A Novel Approach for Pneumatic Pressure Booster

Fedor Nazarov, Elvira Rakova, Jürgen Weber, Amir Rafaee Vardini

Dresden University of Technology, Institute of Fluid Power (IFD), Helmholtzstr. 7a, D-01062 Dresden, Germany
E-Mail: fedor.nazarov@tu-dresden.de

Pneumatic pressure boosters are widely applied in handling systems to increase the network pressure. Although they may enable a considerable energy saving for the entire pneumatic system, there is still a large potential for performance improvement. However, the boosting technologies in other domains, as the electrical DC-to-DC converters, present high efficiency. In the given study transferability of electrical DC-to-DC converters into pneumatics was investigated and the potentials of new circuits were researched. Based on the lumped parameters simulation results the most prospective concepts were identified using the three criteria: maximal pressure gain, energy efficiency, and mass flow rate. The prototypes were implemented on a test rig to verify the simulation results and to compare them with each other.

**Keywords:** Pressure booster, Pneumatics, Efficiency

**Target audience:** Pneumatic Components, Pneumatic Systems

1 Introduction

Pneumatic systems are widely used in a modern automation technology to fulfill different motion tasks. They distinguish themselves through their robust and flexible design as well as low investment costs. However, compressed air is still a source of high energy consumption. In general, pneumatic drive technology demands about 7% of an overall German industrial electricity expenditure, which is more than 17 billion kilowatt-hours /1/. With regards to an energy consumption of pneumatic systems, high energy costs calculated through a long-running time raise a question of efficiency.

The energy consumption of pneumatic systems depends directly on the pressure level and the volume flow rate. The supply pressure of each actuator is defined by the technological process itself, and its peak value determines the pressure requested from the compressor plant. However, not all components require the maximum pressure level. For that purpose, it can be adjusted using pressure regulators. At the same time, energy loss occurring during the gas compression is the greater, the higher the gained pressure is. In case of an oversized drive, system pressure reduction results in energy saving of about 10% per 1 bar. Furthermore, the gas leakages that alone may cause compressed air loss up to 30%, also increase proportionally with the system pressure /1/.

Therefore, to reduce an overall energy consumption, the supply pressure must be decreased up to the level needed by the major actuators. To meet the demands of high-pressure actuators pneumatics boosters are used to increase the network pressure locally using only pneumatic power

2 State of the Art

There are several booster technologies available on the market /2-4/. They implement a similar pressure multiplier approach based on a summation of piston areas to enhance the pressure at the expense of the mass flow loss. The output pressure level depends mainly on the total area difference. An operating principle of pressure multiplier is shown in Figure 1.

![Figure 1: Pneumatic pressure multiplier: 1, 4 – driving chambers; 2, 3 – boosting chambers; \( p_1 \) – supply pressure; \( p_{PR} \) – pressure adjusted by the pressure regulator \( PR \); \( p_b \) – booster output pressure; \( DV \) – directional valve.](image)

Presented pressure multiplier consists of two equal cylinders in a shared shell and two pistons coupled by a shaft. Therefore, there are four chambers that may have equal or various areas \( A_1 \), \( A_2 \), \( A_3 \), \( A_4 \) as defined in Figure 1. The directional valve \( DV \) connects one of the driving chambers 1 or 4 with a source of compressed air under the pressure \( p_1 \) adjusted by the pressure regulator \( PR \) up to \( p_{PR} \). In an ideal case, i.e., without energy loss, a pressure gain \( N \) is constant for the given \( p_1 \). However, any output pressure in the range \( p_1 < p_b < p_{b,\text{max}} \) is achievable when reducing the supply pressure means of \( PR \), where \( p_b \) is an absolute output pressure and \( p_{b,\text{max}} \) is a maximal output pressure determined by the gain factor \( N \):

\[
N = \frac{p_{b,\text{max}}}{p_1} = \frac{A_2 + A_4}{A_3} \cdot \frac{p_{PR} \cdot A_1}{A_3 \cdot p_1}
\]

(1)

where \( p_b \) is an ambient pressure. Boosting chambers 2 and 3 have a permanent pressure supply through the check valves directly from the network. Assuming the both cylinders are equal, hence \( A_1 = A_4 \) and \( A_2 = A_3 \), a driving force appears due to the pressure difference in the driving chambers (in the depicted booster: \( p_{\text{Chamber1}} = p_1 \) and \( p_{\text{Chamber4}} = p_{PR} \)) and piston moves up to the end position and compresses the air in the adjacent boosting chamber (chamber 3 in the given case). When the end position is reached, a part of the compressed air is used to switch the valve \( DV \) and the process repeats until the output pressure reaches \( p_{b,\text{max}} = p_{PR} \) and the forces on the pistons are balanced.

