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Investigation and simulation based optimization of an energy storage system with pressurized air

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Abstract As a central goal of the energy transition in Germany, the share of renewable energies is to be increased to over 80% by 2050. Due to fluctuating wind conditions or the day-night cycle, storage systems must be integrated into the supply grid for a continuous regenerative power supply from wind and solar energy. In addition to pumped storage systems, batteries and Power2Gas approaches, compressed gases (optimally air) can also be used for this purpose. The aim of the research and development project presented is to develop such a storage unit with the best possible efficiency and long service life. To this end, basic calculations were first made on possible efficiencies depending on the assumed changes in the state of the working gas. Furthermore a piston compressor for compressed air generation was investigated experimentally with regard to its efficiency. In addition, the compressor was modelled and simulated in a corresponding software. Thus, on the one hand, the efficiency of the existing piston compressor could be determined experimentally and, on the other hand, the simulation model could be assessed with regard to its suitability for the purpose of simulation-based optimization. Measures to increase efficiency can be derived from the results. In addition, it becomes possible to forecast the achievable overall efficiency of such an energy storage system with compressed air.

Keywords: Renewable energy; Pressurized air; Storage system; Efficiency study; Simulation based investigation and optimization

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1 Introduction

The energy turnaround triggered by climate change is leading to the replacement of fossil fuels with renewables in Germany and worldwide. The aim is to achieve greenhouse gas neutrality by 2050 [1]. Renewable energy sources include wind, solar and tidal energy. These forms of energy are subject to natural daily and annual fluctuations in their output. To ensure a continuous supply of electricity to the population, storage systems for these forms of energy must be developed that can compensate for the natural fluctuations. To this end, various approaches to storage are being worked on worldwide, such as pump storage systems, batteries and Power2Gas-concepts. Another possibility is to store energy in the form of compressed gases. Since air is available in sufficient quantities everywhere: on earth, energy storage systems with this gas would be usable worldwide. Moreover, this medium poses no danger to the environment, e.g. in the event of a leak.

The use of compressed air as an energy source gives rise to various problems. One of the biggest is the temperature increase or decrease that occurs during the compression or expansion of the air. The resulting temperatures must be limited both upwards and downwards during compression or expansion to avoid damage to the system. In order to remove the heat from the air during compression, it is either dissipated as heat loss (as with CAES – compressed air energy storages) or used to heat suitable heat accumulators (as with AA-CAES – advanced adiabatic compressed air energy storages) [2, 3]. Since the temperature drops sharply during the expansion of the gas and this can have a damaging effect on other system components, heat energy must be supplied to the gas before the expansion in order to increase the temperature. In an AA-CAES, the heat for this can be used from the heat accumulator. This also makes it possible to achieve a significantly higher overall efficiency compared to a CAES. However, the construction of an AA-CAES is more complex than that of a CAES, as a heat storage tank and corresponding heat exchangers (including a heat pump if necessary) are also required [4].

Another problem is the compression process itself. In such a system, to achieve the highest possible efficiency, the compression or expansion should take place in several stages with isobaric intercooling/heating [5]. For this purpose, appropriate compressors/motors and heat accumulators must be used or developed.

2 Task

The overarching goal of the R&D project, in which this research work is integrated, is the development of an AA-CAES with adequate efficiency. The system is to be modular and freely scalable, so that the area of application ranges from single-family and multi-family houses to commercial and industrial enterprises to wind and solar parks. Apart from sufficient space, no other requirements are placed on the location. The focus of the development is on the highest possible overall efficiency, the lowest possible environmental impact in the extraction of the raw materials and in the production of the system, as well as very long service life of the storage system.

In the research work presented here, the necessary calculation bases for the design of an AA-CAES were first compiled. With the formal correlations, the theoretically achievable efficiencies with different system parameters (stage pressure ratio, number of stages, intercooling, etc.) can first be estimated for the charging process.

The efficiency was then determined experimentally on a multi-stage piston compressor and compared with the calculated efficiency. For this purpose, the piston compressor was equipped with appropriate measurement technology and the efficiencies were determined at different accumulator sizes and discharge pressures based on the measurement data. From the results, the flashover calculation can be evaluated in terms of usability. In addition, initial improvement measures regarding the design of an optimised multi-stage piston compressor can be derived from this.

In parallel, a model of the pressure storage unit with piston compressor was built in the Simcenter Amesim [14] software and the model was parameterised according to the real system. The measurement and simulation results should show good agreement. This means that the model can be used for later simulation-based optimization of the storage and the storage can be examined virtually in different variants. Measures to increase efficiency can be derived from the results. Furthermore, it is possible to forecast the achievable overall efficiency of such an energy storage system with compressed air.

