Exergy-Based Energy Efficiency Analysis of the Pneumatic Drive with Controlled Braking and Compressed Air Recuperation

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Abstract. The main objectives of the study are to develop a methodology for calculating the energy efficiency of a pneumatic drive with controlled braking and compressed air recuperation based on a three-dimensional CFD model, as well as to optimize the control algorithm of the pneumatic system operating mode in order to maximize the utilization of the internal energy of the working medium. To achieve these objectives, the following tasks were accomplished: mathematical modeling of pneumatic drive dynamics was performed to determine the energetically justified switching point to the braking mode; the rational structure of switching connections between cylinder chambers was optimized to ensure implementation of compressed air recuperation into the supply line; three-dimensional modeling of unsteady gas-dynamic processes was carried out in ANSYS Fluent using a dynamic mesh; an exergy-based approach to assessing the system's energy balance was implemented. The most significant results include the development of a three-dimensional model that enables accurate determination of the pressure in the exhaust chamber; quantitative confirmation of the substantial influence of temperature fluctuations and internal energy redistribution on the efficiency of the pneumatic drive; refinement of the actual pressure level in the exhaust chamber and the mass flow rate of the recuperated air; and obtaining an exergy efficiency value of 48.37%, which is 1.5 times higher than the result calculated using a simplified isothermal model. The significance of the obtained results lies in the development of a scientifically substantiated methodology for calculating the energy efficiency of pneumatic drives based on three-dimensional modeling of transient processes.
Keywords: pneumatic drive, braking, commutation, energy efficiency, exergy efficiency.

DOI: https://doi.org/10.52254/1857-0070.2026.2-70.06
UDC: 621.05

Analiza exergetică a eficienței energetice a unui sistem de acționare pneumatică cu frânare controlată și recuperarea aerului comprimat

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Rezumat. Scopul acestui studiu este fundamentarea parametrilor de proiectare ai echipamentelor spiral-vortex care asigură desulfurarea eficientă din punct de vedere energetic a gazelor de eșapament ale centralelor electrice autonome cu piston pe gaz (GPPP) de capacitate medie. Actualitatea studiului este determinată de necesitatea reducerii rezistenței aerodinamice a sistemelor de epurare a gazelor pentru a minimiza sarcina parazitară asupra unităților de putere. Pentru a realiza acest lucru, au fost abordate următoarele sarcini: elaborarea unui model matematic al mişcării fazei gazoase într-un canal curbiliniu la scară macro; efectuarea modelării numerice a evoluției structurilor secundare macro-vortex; și determinarea influenței unghiului de înclinare spirală și a regimurilor de viteză asupra intensității transferului de masă. Metodologia de cercetare se bazează pe abordări de dinamică computațională a fluidelor (CFD) pentru analiza fluxurilor turbulente sub forțe centrifuge care acționează asupra gazelor de eșapament. Cel mai semnificativ rezultat este stabilirea posibilității formării unei pelicule lichide stabile şi a macro-vortexurilor intense la rezistență aerodinamică scăzută (până la 400 Pa). Se demonstrează că la o viteză optimă de curgere de 6,0 m/s și o secțiune transversală a scruberului de 0.9 m², se obține o eficiență maximă de absorbție a $SO2$ (peste 96%) fără ventilatoare auxiliare cu tiraj indus. Semnificație. Importanța științifică și practică constă în crearea unei metodologii de proiectare inginerească pentru echipamente compacte de purificare a gazelor pentru generarea distribuită de energie electrică. Soluțiile de proiectare propuse oferă economii anuale de energie electrică de 65-95 mii kWh per unitate de putere de 2.6-3.0 MW, crescând semnificativ profitabilitatea generală și siguranța de mediu a centrelor energetice autonome.
***Cuvinte-cheie*:** acționare pneumatică, frânare, comutație, eficiență energetică, eficiență exergetică.

Эксергетический анализ энергоэффективности пневматического привода с управляемым торможением и рекуперацией сжатого воздуха

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Аннотация. Основные цели исследования заключаются в разработке методики расчёта энергоэффективности пневматического привода с управляемым торможением и рекуперацией сжатого воздуха на основе трёхмерной CFD-модели, а также в оптимизации алгоритма управления режимом работы пневмосистемы с целью максимального использования внутренней энергии рабочего тела. Для достижения поставленных целей были решены следующие задачи: выполнено математическое моделирование динамики пневмопривода для определения энергетически обоснованной точки переключения в режим торможения; оптимизирована рациональная структура коммутационных связей камер цилиндра, обеспечивающая реализацию режима рекуперации сжатого воздуха в магистраль; проведено трёхмерное моделирование нестационарных газодинамических процессов в ANSYS Fluent с использованием динамической сетки, что позволило учитывать пространственно-временное распределение давления, температуры и скорости и точно определить массовый расход рекуперируемого воздуха; реализован эксергетический подход к оценке энергетического баланса системы. Наиболее важными результатами являются разработка трёхмерной модели, которая позволяет корректно определить давление в выпускной камере; количественное подтверждение существенного влияния температурных колебаний и внутреннего перераспределения энергии на величину КПД пневмопривода; уточнение фактического уровня давления в выпускной камере и величины массового расхода рекуперируемого воздуха, а также получение значения эксергетического КПД пневмопривода 48.37%, что в 1.5 раза превышает результат, рассчитанный по упрощённой изотермической модели. Значимость полученных результатов состоит в получении научно обоснованной методики расчета энергоэффективности пневмоприводов на основе применения трёхмерного моделирования переходных процессов. Методика принципиально повышает точность определения КПД пневмосистем с рекуперацией энергии и позволяет обоснованно оптимизировать процессы управления пневмоприводами на этапе проектирования, обеспечивая реальное повышение энергоэффективности промышленных процессов.

