Визначена структура пневмопривода для автоматизації логістичних комплексів, що ґрунтується на зміні комутаційних зв'язків, яка забезпечує ефективне використання працездатності стисненого повітря. Проведений кількісний і якісний аналіз втрат енергії у пневмоприводі. Виділена область застосування енергозберігаючої схеми пневмопривода, яка дозволяє реалізувати інерційні навантаження, що у 10 разів вищі допустимих навантажень при дросельному гальмуванні, і підвищити надійність його роботи

Ключові слова: пневмоприводи, гальмування, структура комутаційних зв'язків, збільшення інерційних навантажень

Определена структура пневмопривода для автоматизации логистических комплексов, основанная на изменении коммутационных связей, обеспечивающая эффективное использование работоспособности сжатого воздуха. Проведен количественный и качественный анализ потерь энергии в пневмоприводе. Выделена область применения энергосберегающей схемы пневмопривода, которая позволяет реализовать инерционные нагрузки, в 10 раз превышающие допустимые нагрузки при дроссельном торможении, и повысить надежность его работы

Ключевые слова: пневмоприводы, торможение, структура коммутационных связей, увеличение инерционных нагрузок

UDC 621.5
DOI: 10.15587/1729-4061.2018.126635

IMPROVING POWER EFFICIENCY OF PNEUMATIC LOGISTIC COMPLEX ACTUATORS THROUGH SELECTION OF A RATIONAL SCHEME OF THEIR CONTROL

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 National Technical University «Kharkov Polytechnic Institute»
Kyrpychova str., 21, Kharkiv, Ukraine, 61002

1. Introduction

When solving the problem of automating handling operations in warehouses, portal type manipulators are increasingly being used. In implementation of translational motion in manipulators of this type, the use of rodless long-stroke pneumatic cylinders is the most rational [1]. In contrast to conventional cylinders, they provide ultimate compactness to the manipulator, have high radial rigidity and act as guides. It is also necessary to take into account that pertaining to lump-sum costs, this is the cheapest type of actuators characterized by simplicity, environmental friendliness and convenience of transmission and utilization of energy carriers. On the other hand, compressed air itself is one of the most expensive energy carriers. Therefore, when using pneumatic actuators in logistics complexes, solving the following problems is relevant:

1) energy saving and ensuring smooth shock-free braking;
2) positioning of the actuator work member in conditions of rather high inertial loads.

2. Literature review and problem statement

The progress in the field of automation of production processes features an increasing spread of pneumatic automation means. It is noted in [2] that production of compressed air in industrialized countries accounts currently for about 10 % of their total energy balance. Despite the fact that compressed air is one of the most expensive energy carriers, the energy saving issue of this carrier remains one of the least studied. Various methods of saving compressed air are proposed in [2, 3]. However, among them, there is not a single method associated with selection of a rational way of braking and positioning of the working member of the pneumatic motor from the point of view of energy saving. One way to save compressed air for the entire enterprise is proposed in [4]. It involves refusal to use a single compressor and the synchronization of activation and deactivation of a group of compressors depending on consumption of compressed air in the enterprise network. However, the issue of energy efficiency of each individual consumer (pneumatic actuator) is not given attention. In contrast tp previous works, the problem of improving energy efficiency of an individual pneumatic actuator is being solved in [5]. This is achieved by replacing the standard four-line pneumatic distributor with two three-line distributors and selecting a rational program of their control. Shortcomings of the work include absence of theoretical studies based on mathematical modeling. This excludes the possibility of generalization and determining the scope of rational use of this solution. It is noted in work [6] that the problem of energy saving is necessarily solved in all newly created projects in the last 20 years. However, with regard to pneumatic systems, this is not so easy to do because of complexity of the processes taking place in the pneumatic actuators. Therefore, it is proposed to use mathematical modeling when solving the issue of energy saving in these systems. However, it should be noted that the model proposed in this work is based on outdated representations, in particular, on the assumption of isothermal nature of thermodynamic processes. At present, it is well known (and it was proved) that a polytropic process with a variable polytropic index for describing operation of the pneumatic actuator is the most proper process. Paper [7] draws attention to the fact that irrational use of the increased pressure level leads to the growth of power input. It is suggested that in a necessity, pressure to the consumer should be reduced or, conversely, use of a local device similar to a hydraulic multiplier is advisable if a higher pressure is required. However, there is no proper theoretical solution, which makes it possible to choose rational size of such devices.