3 Theoretical consideration of the compression process

Thermodynamically, the compressor is a quasi-stationary open system. In this system, the working medium (in this case air) is compressed in one or more stages. With each compression, there are various losses that influence the efficiency of the compressor. In order to be able to assess the overall efficiency and derive possible optimization approaches, the compression process is first explained theoretically.

3.1 Multistage compression process with harm volume and intercooling

The compressor intended for the energy storage unit is a multistage piston compressor with isobaric intercooling, as this type of compressor is very well suited for pressures up to 300 bar and beyond at appropriately high volume flows [6]. The pistons in the compressor oscillate between top and bottom dead centre during operation. During the downward movement of a piston, the air is delivered into the respective cylinder via a corresponding inlet. During the upward movement, this air in the cylinder is compressed and, when a corresponding pressure is reached, is pushed out via the provided outlet into the next pressure range. In the process, fresh air is circulated around the system via a fan and thus cooled [7]. Intermediate cooling reduces the mass-specific internal work to be done. On the other hand, the final temperature of the compressed air between the stages is considerably lowered by the intercooling and thus the thermal load on the components of the compressor is reduced. To illustrate this, a single-stage polytropic compression process and the three-stage polytropic compression process with isobaric intercooling presented here are compared schematically in Fig. 1.

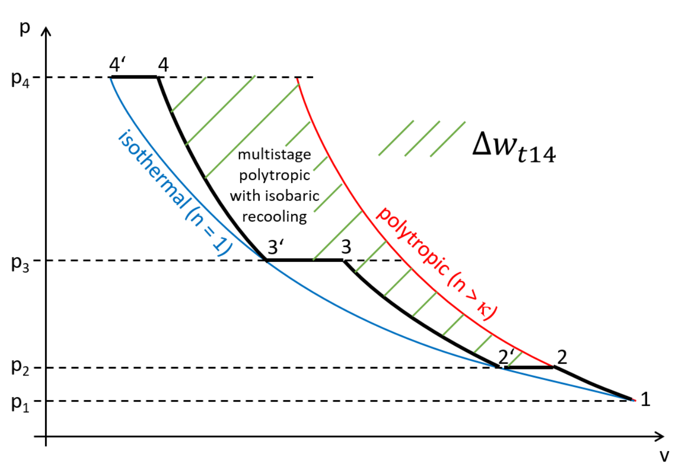

Figure 1: Comparison of single-stage compression (polytropic and isothermal) and multistage compression [5].

It becomes clear that a single-stage compression involves a considerable amount of additional internal work. In addition, it becomes clear that isothermal compression requires the smallest amount of work, which, however, is hardly feasible in practice. Therefore, only multi-stage compression with isobaric intercooling will be discussed further in the following, and the calculation principles necessary for determining efficiency will be explained.

Assuming that the changes in the kinetic and potential energy components can be neglected, the mass-specific inner work ($w_{i,xy}$; $x = 1,\ 2$ or 3 and $y = x + 1$) per stage is obtained (assuming a constant stage pressure ratio)

$$w_{i,xy} = h_{xy} - q_{xy} = w_{i12} = w_{i2'3} = w_{i3'4}\,, \tag{1}$$

where: $w_{i,xy}$ – mass-specific inner work per stage in J/kg, h_{xy} – mass-specific change of enthalpy per stage in J/kg, q_{xy} – mass-specific transitional heat per stage in J/kg.

The mass-specific inner work results from the sum of the mass-specific pressure change work and the mass-specific dissipation work. The following applies:

$$w_{i,xy} = w_{t,xy} + w_{\mathrm{Diss},xy}, \tag{2}$$

where: $w_{t,xy}$ – mass-specific work of pressure change per stage in J/kg, $w_{\mathrm{Diss},xy}$ – mass-specific dissipation work per stage in J/kg.

The following applies to the mass-specific work of pressure change per stage:

$$w_{t,xy} = \int\limits_x^y v\,dp = w_{t,12} = w_{t,2'3} = w_{t,3'4}\,, \tag{3}$$

where: v – mass-specific volume of gas in m^3/kg, dp – differential of pressure change.

