ENERGY EFFICIENCY ANALYSIS AND EXPERIMENTAL TEST OF A CLOSED-CIRCUIT PNEUMATIC SYSTEM

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ABSTRACT

In a closed-circuit pneumatic system the air from the pneumatic cylinder is not exhausted directly in the atmosphere, as done in traditional open-circuit systems, but is captured and fed to the compressor inlet. This provides an opportunity to increase the compressor inlet pressure above the atmospheric and consequently enables both higher compression efficiency and volumetric flow rate. Until the present, the advantages of the closed-circuit pneumatics were assessed only within theoretical studies without being verified experimentally due to uncertainties in performance of a real compressor under oscillating pressure and flow rate, as well as due to a risk of mutual cylinder interference caused by differences in load and required pressure profiles. Hence, this study focuses on experimental investigation of energy efficiency and practicability issues of the closed-circuit pneumatics. It is shown that simultaneous operation of two pneumatic cylinders of different size (Ø32×200 and Ø50×200) performing different tasks (high-dynamic mass handling and extension against the constant force) does not have any negative effect neither on the cylinder dynamics nor on the compressor performance. Inlet pressure increase up to 1.5 bar_{rel} leads to by factor 2.5 higher volumetric flow rate and total efficiency gain of 72 % (increase from 10.7 % to 18.4 %). The experimentally obtained results show a great potential of the closed-loop pneumatics and indicate the need in further research into design and control methods of such systems to enable technology deployment to the industry. Industrial applications can profit from the reduced energy consumption especially in case of pneumatic systems with decentralized air supply.
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INTRODUCTION

Pneumatics is commonly used in modern automation, robotic and nutrition sectors. It offers such advantages as low investment costs, high robustness, ease of use, and lightweight components. Alongside the Industry 4.0 transition, improving the energy efficiency of pneumatic systems has been one of the most challenging and important research fields in the pneumatics within the last decades, essential to keep this technology environmentally friendly and competitive on the global market.

The total efficiency of a pneumatic drive depends on the energy losses brought about by: i) energy conversion (prime mover, compressor, pneumatic actuator), ii) transportation (pressure drops throughout the air cooler, dryer, filter, pipeline system), and iii) power management (pressure and flow rate control, usually by throttling). An efficient design of a pneumatic system implies that all the components needed for energy conversion, transportation, and power management are selected and tuned for an operation with minimal energy losses. The contribution of energy saving measures feasible within each of the group i-iii) was evaluated and assessed within several studies such as [1] and [2]. However, a simultaneous implementation of all these energy saving measures in practice can appear as tedious and costly when retrofitting an already existing pneumatic system. Besides, pneumatic drives are often not the only consumers of compressed air delivered by the compressor plant. Therefore, an exact quantification of energy losses and total efficiency of one particular drive, counting from the plug of the compressor power unit up to the mechanical coupling on the cylinder shaft, is very challenging. The consequence is a common splitting of calculation of energy consumption into two parts: firstly, estimation of how efficiently the air is compressed, prepared, and transported, and secondly, how efficiently is it consumed. The first part provides information about the energy needed to compress a unit quantity of air to nominal pressure, to cool, to dry, and to transport it (usually referred as "specific energy consumption" or "compressed air index", measured in

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kWh×Nm^{-3}) [2]. The second part tells how much air is consumed by a pneumatic drive or a group of drives (measured in Nm^3 per time unit or per cycle) [3].

Once knowing the specific energy consumption, this approach provides an easy and transparent assessment of energy consumption by pneumatic drives without having to take into consideration other air consumers and having an exact knowledge about efficiency of each process involved in the air compression and preparation. On the other side, this reduction of complexity displaces the focus of the research in pneumatics to the question "how efficient is the air consumed", whilst often leaving out the question "how efficient the air is compressed and prepared". This can hinder a generation and investigation of new high-efficient methods and system structures, where the compressed air production and consumption are coherent and both questions are equally important. An example of such a system, considered within this paper, is a closed-circuit pneumatic system, where the exhausted air is fed back to the compressor inlet port and hence circulates in the circuit without leaving it.

THEORETICAL ADVANTAGES OF A CLOSED-CIRCUIT AIR COMPRESSION

Theoretical advantages of a closed-circuit pneumatic system rely on a simple observation that theoretical work W_{compr} needed for an adiabatic compression of air with mass $m = \rho_1 \times V_1$ from inlet pressure p_1 to outlet pressure p_2 depends on the relation of these pressures $\varepsilon = p_2 / p_1$, whereas the work W_{displ} performed by the displacement machine, such as a pneumatic cylinder acting against the constant force F_{load}, depends on the pressure difference in the cylinder chambers. Assuming a lossless air transportation from the compressor to the frictionless cylinder with an infinitely large stroke, this pressure difference equals to $\Delta p = p_2 - p_1$. For the sake of simplicity no losses are considered in this section.

$$W_{compr} = V_1 \times \rho_1 \times R \times T_1 \times \frac{\kappa}{\kappa - 1} \times \left[\left(\frac{p_2}{p_1} \right)^{\left(\frac{\kappa - 1}{\kappa} \right)} - 1 \right] \tag{1}$$

$$W_{displ} = V_2 \times (p_2 - p_1) = \frac{m \times R \times T_1}{p_2} \times (p_2 - p_1) \tag{2}$$

From Equations (1) and (2) can be obtained that an increase in p_1 at constant pressure difference Δp increases the relation of the theoretical cylinder displacement work (Equations (2)) to theoretical compression work (Equations (1)). This relation can be used as a measure of total efficiency $\eta = W_{displ} / W_{compr}$ of the system. Hence the idea of the closed-circuit pneumatics is to increase the η by raising the inlet compressor pressure over the atmospheric pressure while keeping constant the pressure difference Δp, relevant for the application.