There is a relatively small number of publications in which the issue of energy saving is associated with the process of braking and positioning of the work member in the pneumatic actuator. Conventional methods of braking are based on the use of external or internal throttling devices, that is, the methods belonging to exclusively dissipative methods of braking when the kinetic energy of moving parts is transformed to thermal energy [8].

In addition to low energy efficiency, such methods suffer from a lack of operational flexibility which makes it difficult to use them in present-day mechatronic systems [8, 9]. The methods of braking based on the change in the structure of commutation links where throttling devices are not used enable expanding of the scope of application of power pneumatics in a direction of a significant increase in inertial load. Also, due to the use of these braking methods, it is possible to achieve much more effective realization of operational availability (exergy) of compressed air [10]. The authors of [5] drew attention to the important feature of non-dissipative braking through switching of commutation contacts, the possibility of using braking energy in a form of potential energy of compressed air in the brake chamber. This energy is then recuperated into the network or used to reverse the work member. However, the material presented in [5] does not contain an analysis of the field of application and a study of the qualitative and quantitative nature of energy losses.

3. The aim and objectives of the study

This work objective was to develop and substantiate structure of the pneumatic actuator (PA) which ensures efficient use of compressed air working capacity and determine the fields of rational use of the energy saving scheme of the PA based on qualitative and quantitative analysis of energy losses in the PA.

To achieve the objective, the following tasks were set:

– substantiate the most rational structure of commutation links for all phases of motion of the PA work member which ensure minimization of nonproductive power inputs;

– develop an energy-saving scheme of the PA and an algorithm of its control;

– define and quantify all components of power inputs in operation of a discrete PA;

– conduct computer simulation on the basis of the developed mathematical model to define the field of the most rational use of the energy saving scheme of the PA.

4. Materials and methods for studying effectiveness of the pneumatic actuator by selection of a rational scheme of braking and positioning of the work member

When braking and positioning the PA work members with a large inertial load, the braking methods based on the change in the structure of commutation links are increasingly used. The advantage of these braking methods is the ability to realize the most rational commutation links for all phases of motion of the work member of the PA. As a result, it becomes possible to provide the most favorable braking law and realize a shock-free operation of the PA with a large inertial load. Also, the most complete use of compressed air energy is possible in this case.

When creating a pneumatic control system (braking) by changing the structure of commutation links, two diametrically opposite approaches are possible. The first is focused on minimizing lump-sum costs, that is, it requires a minimum of apparatuses to implement a sufficiently effective braking of the PA work member. The second is aimed at minimizing operating costs, that is, it provides a minimum of compressed air for a shock-free high-speed operation of the PA.

As a basic (elementary) structure of the PA with braking by changing the structure of commutation links, consider a PA with two three-linear distributors (Fig. 1). Despite the presence of two pneumatic distributors, the set of commutation links for ensuring radical braking is small and is actually limited to only two variants (Table 1). The state of the control electromagnets is described by the Boolean variables T_1 and T_2 (1 for electric signal given, 0 for no electrical signal).

Along with a reliable braking effect, this scheme provides high speed and requires a minimum of devices, so it is often considered as an effective scheme of the PA with large and medium inertial loads.

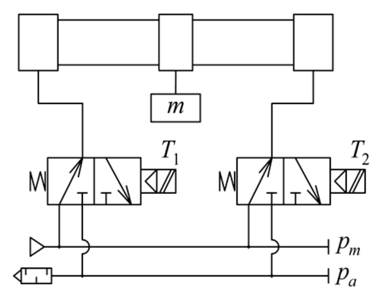


Fig. 1. Basic (elementary) scheme of the pneumatic actuator (scheme No. 1): m is mass of moving parts of the PA; p_m is pressure in the feed line of the PA; p_a is atmospheric pressure; T_1, T_2 are Boolean variables determining the states of the control electromagnets ($T=1$ for current on, $T=0$ for current off)

