International Journal of Applied Engineering Research ISSN 0973-4562 Volume 12, Number 14 (2017) pp. 4599-4603


# Decrease in Power Inputs in Pneumodrive Weighing-and-Packing Machine

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**Abstract**

A mathematical model of gas-dynamic processes of improved pneumatic actuator of table turn of packaging and filling machine "Alur-1500" is considered. To reduce the energy consumption the method of backpressure deceleration with energy recovery in an additional volume was used.

**Keywords:** weighing-and-packing machine, pneumodrive, mathematical model, method of backpressure deceleration.

**Abstract**

Modern production is impossible without mechanization, automation of process equipment, which determines its versatility and performance. If the time of the main and supporting operations is comparable, it is advisable to mechanize or automate not only main, but also supporting operation, which is often less demanding on the accuracy of the positioning of the executive bodies and has no speed limitations of the main operation. For this purpose, pneumatic actuators are widely used, but the inertia of the output unit and the compressibility of the working environment make it difficult to decelerate and stop the working body within the specified accuracy. Because of this feature it is necessary to limit the speed of movement of the working body, which adversely affects the performance of the equipment.

In a number of cases [1], the inertia of the output units of pneumatic actuator and compressibility of a working environment, allows to apply deceleration and stopping of working body by backpressure. Either exhaust cavity of pneumatic engine or attached to it, additional volume [2] is used as the brake cavity. The accumulated potential energy of the compressed air is used for subsequent acceleration of the working body or for the reverse of its movement [3].

Comparability of the main and supporting technological time is characteristic for the filling and packaging operations, including food equipment to mechanize operations of which, because of the known benefits of pneumatic actuators are used extensively. Pneumatic actuator of the packaging machine "Alur 1500" carries out the following operations: the supply of empty containers from the loading tray to the position of the table; rotation of the packaging platform; filling containers with milk product; soldering of cover; putting down the date; unloading the packaged product on the conveyor [1]. Ensuring non-percussive shutdown of pneumatic working bodies is carried out by throttling. From ongoing operations, rotation of the packaging platform, requires compressed air consumption of 0,46 $m^3$ / h, which is comparable to main, dosing and filling operation. Rotation does not require high precision of deceleration and has no speed limitations specific to dosing operation. These features allowed modernizing the pneumatic actuator of the turn of the packaging platform, using the method of backpressure deceleration. To ensure flexibility of the process, to the exhaust cavity at the time of deceleration, was connected additional volume, the exhaust from the cavity was overlapped. Then the deceleration of the pneumatic engine proceeds with the air compression in its exhaust cavity and the additional volume, the initial parameters of the latter are set before deceleration, depending on the given coordinates of the shutdown and the payload. Accumulated during deceleration energy is used for the subsequent reversal of the working body.

The design scheme of the suggested pneumatic actuator [3], is represented in the Figure 1. Quasi-stationary mode at a constant pressure of $p_m$ power without taking into account the heat exchange with the environment is considered. The processes of filling and expiration of air in the cavities of the pneumatic cylinder are taken as adiabatic with supercritical and subcritical modes.

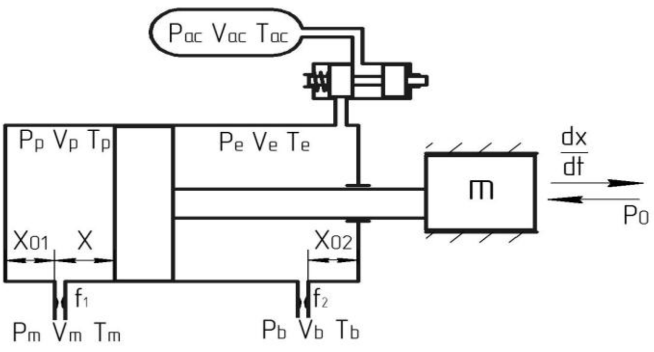


**Figure 1:** Design scheme of pneumatic actuator with compressed air energy recovery into an additional volume.

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For the taken conditions the mathematical model of pneumatic actuator is described as a system of nonlinear differential equations 1 ... 10. Equations 1 ... 5 describe the process of acceleration of the pneumatic cylinder [4], and the equations 6 ... 10 – are characteristics of pneumatic actuator when decelerating after the connection of additional volume.