Ключевые слова: пневматический привод, торможение, коммутация, энергоэффективность, эксергетический коэффициент полезного действия.

INTRODUCTION

Pneumatic drives are widely used in industrial automation due to their simple design, high reliability, safety in explosive environments, and the possibility of using compressed air as the working medium.

However, despite these advantages, pneumatic systems are characterized by relatively low energy efficiency. This drawback is mainly associated with the high energy consumption required for air compression, significant throttling losses, and the dissipation of compressed-air energy during piston braking at the end of the stroke.

According to industrial energy audits, compressed air systems account for a significant share of electricity consumption at manufacturing enterprises, while their overall efficiency often does not exceed 20–30% [1–3].

At the same time, numerous studies show that a considerable portion of these losses can be avoided through rational system design and the application of advanced control strategies [4–7].

Current trends in improving the energy efficiency of pneumatic drives include the recirculation of exhaust air, limiting the air supply depending on the pressure level, elimination of leaks, and other measures, which reduce compressed air consumption while maintaining dynamic performance.

However, the evaluation of energy-efficient operating modes is often based on simplified one-dimensional models that neglect internal energy redistribution and the actual pressure levels in the cylinder chambers, which becomes particularly significant when air recuperation is present.

The application of exergy analysis [8] in combination with three-dimensional CFD modelling (Computational Fluid Dynamics) of transient processes makes it possible to quantitatively evaluate not only the energy parameters but also their irreversibility, which is particularly important for compressed air systems characterized by significant thermodynamic losses.

RESEARCH OVERVIEW

Classical approaches to modelling the dynamics of pneumatic drives are based on one-dimensional unsteady mathematical models that describe the variation of pressure, flow rate, and piston velocity using simplified thermodynamic assumptions, in particular the isothermal or polytropic nature of the processes [9, 10]. Similar approaches are also widely applied in hydraulic drives, where a significant influence of the geometry of valve elements and manufacturing tolerances on the flow capacity and dynamic characteristics of the system has been demonstrated [11, 12].

Such models are computationally efficient and are widely used at the preliminary design stage; however, they do not allow for a correct evaluation of internal energy redistribution and irreversible losses in the system, nor do they enable accurate determination of the mass flow rate of recuperated air.

A separate research direction is related to improving energy efficiency through the optimization of piston braking modes and the use of pneumatic damping schemes. In studies [13–15], it has been shown that controlled braking makes it possible to reduce impact loads and improve the dynamic characteristics of the drive; however, the energy effect of such solutions is usually evaluated only in terms of compressed air consumption, without considering exergy losses.

Considerable attention in recent research has been devoted to compressed air recuperation schemes, which involve the reuse of the energy of the exhaust working medium by returning it to the supply line or bypassing it between the chambers of the pneumatic actuator [16–18]. Experimental and numerical results indicate that such approaches can significantly reduce air consumption; however, their effectiveness largely depends on the actual pressure and temperature levels in the chambers of the pneumatic cylinder.

Another research direction involves the application of three-dimensional CFD modelling to analyse unsteady gas-dynamic processes in pneumatic systems [19]. Studies [20–22] have shown that the CFD approach makes it possible to account for the spatial distribution of velocity, pressure, and temperature, as well as local losses and gas compressibility, which is critical for the correct determination of the mass flow rate and the energy characteristics of the system. At the same time, as demonstrated in [23], in unsteady flows an additional role is played by temporal effects associated with the formation of shear stresses, which cannot be correctly described within the framework of quasi-steady approaches.

Most CFD studies focus primarily on the analysis of flow dynamics and heat and mass transfer, without integrating the obtained results with a thermodynamically justified evaluation of energy efficiency. To overcome this limitation, the application of exergy analysis appears promising, as it makes it possible to account for energy quality and quantitatively evaluate irreversible losses in compressed air systems [24, 25].

Thus, the analysis of the literature indicates that a comprehensive combination of one-dimensional mathematical modelling, three-dimensional CFD analysis, and the exergy-based approach for pneumatic drives with controlled braking and compressed air recuperation makes it possible to account for energy quality and irreversible losses in compressed air systems.

PROBLEM STATEMENT

The calculation of dynamic characteristics of pneumatic systems with braking achieved by modifying the structure of switching connections and by recuperating part of the compressed air is traditionally performed using simplified energy-based or flow-rate indicators, which do not reflect the actual conversion of the kinetic energy of moving masses into the internal energy of the working medium during braking. As a result, the real contribution of temperature effects, local pressure drops, and internal energy redistribution to the formation of the overall efficiency of the pneumatic drive remains undefined.

An additional difficulty is the absence of a consistent approach to the quantitative evaluation of the recuperated energy of compressed air under conditions of variable pressure and temperature. The use of one-dimensional models alone does not allow the correct determination of the onset of recuperation, the actual pressure level in the exhaust chamber, or the mass flow rate of air returned to the network, which in turn leads to significant errors in the determination of efficiency. Similar limitations of one-dimensional dynamic models for drives with complex control structures and feedback loops have also been noted in studies devoted to electrohydraulic systems, where correct description of transient processes requires consideration of unsteady hydrodynamic effects [26].

Thus, the aim of this study is to develop a reliable methodology for evaluating the energy efficiency of a pneumatic drive with controlled braking and recuperation that accounts for the unsteady nature of gas-dynamic processes, variations in the temperature of the working medium, and the quality of energy. The objectives of the study are to combine one-dimensional calculations of piston motion dynamics with three-dimensional flow modelling, as well as to apply the exergy-based approach as the basis for an objective evaluation of the system's energy efficiency.

RESEARCH METHODS

As the calculation scheme, a double-acting pneumatic drive (Fig. 1) with the possibility of implementing an energy recuperation process is considered.

