EXERGY ANALYSIS FOR THE INTERMITTENT AIR SUPPLY IN PNEUMATIC MACHINES

Dominik Gryboś[1]*, Jacek Leszczyński[1]

[1] *AGH University of Krakow, Faculty of Energy and Fuels, Department of Department of Thermal and Fluid Flow Machines, Av. Mickiewicza 30, 30-059 Krakow, Poland*

* Corresponding author: Tel.: +48 12 617 33 89; E-mail address: dgrybos@agh.edu.pl

ABSTRACT

Pneumatic systems are widely used to automate production lines in manufactured plants. Their big disadvantage is their low energy efficiency, 10-20%. It is mainly due to the overconsumption of compressed air by the oversized pneumatic actuator and other components. Reducing air consumption at the utilisation stage can result in significant exergy savings. This can be done by lowering the supply pressure, introducing back pressure, or using expansion energy in the actuator. The most promising is the last method that can be implemented using the intermittent air supply in the pneumatic actuator. The literature lacks an exergy analysis of the utilisation stage of pneumatic and intermittent air supply systems. There is also no optimisation of the control algorithm of intermittent air supply control algorithm in terms of minimising exergy consumption. In this paper, we demonstrate a mathematical model and exergy analysis as a tool for assessing the efficiency of the utilisation of pneumatic system and conduct computer simulation. Exergy analysis showed that for intermittent air supply the reduction in exergy consumption decreased by more than 60% compared to the classic oversized system. The results of the computer simulation give the opportunity to optimise the operation of the utilisation stage in pneumatic systems. Furthermore, exergy analysis can be a useful tool for energy analysis and assessment of pneumatic systems, as well as providing information on the desired direction of changes in the installation.

Keywords: Compressed air system, pneumatics, air use, intermittent air supply, Exergy analysis, sensitivity analysis

1. INTRODUTION

Compressed air systems (CAS) (pneumatic systems) are widely used in numerous industrial sectors such as light, food, paper, plastic and automotive. It is estimated that pneumatic systems consume 10% of the electricity consumed in the industrial sector in the United States, the EU and China. The use of compressed air to power machines and production lines is characterised by high power density, low investment costs, simple expansion, and high reliability [1]. However, pneumatic systems have very low energy efficiency, approximately 5%-10%, cite [2]. Furthermore, the compressed air system is characterised by poor controllability [3] and high noise [4]. Energy losses in CAS occur during compression, transmission due to leaks [5] and the utilisation stage through excessive air consumption [6]. Methods to improve efficiency in compressors and air leaks are well described in the literature and technology [7–9]. However, methods for reducing compressed air consumption are less well presented and can achieve an increase in energy efficiency by up to 40%.

The oversizing of pneumatic actuators is the main reason for the overconsumption of compressed air by CAS due to the design and selection process of pneumatic actuators and their standardisation [10, 11]. The pneumatic actuator is selected for at least 33% greater or even 50% greater force generated from the load and a piston speed condition of 0.3 to 0.5 m/s [10, 12]. For operational reliability and safety, it is customary to select a one-size larger pneumatic actuator. The degree of actuator oversizing can be determined by the pneumatic frequency factor presented by [13]. Additionally, dead volumes in system components also result in excessive energy consumption of compressed air [14].

In the literature, there is a preliminary classification of methods for reducing air overconsumption: reducing the supply pressure [11, 15], increasing the back pressure [16, 17] or using the expansion power in the actuator [18]. This results directly from the reduction of the compressed thermodynamic cycle field [19]. Reducing the supply pressure is the most commonly used method by using pressure reducers or, rarely, a meter in systems. The potential energy savings of compressed air range from 20% to even 40% [11, 16]. Increasing back pressure involves introducing a certain pressure greater than the ambient pressure at the outlet ports of the pneumatic actuator. As a result, the pneumatic actuator operates within the limit of the bottom pressure range. Back pressure air is stored in a buffer tank and its reuse allows for energy savings of 40-75%. In the commonly used algorithm for supplying compressed air to actuators, only air transmission power is used, while air expansion power is lost. The method of using expansion power is implemented through intermittent air supply and a directional control valve bridge system [18, 20]. Air expansion power is used through an appropriate algorithm controlling the suppling and cutting off of compressed air and monitoring the position of the piston and the pressure in the chambers during stroke. This method can achieve energy savings of up to 85% [18]. However, these systems require complex control algorithms, additional sensors, and a fast switching direction control valve [21]. Although this is the method that gives the greatest savings, the proper optimisation of compressed air supply is complex.

Assessment of the energy efficiency of a compressed air system is complex, and therefore it is advisable to use a system analysis approach. The first law of thermodynamics for a compressed air system (energy analysis) has limited application due to near-isothermal transformations and slight changes in air temperature and energy [22]. Much more adequate is the second law of thermodynamics and exergy analysis, which takes into account not only changes in temperature but also pressure and the irreversibility of changes. In the literature there is some static exergy analysis of the air compression and preparation stage [23–27]. These analyses lack an estimate of the actual consumption of compressed air, which is defined by the final devices in the utilisation stage. Rakova et al. [14] present a calculation of the accumulated exergy of compressed air in separate pneumatic components. Merkelbach and Murrenhoff [28] conducted experimental and computer comparisons of exergy consumption for several air savings configurations of the compressed air system and obtained a reduction in exergy consumption of 20-50%. The temporary consumption of compressed air results from the non-stationary transition of compressed air from state A to B, which is then repeated cyclically. Therefore, the tools presented are not suitable for a detailed analysis of processes at the utilisation stage.

