Efficiency Improvement by Air Recuperation through the Use of Ejectors

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Abstract

Pneumatic systems are very common in industrial automation, because they feature good dynamic properties as well as a simple and flexible system setup. The main disadvantage of pneumatics compared to electrical systems is the effort required to achieve the same level of energy efficiency. Air recuperation to increase the efficiency of pneumatic systems usually requires a complex system setup, thus diminishing the advantage of pneumatics. In scope of this paper a newly developed system design is presented, which allows operating pneumatic systems in a virtually closed loop circuit. Thereby a complex circuitry is avoided and a flexible system layout with all its benefits is preserved. An optimization of the ejector which is used to recharge the closed loop circuit is presented. Furthermore, the potential energy savings of the new system design are approximated.

KEYWORDS: efficiency, air recuperation, ejector, optimization, CFD

1. Introduction

The automation of manufacturing processes is a key element of today's industrial production. To accomplish typical tasks, such as positioning, moving, grabbing and clamping, pneumatic solutions are often implemented. Compared to electrical solutions pneumatics feature good dynamic properties, a simple overload protection as well as a simple and cost effective system design, which makes pneumatics the first choice for a wide range of applications. The main disadvantage of pneumatic systems compared to electrical systems is the required effort to achieve the same level of energy efficiency in most applications. In a typical pneumatic system the prevailing losses can be traced back to the thermodynamic compression process. Therefore, discharging the compressed air into the environment, as usually done in typical pneumatic systems, is energetically unfavourable. Hence different approaches to a more energy conserving system setup have been developed /1/, /2/, /3/, /4/, /5/. The typical approach to increase the efficiency of pneumatic systems is the use of efficient compressors with variable speed as well as heat recuperation. Furthermore, the reduction of moving masses and dead volumes improves the system efficiency. This includes the right dimensioning and installation of pneumatic lines and actors as well as regular maintenance of all components to prevent the occurrence of leakage. The separation of currently inactive sub branches of large pneumatic systems from the pressure supply allows the reduction of leakage in the separated system parts. Further efficiency improvements usually require major circuit modifications or complex valve controls which need a deep understanding of the system behaviour. The complex valve control for example allows to separate the pressurized chamber of a linear actor from the pressure supply during motion and to utilize the expansion energy stored in the separated chamber on the remaining stroke. One of the rather simpler circuitry modifications is the implementation of compressed air networks with different pressure levels, realized by pressure control valves. This allows an adequate pressure supply for every connected actor based on the required forces. More complex modifications facilitate the temporary storage of energy and therefore allow the energy recuperation for following actuation cycles. All these circuit modifications unite the disadvantage of a significant increase in the number of needed system components and actuation signals of valves. To circumvent these disadvantages a new concept was developed at the Institute for Fluid Power Drives and Controls (IFAS) of RWTH-Aachen University. The concept allows operating pneumatic systems in a virtually closed loop circuit. Thereby a complex circuitry is avoided and a flexible system layout with all its benefits is preserved.

2. Operating pneumatics in a closed loop circuit

Operating pneumatics in a fully closed circuit would not be beneficial, because air losses through leakage or intended withdrawal must be compensated. To recharge the quasi closed loop circuit and compensate air losses, ejectors are utilized in the investigated case. **Figure 1** depicts a simple pneumatic system on the left and the same system extended with the ability to use air recuperation utilizing an ejector on the right. The velocity of the pneumatic actors is typically adjusted by meter-out controls utilizing exhaust air throttles. The throttle leads to a pressure build-up on the exhaust side of the meter-out controlled actor. In a conventional pneumatic system the energy stored in the pressurized air is completely wasted. The new concept utilizes this energy to raise the pressure level before the compressor.