Table 1

Map of states of electromagnets of valves for scheme No. 1

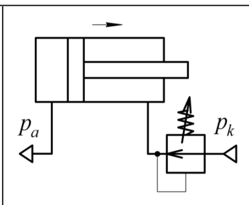
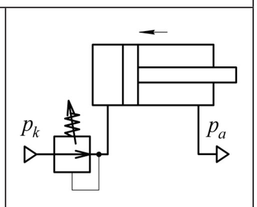
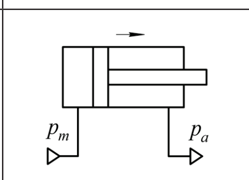
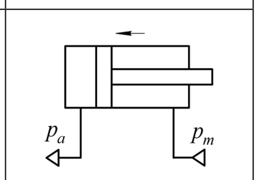
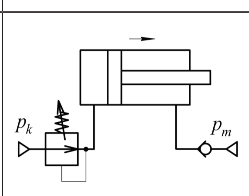
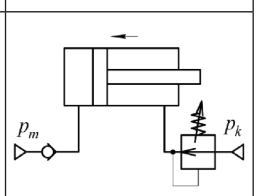
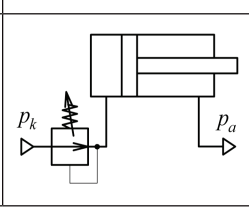
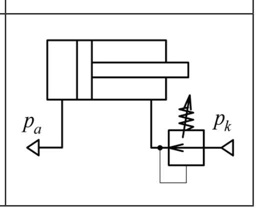
Situation		Electro-magnet state		Situation		Electro-magnet state	
		T_1	T_2			T_1	T_2
Motion to the right	acceleration	0	1	Motion to the left	acceleration	1	0
	braking	1	0		braking	0	1
	fixation	0	1		fixation	1	0

4. 1. Structural synthesis of an energy-saving PA scheme

The principle of synthesizing an energy-saving PA scheme consists in that the most rational switching situations in terms of energy saving and maximum performance must correspond to each motion phase. Represent them in a table form (Table 2).

Table 2

Commutation situations for each phase of the PA motion

Motion to the right		Motion to the left	
Fixation of the left position		Fixation of the left position	
Acceleration		Acceleration	
Braking		Braking	
Fixation of the right position		Fixation of the right position	

The energy-saving scheme of the PA with braking by changing the structure of commutation links ensuring implementation of all switching situations of Table 2 is shown in Fig. 2 and the map of rational control of distribution valves of this scheme is given in Table 3.

For an objective comparison of effectiveness of these methods of the PA control, a universal mathematical model has been developed in a dimensionless form with distinguishing the main criteria for dynamic similarity [10].

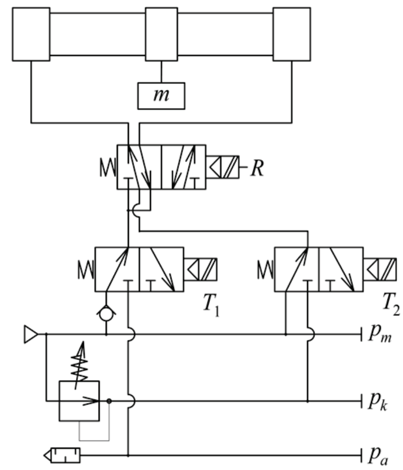


Fig. 2. Energy-saving scheme of the pneumatic actuator (scheme No. 2)

Table 3

Map of states of distributor electromagnets for scheme No. 2

Situation		Electro-magnet state			Situation		Electro-magnet state		
		T_1	T_2	R			T_1	T_2	R
Motion to the right	accelera-tion	1	0	0	Motion to the left	accelera-tion	1	0	1
	braking	0	1	0		braking	0	1	1
	fixation	1	1	0		fixation	1	1	1

4. 2. The criteria of dynamic similarity and analysis of power inputs in the PA with braking by changing the structure of commutation links

The following is used as the main criteria of dynamic similarity:

– the inertance criterion (dimensionless mass) $\beta = \dfrac{mL}{t_b^2 p_m F_1}$ is numerically equal to the ratio of the inertia force of the moving parts of the PA at basic acceleration L/t_b^2 to the maximum force developed by the piston, $p_m F_1$ (where p_m is the feed pressure, F_1 is the piston area, L is a complete stroke of the piston; t_b is the basic time unit);

– static load parameter $\chi = P/p_m F_1$;

– the basic time unit $t_b = \dfrac{F_1 L}{f_1^e \sqrt{kRT_m}}$ numerically equal to the time of filling of the cylinder working volume ($F_1 L$) by air moving at the speed of sound ($a_m = \sqrt{kRT_m}$) through an opening equal to the effective area of the intake path f_1^e.