The pressure increase of the air during compression is polytropic in real terms, whereby the corresponding polytropic exponents strongly depend on the respective compressor and its quality. This results in the mass-specific work of pressure change per stage

$$w_{t,xy,\mathrm{pol}} = \frac{n}{n-1} R_L T_x \left[\left(\frac{p_y}{p_x}\right)^{\frac{n-1}{n}} - 1\right], \tag{4}$$

where: $w_{t,xy,\mathrm{pol}}$ – mass-specific polytropic work of pressure change per stage in J/kg, n – polytropic exponent in 1/1 (in case of piston compressors $n > \kappa$), R_L – specific gas constant for air in J/(kg K), T_x – ambient/start gas temperature in K, p_y – pressure after compression in Pa, p_x – pressure before compression in Pa. The specific gas constant R_L for air is approximately 287 J/(kg K). The ratio of final stage pressure p_y to stage input pressure p_x represents the stage pressure ratio π_{st}. It results from the total pressure ratio π_{tot} and the number of stages z and is determined as

$$\pi_{\text{st}} = (\pi_{\text{tot}})^{\frac{1}{z}} = \left(\frac{p_{\text{max}}}{p_1}\right)^{\frac{1}{z}}, \tag{5}$$

where: π_{st} – pressure ratio per stage in 1/1, π_{tot} – total pressure ratio in 1/1, z – number of stages in 1/1, p_{max} – maximum pressure (maximum storage pressure) in Pa, p_1 – minimum pressure (ambient pressure) in Pa.

Isothermal compression/expansion represents the most energetically favourable compression/expansion of the working medium and shall therefore serve as a reference value in the following. For isothermal compression, the following applies with regard to the mass-specific pressure change work to be applied

$$w_{t,xy,i\text{th}} = p_x v_x \ln \frac{p_y}{p_x} = R_L T_x \ln \pi_{\text{st}}, \tag{6}$$

where: $w_{t,xy,i\text{th}}$ – mass-specific isothermal work of pressure change in J/kg, v_x – mass-specific volume before compression in m^3/kg, π_{st} – pressure ratio per stage in 1/1.

Note The following applies accordingly for an isentropic change of state:

$$w_{t,xy,s} = \frac{\kappa}{\kappa - 1} R_L T_x \left[(\pi_{\text{st}})^{\frac{\kappa-1}{\kappa}} - 1\right], \tag{7}$$

where: $w_{t,xy,s}$ – mass-specific isentropic work of pressure change in J/kg, κ – isentropic exponent in 1/1 ($\kappa \approx 1.4$ for dry air).

From the equations, it can be seen that, compared to isothermal compression, isentropic or polytropic compression always requires more work the greater the polytropic exponent. However, due to the speed at which a piston compressor is actually operated (2000 compressions per minute and more), the change of state taking place in the compressor approaches an isentropic change of state, or the real change of state takes place with a polytropic exponent that is generally greater than the gas-dependent isentropic exponent. This is caused by the dissipation work that arises. The internal work of a polytropic compression under the assumption of an adiabatically running process ($q_{xy} = 0$, this assumption is based on the previously mentioned velocity of the process) can be determined as follows:

$$w_{i,xy} = h_{xy} = c_p \left(T_y - T_x\right), \tag{8}$$

where: c_p – mass-specific heat capacity at constant pressure in J/(kg K), T_y – gas temperature after (polytropic) compression in K.

The mass-specific heat capacity at constant pressure (c_p) for air is about 1004 J/(kg K). The temperature of the air after compression is to be determined via the stage pressure ratio and the respective polytrope exponent. The following applies:

$$T_y = T_x \left(\pi_{\mathrm{st}}\right)^{\frac{n-1}{n}} . \tag{9}$$

With approximately isothermal compression, compression work could be saved. For this reason, among others, the piston compressor used here operates with a multi-stage compression process with intermediate cooling.

A minimum for the total mass-specific pressure change work to be applied results from the realisation of a constant stage pressure ratio (assuming isobaric recooling to the initial temperature, cf. [6]). Thus, the mass-specific internal work to be applied over the entire compression process results in

$$w_{i,\mathrm{tot}} = z w_{i,12,\mathrm{pol}} = z c_p \left(T_2 - T_1\right) , \tag{10}$$

where $w_{i,\mathrm{tot}}$ is the total mass-specific inner work in J/kg.

The corresponding total pressure change work performed for a given number of stages and a given total pressure ratio is given by

$$w_{t,\mathrm{tot,pol}} = z w_{t,12,\mathrm{pol}} = z \frac{n}{n-1} R_L T_1 \left[\left(\pi_{\mathrm{st}}\right)^{\frac{n-1}{n}} - 1\right], \tag{11}$$

where $w_{t,\mathrm{tot,pol}}$ is the total mass-specific polytropic work of pressure change in J/kg.

The dissipation work can be determined from the difference between the internal work and the pressure change work

$$w_{\mathrm{Diss,tot}} = z \left(w_{i,12,\mathrm{pol}} - w_{t,12,\mathrm{pol}}\right), \tag{12}$$

where: $w_{\mathrm{Diss,tot}}$ is the total mass-specific dissipation work in J/kg.