There is a great variety of pressure booster constructions distinct in the number of chambers, area difference, opening pressure levels, an art of the directional valve control and so on. One of the disadvantages of such boosters is a high amount of exhaust air loss. The described operating principle presupposes at least 50% input air loss caused by the driving chamber depressurization at every stroke directly in the atmosphere. On the other hand, pressure multipliers are easy to use and simple to manufacture as well as they can to work stand-alone because no additional electrical connections are required.

All in all, the commercially available boosters permit to gain the input pressure up to 4 times and even higher, but they are not predestined for a long continuous work and demonstrate a moderate efficiency of about 40-50%. However, the use of the pressure boosters promises a vast potential for performance improvement of pneumatic systems /1/.

Although a number of researches is already dedicated to the efficiency improvement, only the above-described operating principle is discussed in all the studies as a basis for some new modifications. For example, Y. Fan et al. propose a booster with an energy recovery implemented using two additional driving chambers /5/. This results in a boost ratio increase of 15-25% and efficiency improvement of 5-10% compared to the conventional model. S. Liu et al. investigated a booster that uses an air expansion energy with the efficiency improvement up to 10% /6, 7/. However, those concepts are complex.
The manufactures mainly limit themselves to some certain constructive solutions as dead volumes reduction or performing the seal materials /2-4/. Any further efficiency improvements of a pressure multiplier are bounded by physics of the operating principle. Therefore, the further development of boosters should be done by investigating a new work principals. Having said that, boosting principles in other domains like hydraulics and electrics show high performance level. In the meanwhile boosting technologies in other domains, for example, DC-to-DC converters in the electrical engineering, perform both high efficiency and robustness. Electrical DC-to-DC converters present a vast variety of circuits implementing different ways to gain a higher voltage. Moreover, some studies show the possibility of implementation of electrical amplifier concept in hydraulics /8/. Development of novel boosting concepts for pneumatic inspired by the electric amplifying principals promises a significant potential for the improvement of system performance.

3 Analogy Approach

In this case, the aim of the study is a pneumatic booster concepts development based on the electrical amplifying circuits and analogies between pneumatic and electrical components.

The analogy approach bases on the similarities of fundamental laws in the diverse domains, although the physical principals often show different nature. Among the analogies between mechanics, hydraulics and electronics, the theory was also developed for low-pressure pneumatic logic components application. This theory aimed at simplifying a fluidic circuits design for communication and computation technology using the available electronic circuits /9, 10/.

Some aspects of this theory were applied in the current study as theoretical fundamentals. For example, to describe electrical and pneumatic circuits, flow and potential quantities should be defined in each domain. Current $I$ in electronics and mass flow $m$ in pneumatics represent therefore the flow quantities and electrical potential $U$ and pressure $p$ character the potential quantities. Flow and potential quantities are subjected to the similar rules in both domains, such as a Kirchhoff’s circuit law in electronics or a mass balance in pneumatics, positing that the algebraic sum of all flow quantities meeting at a network junction point is zero, so $\sum I = 0$ or $\sum m = 0$.

A closer look at some important equations, which describe the transient wave propagation (2), (3), for the distance $dx$ in an electrical conductor of length $l$ using relations between the flow and potential quantities and conductor properties, such as resistances $R$, capacities $C$ and inductivities $L$, shows similarities with the flow equations in pneumatics (4), (5), (9), (10).

\[
\frac{dU}{dx} + \frac{R}{l} \frac{dI}{dx} + \frac{1}{C} \frac{dI}{dt} = 0, \tag{2}
\]

\[
\frac{dI}{dx} + \frac{C}{l} \frac{dI}{dt} = 0, \tag{3}
\]

\[
\frac{dp}{dx} + \frac{R_p}{l} \frac{dp}{dx} + \frac{L_p}{l} \frac{dI}{dt} = 0, \tag{4}
\]

\[
\frac{dm}{dx} + \frac{C_p}{l} \frac{dp}{dt} = 0. \tag{5}
\]

In case of pneumatics, resistance $R_p$, capacities $C_p$, $C_v$ and inductivities $L_p$, $L_v$ are determined as shown in Table 1. Several basic pneumatic elements may be defined as net resistances (for example, a throttle or a nozzle) or net capacities (gas receiver, dead volumes). In contrast, another components, such as pneumatic cylinders, embody a complicated system of constant $C_p$ and variable $C_v$ capacities, resistances $R_p$ as well as pneumatic $l_p$ and mechanical $l_v$ inductivities.

