The use of pneumatic cylinders with return springs when creating mechanical drives with recuperative energy

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Abstract. The use of pneumatic cylinders with return springs in mechanical drives with reusable energy for reusable and continuum movements, fixing the output link in the extreme positions are considered. A defined range of work operations can be used by these cylinders. The use of pneumatic actuators with return springs in mechanical drives based on linear spring batteries with two springs is proposed. Algorithms for compensating dissipative losses in mechanical spring drives are considered. The maximum mass values that can be moved by such mechanical drives for each size of the pneumatic cylinder are determined. It is proposed to use pneumatic cylinders with return springs in mechanical drives with energy recovery based on nonlinear spring batteries. A mechanical spring drive for unwinding rolls of packaging materials with discrete modes. In tasks of reciprocating movement of objects with a controlled stand in extreme positions, the reduction of energy costs is achieved when using mechanical spring drives with energy recovery. Energy costs can be reduced several times. Traditionally, a spring-loaded drive contains a linear or non-linear spring-loaded battery, a control system, controlled clips and a motor to compensate for dissipative losses. The use of electric motors to compensate for dissipative losses is also limited by low speed, as compensation for dissipative losses occurs throughout the displacement and with high speed increases engine power and requires transmission with a large gear ratio.

1 Introduction

In tasks of reciprocating movement of objects with a controlled stand in extreme positions, the reduction of energy costs is achieved using mechanic spring drives with energy recovery [1-3]. Energy costs can be reduced several times [4,5]. Traditionally, a spring drive contains linear or non-linear spring battery, a control system, controlled clamps and a motor to compensate for dissipation losses [6,7]. The use of electric motors to compensate for dissipation losses is also limited by low speed, since compensation for dissipation losses occurs throughout the displacement and with high speed. increases engine power and requires transmission with a large gear ratio [1]. At high speed, it is preferable to use pneumatic cylinders to compensate for dissipation losses, fix the output link in extreme positions and work in the presence of technological loads [4]. With small masses, preferring to use drives with return springs, with which the execution springs are fixed.

This paper discusses mechanic drives with energy recovery, built on the basis of pneumatic cylinders with return springs [8-10]. Usually such cylinders are used in drives with one-sided load.

2 Object of study

In regenerative drives, both linear and non-linear spring batteries are used. In linear spring batteries, the force or moment in the working range of movements or turns have a linear relationship. The main feature of the considered drives is the need to provide a controlled dwell, which is determined by the technological process. Usually, managed ticks are used for this purpose. In addition, an additional motor is needed to compensate for dissipation losses. In this work, the following scientific position is protected: The use of a pneumatic motor allows us to simultaneously solve the following problems: 1. Compensation of dissipation losses, 2. Fixing the drives in extreme positions 3. Maximum speed compared to hydraulic and electric drives. The power element of the considered drives is a spring battery. Figure 1 shows two spring batteries used in spring drives for reciprocating movements. Figure 2 shows a non-linear spring battery with a transnational pair based on a compression spring, this drive has variable loads with energy recovery. Characteristics of spring batteries determine the dynamic properties of spring drives.

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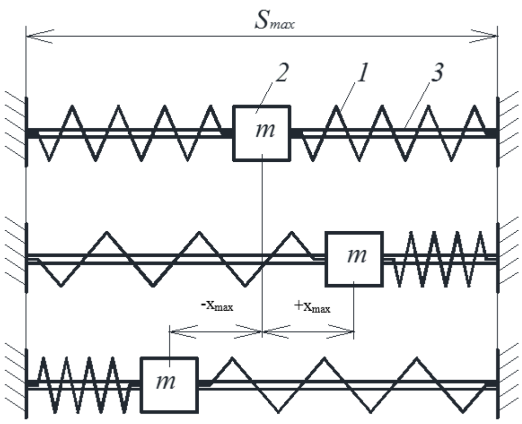

Fig. 1. Spring battery with two compression springs.

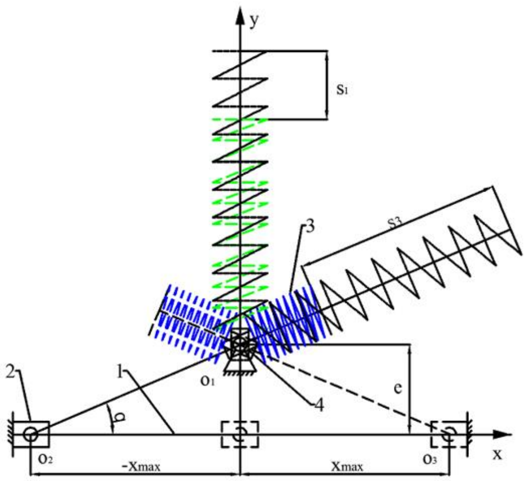

Fig. 2. Non-linear spring battery with a translational pair based on a compression spring (variable loads with energy recovery): 1 - guide; 2 - carriage (slider); 3 – spring.