The procedure of rationing differential equations describing the work processes in the PA is based on the use of dimensionless time $\tau = t/t_b$ and the criteria of dynamic similarity. This procedure allows one to reduce to a limit a large number of design parameters in the mathematical PA model replacing them with independent parameters that determine its dynamics.

To assess the degree of energy perfection of the PA, the concept of exergy, that is, specific operability is used [9]:

$$l_r = RT_m \ln \frac{p_m}{p_0} = U + \frac{RT_m}{p_m}(p_m - p_a). \qquad (1)$$

The first term on the right corresponds to the potential energy of expansion, the second term represents the specific work of pushing through, that is, the so-called transit working capacity.

Let us estimate the degree of energy perfection of the PAs loaded with a static and an inertial load with a complete braking of the work member at the end of the stroke. It is advisable to make this estimation with the aid of the efficiency (η_{cf}) averaged over the cycle and the dimensionless mass of compressed air consumed by the PA in one actuation ($\bar{M}$) [9]:

$$\eta_{cf}=\frac{\frac{\beta}{2\tau'}\int\limits_0^{\tau'}\dot{\xi}^2\mathrm{d}\tau+\chi}{\ln(1/\sigma_a)}, \qquad (2)$$

where τ' is the dimensionless time of piston motion from one position to another; $\dot{\xi}$ is dimensionless piston speed; $\bar{M}$ is a dimensionless quantity of compressed air consumed by the pneumatic actuator; σ_a is dimensionless atmospheric pressure:

$$\bar{M}=\frac{\int\limits_0^{\tau}G\mathrm{d}\tau}{F_1 L\rho_m}=\int\limits_0^{\tau}\varphi(I_1)\mathrm{d}\tau, \qquad (3)$$

where ρ_m is the density of air at its parameters in the feed line; G is the mass flow rate; τ is dimensionless time of complete PA actuation; $\varphi(I_1)$ is the consumption function:

$$\varphi(I_1)=\frac{1+sign(I-0.528)}{2}\sqrt{\frac{2}{k-1}\left(I^{2/k}-I^{(k+1)/k}\right)}+$$
$$+0.579\frac{1-\mathrm{sign}(I-0.528)}{2}, \qquad (4)$$

where $I_1=(\sigma_1/\sigma_{m1})^{\mathrm{sign}(\sigma_{m1}-\sigma_1)}$ is the ratio of pressures at the ends of the pipeline (σ_{m1} is the dimensionless pressure in the object of commutation of the work cylinder chamber, σ_1 is the dimensionless pressure in the work cylinder chamber).

Fig. 3 shows a comparative diagram of compressed air consumption for the pneumatic actuators operating according to the schemes in Fig. 1 (solid line) and in Fig. 2 (dotted line). The results were obtained for the PA at $\beta=5$, $\chi=0.1$, $\sigma_a=0.2$.

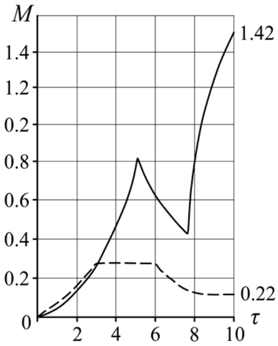


Fig. 3. Consumptions of compressed air in actuation of the PA with the base scheme No. 1 (solid line) and the energy-saving scheme No. 2 (dotted line)

The high consumption of compressed air for the scheme in Fig. 1 is a consequence of its simplicity when there are just two commutation situations for the PA having three phases of forward motion and three phases of backward motion.

