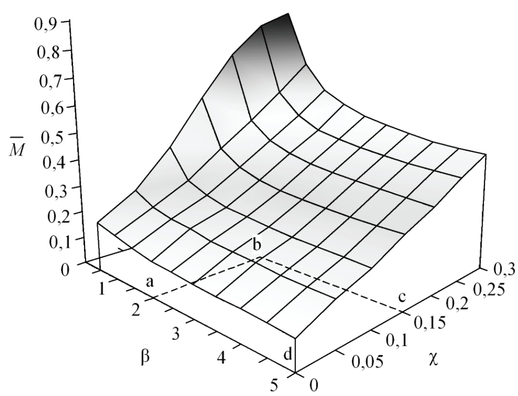

Fig. 8. Determining the area for rational utilization of energy-saving PA

Based on the dimensions of the area of rational utilization (S_{abcd}), diameter of pneumatic cylinder at the assigned load (P, m), stroke length (L), as well as effective area of the pneumatic feed line (f_1^e), is found from inequalities:

$$\sqrt{\frac{4}{\pi}\sqrt[3]{\frac{m\left(f_1^e\right)^2 kRT_p}{2Lp_p}}} \geq D \geq \sqrt{\frac{4P}{0,15\,\pi p_p}}. \tag{10}$$

In the experimental research into proposed methods of braking, we used the long-stroke cylinder made by SMC (Japan) with the precision stopper C92LAB50-1000P (D_p=50 mm; L = 1000 mm; d_s = 20 mm), put on the lathe bed. As the loading device, we used a cart with loads, which could move along the bed's guides. The load on the cart could be collected in the form of flat kettlebells with mass up to 20 kg up to the load in 400 kg (mass of cart was 10 kg).

Fig. 9, 10 show oscillograms of the transient process, obtained at loading the actuator with mass of 30 kg (medium load) and 190 kg (large load), respectively.

The oscillograms received confirm the fact that at braking according to the proposed circuits (Fig. 4, 5) with the help of reduction valve and non-return valve, connected to feed line, it is possible to maintain a constant drop in the piston pressure in the braking mode. This makes it possible to create constant deceleration.

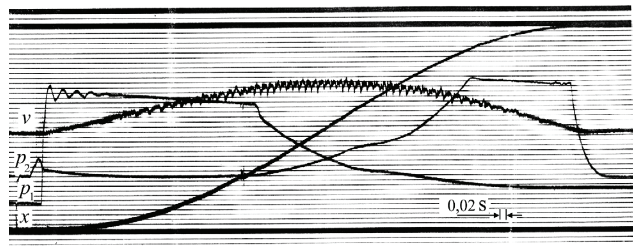

Fig. 9. Oscillogram of transient process for the energy-saving circuit (Fig. 5) at rod extension (p_p=0.45 MPa; p_v=0.1 MPa; x_{01}=x_{02}=0,08 m; m=30 kg, L=1000 mm)

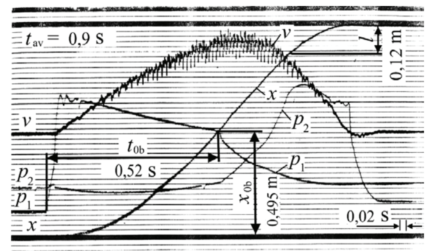

Fig. 10. Oscillogram of transient process for the energy-saving circuit (Fig. 5) at rod extension (p_p=0.45 MPa; p_v=0.1 MPa; x_{01}=x_{02}=0,08 m; m=190 kg, L=1000 mm)

In this case, the nature of velocity change is close to the most rational cycloidal without the signs of oscillation. This is distinctly evident both in the oscillograms of PA with a medium load (Fig. 9) and in the oscillograms of PA with the large inertia load. Moreover, with an increase in the load, the braking mode becomes more stable, which indicates a significant expansion of the area for applying pneumatics towards large inertia loads.

The oscillograms also show the sections, which testify to the effective use of potential expansion energy of compressed air in the working cavity of PC, as well as recuperation into the network of the compressed air from the brake cavity. This is the confirmation of essential improvement in the energy characteristics of PA.

7. Conclusions

1. The choice of criterion for evaluating the degree of energy perfection of pneumatic actuator is performed; basic components of unproductive power consumption are determined based on computer simulation. It is established that they include:

– losses due to the incompleteness of air expansion in the working cavity;

– losses in the dead space;

– losses for throttling;

– additional losses for the fixation of operating unit;

– losses, related to the work to eject compressed air from the exhaust cavity.

2. Requirements for the energy-saving structure of pneumatic actuator are formulated: the use of diversity scheme, which enables the most rational commutation relations for each motion phase. Two energy-saving circuits are designed with the use of commercially available pneumatic equipment.

3. A mathematical model of pneumatic actuator is constructed, underlying which are the thermodynamic dependences of the body of variable mass. The model takes the

form of a system of differential equations in the dimensionless form and makes it possible to analyze not only the dynamic, but also energy characteristics. The dimensionless form of this model allowed us to minimize the number of independent variables and to obtain criteria of dynamic similarity.

4. Computer simulation of transient processes made it possible to solve the problem on determining the region of the most rational utilization of the proposed circuits of pneumatic actuators. The chosen region in the plane of criteria of dynamic similarity, where it is possible to decrease power consumption by 4÷10 times, was used as the basis, and through the return from the dimensionless region into dimensional, we realized parametric synthesis of energy-saving actuator.

5. Based on the dimensions of the area of rational utilization of energy-saving pneumatic actuator, we obtained analytical expression for selecting the diameter of pneumatic cylinder, which, at the assigned load, enables unconditional positioning of actuator in the zone of the most effective energy saving. The structure of this expression is the inequality, which considers the assigned load of actuator, stroke length, as well as the effective area of pneumatic feed line.

It is demonstrated that the application of the proposed schemes for braking of operating unit makes it possible to increase permissible inertia loads for actuator by an order of magnitude in comparison with the pneumatic actuators, where traditional throttle braking is employed.

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