Double transmission double expansion technology as a method for reducing energy losses associated with oversizing of industrial compressed air systems

Dominik Gryboś
AGH University of Science and Technology
Faculty of Energy and Fuels
Department of Thermal and Fluid Flow Machines
Mickiewicza Ave. 30
30-059 Krakow
Poland
dgrybos@agh.edu.pl

Jacek S. Leszczyński
AGH University of Science and Technology
Faculty of Energy and Fuels
Department of Thermal and Fluid Flow Machines
Mickiewicza Ave. 30
30-059 Krakow
Poland
jale@agh.edu.pl

Keywords

industrial energy saving, compressed air, alternative energy systems (AES)

Abstract

Compressed Air Systems (CAS) are successfully used in industrial processes and production lines in almost all sectors due to low investment costs, durable construction, high power density, etc. The biggest disadvantages of these systems are high operational costs due to the very low energy efficiency (10–20 %). Such large energy losses in these systems significantly affect the economic aspects of the industrial sectors and the natural environment. Energy losses in compressed air systems can be differentiated by their place of occurrence, i.e. in the compressor, in pneumatic lines and in pneumatic machines. Compressed air energy losses in pneumatic machines account for approximately 20–30 % of all losses in CAS. They result from oversizing of those machines and process parameters. Such losses lead to significantly higher energy consumption in CAS compared to the actual energy demand for a given production task. There are several solutions to improve efficiency in final pneumatic machines, but neither have been implemented on an industrial scale. In this work we would like to introduce the Double Transmission Double Expansion (DTDE) technology. This technology is based on the accumulation of air exhausted from the pneumatic machine. Then accumulated exhaust air is used in another pneumatic machine and, for example, converted into electricity. We present results from a lab-scale demonstration unit consisting of a 1 kW prototype of this technology connected to a pneumatic machine. The paper also presents achievable energy benefits from the usage of this technology.

Introduction

The European Union in the climate and energy policy until 2030 not only assumes to decarbonise the energy sector, but also draws attention to increasing the energy efficiency of machines and industrial processes. The target of the EU energy and climate plan is to increase energy efficiency by 37.5 % compared to the level from 1990 (European Commission 2013). The Compressed Air Systems (CAS) in industry sector constitute one area where efficiency improvements are necessary in order to achieve this target. It is worth noting that the overall energy efficiency of CAS is only 10–20 % (Zhang et al. 2013; Saidur et al. 2010). Moreover, the CAS accounts for around 10 % of annual electricity consumption in the EU, which shows the popularity of those systems. (Saidur et al. 2010). Therefore, it is important to pay attention to the CAS and show that improving them will not decrease production capacity, but rather will save significant amounts of energy (Andersson et al. 2018; Zhang et al. 2013; Wang 2014; Nehler et al. 2018).

There is no single method to improve the efficiency of the CAS due to the high complexity and high energy losses (Saidur et al. 2010; Nehler et al. 2018). In the CAS energy losses can be grouped into three main categories, classified by location: losses in compressors and air preparation devices, losses in pneumatic lines and losses in air consumption elements, mostly actuators. While, the energy efficiency improving methods could be narrowed down to: prevent energy losses, reduces energy input and recovery energy waste (Harris et al. 2014; Hepke and Weber 2013).

The largest energy losses in the CAS arise in the process of air compression. It is estimated that average up to 20–50 % of electricity is lost in the compressor (Krichel et al. 2012; Saidur et al. 2010). To decrease these energy losses heat recovery systems are used, which are described in the literature very well (Broniszewski and Werle 2018; Saidur et al. 2010). Furthermore, the compressor can be optimized in terms of operation (Hu et al. 2017; Wang 2014; Zhang 2013) and size (Marshal 2011; Zhang 2013) in order to achieve the best efficiency point. Next method ensuring efficient operation of the CAS is periodic technical inspection of pneumatic installation. Particularly in search of air leakages and pressure drops in pneumatic lines (Saidur et al. 2010). Negligence may result in higher maintenance costs of the CAS. Another energy savings solution is presented by Sambandam et al. (2017) and concerns geometry modification of pneumatic pipes joints to limit pressure drops in networks. Regarding losses in pneumatic machines, despite numerous publications and methods to improve their efficiency, knowledge on this subject is not yet systematized (Du et al. 2018; Harris et al. 2012; Harris et al. 2014; Hepke and Weber 2013). There are many unknowns about the mechanisms of losses in actuators and methods of their elimination. Moreover, the knowledge about impact of these methods on the normal operation of pneumatic machine is also poorly disseminated. The most important factor influencing the efficiency of pneumatic machines is the oversizing of pneumatic actuators used for their construction (Leszczynski and Grybos 2019; Nehler et al. 2018). Due to the over-consumption of compressed air, up to 40-60% of compressed air energy can be lost in pneumatic actuators (Leszczynski and Grybos 2019).

In this article, we pay attention to losses occurring in pneumatic actuators. The most relevant losses is over consumption of compressed air associated with oversizing of pneumatic actuator. We present a comparison of pneumatic actuator in the CAS in four variants: conventional, closed system, dual pressure and double transmission double expansion (DTDE). Oversized, air consumption savings and air energy savings factors are used for the assessment of pneumatic actuator operational parameters. Moreover, in order to assess energy benefits in dual pressure and DTDE variants we introduced compressor electricity savings factor. We also present our own implementation of the DTDE called EH unit (Leszczynski and Grybos 2019) and a proposal for its adaptation on an industrial scale.

Theoretical background

Figure 1 specifies energy losses classification in a pneumatic actuator. The two main energy losses in actuators come from its oversizing and control algorithm (Harris et. al 2011). Less significant energy losses occur as a result of dead volumes in the installation. Negligible, in relation to the three previously mentioned, are energy losses of friction and air leakage of actuator seals.