In the paper, we show the exergy analysis for compressed air system and intermittent air supply. For this purpose, an original mathematical model of compressed air systems was used and computer simulations were performed. The level of exergy consumption is shown for various parameters that shape the air supply signal to the actuator. This tool has the ability to use an exergy criterion to optimise the operation of pneumatic systems seems to be the key novelty. This work focused on the exergy analysis of only the utilisation stage, and a mathematical model of system dynamics was implemented only for this stage.

2. METHODS

Industrial compressed air systems (see **Figure 1**) consist of three stages: compression and preparation, storage, and utilisation. Each CAS stage consists of characteristic elements: a compressor, a heat exchanger, a dryer, and a filter in the air compression and preparation stage. Air tank with equipment in the storage stage and pneumatic lines, directional control valves, pneumatic actuator, and muffler. According to the theory of air power [29], it can be divided into transmission and expansion power, where only the former is used in classic control.

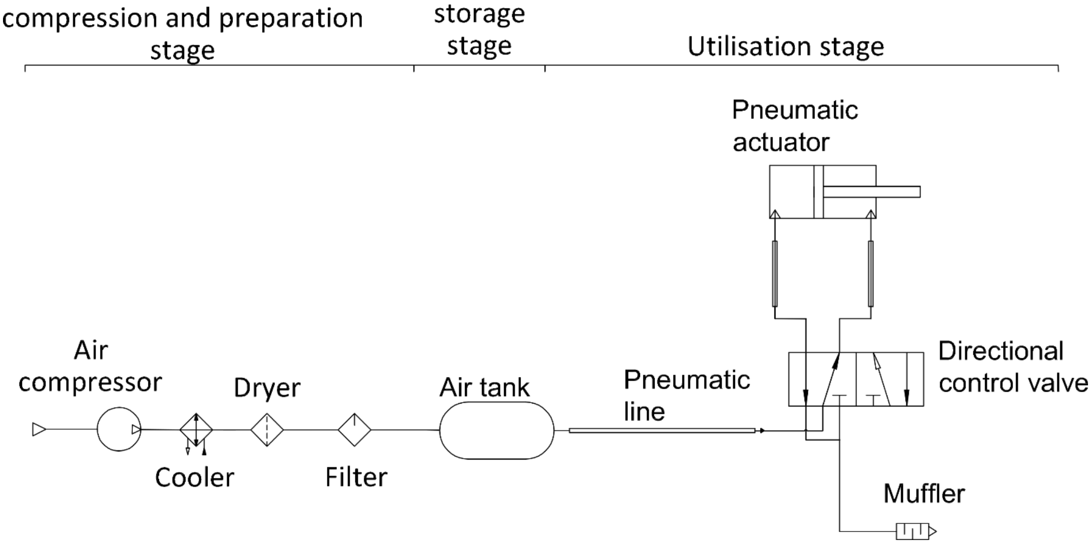

Figure 1: Compressed air system.

2.1. Mathematical model

A detailed description of the the mathematical model of individual components of the compressed air system in the utilisation stage is presented in [30]. The compression and utilisation stages in the compressed air system operate indirectly, independently of each other, with the buffer as a tank. Therefore, the work of individual stages can be considered separately with the connecting element, which is the tank. The mathematical model of the dynamic of the the utilisation stage of compressed air system is based on the flow of compressed air between certain volumes (constant – tanks, valves, mufflers) and variables (actuator) through the transmission line. For each volume, a system of three differential equations should be written describing the mass balance (1), the energy balance (2) and, for variable volumes, an additional equation for the dynamics of motion (3). An example system of equations for the i-th volume and where the index 1 is the inlet and 2 is the outlet:

$$\frac{\mathrm{d}m_i}{dt} = \dot{m}_{i,1} - \dot{m}_{i,2} \tag{1}$$

$$\frac{dT_i}{dt} = \frac{1}{m_i\ C_{v_i}}\ \dot{m}_{i,1}\left(h_{i,1} - h_i\ \right) - \dot{m}_{i,2}\left(h_i - h_{i,2}\right) + \dot{Q}_i - \dot{W}_i \tag{2}$$

$$\frac{d^2x_i}{dt^2} = A_1\left(p_{i,1} - p_{i,2}\right) - A_2p_0 + F_c + F_{fr} \tag{3}$$

where: m is air mass in the volume, $\dot{m}$ is air mass flow, T is air temperature, C_{v_i} is specific heat capacity at constant volume, h is air enthalpy, $\dot{Q}$ is heat flux, $\dot{W}$ is work flux, x is piston position, A_1 is piston cross section, A_2 is piston bore cross section, p is air pressure, F_c is contact force, F_{fr} is Columbus-viscous friction force with Stribeck effect. In addition, the pressure equation (4) and the air mass flow equation should be written for each volume. More details about the developed model and its validation can be found in [30].

$$p_i = \frac{RT_i\ m_i}{V_i} \tag{4}$$

where: R is individual gas constant for air and V is volume. The air mass flow model is calculated by formula presented by Leszczyński i Gryboś [31] where ζ_1 is the reduction coefficient related to flow contraction phenomenon and $\zeta_2$is geometry coefficient of outlet/inlet port.

$$\begin{cases} \zeta_1 A_i \sqrt{\frac{2\gamma}{\gamma-1}\rho p_1\left(\left(\frac{p_2}{Zp_1}\right)^{\frac{2}{\gamma}}-\left(\frac{p_2}{Zp_1}\right)^{\frac{\gamma+1}{\gamma}}\right)} & \text{for } \frac{p_2}{p_1}>\beta \\ \zeta_1 A_i \sqrt{\gamma\rho p_1\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}} & \text{otherwise} \end{cases} \tag{5}$$

where: A_i is inlet/outlet port cross section, γ is polytropic exponent, ρ is air density and Z is scaling function. The ζ_2=1 means ideal smooth orifice edges and results in the critical pressure ratio β movement to lower values.

