DEVELOPMENT OF RECIPROCATING AIR EXPANDER FOR μ - CAES TECHNOLOGY

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ABSTRACT

As renewable energy sources (RES), such as solar and wind, continue to grow in the energy mix, it becomes crucial to address their inflexibility and daily demand-production gap. Energy storage is vital, allowing storage during low demand and supplying power when needed. The authors investigated compressed air energy storage (CAES) as a mechanical solution, focusing on micro-CAES (μ-CAES) for smaller industries or housing estates. This research aimed at optimizing μ-CAES efficiency through modifications to the air expander's construction (piston parameters) and process parameters (air pressure, temperature). Initial results with a 3-cylinder expander showed an average power of 2.2 kW and 0.45 efficiency. Further enhancements, such as compressed air supply control and waste heat utilization during expansion expects to affect on increase of efficiency to 0.8.

Keywords: Compressed air, Air expander, Pneumatic system, Energy storage, Integrated design

1. INTRODUCTION

With the continuous increase of renewable energy sources in the energy mix, it becomes necessary to address the issue associated with their intermittency (solar, wind) and the mismatch between daily energy demand and production [1, 2]. The way to overcome this challenge is energy storage. Storage systems can be divided into mechanical, chemical, electrochemical, and electrical systems [3].One of the mechanical energy storage systems is Compressed Air Energy Storage (CAES), which is dynamically evolving and highly promising [4]. Micro-CAES is an energy installation with a rated power of up to 1 MW, typically fluctuating in the range of tens of kW [5]. Its role involves effective storage and delivery of electrical energy, contributing to the stability and efficiency of the entire energy infrastructure. This technology also finds its use in independent energy systems for individual homes [6]. CAES can also directly collaborate with renewable energy sources such as photovoltaics to optimize the energy efficiency of a building [7]. As there are issues with efficiency or start-up time of those systems a lot of hybrid combinations can be seen in literature. Authors [8] connects CAES with super capacitors creating CAES-SC to achieve efficiency of 57,9 %. Another solutions requires additional heat storage to optimize thermodynamic processes during compression and expansion in systems operation [9].
The design process of entire Micro-CAES system can be divided into specific stages that include analyses of individual subsystems. Generally, we can categorize the design into Conceptual, Pneumatic, Mechanical and Electrical. Each of these steps is continuously reanalysed throughout the entire design process and reconsidered after each modification introduced in another stage. **Figure 1** show integrated design patch scheme.

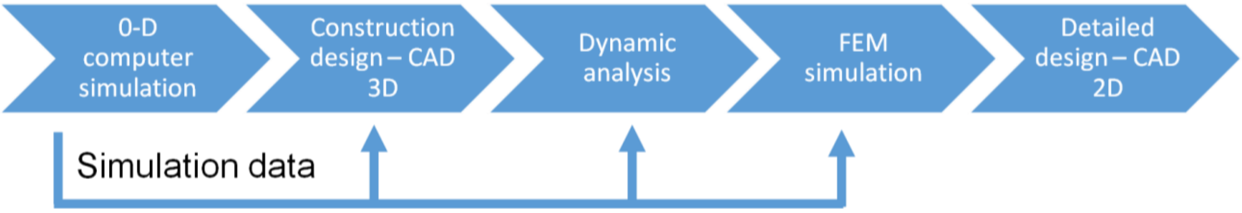


Figure 1: Integrated design patch scheme.

Conceptual Design involves analysing various aspects, starting from determining the overall desired parameters of the system that will impact its functionality. At this stage, system components are being preliminarily selected. The choice of optimal solutions, especially the type of expander, plays a significant role in achieving the highest efficiency [5]. The focus then shifts to Pneumatic System Design, where analysis is conducted on elements such as the air tank, piston, and valves, as well as on process parameters like supply pressure, storage pressure, and air control system, to meet the goals set in the conceptual phase. The subsequent step is Mechanical Construction Design, where elements related to mechanical drive and transmission are analysed. This includes evaluating the strength of each component to ensure the durability, reliability, and efficient operation of the device. Those stages allow for the creation of a coherent CAD design of the mechanical drive of the piston expander, shown in **Figure 2**