In the conventional CAS, the pneumatic actuator uses only air transmission energy to perform work, while the air expansion energy is lost outside the actuator (Cai et al. 2006; Du et al. 2018; Yu et al. 2015). These losses result from zero-one actuator control system. It depends on switching positions of directional control valves in order to alternately supply and vent chambers in actuator. In literature, Harris et al. (2014) and Doll et al. (2011) present that apply the system of directional control valves bridge and modify switching algorithms result in better utilize both air transmission and expansion energy in the actuator. The application of this method can achieve up to 85 % of potential air energy savings (Doll et al. 2011). The drawback of this method is significant impact on control and automation system in pneumatic machines. The next energy losses in actuator is over consumption of compressed air which results from its oversized. The actuator uses much more compressed air energy than it needs to perform the production task. We describe the oversizing effect of pneumatic actuator in the next section. It is also worth to mention that some methods of reducing oversizing further reduce the effect of dead volumes in the CAS.

PNEUMATIC ACTUATOR OVERSIZE

The effect of the oversizing of pneumatic actuator is a complex phenomenon and stems from following parameters: piston diameter, inlet/outlet diameter, stroke, supply pressure, back pressure, load weight and stroke time (Leszczynski and Grybos 2019). Figure 2 shows pressure profiles in supplied and venting chambers of pneumatic actuator. The Δp parameter is a pressure difference between supple pressure p_A and ambient pressure p_0 ($\Delta p=p_A-p_0$). While the pneumatic force F is proportional to the pressure difference between both sides of the piston. If the force F needed to move a certain mass of the load with assumed average velocity v by the actuator is smaller than that resulting from the differential pressure Δp, then the actuator will create a correspondingly smaller pressure difference Δp^*. It turns out that the pneumatic force F adapts to load condition instead of static supply conditions (pneumatic force F is proportional to Δp^*). A visible result of this phenomenon will be higher speed of actuator during the stroke. It is worth noting that design average velocity v of actuator is in range 0.2–0.5 m/s. For this velocity range actuator achieves the best efficiency point (Fleischer 1995). Whereas the consumption of compressed air is not sensitive to load conditions, because each stroke the actuator finishes inflating the supplied chamber of stroke volume to the supply pressure p_A (process 3–4). Therefore, the energy consumption of compressed air by the actuator is constant. A certain static determinant of oversizing of the actuator can be defined as a difference of Δp-Δp^*. However, the phenomenon is much more complex and is based on the dynamics of the actuator movement. A fairly good indicator of the assessment of oversizing is presented by Doll et al.(2015) and it is called the Pneumatic Frequency Ratio (PFR) Ω. It is defined as:

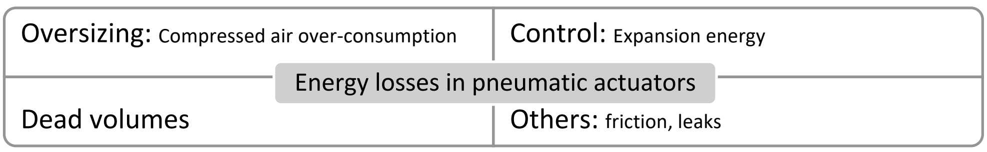

Figure 1. The classification of energy losses in pneumatic actuators.

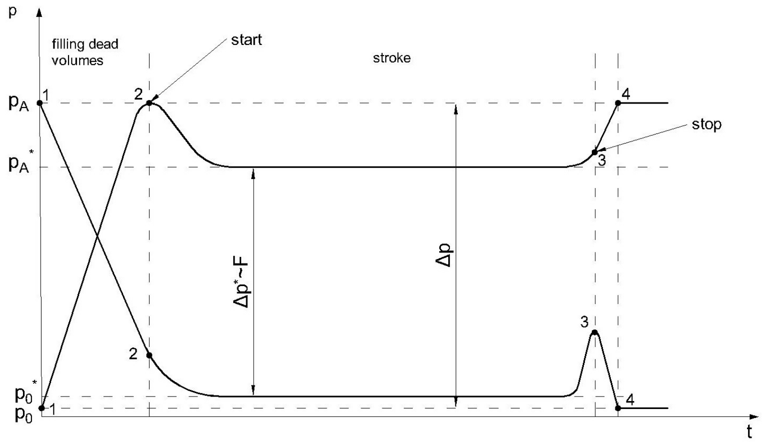

Figure 2. Pressure profiles in the supplied and venting chamber of the pneumatic actuator during the stroke. 1 – valve switch, 2 – piston start, 3 – piston stop, 4 – inflating the supplied chamber.

$$\Omega = \frac{t}{\pi}\sqrt{\frac{A\Delta p}{ms}} \tag{1}$$

where:

t transition time,

A piston cross section,

Δp pressure difference between supply pressure and back pressure,

m mass load and s stroke.

When parameter Ω is higher than 2.2 the actuator is significant oversized, from 1.7 to 2.2 is slightly oversized and for 1.1 to 1.7 is well-dimensioned (Doll et al. 2015). However, given by Doll et al. (2015) the PFR Ω ranges are only theoretical. For actuators operating in the industrial CAS, the optimal PFR Ω value is 1.6–2.0. The PFR Ω is used in the paper to evaluate oversizing of the actuator for various CAS variants.

The compensation of the oversizing effect can be achieved by adjusting the supply conditions, i.e. pressure and air flow (Leszczynski and Grybos 2019; Harris et al. 2014; Cummins et. al 2017), or by optimizing the construction parameters of the pneumatic actuators (Hepke and Weber 2013; Du at al. 2018; Yu at el. 2015; Zhang et al. 2013). The former method consists of matching the pressure difference Δp^* prevailing in the actuator's dynamics with the pressure difference Δp resulting from the supply conditions.