If the air is isobarically cooled back to the initial temperature after each stage, the following heat must be extracted from the gas per recooling:

$$q_{yy'} = c_p \left(T_{y'} - T_y\right) = q_{22'} = q_{33'} = q_{44'}, \tag{13}$$

where: $q_{yy'}$ – mass-specific transitional heat in J/kg, $T_{y'}$ – gas temperature after re-cooling (is equal to T_1) in K.

If we now relate the work $w_{t,14',ith}$ (single-stage compression) to be theoretically applied in isothermal compression to the internal work applied, we obtain the isothermal efficiency of the compressor

$$\eta_{C,i\text{th}} = \frac{w_{t,14,i\text{th}}}{w_{i,\text{tot}}}, \tag{14}$$

where: $\eta_{C,ith}$ – isothermal compressor efficiency in 1/1, $w_{t,14,ith}$ – mass-specific isothermal technical work in J/kg.

This efficiency describes the proportion of the work to be done to compress the air that could be maximally recovered by isothermal expansion (the recoverable technical work is at a maximum with isothermal expansion). In addition, the mechanical-electrical compressor efficiency must also be taken into account. This represents the ratio of internal work and supplied electrical work. The following applies:

$$\eta_{meC} = \frac{w_{i,\text{tot}}}{w_{el}}, \tag{15}$$

where: η_{meC} is the mechanical compressor efficiency in 1/1.

Here, all efficiencies in the conversion chain from supplied electrical energy to the output of internal work must be taken into account (for example, engine efficiency, gearbox efficiency, losses due to cooling fans and oil pumps, etc.).

3.2 Estimation of achievable efficiencies

In the following, the achievable efficiencies of a piston compressor with a given total pressure ratio and the corresponding number of stages are estimated using the derivations for the compression process. For this purpose, the following data of the experimentally investigated piston compressor are used as an example (Table 1).

Table 1: Coal composition.

Parameter	Value
Maximum pressure	$p_{\max} \approx 300$ bar
Number of stages	$z = 3$
Assumed polytropic exponent	$n \approx 1.65$
Ambient pressure	$p_1 \approx 1$ bar
Ambient temperature	$T_1 \approx 293$ K

First, the optimum stage pressure ratio must be determined. This results in

$$\pi_{\mathrm{st}} = (\pi_{\mathrm{tot}})^{\frac{1}{z}} = \left(\frac{p_{\max}}{p_1}\right)^{\frac{1}{z}} = \left(\frac{300\ \mathrm{bar}}{1\ \mathrm{bar}}\right)^{\frac{1}{3}} = 6.69. \tag{16}$$

This gives the maximum temperature after compression

$$T_2 = T_1\,(\pi_{\mathrm{st}})^{\frac{n-1}{n}} = 293\ \mathrm{K}\cdot 6.69^{\frac{1.65-1}{1.65}} = 619.5\ \mathrm{K}\ \ (\approx 346^\circ\mathrm{C}). \tag{17}$$

From the data, the total mass-specific internal work to be applied can be calculated as

$$\begin{aligned} w_{i,\mathrm{tot}} &= z w_{i,12} = z c_p\,(T_2 - T_1) = \\ &= 3\cdot 1.004\,\frac{\mathrm{kJ}}{\mathrm{kg\,K}}\cdot(619.5-293)\ \mathrm{K} \approx 983\,\frac{\mathrm{kJ}}{\mathrm{kg}}\,. \end{aligned} \tag{18}$$

In addition, the mass-specific pressure work of change for isothermal compression can be determined as

$$w_{t,14',i\mathrm{th}} = R_L T_1 \ln \pi_{\mathrm{tot}} = 287\,\frac{\mathrm{J}}{\mathrm{kg\,K}}\cdot 293\,\mathrm{K}\cdot \ln\frac{300}{1} \approx 480\,\frac{\mathrm{kJ}}{\mathrm{kg}}\,, \tag{19}$$

This gives the isothermal compressor efficiency

$$\eta_{C,i\mathrm{th}} = \frac{w_{t14',i\mathrm{th}}}{w_{i,\mathrm{tot}}} = \frac{480}{983} = 0.488\ (= 48.8\%). \tag{20}$$

In the present case of the piston compressor, an asynchronous motor with $n = 2850\ \mathrm{min}^{-1}$ is used to drive the compressor via a V-belt drive. The speed of the piston compressor is then nominally $2300\ \mathrm{min}^{-1}$ due to the reduction. Assuming an efficiency of 85.9% for the electric motor (according to DIN EN 60034-30-1: IE3 for 2-pole electric motor with 230 V and 2.2 kW [8]), V-belt drive efficiency of 93% [9], mechanical efficiency of the piston compressor of approx. 90% [10] and an additional loss rate for the cooling air fan drive of 10% (efficiency 90%) [10], the mechanical-electrical compressor efficiency is estimated as follows:

$$\eta_{meC} = 0.859\cdot 0.90\cdot 0.93\cdot 0.90 = 0.647\ \ (= 64.7\%). \tag{21}$$

