FEASIBILITY STUDY AND EXPERIMENTAL VALIDATION OF A NOVEL COMBINED THROTTLING APPROACH

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ABSTRACT

In the field of pneumatic automation, downstream throttled pneumatic drives are commonly used for motion tasks due to their cost-effectiveness, durability, and robustness. However, this type of system is often regarded as being inefficient. Consequently, many researchers have focused on developing more efficient control strategies. This paper presents the results of an experimental investigation of three distinct pneumatic circuits: a novel combined throttling approach, the control by a commercially available downstream throttle with quick exhaust function and conventional downstream throttling. A metric for comparing energy savings adjusted to changes in cycle time is introduced for objective evaluation. The findings highlight a significant reduction in normalized air consumption with the novel circuit compared to both state of the art control schemes.

Keywords: Pneumatics, Energy Savings, Sustainable Systems

1. INTRODUCTION

Currently, downstream throttling is the primary method used to control the speed and cycle time of pneumatic drives. In the typical circuit configuration, the drive chamber is pressurized with the full supply pressure using a 5/2 directional control valve. The opposing chamber employs downstream throttling to regulate the speed by building up a pressure that opposes the movement, referred to as 'back pressure' below. With this setup, the cylinder always consumes the maximum amount of compressed air due to its compressibility and pressurization of the actuator chamber to full supply pressure.

In upstream throttling, the driving cylinder chamber is pressurized only to the extent required for the load to move at the desired speed, while the opposing chamber is vented to the ambient atmosphere. Due to the inherent resistance of the venting paths, only minimal back pressure occurs. As a result, upstream throttling can represent a highly efficient implementation of pneumatic movement. Provided that a further supply of compressed air is prevented when the end position is reached it consumes less compressed air than downstream throttling due to the lower actuator pressure. This crucial feature can be achieved by using a 5/3-way valve with a closed neutral position.

A key difference between the two systems is the back pressure in the exhaust chamber at the end of the movement, which has a major influence on pneumatic end-cushioning. End-cushioning is essential to prevent damage resulting from high-speed piston impacts and plays a crucial role in the dimensioning of pneumatic actuators concerning their cycle time [1][2]. Inherently, upstream throttled drives exhibit low back pressure, which can have an adverse effect on end-cushioning function depending on the load condition. This often results in unacceptably longer cycle times when employing upstream throttling.

Numerous researchers have proposed control strategies with the aim of enhancing the efficiency of systems employing pneumatic drives. Systems designed to maximize the utilization of the expansion energy within compressed air have been examined in references such as [3], [4], [5], and [6]. One especially noteworthy example is the Festo motion terminal ([7][8]), which uses four proportional valves arranged in a Wheatstone bridge configuration. However, it's worth noting that these methods do add complexity to the system and increase the procurement costs and the workload during commissioning, which could pose challenges for broader implementation. Other simpler approaches to energy savings have also been proposed by some researchers, such as the intermediate storage of compressed air, as suggested in [9][10], or the reuse of compressed air via a rapid traverse and feed circuit [11]. The design of these approaches is highly application specific and therefore may have limited transferability. A promising approach for optimizing the efficiency of pneumatic drives has already been proposed by the authors [12], which involves combining downstream and adaptive upstream throttling. In the scope of this paper, a novel configuration of this concept is investigated.

2. INVESTIGATED SYSTEMS

The left side of **Figure 1** depicts the switching scheme originally presented at FPMC2020 [12]. This approach aims to combine the advantages of downstream and upstream throttled systems. Downstream throttles control the cylinder speed, while upstream proportional throttles adaptively regulate the air supply. If it is possible to achieve satisfactory pneumatic end-cushioning and, consequently, acceptable cycle times with a lower drive pressure compared to downstream throttling, there is significant potential for compressed air savings. The use of a 5/3-way valve with a closed neutral position is necessary in both circuits. Furthermore, the switching of the valve to the neutral position must be timed with the cylinder reaching its end position to prevent further unnecessary filling of the driving cylinder chamber after the motion has been completed.

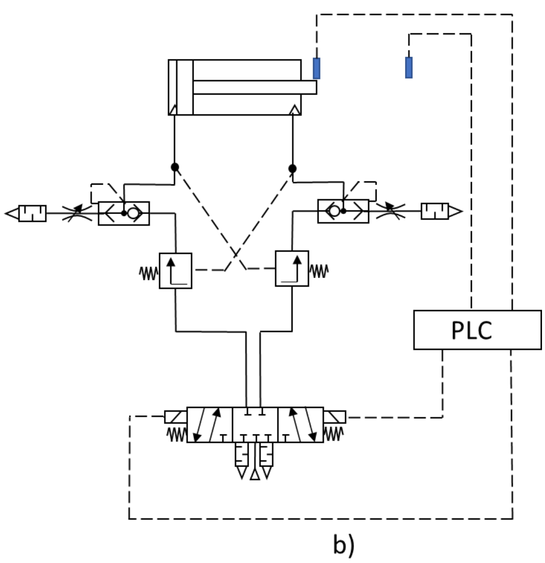


Figure 1: Adaptive upstream throttling configurations: a) original circuit and b) circuit with quick exhaust

The alternative design of the adaptive upstream throttling in Figure 1 (right) is based on the exhaust air flow being discharged into the environment by means of a quick exhaust valve. In contrast, in the original circuit diagram on the left side of **Figure 1**, the exhaust air flows through the directional control valve. The quick exhaust valve combines the logic structure of both check valves connected in parallel in the original switching scheme, resulting in a more compact design while simultaneously reducing pressure losses, especially when using longer pneumatic tubes.

