A Thesis submitted for the degree Master of Science

Study of Compressed Air Energy Storage (CAES) for Domestic Photovoltaic Systems

B.Eng. Moritz Raible

Advisor
Prof. Dr. Paulo Fontes

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Instituto Politécnico de Setúbal, Escola Superior de Tecnologia de Setúbal

I Dimensions

A	$[m^2]$	Area
c, c_p, c_v	$[\frac{J}{kgK}]$	Specific heat capacity, isobar or isochoric
E	[J]	Energy
E_{el}	$\left[\frac{kWh}{d}\right]$	Electric energy
F	[N]	Force
h	$\left[\frac{J}{kg}\right]$	Specific enthalpy
h_{solar}	[h]	Solar hours
I_{solar}	$\left[\frac{W}{m^2}\right]$	Solar radiation
m	[kg]	Mass
$\dot{m}$	$\left[\frac{kg}{min}\right]$	Mass-flow
n	[pcs]	Number, quantity
n	$[-]$	Polytrophic exponent
p	[bar]	Pressure
p_a	[bar]	Ambient pressure
δp	[bar]	Pressure difference
P	[W], $\left[\frac{J}{s}\right]$	Power
P_a	[W], $\left[\frac{J}{s}\right]$	Adiabatic power
Q	[J]	Heat
$\dot{Q}$, $\frac{\delta Q}{\delta t}$	[W], $\left[\frac{J}{s}\right]$	Heat-flow
R_i	$\left[\frac{J}{kg \cdot K}\right]$	Individual gas constant
s	[m]	Distance
t	[s]	Time
T	[K], $[°C]$	Temperature
T_{sc}	[K], $[°C]$	Temperature state change

U	[J]	Internal energy
u, v	$\left[\frac{m}{s}\right]$	Velocity
V	$[m^3]$, [l]	Volume
$\dot{V}$	$\left[\frac{m^3}{min}\right]$	Volume-flow
W	[J]	Work
W_t	[J]	Technical work
W_U	[J]	Useful work
W_V	[J]	Volumetric change work
W_a, L_a	[J], [mkg]	Adiabatic work
W_{is}, L_{is}	[J], [mkg]	Isentropic work
α	$\left[\frac{W}{m^2K}\right]$	Heat transfer coefficient or convection coefficient
η	$[-]$	Efficiency
λ	$\left[\frac{W}{m^2K}\right]$	Heat transfer coefficient or conduction coefficient
μ	$\left[\frac{m^2}{s}\right]$	Kinematic viscosity
ν	$\left[\frac{kg}{m \cdot s}\right]$	Dynamic viscosity
∇	$[-]$	Nabla differential operator (here for divergence)
κ	$[-]$	Isentropic exponent
ρ	$\left[\frac{kg}{m^3}\right]$	Density
ζ	$[-]$	Dimensionless pressure coefficient

II Abbreviations

ASME	American Society of Mechanical Engineers
CAES	Compressed Air Energy Storage
CERN	Conseil Européen pour la Recherche Nucléaire
DN	Diamètre nominal in [mm]
HX	Heat Exchanger
ISO	International Standard Organisation or International Organisation for Standardization
PN	Pressure nominal in [bar]
PV	Photovoltaic
RPM	Rounds Per Minute, drive or revolution

III Agradecimentos

This work originated because of two main things: the wish to live in another southern European country and the ideal that everybody has the chance to make the world a little bit better. In my case this improvement focused on technical things because of my profession and passion. These two wishes could not have happened without the successful interaction of several different people and opportunities, to which I am deeply grateful. Thanks to our technical progress, it was possible for me to travel between Portugal and Germany. In this context I want to thank mainly the airline TAP Portugal, which has great connections between Lisbon and Zürich. Thanks to European political progress and solidarity it was possible to live and study in Portugal. I am thankful to live and be part of such a great and peaceful community, the European Union, which enables easy movement in all the member states. As we see daily in the news, this achievement cannot be taken for granted. It has to be encouraged through active exchange and protected by us, to be saved for future generations. I want to thank the pleasant Portuguese people, especially my fellow students and neighbours in Oeiras, for the great time in a beautiful country. In particular I want to thank Prof. Dr.-Ing. Michael Butsch from HTWG Konstanz, who gave me the idea to contact the University of Applied Science "Instituto Politécnico de Setúbal" for a Masters course, which is a partner of HTWG Konstanz where I received my Bachelor of Engineering in Mechanical Engineering. Prof. Dr. Luís Coelho who welcomed me warmly in the Politécnico in Setúbal and always had an open ear and a helping hand for any challenges regarding the Masters studies in Energy or Mestrado em Energia. My line manager, Jörg Hunnekuhl, from Georg Fischer Piping Systems in Schaffhausen, Switzerland, who understood my reasons for leaving the company for the Masters course and for living in another country. I was allowed to work externally for his Research and Development Group which allowed me to finance this plan. Especially I want to thank my advisor Prof. Dr. Paulo Fontes, who was interested from the first idea in this subject. He always supported me in technical and organisational challenges. After the initial phase I moved back to Germany. The distance was never a problem. Every time I was in Portugal during this time, we had good constructive meetings.

IV Resumo

Este trabalho é o projeto final do Mestrado em Energia. A mudança de combustíveis fósseis para energias renováveis é uma grande tarefa para a nossa e as futuras gerações de modo a parar o aquecimento global e impedir mudanças drásticas no ambiente no nosso planeta. Como o armazenamento de energia é um parâmetro muito importante na utilização de energia renovável, este estudo tem por objetivo estudar em termos termodinâmicos um destes sistemas. Para a implementação de um sistema de armazenamento de energia por ar comprimido este trabalho permite também fornecer uma base para o estudo económico e de rentabilidade. Neste trabalho é calculado e dimensionado um sistema adiabático de armazenamento de energia por ar comprimido (AA-CAES) para um sistema fotovoltaico doméstico localizado em Lisboa, Portugal. Apresenta-se uma breve história sobre a compressão de ar, estado da arte científico e os desenvolvimentos atuais. O contexto teórico para a compressão de ar é explicado com base num sistema AA-CAES, sendo apresentadas as fórmulas usadas para dimensionar um sistema doméstico e apresentadas e explicadas as suposições assumidas. O sistema é dimensionado numa folha de cálculo Microsoft Excel tendo sido selecionado um compressor de mergulho e estudado o seu funcionamento com temperaturas ambiente entre 0 $°C$ e 50 $°C$. As possibilidades de armazenamento de calor são explicadas e o processo de expansão na válvula de redução de pressão selecionada é simulado através de CFD.

V Abstract

This work is the final Thesis for the Mestrado em Energia study.
Changing from fossil fuels to renewable energy is a big task for our and for future generations. Stop global warming to prevent our planet from drastic changes in the environment. As energy storage is a big challenge in Renewable Energy, this study is made to estimate the thermodynamic sense of such a system. In order to set up the system, this work also gives a basis on which to make a rough economic calculation of the rentability of such a system. Because of this, a possibility to store energy when it is produced for example from photovoltaic cells until it is required is necessary. Does it make sense to sell energy to the grid? Is the possibility to store energy in pressurised air meaningful? To compress air with a compressor when there is energy to run it, and re-use the pressure energy via a turbine when it is required sounds a little bit like a pump storage power plant with air instead of water and without a mountain to get the potential energy. Air compressors produce heat compared to water pumps which work almost isotherm because water is incompressible. How can the waste heat of air compression be used to increase the system efficiency - an adiabatic system? This study is made to estimate the thermodynamic frame of such a system. How does the compression of air work? Which components are required for the full system? Which would be the size of such a system for one person? Does it fit in a cellar and how much Photovoltaic area does it need to be fed? How high is the temperature, density, pressure in the different stages?
An Advanced Adiabatic - Compressed Air Energy Storage for a domestic photovoltaic system in the Lisbon area, Portugal, is calculated and given dimensions in this work. A brief overview over the history of air compression is given, actual scientific works and developments are shown. The theoretical background for air compression is explained with focus on an AA-CAES system. The formulas used to plan the dimensions of a domestic system and all assumptions are shown and explained. The system is calculated in Microsoft's Excel for one chosen scuba dive compressor with surrounding temperatures from 0 to 50 $°C$. The heat storage possibilities are explained. The expansion process in the pressure reduction valve is calculated and simulated with CFD.

Contents

List of Figures

List of Tables

1. Introduction

1.1. Technical

With the industrial revolution, many things changed in our lives. Machines were constructed to help people to do their work. People used machines to work more efficiently. More things could be done in the same time. Machines were always controlled by humans because they had no "brain". Technological development in the past decades brought many faster, more precise and more efficient machines, but thanks to information technology, machines also gained something like intelligence. Simple machines just have a measure control system, which compares a real value with a given setpoint. This must come from humans who program the machine. This development goes on. The newest machines have artificial intelligence. They are learning from past works and implement this knowledge into new work. Nowadays some machines can totally replace humans in some areas, but these are still defined actions in a given control volume. In the future, machines will be more connected and share their "knowledge". This leads to a steep learning curve. It is only a question of time until machines will make decisions about how to act, and estimate and judge the influence of their decisions for the future.
Machines, tools, devices, or any technical equipment need energy. This energy is often transferred as electricity, because electricity can be generated and transferred by wires and cables over short and long distances. Machines don't need to have their own energy source, which makes them much more flexible and smaller.
One challenge, since the beginning of electricity is its storage.
Electricity flows when it is produced or generated by a generator, magnetic field or a photovoltaic cell. When the generator, the magnetic field or the sun stops, the electricity also stops.
The end user needs his energy source electricity, whenever it is required. Due to sleep electricity is required less at night. During the day behaviour patterns like going to school or work and having lunch-break at midday lead to typical average electricity needs. Figure 1.1 shows the typical electricity use in a German residential area over 24 hours. The typical curve is different on weekdays and weekends, and variation is part of the daily work of electricity suppliers. Curve "a" shows typical consumption on week days. The peak is at midday, presumably because of food preparation. Curve "b" shows the average consumption on weekends. The consumption begins at equal time like week days between midnight and midday but doesn't rise so high. The curve also falls down after midday with ups and downs until late evening when it drops down to the low night level.