Some other mostly used electronic components are also transferable into pneumatics: electrical switch fulfills the same function as a directional valve; diode represents the functionality of a check valve and pressure may be converted by multiplicator in about the same way as a voltage by means of transformer (Table 1).

<table>
<thead>
<tr>
<th>Components</th>
<th>Electronics</th>
<th>Pneumatics</th>
</tr>
</thead>
<tbody>
<tr>
<td>Illustration</td>
<td>$R = \frac{\Delta U}{I}$</td>
<td>$R_p = \frac{\Delta p}{\Delta m}$</td>
</tr>
<tr>
<td>Properties</td>
<td>$I = C \frac{dU}{dt}$</td>
<td>$\frac{dU}{dt} + \frac{1}{C_v} = \frac{V}{c_v}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$C_v = \frac{AV}{c_v}$</td>
</tr>
<tr>
<td>Inductivity</td>
<td>$U = L \frac{dI}{dt}$</td>
<td>$dI = \frac{L_p}{l_p} \frac{dp}{dt}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$L_p = \frac{ML}{\frac{dp}{dt}}$</td>
</tr>
<tr>
<td>Switch</td>
<td>$I = \begin{cases} 1 &amp; \text{if } x = 0, \ 0 &amp; \text{if } x = 0 \end{cases}$</td>
<td>$I = \begin{cases} 1 &amp; \text{if } x = 0, \ 0 &amp; \text{if } x = 0 \end{cases}$</td>
</tr>
<tr>
<td>Diode</td>
<td>$U_1 &lt; U_2$</td>
<td>$p_1 &lt; p_2$</td>
</tr>
<tr>
<td>Transformer</td>
<td>$N_1 \frac{U_1}{U_2}$</td>
<td>$A_1 \frac{p_1}{p_2}$</td>
</tr>
</tbody>
</table>

Table 1: Basic electronic components and their pneumatic analogues /9, 10/.

However, there are some restrictions bounding the applicability of these analogies to pneumatics. So, a pneumatic resistance $R_p$ remains constant only by the laminar flow, which occurs in pneumatics very seldom due to the low air viscosity and relative high stream speeds. In case of a critical flow, $R_p$ depends on the gas temperatures, and for the subcritical flow, it is also a function of pressures and critical pressure ratio. Another aspect is an absence of negative absolute pressure; therefore pneumatic capacities possess only one pole. Besides that, low air density restricts the influence of pneumatic inductivity $L_p$. Thus, the inductive component in (4) is usually neglected against the resistive one. As an alternative, inductivity effects in a pneumatic system can be reached by use of mechanical inertial load, which inductivity $L_{in}$ is usually factor 10^3-10^5 higher than the pneumatic one.

Depending on the working principle electrical converters are divided into capacitive, inductive and resonant /11/. Three basic DC-to-DC converter circuits representing each principle were chosen for the further transfer into pneumatics as to be seen in Table 2. Applying the previous specifics and boundaries, functionally similar pneumatic networks for new booster concepts were derived from the electrical circuits. A pressure supply implies a voltage source, an exhaust pressure – a ground and a receiver pressure – an output voltage.

The charge pump (Table 3) uses electrical switches $S_1$ and $S_2$ to connect the plates of capacitor $C_p$ in series and in parallel with a supply and a load within the two cycles. Depending on the number of stages and the control of switches, the charge pump can double, triple, halve and also invert voltages.
Flyback converter uses a transformer to store the energy in the form of electromagnetic field. When closing the switch S the primary winding of the transformer is connected to the input voltage and magnetic flux in the transformer increases. The diode is blocked and the output capacitor C_out supplies the load. After opening the switch S primary current and magnetic flux weaken and the secondary voltage is positive, allowing current to flow from the transformer through the diode. The energy from the transformer core recharges the capacitor again.

\[ \zeta = \frac{E_{\text{out}}}{E_{\text{in}}} \]  

Several assumptions were made to simplify the simulation models. Due to the air compression in cylinder chambers and the receiver, both pressure and temperature rise. That leads to intensive heat exchange through a cylinder wall with ambient at a lower temperature. To consider this effect in the model, the process has been assumed as near to isothermal with further calculation of external heat flow \( P_{\text{b}} \) according to the equation

\[ P_{\text{b}} = C_{\text{w}} \cdot A_{\text{wall}} \cdot (T - T_{\text{wall}}) \]  

where \( C_{\text{w}} \) is a heat transfer coefficient, \( A_{\text{wall}} \) – surface area, \( T \) and \( T_{\text{wall}} \) – gas and wall temperatures.