The use of valve terminals is established in pneumatic systems, combining several pneumatic valves in one module. This leads to a reduction in installation and cabling effort. However, the use of this technology also increases the need for longer pneumatic tubes to reach the actuators. Therefore, this paper aims to also investigate the impact of tube sizes on the dynamics of pneumatic drives and air consumption. **Figure 2** illustrates the systems investigated in this paper. In addition to the conventional downstream throttled system a), the effects of quick exhaust on such systems is considered by means of the system b) to enable an objective comparison of all systems.

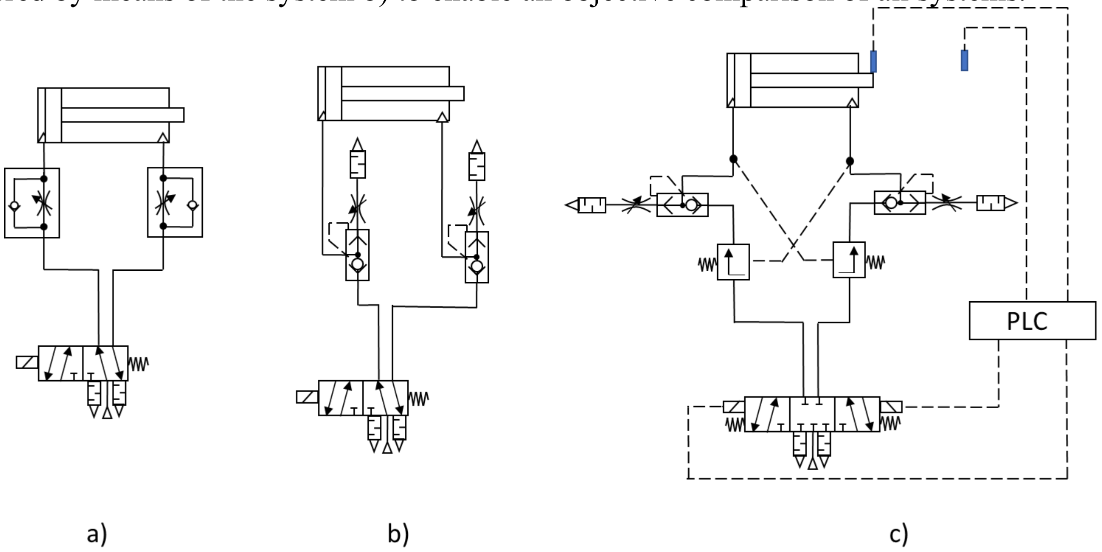


Figure 2: Investigated Systems: a) downstream throttling (DT), b) downstream throttling with quick exhaust (DTQ) and c) adaptive upstream throttling with quick exhaust (CTQ).

3. DESIGN OF THE ADAPTIVE UPSTREAM THROTTLING CONCEPT

Figure 3 shows the mechanical design of the concept with quick exhaust valves, whose circuit diagram was presented in Section 2. All the necessary components are combined in a common housing. The concept divides the circuit branches for each cylinder chamber into two levels, with the upper level shown in Figure 3 through a sectional view. To achieve a more compact design, upstream proportional controllers with a sleeve-spool design were chosen. All the required fluid pathways are integrated into the housing, including pressure signal holes for actuating the upstream controllers.

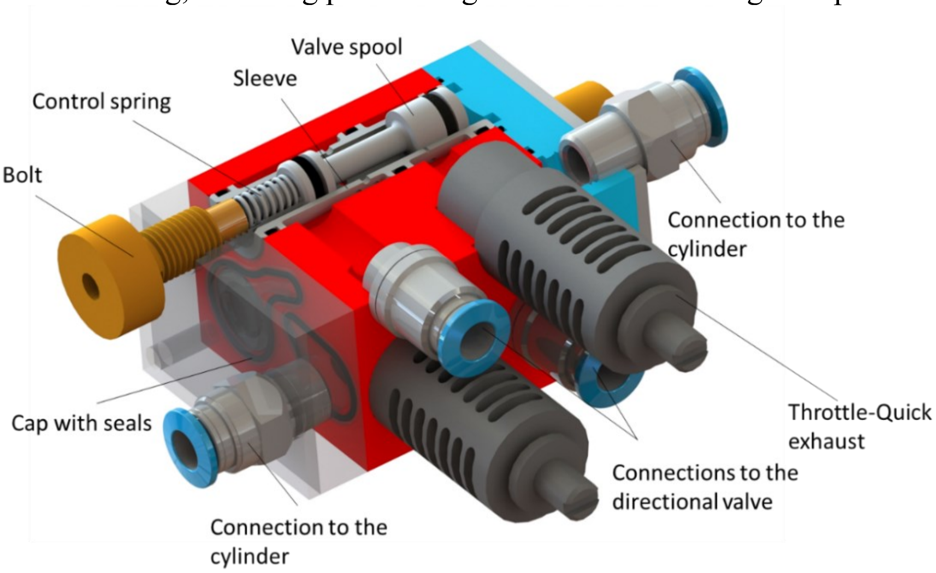


Figure 3: Mechanical design of the novel switching scheme

The manufactured prototype, as depicted in **Figure 4**, is used for the experimental tests conducted in this paper. The housing is designed from aluminium, while the cap is 3D printed using MSLA technology. The block's design is symmetrical, allowing for the use of the same cap shape on both sides. The model of the quick exhaust throttles is ASV 310 F-01-06 S from the SMC company, which were modified to be glued into the corresponding housing’s bores, with the main functionality remaining unchanged. The same exhaust throttle model was later employed as a standalone component to test the impact of quick exhaust on downstream throttled drives (see **Section 4**).