Because of this requirement, the electric grid has been invented with central power stations that run a big generator and produce electricity 24 hours a day, 365 days a

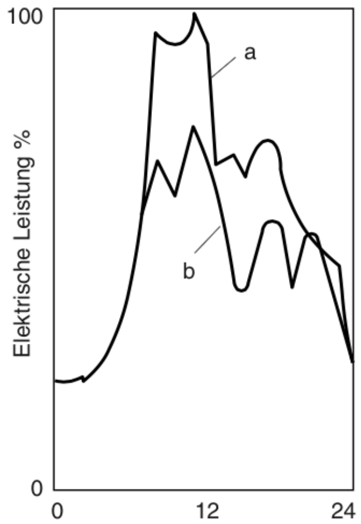

Figure 1.1.: Load course in german residential areas; a - working day, b - weekend [24, p.20]

year. Traditionally they run with fossil fuels like coal, oil or nuclear power. These types of power stations have the characteristic that they can't be switched off and on within seconds. The start up and shut down processes can take some hours. The problem with electricity is that once produced and given into the grid, the electricity has to be used. The grid can't handle unrequired energy because it has no storage, it is just the transporter. Thus, it is very important for the electricity provider, to have some compensation zones. A very good possibility are pump storage power plants. If there is too much currency in the grid, a generator is run which transports water from a lower to a higher level, and creates potential energy. If the grid has a need of currency, water from the higher basin forces the turbine and produces electricity. Because everywhere does not have mountains, which are needed to create sufficient potential energy, and not everywhere has enough water and space, the industry is looking for other possibilities to save and release energy for short term use. One possibility is CAES.

1.2. General

This work corresponds to the technical energy sector. CAES (Compressed Air Energy Storage) systems are a possibility to store energy, primarily from renewable energy sources, when it is available for short or long term until it is required. Energy storage is a big challenge of our time because there is not a one and only good technology. Furthermore this challenge only appeared a few years ago. Our energy sector is constantly changing from fossil fuels to renewable, more sustainable energy sources. While electricity produced from fossil fuels can be regulated in time and amount, renewable energy like sun and wind can't be switched on and off. Thus, sustainable energy storages have to be developed to support renewable energies to grow and be a reliant partner in our daily life. CAES storages use electricity to run a compressor whenever it is available. This compressor feeds an air storage with pressurised air. This pressurised air can be expanded through a turbine to produce electricity, whenever it is required. The challenge of all energy storages is a good efficiency, reliability, economy and sustainability to ensure ecological and political senses but also to ensure market acceptance. Compressing air is an old technology compared for example to Information Technology and Micro Electronics. This gives it a touch of antiquity, but it can be combined with new technologies regarding noise emission, installation volume, efficiency, inspection intervals and time of using. Until batteries are not further developed regarding infinite loading cycles or high percentage of recyclability, other promising technologies like CAES should be forced for technological development. In air compression there is one main task to solve, which is that compression is an exothermal process. When air is compressed, a big part of the compression work is transferred into heat. Thus, this heat must be used to reach an acceptable efficiency. As shown later, warm air increases it's volume which leads again to more compression work. One option to use the compression heat is an adiabatic system, called AA-CAES (Advanced Adiabatic CAES) System. Another set up is Quasi Isothermal compression. Both technologies are shown in this work, an AA-CAES system is designed for a domestic application.

2. Bibliographic Revision of CAES

2.1. Early Use of Pressurised Air as Energy Storage and Power Source

Bern in Switzerland already used compressed air as energy source around the year 1900. Air driven trams were used for public transport. They were charged in a charging station and had a mobile tank in the rail cars. The city saw the gain in exhaust free streets.

2.2. State of the Art CAES Power Plants, Example Huntorf

At the moment there are two big commercially used compressed air energy storage power plants. Power plant Huntorf in Germany which was installed in the 1970s with a capacity of about 300 MW and McIntosh power plant in the USA installed around 1990 with 110 MW. In Sesta, Italy, in the 80s a test plant was installed with 25 MW, but after an earthquake it was closed in the early 90s. There was a second planned CAES power plant in Germany which should have been started in 2015 under the name ADELE. The concept was to use the experience from the Huntorf power plant and improve the efficiency with an adiabatic process. This means to store the heat during compression and use this heat during the expansion instead of heating with gas, which decreases the overall efficiency and environmental friendliness a lot. The Huntorf power plant is located in the north west of Germany. It is an air storage gas turbine system with a salt cavern as air storage. The power plant was built to store overproduced energy which is produced but not required at this moment. This energy can be released into the electric grid when peak needs are very high. This allows regular power plants to run more constantly on a lower level. It takes about eight hours to fill the cavern with around 72.000 t pressurised air. The turbo-group is designed in one shaft, as you can see in figure 2.1. The synchronous machine is positioned in the middle and can be connected to the thermal block on the left, or the compression group on the right side by a clutch, as you can see in figure 2.2. The compression group has a 4:1 charge ratio with 60 MW nominal power, a mass flow of 108 kg/s and 70 bar maximum pressure. The charge ratio is the ratio between charging and discharging time. Economic operation is reasonable between 1:1 and 4:1, [17, p843]. The low pressure compressor has 20 stages which are connected directly. The high pressure compressor runs at 7622 rpm with six radial stages. After the low pressure compressor and after every second hp stage, air-water-coolers are installed. The synchronous machine is a standard generator with water cooling in the stator and a hydrogen cooled rotor. The most important parts of the thermal block are two turbine groups in series each with a previous combustion chamber. The high pressure turbine has

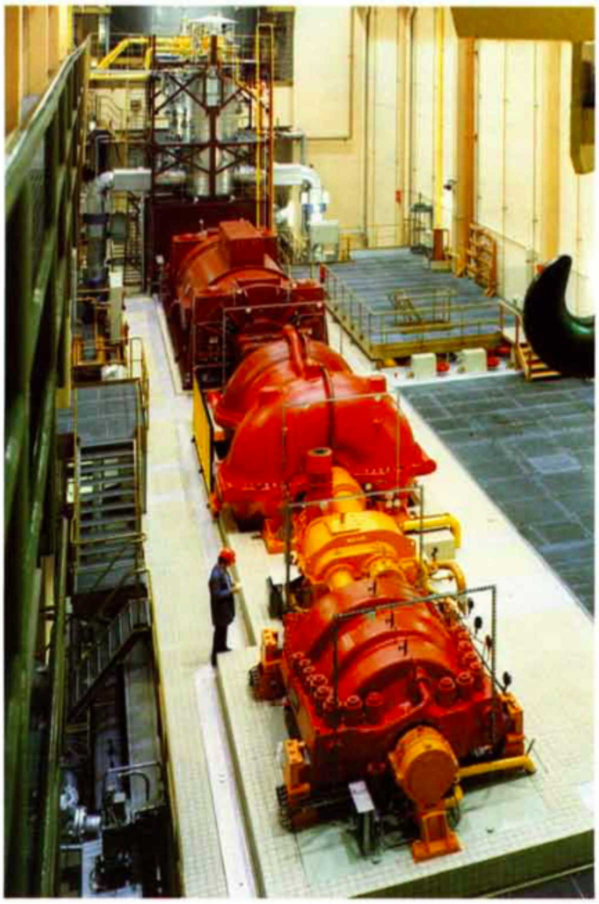

Figure 2.1.: The main shaft of the Huntorf compressed air power plant [7, p.467]

an entrance pressure of 42 bar and 550 $°C$ entrance temperature, the low pressure turbine has 11 bar and 825 $°C$ entrance pressure and temperature. The low pressure turbine is a standard opened gas turbine with five stages and 76 MW. Instead of the combustion air compressor, there is a six stage high pressure turbine before. The entrance temperatures are made by two combustion chambers, fired with gas. Also a recuperator is used from the exit of the low temperature turbine (400 $°C$) to preheat the stored air before the entrance of the first combustion chamber. The whole system was built with known parts, but because of the special use there where the following interesting points that had to be proofed before. The two turbines on one axel, a combustion chamber with 42 bar, initial firing with high combustion-air velocities, quick changing temperatures in the high pressure turbine and the synchronous machine normally only used as generator is used as motor.

2.3. Heat Recovery

Heat recovery means a technique to recover waste heat from an outgoing mass flow of a system. Recovering waste heat to use it for the same system or for another one, increases the overall efficiency because the heat is not "lost" to the surrounding. The techniques can be classified by their heat exchanger. There are for example regenerative and recuperative systems. Recuperative heat recovery systems are transferring heat directly from exhaust air to the fresh air through the wall of a heat exchanger. Regenerative systems are using a liquid or solid intermediary thermal heat carrier. Heat is stored in a solid body and later given to the inlet air.