Table 1 presents the characteristics of pneumatic cylinders with return springs of the Italian company Pnevmax.

Where F_H - initial force, with extended stem; F_K - maximum force with spring retracted; c - spring stiffness. Define the scope of the pneumatic cylinders with recoil springs, presented in Table 1, in mechanic recuperative drives constructed using a linear spring accumulator circuit with two springs. The time of movement of the working body from one extreme position to another, for a linear spring battery with two springs, without taking into account dissipation losses, is determined from the expression

$$t=\pi\cdot\sqrt{m/2\,c}, \qquad (1)$$

For a spring battery, when using pneumatic cylinders with return springs, the amount of decompression is determined by the following expression.

$$L_{min} = F_H/c=h, \qquad (2)$$

Then the minimum potential energy of the spring is

$$P_{min}=0.5F_H^2/c, \qquad (3)$$

The maximum potential energy of the spring is equal to

$$P_{ef}=P_{max}-P_{min}=0.5c(S^2+S\cdot L_{min}), \qquad (4)$$

We introduce the concept of the effective potential energy P_{max}, equal to the difference between the maximum and minimum potential energies

$$P_{max}=0.5c(S+L_{min})^2, \qquad (5)$$

Which allows you to determine the maximum carriage speed in the middle position.

$$\dot{x}_{max}=\sqrt{2P_{ef}/m}, \qquad (6)$$

Equation (1) allows to determine the maximum mass, depending on the specified travel time *t*.

Table 1. Characteristics of pneumatic cylinders with return springs of the Italian company Pnevmax.

Series	Microcylinders series ISO 6431-1260 (travel 0-40mm)			Microcylinders series ISO 6431-1280 «MIR» (travel 0-50mm)		
piston diameter (mm)	F_H (N)	F_K (N)	c (N/m)	F_H (N)	F_K (N)	c (N/m)
ø 8				2	4	40
ø 10				2	4	40
ø 12	10	27	415	4	9	94
ø 16	11	23	295	8	21	270
ø 20	11	23	295	11	22	220
ø 25	8	49	1030	17	31	284
ø 32	20	53	833	23	53	590
ø 40	39	106	1668			
ø 50	39	106	1668			
Series	Cylinders Series ISO 15552-1319-20-21 (travel 0-50mm)			Compact cylinders « Europe » (travel 0-25mm)		
Piston diameter (mm)	F_H (N)	F_K (N)	c (N/m)	F_H (N)	F_K (N)	c (N/m)
ø 12				4	9	216
ø 16				4	18	532
ø 20				5	18	528
ø 25				10	26	628
ø 32	17	42	490	12	34	880
ø 40	25	83	1176	17	44	1096
ø 50	51	115	1276	28	51	940
ø 63	51	115	1276	37	64	1060
ø 80	98	194	1922	59	99	1600
ø 100	98	194	1922	101	142	1624

Fig. 3 shows the graphs of the change in the maximum mass of the load m from the given time of movement *t*.

In work [5] it is shown that the spring battery turns into a linear harmonic oscillator and the time of movement from one extreme position to another is determined by equation (1), if the following relationship between the characteristics of a spring battery.

$$h=m\cdot g/c=F_H/c, \qquad (7)$$

where h - a constructive size equal to the minimum distance between the axes of the hinged connections of the spring accumulator.

The mass of the slider in this case should be equal to

$$m=FH/g, \qquad (8)$$

Travel time is [5]