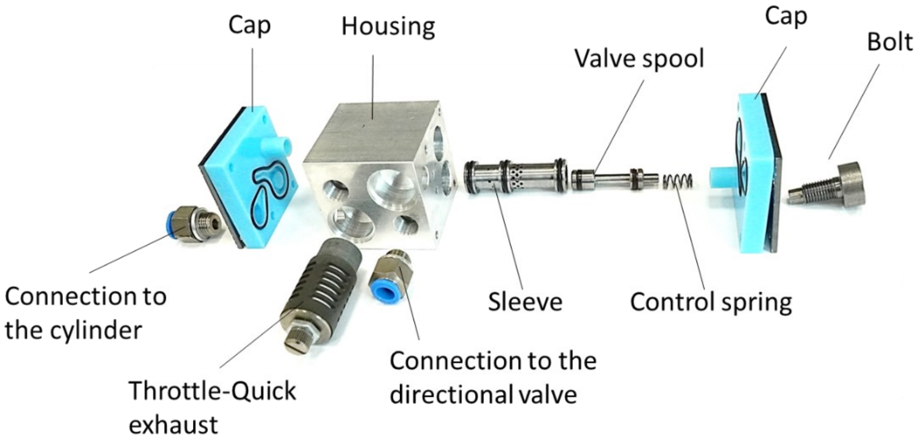


Figure 4: Exploded view of the manufactured components

3.1. Metering Edge Design

An essential aspect of controller design involves the careful selection of an appropriate metering edge geometry, since it has big implications on the metering characteristics of the valve and subsequently on the closed loop stability. **Figure 5** shows two possible implementations of the desired metering edge geometry.

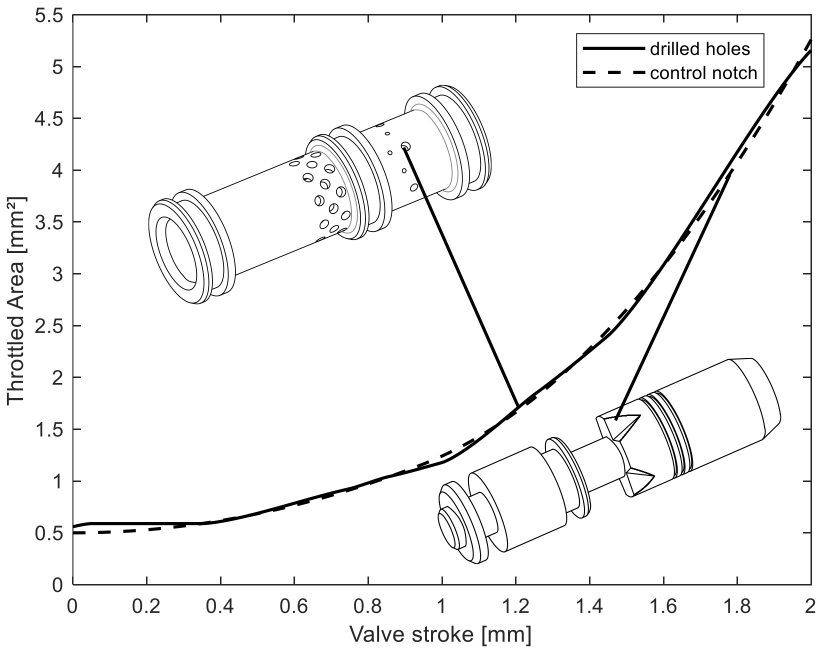
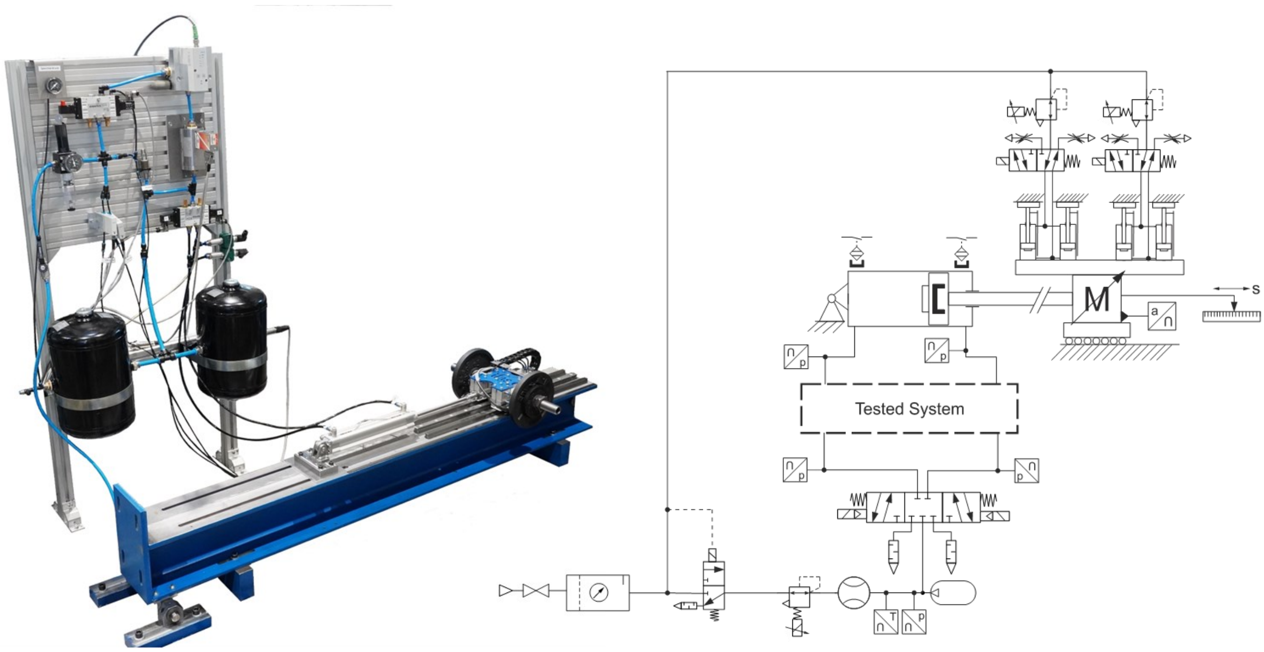