$$\beta = \zeta_2\left(\frac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}} \tag{6}$$

$$Z = \frac{1-\zeta_2}{1-\beta^{2\zeta_2}}\left(\frac{p_1}{p_2}\right)^{2\zeta_2} + \frac{\zeta_2-\beta^{2\zeta_2}}{1-\beta^{2\zeta_2}} \tag{7}$$

For a pneumatic muffler, the air mass flow is calculated using the Ergun formula [32] as a phenomenon of flow through a porous surface. The outlet mass flow $\dot{m}_{\mathrm{i},2}$ from a pneumatic can be determined as the positive square root of the quadratic function:

$$c_3 \cdot\left(\dot{m}_{i,2}\right)^2 + c_4 \cdot\left(\dot{m}_{i,2}\right) - \frac{p_{i,2}-p_i}{\delta_i} = 0 \tag{8}$$

Where: where δ_{i} is thickness of the porous layer, c_3 is internal losses in porous material and c_4 viscous losses in porous material. The mathematical model of dynamic flow through the pneumatic line based on one-dimensional ordinary differential equations for pressure and mass flow for each segment [11, 33]:

$$\frac{dp_j}{dt} = -\frac{p_j}{\rho_j A_j}\frac{\dot{m}_{i+1}-\dot{m}_i}{\Delta x} \tag{9}$$

$$\frac{d\dot{m}_i}{dt} = -A_i\frac{p_{j-1}-p_j}{\Delta x} - \frac{\lambda_i}{(\rho_{j-1}+\rho_j)A_i d_i}\dot{m}_i|\dot{m}_i| \tag{10}$$

2.2. Exergy analysis

The complete exergy balance is presented in [30]. The change in exergy over time of an air in a given control volume of a compressed air system is the sum of the exergy flow through this volume transported with mass ($\dot{X}_i^k(t) - \dot{X}_{i-1}^k(t)$), work $\dot{X}_{w\,i}^k(t)$, heat $\dot{X}_{Q\,i}^k(t)$ and exergy destruction $\dot{X}_{d\,i}^k(t)$. The sign before the exergy term indicates the direction of exergy flow ("+" to the system, "-" from the system) and k is component pointer. Compressed air systems are open thermodynamic systems, therefore exergy equation should be:

$$\frac{dX_i^k(t)}{dt} = \dot{X}_i^k(t) - \dot{X}_{i-1}^k(t) + \dot{X}_{w\,i}^k(t) - \dot{X}_{Q\,i}^k(t) - \dot{X}_{d\,i}^k(t) \tag{11}$$

In the exergy analysis of the complete operating cycle of the pneumatic system, understood as the extend and retract movement of the actuator. It is assumed that the final air parameters at each point of the system return to their initial values in the ideal case. Therefore, the actuator movement is for extend movement from t_b to t_e, (from t_b to t_m) and retract movement (from t_m to t_e). Furthermore, constant ambient conditions (reference conditions) were assumed throughout the entire analysis period and the kinetic and potential energy of the air in the system were neglected. Taking into account the destruction exergy, the total change in exergy in a given volume over the complete actuator cycle from t_b to t_e is equal to zero $\frac{dX_i^k}{dt} = 0$. Therefore, the exergy of destruction in a given volume of the system can be defined as the balance of the remaining exergy streams in the system.

$$X_d^k = \int_{t_b}^{t_m} \left(\sum_{i=1}^{n} \left(\dot{X}_i^k(t) - \dot{X}_{i-1}^k(t) - \dot{X}_{Q_i}^k(t) + \dot{X}_{w_i}^k(t) \right) \right) dt + \int_{t_m}^{t_e} \left(\sum_{i=1}^{n} \left(\dot{X}_i^k(t) - \dot{X}_{i-1}^k(t) - \dot{X}_{Q_i}^k(t) + \dot{X}_{w_i}^k(t) \right) \right) dt \quad (12)$$

2.3. Computational setup

Computer simulations were carried out for two stages of utilisation of compressed air systems: classic and intermittent air supply, shown in **Figure 2**.

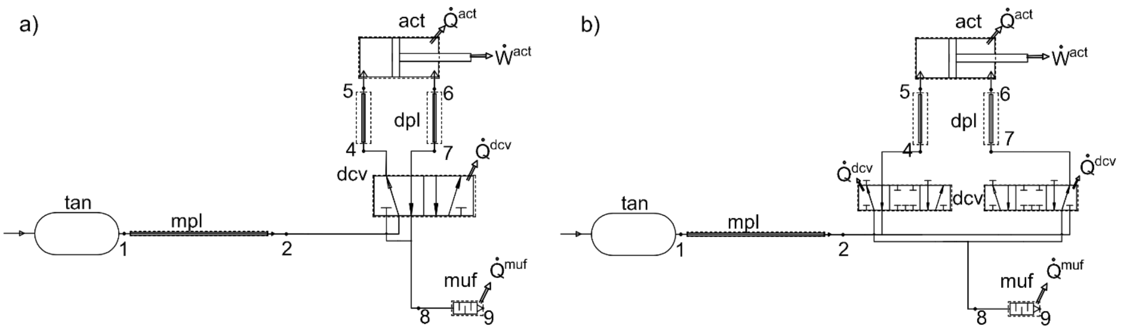

Figure 2: Stage of the compressed air system: a) Classical; b) Intermittent air supply. Where: tan – tank, mpl – main pneumatic line, dcv – directional control valve, dpl – distributed pneumatic line, act – actuator, muf – muffler.

The difference in the structure of the systems is the use of a directional control valve bridge in the case of intermittent air supply. The parameters of the system components and the process parameters are presented in **Table 1**.