Further, the friction effects influence sufficiently the cylinder piston dynamics and have to be considered in the model. Due to the complex nature of friction forces, there is no universal model able to describe it with a high accuracy and by different operating conditions. However, the Striebeck model is commonly used to give a comprehensive picture of friction effects /14/.

\[ F_{\text{fr}} = (F_{\text{fr}} - F_{\text{C}}) \cdot e^{-\frac{T}{T_{\text{fr}}}} + k_{\text{fr}} \cdot |x|^{1.5} + F_{\text{C}} + k_{\text{fr}} \cdot \Delta p \]  

where the first term is a mixed friction, characterized by Striebeck force \( F_{\text{C}} \), Coulomb forces \( F_{\text{C}} \), piston velocity \( \dot{x} \) and Striebeck velocity \( T_{\text{fr}} \). The second term is a hydrodynamic or viscous friction force obtained from the piston velocity and parameters \( k_{\text{fr}} \) and \( \Delta p \).

The last value is a pressure depended term as a product of pressure difference \( \Delta p \) in the cylinder chambers and a sealing-specific constant \( k_{\text{fr}} \).

### 4.1 Charge pump

An electrical charge pump is a capacitive converter. Due to the specific noticed in the previous chapter, pneumatic capacity possess only one pole for the reason that no negative pressure, i.e. below the absolute vacuum, exists. Although this complicates a pneumatic circuit design using a direct transfer from the electronics, a pneumatic network for a charge pump was implemented based on a combination of constant and variable capacities as shown in Figure 2. The principle of the charge pump is based on the volumes summation of simultaneously working cylinder. The simulated pressure increase in the middle capacity \( p_{\text{m}} \) and the receiver \( p_{\text{r}} \) as well as an exergy efficiency \( \zeta \) for a two-stage pneumatic charge pump, as shown in Figure 2. The single-acting cylinders with a stroke \( 50 \text{ mm} \), piston diameter \( d = 16 \text{ mm} \) and shaft diameter \( d_{\text{a}} = 8 \text{ mm} \) were used.

A simple structure of this booster allows a modular units application, each consisting of a 3/2 directional valve, single-acting cylinder, capacity and two check valves, to increase the output pressure up to the required level. The disadvantage of a multistage architecture is a sufficient exhaust air loss. In the given case a two-stage construction results in an energy loss of about 88 % that makes a pneumatic charge pump energetically unfavorable.

![Figure 2: Pressure increase in the intermediate capacity (p_m) and in the receiver (p_r) as well as an exergy efficiency simulated for a pneumatic charge pump](image)

Another drawback of a considered circuit is that the pressure increase in the receiver is a result of a multiplying effect due to the cylinder areas difference and not the result of a merely capacitive principle as in an electric charge
pump. The reason for this is a unipolarity of pneumatic capacitances, whereas it seems to be impossible to charge a pneumatic capacity up to the pressure, higher than the input one, on the contrary to the electric circuits, where the potential difference between both capacitor plates is used.

4.2 Flyback booster

A pneumatic flyback booster implements a pressure multiplier principle that is currently used by the pneumatic booster manufacturers. The action has been considered in details in Chapter 2. The simulation model implies dimensions of a serial booster with a stroke \( h = 50 \text{ mm} \), cylinder diameter \( d = 40 \text{ mm} \) and shaft diameter \( d_s = 8 \text{ mm} \). Figure 3 depicts simulation results for receiver pressure and energy efficiency response.

![Figure 3: Receiver pressure and energy efficiency simulated for a pneumatic flyback booster](image)

As mentioned above, at least 50% air loss causes a moderate energy efficiency in comparison with the original electrical converter. According to (1), driving chamber area increase and boosting chamber area reduction (Figure 1) enhance the maximal output pressure but also causes a lower efficiency due to the higher exhaust loss.