Figure 5: Metering edge design

To achieve satisfactory system performance, it is crucial to maintain closed-loop stability while ensuring sufficient conductivity to generate the necessary back pressure. The authors demonstrated that a linear opening characteristic is inadequate for reliable operation across a broad range of parameters [13]. Consequently, an alternative design approach incorporating a non-linear control notch geometry was proposed. This design method utilizes a gain-scheduling approach, where controller gains are adjusted based on the steady-state error at different operating points, using a linear model of the system dynamics. A comprehensive explanation of this design method exceeds the scope of this paper; therefore, readers are referred to prior work [13] for more in-depth information. As mentioned earlier, a sleeve-spool design was selected to implement the metering edge of the controller due to its advantages in achieving a more compact design. Consequently, the opening geometry of the control notch design was approximated using drilled holes of varying sizes in the sleeve, as illustrated in Figure 5. The valve spool gradually exposes these holes to the airflow, thereby increasing the flow cross-section in the desired manner.

4. EXPERIMENTAL VALIDATION

For the experimental validation the test rig shown in **Figure 6** was used. The test rig allows for the air consumption and dynamic behaviour of the cylinder to be measured. Depending on the system, the 5/3 directional control valve is controlled as a 5/2 directional valve if the adoption of the neutral position is not required by the system. In this way, different systems with and without shut-off can be examined without change in the test rig components.



a) b)

Figure 6: Test rig: a) implementation and b) circuit

4.1. Test plan

The investigated system parameters are listed in **Table 1** below. Three cylinder sizes were tested with varying tube lengths. The conventional downstream throttling (DT) was dimensioned using the sizing tool provided by a pneumatic components manufacturer [14]. It should be noted that with longer tube lengths, the tube diameter must be increased with such system due to the higher resistances in the return line. This can impact the proper functioning of the downstream throttle. In contrast, the systems with quick exhaust (DTQ and CTQ) do not experience an increase in downstream resistance and will therefore be tested with only one tube diameter. This approach allows for the evaluation of the effect of lower dead volume and higher upstream resistance on air consumption and cycle time. Furthermore, during the experimental investigation, all systems utilized the air cushion to its maximum loading capacity. Consequently, the minimal achievable cycle time was measured for each system, thus allowing for an objective comparison.

Table 1: Test matrix with the corresponding tube diameters

		DT			DTQ/CTQ		
		Tube length [m]			Tube length [m]		
Cylinder size [mm]	Mass [kg]	1	4	6	1	4	6
Ø 25 x 250	5	Ø 6	Ø 6	Ø 6	Ø 6	Ø 6	Ø 6
Ø 32 x 200	10	Ø 6	Ø 8	Ø 8	Ø 6	Ø 6	Ø 6
Ø 32 x 320	10	Ø 6	Ø 8	Ø 8	Ø 6	Ø 6	Ø 6

4.2. Dynamic Behaviour

In this section, an exemplary comparison of the dynamic behaviour of all investigated systems using a cylinder with dimensions Ø 32 x 200 is provided. Quantitative results for all systems and cylinder sizes will be presented in Section 4.3. **Figure 7** shows the cylinder's speed and pressure curves within its chambers using the different systems listed in Table 1, with a used tube length of 1 meter.