Figure 1 illustrates these relations by comparison of p-V diagrams of a compression cycle from 1 bar$_{abs}$ to 5 bar$_{abs}$ with compression cycle of the same air mass from 3 bar$_{abs}$ to 7 bar$_{abs}$. Both scenarios provide the same differential pressure $\Delta p = 4$ bar allowing the cylinder to overcome the force F_{load}. The orange-colored area on the diagrams a) and b) corresponds to the work of adiabatic compression cycle composed of the compression work, displacement work and suction work (first, second and third terms in Equation (3) respectively), and is equal to the result of Equation (1).

$$W_{compr}^{ad} = \frac{p_1 \times V_1}{\kappa - 1} \times \left[\left(\frac{p_2}{p_1} \right)^{\left(\frac{\kappa - 1}{\kappa} \right)} - 1 \right] + p_2 \times V_2 - p_1 \times V_1 \tag{3}$$

Adiabatic compression is taken here as a simplification of a real compression process that can proceed as a polytropic (with partial heat flow from compressed gas to environment, calculated with polytropic exponent $n < \kappa$) or superadiabatic process (with heat flow from environment, calculated with polytropic exponent $n > \kappa$) [4]. From diagrams a) and b) in Figure 1 can be obtained that compression with $\varepsilon = 7/3$ (closed circuit) takes less than the half of the energy needed to compress the same air quantity with $\varepsilon = 5/1$ (open circuit).

After being compressed, the air needs to be cooled down to the ambient temperature and the absorbed heat is extracted from the system. The air volume shrinks from $V_{2.ad}$ to $V_{2.is}$ and takes the state equivalent to the result of an isothermal compression (Figure 1, c) and d)):

$$W_{compr}^{is} = m \times R \times T_1 \times \ln \left(\frac{p_2}{p_1} \right) \tag{4}$$

From the diagrams c) and d), Figure 1, is visible that the heat losses also decrease with the reduction of compression ratio. In the given case the heat loss in closed circuit is almost four times lower than that after compression from the atmospheric pressure.

Because the air volume V_2 at pressure p_2 = 7 bar_{abs} is smaller than at p_2 = 5 bar_{abs}, it can perform lower displacement work, as can be seen from Figure 1, e) and f). Nevertheless, when considering the relation $\eta = W_{displ}/W_{compr}^{ad}$ the closed-circuit system is about 50 % more efficient then the open one.

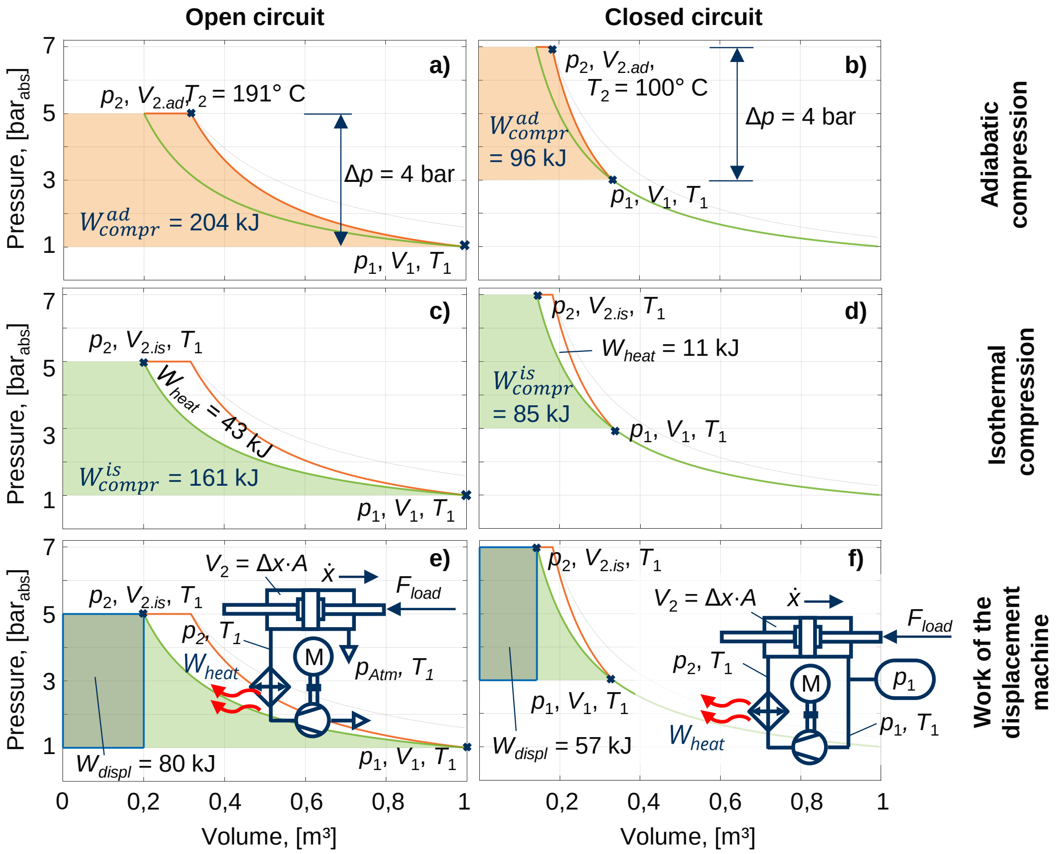

Figure 1 – p-V-diagrams for compression from atmospheric inlet pressure (open circuit, left column) and from inlet pressure of p_1 = 3 bar_{abs} (closed circuit, right column)