Figure 3 shows CAS in 4 variants: conventional, closed system, dual pressure and double transmission double expansion (DTDE). Each CAS consists of pneumatic machines represented by squares shown in Figure 3. In the conventional CAS pneumatic machines consists of actuators (1), directional control valves (2) and silencers (3) (Figure 3a). Compressed air is supplied to pneumatic actuators in the machines from the compressor and then expands by pneumatic silencers to the ambient. The method presented in Figure 3b is called closed system. This method involves returning exhausted air from pneumatic machine back to compressor instead of spreading it locally by the pneumatic silencers. Figure 3c shows the variant of the CAS called dual pressure which reduces supply pressure in pneumatic actuators. By using two additional pressure regulators (4) for each actuator in each machine, the supply pressure can be individually adapted to the extend and retract motion of the actuator (Harris et al. 2014). This method is widely described in Fleischer (1995). The last method (Figure 3d) is called double transmission double expansion (DTDE) and this term was introduced by Leszczynski and Grybos (2019). It consists of staging the work of compressed air in the CAS. In the first stage, the air expands from supply pressure p_A to back pressure p_B, and in the second stage from back pressure p_B to ambient pressure p_0. The DTDE system divides the CAS into high-pressure and low-pressure subsystems. This variant increases complexity of the CAS only by tank (5), directional control valve (6) and actuator (7). The DTDE variant can be used both for a single actuator in a pneumatic machine or for groups of actuators in several pneumatic machines as shown in Figure 3d. Authors present their own solution of the DTDE in the form of a device called energy harvester (EH) unit, which converts the waste energy of compressed air into electricity (Leszczynski and Grybos 2019). Other methods of adapting the DTDE technology are presented for example by Cummins et al. (2017), Hepke and Weber (2013) or Luo et al. (2013). All of them take into account only a single actuator in the DTDE variant, which significantly increases the complexity of the complete CAS.

MATHEMATICAL MODEL AND COMPUTER SIMULATION

To compare these four CAS variants, a dynamic model of the actuator was created. The basis of the model is a system of three equations (2): the equation of mass conservation in both supplied m_1 and venting m_2 actuator's chambers and the piston equation of motion. The model was described and validated in our previous paper (Leszczynski and Grybos 2019). The actuator force system consists of pneumatic forces (F_1, F_2), gravity (F_g) and Columb-Viscous friction (F_r). The model below assumes no heat transfer, no leaks and air as ideal gas. (Yu i in., 2015; Beater, 2007).

$$\begin{cases} \dfrac{dm_1}{dt} = \dot{m}_1 \\ \dfrac{dm_2}{dt} = \dot{m}_2 \\ m\dfrac{d^2x}{dt^2} = F_1 + F_2 + F_g + F_{fr} \end{cases} \tag{2}$$

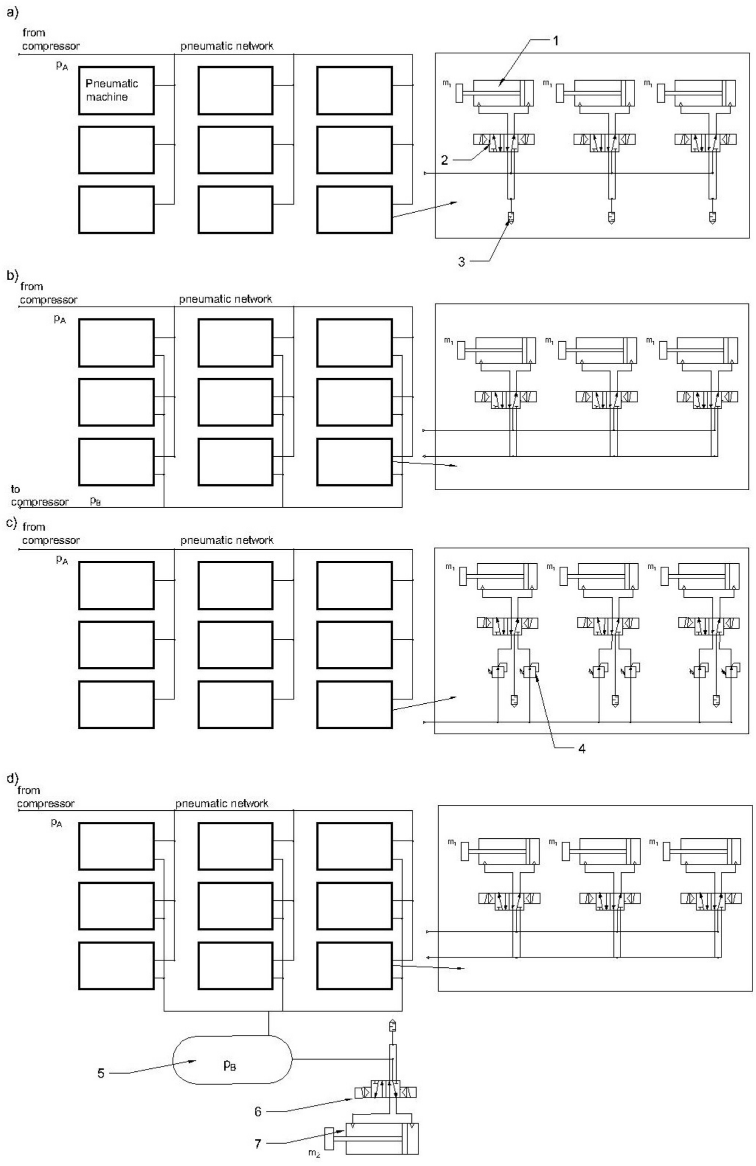

Figure 3. Pneumatic machines in the CAS: (a) Conventional circuit; (b) Closed cycle circuit; (c) Dual pressure circuit; (d) Double transmission double expansion circuit.