Table 1: Parameters of the compressed air system

Parameter	Value
Supply gauge pressure	6.3 bar
Piston diameter	63 mm
Stroke	300 mm
Extend mass load	110 kg
Port size	G3/8
Main/distributed line diameter	12/8 mm
Main/distributed line length	5/2 m
Pneumatic frequency ratio	1.89

For intermittent air supply, a rigid algorithm was used to air supply to the actuator to examine the parameters of compressed air pulses and the sensitivity of the movement dynamics. The valve control signal (VOS) is based on the pressure signal (optional, can be omitted) in the chambers and the piston position and is defined as:

- For extend stroke

$$VOS == \begin{cases} 1 \text{ for } (x \leq D) \cup (x \leq a_1 \cap x \leq (a_1 s + a_2 D)) \cup (x \leq b_1 s \cap x \leq (b_1 s + b_2 D)) \\ 0 \; for \; otherwise \end{cases} \quad (13)$$

- For retract stroke

$$VOS = \begin{cases} 1 \text{for} (x \leq D) \cup (p_2 > p_1) \\ 0 \, for \; otherwise \end{cases} \tag{14}$$

where x is piston position, p_2 is pressure in retract chamber, p_1 is pressure in extend chamber and D is duty factor:

$$D = \frac{\Delta l}{s} \tag{15}$$

where: s is actuator stroke and Δl is the distance at which the actuator is supplied by compressed air. Figure 3 shows the shape of the compressed air supply signal for the extend and retract movement.

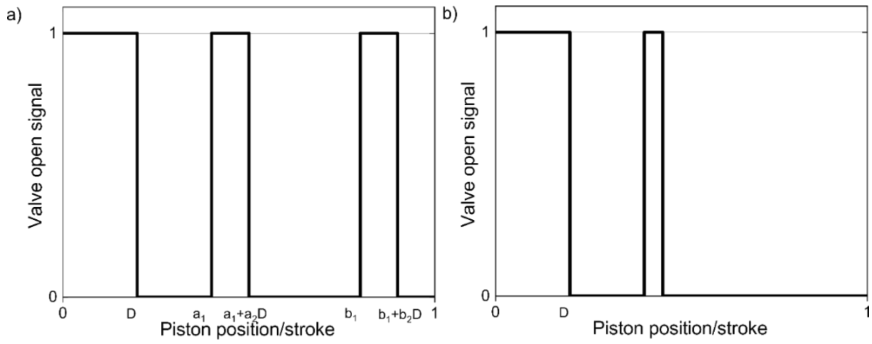

Figure 3: Valve open signal for: a) Extend stroke; b) Retract stroke.

Individual parameters defining the valve open signal are defined in **Table 2**.

Table 2: Sensitivity analysis of valve open signal

Parameter	Value
D	0.01/0.03/0.06
a_1	0.4
a_2	10/30/60
b_1	1
b_2	5

A sensitivity analysis was performed for 3 values of the D parameter and 3 values of the c parameter. A detailed model of the mathematical utilisation stage of the compressed air system was presented, implemented, and validated in [30]. The computer simulation was conducted in MATLAB~R2021b software with Runge-Kutty-Fehlberg solver. Furthermore, the thermodynamic library of moist air ASHRAE RP-1485, Thermodynamic Properties of Real Moist Air, Fog, Ice Fog, Dry Air, Steam, Water, and Ice, MATLAB version (FluidLAB) was used.

3. RESULTS

Computer simulations were carried out for a classic compressed air system and for an intermittent air supply with D = 0.01 and a_2 = 10. For such a case, the dynamics of the actuator operation was compared, and the changes in pressure, position, speed, and exergy flux were presented. The results for classical and intermittent air supply are presented in **Figure 4** and **Figure 5**. The use of an intermittent air supply instead of a classical system did not cause any disruptions in the actuator's operation. This means that the jump time remained practically unchanged and did not exceed 0.6 s, which means the average speed is more than 0.5 m/s. Additionally, during the extend movement, the piston speed increases faster, and the stroke remains constant for a longer period of time.