Furthermore, the increase in inlet pressure results in higher compressor delivery rate. In case of a real compressor, the geometrical suction volume V_1 is always constant. When referring to the example on diagrams b) and d) in Figure 1, the starting point of compression would always have an abscisse of V_1 = 1 m³ regardless of the inlet pressure. Hence, higher air mass is inhausted within the suction stroke at increased inlet pressure because of higher inlet air density. This amplifies delivered mass flow rate $\dot{m}$ and volumetric flow rate $\dot{V}_2$ at the outlet port, giving an opportunity to supply more pneumatic cylinders with the same compressor.

$$\dot{m} = \frac{V_1 \times \rho_1}{\Delta t} = \frac{V_1 \times p_1}{\Delta t \times R \times T_1} \tag{5}$$

$$\dot{V}_2 = \frac{\dot{m}}{\rho_2} = \frac{V_1 \times \rho_1}{\Delta t \times \rho_2} = \frac{V_1 \times \rho_1}{\Delta t \times \rho_2} \tag{6}$$

Although an increase in inlet air density gains the energy consumption according to Equation (1), the relation $\eta = W_{displ}/W_{compr}^{ad}$ remains in favor of the closed circuit. The advantage is that the compressor (but not the prime mover) needed to supply a given number of pneumatic drives can be downsized when operating in a closed-circuit system. A further notable effect is a decrease in discharge temperature brought about by reduction of the compression ratio:

$$T_2 = T_1 \times \varepsilon^{\left(\frac{\kappa-1}{\kappa}\right)} \tag{7}$$

Low discharge temperature contributes to lower thermal strain in the piston block and valve plates, reduces the wear of the sealing rings and carbonizing in oiled compressors. Hypothetically, bringing ε down to $\varepsilon \approx 1.8$ enables discharge temperatures of about $T_2 \approx 50...60$ °C. These temperatures are safe in case of a skin contact (with regards to the safety regulations ISO 13732-1) as well as tolerable by almost all pneumatic components. This allows to operate the compressor without a cooler and hence almost without any heat losses. However, very high inlet pressure, needed to reach the compression ratios of $\varepsilon < 1.8$, narrows the usable pressure difference to $\Delta p < 5$ bar, because the maximum operational pressure of most pneumatic cylinders and valves is usually limited to ~10 bar_{rel}.

STATE OF THE ART AND THE AIM OF THE STUDY

The observations noted and illustrated in the previous section are well-known in the literature. The most relevant for pneumatics theoretical study is carried out by Weiß [6]. He analyses the thermodynamic relations, considered above, and assesses the benefits of the closed-circuit compressed air systems. He shows that to achieve a typical for pneumatics differential pressure of 6 bar compression from 4 to 10 bar_{rel} brings significant benefits compared to the conventional compression from 0 to 6 bar_{rel}. In detail, the delivered flow rate, normalized to the normal conditions according to ISO 1217, increases by factor of five, whilst compression power is only doubled. This results in a significant reduction of specific energy consumption: 1.31 instead of 3.24 $\text{kW}\times\text{m}^{-3}\times\text{min}^{-1}$. A practical implementation of a closed-circuit system is expected to be challenging due to the effect of pressure oscillations caused by the non-continuity of air demand from the cylinders on efficiency of the continuously operated compressor. The components need to be optimized for one specific pressure ratio. To ensure an efficient air transportation (i. e. low pressure drop in the pipeline) piping tubes must be upsized because of higher volumetric flow rate and air density. Moreover, extra low-pressure piping line is needed to collect the exhausted air from the cylinders and convey it to the inlet port of the compressor, being an extra source of leakage and pressure loss. The closed-circuit system can only be applied, if no exhaust in the atmosphere is needed by the application, that is no vacuum ejectors or air-blasting are used. Finally, a technical solution for compensation of some inevitable air leakage from the closed system is needed.

A positive effect of the closed-circuit system is observed by authors of [7]. Here a closed system was implemented to supply the pneumatic tools typically used in the car repair shops. The toolkit proposed in [7] is suitable for modification of any compressor for an operation at inlet pressure up to 2 bar_{abs}. The influence of the inlet pressure increase on torque of the prime mover is not discussed. According to the authors the closed-circuit system enables energy costs savings of up to 40 %.