The model is supplemented with a modified equation of mass outflow depending on the pressure ratio. The introduced formula with special scaling function modifies the flow characteristics depending on real conditions, see (Leszczynski and Grybos 2019).

$$\begin{cases} A\sqrt{\dfrac{2\kappa}{\kappa-1}\rho p_1\left(\left(\dfrac{p_2}{Zp_1}\right)^{\frac{2}{\kappa}}-\left(\dfrac{p_2}{Zp_1}\right)^{\frac{\kappa+1}{\kappa}}\right)} & \text{for } \dfrac{p_2}{p_1}>\beta \\ A\sqrt{\kappa\rho p_1\left(\dfrac{2}{\kappa+1}\right)^{\frac{\kappa+1}{\kappa-1}}} & \text{otherwise} \end{cases} \tag{3}$$

Where, critical pressure ratio:

$$\beta = \zeta\omega\left(\frac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}} \tag{4}$$

Scaling factor:

$$Z = \frac{1-\zeta}{1-\beta^{2\zeta}}\left(\frac{P_A}{p_B}\right)^{2\zeta} + \frac{\zeta-\beta^{2\zeta}}{1-\beta^{2\zeta}} \tag{5}$$

In order to evaluate and compare methods for reducing the oversizing of pneumatic actuators, the compressed air savings indicator α and air energy savings indicator γ is used. The compressed air savings α of compressed air is the ratio of the volume flow of the compressed air consumed in a given method $\dot{V}_0^*$ to the volume flow in the conventional CAS $\dot{V}_0$.

$$\alpha = \frac{\dot{V}_0^*}{\dot{V}_0} \tag{6}$$

The air energy savings indicator γ shows energy savings in given variant of CAS in relation to the energy consumption in the conventional CAS. The mathematical formula is different for each of CAS variants. To determine these values, the definition of the pneumatic work W consisting of transmission W_T and expansion W_E part is used (Cai et al. 2006):

$$W = p_A V_A \ln\frac{p_A}{P_0} = W_T + W_E \quad (7)$$
$$= p_A V_A \left(1 - \frac{p_0}{P_A}\right) + p_A V_A \left(\ln\frac{p_A}{P_0} + \frac{p_0}{P_A} - 1\right)$$

Theoretical pressure-volume charts for all CAS variants are shown in Figure 4. On them the basis the following formulas are derived. For the closed system variant air energy savings γ is defined as the ratio of the pneumatic work W of cycle 1–1*–4*–4 shown in Figure 4b to the pneumatic work W of conventional CAS cycle 1–2–3–4 shown in Figure 4a.

$$\gamma_{CC} = \frac{W_{1-1*-4*-4}}{W_{1-2-3-4}} = 1 - \frac{\ln\frac{p_B}{p_0}}{\ln\frac{p_A}{p_0}} \quad (8)$$

For the dual pressure variant, air energy savings γ is defined as the ratio of the pneumatic work W of cycle 2*–2–3–4–3* shown in Figure 4c to the pneumatic work W of conventional CAS cycle 1–2–3–4 shown in Figure 4a.

$$\gamma_{DP} = \frac{W_{2*-2-3-4-3*}}{W_{1-2-3-4}} = 1 - \frac{p_A^* \ln\frac{p_A^*}{p_0}}{p_A \ln\frac{p_A}{p_0}} \quad (9)$$

For the DTDE variant, air energy savings γ is defined as the ratio of the transmission pneumatic work W_T of cycle 5–6–7–8 shown in Figure 4d to the pneumatic W of conventional CAS cycle 1–2–3–4 shown in Figure 4a.

$$\gamma_{DTDE} = \frac{W_{T_{5-6-7-8}}}{W_{1-2-3-4}} = \frac{1 - \frac{p_0}{p_B}}{\ln\frac{p_A}{p_0}} \quad (10)$$

where
p_A supplied pressure,
p_A* reducted supplied pressure,
p_0 ambient pressure,
p_B backpressure,
V_A actuator total volume,
V_D actuator dead volume.

Simulation and Experimental setup

COMPUTER SIMULATION

Computer simulations were made in the Matlab R2017 program using the Runge-Kutta-Fehlberg solver. Table 1 presents the parameters of the simulation of pneumatic actuator movement in four variants: conventional CAS, closed system, dual pressure and DTDE. In order to visualize the impact of individual modifications, a single actuator was simulated. The pneumatic actuator with 50 mm piston diameter and 14 mm piston rods diameter was used. The pneumatic actuator's task was to lift the weight of 60 kg to a height of 0.6 m, with a minimum average speed of 0.5 m/s. Then, in retract motion fall down without load the expected return time is 0.37 s.

TEST RIG

One of the possibilities of implementing the DTDE system is presented in Grybos and Leszczynski (2019). The idea of the device called the Energy Harvester (EH) unit is the conversion low-pressure air energy of the CAS second stage into electricity. The EH unit is used as a separate additional device for pneumatic machines that is attached to the outlets of an exhausted air. Therefore, it is very beneficial for existing pneumatic machines, in which it is not possible to modify their internal structure or control algorithm. The electricity produced can assist to air compressor reducing electricity consumption from the

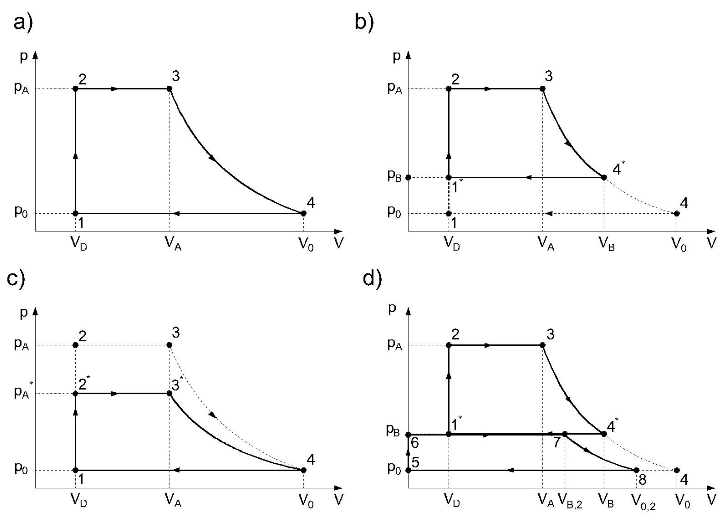

Figure 4. Theoretical pressure-volume cycles of pneumatic actuator: (a) Conventional circuit; (b) Closed cycle circuit; (c) Dual pressure circuit; (d) Double transmission double expansion circuit.