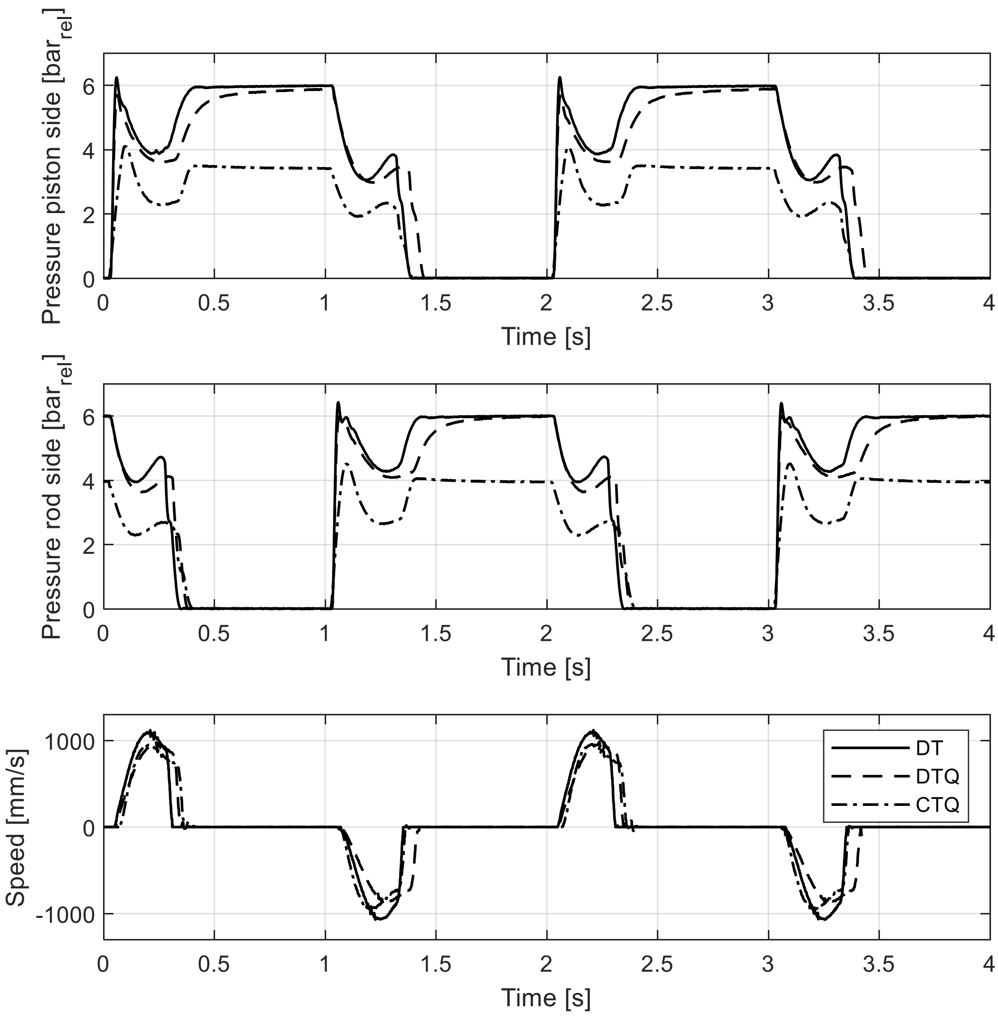


Figure 7: Dynamic behaviour: Ø 32 x 200, M = 10 kg and tube length = 1 m

The Combined Throttling Circuit (CTQ) exhibits lower back pressures and, consequently, lower driving pressure during cylinder movement compared to the other systems. Additionally, in systems utilizing the directional valve as a 5/2 directional valve (DT and DTQ), the pressure in the active chamber rises to the supply pressure (6 bar_{rel}), whereas in CTQ, this is prevented by using the center position of the 5/3 directional valve as a shut-off. It is worth noting that the pressure rise at the end of the movement is much slower when using the quick exhaust throttle (DTQ) compared to the standard downstream throttling (DT), which indicates higher upstream pressure losses. Surprisingly, the cylinder experiences a lower start-up acceleration with the quick exhaust throttle (DTQ) than with the other systems. This may be attributed to higher upstream pressure losses resulting from the compact design of this commercially available valve, combined with the characteristic high backpressures associated with downstream throttling. On the other hand, CTQ exhibits a very rapid start of the movement especially on the return stroke, possibly due to the overall lower backpressure.

All the systems exhibit slower pressure build-up with a longer tube, as shown in **Figure 8**, due to the higher dead volumes. However, due to the larger tube diameter used in conventional downstream throttling (DT), an even higher driving pressure can be observed compared to Figure 7. Notably, the quicker start of the novel circuit (CTQ) on the return stroke remains observable despite the longer tubes.