4.3 Resonant converter

Unlike a charge pump, a resonant booster retains an original electronic converter operating principle. For this purpose four 2/2 valves are connected in a bridge circuit (Figure 4, A) to control the pressure in the cylinder chambers and to keep the piston oscillating in a parametric resonance with an eigenfrequency:

\[
f = \frac{1}{2\pi \sqrt{L_m C_x}},
\]

where \( C_x \) includes both constant dead volume capacities \( C_{\text{dead}} \) and a variable capacity \( C_{\text{var}} \) representing the cylinder chamber. As mentioned in Chapter 2, a pneumatic inductivity \( L_m \) is negligible in comparison with the mechanical one, thus only \( L_m \) is considered for the eigenfrequency and oscillation period \( T = 2\pi f \) calculations.

A parametric resonance occurs in a damped oscillating system if the periodic energy supply, called modulation, is provided to compensate for the damping energy dissipation through the damping \( \xi \). In electrical systems parametric resonance can be achieved by changing the capacitor capacitance or coil inductivity. Equivalently in pneumatics parametric resonant oscillations may be initiated by changing pneumatic inductivity or capacity. In this case, either a mass load or a cylinder capacity needs to be affected in the process. Changing the mass load of a linear oscillating system is inconvenient because additional mechanical parts are required, whereas cylinder chamber capacities can be easily adjusted by varying a pneumatic spring stiffness \( s \) (see Table 1). In the given case fast-switching 2/2 valves are applied to control the cylinder pressure and, therefore, the pneumatic cushion stiffness by inflating and deflating the cylinder chambers:

\[
s = -\frac{dp \cdot A^2}{dv}.
\]

In the system to be considered energy outflow characterizes a damping and energy inflow into the booster – modulation. Energy outflow from the cylinder is divided into the energy loss and exergy, flowing to the receiver. Energy loss occurs mainly due to the friction in cylinder seals, exhaust air loss and entropy loss. Both values, damping and modulation, are controlled by the valves V1…V4, Figure 4, A.

A resonant booster operating principle, illustrated in Figure 4, B, proposes a double energy conversion. By operation of valves V1 and V3 the supply air energy is converted into the kinetic energy of a piston with the mass load. At the moment \( W_1 \) valve V1 closes to reduce an input air consumption. During the time \( W_1 \)... \( W_2 \) air expands in the chamber A and pressure \( p_A \) decreases, while the piston still accelerates because of \( p_B > p_A \) (see Figure 4, C). At the moment \( W_2 \) valve V3 closes, so that chamber B volume grows smaller and pressure \( p_B \) rises and decelerates the piston. Then \( p_B \) exceeds receiver pressure \( p_R > p_B \), check valve opens, and the compressed air is pushed into the receiver until the piston completely loses its kinetic energy and stops. Therefore, a reverse energy conversion occurs in the time span \( W_2 \)... \( T/2 \), i.e., the kinetic energy of the piston and inertial load is transformed back to the internal energy of the air compressed up to a higher pressure \( p_B > p_R \). Then the same sequence repeats for the valves V2 and V4 on the return stroke.

![Figure 4: Resonant booster (with isobaric receiver): A – pneumatic circuit; B – Valves V1…V4 statements; C – Cylinder stroke, velocity, receiver and cylinder chambers pressure changes for piston moving rightwards](image)

Considering an input and output energy ratio, pulse width \( W_1 \) can be used to characterize the amount of energy supplied to the booster from the network:

\[
E_{\text{in}} = \int_{0}^{W_1} m_B R_T \frac{dP_A}{P_0} \cdot dt + \int_{0}^{T/2} m_B R_T \frac{dP_B}{P_0} \cdot dt = 2 R_T m_B \left( \frac{P_R}{P_0} - 1 \right) \int_{0}^{W_1} \frac{dP_A}{P_0} \cdot dt,
\]

and \( W_1 \) to characterize the amount of energy, delivered from booster to the receiver:

\[
E_{\text{out}} = \int_{0}^{W_1} m_B R_T \frac{dP_B}{P_0} \cdot dt + \int_{0}^{T/2} m_B R_T \frac{dP_A}{P_0} \cdot dt = \int_{W_1}^{T/2} m_B R_T \frac{dP_A}{P_0} \cdot dt.
\]

According to the equations (11) and (12) and variable capacity definition (see Table 1) the oscillating period \( T \) depends on the pressure in cylinder chambers. During the receiver filling process, the output pressure \( p_B \) and, therefore, the maximal pressure in the cylinder chambers increase, hence the cylinder eigenfrequency \( f \) increases too. To simplify the simulation model, period \( T \) was predefined and kept constant for each particular simulation process. Also for a better clarity duty factors, \( K_{\text{eff}} = 2W_1/T \) and \( K_{\text{ret}} = 2W_2/T \), were used instead of the pulse width \( W_1 \) and \( W_2 \).