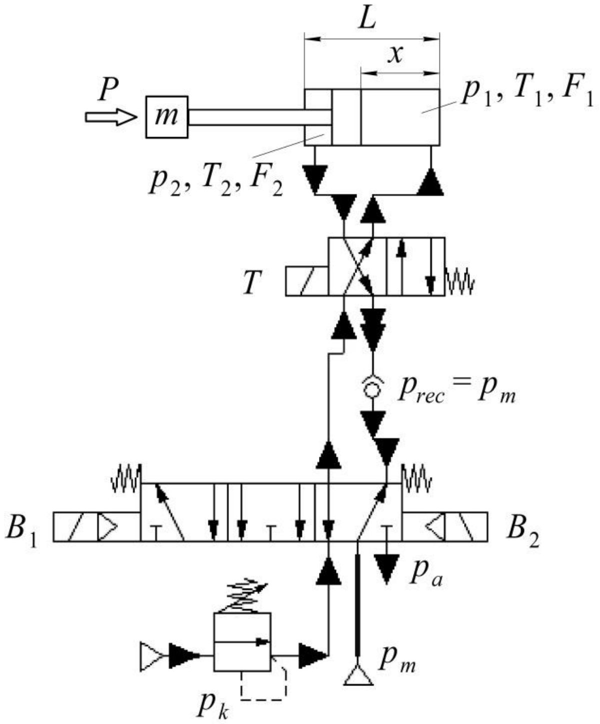

Fig. 1. Calculation diagram of the pneumatic drive.

The working process in the pneumatic drive was investigated in [27], where the influence of the pneumatic system structure on its dynamic characteristics and the possibility of implementing an energy-efficient braking mode with energy recuperation were analysed. As a result of the study, a pneumatic system structure with relay control and compressed air recuperation to the supply network was determined to be optimal from the standpoint of implementing an energy-efficient operating mode (Fig. 1).

In [18], a mathematical model of the pneumatic system (Fig. 1) was developed based on the energy balance equations for a body with variable mass. As an assumption, the processes in the system were considered to be isothermal. In order to unify the calculation results, the model was presented in a dimensionless form.

When modelling the working process, the transition point from the acceleration phase to the braking phase (the moment of switching the directional control valve and connecting the exhaust chamber to the pressure source) was chosen based on the condition of minimizing the final piston velocity and represents the condition of the pneumatic system energy balance (Fig. 1):

$$\Pi_i + T_i = \Pi_{1i} + A_i, \tag{1}$$

where Π_i – the potential energy of expansion of the compressed air in the working chamber of the pneumatic cylinder over the braking distance:

$$\Pi_i = \int_{x_i}^{l_1+x_i} F_1 \cdot p_1 dx + p_k \cdot F_1 \int_{l_1+x_i}^{L} dx = \\ = \frac{F_1 \cdot p_{1i}\left(x_{01}+x\right)}{k-1}\left[1-\left(\frac{x_{01}+x_i}{x_{01}+x\ +l_1}\right)^{k-1}\right] + \\ + p_k F_1\left(L-x-l_1\right), \tag{2}$$

where k – adiabatic index; l_1 – the distance travelled by the piston from the beginning of the braking path to the moment when the pressure-reducing valve is actuated; L – piston stroke; p_1 – pressure in the working (piston) chamber; p_{1i} – pressure in the working chamber at the beginning of braking; p_k – pressure setting of the pressure-reducing valve; F_1 – piston area; x – current piston coordinate; x_{01} – initial piston coordinate (dead volume); x_i – piston coordinate at the beginning of the braking path.

T_i – kinetic energy of the moving parts of the drive at the beginning of braking:

$$T_i = \frac{m \cdot v_i^2}{2}; \tag{3}$$

where m – weight of the moving masses reduced to the piston inertia axis; v_i – piston velocity at the beginning of the braking path.

Π_{1i} – the potential energy of air compression in the exhaust (braking) chamber of the pneumatic cylinder over the braking distance:

$$\Pi_{1i} = \int_{x_i}^{L-l} F_2 p_2 dx + p_m F_2 \int_{L-x}^{L} dx = \frac{F_2 p_{2i} \left(L + x_{02} - x_i \right)}{k-1} \times \left[\left(\frac{L + x_{02} - x_i}{x_{02} + l} \right)^{k-1} - 1 \right] + F_2 p_m l; \tag{4}$$

where p_m – pressure in the supply line; p_2 – pressure in the exhaust (rod-side) chamber; p_{2i} – pressure in the exhaust chamber at the beginning of braking; l – the distance travelled by the piston from the moment the check valve opens to the end of the stroke; F_2 – area of the rod-side chamber; x_{02} – final piston coordinate (dead volume).

A_i – the work required to overcome the static resistance force:

$$A_i = \int_{x_i}^{L} P dx = P\left(L - x_i \right). \tag{5}$$

where P – static resistance force (load acting on the piston).

At the final stage of the study, numerical modelling of the unsteady gas flow in the pneumatic cylinder was performed in a three-dimensional formulation using the ANSYS Fluent software environment (Fig. 2). To ensure consistency between the mechanical and gas-dynamic parts of the problem, the results of the preliminary piston motion calculation were integrated into the CFD model in the form of dynamic meshes corresponding to the piston motion. This made it possible to avoid errors associated with approximating the piston position and improved the reliability of the calculated thermodynamic parameters. The piston displacement law was specified via a profile file (prof-file) used during the simulation.