2.4. Isothermal Compression with a Liquid - Company CAEstorage

The Company CAEstorage developed a system which should work isothermal. Main parts are gas cylinders, filled with air on one side and a hydraulic liquid on the other side. In compression mode, a compressor increases the pressure of the liquid on the one side of the cylinder. Thus, the piston moves in the other direction and minimises the volume of the closed room with air, which leads to pressure increase. Reverse to generate electricity the air presses the piston into the fluid which escapes through the turbine. Because this process is on the one side an incompressible liquid and on the other side very slow, the company speaks of an isothermal process.

2.5. AA-CAES

Adiabatic CAES Systems intend to use the heat which is produced during compression for the pre heating of the pressurised air before the expansion. When no heat is lost to the surrounding it is adiabatic. State of the art CAES power plants like Huntorf are loosing a lot of energy because this heat is not used. Compression heat is given to the surrounding with cooling fans to cool the compression temperature for better compression efficiency and not overheating the compressor, but before the expansion pre heating is done with fire from natural gas. This required heat has to be counted into the system efficiency which drastically drops the overall efficiency and also the environmental friendliness. Advanced Adiabatic CAES system promise a much better efficiency and sustainability. AA-CAES power plants vary between 52 and 60 % after simulation and analysis of Hartmann et al. [16, p.541], and reach a value of 70 % in literature [16, p.541]

2.6. Isothermal Compression with Water Injection - Company LightSail

In California U.S., is a new company which is developing a compressed air energy storage system, to store energy when the grid has more energy than there is demand. The approach of the company is the optimum isothermal process, without volume increases as a result of the increasing temperature which leads to losses in the efficiency. The website [1] explains “We inject a fine, dense mist of water spray which rapidly absorbs the heat energy of compression and provides it during expansion.” Figure 2.3 shows the scheme. This water is evaporating from liquid to gaseous directly when it is injected. The phasechenge of a liquid to gaseous is an endothermic process. The company says that the amount of water and the size of the drops for the surface is very important for a good process. The warm water is injected into the expansion process to heat the exothermal expansion, shown in figure 2.4. The whole process is shown schematic in 2.5 To get the warm water which is sprayed in during expansion, the compressed air normally has to be dried with a condenser. The company also plans the transport of the heat energy generated during compression for district air conditioning (heating and cooling). Two prototypes were built until 2015. LightSail collected around 50 Mio. dollars private capital for example from Bill Gates (CEO Microsoft) and Peter Thiel (Famous Investor). [$Quelle_green.wiwo$] The compressor is a double piston boxer system which is also working as turbine. During a presentation in the CERN, the CTO claims it is the most efficient compressor ever built. Danielle Fong said that there will be five prototype installations until 2016 in the U.S. and one in France.

According to the website “air can be stored in simple, low cost air storage tanks, packed in a convenient shipping container from factor using industry standard pipes and matching ASME and ISO safety standards.” This is shown by figure 2.6.

Furthermore, LightSail says that for “truly massive installations, air can be stored in underground caverns which are the standard for large scale natural gas storages.” LightSail claims that experimental results have offered 300+ hours of reliable operation, around 10 $°C$ final temperature difference, 1000 RPM reciprocating piston compressor/expander and 250 kW highest power achieved. LightSail says that they “achieve very high thermodynamic efficiencies without sacrificing performance. "We have achieved these high thermodynamic efficiencies at higher RPMs than many thought possible. This is crucial to achieve low cost: the higher the RPM, the higher the power of the same machine, and the lower the cost per kW.” [1] According to the website, experimental results delivered more than 300 hours energy, the final temperature difference is around 10 $°C$, the reciprocating piston compressor/ expander operates with 1000 RPM and 250 kW was the highest power achieved.

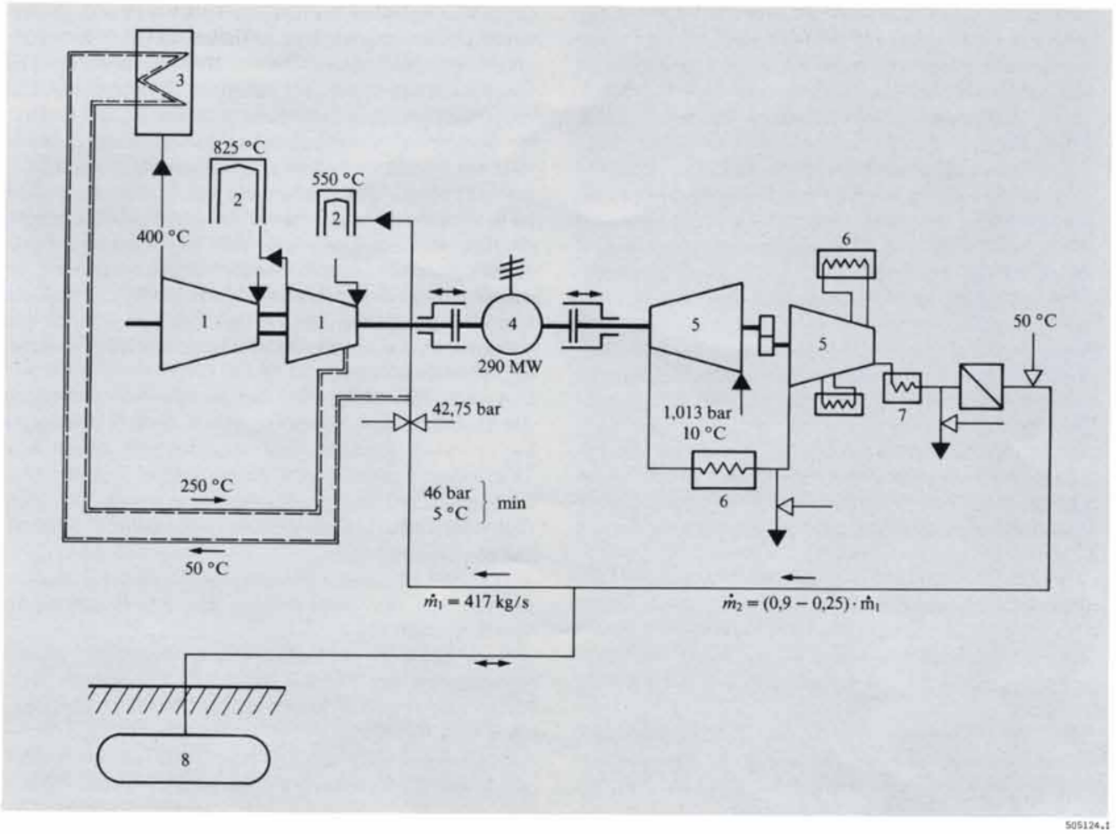

Bild 3 – Schema des BBC-Luftspeicher-Gasturbinen-Konzeptes

1 = Turbine turbine
2 = Brennkammer combustion chamber
3 = Rekuperator recuperator
4 = Generator generator
5 = Verdichter compressor
6 = Zwischenkühler intercooler
7 = Nachkühler aftercooler
8 = Luftspeicher mit konstantem Volumen (55…75 bar, 15…80 °C, 230 000 m³) oder konstantem Druck (50 bar, 65 000 m³), ausgelegt für 2 h Energieerzeugungsbetrieb air reservoir

$\dot{m}_1$, $\dot{m}_2$ = Massendurchsatz

Figure 2.2.: Technical scheme of the Huntorf power plant
[7, p.468]

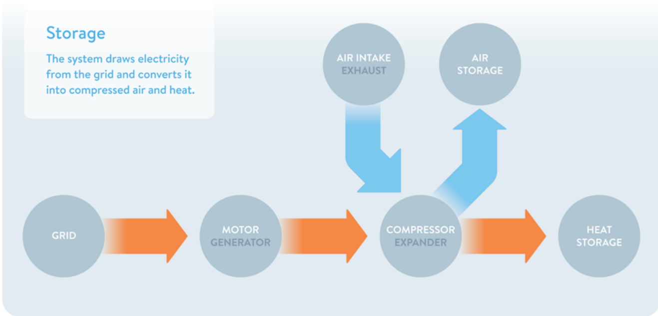

Figure 2.3.: LightSail compression system
[1]

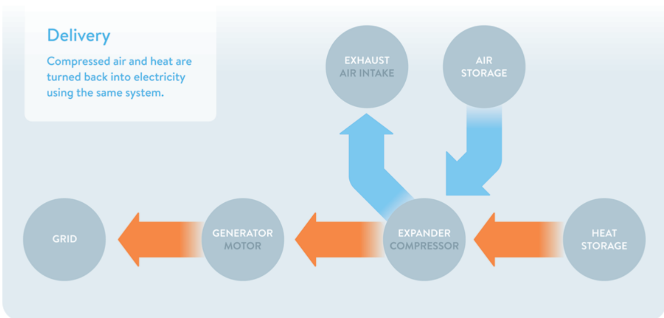

Figure 2.4.: LightSail expansion system
[1]