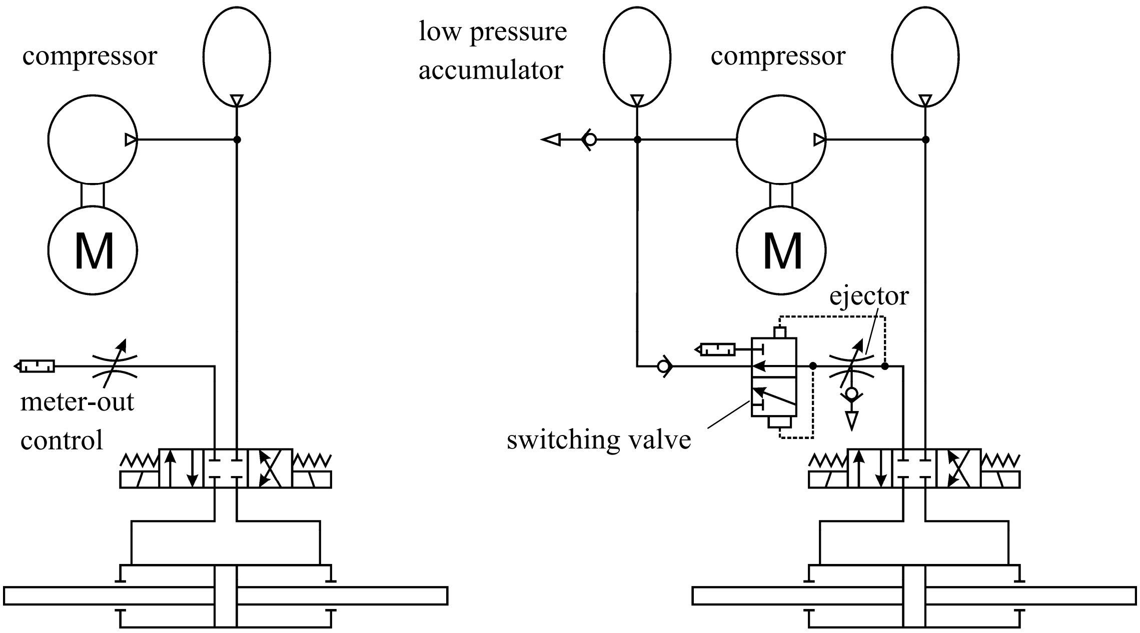

Figure 1: Comparison of a conventional pneumatic system (left) with the same system extended with air recuperation (right)

The activated ejectors are capable to feed a mass flow from the environment to the quasi closed circuit with an elevated pressure level. In this case the air compressor sucks in air from the low pressure accumulator with an elevated pressure level instead of the environment with ambient pressure. This reduces the energy consumption of the polytropic air compression to a constant pressure level in comparison to a conventional pneumatic system. If the mass losses would not be compensated the system efficiency would be reduced to the efficiency of a conventional system. Conventionally, the velocity of pneumatic actors is adjusted by meter-out controls, exhausting the air into the environment. The new concept with air recuperation, depicted in figure 1 (right), allows the recirculation of the compressed air into a low pressure accumulator, using the ejector instead of the exhaust air throttle. To preserve the motion characteristics of a conventional system with meter-out controls, a switching valve is used, which connects the ejector with the low pressure accumulator as long as the ejector is critically perfused. If the flow through the ejector reaches a non-critical condition, the switching valve connects the ejector exhaust port to the environment, which results in the flow properties of a conventional meter-out control. The switching valve is pressure actuated with an area ratio adjusted to the critical flow condition of the ejector motive flow nozzle. In order to keep the system layout with air recuperation as simple as possible, all further components, except the low pressure accumulator, are supposed to be integrated into a single assembly. This allows a simple implementation into new pneumatic systems and furthermore provides a possibility to upgrade existing systems.

3. Optimizing the ejector geometry

The ejector is a vital component, because the achievable pressure level and therefore the attainable efficiency improvements are directly related to the performance of the ejector. Hence, an optimization of the ejector geometry for high output pressures is conducted. **Figure 2** depicts the layout of a typical ejector.

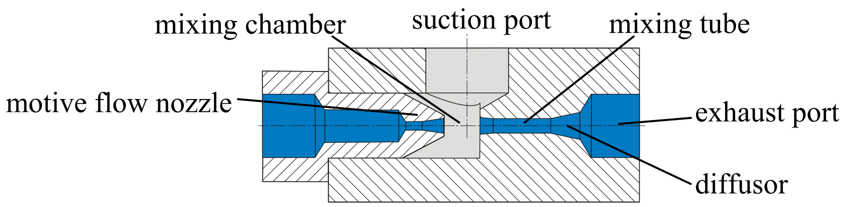

Figure 2: Ejector schematic

The flow from the meter out controlled actor is directed through the motive flow nozzle into the mixing chamber. If the ejector is critically perfused, the air flow reaches sonic speed in the smallest diameter of the motive flow nozzle. Downstream the flow is accelerated further in the divergent duct of the motive nozzle until the stream is expelled into the mixing chamber with supersonic speed. The high velocity of the motive stream creates a low pressure zone that draws in air from the suction port. The sucked in air is accelerated through the exchange of impulse with the motive stream. Further downstream in the mixing tube, when the velocity of the streams is equalized, the stream is decelerated in the diffuser. Decelerating the stream converts its high velocity in a higher static pressure at the exhaust port /6/. The geometry of a commercially available ejector serves as reference for the optimization process. Herein, first the motive flow nozzle is investigated.