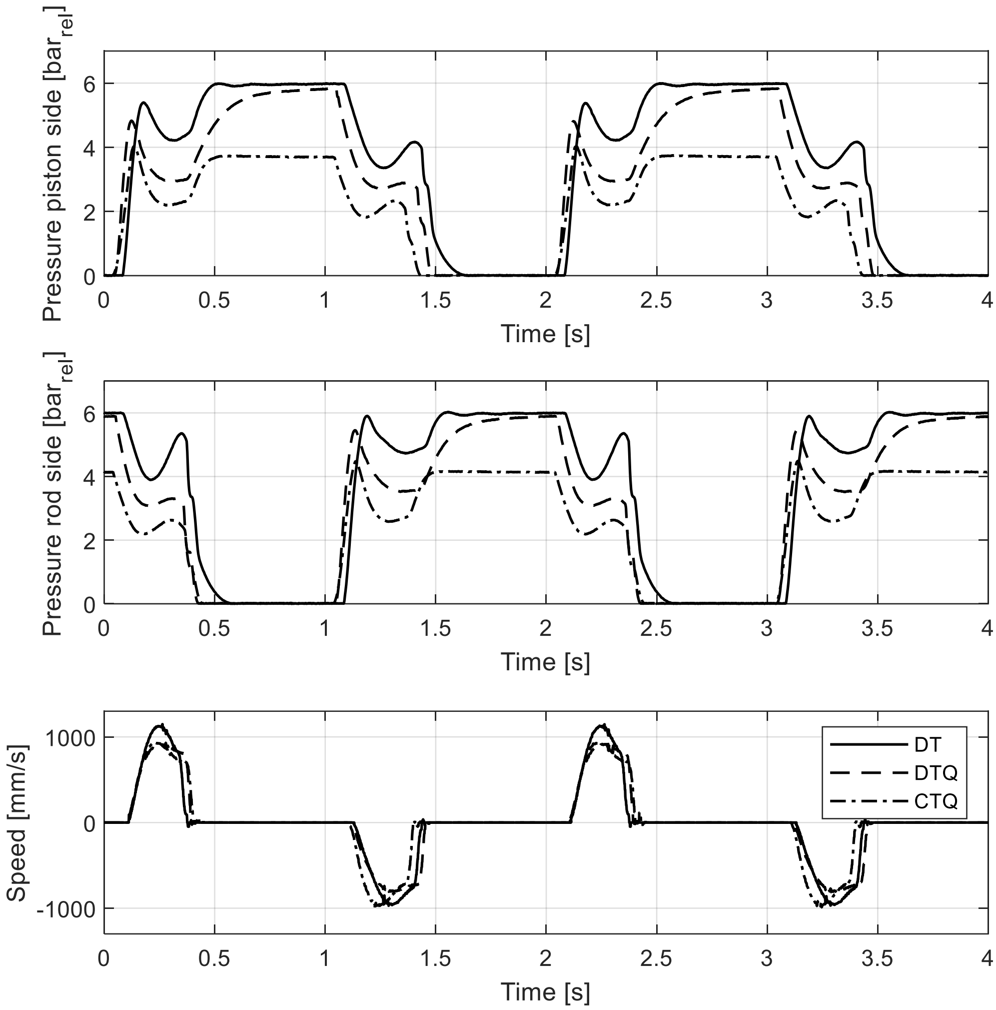


Figure 8: Dynamic behaviour: Ø 32 x 200, M = 10 kg and tube length = 6 m

4.3. Evaluation of the results

In this section, the experimental results are summarized. To facilitate a fair comparison between the systems, it is essential to account for the interaction between an increase in cycle time and potential compressed air savings. Therefore, in addition to absolute performance metrics such as cycle time and air consumption, this paper will also present a normalized air consumption adjusted for changes in cycle time. For that, the sizing guideline for downstream throttled drives proposed by Doll et al. will be used [2], which is based on a "pneumatic frequency ratio" Ω. The value for Ω can be calculated using (1) and relates the reciprocal of the eigenfrequency (t_0) of the piston to the transfer time (t_f). A well dimensioned cylinder exhibits Ω values between 1,1 and 1,7.

$$\Omega = \frac{t_{f,DT}}{t_0} = \frac{t_{f,DT}}{\pi}\sqrt{\frac{A_p \cdot p_s}{M \cdot (L + 2 \cdot L_{dead})}} \tag{1}$$

Furthermore, the air consumption can be approximated with (2), with α being the cylinder area ratio.

$$m_{Air,DT} \approx \frac{A_p \cdot (L \cdot (1+\alpha) + 2 \cdot L_{dead})}{R \cdot T_0} \cdot p_s \tag{2}$$

By substituting equation (1) into (2), an expression for the compressed air consumption as a function of the transfer time can be obtained (3).

$$m_{Air,DT} = \frac{M \cdot (L \cdot (1+\alpha) + 2 \cdot L_{dead}) \cdot (L + 2 \cdot L_{dead})}{R \cdot T_0} \cdot \left(\frac{\Omega \cdot 2 \cdot \pi}{t_{f,DT}}\right)^2 \tag{3}$$

Assuming, that neither the operating temperature nor Ω change, the relationship between compressed air requirement and cycle time in equation (4) results. Here ${m_{Air,DT}}^*$ corresponds to the compressed air consumption at a different transfer time ${t_{f,DT}}^*$. Furthermore, we will use the complete cycle time instead of just the transfer time, under the assumption that the ratio of transfer time to dead time of the downstream throttled drive remains relatively constant at different pressure levels.

$$\frac{{m_{Air,DT}}^*}{m_{Air,DT}} = \left(\frac{t_{f,DT}}{{t_{f,DT}}^*}\right)^2 \approx \left(\frac{t_{cycle,DT}}{{t_{cycle,DT}}^*}\right)^2 \tag{4}$$

Finally, the normalized air consumption can be calculated as the ratio of the air consumption of a given system $m_{Air,i}$ to the corrected air consumption of a downstream throttled drive at the corresponding cycle time ${m_{Air,DT}}^*$.

$$Normalized\ m_{Air,i} = \frac{m_{Air,i}}{{m_{Air,DT}}^*} = \frac{m_{Air,i}}{m_{Air,DT}} \cdot \left(\frac{t_{cycle,DT}}{t_{cycle,i}}\right)^2 \tag{5}$$

Figure 9 to **11** display the results of the different cylinder sizes with the measured Ω for the downstream throttling. In Figure 9, all the systems used the same tube diameters, despite varying tube lengths as indicated in Table 1. As expected, both systems without a shut-off function (DT and DTQ) exhibit similar air consumption for all tube lengths. Interestingly, the normalized air consumption is higher for DTQ and CTQ with the shortest tube length, primarily due to their longer measured cycle time. However, this trend reverses for longer tube sizes, as the advantages of quick exhaust on cycle time become more pronounced.

The differences in the absolute air consumption between DT and DTQ in Figure 10 are solely caused by the variances in tube diameter. Thus, this difference can be understood as savings resulting from reduced dead volumes. The combined throttling circuit (CTQ) exhibits remarkable results in terms of normalized air consumption for this cylinder size, especially with longer tubes. Therefore, it can be concluded that the novel system is particularly advantageous with this specific cylinder size. This can be partially attributed to its faster dynamic behaviour at the beginning of the return stroke, as demonstrated in Figures 7 and 8.