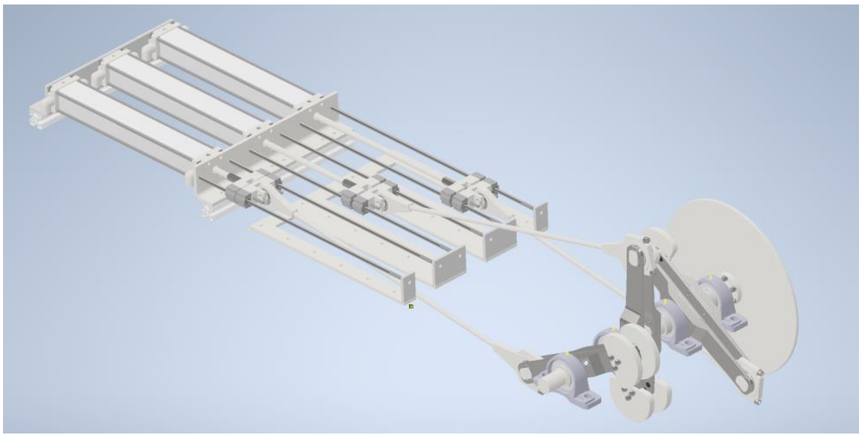


Figure 2: Mechanical construction of multi-cylinder air expander

As a result, the system design process involves a gradual transition from general parameters, through construction specifications, to a detailed consideration of individual elements, ensuring consistency and efficiency throughout the design process.

2. MATHEMATICAL MODEL

Our μ-CAES solution involves the utilization of a compressor, a storage tank, and a reciprocating engine. Actuators driven by compressed air initiate rotary motion of the shaft through a crankshaft. The shaft is connected to a high-speed generator via a mechanical gearbox, ensuring the required rotational output speed. Considering micro-scale installations, potential issues with excessively long start-up times or high resistance during intermittent operation are avoided. Therefore, the decision was made not to install a gas turbine, which is not resistant to varying mass flow rates of the supplying air and would also significantly increase installation costs. Instead, a reciprocating engine was installed, capable of operating with variable loads, pressures, and mass flow rates during a single tank discharge. **Figure 3** below illustrates the micro-CAES scheme.

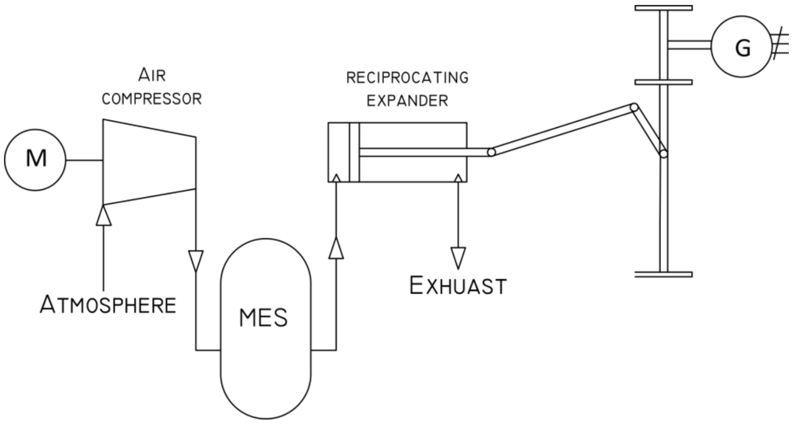


Figure 3: Mechanical construction of micro-CAES

The mathematical model is highly important in the real system design process as it provides force necessary data for conducting dynamic strength analysis of the system components. Model was based on differential equations described by the authors [10].

$$\begin{cases} J\frac{d^2\alpha}{dt} = \sum T \\ \omega = \frac{d\alpha}{dt} \\ L\frac{dI}{dt} = K\omega - R_G I \end{cases} \tag{1}$$