The study of systems of equations 1 ... 10 is carried out by the numerical method of Runge-Kutta with automatic step selection. As a result of studies of the suggested device dependencies of the pressure and temperature in the exhaust and working cavity of the pneumatic cylinder, velocity and acceleration of the output unit on the time and coordinates of the piston movement are obtained. The testing was conducted on a specially developed for this test bench that simulates the work of PA installation for filling and packaging dairy products "Alur-1500" [5]. Stop of the table was conducted by overlapping exhaust pipe and subsequently air compression in the additional volume with a predetermined pressure and volume.

$$\begin{cases} m\dfrac{d^2x}{dt} = p_P \cdot F_P - P_0 \cdot \mathrm{sign}\dfrac{dx}{dt} - \kappa_V \cdot \dfrac{dx}{dt} - p_E \cdot F_R & (1) \\ \dfrac{dp_P}{dt} = \dfrac{k \cdot \mu_1 \cdot f_1 \cdot p_M \sqrt{\dfrac{2k}{(k-1)} R \cdot T_M \left[ \left(\dfrac{p_P}{p_M}\right)^{\frac{2}{k}} - \left(\dfrac{p_P}{p_M}\right)^{\frac{k+1}{k}} \right]}}{F_P (x_{01} + x)} - \dfrac{k \cdot p_P}{(x_{01} + x)} \dfrac{dx}{dt} & (2) \\ \dfrac{dp_E}{dt} = -\dfrac{k \cdot \mu_2 \cdot f_2 \cdot p_E^{\frac{3k-1}{2k}} \sqrt{\dfrac{2k}{(k-1)} R \cdot T_E \left[ \left(\dfrac{p_B}{p_E}\right)^{\frac{2}{k}} - \left(\dfrac{p_B}{p_E}\right)^{\frac{k+1}{k}} \right]}}{F_R (S + x_{02} - x) p_B^{\frac{k-1}{2k}}} + \dfrac{k \cdot p_E}{(S + x_{02} - x)} \dfrac{dx}{dt} & (3) \\ T_E = \dfrac{p_E^{\frac{k-1}{k}} \cdot T_{|0E}}{p_{0E}^{\frac{k-1}{k}}} & (4) \\ \dfrac{dT_P}{dt} = (k \cdot T_M - T_P) \dfrac{R \cdot T_P \cdot \mu_1 \cdot f_1 \cdot p_M \sqrt{\dfrac{2k}{(k-1)} \dfrac{1}{R \cdot T_M} \left[ \left(\dfrac{p_P}{p_M}\right)^{\frac{2}{k}} - \left(\dfrac{p_P}{p_M}\right)^{\frac{k+1}{k}} \right]}}{p_P \cdot F_P (x_{01} + x)} - & \\ -(k-1) \dfrac{T_P \cdot dx}{(x_{01} + x) dt} & (5) \end{cases}$$

$$\begin{cases} m_\Pi \dfrac{d^2x}{dt} = p_P \cdot F_P - P_0 \cdot \mathrm{sign}\dfrac{dx}{dt} - \kappa_V \cdot \dfrac{dx}{dt} - p_E \cdot F_R & (6) \\ p_P = \left( \dfrac{x_{01} + x_{SP}}{x_{01} + x} \right)^k p_{PSP} & (7) \\ p_E = \left( \dfrac{(s + x_{02} - x)(p_{P\,SP})^{1/k} + hh\left((p_{PB})^{1/k}\right)}{(s + x_{02} + hh - x)} \right)^k & (8) \\ T_E = \left( (s + x_{02} + hh - x(t)) \cdot \dfrac{p_E \cdot T_{E\,SP} \cdot T_{PB}}{(s + x_{02} - x(t)) T_{PB} + hhT_{0E}} \right) & (9) \\ T_P = \left( \dfrac{p_{P\,SP}}{p_P} \right)^{\frac{k-1}{k}} T_{P\,SP} & (10) \end{cases}$$

where: m - the reduced mass of progressively moving parts; $F_P$, $F_R$ - cross-sectional areas of the piston of exhaust and working cavities, respectively; $p_B$, $p_M$, $p_p$, $p_B$, $p_{PB}$ - pressure: atmospheric, in pipe-line, work and exhaust cavities of PC and