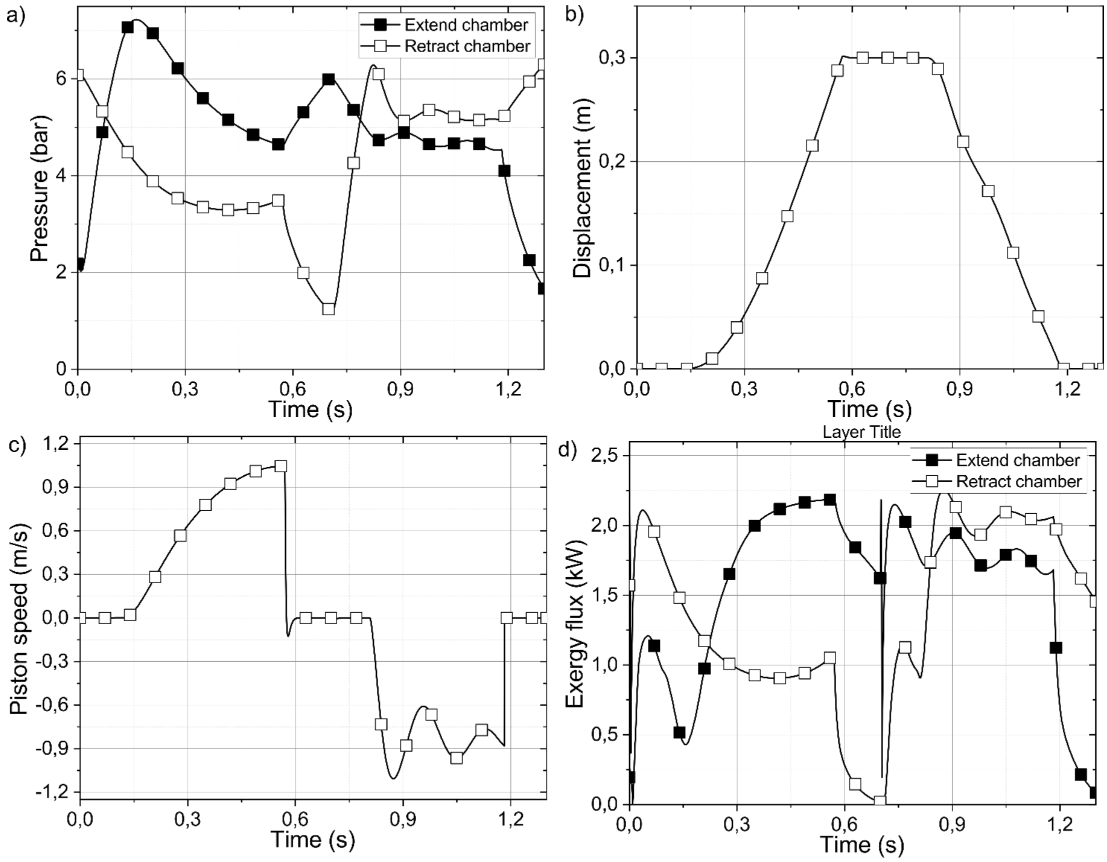

Figure 4: Pneumatic actuator dynamics for classical compressed air system: a) Pressure; b) Displacement; c) Speed; d) Exergy flux.

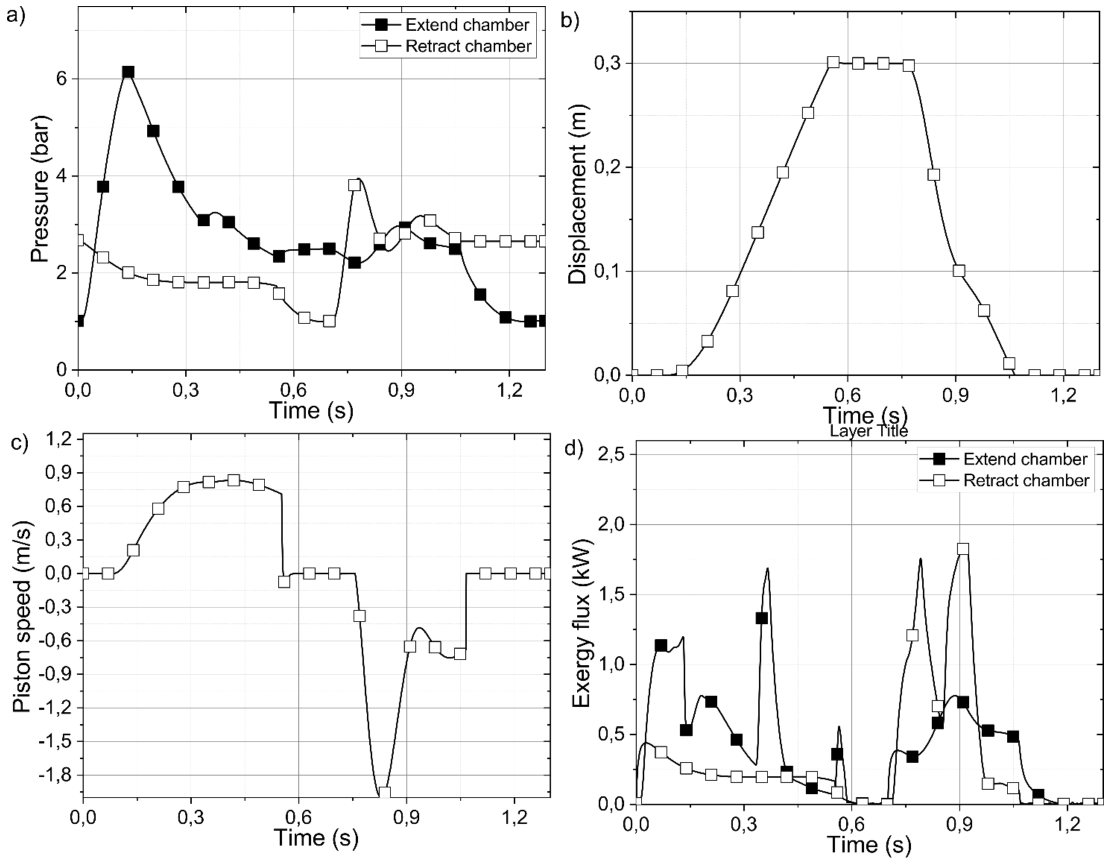

Figure 5: Pneumatic actuator dynamics for intermittent air supply: a) Pressure; b) Displacement; c) Speed; d) Exergy flux.

This is due, among other things, to the pressures that prevail in both chambers. In the case of intermittent air supply, the pressure during the retract movement is lower than the pressure in the network, and therefore the actuator needs less time to reload the harmful volumes. It also has a positive effect on the exergy flux, which is even 5 times smaller than in the classical case. The retract movement, although it is without load in both cases, is performed faster for intermittent air supply. **Figure 6** shows the Sankey diagram for the energy consumption of the classic system and intermittent air supply.