$$t=\sqrt{h}, \qquad (9)$$

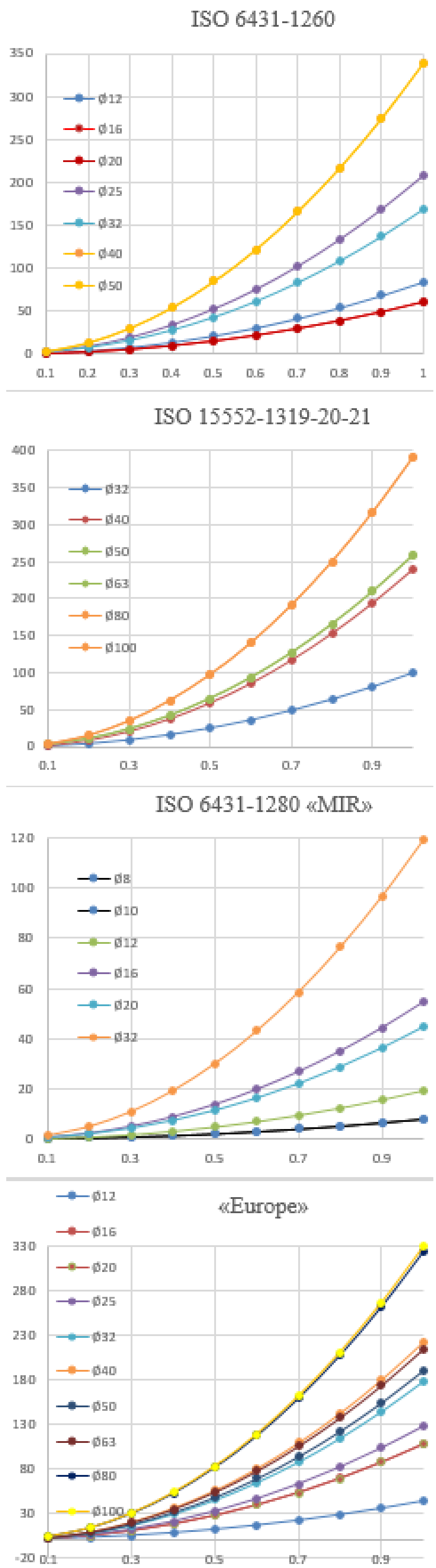

Fig. 3. Graphs of changes in the mass of the load m from a given time of movement t.

The maximum deviation of the carriage from the average position x_{max} is determined by the following expression

$$x_{max}=\sqrt{s^2+2\cdot e\cdot s}, \tag{10}$$

where s - cylinder stroke.

With such characteristics of the drive, throughout the movement of the carriage, the total load on the guideline is zero.

Energy costs are determined only by the losses due to internal friction in the spring, friction in the hooks and guides, and friction forces in the pneumatic cylinder.

Knowledge of the total dissipation losses is necessary to determine the compensation of power pulses, depending on the adopted algorithm for applying pressure in the cavity of the pneumatic cylinder.

In this case, the time t can be obtained in accordance with the work [9] from the solution of the following equation

$$t=\sqrt{m/2}\cdot\int_{-x_{min}}^{x_{max}}\frac{dx}{\sqrt{P_{max}-P_{min}}}, \tag{11}$$

where Π_{max} is the current value of the potential energy defined by the x coordinate.

After the transformation of equation (11) takes the following form

$$t=\sqrt{m/c}\cdot K_{te}, \tag{12}$$

where K_{te} is the dimensionless coefficient obtained when solving an integral equation.

$$K_{te}=\sqrt{m/c}\int_{-1}^{1}\frac{d\bar{x}}{\sqrt{\left(\sqrt{1+\bar{e}^2}-\bar{e}\right)^2-\left(\sqrt{\bar{x}^2+\bar{e}^2}-\bar{e}\right)^2}}, \tag{13}$$

where $\bar{e}=e/x_{max}$, $\bar{x}=x/x_{max}$

The energy loss to overcome the resistance force determined by the internal friction in the spring is equal to [4]

$$A_1=\psi\cdot P_{ef}, \tag{14}$$

The dispersion coefficient ψ can be taken equal to 0.1.

Regardless of the design of the drive

$$A_2=\int_{-x_{min}}^{x_{max}}F_{mp}^{np}(x)dx, \tag{15}$$

where F_{mp}^{np} is the friction force in the articulated joints, reduced to the slider.

We define F_{mp}^{np}, for this we write the equation of instantaneous power system

$$N_1=N_2, \tag{16}$$

where N_1 is the instantaneous power of the reduced friction forces in the articulated joints of the spring; N_2 - instantaneous power of friction forces in the articulated joints of the spring.

The instantaneous power of the reduced friction force of the spring, F_{mp}^{np} reduced to a slider, is determined from the expression

$$N_1=F_{mp}^{np}\cdot\dot{x}, \tag{17}$$

The instantaneous power of the friction forces in the articulated joints of the spring in accordance with [4] is determined from the expression

$$N_2=2F_{np}\cdot f\cdot d/2\cdot\dot{q}, \qquad (18)$$

where F_{np} is the current spring force; f is the coefficient of friction in the articulated joints of the spring; d is the diameter of the swivel axis; $\dot{q}$ is the angular speed of rotation of the spring relative to the slide.