Driving torque:

$$T_D = r \cdot F_1 \sin(\alpha)\left(\left(\frac{r}{l}\right)^2 \sin^2(\alpha) + \frac{r}{l}\cos(\alpha)\sqrt{1-\left(\frac{r}{l}\right)^2 \sin^2(\alpha)} - 1\right) \tag{2}$$

The braking torque arising from the friction in the bearings:

$$T_F = f_1 \cdot Pz \cdot d_b + f_0(\nu \cdot \omega)^{\frac{2}{3}} \cdot d_b \cdot 10^{-7} \tag{3}$$

Generator breaking torque:

$$T_G = \frac{3}{2} \cdot I \cdot \Psi \cdot N_{pb} \tag{4}$$

The angular speed of the shaft:

$$\omega(t) = \frac{d\alpha}{dt} \tag{5}$$

The mass flow is calculated by formulas presented by Leszczyński and Gryboś [11]:

$$\dot{m} = \begin{cases} A\sqrt{\frac{2\kappa}{\kappa-1} \cdot \rho p_1 \left(\left(\frac{p_2}{p_1}\right)^{\frac{2}{\kappa}} - \left(\frac{p_2}{p_1}\right)^{\frac{\kappa+1}{\kappa}}\right)} \; for \; \frac{p_2}{p_1} > \beta \\ 0 \quad for \quad p_1 < p_2 \\ A\sqrt{\kappa\rho p_1 \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa+1}{\kappa-1}}} \quad for \; \frac{p_2}{p_1} \leq \beta \end{cases} \tag{6}$$

As noted in [11] the value of β equals:

$$\beta = \zeta((\frac{2}{\kappa+1})^{\frac{\kappa}{\kappa-1}} \tag{7}$$

Where ζ takes values from 0 to 1 depending cross section edges smoothness.

The power output is calculated as follows:

$$P_{out} = I^2 R_G \tag{5}$$

Estimated power of compressed air [12]:

$$P_{ca} = p_a \dot{V}_a \ln \frac{p_a}{p_b} \tag{6}$$

Round trip efficiency compares the output power of the expander and power of used compressed air :

$$\eta = \frac{P_{out}}{P_{air}} \tag{7}$$

To verify the accuracy of the mathematical model, experimental data from a single-cylinder expander were compared in previous study by authors [10]. Obtained data from the simulation was cross-referenced and compared with real-world observations or measurements. This step is crucial to confirm the reliability of the simulation data used for designing a multi-cylinder prototype. The validation process substantiated and affirmed the reliability and accuracy of the simulation results. The consistency between the simulated outcomes and the actual performance provided a robust validation of the mathematical model.

3. RESULTS

To select components such as a generator or mechanical gear, computer simulations were performed to illustrate the unit's characteristics. Proper selection of parameter values will allow the expander to operate in the highest efficiency ranges. **Table 1** below shows desired parameters of three piston air expander.

Table 1: Desired parameters of three piston air expander

Parameter	**Value**
Mechanical power of Expander	1-2 kW
Energy storage	0,9 kWh (2x1m^3)
Roundtrip efficiency	Over 50%
Discharge time	20-30 min @ 7 bar
Storage pressure	11 bar
Expansion pressure	6-11 bar

In the first step, in order to select the appropriate control strategy characteristics of expander with implemented PWM (Pulse Width Modulation) function were carried out. PWM control function regulate the injection of compressed air into the actuator chamber. Depending on the specified condition, the supply of compressed air is somewhat restricted, resulting in reduced consumption with a slight decrease in output power. Due to the expansion of air in the chamber, even when the supply is cut off, its velocity does not change significantly, and it is capable of performing work. **Figure 4 and 5** shows characteristics of expander with implemented PWM function and three actuators with 50mm of piston diameter.