The impact of the back-pressure on the cylinder with a servo control as well as evaluation of energy savings are discussed in [8]. Apart from efficiency increase, a lower noise level due to absence of exhaust is noted by the authors as a further benefit. Another closely related approach for boosting the inlet pressure without closing the pneumatic circuit completely is proposed and successfully implemented by [9]. A special unit is designed to forward the exhausted air from the cylinder chamber through an ejector nozzle forcing the atmospheric air to pass to a low-pressure tank (~1.5 bar_{abs}) at the inlet port of compressor. On this way the energy intake by compressor can be reduced up to 15 %. The project is focused primary on the efficient ejector nozzle design capable of operating at different pressures.

Summing up, the evidence of thermodynamic advantages of pneumatic closed-circuit systems is shadowed by their possible drawbacks noted in the above-mentioned studies. These drawbacks put a question, whether a closed-circuit system can operate reliably and is economically sound. Up to the present moment no successful implementation of the closed-circuit pneumatics could be found in the published sources. In this context, the aim of this research is to study experimentally the performance of a compressor with increased inlet pressure as well as to implement and to test a pneumatic system operating in the closed circuit. Basing on the practical experience, the conclusions about applicability, prospects, and need in further research of such systems are drawn.

EXPERIMENTAL STUDY ON COMPRESOR PERFORMACE

Basically, any positive displacement compressor can operate with inlet pressure above the atmospheric, provided the suction line is mechanically strong enough to resist it. However, a compressor in closed circuit is more likely to operate at variable compression ratio ε because of temporary discrepancies between the volumetric flow rate delivery and demand. In this case, piston compressors, though being in general less efficient than rotary-screw compressors, are a better option, for their compression ratio is defined by the outlet pressure. That means, air is compressed in the piston chamber until the pressure level of discharge piping is reached, and a discharge valve opens. In case of rotary-screw or gear machine, the pressure ratio is defined by the inner geometry and is constant. If pressure in discharge line is higher or lower than pressure reached by the internal compression, the pressurized air volume shrinks or expands respectively, causing vibrations and energy losses. For this reason, a piston compressor was chosen for this experimental study. The aim of the experiment was to obtain the information about compressor performance at various inlet pressures and pressure differences, and to compare the open and the closed circuits. The pneumatic plan of the test rig is shown in Figure 2. Pressures and temperatures at inlet port, discharge port and after the cooler were measured along with normal flow rate $\dot{V}_0$ and real power $\dot{W}$ of the asynchronous motor. The rotational speed n was estimated from the Fourier analysis of the pressure oscillations. The inlet pressure was set by a pressure regulator to $p_1 \in \{0, 0.5, 1, 1.5, 2\}$ bar_{rel} and kept constant during each measurement. The pressure difference was adjusted by a throttle until outlet pressure of $p_2 \approx 8$ bar_{rel} was reached or the motor was overloaded.

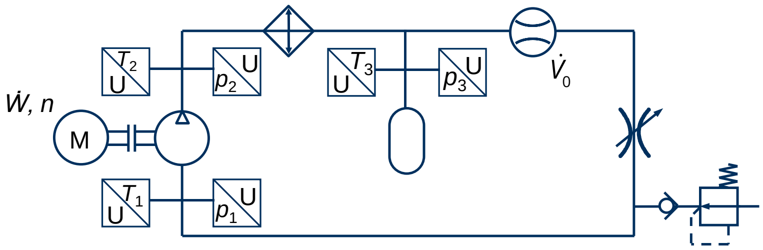

Figure 2 – Test circuit for experimental investigation of the 500-W-compressor performance

The data were obtained within at least 2 minutes of a steady compressor operation, i. e. after pressures and flow rate have settled down to a constant value. The results calculated from the mean values measured over this time are plotted in Figure 3.