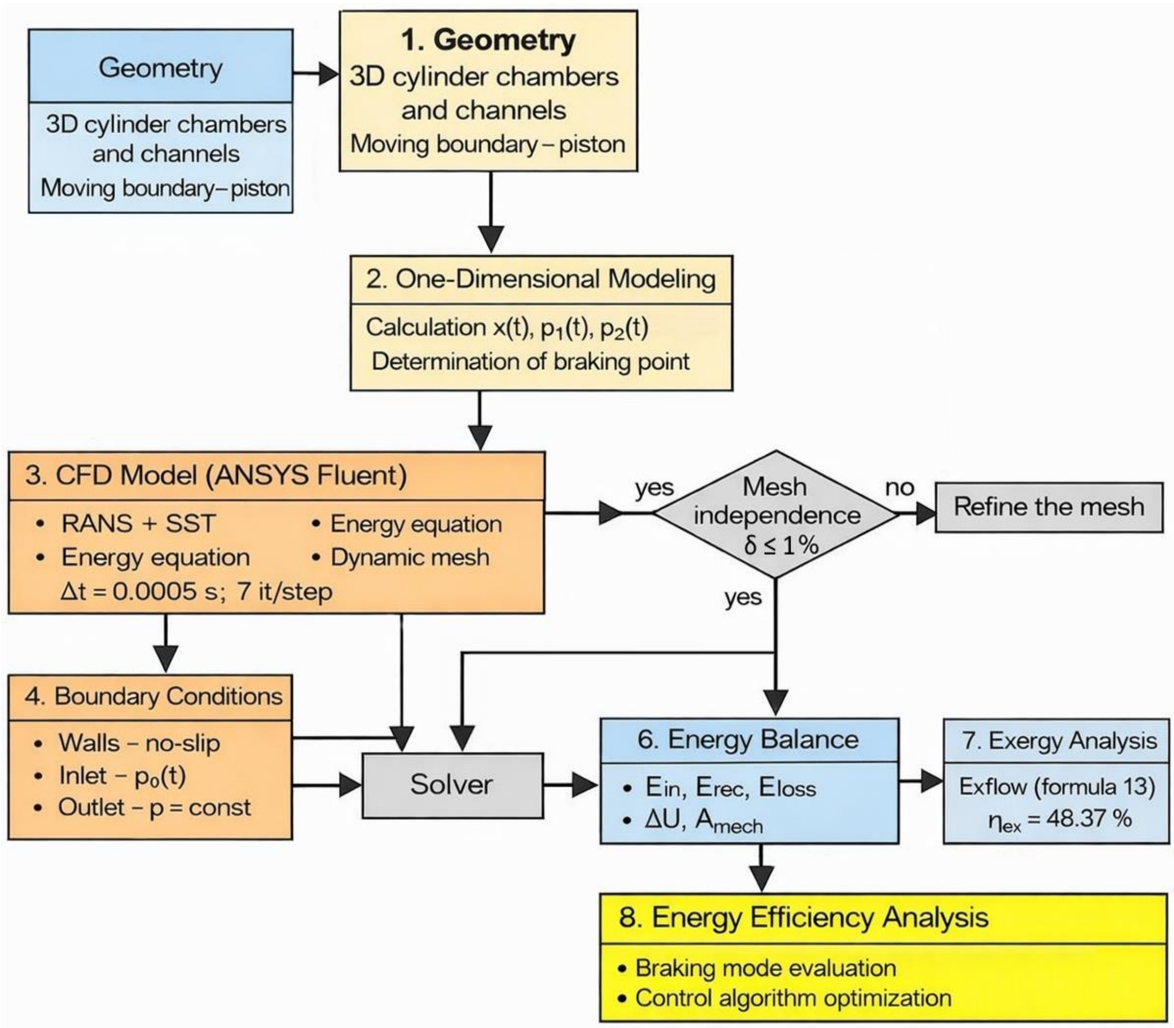

Fig. 2. Research Algorithm for Energy Efficiency Analysis of a Pneumatic Drive.