The simulation model was parametrized with a foresight to a prototype design. A booster performance is determined by some parameters such as inertial mass \( M \), maximal cylinder stroke \( h \), duty factors \( K_{\text{eff}}, K_{\text{ret}} \) and
oscillation period $T$. This study considers a variation of $K_{df1}$, $K_{df2}$ and $T$, that are easily adjustable within the experiments. Their influence on the pressure ratio and efficiency was evaluated.

A presented simulation model of a resonant booster is strongly nonlinear, hence the optimum for the variable parameters cannot be calculated analytically. Therefore, a parameter variation was used within this study. The analytically calculated values of $T$ for estimated minimal and maximal pressures in the cylinder chambers were used as variation boundaries for the oscillation period. Duty factors $K_{df1}$ and $K_{df2}$ both are bounded between 0 and 1. Thus firstly the parameter variation within these constraints was attempted. Then the boundaries were narrowed, and the variation step was fixed. Figure 5 illustrates the results of a parameter variation for a pneumatic resonant booster with an optimal period $T = 0.35 \text{s}$, stroke $h = 400 \text{ mm}$, piston diameter $d = 18 \text{ mm}$ and inertial load $M = 4.3 \text{ kg}$ (consisting of $0.3 \text{ kg}$ cylinder piston mass and $4 \text{ kg}$ additional load). The aim was to determine a pressure $p_k$ and energy efficiency $\zeta$ for each parameter set when filling a receiver $V_k = 0.75 \text{ l}$ from the network pressure $p_n = 5 \text{ bar}$, within 20 s simulation time.

![Graph](image)

**Figure 5:** Field of characteristics of a resonant booster for oscillation period $T = 0.35 \text{s}$

It is evident that the gained pressure is the higher, the more exergy is conveyed to the booster. This corresponds to the significant duty factor $K_{df}$ values. High values of $K_{df}$ enable a more extended acceleration phase of mechanical inductivity and therefore, a higher momentum provides the pressure increase after closing the valve V3 or V4. Disadvantages of high $K_{df1}$ and $K_{df2}$ are a high supply air consumption and a low output mass flow that results in a low efficiency. And vice versa high efficiency is reached with low $K_{df1}$ and $K_{df2}$ caused by a lower compressed air consumption $h_{m, in}$ and higher air flow $h_{m, out}$ to the receiver. However, as an output pressure $p_k$ increases, the energy stored in the mechanical inductivity is not enough to keep the piston moving up to the end position and to push the entire compressed air mass out of the chamber against the back-pressure $p_h$. As a result, the piston oscillates in the middle of the cylinder and not delivering the air anymore. Thus the compromise between the gained pressure and efficiency must be found to settle the duty factors depending on particular booster application. Otherwise, the resonant booster performance can be illustrated by simulation for fixed $K_{df1}$ and $K_{df2}$ (Figure 6).

![Graph](image)

**Figure 6:** Pressure in the receiver and exergy efficiency simulated for a resonant booster; $K_{df1} = 0.4; K_{df2} = 0.45; T = 0.35 \text{s}$

As is seen from Figure 6, exergy efficiency decreases with increasing pressure. A longer filling time and therefore, a lower delivery in comparison with the flyback booster can be explained by a small cylinder diameter (only 18 mm to 40 mm for the flyback booster). Advantages of this concept are a theoretically unlimited pressure gain as well as a significant potential to enhance the efficiency by estimating not only optimal duty factors, but also an inertial mass and stroke.

5 Experimental Research

Since the resonant booster enables a high-pressure gain and the flyback booster theoretically possess relative high exergy efficiency, compared with the other concepts derived from the electronics, both boosters were chosen for the experimental research. Firstly, a resonant booster prototype, depicted in Figure 7, was built to prove its functionality and to study the influence of the duty factors experimentally.

![Graph](image)

**Figure 7:** Prototype of a resonant booster

For the prototype only standard components were used. Fast switch valves are controlled by the PLC accordingly to the Figure 4. B. Supply air flow rate, input pressure $p_i$ and output receiver pressure $p_r$ were measured to estimate the exergy efficiency.