3.1. Optimization of the motive flow nozzle

The motive flow nozzle directs the flow from the actor into the mixing chamber. The optimization objective is to maximize the thrust of the critically perfused motive flow nozzle to provide the mixing chamber with airflow of high kinetic energy. Furthermore fluidic restrictions must be considered. The opening angle of the motive flow nozzle should not exceed 12° /6/. Higher opening angles could cause a flow separation from the nozzle wall, which would lead to high energy losses. Therefore the optimization process can be reduced to identifying an optimal area ratio of smallest cross section to outlet cross section. **Figure 3** depicts the inner shape of an arbitrary motive flow nozzle with straight transitions and the characteristics of the Mach number *Ma* associated to the geometry as well as the area ratio A^*/A. The Mach number is calculated using equation 1.

$$\frac{A^*}{A} = \frac{Ma}{\left[\frac{2}{\kappa+1}\left(1+\frac{\kappa-1}{2}Ma^2\right)\right]^{\frac{\kappa+1}{2(\kappa-1)}}} \quad (1)$$

Figure 3 (right) depicts the Mach number characteristic associated to the geometry

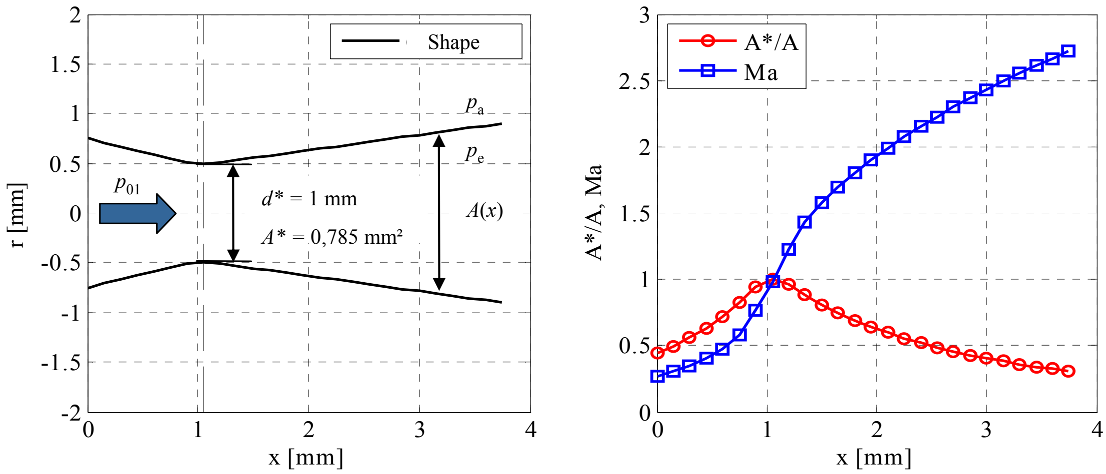

Figure 3: Inner shape of a motive flow nozzle (left) and the associated Mach number characteristics (right).

The thrust F_{Thrust} of the motive flow nozzle is depicted in equation 2 and will be used for the optimization.

$$F_{Thrust} = \dot{m} \cdot v_e + (p_e - p_a) \cdot A_e \quad (2)$$

The outlet pressure p_a is assumed to be 1 bar_{abs} because the mixing chamber is directly connected to the environment with atmospheric pressure. Assuming the flow is isentropic and air can be described as ideal gas the required flow variables can be calculated along the shape of the nozzle geometry based on the Mach number using equations 3 through 5.

$$\frac{p}{p_0} = \left(1+\frac{\kappa-1}{2}Ma^2\right)^{-\frac{\kappa}{\kappa-1}} \quad (3)$$

$$\frac{T}{T_0} = \left(1+\frac{\kappa-1}{2}Ma^2\right)^{-1} \quad (4)$$

$$\frac{c}{c_0} = \left(1+\frac{\kappa-1}{2}Ma^2\right)^{-\frac{1}{2}} \quad (5)$$