The energy saving effect for the scheme in Fig. 2 consists in that the most rational commutation links are realized for each motion phase. In the phase of acceleration, only the transit working capacity of the incoming compressed air (1) is used. But since the initial pressure differential on the piston during fixation was small (p_k-p_a), nonproductive work of pushing compressed air from the exhaust chamber was minimal which also contributes to an increase in the speed of the PA. In the braking phase, the switching situation is such that the potential energy of air expansion in the working chamber starts to be used. At the same time, compressed air is recuperated from the exhaust (braking) chamber into the network after opening the return valve.

In the mode of fixation, the additionally compressed air from the network is not consumed more and the piston is retained by the minimum pressure differential (p_k-p_a).

The energy balance for the compressed air consumed by the PA in the course of one actuation can be represented by the following dependence:

$$E_s-E_r=R_1+E_{ie}+E_{lc}+E_{lv}+E_{th}+E_f, \qquad (5)$$

where E_s is the full working capacity of the compressed air consumed by the PA in the course of one actuation; E_{ie} is loss of working capacity of the compressed air because of incompleteness of expansion in the working chamber of the cylinder; E_{lc} is loss of working capacity when the real process of expansion does not correspond to the ideal (isothermal) process; E_{lv} is loss of working capacity in the dead volume of the cylinder; E_{th} is loss on throttling; R_1 is external mechanical work of compressed air; E_f is the loss associated with fixation of the work member by compressed air in the final position; E_r is working capacity of compressed air returned to the network as a result of recuperation.

A comparative diagram of components of energy consumption for schemes No. 1 and No. 2 (Fig. 4) was obtained under the same conditions as the diagram in Fig. 3. The values are given in a dimensionless form. Basic unit of energy (work) $E_b=p_m F_1 L$.

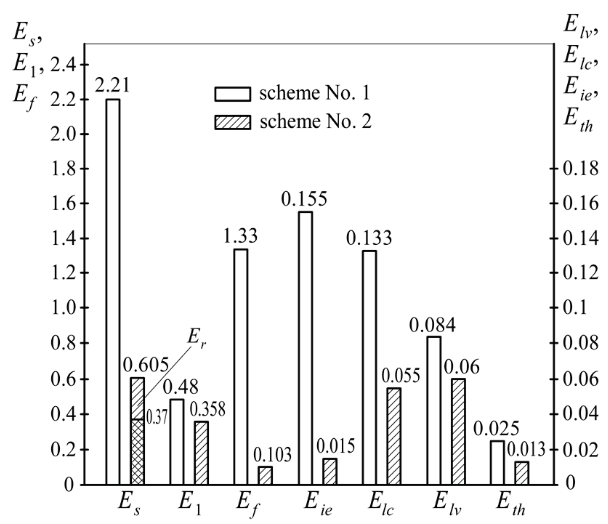


Fig. 4. Comparative diagram of energy consumption components for schemes No. 1 and No. 2

4. 3. Determination of the rational use of the energy-saving pneumatic actuator

To determine the scope of rational use of the energy-saving scheme, computer simulation of the work processes occurring in the PA has been carried out. The work processes reflect a different approach when solving the problem of braking the PA work members with medium and large inertial loads. The calculations cover a rather wide area of application of a PA represented by a space of criteria of dynamic similarity β and χ (Fig. 5, 6).

Fig. 5 shows dependence of the dimensionless response time τ, the dimensionless braking distance ξ_t, cycle mean efficiency η and the relative mass of compressed air on the criteria for dynamic similarity β and χ.

Analysis of the graphs in Fig. 5 shows that scheme No. 2 provides an unconditional reduction in power input in comparison with the scheme No. 1 in the whole region of existence of the PA. The most significant reduction in energy input is achieved when $\chi = 0 \div 0.1$, and when $\chi = 0.15 \div 0.3$, the decrease is significant only at a large inertial load $(\beta = 2 \div 5)$. At large values of χ $(\chi > 0.3)$ and small values of β $(\beta < 0.5)$, the use of the PA with an energy-saving structure becomes inexpedient because it does not result in a significant reduction in energy input. The decrease in the energy efficiency of the PA operating according to scheme No. 2 in this region is explained firstly by the lack of recuperation into the network because of a small braking distance and secondly by an insufficiently complete expansion of the compressed air in the working chamber. In addition, the nature of the dominant energy inputs itself is changing. In the phase of acceleration, the transit working capacity of the compressed air (the pushing operation) is mainly used when air is just a kinematic link between the compressor and the pneumatic cylinder.