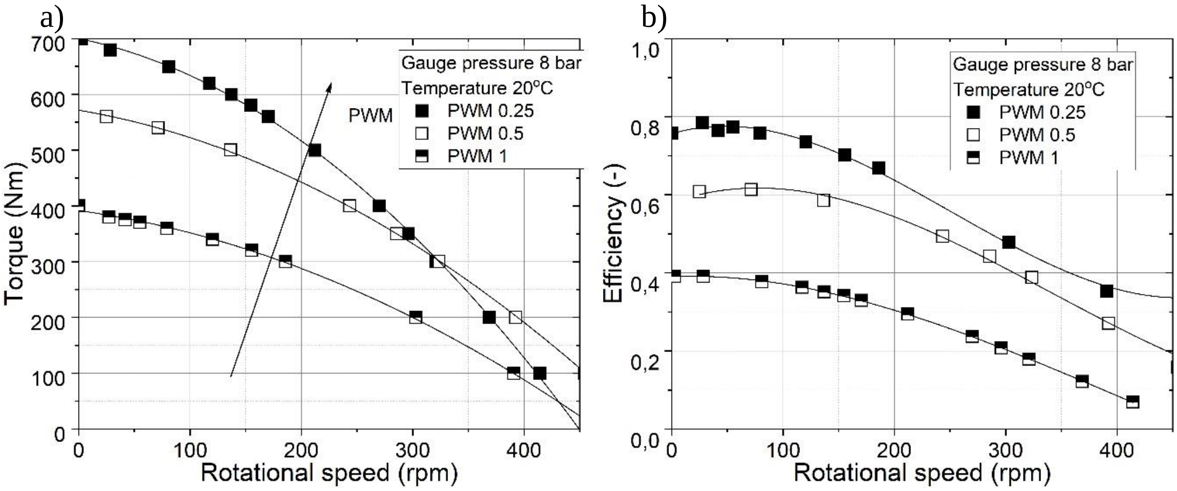


Figure 4: Characteristics of multi-piston expander with PWM function: a) torque and b) efficiency.

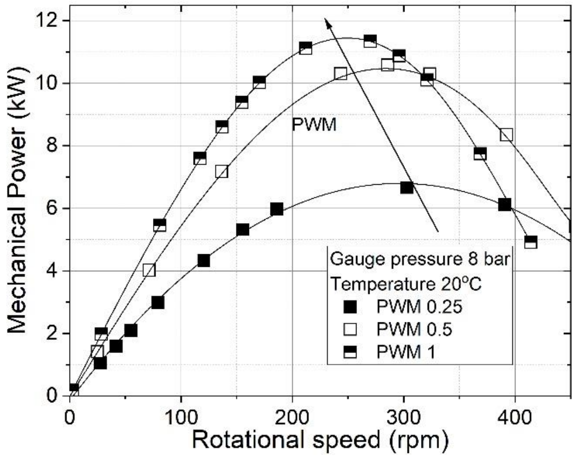


Figure 5: Characteristics of multi-piston expander with PWM function: mechanical power.

For each subsequent lower PWM coefficient, the mechanical power decreases. With gauge pressure of 8 bar and PWM 0.25 efficiency of 0.8 was obtained with generated mechanical power of 1.2 kW. In further stages it was decided to analyze system with multi-piston expander with implemented PWM 0.25 function to seek the highest possible efficiency. Characteristics shown on **Figure 6** were determined for three different supply gauge pressure values: 6, 8, and 10 bars , in order to select the appropriate supply pressure.