The friction force brought to a slider in hinged joints is obtained from the joint solution of the system of equations (17) and (18)

$$F_{mp}^{np}=F_{np}\cdot f\cdot d\cdot\dot{q}/\dot{x}, \qquad (19)$$

Find the relation $\dot{q}/\dot{x}$. In accordance with Figure 3, we have

$$tg\,q=h/x, \qquad (20)$$

Taking the derivatives of the left and right sides of equation (20) we get

$$1/cos^2q\cdot\dot{q}=h/x^2\cdot x, \qquad (21)$$

or

$$\dot{q}/\dot{x}=h/x^2\cdot cos^2q=h/x^2\cdot x^2/(x^2+h^2)=h/(x^2+h^2), \qquad (22)$$

The current spring force F_{np} is determined from the equation

$$F_{np}=c\cdot(\sqrt{x^2+h^2}-h), \qquad (23)$$

The friction force applied to the carriage at the articulated joints of the spring is determined by the formula [4]

$$F_{mp}^{np}=c\cdot f\cdot d\cdot\frac{h}{x^2+h^2}\cdot\left(\sqrt{x^2+h^2}-h\right)=c\cdot f\cdot d\cdot h\cdot\left(\frac{1}{\sqrt{x^2+h^2}}-\frac{h}{x^2+h^2}\right), \qquad (24)$$

We introduce the dimensionless parameters $\bar{h}=h/x_{max}$ and $\bar{x}=x/x_{max}$, then equation (24) takes the form

$$F_{mp}^{np}=c\cdot f\cdot d\cdot\bar{h}\cdot\left(\frac{1}{\sqrt{\bar{x}^2+\bar{h}^2}}-\frac{\bar{h}}{\bar{x}^2+\bar{h}^2}\right), \qquad (25)$$

The work required to overcome the friction forces in the hinge joints is determined by the following expression

$$A_2=c\cdot f\cdot d\cdot\bar{h}\cdot\left(\int_{-1}^{1}\frac{1}{\sqrt{\bar{x}^2+\bar{h}^2}}dx-\int_{-1}^{1}\frac{\bar{h}}{\bar{x}^2+\bar{h}^2}dx\right), \qquad (26)$$

and after the transformations we get

$$A_2=2c\cdot f\cdot d\cdot\bar{h}\left[ln\left(1+\sqrt{1+\bar{h}^2}\right)-\bar{h}\cdot arctg\frac{1}{\bar{h}}-ln\,\bar{h}\right], \qquad (27)$$

The friction force in the guides is determined by the total mass of the slide and the working body and the load on the guides from the vertical component of the spring force in general form is equal to

$$F_{mp.}=m\cdot g\cdot f-F_{np}\cdot sin\,q, \qquad (28)$$

If the drive parameters ensure that the ratio is met, then the friction loss in the guides is zero.

This paper discusses spring drives with energy recovery based on linear spring batteries, shown in Figure 4. The considered spring drive consists of four compression springs 1, installed in pairs on both sides of the slide 2 moving along the guides 3. To compensate for dissipation losses, pneumatic is used cylinder 4, the working cavities of which are connected to the distributor 5, the electromagnetic actuators *Y1* and *Y2* of which are connected to the control system 6. In addition, on the pneumatic body cylinder 4, position sensors *D1*, *D2*, *D3* and *D4* are installed, the outputs of which are connected to the control system 6. The drive is also equipped with an additional distributor 7, the electromagnetic actuator *Y3* of which is also connected to the control system 6. The input of the pneumatic distributor 7 is connected to one of the pneumatic output distributor 5, and the second inlet of the distributor 7 through an adjustable choke 8 is connected to the atmosphere.

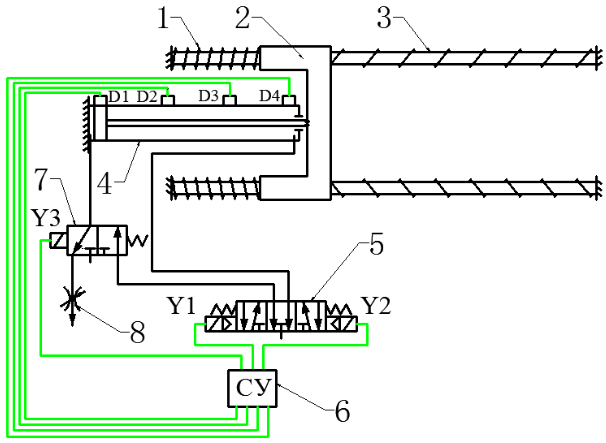

Fig. 4. Scheme spring drive with energy recovery
1 - springs; 2 - slider; 3 - guide; 4 - cylinder; 5 - pneumatic distributor 5/3; 6 - exercise system; 7 - pneumatic distributor 3/2; 8 - adjustable damsel.