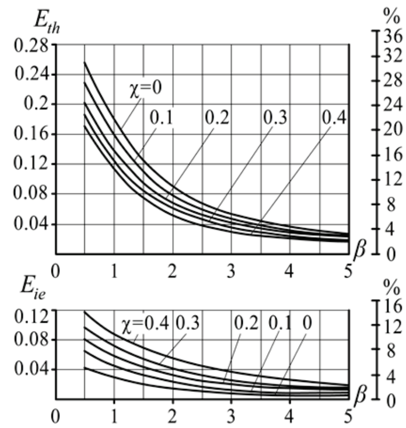


Fig. 6. Dependence of the loss of working capacity of compressed air for throttling (E_{th}) and because of incompleteness of air expansion in the working chamber (E_{ie}) on the β and χ criteria

The main type of loss of working capacity is the throttling loss. As the β parameter decreases (with a decrease in inertance), the piston speed increases and the braking distance is shortened. This increases the throttling losses and the losses connected with incompleteness of air expansion in the working chamber. The graphs reflecting growth of these losses with a decrease in β are shown in Fig. 6.

The increase in throttling losses (E_{th}) occurs according to an exponential law. For example, when $\chi = 0.1$ and β decreases from 5 to 0.5, there is an 11-fold increase in throttling losses making up $1/4$ of the entire working capacity of the compressed air flow, i. e. becoming the main loss item. The growth of losses because of incompleteness of air expansion is also significant and increases almost 3.5 times under the same conditions.

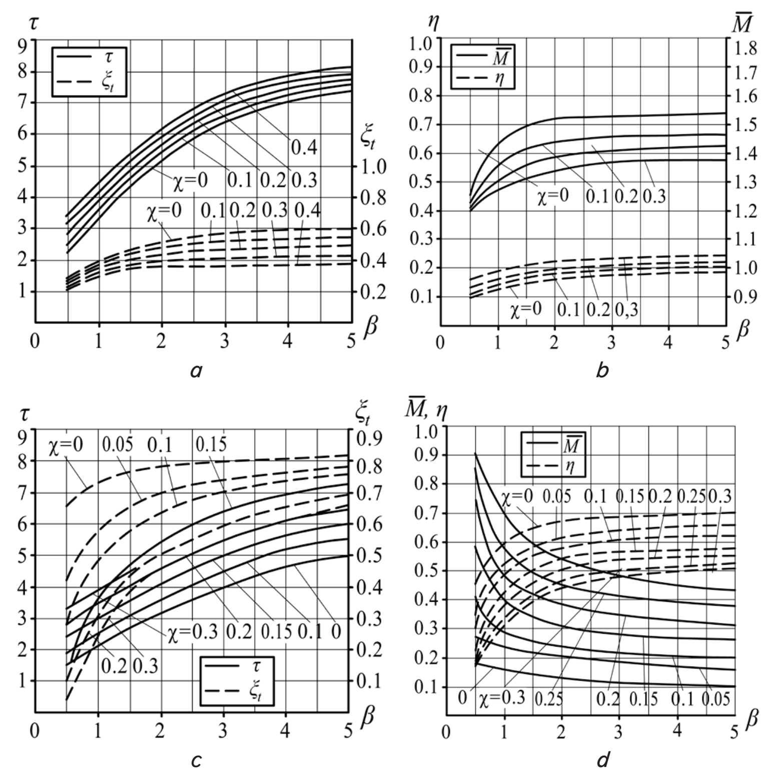


Fig. 5. Dependence of the main operational characteristics of the PA on the criteria of dynamic similarity β and χ: scheme No. 1 (*a*, *b*); scheme No. 2 (*c*, *d*)

The graphs in Fig. 5 allow us to make both qualitative and quantitative assessment of efficiency of the energy saving scheme. This makes it possible to compare lump sum and operating costs and make a substantiated decision.