Figure 8, A, shows measured and simulated receiver pressure responses. A maximum relative error of about 5 % within the simulation time $t_s = 180 \text{s}$ is caused foremost by a friction forces in a sealing band of a rodless cylinder, which was not considered in the simulation due to the lack of appropriate models but appreciably increases the friction force. Other reasons may be a rough evaluation of cylinder dead volumes as well as the leakages in the fittings and valve plugs. Faster pressure increase measured in the very beginning of the filling process may be caused by a temperature rise in the receiver due to a worse heat exchange with an environment as it was simulated.
Figure 8, B and C, show that both efficiency and pressure gain responses are inferior by low supply pressures because the friction force ratio related to the driving force grows higher with the decreasing supply pressure.

Due to the pressure-dependent cushion stiffness and therefore, variable cylinder eigenfrequency duty factor $K_{RF}$ was empirically determined and kept constant to prohibit the piston impact on the cylinder head. Thus, only $K_{RF}$ influence on the pressure and efficiency responses was studied, see the Figure 9.

The prototype was tested with a duty factor $K_{RF} = 0.5 \ldots 0.78$ and a constant $K_{RF} = 0.56$. According to the (13) the lower is $K_{RF}$, the higher is an efficiency but as mentioned before, the slower is a pressure increase. Thus, it takes the booster about 115 s to reach the pressure $p_R = 6 \text{ bar}$ with $K_{RF} = 0.5$ and only 60 s with $K_{RF} = 0.71$. Although the air consumption per air stroke is lower in the first case, the integral supply air mass for the whole filling process and thus the exergy efficiency is greater with a higher $K_{RF}$. Any further increase above about $K_{RF} = 0.71$ does not lead to a higher pressure response slope and only enhances the air consumption. Hence, when considering a filling of the receiver up to some defined pressure, an optimum for the duty factor can be found.

Figure 9: Receiver pressure increase and exergy efficiency characteristic ($\xi(p_R)$) for variable $K_{RF}$ values

While a pressure multiplier (flyback booster) is already implemented by pneumatics manufacturers, the simulated serial model was used as a reference for the resonant booster. The test setup and measurements were analog to the resonant booster experiments. As to be seen from Figure 10, although the exergy efficiency is about 12 % lower as was simulated (Figure 3) due to an exhaust air loss for the needs of a directional valve control (DV), Figure 1), it amounts about $\xi = 0.5$ by a pressure gain factor $N = 1.6$. A developed prototype of a resonant booster is able to increase the pressure with a comparable efficiency only with a gain factor $N = 1.2 \ldots 1.3$. However, the maximum gain factor for the standard boosters with equal cylinders usually does not exceed $N = 1.65$. The novel approach theoretically enables higher gain factors. Experimentally $N = 1.7$ was reached. A further performance increase needs a specific components application, such as low-friction sealing, cylinder head with reduced dead volumes and integrated check valves. The constructive parameters, such as cylinder length and mass load, should be comprehensively investigated to explore the concept limitations and further potentials.

Figure 10: Measured receiver and supply pressure and exergy efficiency for a pressure multiplier

6 Summary and Conclusion

Pneumatic boosters are used to increase the efficiency of compressed air systems. They have certain advantages and drawbacks. Constructive measures nowadays limit the further improvement of standard boosters. On the other hand, electric DC-to-DC boosters present a wide variety of circuits and functioning principles as well as a high robustness and efficiency. The study proposes a novel approach for pneumatic pressure booster design inspired by the electrical circuits. Applying the theory of electronic-pneumatic analogies, originally developed for a fluid-jet control systems design, some basic electrical converters were transferred into pneumatics. This transfer is confined by the physical differences between the domains, which are not acute for a low-pressure fluidics, but considerably bound the analogies applicability for conventional pneumatics.

For this reason, only two out of three electric circuits were successfully implemented in pneumatics, whereas one of them represents an already existing pressure multiplier further used as a reference. The novel concept, a pneumatic resonant booster, was investigated using the lumped parameters simulation. The parameter variation was applied to estimate the output pressure and exergy efficiency response depending on the control parameters. After validation an influence of control parameters and supply pressure on the booster performance was experimentally studied. A novel concept presents a comparable exergy efficiency with the reference in a restricted pressure gain ratio of about $N = 1.2 \ldots 1.3$, while a serial pressure multiplier enables $N = 1.6$ with an exergy efficiency of about $\xi = 0.5$. However, an advantage of the proposed concept is a higher maximal achievable pressure gain. Moreover, a resonant booster offers a greater potential for the further researches in the field of efficiency and pressure gain enhancement using construction improvement and study of alternative inducivity implementations in pneumatics.