The second example of the use of the considered pneumatic cylinders with return springs is the drive for pulling packaging material in automatic packaging machines, shown in fig. 5.

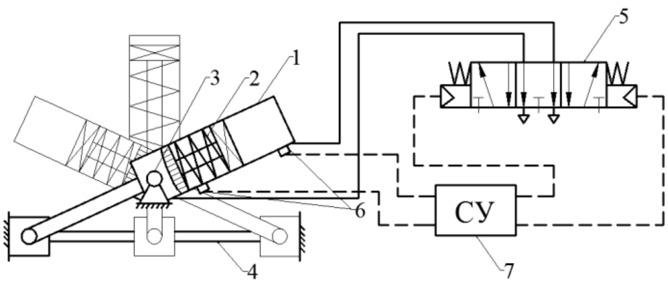

Fig. 5. Spring drive mechanism for pulling packaging material
1 - pneumatic cylinder with compression spring; 2 - compression spring; 3-side hinges; 4 –directing; 5 –pneumatic distributor 5/3; 6 - reed switches; 7 - exercise system.

The spring actuator consists of a pneumatic cylinder 1, equipped with a compression spring 2 installed in the rod end of the cylinder 1. With the help of side hinges 3 the cylinder is fixed to the drive housing. The cylinder rod 1 is pivot ally connected to the guide 4. The drive is equipped with a pneumatic distributor 5/3 (position 5). On the pneumatic cylinder 1 installed position sensors -

reed switches 6. When moving the slider to any of the extreme positions of the spring 2 is compressed. The work is as follows. The signal from the controller enters the distributor 5, which moves to the leftmost position. Air from the distributor is supplied to the pneumatic cylinder 1, and the movement begins under the action of the force of the spring and the force acting on the piston of the cylinder. With the passage of the piston of the cylinder past the sensor 6, the controller 7 receives a signal, which in turn applies voltage to the electromagnetic input of the distributor 5, which goes into the neutral position and the cavities of the pneumatic cylinder are connected to the atmosphere. Further movement of the carriage occurs only under the action of the force of the spring 2. This continues until a configuration occurs, when the pneumatic cylinder takes a vertical position and further movement occurs under the action of inertial forces and the potential energy begins to accumulate in the spring 2. When the carriage approaches the extreme to the right position, position sensor 6 is triggered and the distributor switches to the extreme right position, air is supplied to the roadless cavity of the cylinder and the carriages are fixed and at the extreme right. Then begins the movement in the opposite direction. The presence of a pneumatic distributor 5 allows you to turn on and off the supply of compressed air in the cavity of the pneumatic cylinder 1 with different configurations of the drive and to ensure the required laws of motion of the output link. Fixing the carriage in the extreme positions is ensured by the fact that the spring force in the extreme positions of the output link is less than the force of the pneumatic cylinder. Spring battery 2 is a non-linear oscillatory system with a moving mass and with a given amplitude of oscillation, in which the time of movement from one extreme position to another depends on the spring stiffness and design parameters.

5 The results of the research

The characteristics of the working operations that can be obtained when creating mechanic spring drives with energy recovery are defined, in which favorable dynamic conditions are provided and the energy costs are reduced several times. Recommendations on the use of pneumatic cylinders with return springs in mechanic spring drives using nonlinear spring accumulators have been developed, which allows reducing the size of the drive in the direction of movement of the working body and reducing the load on the guides. Calculation formulas are given for determining dissipation losses. It is shown that when the stroke of the rod is 40-50 mm, the output link can move more than 2 times or more than the stroke.

The maximum values of the masses of products that can move only due to the springs in the considered pneumatic cylinders at the time of displacement of 1s are shown For cylinders according to «ISO 6431-1260» and «ISO 15552-1319-20-21», the maximum weight can reach 400 kg, and for cylinders «ISO 6431-1280» and «Europe» the maximum weight can reach values up to 200 kg.

The developed drive for pulling the packing tape using a spring-loaded pneumatic cylinder reduces energy costs and provides favorable dynamic operating conditions.

The results of the work were used in the development of the mechanic pneumatic system of the drives of the filling and packaging machine for bulk products in the range from 0.5 g to 5 g. The results of the work can take advantage of developers of energy-saving technological equipment.

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