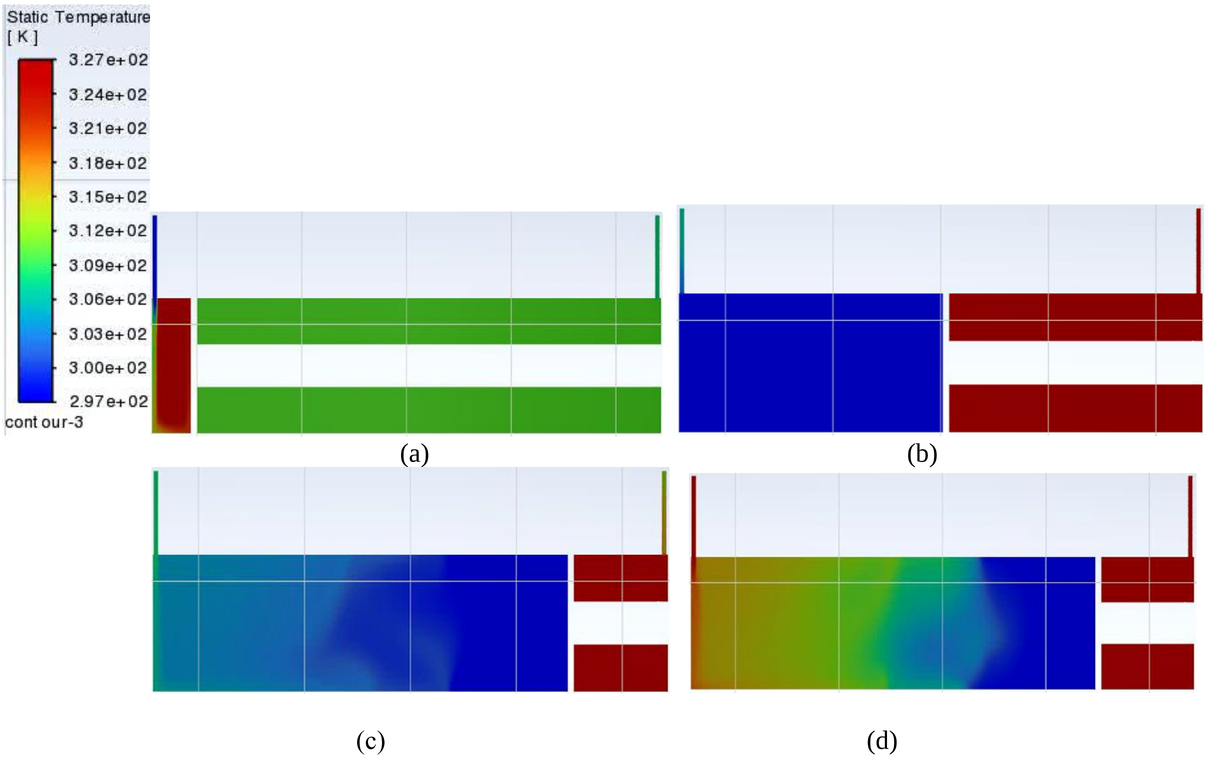

Fig. 3. Air temperature in the pneumatic cylinder: (a) time 0.1 s; (b) 0.4 s; (c) 0.7 s; (d) 0.75 s.

The numerical model is based on the system of Reynolds-averaged equations of motion, the conservation equations of mass and energy, and the corresponding equations of state of the working medium. The mathematical formulation of the problem is described in detail in [28, 29]. Closure of the system of equations was achieved using the Shear Stress Transport (SST) turbulence model (Fig. 2).

A specific feature of the problem is the presence of moving boundaries of the computational domain caused by the reciprocating motion of the piston, which results in a continuous change in the volumes of the working chambers. The use of a stationary computational mesh in such a formulation is unacceptable because it does not allow adequate representation of the time-dependent change of the flow geometry and gas compressibility. Therefore, the CFD simulations were performed using a dynamic (adaptive) mesh that is automatically reconstructed according to the prescribed piston displacement. This approach ensured correct modelling of gas-dynamic and thermal processes in variable volumes and enabled the determination of air temperature (Fig. 3), pressure, and velocity during transient operating conditions of the cylinder. As can be seen from Fig. 2, during piston motion the temperature range in the chambers varies significantly; therefore, the assumption of isothermal processes adopted in the one-dimensional model introduces considerable inaccuracies.

Boundary conditions were defined according to the control scheme of the pneumatic drive and the conditions of the transient process. A no-slip boundary condition was imposed on all solid surfaces. At the inlet of the supply channel, total pressure values were specified and varied over time according to the control algorithm, while a fixed static pressure condition was applied at the outlet channel. The computational mesh contained approximately 0.5 million control volumes, with the value of the parameter y^+ close to 1 [28], which ensured correct application of the SST model without the use of wall functions.

To verify the independence of the results from discretization parameters, a series of simulations was performed using meshes containing 0.1 million, 0.3 million, and 0.5 million elements. Comparative analysis showed that the differences in integral characteristics, particularly the outlet mass flow rate, between the two most refined meshes did not exceed 1%, indicating the achievement of mesh convergence. In addition, the influence of the number of iterations per time step of 0.0005 s was analysed. It was found that starting from 7 iterations per time step, further increase in the number of iterations had practically no effect on the numerical simulation results.

RESEARCH RESULTS AND THEIR DISCUSSION

The phenomenon of compressed air recuperation occurs when the pressure in the chamber is higher than the supply line pressure ($p_2 > 0.6$ MPa). n this case, the check valve in the exhaust line opens, and the compressed air flows into the supply line (receiver). As soon as the pressure drops below the supply line pressure ($p_2 \le 0.6$ MPa) – the check valve closes (at this moment, the rod-side chamber is connected to the atmosphere, and the pressure p_2 rapidly decreases to atmospheric level). The mass of air recuperated into the network is:

$$m_{rec} = \int_{t_s}^{t_f} \frac{p_2(t) Q_{rec}(t)}{RT_2(t)} dt, \tag{6}$$

where t_s, t_f – the start and end times of the recuperation process; Q_{rec} – volume of air being recuperated; T_2 – temperature in the exhaust chamber.

The flow rate during the recuperation process Q_{rec}, and, therefore, the mass flow m_{rec} depend on the pressure difference between the cylinder exhaust chamber and the receiver. Experimental studies [18] indicate that, in real conditions, the pressure in the exhaust chamber is higher than the supply line pressure. However, in the mathematical model, it is assumed that the pressures in the cylinder chamber and the supply line are equal during the recuperation process: $p_2 = p_m$ (the horizontal segment at 0.6 MPa in Fig. 6). Therefore, the calculation of the mass flow (and, accordingly, the energy) of recuperated air using the one-dimensional model can be carried out only in a simplified manner, considering as recuperated the volume of air displaced from the piston chamber during the period of recuperation:

$$E_{rec} = m_{rec}\, R\, T_m \ln \frac{p_1}{p_a}, \tag{7}$$

where $m_{rec} = \dfrac{p_2 \cdot V_{rec}}{R \cdot T_m}$ – mass of air recuperated into the supply network;

$V_{rec} = \dfrac{\pi \cdot \left(d_p^2 - d_s^2\right)}{4} \Delta x_{rec}$ – volume of air recuperated;

Δx_{rec} – piston displacement during the recuperation process.

Then, the efficiency of the pneumatic system is defined as:

$$\eta = \frac{E_{mech}}{E_{in} - E_{rec}}, \tag{8}$$

where $E_{mech} = P \cdot L$ – mechanical energy spent to move the load; $E_{in} = m_{in} \cdot R \cdot T_m \ln \dfrac{p_1}{p_a}$ – energy of compressed air delivered into the working chamber.