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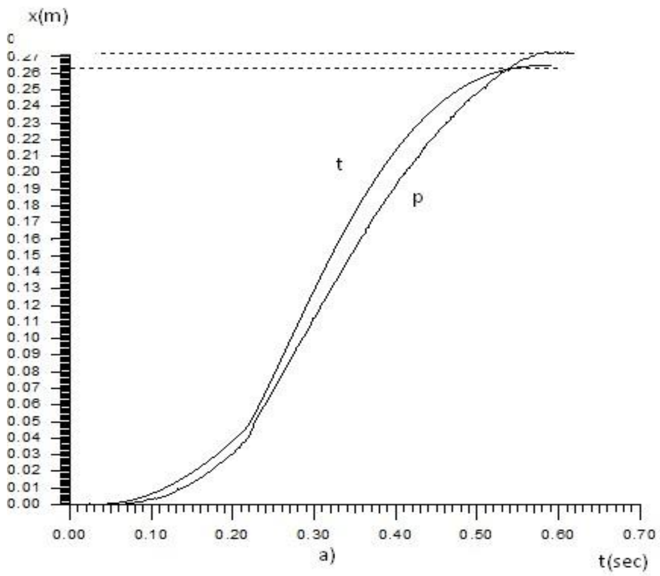
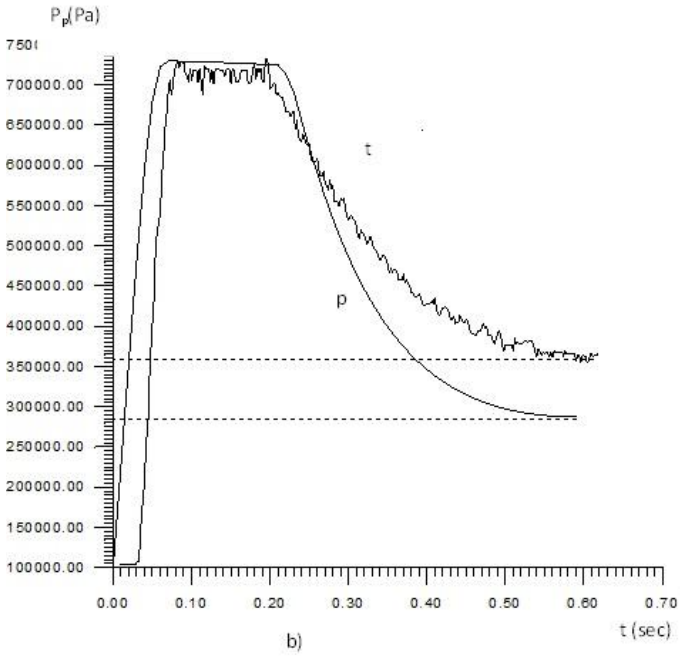
battery, respectively; x, dx / dt, $d^2x / dt^2$ - movement, velocity, acceleration of the piston; s – power stroke of the piston; $X_{01}$, $X_{02}$ - given value of x corresponding to the initial volumes $V_{01}$, $V_{02}$; t - time of movement; $P_0$ - the dry friction force; k - adiabatic index; $T_B$, $T_M$, $T_p$, $T_E$, $T_{PB}$ - the absolute temperature in the atmosphere, pipe-line, work, exhaust cavities of the pneumatic cylinder and battery, respectively; $T_{0E}$ - the initial value of the temperature of the exhaust cavity at the start of movement; $T_{SP}$, $T_{E\ SP}$ - the temperature in the working cavity and its initial value during deceleration, respectively; $p_{0E}$ - the initial value of the pressure of the exhaust cavity of the pneumatic cylinder at the time of the movement; $p_{E\ SP}$, $p_{P\ SP}$ - the initial value of the pressure when decelerating in the exhaust and the working cavities, respectively; $\mu_1$, $\mu_2$ - flow coefficients of pneumatic lines; $k_V$ - viscous friction coefficient; $f_1$, $f_2$ - equivalent areas of pneumatic lines; R - universal gas constant; hh - equivalent additional controlled volume of exhaust cavity; $x_{SP}$ – coordinate of the switching of the control device for deceleration.

Testing of the correctness of the mathematical model consisted of two parts [5]: testing for the characteristic alignment between the theoretical and practical dependencies; testing for the adequacy of the mathematical model. Tests of the pneumatic actuator, piston diameter of pneumatic cylinder of which was 32 mm, power stroke - 265 mm, were carried out for the pressure in the additional volume - 630,000 Pa. The value of the additional volume was 42 $cm^3$.

Graphical layouts of obtained dynamic processes are shown in Figure 2. Lines with index «t» on the experimental graphical layout show the theoretical calculation lines (Figure 2.) obtained by solving the system of equations for calculation of these parameters (Table 1.).



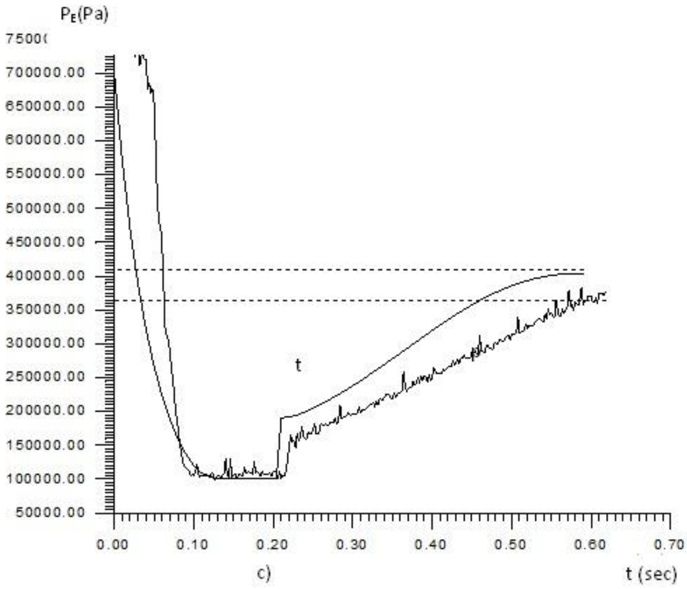




**Figure 2.** Graphical lines of gas dynamic processes of the table rotation pneumatic actuator (t - theoretical dependence, p - practical dependence, a) change in time movement; b) change of pressure in the working cavity of the pneumatic cylinder; c) change of pressure in the exhaust cavity of the pneumatic cylinder).