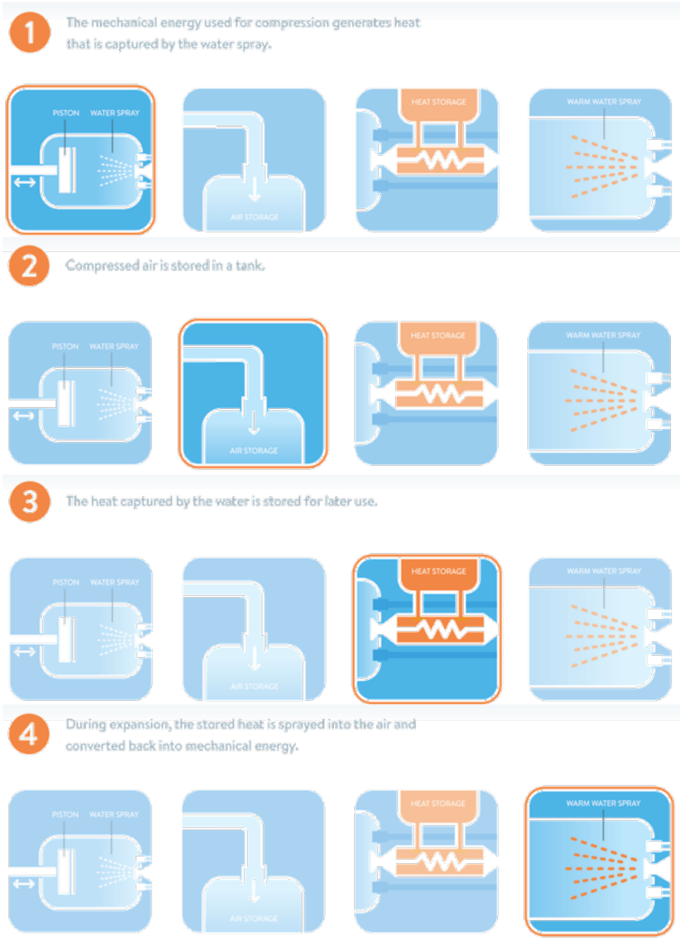

Figure 2.5.: LightSail scheme
[1]

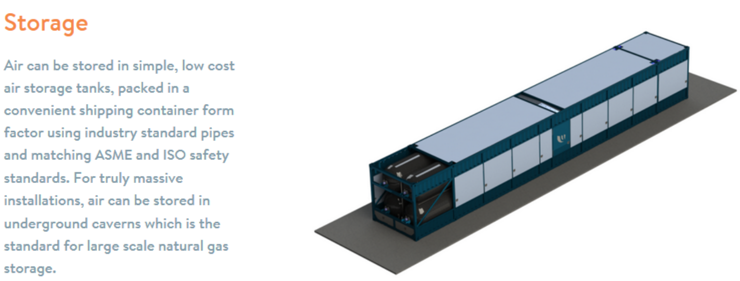

Figure 2.6.: Lightsail storage tank
[1]

3. Technical Background

For a better understanding of the following chapters, some key elements of a CAES System and definitions are explained in this chapter. It is important to have knowledge about the used techniques, the mathematical background and also the development of the economics related to this specific topic, to design and then judge such a system.

3.1. Photovoltaic Cells

Photovoltaic cells are made of semiconductor materials. Semiconductors have the characteristic, to be conductor or dielectric dependent on the temperature. The conductivity rises with increasing temperature. The typical material to produce PV cells is silicon. Silicon is made out of sand. It is possible to dope silicon with other elements, to change the characteristic. To create a PV cell which means an electrical flow, potential difference is required. That means, one side of the cell should be positive, the other negative. Therefore it is possible to dope silicon with boron. The resulting material is called p-Type. There are electron "wholes", because boron has only two valence electrons. This means, that there is space for two other electrons. A positive loaded side is created, because the positive protons in the core predominate. When silicone is doped with phosphors, the resulting material is called n-Type. In the n-Type, the 5th valence electron is not bound with silicon, and can move free. This means a negative side with electron dominance. When these two materials are put in contact, the free electrons from the n-Type will bond the free holes in the p-Type. The n-Type loses electrons, and the p-Type loses holes in the boundary layer. When the n-Layer is now emitted to the sun, electrons will go out of the n-material (semiconductor characteristics), which is called photoelectric effect. Because of the electron loss, the now also free "whole" goes back into the p-layer. The escaping electron flows through a bulb into the p-layer, to re fill the free "whole". See therefore figure 3.1. The bulb will light because electric flow is generated. The efficiency is steadily increasing, see diagram 3.2.

3.2. Economic Situation of Domestic PV Systems

Photovoltaic systems for domestic applications depend on the location and can be designed by the needs of every owner. In Europe, based in the northern hemisphere, the further south one goes, the more sun radiation one will find in average. Sun radiation is transferred into electrical energy by the PV cells, depending on the cell area. That means that one needs less cell area further south for the same amount of electricity, in

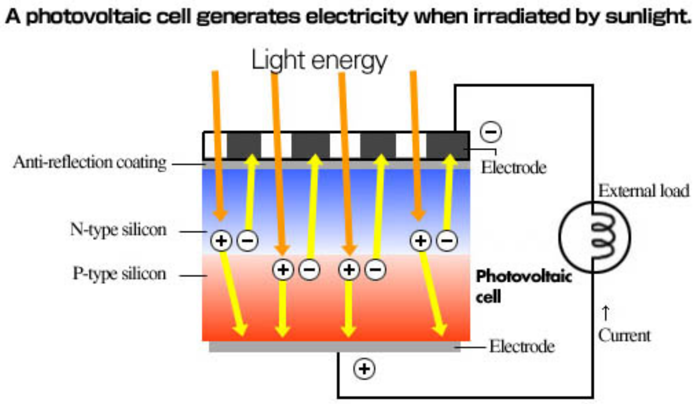

Figure 3.1.: Function principle of a PV cell
[2]

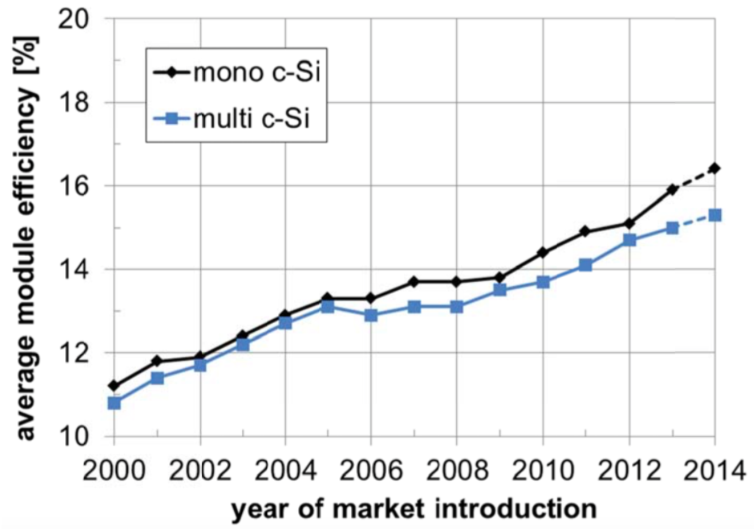

Figure 3.2.: Development of average efficiencies of PV-modules based on mono or multi crystalline cells
[21, p.40]

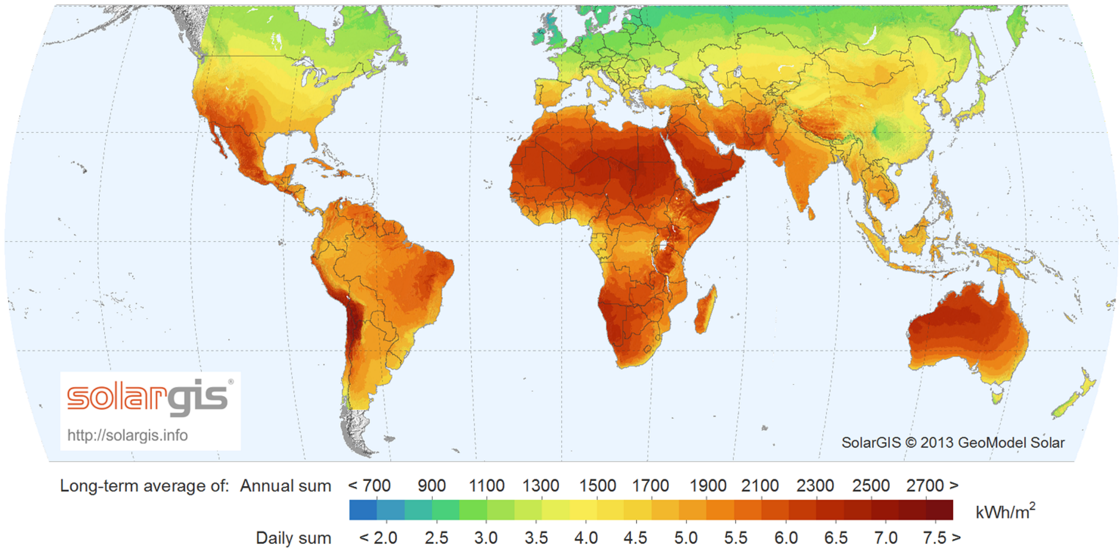

Figure 3.3.: Global annual solar radiation map
[20]

average. There are times or regions when this rule can't be used. Figure 3.3 shows the average sun radiation in Europe in $\frac{kWh}{m^2}$ per year and day.