Table 1. Parameters of the simulation of pneumatic actuator movement in four circuits variant: conventional CAS, closed system, dual pressure and DTDE.

Parameter	Conventional CAS	Closed system	Dual Pressure	DTDE
Piston diameter (mm)	50			
Bore diameter (mm)	14			
Inlet/outlet diameter (mm)	9			
Stroke (m)	0.6			
Load (kg)	60			
Supply gauge pressure p_A (bar)	6.3	6.3	4.8/5.7	6.3
Back gauge pressure p_0 (bar)	0	1.75	0	1.75

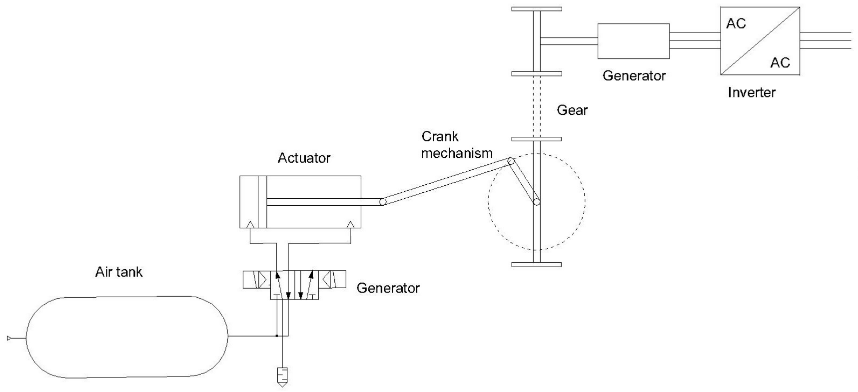

Figure 5. Schematic of proposed Energy Harvester unit.

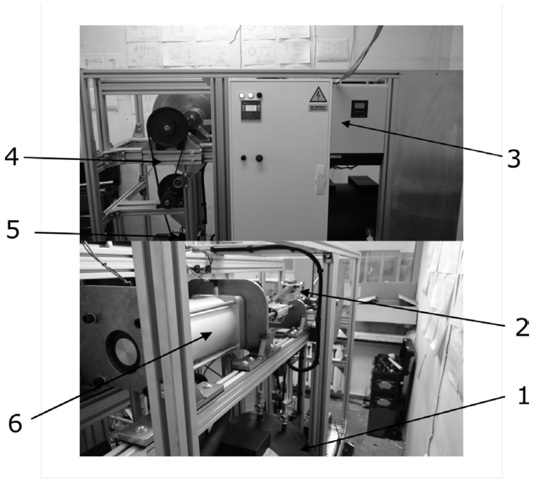

Figure 6. The EH unit prototype as an implementation of the DTDE CAS concept: air tank (1), crank mechanism (2), Inverter (3), gear (4), generator (5), actuator (6).

power grid. The schematic of current version of the EH unit is shown in Figure 5. Own construction of the EH unit prototype is shown in Figure 6. This version of the prototype is significantly modified as compared to the version presented in our previous work (Leszczynski and Grybos 2019).

The internal construction of the mechanical power transmission was changed on a crankshaft slider (2) with a two-stage belt transmission (4). The system consists of a 200 litre tank (1) and actuator (6) with 200 mm piston diameter and 300 mm stroke. The device has the ability to work in on-grid mode, i.e. feeding electricity into the power grid. A permanent-magnet generator (5) with a capacity of 1 kW and a rotational speed of 650 rpm was installed in the system. In addition, an on-grid inverter (3) was also installed in the system.

The measuring system was equipped with a computer and National Instrument USB 6211 measuring card. The voltage and current measurements were carried out using LEM LV25 and LA50 transducers, respectively. The gauge pressure measurements in the tank and in the actuator chambers were made with an Introbar 20 pressure transducer. The volume flow was calculated on the basis of pressure measurement in the actuator's chambers.

Results

The aim of simulations is to compare thermodynamic efficiency of the four CAS variants described in section 2: the conventional circuit, the closed system, the dual pressure and the DTDE. The purpose of the simulations was to minimize the piston velocity to assumed 0.55 m/s for extend and 1.60 m/s for retract movement. It is worth noting that typical average actuator velocity in its operation point should be between 0.2 and 0.5 m/s. The actuator achieves maximum efficiency for this velocity range. We additionally assumed the safety factor of the actuator in the form of increased velocity by 0.05 m/s. It follows that the duration of the extend and retract movement will be 1.1 s and 0.37 s, respectively. For the thermodynamic comparison of these methods, constant actuator movement parameters for the closed system, the dual pressure and the DTDE variants were assumed. Figures 7 and 8 show the actuator movement parameters for the extend and retract movements, respectively. Figure 7 and 8 show the pressure change in the actuators chambers p, displacement x and piston velocity v, and air mass flow $\dot{m}_1$. It is worth noting that for the closed cycle and the DTDE variants, the piston velocity profile in extend stroke is the smoothest and the maximum velocity is the lowest (Figure 7c). It is directly translated to the extended lifetime of pneumatic actuator (Seong-woo 2018). In the conventional CAS variant, the actuator made the extend stroke in 0.8 s while the retract in 0.34 s. Compared to the designed extend movement times, the actuator performs its tasks much faster than assumed in the machine control system by designer. It follows that the actuator in the conventional CAS is oversized, which was confirmed by the determined PFR Ω indicator, which is 2.3 (Table 2). This value indicates the oversized actuator. The closed system and the DTDE variant of the CAS have identical effect on the dynamics of the actuator movement because they introduce the same back pressure in venting chamber. The difference between these variants results from the utilizing of exhausted air from the actuator. In the close system variant, the exhausted air is returned to the compressor. In the DTDE system, the exhausted air is directed to the next actuator in EH unit where it performs further work.