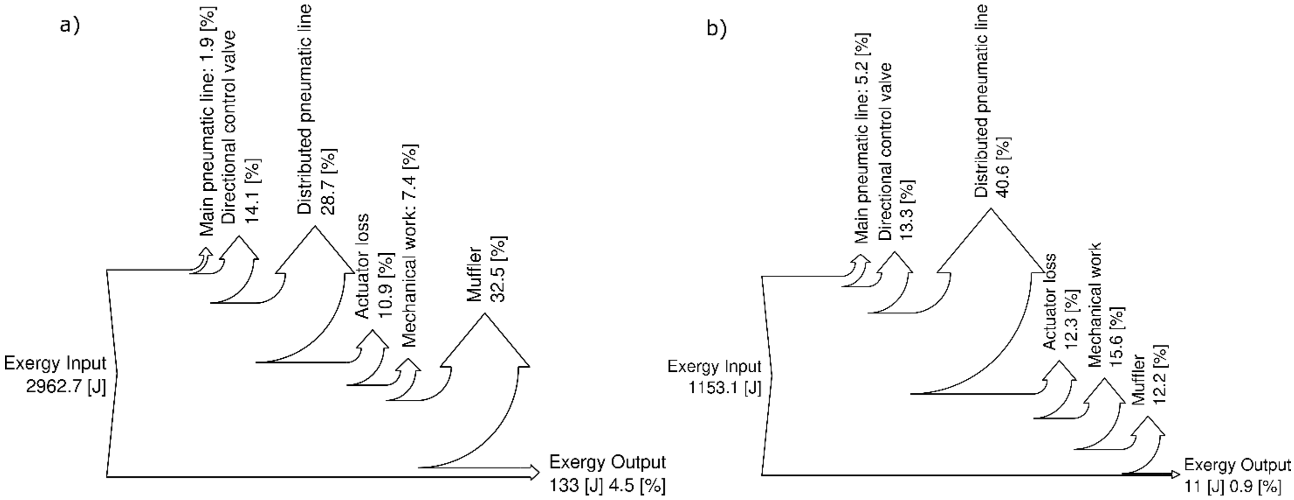

Figure 6: Sankey diagram: exergy flow in a) Classical; b) Intermittent air supply.

Exergy consumption decreased from 2962.7 J for the classical system to 1153.1 for intermittent air supply. These are savings in exergy consumption of 61%, which is caused by the compensation of dead volumes and the use of expansion air power during actuator operation. The burden of exergy losses has been transferred from the muffler and output to the distributed pneumatic line, and the exergy used in the actuator has been increased relative to the input exergy.

Then, a sensitivity analysis was performed for the shape of the valve open signal in terms of parameters D and a_2 (see Table 2). **Figure 7** shows the impact of changing these parameters on actuator dynamics.

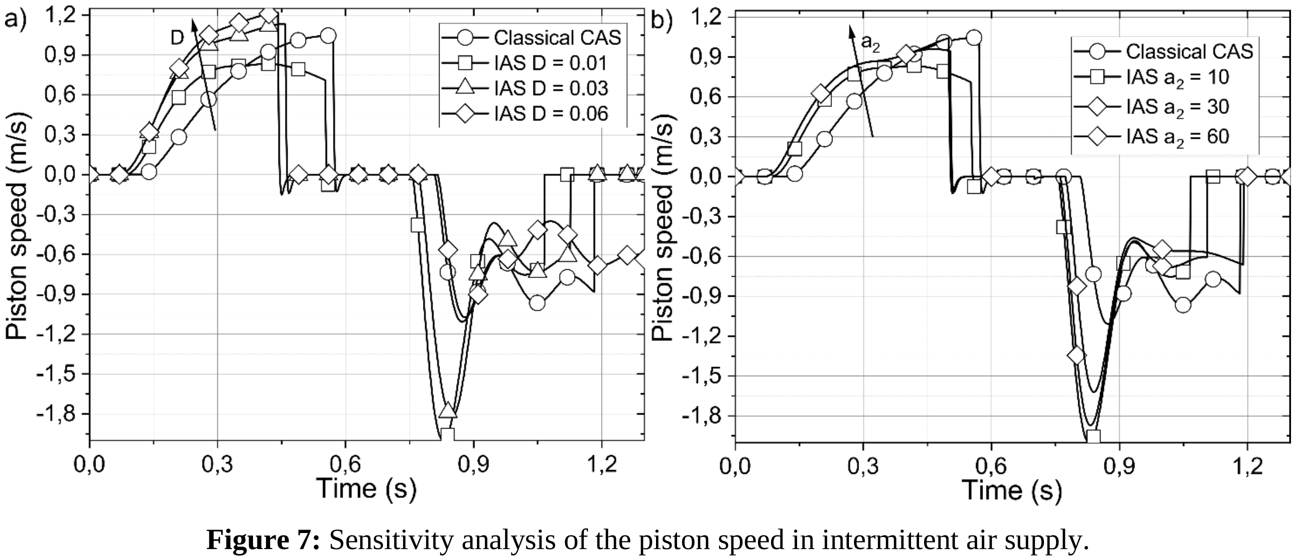

Figure 7: Sensitivity analysis of the piston speed in intermittent air supply.

As the D parameter increases, the actuator speed reaches higher values and becomes steeper. Additionally, the average speed of extend movement increases. However, the a_2 parameter affects the final speed of movement. With the larger parameter a_2, that is, with the extension of the second part of air supply, the final speed of the actuator is higher. This allows you to achieve the shock speed parameters for the process. However, as shown in **Figure 8**, with increasing parameters *D* and *a2*, exergy consumption increases. For the parameters a_2 the increases are much smaller than for *D*.