Let, for example, a pneumatic cylinder with $D=100$ mm and $L=400$ mm at $p_m=0.5$ MPa and with an attached mass of $p_m=0.5$ kg overcomes the built-in static load of $P=390$ N and the effective area of the intake path $f_1^e=0.71\cdot 10^{-4}$ m^2. Under these conditions, $t_b=0.128$ s; $\beta=2.54$; $\chi\approx 0.1$. The relative mass of the consumed air in accordance with the graphs in Fig. 5 for the scheme No. 1 is $M_1=1.44$, and $M_2=0.22$, for the scheme No. 2, the basic unit of mass of air:

$$M_b=\frac{F_1 L p_m}{R T_m}=\frac{0.00785\cdot 0.4\cdot 5\cdot 10^5}{287\cdot 293}=0.0186 \text{ kg}.$$

The economy of compressed air for one cycle in the transition to an energy-saving scheme is:

$$\Delta\bar{M}=M_b\left(\bar{M}_1-\bar{M}_2\right)=$$
$$=(1.44-0.22)0.0186=0.0227 \text{ kg}.$$

At the lifetime of the pneumatic cylinder equal to $2\cdot 10^6$ actuations, the economy will be $M_s=0.0227\cdot 2\cdot 10^6=45400$ kg or 7643.1 m^3 of compressed air at $p=0.5$ MPa for the whole operation time.

If price for 1 m^3 of compressed air and cost of pneumatic devices are known, it is possible to compare operational and lump sum costs. Comparative analysis will make it possible to make a decision on the expediency of using an energy-saving actuator scheme.

5. Discussion of the results: the field of effective use of power pneumatics

In the case of conventional throttling braking, the process of braking the work member at an average and large inertial load ($\beta\geq 0.5$) is accompanied by an uncontrolled nature of the pressure changes in the cylinder chambers and the piston motion with a developed oscillatory process. The proposed braking scheme provides an optimal form of the transient process. A high braking effect is achieved due to a simultaneous pressure growth in the exhaust chamber and a pressure drop in the working chamber up to a constant negative pressure differential p_m-p_k at which there is a constant negative acceleration during braking. The level of this pressure differential depends on the adjustment of pressure p_k in the reducing valve. The greater the inertial load, the more stable this differential is maintained. Owing to this fact, the field of use of power pneumatics can be expanded by almost an order of magnitude in the direction of increase in the inertial load (up to the value of the inertance criterion $\beta=5$). Using such a scheme, it is possible to achieve much more efficient use of working capacity of the compressed air due to the fact that:

– in the phase of braking (Table 2, Fig. 2), not only the transit working capacity is used but also the potential energy of expansion of the compressed air in the working chamber (1) which is completely impossible in pneumatic actuators with a conventional throttle braking scheme and a full filling of the working volume;

– the braking energy, that is, compressed air from the braking chamber is not irrevocably transformed into thermal energy like in the actuators with throttle braking. It returns through the open return valve at $p_2\geq p_m$ into the feed line, i. e. the recuperation mode is realized (dotted line in Fig. 3);

– the compressed air consumption for fixing the piston in the final position at a minimum pressure differential p_k-p_a, is significantly reduced and pressure p_k is considerably smaller than the main pressure p_m;

– owing to the minimum pressure p_k in the exhaust chamber at the initial moment of the piston motion, the nonproductive work of pushing the compressed air from the exhaust chamber is substantially reduced.

Such a complex character of reducing nonproductive energy inputs creates an energy saving effect that enables a 4- to 10-fold reduction of energy inputs in the scope of rational use of this actuator ($\chi<0.2$ and $\beta<2$).

A distinctive feature of the foregoing in comparison with similar publications on this subject is a much higher level of generalization of the results obtained. Due to the use of dynamic similarity criteria instead of physical parameters, it was possible to extend the results obtained practically to the entire region of existence of such actuators (the graphs in Fig. 5, 6) that has made it possible to effectively and clearly identify the scope in which the use of a discrete actuator is rational.

The concrete numerical example has demonstrated the engineering procedure of using these graphs in solving the main problem of functional and cost analysis which consists in comparison of lump sum and operating costs when deciding whether to use the new solution in practice. This approach assumes the use of the concept of economic expediency as the main criterion of the solution efficiency.