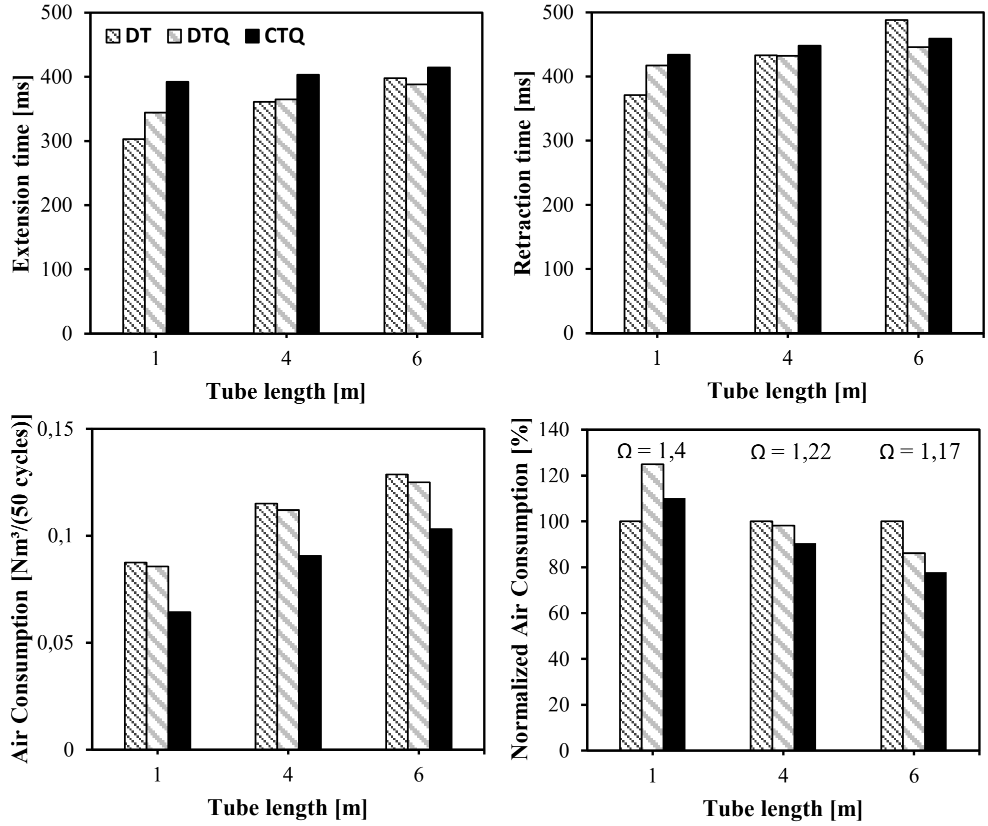


Figure 9: Measured results for Ø 25 x 250 with M = 5 kg

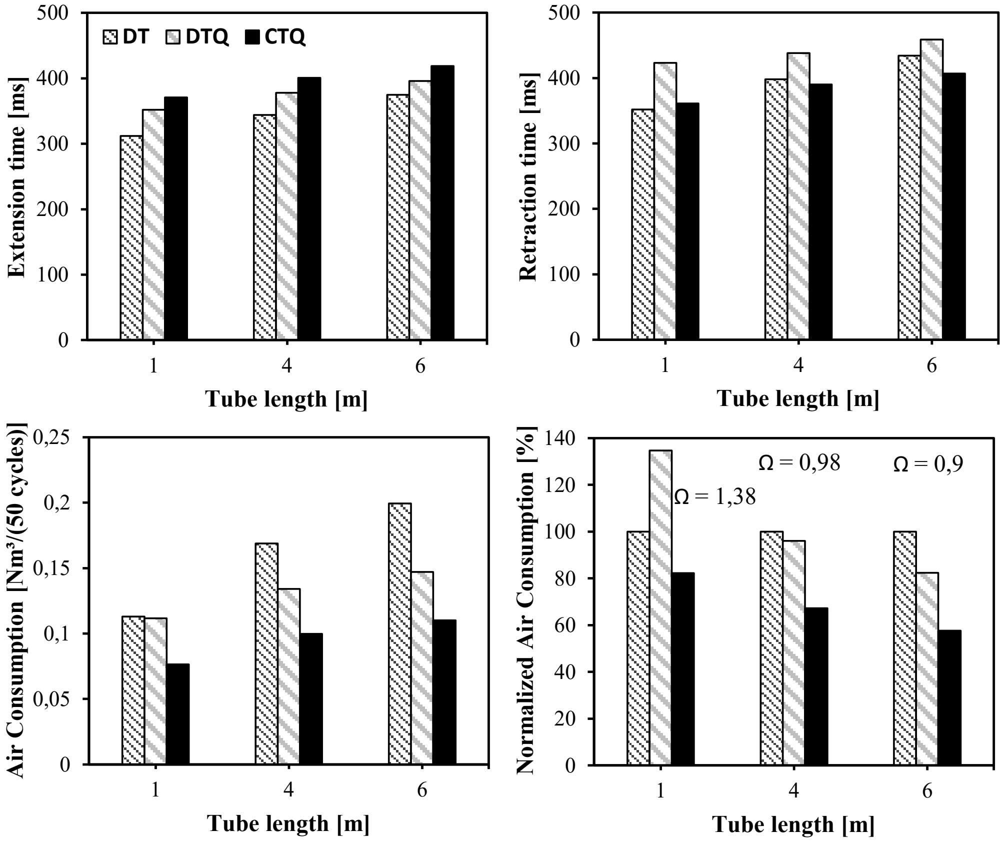


Figure 10: Measured results for Ø 32 x 200 with M = 10 kg

Figure 11 displays the results for the largest cylinder. Interestingly, despite the compressed air savings resulting from the smaller dead volumes used in DTQ compared to DT, DTQ performs worse than

DT in terms of normalized air consumption. One possible explanation is that the relative savings due to dead volume become lower with the larger cylinder and no longer outweigh the increase in cycle time. The normalized savings of the novel combined throttling approach (CTQ) remains largely unchanged with different tube sizes.