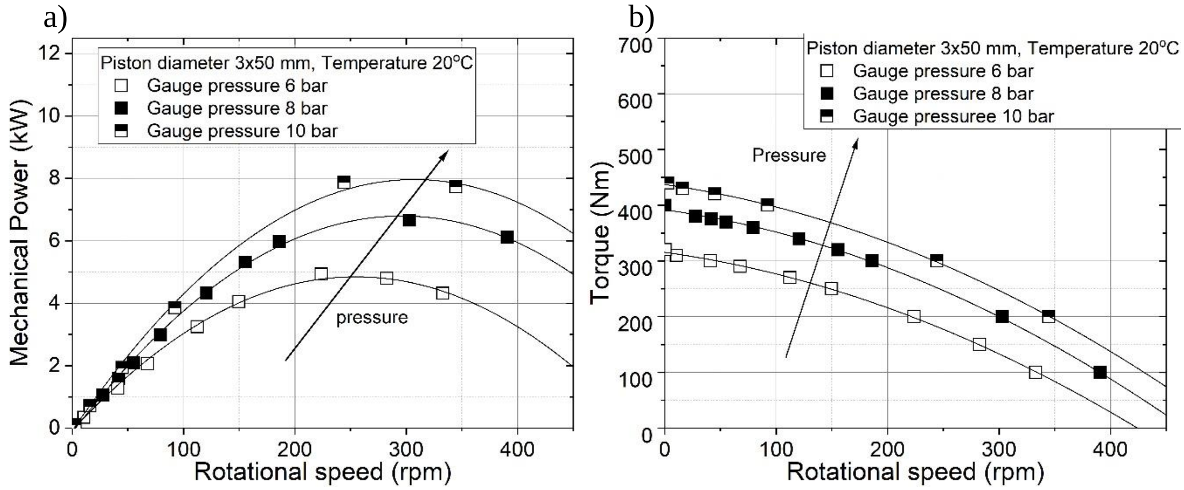


Figure 6: Characteristics of multi-piston expander: a) mechanical power and b) torque.

The maximum mechanical power obtained increases with the supply pressure, reaching from 5 kW for gauge pressure of 6 bar to 8kW for gauge pressure of 8 bar. Since the anticipated highest efficiency is expected to occur at low rotational speeds, it has been decided that further considerations will focus on a system powered by a gauge pressure of 8 bars in order to meet the intended output power. The characteristics (**Figure 7 and Figure 8**) were conducted for a supply pressure of 8 bar for three different sizes of actuators.

a) b)

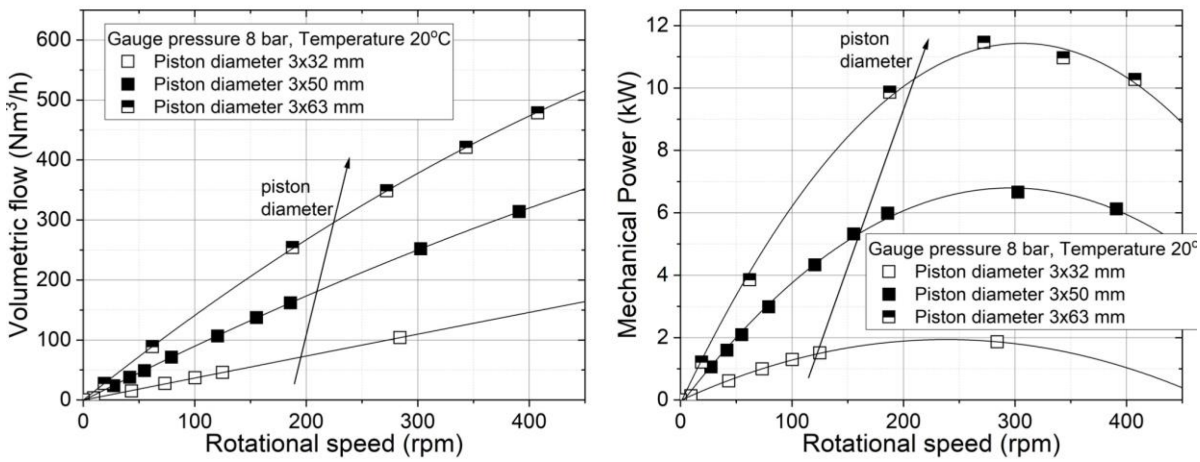


Figure 7: Characteristics of multi-piston expander: a) volumetric flow and b) mechanical power.

a) b)