Depending on the idea behind an installation (commercial, private or combined) and the average consumption, it is possible to calculate the dimension of the photovoltaic cell area and system size with map 3.3 and special tables available for each region . For example for a given output Energy, an installation in the Sahara (Africa) needs roughly $\frac{2}{3}$ of the cell area than in Portugal. And in Portugal roughly $\frac{2}{3}$ of the area than required in Germany. Formula 3.1 shows how to calculate the cell area required.

$$E_{out} = \eta_{cells} \cdot A_{cells} \cdot I_{solar} \tag{3.1}$$

The following calculation shows an example for how much area is required for 5 kWh per day in average in Portugal. According figure 3.3, mid Portugal has an average yearly sun radiation of 1700 $\frac{kWh}{m^2}$. The average daily radiation is 4,6 $\frac{kWh}{m^2d}$. With an average cell efficiency of 16 %, the required area is 18,4 m^2. In the past years, when new installations where supported by the European Union, installations where made to use the produced electricity directly by the owner but also as small power plants to sell electricity to the grid. The financial support for new installations, and guaranteed prices for 20 years after the installation, politics wanted a change the whole energy sector to green energy with less emissions and less nuclear power. After years of rentable selling of such produced electricity, it is no longer rentable to sell PV produced electricity. Figure 3.4 shows this development in Germany. The continuous lines are the governmental decided payments for new PV installations secured for 20 years after installation by year. The dotted lines is the development of electricity costs. The blue line is for small buildings, the

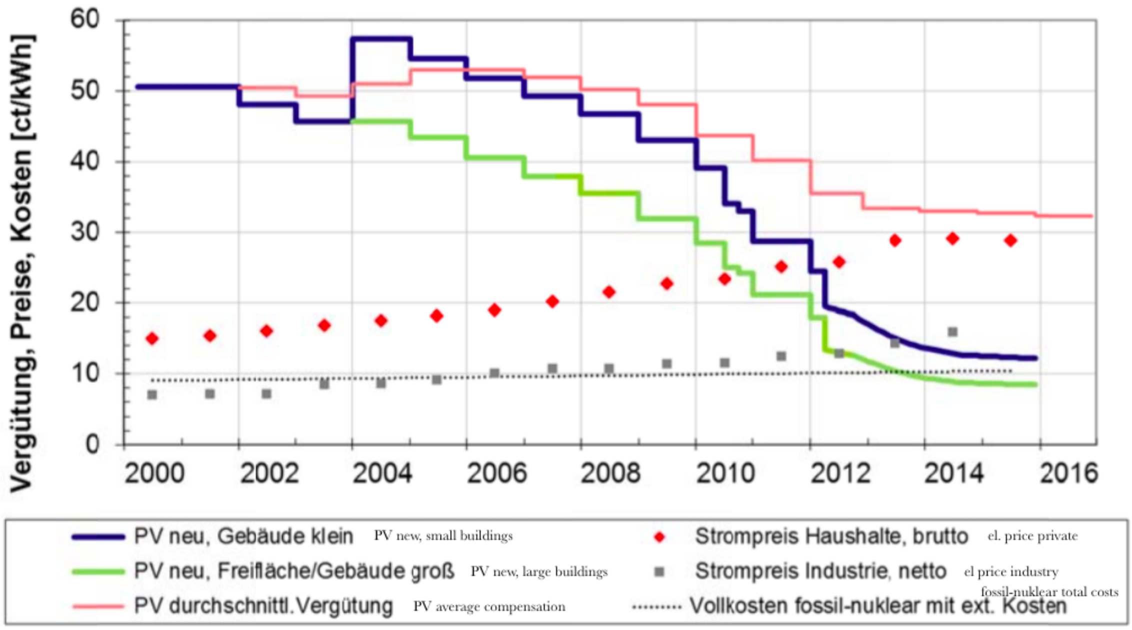

Figure 3.4.: Development of electricity costs and payments under EEG (Renewable Energy Act)

[21, p.10]

green line for installation on fields, the rose line is the average compensation. This line is higher because the payments are guaranteed for 20 years after installation. The red dotted line is the pre-tax price development for private households, the blue dotted line for industry. These curves are very important for new installations nowadays because they show the development to non rentability of selling electricity. New buildings with new PV installations earn around 10 $\frac{ct}{kWh}$ but pay around 30 $\frac{ct}{kWh}$. This is an efficiency of 33,33 %. Compensations where supported by the politic in the beginning of the new green technology to boost the development and installation of PV cells. Now that there are enough capacities, financial support is restricted. It is almost to speak of penalties for new installations, looking at the development.

3.3. Important Definitions

3.3.1. Heat Capacity

The heat capacity is a material property. Looking at a closed system with feed charge of heat, and work, the internal energy will increase as the first thermodynamic law for closed systems shows.

$$dQ + dW = dU \tag{3.2}$$

When we bring heat into a system, it will react without phase change with an increase of temperature.

$$dQ = C \cdot dT = m \cdot c \cdot dT \tag{3.3}$$

$$Q_{12} = m \cdot c \cdot (T_2 - T_1) \tag{3.4}$$

If the state change has constant pressure, isobar, we use c_p. For isochoric state changes c_v. The specific heat capacities c_p and c_v are temperature and pressure independent constants. The definition of the isentropic exponent is

$$\kappa = \frac{c_p}{c_v} \tag{3.5}$$

3.3.2. State Changes

Thermodynamic systems are divided in different phases or states between which state changes are occurring. If enough variables are known (e.g input ant output pressure) and the kind of state change is known, it is possible to calculate an unknown variable. Following are listed state changes and equations that are required to calculate the described system. A basic formula for all state changes is the ideal gas rule.

$$p \cdot V = m \cdot R_i \cdot T \tag{3.6}$$

Adiabatic State Change

At an adiabatic state change of a thermodynamic system, heat is not exchanged with the surrounding. Work that is done on the system leads to higher inner energy U.

$$\Delta U = W \tag{3.7}$$

$$p_1 V_1^{\kappa} = p_2 V_2^{\kappa} \tag{3.8}$$

With ideal gas equation 3.6 results

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{\kappa-1} \tag{3.9}$$

$$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{\kappa-1}{\kappa}} \tag{3.10}$$

$$\rho_1 = \rho_2 \left(\frac{p_1}{p_2}\right)^{\frac{1}{\kappa}} \tag{3.11}$$

Isotherm State Change

The temperature stays constant at an isothermal state change.

$$T_1 = T_2 = const. \tag{3.12}$$

With ideal gas equation 3.6 results

$$\frac{V_1}{V_2} = \frac{p_2}{p_1} \tag{3.13}$$

$$W = p_1 \, V_1 \, ln \left(\frac{V_1}{V_2} \right) \tag{3.14}$$

Polytropic State Change

Polytropic state change means the pressure p times specific volume to the power of the polytropic exponent V^n is constant.

$$p \cdot V^n = const. \tag{3.15}$$

Isothermal and adiabatic state changes are both perfect and thus reversible state changes without heat losses. Adiabatic compression means that the walls surrounding the compression room are perfect insulated. This is not or not yet reached in reality. Real compression is somewhere between isothermal and adiabatic. Figure 3.5 shows the T-s Diagram of polytrophic state changes. Isothermal state changes are horizontal and adiabatic changes vertical in the T-s Diagram, see figure 3.6.

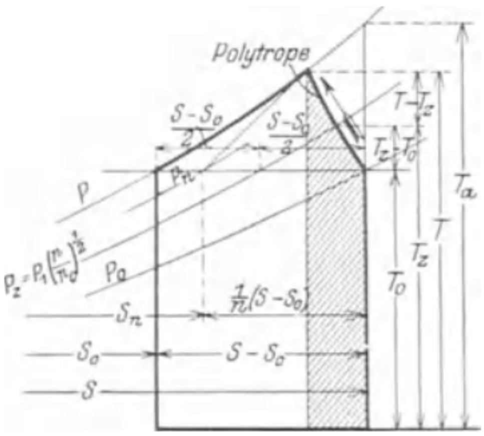

Figure 3.5.: Polytropic state change in the T-s diagram
[14, p.28]

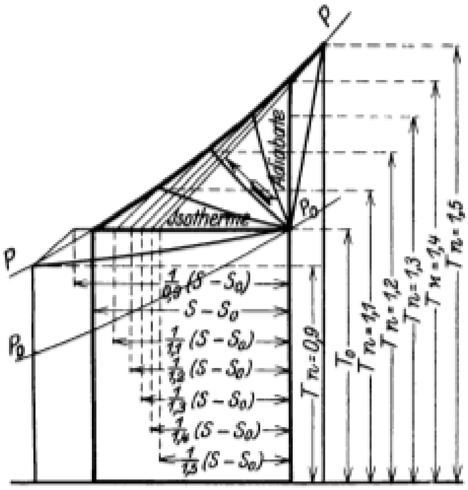

Figure 3.6.: State changes in the T-s diagram
[14, p.29]

3.4. Minor Losses in Pipe Systems

"The losses might not be so minor; e.g., a partially closed valve can cause a greater pressure drop than a long pipe." [18, p. 367]

The following equation describes the overall minor loss for a system with constant diameter.

$$\delta p_{tot} = \frac{\rho u^2}{2} \left(\lambda \frac{l}{d} + \sum \zeta \right) \tag{3.16}$$

To improve the CAES system efficiency, it is important to look at every single part of the system. One part where normally a lot of energy is lost, is the pressure reduction valve (PRV).

3.5. Compression

Compression means to take a given volume with it's amount of a fluid inside, and compress it to a smaller volume. A solid material is normally incompressible because the density is much higher and atoms are much closer. For compression, it is necessary to perform volumetric change work W_V. From the outside mechanical point of view, you can say that work is force times distance.

$$W = F \cdot s \tag{3.17}$$

From the inside mechanical point of view, force is pressure times area.

$$F = p \cdot A \tag{3.18}$$

Figure 3.7 shows volume change work on a simple example.