The characteristics of the flow variables along the geometry are shown in **figure 4**. With known flow variables the mass flow rate and velocity in equation 2 can be replaced by the calculated flow variables, whereby the thrust F_{Thrust} can be expressed along the geometry as a function of the input variables and the geometry itself. Furthermore, equation 6 shows that the thrust is not temperature dependent.

$$F_{Thrust} = A^* \cdot p_0 \cdot \gamma \cdot \left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}} \cdot Ma(x) \cdot \sqrt{\frac{\gamma+1}{2+(\gamma-1)\cdot Ma(x)^2}} + \left[p_0 \cdot \left(1+\frac{\gamma-1}{2}Ma(x)^2\right)^{-\frac{\gamma}{\gamma-1}} - p_a\right] \cdot A(x) \tag{6}$$

By the use of equation 6 the thrust along the geometry can be calculated. Results are depicted in figure 4 for different input pressures p_{01}.

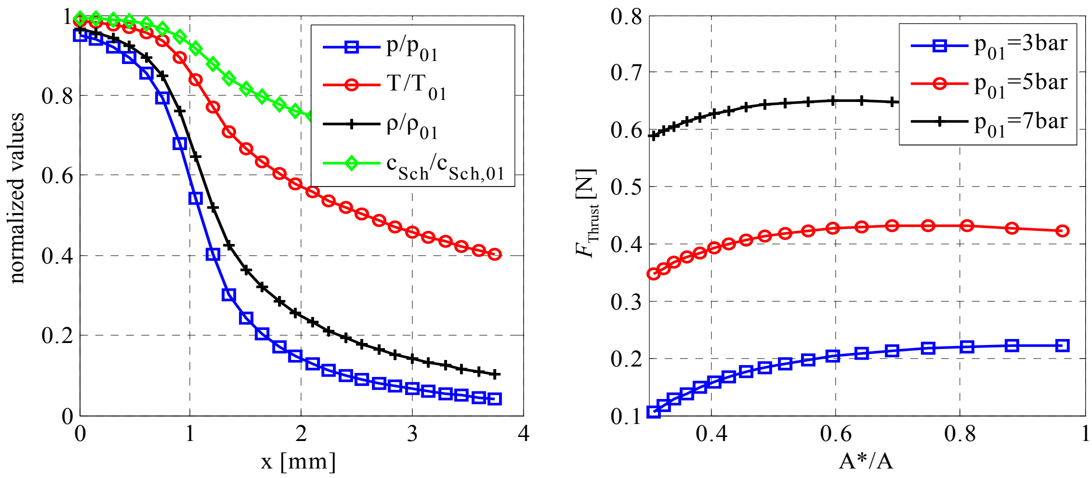

Figure 4: Flow parameters along the ejector geometry.

The maximum thrust at different pressures correlates with the area ratio A^*/A according to figure 4. Hence an optimal value for the area ratio of the smallest cross section to outlet area can be found. **Figure 5** depicts the characteristic of the optimal area ratio A^*/A_e corresponding to the input pressure p_{01}. With increasing input pressure the optimal area ratio decreases.

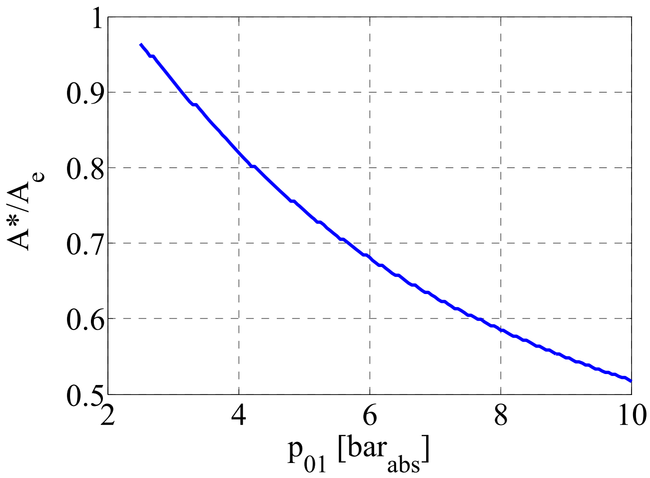

Figure 5: Optimal area ratio of motive flow nozzle over input pressure p_{01}.