From Fig. 6, it can be seen that over the time interval 0…0.35 s, corresponding to the piston displacement of $\Delta x_{1-1} = 175$ mm in the acceleration phase, the pressure in the piston chamber p_1 is close to the supply line pressure. Therefore, the formula for the mass of air entering the working chamber during the acceleration phase is valid:

$$m_{in\_acc} = \frac{p_1 \cdot V_{acc}}{R \cdot T_m}, \tag{9}$$

where $V_{acc} = \dfrac{\pi \cdot d_p^2}{4} 0.175$ – volume of compressed air entering the working chamber in the acceleration phase.

During the braking phase, the piston chamber is connected to the supply source, and the mass flow into the piston chamber is absent.

At the same time, the applied calculation method (7–9) has limited accuracy, as it does not account for the variation in the temperature of the working medium (Fig. 3 shows that the temperature changes significantly during piston motion) in the pneumatic cylinder chambers. As a result, part of the work performed is implicitly converted into the internal energy of the compressed air, which is not reflected in the energy balance of the one-dimensional model.

Further analysis should focus on the quantitative evaluation of the system's energy parameters, namely, the determination of energy consumed, expended, and recuperated during a single working cycle. This approach allows moving from a qualitative description of the process to an accurate comparison of the efficiency of different operating modes and the determination of the pneumatic drive efficiency.

The energy balance of the pneumatic system (Fig. 1) over one working cycle (piston extension under static load, retraction without load) is:

$$E_{in} - E_{rec} + E_{ret} = \\ = E_{out} + \Delta E_{sys} + W_{mech} + E_{kin} + E_{dest}, \tag{10}$$

where E_{in} – total energy supplied to the piston chamber during the half-cycle (extension of the actuator);
E_{rec} – energy of the recuperated air;
E_{out} – energy of the air vented to the atmosphere from the rod-side chamber;
ΔE_{sys} – change in the internal energy of the air in the cylinder chambers over the half-cycle;
E_{mech} – useful mechanical work performed by the drive;
E_{kin} – half-cycle averaged kinetic energy of the moving parts of the drive;
E_{dest} – exergy losses (dissipation).

In this case, the useful component in formula (10) is E_{mech}, while all other terms represent losses of the supplied energy. The efficiency of the half-cycle (piston extension under useful load) is then:

$$\eta_{sys} = \frac{E_{mech}}{E_{in} - E_{rec}}. \tag{11}$$

Taking into account the temperature variation in the chamber, made possible by the 3-D model, the energy of the air supplied to the piston chamber from the network over the working half-cycle is:

$$E_{in} = \int_{t_{in}^{s}}^{t_{in}^{f}} \dot{m}_{in}(t) \cdot \psi\left(p_1(t), T_1(t)\right) dt, \tag{12}$$

where m_{in} – mass flow of air entering the piston chamber (Fig. 7).

The energy of the air recuperated to the network (receiver) is:

$$E_{rec} = \int_{t_{rec}^{s}}^{t_{rec}^{f}} \dot{m}_{rec}(t) \cdot \psi\left(p_2(t), T_2(t)\right) dt. \tag{13}$$

where $\psi(p,T)$ – exergy of the air flow;
m_{rec} – mass flow of recuperated air (Fig. 7).

In the general case, the exergy of an ideal gas flow is ψ :

$$\psi(p,T) = c_p\left(T - T_0\right) - \\ -T_0\left(c_p \ln\frac{T}{T_0} - R\ln\frac{p}{p_0}\right), \tag{14}$$

where c_p – specific heat capacity of air; R – gas constant for air; T, p – temperature and pressure in the chamber from which the flow originates; T_0, p_0 – temperature and pressure in the chamber to which the flow is directed.

The useful mechanical work performed by the drive to move the load is:

$$E_{mech} = P \cdot L, \tag{15}$$

where P – static load on the pneumatic cylinder piston; $x_s = x_{01}$ – initial point of the piston trajectory; L – piston displacement.

The energy supplied to the rod-side chamber over the half-cycle of piston retraction is calculated in a simplified manner, assuming that the exergy of the compressed air entering the rod-side chamber equals the exergy of the air in the working chamber during piston extension (14), and the mass flow of air into the rod chamber (assuming isothermal process) is:

$$m_{ret} = \frac{p_m \cdot V}{R \cdot T_m}, \tag{16}$$

where V – volume of the rod-side chamber.

EXAMPLE OF CALCULATING THE EXERGY EFFICIENCY OF A PNEUMATIC SYSTEM

To obtain the characteristics of the working process of the pneumatic system, its transient process was calculated using the mathematical model [18] with a 4th-order Runge-Kutta numerical method and a fixed integration step $0.5 \cdot 10^{-3}$ with respect to the following variables: p_1 – pressure in the piston chamber; p_2 – pressure in the rod-side chamber; x – current piston coordinate; v – piston velocity.