The purpose of modifying the compressed air system is to reduce the effect of the oversizing, while maintaining designed extend and retract movement times of 1.1 s and 0.37 s, respectively. Comparison of oversizing phenomenon of pneumatic actuator in the CAS in conventional, closed cycle, dual pressure and DTDE variants, shown in Table 2. All variants have been able to reduce the effect of oversizing of the actuator, see PFR factor in Table 2. However, the DTDE and the close cycle variants reduced oversizing (Ω=1.92) to a greater extent than the dual pressure variant (Ω=1.97) in condition of constant movements parameters of actuator. The air energy saving indicator γ for the closed cycle, the dual pressure and the DTDE variants are 0.51, 0.21 and 0.34 respectively. It is important that the air energy savings γ for the closed cycle and the dual pressure variants result from the saving of compressed air consumption α. This directly translates into reduced electrical energy consumption by the compressor. However, for the DTDE variant, only part of the air energy saving γ is associated with a reduction in the consumption of compressed air α. It results from the compensation of the phenomenon of dead volumes in the system. This effect strongly depends on the size of these volumes in the system. In the given example, the dead volumes amounted to 10 % of the total volume of the actuator which directly resulted in savings in compressed air consumption α of 0.04. On the other hand, the remaining part of energy savings is related to its subsequent effective use in another actuator.

Comparing all the variants, it comes that the most air energy savings variant turned out to be the closed system (γ=0.51), next the DTDE (γ=0.32) and finally, the dual pressure (average γ=0.21). Although the greatest air energy savings γ are obtained in closed system variant, its implementation in the existing CAS is impractical due to the total cost and size of the investment. This would involve a complete reconstruction of the CAS in manufacture plant which would mean downtime during the investment. Finally, it would turn out that it is not technically possible to translate the entire CAS in existing manufacture plant into the closed system. This would force to maintain two separate CASs: conventional and closed system. Therefore, for the existing CAS, the DTDE variant is better retrofittable method, despite the lower energy savings. In particular, the EH unit as the DTDE variant can be easy installed in the form of overlay at any pneumatic machines. This gives the opportunity to gradually adapt the existing CAS.

Finally, we would like to present our adaptation of DTDE variant in CAS (Leszczynski and Grybos 2019). For this purpose we use our own device called the Energy Harvester (EH) unit described in Section 3 and shown in Figure 6. Figure 9 shows experimental result of output electrical power and compressed air volume flow during time by EH unit with on grid mode. The device operated in on-grid mode with an average power of 400 W and a maximum power of 930 W. During the operation of the EH unit, average air volume flow 66 Nm^3/h was supplied at gauge pressure 1.75 bar. The EH unit energy conversion (from air energy to electricity) efficiency is 0.33. In subsequent development works of the EH technology, the possibility of achieving the efficiency of this device of the order of 0.45–0.6 is visible. Improvements are planned to the mechanical and the control system. Interestingly, the implementation of this device is able to remove noise and oil moist pollution resulting from the expansion of compressed air in pneumatic silencer. Due to remove compressed air expansion outside of the work area in manufactured plant. Preliminary noise tests show a reduction of sound pressure in the manufactured plant from 88 dB to 70 dB which translates into eighteen-fold noise reduction.

The profit assessment of the EH unit (the DTDE variant) and the pressure regulator (the dual pressure variant) implemented in an industrial plant is a more complex issue than the thermodynamic and theoretical considerations presented above. The CAS is topological and operational complex systems due to the large number of elements that operate more or less randomly. From the energy point of view, the EH unit

Table 2. Comparison of oversizing phenomenon of pneumatic actuator in CAS in conventional, closed cycle, dual pressure and DTDE variants.

CAS circuit	Stroke time t (s)		Average velocity u (m/s)		PFR Ω (–)	Air consumption savings		Air energy savings γ (-)	
	Extend	Retract	Extend	Retract		Extend	Retract	Extend	Retract
Conventional	0.8	0.34	0.75	1.76	2.30	0	0	0	0
Closed cycle	1.10	0.37	0.55	1.60	1.92	0.51	0.51	0.51	0.51
Dual Pressure	1.10	0.37	0.55	1.60	1.97	0.29	0.12	0.29	0.12
DTDE	1.10	0.37	0.55	1.60	1.92	0.04	0.04	0.32	0. 32