This results in an expected overall efficiency (only charging process of the compressed air energy storage) of

$$\eta_{\mathrm{tot}} = \eta_{V,ith}\eta_{meC} = 0.488\cdot 0.647 = 0.316\ \ (= 31.6\%), \tag{22}$$

where η_{tot} is the total efficiency in 1/1 or in percentage.

The result leads to the assumption that such energy storage systems generally have low efficiencies in the charging process and have therefore never been further developed for energy storage. However, there are some possibilities for improvement that can significantly influence the efficiency and thus enable a sufficiently high overall efficiency, as will be shown later with the example compressor.

4 Test bed and measurement

To estimate the overall efficiency of the energy storage unit that can be achieved, a piston compressor from Bauer is first examined with regard to its efficiency during the compression process. A compressed air cylinder with a capacity of five or twenty litres is used to store the generated compressed air. A combined voltage/current/power meter with documentation function from the company Voltcraft is used to measure the electrical energy supplied to the compression process. The pressure in the energy storage tank is recorded using a pressure sensor with a voltmeter (with integrated recording function) and an analogue manometer. The structure of the system with the corresponding sensor technology can be seen in the following figure.

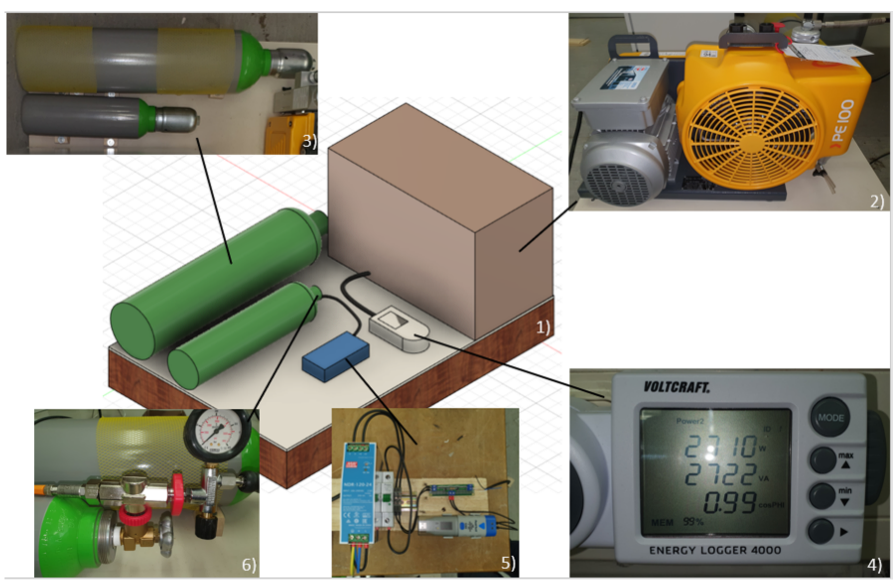

Figure 2: Experimental setup and sensor technology.

In the extracts from the simplified CAD model, the individual components can be seen as follows (Fig. 2):

(1) base plate (Euro pallet with mounting plate, for transportability),

(2) compressor (piston compressor from Bauer),

(3) pressure bottles (one 5-litre and one 20-litre bottle, used separately),

(4) energy logger (measurement of the supplied electrical energy),

(5) current-voltage converter and voltage logger (acquisition of voltage signal pressure sensor),

(6) pressure sensor and pressure gauge (measurement of storage charging pressure).

4.1 Description of the construction and function of the measuring system

The measurement system currently consists of only two parts, an energy meter to measure the electrical energy absorbed and a voltage meter which records the voltage in the pressure measurement circuit (coming from the pressure sensor). The energy cost measuring device is the Energy-Logger 4000 from the company Voltcraft [11]. During the measurement, it documents the recorded current as well as the apparent and active power. Integrated over time, the supplied electrical energy can be determined. The pressure sensor AP020 from the company Autosen is used to determine the pressure inside the compressed air cylinder to be charged [12]. This relative pressure sensor has a measuring range of 0 to 400 bar and generates a proportional output signal of 4 to 20 mA. The signal is converted into a proportional voltage *via* a corresponding shunt resistor and recalculated into the pressure signal.