Further development of the proposed solution is transition to the so-called compression actuation mode (transition to resonant pneumatic actuators). Such units, when operating under conditions of large inertial loads, use the compressed air accumulated in braking directly for the return stroke. Such a transition enables an even more efficient utilization of working capacity of the compressed air in the actuator.

7. Conclusions

1. A rational structure of commutation links in a pneumatic actuator was defined, where each motion phase corresponds to commutation situations the most rational from the point of view of energy saving and maximum speed.

2. Based on the most rational structure of commutation links, a pneumatic actuator scheme and an algorithm for its control were constructed. Energy saving is ensured due to the following:

– in the phase of braking (Table 2, Fig. 2), not only transit working capacity is used but also potential energy of expansion of the compressed air in the working chamber;

– compressed air from the brake chamber is not converted irreversibly into thermal energy but returns to the feed line;

– the compressed air consumption for fixing the piston in the final position is reduced;

– due to the minimum pressure p_k in the exhaust chamber at the initial moment of the piston motion, the nonproductive work of pushing the compressed air from the exhaust chamber is substantially reduced.

3. In functioning of the pneumatic actuator (5), all components of loss of working capacity of the compressed air

leading to nonproductive energy inputs were determined. A comparative quantitative analysis of energy losses was made using the basic and proposed PA schemes.

4. A procedure for determining the scope of rational use of an energy-saving actuator was developed based on defining the criteria of dynamic and energy similarity. The graphs of dependence of the operational characteristics of the PA on the criteria of dynamic similarity (Fig. 5, 6) were plotted, which make it possible to extend the results obtained to the entire field of the PA use. The proposed procedure makes it possible to calculate quantity of compressed air consumed by the PA with a base and an optimal structure of commutation links and compare the results obtained in percentage and in a monetary equivalent.

Refrences

1. SMC Pneumatics. Pnevmaticheskie sredstva avtomatizacii. Katalog firmy SMC. 2011. 867 p.
2. Saidur R., Rahim N. A., Hasanuzzaman M. A review on compressed-air energy use and energy savings // Renewable and Sustainable Energy Reviews. 2010. Vol. 14, Issue 4. P. 1135–1153. doi: 10.1016/j.rser.2009.11.013
3. Parkkinen J., Zenger K. A New Efficiency Index for Analysing and Minimizing Energy Consumption in Pneumatic Systems // International Journal of Fluid Power. 2008. Vol. 9, Issue 1. P. 45–52. doi: 10.1080/14399776.2008.10781296
4. Mousavi S., Kara S., Kornfeld B. Energy Efficiency of Compressed Air Systems // Procedia CIRP. 2014. Vol. 15. P. 313–318. doi: 10.1016/j.procir.2014.06.026
5. Al-Dakkan K. A., Barth E. J., Goldfarb M. Dynamic Constraint-Based Energy-Saving Control of Pneumatic Servo Systems // Journal of Dynamic Systems, Measurement, and Control. 2006. Vol. 128, Issue 3. P. 655. doi: 10.1115/1.2232688
6. Transmission and Consumption of Air Power in Pneumatic System / Chen S., Youn C., Kagawa T., Cai M. // Energy and Power Engineering. 2014. Vol. 06, Issue 13. P. 487–495. doi: 10.4236/epe.2014.613042
7. Analysis of the Energy Efficiency of a Pneumatic Booster Regulator with Energy Recovery / Yang F., Tadano K., Li G., Kagawa T. // Applied Sciences. 2017. Vol. 7, Issue 8. P. 816. doi: 10.3390/app7080816
8. Kudryavcev A. I., Kudryavcev A. A. Pnevmaticheskie sistemy i ustroystva v promyshlennosti. Kharkiv: Format, 2011. 479 p.
9. Krutikov G. A., Strizhak M. G. Svobodnoprogrammiruemyy diskretnyy pnevmoprivod s energosberegayushchim rezhimom raboty // Intehrovani tekhnolohiyi ta enerhozberezhennia. 2012. Issue 1. P. 11–16.
10. Krytikov G., Strizhak M., Strizhak V. The synthesis of structure and parameters of energy efficient pneumatic actuator // Eastern-European Journal of Enterprise Technologies. 2017. Vol. 1, Issue 7 (85). P. 38–44. doi: 10.15587/1729-4061.2017.92833