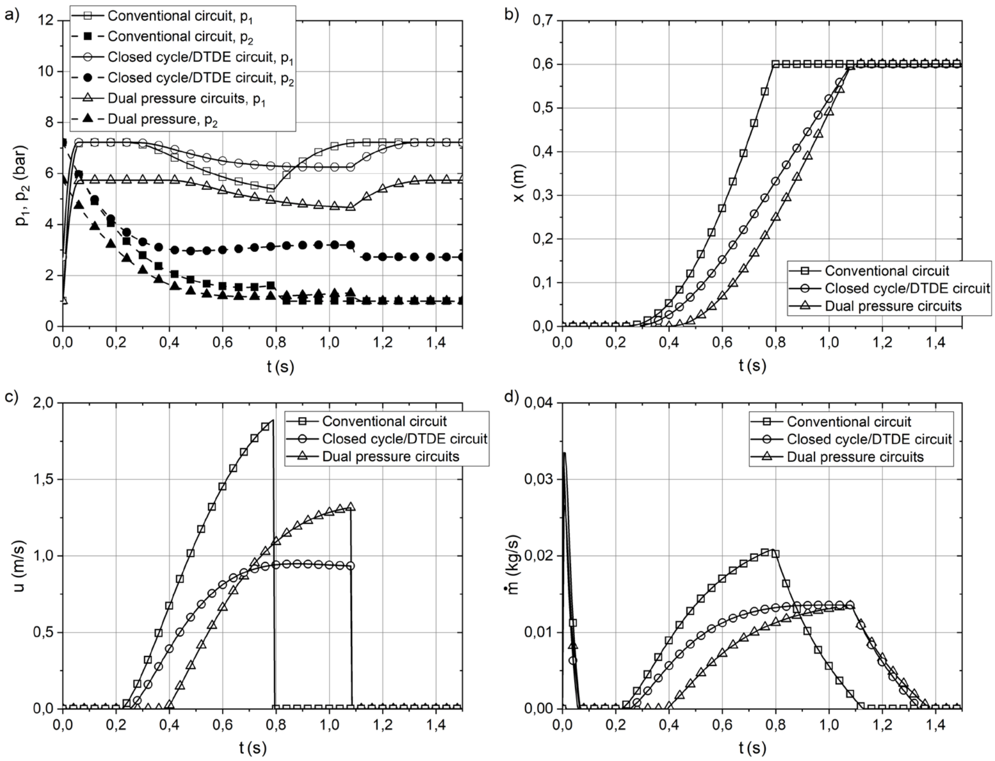

Figure 7. The computer simulation of the pneumatic actuator in extend motion for four CAS variants: (a) Pressure in actuator's chambers; (b) Piston displacement; (c) Piston velocity; (d) Compressed air mass flow.

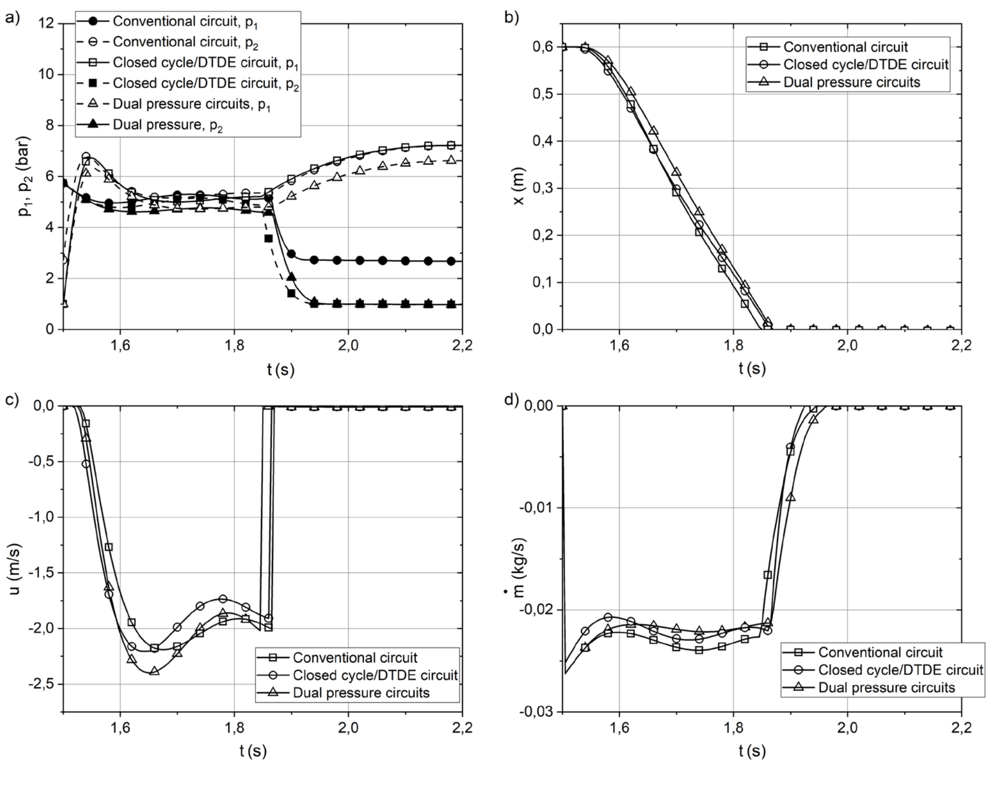

Figure 8. The computer simulation of the pneumatic actuator in retract motion for four CAS variants: (a) Pressure in actuator's chambers; (b) Piston displacement; (c) Piston velocity; (d) Compressed air mass flow.

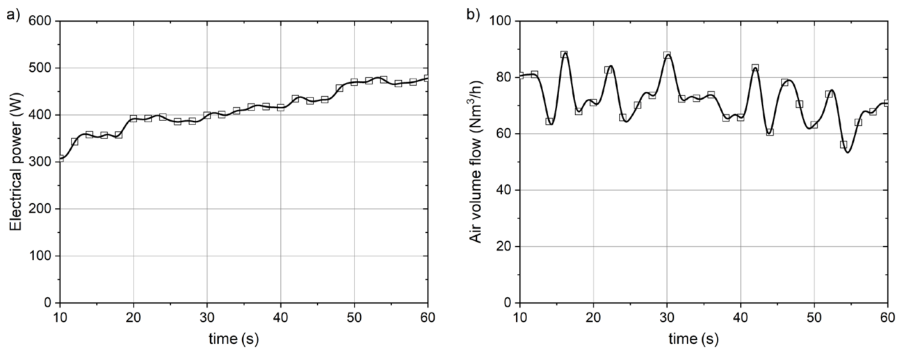

Figure 9. Experimental measurement results of EH unit in laboratory conditions: (a) Electrical power output; (b) Compressed air volume flow.

cannot be directly compared with pressure regulators due to different energy benefits. In the case of the EH unit, this benefit is both reduction of compressed air energy consumption and additional produced electricity. In the case of pressure regulators it only means reduction of compressed air energy consumption. In order to compare these two systems, all energy benefits in these variants should be transferred to the level of electricity consumed by the compressor. In the case of the EH unit (DTDE variant) it was assumed that all produced electricity is supplied to the compressor. Moreover, additional assumptions were made:

- the entire CAS system in an industrial plant is in the DTDE variant with the EH unit or dual pressure variant;
- installed air compressor is fixed speed and works with maximum power during 80 % of time and 30 % of maximum power during 20 % of time with efficiency 0.75 (Mousavi et al. 2014);
- internal efficiency of pressure regulator is 0.84 (Huang et al. 2017; Seslija et al. 2017).