4.2 Test procedure and results

To determine the efficiency, the compressor was prepared according to the operating instructions before each measurement. The compression process was then started and the resulting pressure curves over time and the curves of the electrical power consumption of the compressor were recorded. After reaching a defined storage pressure (the final pressures 100 bar, 200 bar and 300 bar were defined), the system was switched off. The pressure curve was then observed until the accumulator, which had been heated during compression, had returned to approximately ambient temperature.

An example of the power consumption curve and the pressure curve in the accumulator is shown in Figs. 3 and 4, respectively.

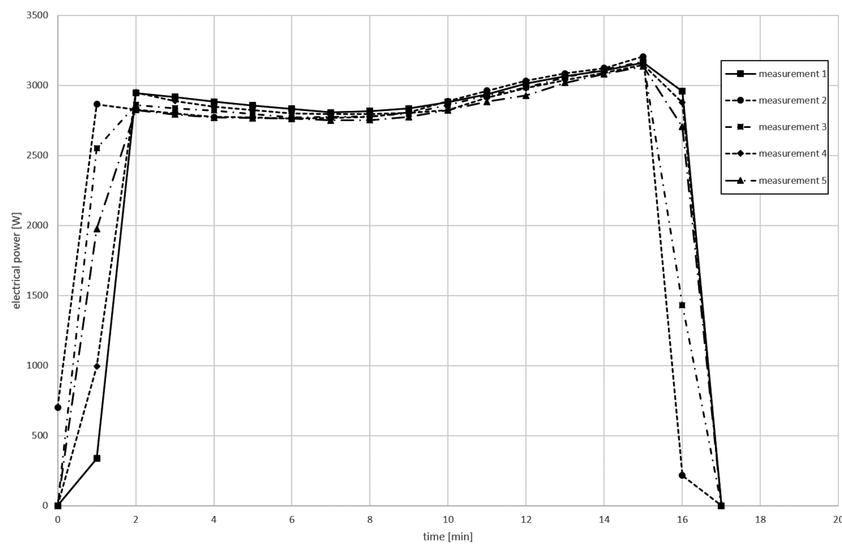

Figure 3: Electrical power *vs.* time for 5 measurements (0–300 bar).

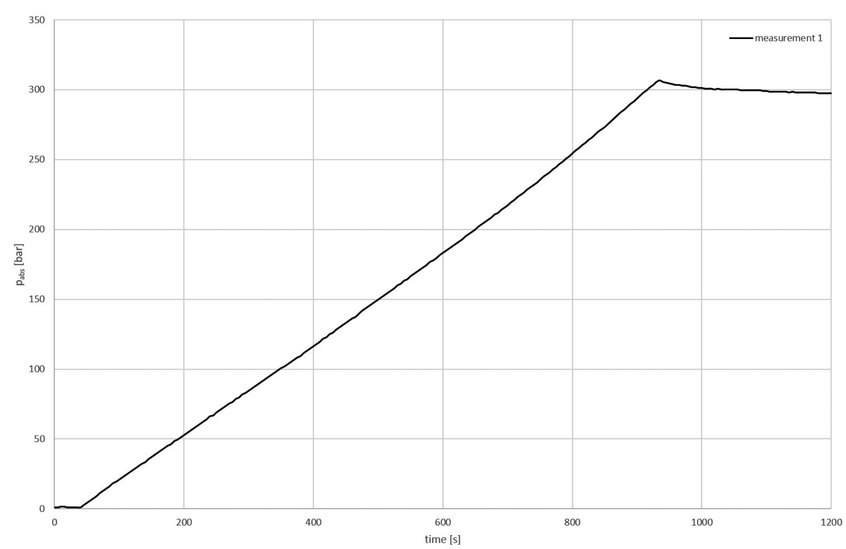

Figure 4: Absolute pressure *vs.* time for 1 measurement (0 bar to 300 bar).

From the measurement data, both the electrical energy supplied and the pressure change work remaining in the pressure accumulator during isothermal expansion can be determined (as a function of the residual pressure after temperature equalisation), since this possible expansion work also corresponds to the compression work to be applied during isothermal change of state.

For the measurements of the apparent power shown here, the average electrical energy supplied (by averaging) is 2.67 MJ. The stored pressure change energy resulting from the residual pressure of approx. 280 bar (after cooling to room temperature) is 0.789 MJ for the pressure bottle with a volume of 5 l.

If the efficiency is now calculated by putting the stored pressure change work concerning the electrical work supplied, the efficiency for the energy storage process here is only 29.6%. This efficiency is very low and illustrates the problems of energy storage using compressed air. However, it should be noted here that this efficiency almost corresponds to the estimated efficiency from the theoretical considerations (which is 31.6%).