Figure 6 shows the comparison of the reference motive nozzle geometry with the thrust optimized motive nozzle. The CFD-simulation of the reference geometry (left) shows an early flow separation from the divergent nozzle wall.

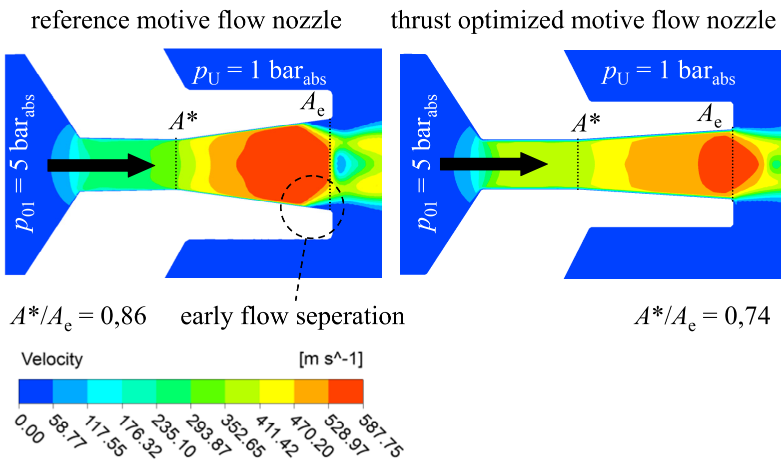

Figure 6: CFD simulation of the motive flow nozzle.

The over expanded nozzle flow is the result of shocks within the divergent nozzle section, which involve energy losses /7/. In contrast to the reference geometry the CFD-simulation of the thrust optimized nozzle shows no early flow separation. Hence energy consuming shocks within the divergent part of the nozzle are prevented.

3.2. Optimization of the ejector mixing chamber and diffusor

A very time efficient way to optimize the ejector geometry is the use of an analytical ejector model. This would allow many simulations in a short period of time and additionally the use of well established numerical optimization routines. The disadvantages of available analytical ejector models are the required simplifications to achieve a reasonably simple mathematical model. These simplifications imply that the analytical models are restricted to a certain set of design parameters. To circumvent these limitations CFD-simulations are used to perform the optimization. Ansys 13 provides a goal driven optimization toolbox, which is used to optimize the ejector geometry. Because every additional design parameter raises the required simulation time significantly, the motive flow nozzle is optimized separately as described in **chapter 3.1**. The optimization process uses the reference geometry as starting point. The flow properties are chosen according **table 1**.

flow parameter for geometry optimization	
static pressure at driving nozzle	p_1 = 5 bar_{abs}
static pressure at ejector outlet	p_2 = 1,5 bar_{abs}
total pressure at suction port	p_U = 1,0 bar_{abs}

Table 1: Flow parameter for ejector geometry optimization

The goal of the optimization is to find an ejector geometry, which maximises the achievable mass flow ratio of sucked in mass flow from the environment to the driving mass flow under the given flow parameters. Every parameter variation requires a full simulation run, which makes the optimization process very time consuming. Therefore the CFD-simulation is performed on a 2-dimensional mesh, representing a circular sector of 6° of the ejector geometry. **Figure 7** shows the forward section of the optimized geometry including the mesh and the geometry parameters used for the optimization.

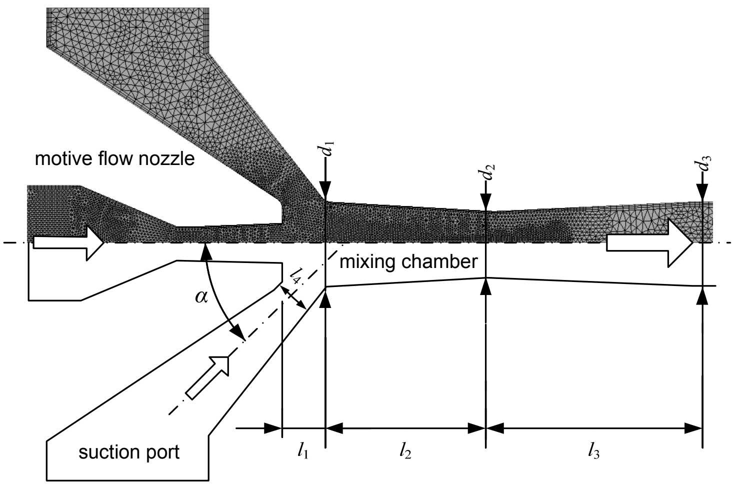

Figure 7: Forward section of the optimized ejector geometry, including the mesh and design parameters

Simulation results with the 2-dimensional mesh show good agreement with those of the 3-dimensional mesh considering that asymmetry caused by the suction port cannot be modelled.