The compressor electricity savings (CES) indicator presented below should be considered as demonstrative. It means how much less electricity the compressor uses from the power grid.

$$CES = \frac{0.7\eta_r\alpha + (\gamma - \alpha)(1 - \alpha)\eta_{EH}\eta_{comp}}{0.7 + \frac{0.3}{\tau}} \tag{11}$$

where,

η_i pressure regulator internal efficiency,
α compressed air savings,
γ air energy savings,
η_{EH} EH unit efficiency,
η_{comp} compressor efficiency,
τ ratio of compressor run time with maximum power to total time (air compressor duty time).

The *CES* indicator for the DTDE variant with presented above the EH unit is 0.10. However, if we assume the efficiency of the EH unit expected in developed technology (η_{EH}=0.60), the *CES* will be 0.16. In the case of the dual pressure variant with pressure regulators, the *CES* indicator is 0.11. The assumed efficiency of pressure regulators 0.84 is the results of air leakage stated by the manufacturer and air expansion energy loss (Huang et al. 2017). Unfortunately, one of the disadvantages of these devices is unsealing during operation time, therefore the actual efficiency will be even less than 0.84. So, the estimated *CES* ratio range for the pressure regulator is between 0.09 and 0.12. It should be noted that the *CES* indicator is estimated for systems parameters described and assumed above. The CES indicator is closely related to back pressure level and dead volumes parameters for the DTDE variant, supplied pressure reduction parameter for dual pressure variant and type of air compressor in the CAS. In extreme cases, the CES values for the dual pressure and the DTDE with the EH unit can even be 0.26 and 0.28, respectively. In addition, non-energy benefits must be considered and in the case of the DTDE it is reduced noise, oil mist contamination and extended actuator life compared to the dual pressure variant. Moreover implementation of the DTDE technology seems to be easily and less time consuming than dual pressure variants. Analysing Figure 3d we can see that in the DTDE variant one EH unit is assembled on the whole CAS consists of 9 machines. Then, the one parameter of back pressure is set. In the case of dual pressure (Figure 3c), two pressure regulators should be fitted at each actuator. In the presented CAS we have 27 actuators, which gives 54 regulators. Moreover, each pressure regulator should be individually adjusted, which takes much more time than adjusting the DTDE variant. So, the DTDE variant with EH unit represent more holistic approach than dual pressure.

Conclusion

The paper presents the phenomenon of oversizing of pneumatic actuators and their impact on their efficiency. Conventional, closed system, dual pressure and double transmission double expansion (DTDE) CAS are compared in thermodynamic analysis. In closed system variant air energy saving in actuator is 0.51. However, its implementation in the existing CAS is very expensive. The DTDE and dual pressure variants are more adaptable in existing CAS and achieved average energy saving indicator γ 0.32 and 0.21, respectively. The paper also presents an example application of the DTDE called Energy Harvester (EH) unit. In order to compare energy benefits for the EH unit and pressure regulators the compressor electricity savings (CES) indicator was introduced. The estimated CES indicator range for presented pressure regulator and the DTDE variant with the EH unit is between 0.09 and 0.12, and 0.10 and 0.16, respectively. In addition, there were non-energy benefits in favour of the EH unit like reducing noise, reducing oil mist contamination and extended actuator life. Moreover, the DTDE variant with EH unit gives more holistic approach than dual pressure in energy optimisation of the CAS. It means that the EH unit is less invasive methods in an existing pneumatic machine than dual pressure variant.

Nomenclature

UPPERCASE LETTERS

$\dot{V}$	Volume flow, m³/s
A	Cross section, m²
F	Force, N
P	Power, W
V	Volume, m³
W	Work, J
Z	Scaling function, –

LOWERCASE LETTERS

$\dot{m}$	Mass flow, kg/s
m	Mass, kg
p	Pressure, Pa
s	Stroke, m
t	Time, s
u	Velocity, m
x	Displacement, m

GREEK SYMBOLS

Δ	Interval, –
Ω	Pneumatic Frequency Ratio, –
α	Air consumption savings, –
β	Critical pressure ratio, –
γ	Air energy savings, –
η	Efficiency, –
κ	Adiabatic exponent, –
ρ	Density, kg
τ	Air compressor duty cycle, –
ζ	Scaling factor, –

SUBSCRIPTS

0	Ambient
A	Supplied
B	Backpressure
D	Dead
E	Expansion
EH	Energy Harvester unit
T	Transmission
Comp	Compressor
fr	Friction
g	Gravity
i	Index, i=...0,1,2
r	Pressure Regulator

CONSTANT

$\kappa = 1.4$	Adiabatic exponent for air, –
$g = 9.81$	Acceleration due to gravity force, m/s²
$p_0 = 101235$	Ambient pressure, Pa
$R_a = 287.1$	Individual gas constant for air, J/(kgK)

ABBREVIATIONS

CAS	Compressed Air System
CES	Compressor Electricity Savings
DTDE	Double Transmission Double Expansion
EH	Energy Harvester
PFR	Pneumatic Frequency Ratio

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Acknowledgements

The work was carried out as part of a research subvention under contact no. 16.16.210.476 supported by the Polish Ministry of Science and Higher Education.