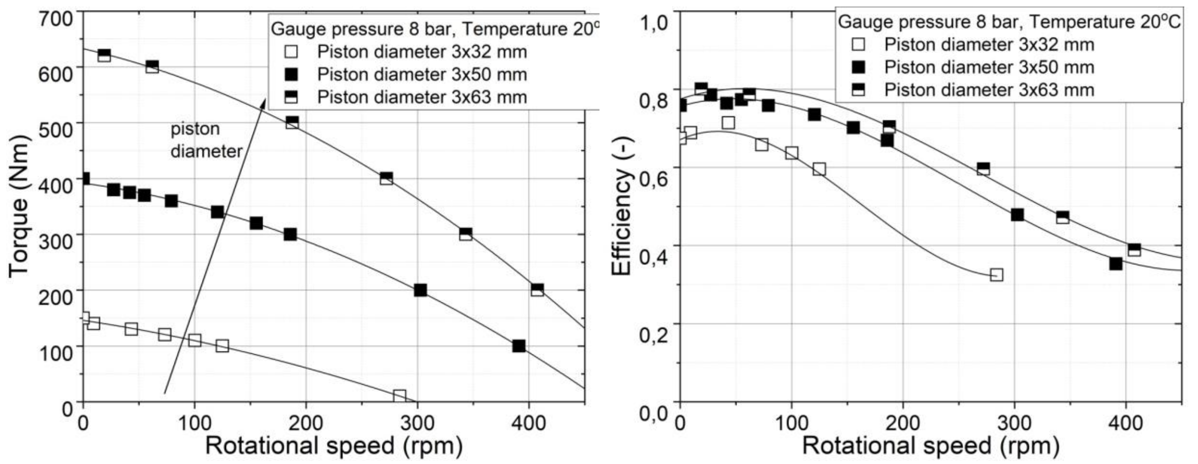


Figure 8: Characteristics of multi-piston expander: a) torque and b) efficiency.

The expander's characteristics are intricately linked to its ability to function optimally at speeds below 60 revolutions per minute. This feature underscores the importance of precision and careful consideration during the design process. In order to achieve the highest efficiency of the expander, it is critical to maintain a rotational speed around 50 revolutions per minute. In the subsequent stages of designing the entire system, the construction planning will be centred on ensuring optimal operation within this particular speed range. It has been emphasized that the power generated by the device is satisfactory, serving as a positive indicator for the overall efficiency and performance of the entire project. **Figure 9** below shows the linear velocity of the piston for three different sizes of actuators powered by a gauge pressure of 8 bars.

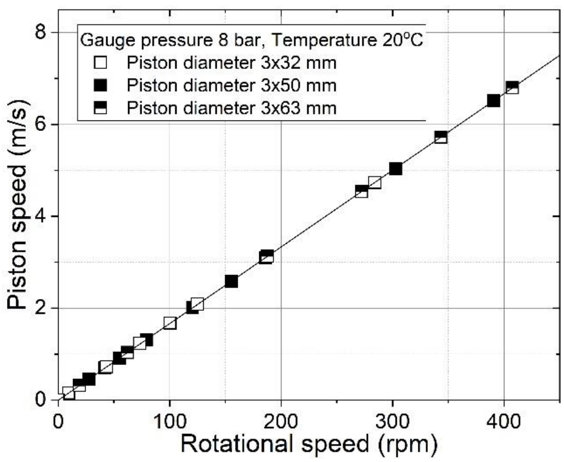
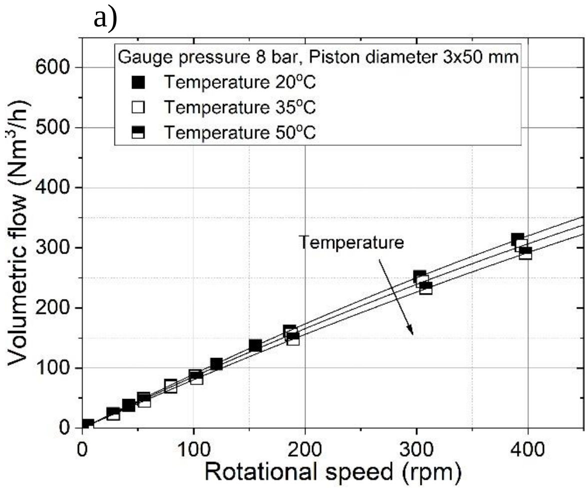


Figure 9: Linear speed of piston for three different actuators sizes.