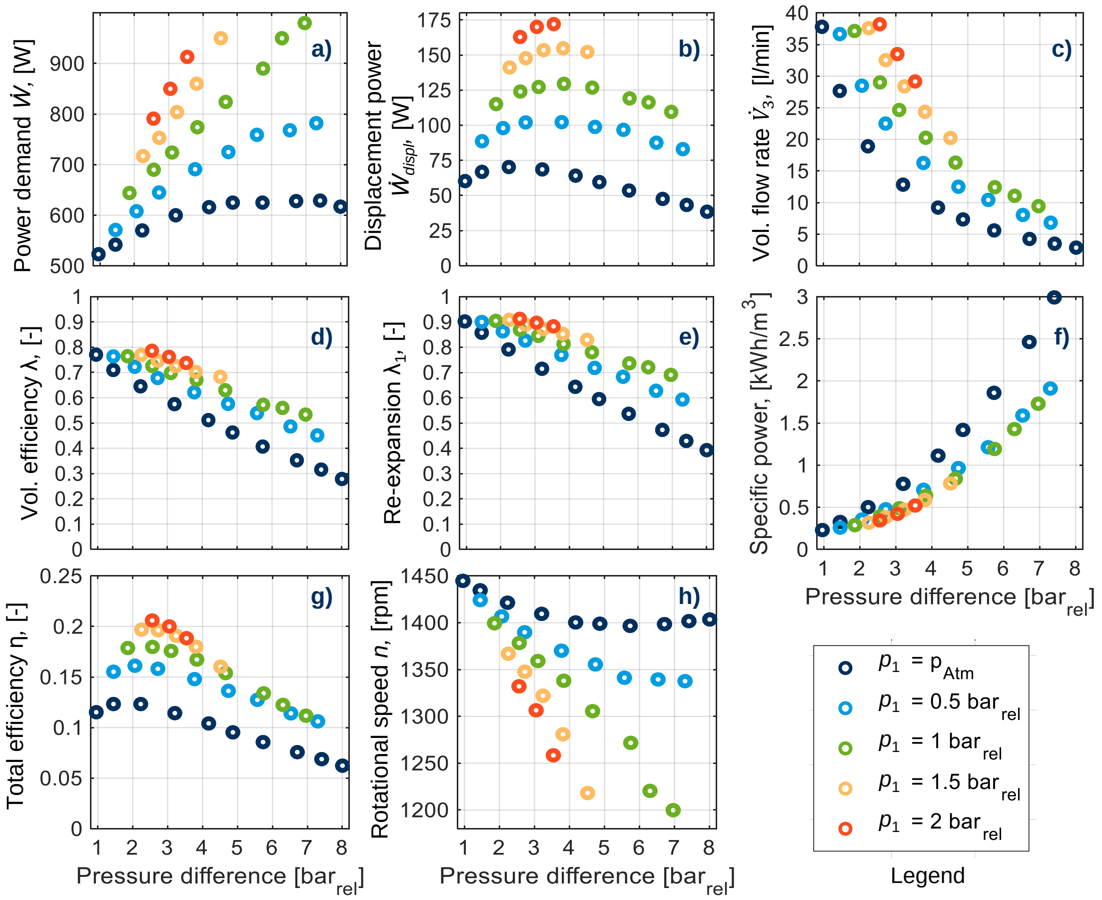

Figure 3 – Experimentally obtained performance characterization of the 500-W-compressor at various inlet pressures p_1 and pressure differences $\Delta p = p_2 - p_1$

Due to compressors under 1 kW designed for operation at inlet pressure above the atmospheric (usually called "booster compressors") are difficult to acquire, the experiments were carried out on a small one-stage, double-piston compressor designed for air intake from the atmosphere only. Despite on sufficient mechanical strength of the inlet port, this matter limited the maximum level of inlet pressure because of an overload of the asynchronous motor at $p_1 \geq 2$ bar$_{rel}$ and $\dot{W} > 900$ W (Figure 3, a)). The motor overload can be also observed by a drastic sagging of rotational speed at high inlet pressures (s. Figure 3, h)).

The theoretical power $\dot{W}_{displ}$ (Figure 3, b) that can be obtained with a displacement machine, such as pneumatic cylinder, driven by the delivered compressed air is calculated from Equation (2). Volumetric flow rate $\dot{V}_3$ (Figure 3, c) is estimated from the measured normal flow rate $\dot{V}_0$, pressure p_3 and temperature T_3 basing on Equation (6). As expected from the theoretical analysis, power demand $\dot{W}$, displacement power $\dot{W}_{displ}$ of a pneumatic cylinder and volumetric flow rate $\dot{V}_3$ increase with inlet pressure. Volumetric flow rate drops with rising discharge pressure due to the compressibility of air and the pressure-dependence of volumetric efficiency λ. Similar relations of volumetric flow rate and power demand to the pressure difference were also observed within tests of a scroll compressor for natural gas, carried out at different inlet pressures [10]. For displacement power $\dot{W}_{displ}$ being a product of pressure difference Δp and flow rate $\dot{V}_3$, the maximum peak is present. It is noticeable that the point of maximum power shifts towards the higher pressure differences with an increase in inlet pressure. This shift is caused by a lower gradient of the Δp-dependent volumetric efficiency λ at higher inlet pressures additionally contributing to increase in flow rate $\dot{V}_3$.

Volumetric efficiency λ is estimated from the displacement volume $V_{displ} = 33.2$ cm³ per piston, and rotational speed n applying Equation (8). According to [5], volumetric efficiency is a product of factors, representing re-expansion of the clearance volume λ_1, inlet air density reduction due to throttling in the inlet valve λ_2 and heat absorption while passing the inlet channels λ_3, as well as the leakage factor λ_4 (Equation (9)). While measuring λ_2, λ_3, and λ_4, requires sophisticated equipment, λ_1 can be estimated from the pressure values and the clearance volume of $V_{clear} = 5.3$ cm³ using Equation (10):

$$\lambda = \frac{\dot{V}_{measured}}{\dot{V}_{theor.}} = \frac{\dot{V}_1}{V_{displ} \times n} \tag{8}$$

$$\lambda = \lambda_1 \times \lambda_2 \times \lambda_3 \times \lambda_4 \tag{9}$$

$$\lambda_1 = 1 - \frac{V_{clear}}{V_{displ}} \times \left[\varepsilon^{\frac{1}{\kappa}} - 1\right] \tag{10}$$

It is remarkable that the volumetric efficiency that mostly depends on the re-expansion term λ_1, rises significantly with increase in inlet pressure. For this reason, the volumetric flow rate is gained even stronger than predicted alone with Equation (6). For example, at $\Delta p = 4$ bar and $p_1 = 1.5$ bar$_{rel}$ compressor delivers more than 2.5 times the flow rate delivered at $p_1 = p_{Atm}$ and the same Δp. This observation dispels the doubts about the poor compressor efficiency at increased inlet pressure. However, it can be expected that at very high inlet pressures $p_1 > 5$ bar$_{rel}$ the volumetric efficiency will be influenced by inlet air density decrease due to pressure losses in the inlet valve (λ_2). That may be especially relevant for the simple feather valves as in case of the studied compressor.