4.3 Discussion of the measurement results and derivation of optimization approaches

The efficiency determined here (only charging the energy storage unit) of approx. 30% is very low and, with a similar efficiency of energy recovery into mechanical and, ultimately, electrical energy represents such a low overall efficiency that further development without optimization approaches does not make sense. However, some optimization measures are possible that can increase the overall efficiency enormously.

First of all, the recovery of waste heat during recooling should be mentioned, whereby two scenarios are debatable [13]:

- If this waste heat is temporarily stored and then reused in the re-expansion, an efficiency increase of at least 80% should be possible. However, due to the different temperature levels, the use of a heat pump may be necessary, which greatly complicates the system and makes it more maintenance-intensive.

- In the case of purely further use of the waste heat (e.g. for domestic or heating water), only a heat exchanger with a corresponding circuit must be used, which enables the transport of the heat into the corresponding heat storage. Preheating of the air before expansion could then be done with heat release (e.g. from fossil fuels), which would ultimately mean climate neutrality despite the use of fossil fuels.

Another optimization measure can be seen in the design. The piston compressor used always compresses to the corresponding final pressure, regardless of the charge state of the accumulator. Only the last stage does not always compress to 300 bar, but a so-called pressure maintenance valve between the 3rd stage and the storage cylinder ensures that at least a pressure of 150 bar is maintained in the area in front of the pressure maintenance valve. This means that at lower charging pressures, the gas simply expands into the pressure accumulator when the pressure retaining valve is opened and the pressure change work released in the process is dissipated. This design implementation is preferred for reasons of moisture extraction from the compressed air (later, the air here can also be used for breathing apparatus and diving cylinders), but is irrelevant for the application as an energy storage device.

In addition, in the case of the piston compressor examined, all pistons always work, even at accumulator pressures that would not require this. Optimization would be possible by switching the delivery paths (at low accumulator pressures, all pistons deliver directly into the pressure accumulator, when a corresponding accumulator pressure is exceeded, the system switches over, etc.). Another solution is to switch off and later switch on compressor stages that are not needed at the respective state of charge of the pressure accumulator.

5 Model of the multi-stage piston compressor

To be able to examine various optimization approaches in a simulation-based manner in the later course of the work, a model of the experimentally examined piston compressor was built in the software Simcenter Amesim [14] and parameterised with the characteristic values of the piston compressor. The model can be roughly divided into 3 areas, which are marked '5.1', '5.2' and '5.3' in Fig. 5. The fluidic part of the system is marked centrally with '5.1'. Above this is the pressure accumulator. The model of the mechanical part of the compressor and the pressure maintenance valve is marked with '5.2' on the left side of the figure. The entire cooling system (here already with water for the purpose of simplification for later variations) for the intermediate cooling of the compressed air is indicated in the figure with '5.3'.

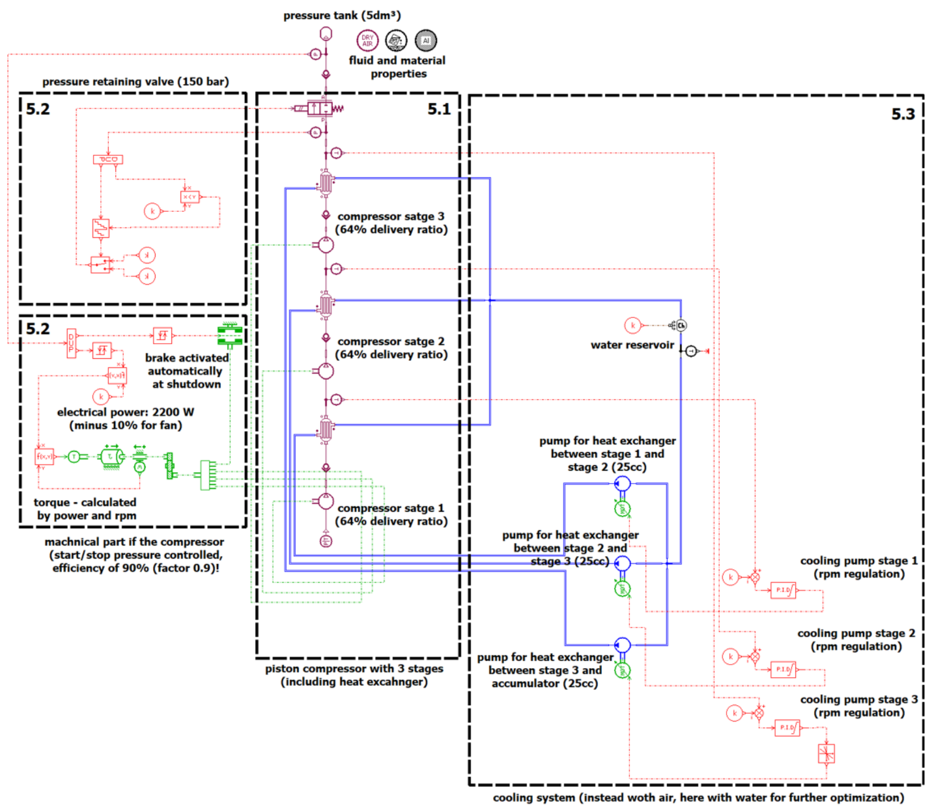

Figure 5: Model of the piston compressor (modelled with Simcenter Amesim).