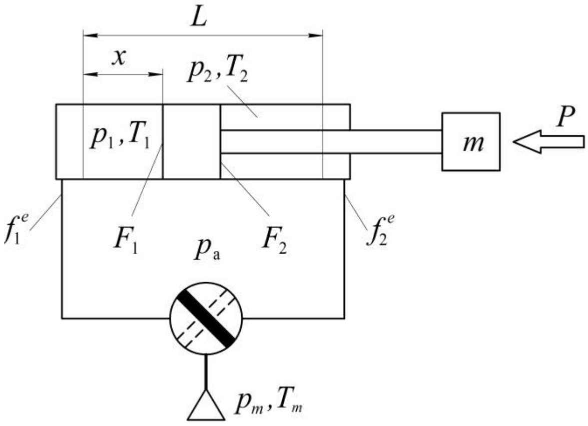

Fig. 4. Pneumatic system calculation schematic

The system's design and calculation parameters (Fig. 4) are: $k = 1.4$; $f_1^e = 14 \cdot 10^{-6}\ m^2$; $f_2^e = 14 \cdot 10^{-6}\ m^2$ – effective flow areas of the inlet and exhaust lines; R – gas constant, $R = 288\ \frac{J}{kg \cdot \deg}$; T_m – air temperature in the supply line, $T_m = 300\ ^\circ K = 27\ ^\circ C$; $F_1 = 3.115 \cdot 10^{-3}$ m^2 (piston area $d_p = 63 \cdot 10^{-3}$ m); $F_2 = 2.801 \cdot 10^{-3}$ m^2 (stroke area $d_s = 20 \cdot 10^{-3}$ m); $p_м = 0.6$ МПа; $x_{01} = 0.01$ м; $x_{02} = 0.015$ м; $L = 0.4$ m; $m = 300$ kg; P – load on the piston, $P = 650$ N; p_a – atmospheric pressure, $p_a = 0.1$ MPa; $p_k = 0.2$ MPa.

The obtained transient characteristics are shown in Figs. 5–6. From Fig. 5, it can be seen that the piston accelerates and decelerates with nearly constant acceleration; there are no sharp velocity changes or pneumatic rebounds. The maximum velocity in the acceleration phase is approximately 0.9 m/s.

Analyzing the graph in Fig. 6, it can be concluded that the potential energy of the compressed air in the working chamber is utilized, since at the beginning of the braking phase the piston chamber is connected to the pressure-reducing valve, and during the piston motion, there is no air consumption until the pressure p_1 reaches the setpoint of the pressure-reducing valve p_k.

Fig. 6 also indicates the presence of compressed air recuperation to the supply line, observed as the horizontal segment of the pressure curve p_2 in the exhaust chamber at approximately 0,6 MPa).

From the pressure change graph p_2 (Fig. 5) the duration of the air recuperation process was calculated as: $t_{rec} = 0.5945 - 0.5455 = 0.049$ s, which is approximately 6.5% of the total working cycle duration. Therefore, it can be concluded that the pneumatic drive scheme (Fig. 1) provides a stable and safe working process and is energy-efficient.

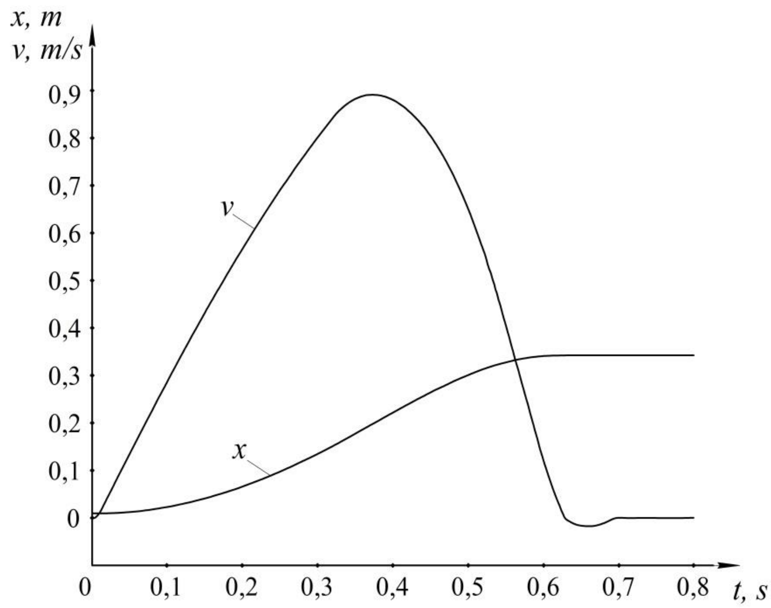

Fig. 5. Transient process in a pneumatic system with energy recuperation calculated using a one-dimensional model, showing the piston velocity and displacement.

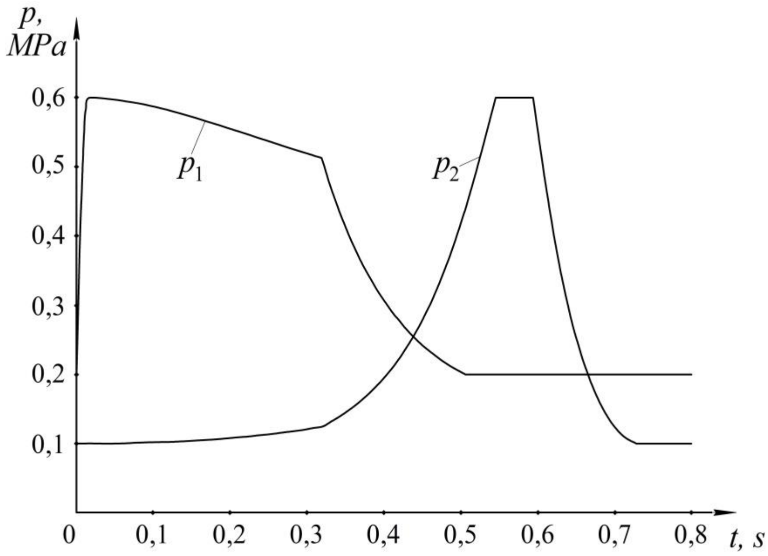

Fig. 6. Transient process in a pneumatic system with energy recuperation calculated using a one-dimensional model, showing the pressures in the piston and rod-side chambers.

The efficiency of the transient process, calculated using the one-dimensional simplified model (8) with air recuperation taken into account, was $\eta = 32.54\%,$ which is at the level of typical industrial pneumatic systems 20-30%.