This discussion leads to analysis of relation of displacement cylinder power $\dot{W}_{displ}$ to power demand $\dot{W}$, noted here as total efficiency η. As in case of $\dot{W}_{displ}$, this relation has a clear optimal point, that moves towards the higher values of Δp with rising p_1. Since inlet pressure reaches $p_1 = 1.5...2$ bar$_{rel}$, the efficiency does not increase significantly anymore. However, this phenomenon can be explained with a poor motor performance in the overloaded region above $\dot{W} > 800$ W. It is expected, that driving the compressor with a more powerful asynchronous motor would enable a further performance increase and operation at higher inlet pressures over $p_1 > 2$ bar$_{rel}$.

Nevertheless, the present study shows that the performance improvement of a closed-circuit compressor is measurable and distinguishes itself by both increasing efficiency and compressor delivery. In the given case, volumetric flow rate was more than doubled, and efficiency was increased by about 50...60 %. Another measure of compressor efficiency, specific energy consumption (Figure 3, f) or relation of power demand to the volumetric flow rate, also illustrates a lower energy consumption in the closed system. In contrary to the standard definition of specific energy consumption, the relation of volumetric flow rate to the normal conditions can be misleading in the closed-circuit systems. Therefore, the volumetric flow rate $\dot{V}_3$ related to the conditions after compression and cooling is used here. Facing the fact that large high-performance rotary-screw compressors have a specific energy consumption of ~0.7...0.9 kWh·m³×h^{-1} at $\Delta p = 7$ bar and the tested compressor needs double this specific energy, must be noticed that the smaller the compressor is the lower is its efficiency. Implementation of a closed-circuit system with a well-sized prime mover and a small compressor of industrial quality with a power range of 1...5 kW makes feasible the decrease in specific energy consumption up to the values currently typical only for the large high-performance rotary-screw compressors. This offers a great market opportunity for small compressors to be used for decentralized air supply in the closed-circuit architecture.

EXPERIMENTAL IMPLEMENTATION OF CLOSED-CIRCUIT PNEUMATIC SYSTEM

In the previous section the compressor performance under the constant load, that is constant pressure difference caused by continuous air consumption, was analyzed. To address the uncertainty and anticipations of a negative effect of oscillating pressure and volumetric flow rate consumption by real consumers with discontinuous behavior two pneumatic cylinders were added to the studied compressor. For the sake of demonstrativeness, the cylinders of different size performing different tasks were chosen, as specified in the Table 1: high-dynamic horizontal handling of a mass within the specified time and extension against the constant force (lifting and descending of a large mass).

Table 1 – Operational cases for two cylinders at pressure Δp = 4 bar$_{rel}$

Load specification for extension and retraction	Horizontal cylinder H, Ø32×200	Vertical cylinder V, Ø50×200
Handled mass m, [kg]	10	65
Force, F_{ext} // F_{restr}, [N]	0 // 0	640 // -640
Travel time, t_{ext} // t_{retr}, [s]	0.34±0.02 // 0.44±0.03	<3 // <3

The cases were parameterized in the way that both cylinders are properly loaded and not oversized for their tasks. The horizontal cylinder was sized using the Pneumatic Frequency Ratio (PFR) approach proposed by [11]. According to that, both PFR Ω_{retr} = 1.37 and Ω_{ext} = 1.77 correspond to a properly sized cylinder that can operate safely with a standard pneumatic end cushioning. The constant force of the vertical cylinder is calculated to be about 80 % of its theoretical pneumatic force, hence leaving some space for acceleration and overcoming the friction force, not included into the load forces in Table 1. The test circuit is shown in Figure 4.

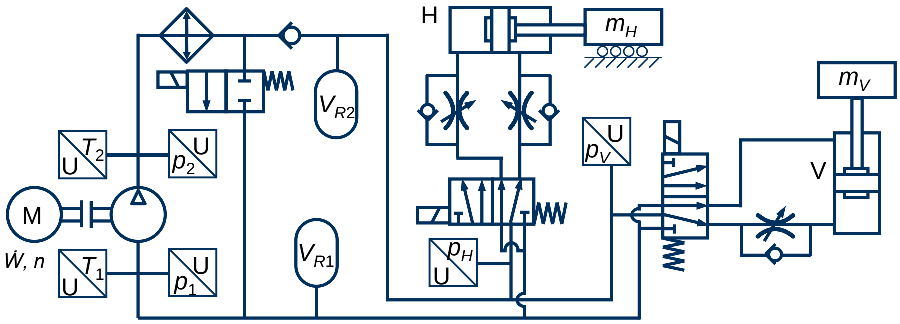

Figure 4 – Test circuit with compressor and two cylinders operating in a closed circuit. Tank volumes on the discharge and suction site are V_{R1} = 9 l and V_{R2} = 2 l respectively.

Before starting the measurements, the system was filled with dry compressed air. The filling pressure depends on the volumes of low-pressure and high-pressure sides (mostly composed of tanks V_{R1} = 9 l and V_{R2} = 2 l), desired steady-state inlet pressure p_1, differential pressure Δp_{cyl} needed for operation of cylinders and their flow rate demand. To estimate Δp_{cyl}, pressure was measured at supply and exhaust ports of the cylinder valve. Differential pressure on compressor Δp is greater than the differential pressure on the cylinder Δp_{cyl} by the value of pressure losses in the piping tubes. Due to no simple method for calculation of the filling pressure is currently present, this pressure was estimated empirically within the tests and amounted p_{fill} = 2…2.65 bar$_{rel}$ for the studied cases. After being filled, the system was disconnected from the pressure source and operated completely autonomous.