With the parameterised model, it is currently possible to calculate the pressure and temperature curve in the storage tank during the charging process on a time basis with sufficient accuracy. For model verification, the simulation results are directly compared with corresponding measurement results, as shown in Fig. 6.

It becomes clear that the simulation model represents the processes in the charging process very well. Smaller deviations are due, for example, to the assumed values for heat transfer coefficients or the heat storage capacity of individual components. In subsequent work, further data (pressures between the individual stages, temperatures upstream and downstream of the respective heat exchangers, air mass flow, etc.) must be recorded experimentally. This will allow the model to be detailed accordingly, and the influences of the individual components on the charging process and its efficiency will become apparent.

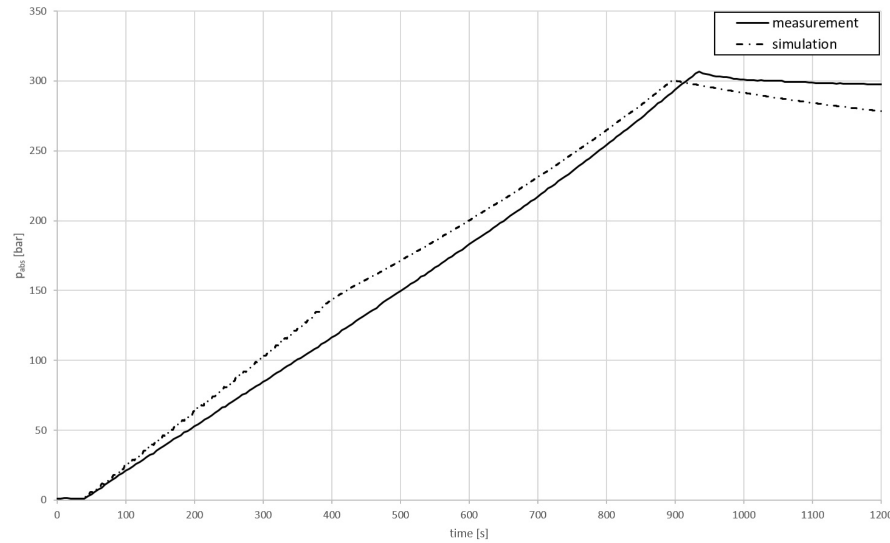

Figure 6: Comparison of the simulated and measured pressure.

6 Summary and outlook

The model-based estimated efficiencies, as well as those calculated from the experimental results of the three-stage piston compressor for realising an energy storage unit by means of compressed air, are very low at approx. 30% (charging process only). Further development of the storage unit without special measures to improve the efficiency in the charging process is therefore not recommended, as the system would have an overall efficiency of less than 10% with similar efficiency in energy recovery. However, some optimization approaches can significantly increase the efficiency when charging the energy storage unit. These include, above all, the use of waste heat in the compression process, which could increase the isothermal compressor efficiency to over 80% (charging process only). However, various implementation options must be investigated and efficiency levels estimated again. Some components (heat exchanger, heat pump, etc.) may have to be specially adapted or developed for the application, e.g. through simulation-based design using Ansys [15]. This is done taking into account all requirements for pressure resistance, tightness, etc. – cf. [16, 17].

For the simulation-based investigation of the respective optimization approaches, a simulation model was built and parameterised accordingly.

The first comparisons of measurement and simulation results show good agreement. However, the model is to be further detailed so that the individual factors influencing the efficiency during the charging process can be considered in a differentiated manner. For this purpose, the measurement setup is currently being equipped with additional sensors and the measurement data acquisition is being realised with an S7 module from Siemens. The measurements will then be carried out again. The results will enable further statements to be made on individual efficiencies, e.g. the stage efficiencies of stages 1, 2, and 3.

Subsequently, the different approaches to optimization with the improved model will be examined and evaluated with the aid of simulation (e.g. with regard to overall efficiency, complexity, service life, economy, etc.). Finally, the most favourable approach will be worked out, missing components will be developed or adapted, the plant will be built and the improved system will be examined again with regard to the overall efficiency.

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