To improve the accuracy of the energy efficiency assessment of the pneumatic system, a simulation model of the pneumatic system was implemented in ANSYS Fluent, which allows consideration of the temporal and spatial variation of thermodynamic parameters of the working medium. Using the CFD model of the pneumatic system, the mass flow of air from both cylinder chambers was determined (Fig. 7).

During the time interval $t = 0.32\ldots0.49$ s (during braking), there is no flow in the system. This is explained by the closure of the exhaust chamber, where, until the pressure reaches the level required to open the check valve and start the recuperation process, the flow is zero. Additionally, the piston (working) chamber is connected to the reduced-pressure supply source – the pressure-reducing valve ($p_k = 0.2$ MPa) at a moment when the chamber pressure is approximately twice as high ($p_1 \approx 0.4$ MPa) (Fig. 6). Therefore, during this period, the piston motion occurs by utilizing the internal energy of the compressed air, which accumulates due to the conversion of the kinetic energy of the moving masses of the drive.

The graphs confirm the presence of air recuperation during the braking phase. It can be seen that the amount of air vented from the exhaust chamber to the atmosphere is comparable to the amount of air recuperated to the receiver (lower curve in Fig. 7).

From the mass flow versus time data in the cylinder chambers obtained from the CFD model (Fig. 7), the total mass of working medium supplied to the piston chamber was calculated: $m_{in} = 0.0069$ kg; and the mass of recuperated air: $m_{rec} = 0.000695$ kg. Using formula (14), the exergy of each of these flows was determined; using formulas (12–16), the energy of the consumed air $E_{in} = 491.65$ J and the energy of the recuperated gas $E_{rec} = 14.58$ J.

The useful mechanical work performed during one half-cycle was determined from formula (15) as $E_{mech} = 230.75$ J. These values allowed the calculation of the working cycle efficiency of the pneumatic system using the exergy method:

$$\eta_{sys} = 48.37\%. \qquad (17)$$

The efficiency value of 48.37% is significantly higher than the typical efficiency of industrial pneumatic drives, indicating the feasibility of using the proposed scheme with the reutilization of excess energy and the optimized operation process. At the same time, this result is 1.5 times higher than the efficiency obtained using the simplified calculation method (7–9). This outcome is a consequence of accounting for the temperature variation of the working medium

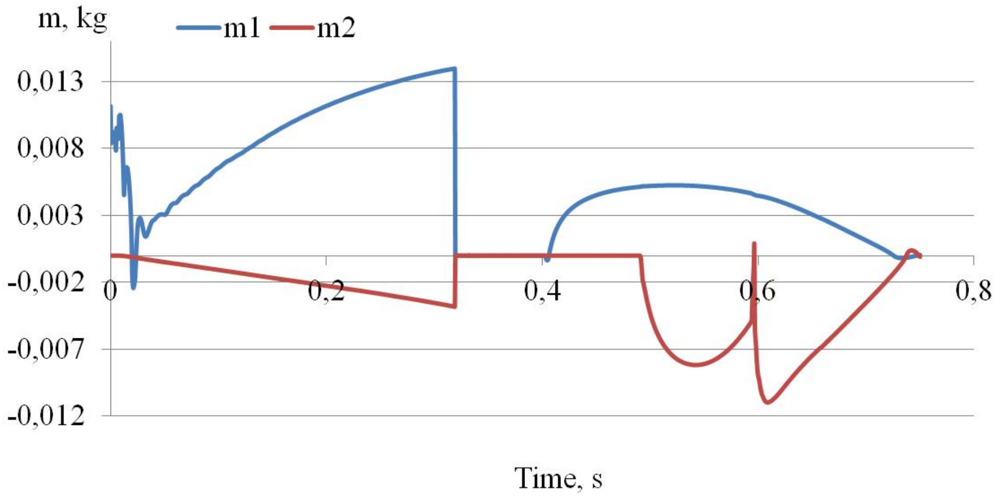

Fig. 7. Mass flow from the pneumatic cylinder chambers during a single actuation (motion from left to right).

and, consequently, the internal energy transitions within the cylinder chambers, demonstrating the promising potential of further application of three-dimensional simulation modelling for pneumatic systems.

CONCLUSIONS

An energy-efficient operating mode of the pneumatic drive with controlled braking was investigated, which is implemented by optimizing the switching algorithm and modifying the structure of the cylinder chamber connections. This approach enables the recuperation of part of the compressed air to the supply line and allows maximum utilization of the internal energy of the working medium during the braking phase.

It was shown that the one-dimensional mathematical model is an effective tool for preliminary analysis of the dynamics and for determining an energy-justified transition point to the braking mode; however, it does not provide a reliable assessment of energy efficiency. The main reason for its limited accuracy is the assumption of isothermal processes, neglect of temperature fluctuations, and the inability to correctly determine the actual pressure in the exhaust chamber and, consequently, the mass flow of recuperated air and internal energy redistribution within the cylinder chambers.

The application of three-dimensional CFD modelling using a dynamic mesh allowed accounting for unsteady gas dynamics and thermodynamic effects, obtaining spatiotemporal distributions of pressure and temperature, and accurately determining the mass flow of recuperated air, which significantly refined the system’s energy balance.

Exergy analysis showed that the efficiency of the pneumatic system reaches 48.37%, which is significantly higher than that of typical industrial pneumatic drives and more than one and a half times greater than the value obtained using the simplified one-dimensional model.

The obtained results confirm that CFD modelling of transient processes in a pneumatic drive allows the evaluation of the energy efficiency of pneumatic systems at the design stage, providing data (cylinder chamber pressures, velocity, and temperature fields) that can otherwise only be obtained experimentally. This, in turn, is a necessary condition for improving the accuracy of efficiency determination in pneumatic systems with energy recuperation and provides a scientifically justified basis for optimizing the control of pneumatic drives during the design stage.

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