Volumetric flow rate, delivered from the compressor, is consumed by the both cylinders, switched every $t_{H.takt}/2$ and $t_{V.takt}/2$ seconds (s. Table 2) in such a way, that cumulative air consumption of the horizontal cylinder is about two times that of the vertical cylinder. The total volumetric flow rate can be determined from the already known compressor performance data shown in Figure 3, c) depending on the inlet pressure and pressure difference. With regards to the different volumes of the cylinders, the horizontal cylinder represents a nearly constant load, and the vertical cylinder brings low-frequent (with frequency of $2/t_{V.takt}$), high-amplitude peak load. In Figure 5 two forms of pressure oscillations caused by overlapping of these different consumption profiles can be easily distinguished.

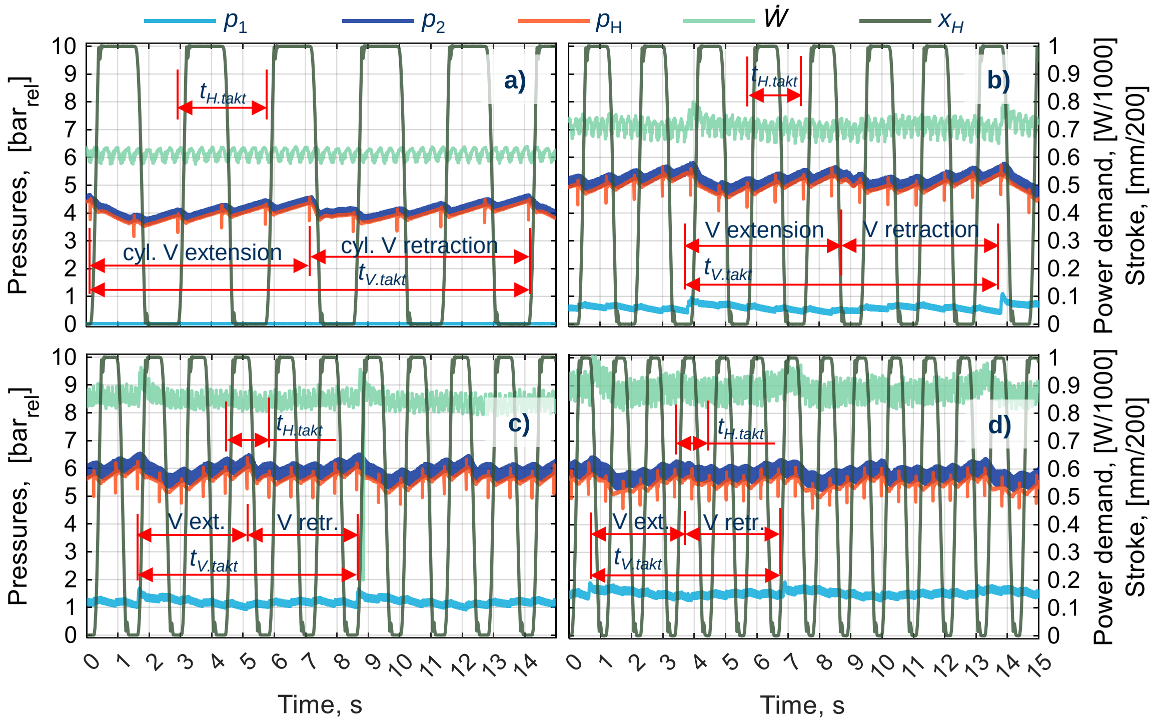

Figure 5 – Fragments of the measured inlet pressure p_1, discharge pressure p_2, supply pressure of horizontal cylinder p_H, power demand $\dot{W}$, and horizontal cylinder stroke x_H at a) p_1 = 0 bar_{rel} (open circuit), b) $p_{1.mean}$ = 0.59 bar_{rel}, c) $p_{1.mean}$ = 1.17 bar_{rel}, d) $p_{1.mean}$ = 1.5 bar_{rel}.

To evaluate the quality of the cylinder operation at different inlet pressures, travel times of both cylinders were captured during the operation for extension and retraction strokes ($t_{H.ext}$ and $t_{H.retr}$ for horizontal and $t_{V.ext}$ and $t_{V.retr}$ for the vertical cylinder in Table 2). Due to the mean differential pressure was slightly above the desired value of $\Delta p_{cyl} = 4$ bar_{rel} in cases b) and c), travel times of the vertical cylinder appears to be slightly lower in these cases. The dispersion of travel times within one measurement session is similar for all experiments with the closed circuit and does not exceed that for the open-circuit operation. Hence, the expectations of disruptive and non-representative dynamic behaviour of cylinders in the closed system could not be confirmed. The closed-circuit pneumatics seems to operate as reliable as the classical open system even with the non-constant load. Remarkable is an amplitude decrease of the high-frequent and low-frequent pressure oscillations with increasing inlet pressure.