3.3. Performance of the ejector with optimized geometry compared to the reference design

The verification of the optimization results requires comparing the mass flow characteristics of the optimized geometry with the reference geometry, because CFD-simulations always come with an uncertainty. **Figure 8** depicts the test rig used to evaluate the ejector performance. The pressure p_1 is adjusted by an adjustable throttle. The outlet pressure p_2 of the ejector is controlled by a pneumatic servo valve.

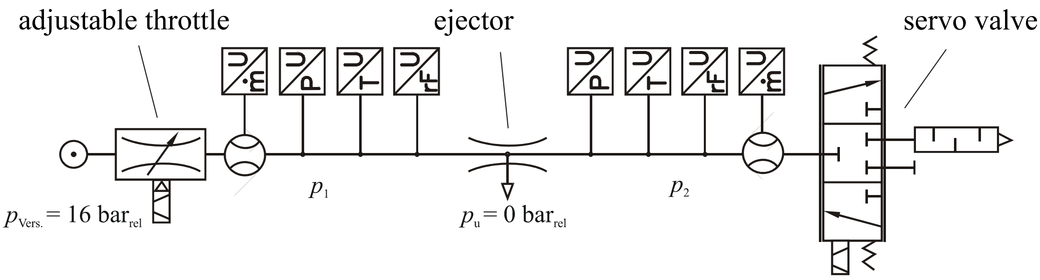

Figure 8: Test rig

Figure 9 depicts a characteristics diagram for the mass flow ratio as a function of inlet and outlet pressure to evaluate the ejector performance in different operating points.

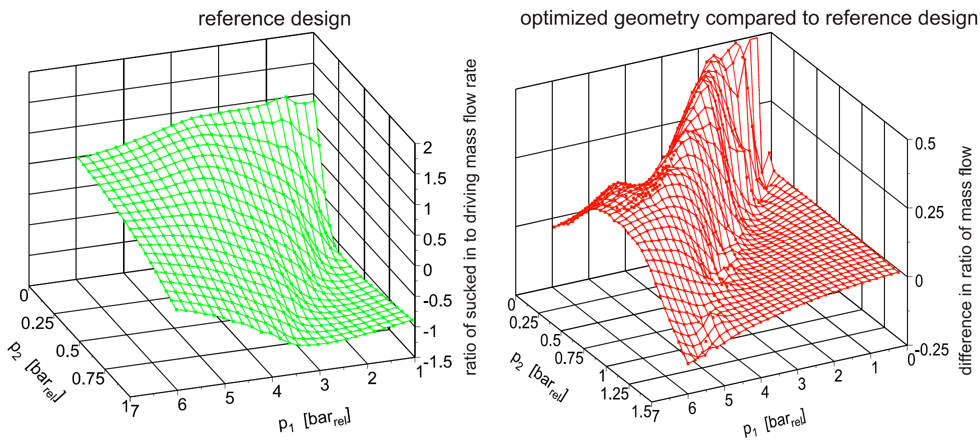

Figure 9: Mass flow characteristics diagram

The difference in mass flow ratio of the optimized geometry to the reference geometry reveals the improvements of flow ratio over the full pressure range of motive flow and exhaust port pressure.

4. Potential energy savings

Based on the measured characteristics diagram a one-dimensional simulation of the ejector assembly is initiated. The potential energy savings are assessed for a pressure compensation operation. An air volume of 8 l at 8 bar is expanded over the ejector assembly against a constant pressure level of 1.3 bar, depicted in **figure 10**. During the suction period air is sucked in from the environment over the suction port. In this time the motive flow nozzle is critically perfused and a mass flow is feed to the pneumatic system. In the conduction period the suction port is closed, so that air coming from the critically perfused motive flow nozzle is conducted through the ejector to the exhaust port. As soon as the motive flow nozzle reaches non-critical flow condition, the ejector valve connects the outlet port to the environment. In this operating condition the pneumatic system discharges the mass flow of the motive flow nozzle into the environment.