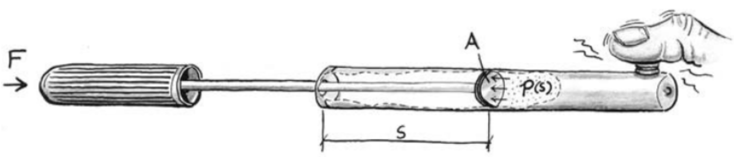

Figure 3.7.: Volume change work on the example of a closed pump [9, p.64]

A bicycle tire inflator or air-pump is working with an inside air volume, given by the circular area $A = \frac{\Pi \cdot d^2}{4}$ times the hub s. If an axial force F is given on the shaft, the inside pressure $p(s)$ is increasing constantly with the force because of the closed exit by the finger in the picture. We want to change the volume of a body against it's pressure. With these two formulas and the knowledge that area times distance is volume, it is possible to write the following universal formula for volume change work.

$$W = p(s) \cdot As = p(s) \cdot \Delta V \tag{3.19}$$

To respect the energy balance, on one side we bring positive counted work into the system (a negative amount work means work generation), and on the other side we minimise the volume which means negative sign. Referred to two stages, the formula gets the following.

$$W_{V,12} = -\int_1^2 p(s)Ads = -\int_1^2 p(s)dV \tag{3.20}$$

In difference to volume change work, effective work also includes the surrounding pressure. For example if you want to fill the bike tyre. The Pressure which is already inside the tyre works against the pump force.

$$W_{U,12} = W_{V,12} - p_a(V_1 - V_2) = -\int_1^2 p\ dV - p_a(V_1 - V_2) \tag{3.21}$$

With constant surrounding pressure p_U

$$W_{U,12} = -p(V_2 - V_1) - p_a(V_1 - V_2) = (p_a - p)(V_2 - V_1) \tag{3.22}$$

The previous examples show closed systems. If we want to use the pump example for an opened system like we have it in a compressor, we have to introduce the technical work. It is the pressure difference which a moved fluid experiences. Figure 3.8 shows the same example as previous but without the thumb. Technical work has to be made to let flow a volume against the ambient pressure.

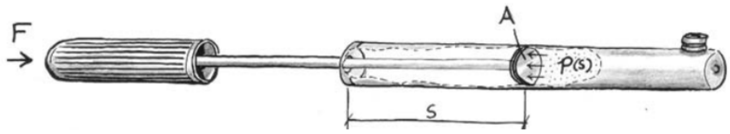

Figure 3.8.: Technical work on the example of an opened pump
[9, p.66]

$$W_{t,12} = \int_1^2 V \; dp \tag{3.23}$$

As the power is the derivation of work, technical power can be written as follows.

$$P_{t,12} = \frac{W_{t,12}}{d\tau} = \int_1^2 \dot{V} \; dp \tag{3.24}$$

A compressor brings work into our system. In the boundaries we want to stay with our CAES system, we can suppose that air is an ideal gas. [14, p.8] Because of the inserted work, our gas makes a state change according the ideal gas rule 3.6 which makes it much easier to calculate with. Because of the gauge pressure compared to the surrounding, the gas is later able to perform work. Compression is a state change and exothermal process. The air increases temperature during this process according to the ideal gas equation. Higher ambient and process temperatures lead to expansion of air which leads to volume increases. Higher volume of air leads to a higher amount of compression work, as you can see in equation 3.23 that work is a function of volume. The p-V diagram 3.9 shows this also qualitative. The work is the area beneath the curves.

A piston compressors can't pull out the whole air it soaked in, the detrimental space V' stays inside. When no heat transfer from this compressed air is done with the surrounding (adiabatic), the whole more compression work is gained back during the back expansion, see therefore figure 3.10. The indicator diagram shows the real compressor diagram for one stage compression. More work compared to the theoretical optimum must be done because of resistances in suction ducts and control ducts during compression and expansion. This more work is pictured hatched in the p-V diagram figure 3.11. The proportion of the hatched area to the theoretical white area is the compression efficiency $\eta_{compression}$

The pressure ratio is the relation between the initial pressure p_a and the pressure after compression p_e and calculated as follows.

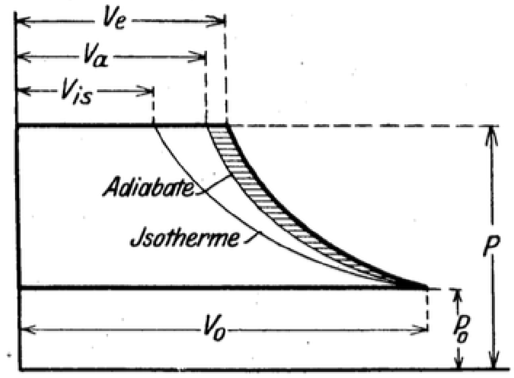

Figure 3.9.: Adiabatic efficiency of uncooled turbo fans in the p-V diagram
[14, p.20]

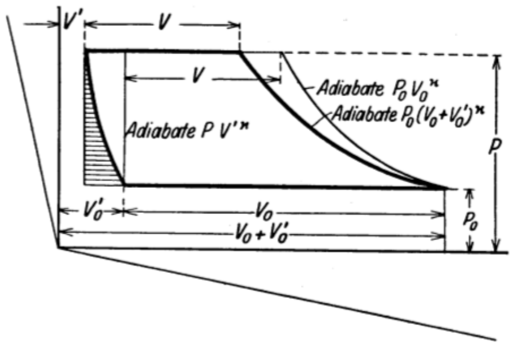

Figure 3.10.: Theoretical compression in the p-V diagram with the influence of detrimental space
[14, p.12]

$$p_r = \frac{p_e}{p_a} \tag{3.25}$$

From the theoretical point of view, the optimum compression and expansion cycle should be isothermal. Due to lower temperatures during the compression, less volume of air has to be compressed which leads to less technical work to reach the same pressure level. Also for machine parts it is good to deal with "normal" temperatures. Normal should be between 0 $°C$ and 50 $°C$. Very high or very low temperatures always mean possible problems with rubber parts like sealing gaskets or viscosity of lubricants. Normal compressors like screw or reciprocating compressors run with asynchronous machines. Due to the net frequency, asynchronous machines have around 1500 rpm. That means that without transmission, a reciprocating compressor compresses it's compression volume 1500 per minute, which means that one compression cycle has 0,02 s because the

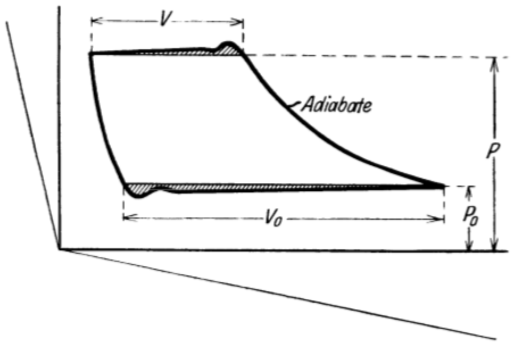

Figure 3.11.: The real compression p-V diagram (indikator diagram)
[14, p.13]

compression is only 180 ° from a full 360 °axle rotation. This means that the air doesn't have enough time in the compression or expansion chamber to transfer the heat to the outside. It is otherwise possible that even more heat comes inside because the compressor heats up because of friction in the system, especially the sealing gaskets with high surface speeds. For example the Bauer PE 100 compressor has a piston stroke of 24 mm and a cylinder bore in the first stage of 60 mm, which gives a volume of round about 68.000 mm^3 with surrounding pressure. Compressed to 7 bars in the first stage, we increase the temperature around 200 ° C. That gives an amount of heat of about 13 W. Also with perfect convection in the inside of the cylinder and perfect heat conduction in the solid material of the cylinder, around 325 W have to be carried away on the outside of the cylinder body with ventilation by forced convection. By perfect forced convection this means a mass of 1,6 kg air or volume of 1,6 m^3 in 1 s. With a ventilator of 25 cm diameter, this means an air speed of around 33 $\frac{m}{s}$ ideal, which is around 120 $\frac{km}{h}$. This speed would causes a lot of noise on the outside edges of the compressor. With cooling blades the speed can be reduced, but it still is a lot of heat that has to be carried away. The compression temperature is defined by the ambient temperature and the pressure ratio $\frac{p_{sc}}{p_a}$. If the inter-cooling is made completely to ambient temperature T_a, the stage compression temperature T_{sc} of each stage is the same, if the pressure ratio is constant. In the previous example, which is a three stage compressor PE-100 from Bauer, stage one pressure ratio is

$$pr_1 = \frac{p_{sc}}{p_a} \tag{3.26}$$

The pressure ratio of n compression stages is

$$p_{r,n} = \sqrt[n]{\frac{p}{p_a}} \tag{3.27}$$

For a domestic house, a compressor with 2.2 kW nominal power like the Bauer PE 100 is a possible pressure source which can be driven by Photovoltaic Cells. The compressor is working in three steps, compressing air to 200 or 300 bar. The schematic figure 3.12 shows the construction of the PE 100. The inter and after cooler is working with an air ventilator.