Table 2 – Cylinder operation in the closed-circuit pneumatic system at $\Delta p \approx 4$ bar_{rel}

$p_{1.mean}$, [bar_{rel}]	Δp, [bar_{rel}]	Δp_{cyl}, [bar_{rel}]	$t_{H.takt}$, [s]	$t_{V.takt}$, [s]	$t_{H.ext}$, [s]	$t_{H.retr}$, [s]	$t_{V.ext}$, [s]	$t_{V.retr}$, [s]	$\dot{W}_{mean}$, [W]	$\dot{V}_3$, [l/min]	η_{cyl}, [-]
0	4.07	3.93	2.8	14	0.36… …0.37	0.43… …0.46	2.3… …2.65	2.64… …2.65	620	10.1	0.107
0.59	4.62	4.42	1.8	10	0.33… …0.34	0.41… …0.43	1.64… …1.67	2.74… …2.77	720	15.2	0.155
1.17	4.82	4.52	1.4	7	0.33… …0.34	0.42… …0.45	1.61… …1.69	2.84… …2.85	851	20.2	0.181
1.5	4.41	4.04	1.1	6.2	0.33… …0.35	0.43… …0.46	1.95… …1.96	2.76… …2.8	906	24.8	0.184

Considering the efficiency as relation of the mean displacement power of both cylinders to the mean power demand of compressor $\eta_{cyl} = \Delta p_{cyl} \times \dot{V}_3/\dot{W}_{mean}$, an efficiency increase of 72 % (from 10.7 % to 18.4 %) can be observed when comparing the open circuit and the closed circuit at p_1 = 1.5 bar_{rel}. These values match good with the measurements carried out on compressor with constant pressure and constant flow rate in the previous section (Figure 3, g)).

CONCLUSION

The aim of this study was firstly, to prove experimentally the earlier analytically outlined theory about the increase in efficiency and delivery of a compressor operating at the boosted inlet pressure, and secondly, to perform the test of a real pneumatic system that is supplied from this compressor and feeds the exhausted air back to its inlet at various pressures within a range from 0 (open circuit) to 1.5 bar_{rel}. The test rig included a small one-stage double-piston compressor with a displacement volume of ca. 33 cm³ per piston and two pneumatic cylinders of 32 and 50 mm in diameter. The experiments have shown a real increase in efficiency of up to 72 % and the volumetric flow rate gain by factor 2.5. The compressor performance is very similar in both standalone test with continuous volumetric flow demand and in the experiments with discontinuous air consumption. In the latter case, a smaller cylinder has imitated a constant load and the larger cylinder a peak load. No disturbances were observed in the cylinder operation. Both types of operational cases, i. e. high-dynamic motion and extension against the constant force, were performed with the repeatability of the classical open-circuit system. Besides, following points should be noted:

- Volumetric efficiency of compressor rises significantly with increasing inlet pressure.
- The prime mover must be properly sized for an optimal operation with a closed-circuit compressor because of higher torque at higher inlet air density.
- An additional benefit is a considerably lower noise level observed during experiments because the air is neither taken from nor exhausted in the atmosphere.
- Increase in pressure level in both low-pressure and high-pressure sides is favourable for high-dynamic tasks because it increases the stiffness of the air springs in the cylinder.
- Increase in pressure losses caused by higher flow rate and air density is measurable (compare the differences between Δp and Δp_{cyl} for various inlet pressures in Table 2), but not crucial and easily avoidable by a proper sizing of the piping system.
- Due to the relatively short test phases in this study (several minutes only), no major issues with the air leakage from the closed system were observed. However, this will be the case in real applications and a system for leakage compensation must be designed.

An economic benefit of the closed-circuit pneumatic system relies on a higher energy efficiency and hence lower energy costs as well as the possibility to downsize the compressor and hence to decrease investment costs. On the other hand, additional piping lines are needed to convey the exhausted air from the valves to the compressor as well as components for compensation of the leakages. Therefore, especially pneumatic systems with closely located drives can profit from the closed-circuit architecture with a local and decentralized air supply. In this case, the further corners can be cut on the absent piping infrastructure and non-present pressure losses throughout the central pipeline. Besides, in a smaller decentralized closed system with a constant air mass, the leakages are easier to detect, access and eliminate, saving further energy costs.

OUTLOOK

The further research is needed to design and validate the methods and algorithms for calculation of large and complex pneumatic closed-circuit systems. Cost-effective strategies for the volumetric flow management for different types of the air consumption profiles are of a highest interest. As already noted, an automated leakage compensation must be designed in order to make the closed-circuit system appliable for real industrial conditions.

NOMENCLATURE

ε	Compression ratio [-]
η	Efficiency [-]
κ	Isentropic exponent [-]
λ	Volumetric efficiency [-]
ρ	Density [kg × m^{-3}]
Ω	Pneumatic frequency ratio [-]
F	Force [N]
m	Mass [kg]
$\dot{m}$	Mass flow rate [kg × s^{-1}]
n	Rotational speed [s^{-1}]
p	Pressure [bar]
R	Specific gas constant [J × kg^{-1} × K^{-1}]
t	Time [s]

T	Temperature [°C]
V	Volume [m³]
$\dot{V}$	Volumetric flow rate [$m^3 \times s^{-1}$]
W	Work [J]
$\dot{W}$	Power [W]
x	Stroke [m]

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