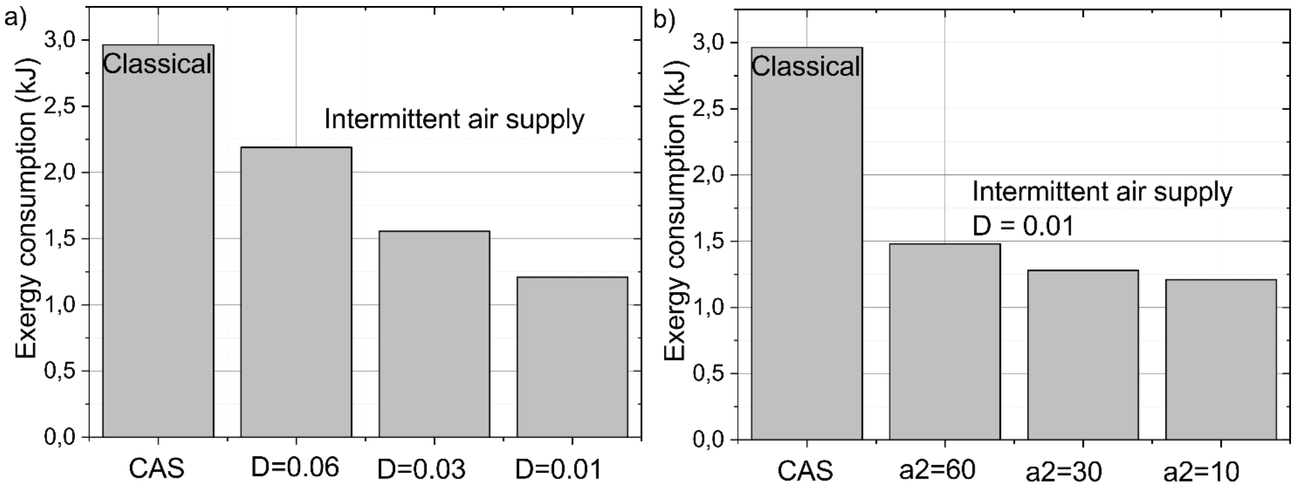

Figure 8: Sensitivity analysis of exergy consumption in intermittent air supply.

The results presented show that intermittent air supply provides large savings in exergy consumption exceeding 60-70%. However, exergy analysis can be very useful when programming the movement of an actuator piston via a valve open signal. By defining the limit conditions for the actuator movement, such as movement time, required average speed,, required final speed or return movement time, it is possible to programme intermittent air supply in such a way as to minimise exergy consumption in the entire system. Furthermore, exergy analysis can be a useful tool for energy analysis and assessment of pneumatic systems, as well as providing information on the desired direction of changes in the installation.

4. CONCLUSION

The paper presents an exergy analysis and a comparison of exergy consumption for the utilisation stage of an oversized classic CAES system and an intermittent air supply system. The intermittent air supply system consists in providing a discontinuous signal to open the air supply to the actuator. A brief description of the mathematical model and the exergy balance of the system is presented. Then, a sensitivity analysis was performed showing the influence of the valve open signal shape parameters on the piston speed profile and exergy consumption by the utilisation stage. Significant reduction in exergy consumption by the system during an actuator operation cycle by more than 60%. This is caused by the use of air expansion power and dead volume compensation in the system. Additionally, the piston speed profile has not changed significantly, and its course is even more favourable. In percentage terms, the highest exergy losses in intermittent air supply are on the distributed pneumatic line as dead volumes. By modifying the parameters of the open valve signal, we influence the speed profile and the energy consumption of the system. Exergy analysis can be used to optimise the engine in the intermittent air supply to minimise the exergy consumption of the system under given process constraints. The use of exergy analysis to optimise actuator operation and obtain air savings is not common. Therefore, preliminary results of the analysis of the sensitivity of the valve open signal to exergy consumption and dynamics will be further investigated. They will include optimisation of the movement of the actuator dynamics with the function of minimising exergy consumption.

ACKNOWLEDGEMENT

Research project partly supported by program ‘‘Excellence Initiative – Research University’’ for the AGH University of Krakow and research subvention supported by the Polish Ministry of Education and Science, Poland grant number 16.16.130.942.

Supported by the Foundation for Polish Science (FNP)

NOMENCLATURE

A	Cross section	m^2
D	Duty factor	-
F	Force	N
V	Volume	m3
$\dot{Q}$	Heat flux	W
T	Temperature	K
$\dot{W}$	Work flux	W
$\dot{X}$	Exergy flux	W
X	Exergy	J
Z	Scaling function	-
a,b	Signal parameters	-
c	constant	-
h	Specific enthalpy	kJ/kg
$\dot{m}$	Mass flow	kg/s
m	Mass	kg
p	Pressure	Pa
s	Stroke	m
t	Time	s
x	Position	m
Δ	Interval	-
β	Critical pressure ratio	-
γ	Adiabatic exponent	-
δ	Thickness	m
ρ	Density	kg/m^3
ζ	Scaling factor	-

CAS	Compressed Air System
IAS	Intermittent Air Supply
VOS	Valve Open Signal

REFERENCES

[1] Cai M, Kawashima K, Kagawa T (2006) Power Assessment of Flowing Compressed Air. J Fluids Eng 128:402. https://doi.org/10.1115/1.2170129

[2] Saidur R, Rahim NA, Hasanuzzaman M (2010) A review on compressed-air energy use and energy savings. Renewable and Sustainable Energy Reviews 14:1135–1153. https://doi.org/10.1016/j.rser.2009.11.013

[3] Saravanakumar D, Mohan B, Muthuramalingam T (2017) A review on recent research trends in servo pneumatic positioning systems. Precis Eng 49:481–492. https://doi.org/10.1016/j.precisioneng.2017.01.014