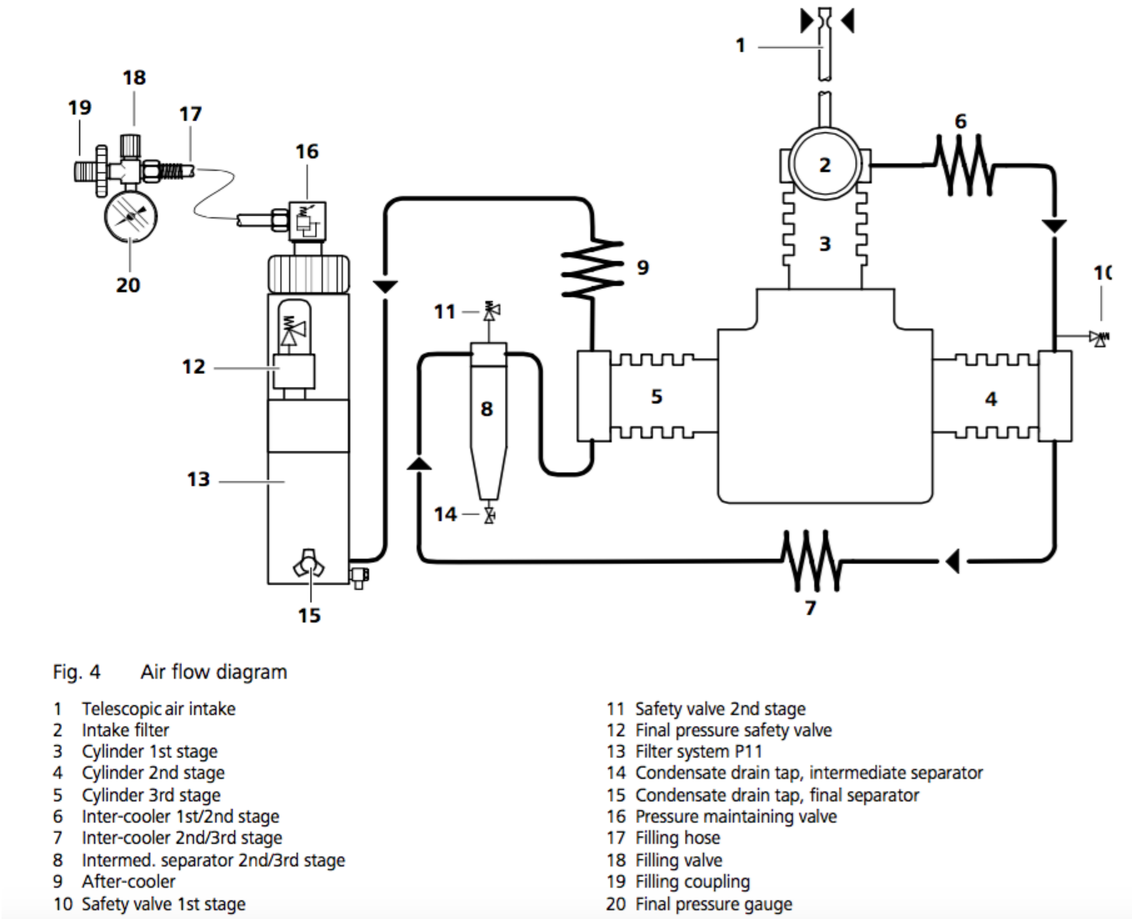

Figure 3.12.: BAUER PE 100 compressor schematic
[13, p.3]

3.5.1. Isentropic Compression

The optimum and reversible isentropic process is without any losses to the surrounding. Isentropic means with constant entropy. In figure 3.13 shows compression in the T-s diagram. From the start point 0 we go vertically up until we cross the $p-e$ line. If we want to let the gas work for us now, we can expand it isentropic, and get out the same amount of work that we had to use for the compression. In reality this process is unfortunately not possible because of dissipation work like friction which leads to heat losses.

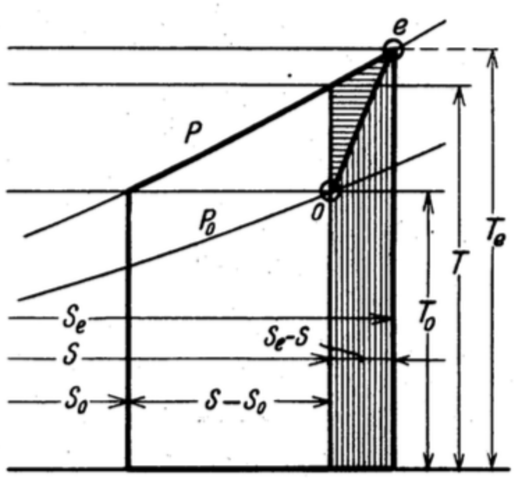

Figure 3.13.: Compression in the T-s diagram of an uncooled turbo fan [14, p.20]

The optimum compression is isothermal or isentropic with a relative low ambient temperature, which is not to see on the p-V diagram. A gas in initial conditions T_0, p_0 and s_0 is compressed by work through the compressor. The pressure rises to p_e. Because compression is an exothermal process the temperature rises to T_e, and entropie $\Delta s = s_e - s$ is generated to s_e. The arising entropy is mostly caused by friction losses.

3.5.2. Isothermal Compression

As shown in figure 3.13 from initial point 0 we go horizontally to the left side until we cross the p_e line. Isothermal means the temperature stays constant. A very slow compression and expansion process without friction would perform like this. The process has to be very slow because of the generated heat during compression, and fluctuating heat during expansion which has to be carried away or brought into the compressor or turbine. If these processes are very slow, the compressor walls have enough time to transport the heat to the surrounding to stay isotherm. The process is also shown in the p-V diagram 3.9. As technical work is the integral of volume over the pressure, see equation 3.23, if the pressure difference is the same, the work is less if the volume is less. It is the difference of the areas under the curves "Adiabatic" and "Isothermal" in figure 3.9.As you can see, the isothermal compression means the minimal possible technical work, which increases the efficiency of such a system. A possibility to compress isothermal in a normal gas compressor with very fast compression times is to inject or spray water into the compression room during the process. The water evaporates immediately which leads to a cooling effect, because evaporating is an endothermic process. This technology has already been discussed in [14, p.31] in 1927: "In practice it was tried to approach the ideal, isothermal compression by spaying in water during the compression. But the fast wearout of the grinding parts led to be content with a cooling jacket." Compressors can also get serious problems like explosion when incompressible media is in the compression chamber. That's why compressors are always designed to stay above the dew-point of air. That means that the spayed in water must totally change into gaseous phase. For example the German carmaker BMW is using this technology in the new 2015 M4 Safety Car to increase the efficiency of the combustion engine. Water is directly sprayed into the combustion chamber. The water comes out of a separate tank in the car. If it is empty, the power of the engine decreases. The workload of an isothermal compressor is

$$L_{is} = \int_{p_0}^{p} V\,dp = p_0\,V_0 \int \frac{dp}{p} \tag{3.28}$$

$$L_{is} = p_0\,V_0\,ln\frac{p}{p_0}\,mkg \tag{3.29}$$

The isothermal workload is very important to know, because this value is the theoretical minimum workload for a compressor to compress in specific conditions. This value can then be compared to get the isothermal efficiency as follows.

$$\eta_{is} = \frac{L_{is}}{L} \tag{3.30}$$

3.5.3. Adiabatic Compression

The previous chapter shown that the technical work of isothermal compression is less than the work of adiabatic compression. In an ideal compression and expansion process, the amount of compression heat that leads to higher volume and due to that higher work decreases the efficiency of the process. As written in the previous chapter, isothermal compression is not possible for normally used compressors like screw compressors or reciprocating compressors because of the high rotation speed. Heat doesn't have the time to flow out or in.

Adiabatic compression means that no heat is transferred with the surrounding of the system. When compressed isentropic, which the ideal compressor does, temperature of the working fluid increases as shown before in figure 3.13. When expanded isentropic, which the ideal turbine does to produce electricity, temperature decreases because expansion is an endothermic process. The heat amount which is produced during the compression is exactly the same like it is negatively counted during expansion. It means that the fluid comes out of the turbine with the same surrounding temperature like it was soaked in before the process. In the p-V diagram 3.9 the compression is also compared with isothermal compression. The area between the two curves ("Adiabate" and "Isotherme") is the heat which is produced. This heat will be re-given into the adiabatic CAES process with the heating of the gas before the expansion, so that this energy is not lost for the system. Because heated gas needs more space than colder gas with the same pressure, it is more economic to store cold gas, because then it is possible to store more mass in the same volume. Normally the volume is limited by space factors and it is necessary to store as

much energy as possible in a given volume, which is called energy density.
To store cold gas, it has to be cooled after the compression with a heat exchanger and give the heat into a heat storage. The heat storing material must have good thermal characteristics like a high specific heat capacity.
Putting

$$V = V_0 \left(\frac{p_0}{p}\right)^{\frac{1}{\kappa}} \tag{3.31}$$

into equation 3.23 , we can convert to the following expression

$$L_a = p_0^{\frac{1}{\kappa}} \cdot V_0 \int_1^2 p^{\frac{-1}{\kappa}}\ dp \tag{3.32}$$

L_a is adiabatic work calculated after Hinz [14].