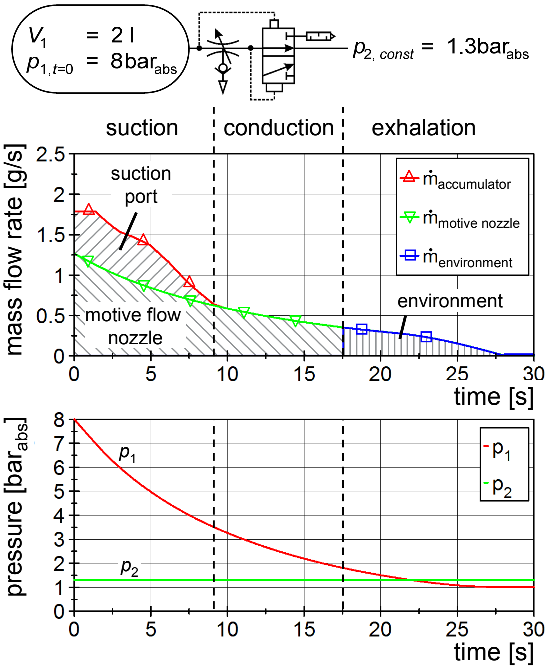

Figure 10: Exemplary case of pressure compensation over the ejector

The ratio of mass flow expelled into the environment to mass flow sucked in from the environment must be less than one in any case to compensate loss of leakage and achieve an increase in efficiency. Integrating the mass flow rates delivers the absolute masses in equation 7.

$$
\begin{aligned}
m_{\text{accumulator}} &= \int_{t=0}^{t} \dot{m}_{\text{accumulator}} \cdot dt = 16.33g \\
m_{\text{motive flow nozzle}} &= 14.42g \\
m_{\text{suction port}} &= 4.20g \\
m_{\text{environment}} &= 2.29g
\end{aligned}
\qquad (7)
$$

This results in a mass gain of 13.2 % based on the motive flow. Hereby must be considered that the pressure compensation is adverse to energy recuperation. Using the ejector as meter-out control is expected to expand the suction period. Therefore higher mass flow to the system or increased outlet pressures can be expected. The specific energy needed for the polytropic compression with $n = 1.3$ is depicted in equation (8). The increase in efficiency at different input pressure levels and a constant outlet pressure of 1.3 bar, used in the example of figure 10, is depicted in equation (9).

$$w_{Comp} = n \cdot \frac{p_1 \cdot v_1}{n-1}\left(\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right) = n \cdot R \cdot T_1 \cdot \left(\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right) \tag{8}$$

$$\Delta\eta = 1 - \frac{w_{Comp,p_1=1,3bar}}{w_{Comp,p_1=1,0bar}} = 1 - \frac{\left(\frac{8\,bar}{1.3\,bar}\right)^{\frac{1.3-1}{1.3}} - 1}{\left(\frac{8\,bar}{1\,bar}\right)^{\frac{1.3-1}{1.3}} - 1} = 1 - 0.85 = 0.15 \tag{9}$$

The required energy intake is reduced by $\Delta\eta = 15$ % based on the increased input pressure upstream of the compressor.

5. Conclusion

This article addresses the energy efficiency of standard pneumatic systems with meter-out controls. An approach to operate these pneumatic systems in a quasi closed loop circuit is presented. Instead of the exhaust air throttle an ejector is utilized to feed a mass flow to the quasi closed loop. Furthermore the approach allows compensating inevitable leakage and intended withdrawal. To maximize the energy efficiency the stream geometry of the ejector is optimized. An analytical approach to optimize the motive flow nozzle is presented. To optimize the ejector geometry an automated screening process based on 2-dimensional CFD-simulations is utilized in Ansys. Measurements show an improved ejector performance of the optimized geometry compared to the reference geometry used as starting point for the optimization process. Based on the measured ejector characteristics the achievable energy savings are approximated.

6. Literature

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7. Symbols

A	flow cross-section	m²
A^*	smallest flow cross-section	m²
Ma	mach number	[-]
κ	polytrophic exponent	[-]
p	static pressure	bar
p_0	total pressure	bar
v	velocity	m/s
T	Temperature	K
m	mass	g
$\dot{m}$	mass flow rate	g/s