[4] Czopek D, Gryboś D, Leszczyński J, Wiciak J (2022) Identification of energy wastes through sound analysis in compressed air systems. Energy 239:1–19. https://doi.org/10.1016/j.energy.2021.122122
[5] Dindorf R, Wos P (2018) Automatic measurement system for determination of leakage flow rate in compressed air pipeline system. Metrology and Measurement Systems 25:159–170. https://doi.org/10.24425/118170
[6] Nehler T (2018) Linking energy efficiency measures in industrial compressed air systems with non-energy benefits – A review. Renewable and Sustainable Energy Reviews 89:72–87. https://doi.org/10.1016/j.rser.2018.02.018
[7] Nourin FN, Espindola J, Selim OM, Amano RS (2022) Energy, Exergy, and Emission Analysis on Industrial Air Compressors. J Energy Resour Technol 144:. https://doi.org/10.1115/1.4051682
[8] Vittorini D, Cipollone R (2016) Energy saving potential in existing industrial compressors. Energy 102:502–515. https://doi.org/10.1016/j.energy.2016.02.115
[9] Doner N, Ciddi K (2022) Regression analysis of the operational parameters and energy-saving potential of industrial compressed air systems. Energy 252:124030. https://doi.org/10.1016/j.energy.2022.124030
[10] Fleischer H (1995) Manual of pneumatic system operation, 1st ed. McGraw-Hill, New York San Francisco Washington
[11] Beater P (2007) Pneumatic Drives. Springer Berlin Heidelberg, Berlin, Heidelberg
[12] Leszczyński JS, Gryboś D (2019) Compensation for the complexity and over-scaling in industrial pneumatic systems by the accumulation and reuse of exhaust air. Appl Energy 239:1130–1141. https://doi.org/10.1016/j.apenergy.2019.02.024
[13] Doll M, Neumann R, Sawodny O (2015) Dimensioning of pneumatic cylinders for motion tasks. River Publishers 16:11–24. https://doi.org/10.1080/14399776.2015.1012437
[14] Rakova E, Weber J (2015) Process simulation of energy behaviour of pneumatic drives. Procedia Eng 106:149–157. https://doi.org/10.1016/j.proeng.2015.06.018
[15] Harris P, O'Donnell GE, Whelan T (2012) Energy Efficiency in Pneumatic Production Systems: State of the Art and Future Directions. In: Proceedings of 19th CIRP International Conference on Life Cycle Engineering. pp 363–368
[16] Hepke J, Weber J (2013) Energy saving measures on pneumatic drive systems energy consumption of pneumatic drives. In: The 13th Scandinavian International Conference on Fluid Power, SICFP2013. Linkoping University Electronic Press, pp 475–483
[17] Hepke J, Weber J (2012) Improving Energy Efficiency of Pneumatic Handling System. In: Proceedings of The 7th International Fluid Power Conference. pp 1–5
[18] Doll M, Neumann R, Sawodny O (2011) Energy efficient use of compressed air in pneumatic drive systems for motion tasks. In: Proceedings of 2011 International Conference on Fluid Power and Mechatronics. IEEE, pp 340–345
[19] Cai M, Kawashima K, Kagawa T (2006) Power assessment of Flowing Compressed Air. Transaction of ASME 128:402–405. https://doi.org/doi:10.1115/1.2170129
[20] Raisch A, Hulsmann S, Sawodny O (2018) Saving Energy by Predictive Supply Air Shutoff for Pneumatic Drives. In: 2018 European Control Conference (ECC). IEEE, pp 965–970
[21] Pfeffer A, Glück T, Schausberger F, Kugi A (2018) Control and estimation strategies for pneumatic drives with partial position information. Mechatronics 50:259–270. https://doi.org/10.1016/j.mechatronics.2017.09.012
[22] Brol S, Czok R, Mróz P (2020) Control of energy conversion and flow in hydraulic-pneumatic system. Energy 194:116849. https://doi.org/10.1016/J.ENERGY.2019.116849

[23] Krichel S, Sawodny O, Hülsmann S, et al (2012) Exergy flow diagrams as novel approach to discuss the efficiency of compressed air systems
[24] Bader, W. T.; Kissock JK (2000) Exergy Analysis of Industrial Air Compression. IETC - Industrial Energy Technology Conference 89–98
[25] Harris P, Nolan S, O'Donnell GE, Meskell C (2013) Optimising Compressed Air System Energy Efficiency - The Role of Flow Metering and Exergy Analysis. In: Re-engineering Manufacturing for Sustainability. Springer Singapore, Singapore, pp 129–134
[26] Taheri K, Gadow R (2017) Industrial compressed air system analysis: Exergy and thermoeconomic analysis. CIRP J Manuf Sci Technol 18:10–17. https://doi.org/10.1016/j.cirpj.2017.04.004
[27] Tong Z, Wang H, Xiong W, et al (2021) Accumulated and transient exergy analyses of pneumatic systems with isochoric and isobaric compressed air storage tanks. Energy Storage 1–10. https://doi.org/10.1002/est2.269
[28] Merkelbach S, Murrenhoff H (2015) Exergy based analysis of pneumatic air saving measures. ASME/BATH 2015 Symposium on Fluid Power and Motion Control, FPMC 2015. https://doi.org/10.1115/FPMC2015-9513
[29] Yu Q, Cai M (2011) Research on expansion ratio of air power engine system. Proceedings of 2011 International Conference on Fluid Power and Mechatronics, FPM 2011 987–990. https://doi.org/10.1109/FPM.2011.6045906
[30] Gryboś D, Leszczyński J (2023) Exergy analysis of pressure reduction, back pressure and intermittent air supply configuration of utilization/expansion stage in compressed air systems. Energy 285:129419. https://doi.org/10.1016/j.energy.2023.129419
[31] Leszczyński JS, Gryboś D (2019) Compensation for the complexity and over-scaling in industrial pneumatic systems by the accumulation and reuse of exhaust air. Appl Energy 239:1130–1141. https://doi.org/10.1016/j.apenergy.2019.02.024
[32] Zhong W, Xu K, Li X, et al (2016) Determination of pressure drop for air flow through sintered metal porous media using a modified Ergun equation. Advanced Powder Technology 27:1134–1140. https://doi.org/10.1016/j.apt.2016.03.024
[33] Krichel S V., Sawodny O (2014) Non-linear friction modelling and simulation of long pneumatic transmission lines. Math Comput Model Dyn Syst 20:23–44. https://doi.org/10.1080/13873954.2013.811268