At a constant supply pressure, the piston velocities remain unchanged regardless of the actuator size increasing with rotational speed. Irrespective of whether the actuator is larger or smaller, it receives the same piston velocity at the same level of supply pressure. The last considered aspect was examining the impact of temperature on the performance of the expander. Characteristics shown on **Figure 10** below were obtained for three different values of the supplied air temperature.



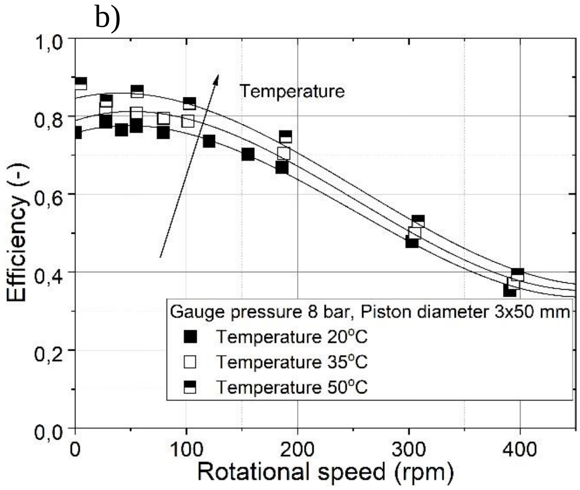


Figure 10: The influence of temperature on the characteristics of the air expander: a)volumetric flow, b) efficiency.

As the temperature increases, the changing air density leads to reduced volumetric consumption, resulting in higher achieved overall efficiency. Heating the air to a temperature of 50 degrees allows to achieve efficiency of up to 0.9 The overarching goal of this project is to optimize the design and functionality of the pneumatic system to seamlessly match the specified power output, ensuring both efficiency and reliability. Implementing proper control of the entire system, using Pulse Width Modulation (PWM) function and introducing additional air heating aim to further enhance of systems round trip efficiency. It is important to emphasize that the pursuit of higher efficiency is not a singular effort but rather a combination of various strategies. Only the integration of all these efficiency-improving methods and an analysis of their optimal collaboration will allow for the best performance outcomes.

4. CONCLUSIONS

The paper introduces a mathematical model essential for establishing design and process parameters in micro-CAES piston expanders. Computer-simulated data from this model serve as inputs for strength calculations, FEM analysis, and the construction of the mechanical drive. A specific case study is showcased, focusing on the design of a 1-2 kW piston expander operating at a gauge pressure of 6-10 bar. The anticipated mechanical efficiency of the air expander is targeted at 0.7. Simulations showed that compressed air supply control and waste heat utilization during expansion can affect on increase of efficiency to 0.8. The synergy between process parameters, PWM control system, and supplementary heating plays a key role in elevating the overall effectiveness of the expander within the CAES system. Only through the proper integration of these efficiency-enhancing methods and a thorough analysis of their optimal collaboration can the system achieve its highest potential.

5. ACKNOWLEDGMENTS

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NOMENCLATURE

A	Cross-section	m^2
I	Current	A
J	Moment of inertia	Nm
K	Generator constant	Wb
L	Inductance	H
P_z	Equivalent load	N
P_{out}	Power output	W
P_{ca}	Compressed air power	W
R_G	Resistance	Ohm
T	Torque	Nm
$\dot{V}$	Volumetric flow	$\frac{m^3}{min}$
d_b	Bearing pitch diameter	m
f_0, f_1	Bearing factors	-
l	Crank arm length	m
$\dot{m}$	Mass flow	kg/s
p_1,p_2	Pressures in actuator chambers	
p_a	Storage pressure	Pa
p_b	Pressure after the expansion process	Pa
r	Crank length	Pa
t	Time	m
α	Angular position of the crank	s
β	Critical pressure ratio	rad
ω	Angular speed	$\frac{rad}{s}$
v	Kinematic viscosity of oil	mm^2/s
κ	Adiabatic exponent for air	-
ρ	Density	kg/m^3
ζ	Numerical factor	-
CAES	Compressed Air Energy Storage	
PWM	Pulse Width Modulation	

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