7 Acknowledgements

The content of this article was developed within a project “Energieeffiziente Konzepte zur Druckwandlung in der Pneumatik” that was funded by the Fluid Power Research Fund of VDMA, the German Engineering Federation, FKM Nr. 703461.
### Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>Surface area</td>
<td>[m²]</td>
</tr>
<tr>
<td>$C$</td>
<td>Electrical capacity</td>
<td>[F]</td>
</tr>
<tr>
<td>$C_c$</td>
<td>Pneumatic constant capacity</td>
<td>[m³/s]</td>
</tr>
<tr>
<td>$C_v$</td>
<td>Pneumatic variable capacity</td>
<td>[m³/s]</td>
</tr>
<tr>
<td>$c$</td>
<td>Sonic speed</td>
<td>[m/s]</td>
</tr>
<tr>
<td>$\gamma_{Wall}$</td>
<td>Heat transfer coefficient of a cylinder or receiver wall</td>
<td>[W/(m²·K)]</td>
</tr>
<tr>
<td>$d$</td>
<td>Diameter</td>
<td>[m]</td>
</tr>
<tr>
<td>$E$</td>
<td>Exergy</td>
<td>[J]</td>
</tr>
<tr>
<td>$F$</td>
<td>Force</td>
<td>[N]</td>
</tr>
<tr>
<td>$f$</td>
<td>Frequency</td>
<td>[Hz]</td>
</tr>
<tr>
<td>$h$</td>
<td>Stroke</td>
<td>[m]</td>
</tr>
<tr>
<td>$I$</td>
<td>Electric current</td>
<td>[A]</td>
</tr>
<tr>
<td>$K$</td>
<td>Duty factor</td>
<td>[-]</td>
</tr>
<tr>
<td>$k_p$</td>
<td>Pressure depending friction coefficient</td>
<td>[m²]</td>
</tr>
<tr>
<td>$k_v$</td>
<td>Viscous friction coefficient</td>
<td>[kg/s]</td>
</tr>
<tr>
<td>$L$</td>
<td>Electrical inductivity</td>
<td>[H]</td>
</tr>
<tr>
<td>$L_p$</td>
<td>Pneumatic inductivity</td>
<td>[m³]</td>
</tr>
<tr>
<td>$I_m$</td>
<td>Mechanical inductivity</td>
<td>[m³]</td>
</tr>
<tr>
<td>$l$</td>
<td>Length</td>
<td>[m]</td>
</tr>
<tr>
<td>$M$</td>
<td>Load mass</td>
<td>[kg]</td>
</tr>
<tr>
<td>$m_{air}$</td>
<td>Air mass, air mass flow rate</td>
<td>[kg], [kg/s]</td>
</tr>
<tr>
<td>$N$</td>
<td>Pressure gain factor</td>
<td>[-]</td>
</tr>
<tr>
<td>$p_0$</td>
<td>Ambient pressure</td>
<td>[bar]</td>
</tr>
<tr>
<td>$p_s$</td>
<td>Supply Pressure</td>
<td>[bar]</td>
</tr>
<tr>
<td>$p_{max}$</td>
<td>Pressure adjusted by the pressure regulator</td>
<td>[bar]</td>
</tr>
<tr>
<td>$p_b$</td>
<td>Booster output or receiver pressure</td>
<td>[bar]</td>
</tr>
<tr>
<td>$Q$</td>
<td>Volume flow rate</td>
<td>[m³/s]</td>
</tr>
<tr>
<td>$R$</td>
<td>Electrical resistance,</td>
<td>[Ω]</td>
</tr>
<tr>
<td>$R$</td>
<td>Specific gas constant</td>
<td>[J/(kg·K)]</td>
</tr>
<tr>
<td>$R_p$</td>
<td>Pneumatic resistance</td>
<td>[m³/s]</td>
</tr>
<tr>
<td>$s$</td>
<td>Spring stiffness</td>
<td>[N/m]</td>
</tr>
<tr>
<td>$T$</td>
<td>Oscillation period</td>
<td>[Hz]</td>
</tr>
</tbody>
</table>

### References