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**Table 1:** Parameters of pneumatic actuator

| PC cavities | Area of passes, $m^2$ | Flow coefficient, $\mu$ | Detrimental volumes, $m^2$ | Additional volume, $m^2$ |
|---|---|---|---|---|
| Piston | $7,85 \cdot 10^{-5}$ | 0,24 | $1,71 \cdot 10^{-4}$ | ——— |
| Rod | $7.85 \cdot 10^{-5}$ | 0,35 | $7,56 \cdot 10^{-5}$ | $4.2 \cdot 10^{-5}$ |

The time difference between the experimental index "n", and the theoretical index «t», graphs is - 5%, the final break point - 5%, the pressure in the pressure cavities - 20% and in the exhaust cavity - 23% under the condition of experienced preliminary determination of flow coefficients. During the testing of the adequacy of the mathematical model controlled and independent from each other parameters varied: the pressure in the additional volume ($p_{PB}$ (Pa)), the size of the additional volume (V (cm3)), resulting in a changed value of movement, velocity and energy consumption. Final point of table shutdown (x (m)) (table 2) was adopted as parameter of evaluation of the tests quality. Pressure of compressed air in pipe-lines was $p_m$ = 630000 Pa. In both tests, the total weight of the table has not changed and amounted to m = 34 kg.

Planning for testing the adequacy of the mathematical model was carried out by the method of "Latin square", the intervals between the levels of variation were chosen equal to facilitate the computational work.

**Table 2.** Data verification of the adequacy of the mathematical model

| Additional volume, ($cm^3$) | Primary pressure in additional volume, (Pa) | Shutdown coordinate (м) | | | | | $\sum_{i=1}^{3}\left(x_i - x_{cp}\right)^2$ |
|---|---|---|---|---|---|---|---|
| | | Experimental | | | Average | Theoretical | |
| V($cm^3$) | $p_{PB}$ | $x_1$ | $x_2$ | $x_3$ | $x_{med}$ | $x_t$ | |
| 42 | 700000 | 0,4036 | 0,3985 | 0,4012 | 0,4011 | 0,3974 | $1,302 \cdot 10^{-5}$ |
| 42 | 400000 | 0,4308 | 0,4213 | 0,4265 | 0,4262 | 0,4225 | $5,268 \cdot 10^{-5}$ |
| 42 | 100000 | 0,4485 | 0,4529 | 0,4495 | 0,4503 | 0,4468 | $1,063 \cdot 10^{-5}$ |
| 63 | 700000 | 0,3768 | 0,3892 | 0,3816 | 0,3825 | 0,3791 | $7,818 \cdot 10^{-5}$ |
| 63 | 400000 | 0,4232 | 0,4173 | 0,4265 | 0,4223 | 0,4176 | $4,344 \cdot 10^{-5}$ |
| 63 | 100000 | 0,4439 | 0,4581 | 0,4523 | 0,4514 | 0,4491 | $1,019 \cdot 10^{-4}$ |
| 84 | 700000 | 0,3721 | 0,3785 | 0,3754 | 0,3753 | 0,3715 | $2,048 \cdot 10^{-5}$ |
| 84 | 400000 | 0,4245 | 0,4112 | 0,4192 | 0,4183 | 0,4147 | $8,966 \cdot 10^{-5}$ |
| 84 | 100000 | 0,4626 | 0,4587 | 0,4617 | 0,461 | 0,4567 | $8,339 \cdot 10^{-6}$ |

Adequacy testing was carried out by the methods of dispersive analysis. Since the number of replicates of tests is the same in all the rows of the matrix (table 2), to assess the homogeneity of dispersions was used Cochran's criterion. After this the assessment of equivalence of dispersions was performed by Fisher's criterion for the confidence level of $\alpha$ = 0,95 and the maximum permissible error $\varepsilon$ in fractions of standard deviation $\pm 3\sigma$. The total number of tests was determined based on the levels of variation of factors - three of them (Table 2.) and a number of factors - two of them (the initial pressure in the additional volume and its size).

The value of calculated Fisher's criterion was 2.67, and the table - 2.7 at a significance level $\alpha$ = 0,05, which confirmed the adequacy of the mathematical model to practical results.

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