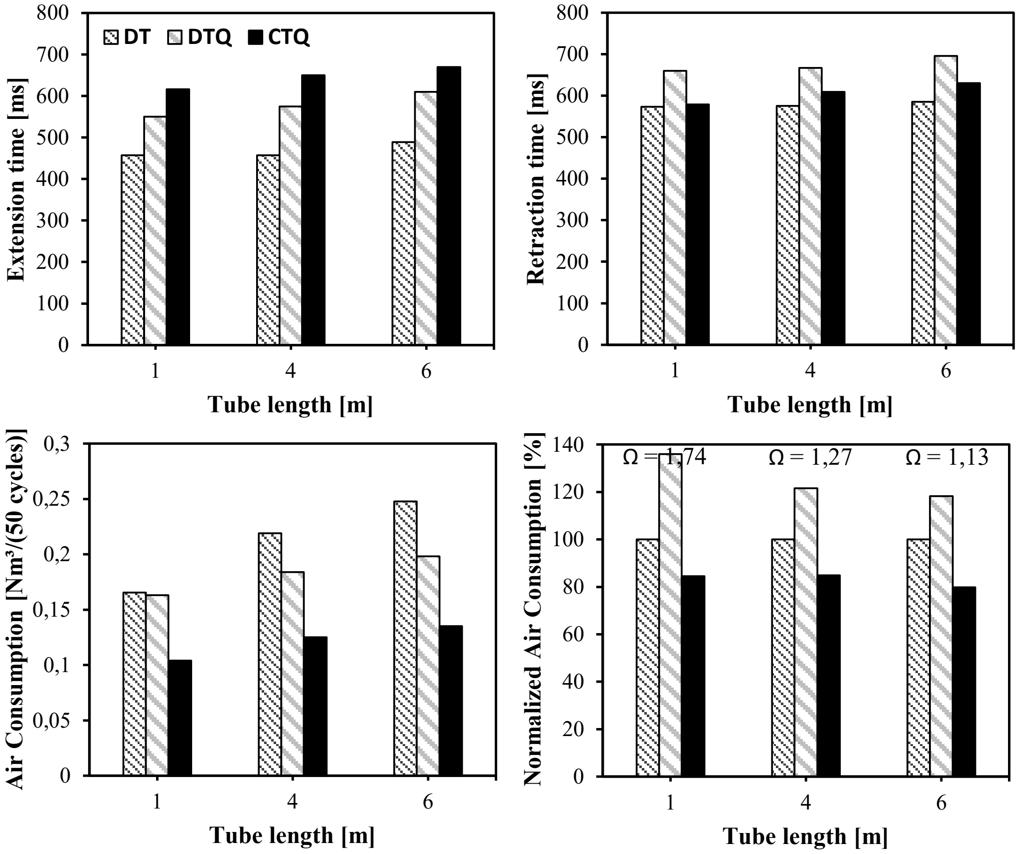


Figure 11: Measured results for Ø 32 x 320 with M = 10 kg

5. CONCLUSION AND OUTLOOK

The paper presents an experimental investigation of three different pneumatic circuits: a novel implementation of a combined throttling switching scheme, a state of the art downstream throttling with quick exhaust, and a conventional downstream throttled drive. Furthermore, a metric for comparing energy savings adjusted to changes in cycle time was introduced, allowing for an objective comparison between the systems. The results demonstrate a significant improvement in normalized air consumption with the novel circuit compared to conventional downstream throttling, although a high variation in performance was observed with different cylinder sizes. The results also indicate that the use of downstream throttling with quick exhaust can be beneficial in some applications, but its performance varies widely depending on the ratio of cylinder size to dead volumes in the lines, with the system achieving the highest normalized savings with smaller cylinders and longer lines.

In future work, the cost-effectiveness of the energy-saving circuits should be analysed. Additionally, the use of a shut-off function, implemented using a 5/3 valve in combination with the downstream throttle with quick exhaust, should be further investigated. Furthermore, exploring the performance of these systems with a broader range of cylinder sizes and moving masses could lead to a deeper understanding of the opportunities and limitations associated with the use of such circuits.

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NOMENCLATURE

A_p	Piston area	m^2
L	Cylinder stroke	m
L_{dead}	Dead volume expressed as cylinder stroke	m
M	Moving mass	kg
m_{air}	Air mass	Nm^3
p_s	Supply pressure	Pa
R	Specific ideal gas constant of air	J/(kgK)
t_{cycle}	Cycle time	s
t_f	Transfer time	s
T_0	Temperature at standard conditions	K
α	Cylinder area ratio	-
Ω	Pneumatic frequency ratio	-
PLC	Programmable Logic Controller	
DT	Downstream Throttling	
DTQ	Downstream Throttling with Quick-Exhaust	
CTQ	Combined Throttling with Quick-Exhaust	

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