$$L_a = p_0^{\frac{1}{\kappa}} \cdot V_0 \frac{\kappa}{\kappa - 1} \left[p^{\frac{\kappa-1}{\kappa}} - p_0{}^{\frac{\kappa-1}{\kappa}}\right] \tag{3.33}$$

$$L_a = p_0 \cdot V_0 \frac{\kappa}{\kappa - 1} \left[\left(\frac{p}{p_0}\right)^{\frac{\kappa-1}{\kappa}} - 1\right] mkg \tag{3.34}$$

The unit $[mkg]$ is the old unit "meter-kilogram". The force is expressed in $[kg]$ instead of $[N]$. A Joule can be expressed as follows.

$$1[Nm] = 1 \left[\frac{kg \cdot m^2}{s^2}\right] = 1[J] \tag{3.35}$$

$[mkg]$ can be transferred into the standard unit nowadays $[J]$ as follows.

$$W[mkg] = 9.81[J] \tag{3.36}$$

$$W[kJ] = W[mkg] \cdot 9.81[J] \cdot 10^{-3} \left[\frac{kJ}{J}\right] \cdot \frac{1}{60} \left[\frac{min}{s}\right] \tag{3.37}$$

The given volume flows are always per minute, the equation for the power is.

$$P[kW] = \frac{W}{60} \left[\frac{kJ}{s}\right] \tag{3.38}$$

The compression temperature T_1 (T_z in figure 3.15) can be calculated as follows according to Hinz [14]

$$T = T_1 = T_0 \frac{p_1}{p_0}^{\frac{\kappa-1}{\kappa}} \tag{3.39}$$

With n compression stages and inter-cooling, the compression temperature which is equal for every stage can be calculated as follows.

$$T = T_1 = T_0 \sqrt[n]{\frac{p_1}{p_0}^{\frac{\kappa-1}{\kappa}}} \tag{3.40}$$

The power consumption for the first stage is according to Hinz [14]

$$L_{a,sc1} = p_0 \cdot 10^5 \cdot V_0 \frac{\kappa}{\kappa - 1} \left[\sqrt[n]{\left(\frac{p}{p_a} \right)^{\frac{\kappa-1}{\kappa}}} - 1 \right] \tag{3.41}$$

The total power consumption is [14]

$$L_{a,nsc} = n \cdot p_0 \cdot 10^5 \cdot V_0 \frac{\kappa}{\kappa - 1} \left[\sqrt[n]{\left(\frac{p}{p_a} \right)^{\frac{\kappa-1}{\kappa}}} - 1 \right] \tag{3.42}$$

3.5.4. Stage Compression and Inter Cooling

In practice was tried to reach the ideal isothermal compression by spraying in cold water into the compression chamber. The rapid wear-out of the frictional parts led to only use a cooling jacket. The positive influence of a cooling jacket is only measurable at small compressor dimensions. The time during compression is too short for the cold from the outside to influence the temperature in the centre. That means in big machines with a diameter bigger than 500 mm [14, p.31] is the compression process adiabatic despite cooling. The air is almost stagnating during the compression in a piston, which means heat flux is only out of free convection with low heat conduction coefficient α. Roughly interpolating the h-s diagram of air (Figure 3.14), isentropic compression to 100 bar leads already to a temperature of round about 500 $°C$. Compression up to 200 bar or more leads to very high temperatures which can exceed 1000 $°C$. That can only be handled with special materials which would make the system very expensive and would lead to very low efficiencies due to the volume or pressure increase.

Because this problem already exists since a long time, stage compression with inter-cooling has been developed. To compress a gas to higher pressures and reduce the temperatures, volumes and the technical work it is possible to use stage compression with inter-cooling. This means that the compressor has two or more compression stages, where the pressure increases to e.g. 7 bar in the first stage, is then cooled down to ambient temperature and then compressed to 50 bar in the second stage. Figure 3.15 shows the T-s diagram. If you start compression at T_0 and p_0 and than straight compress until final pressure p, a horizontal line in the point of interception marks the high temperature in the axis of ordinate. Then temperature decreases in the storage after some time. The area (Integral of V dp) underneath these lines is the technical work, see equation 3.23. If you compress to p_2 (with T_2), then intercool back to T_0 and then compress in a second compression stage to final pressure p, the area under the curves is smaller which means less technical work for the same pressure ratio or a better efficiency.

The cooling jacket, often constructed as cooling blades with an air fan, has to keep the frictional walls and the lubricant cool. After the compression during the ejection cooling is much better because of smaller pipe circumferences and turbulences in the fluid. Lid cooling prevents heating of the inflowing air into the cylinder and thus increases the

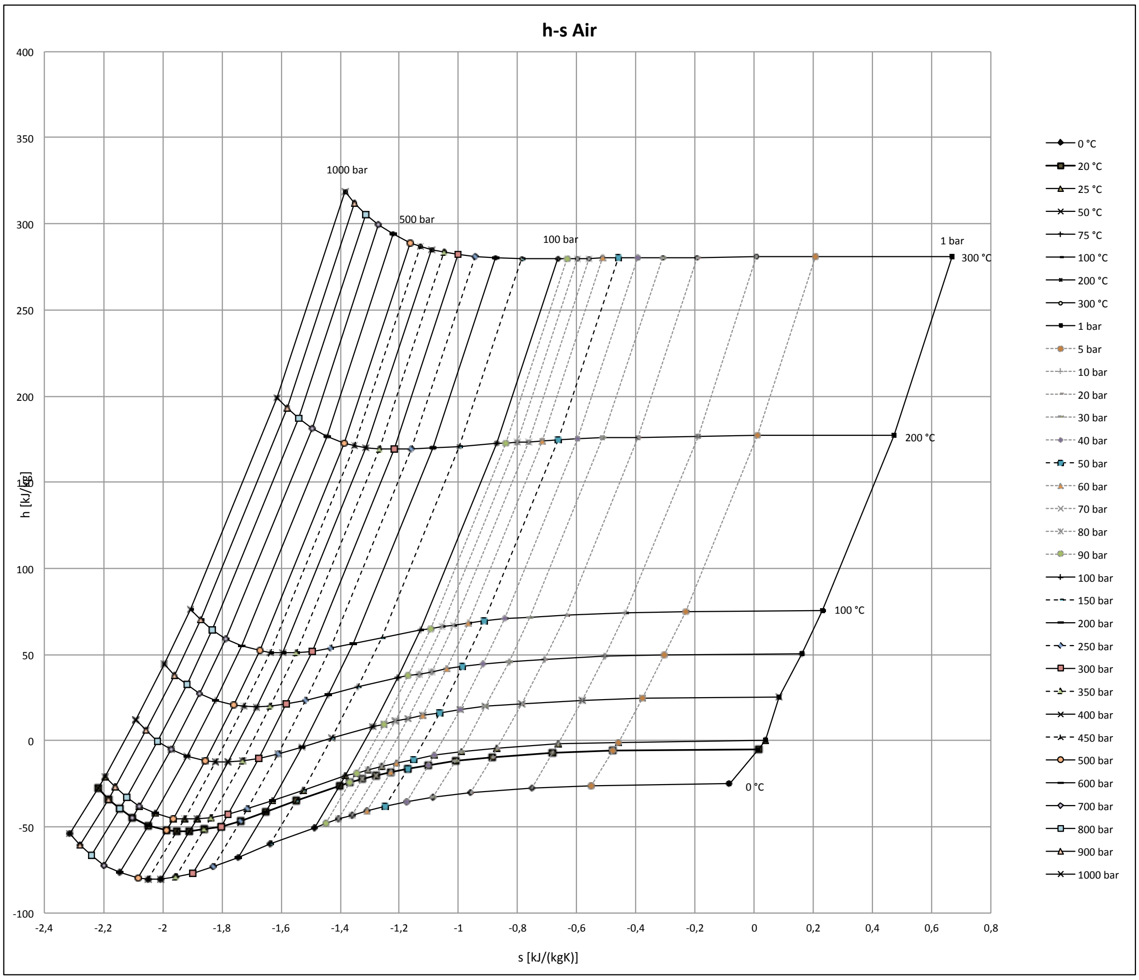

Figure 3.14.: The h-s diagram of air, data from VDI [15]

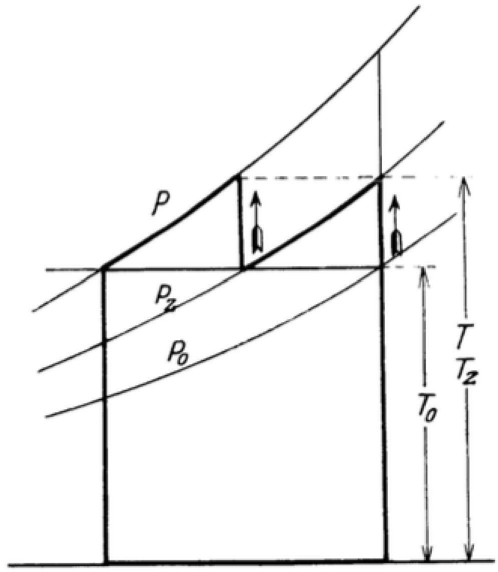

Figure 3.15.: The T-s diagram for two stage compression
[14, p.31]

volumetric efficiency. It is possible to calculate the temperature of each stage compression with the following equation according to Hinz [14].

$$T_{sc} = T_0 \frac{p_{sc}}{p_0} \tag{3.43}$$

During the compression and therefore temperature rise and therefore volume or pressure increase

3.5.5. Incomplete Inter-Cooling

If the outlet air temperature of one stage is higher than the inlet temperature was, the following stage has to do more work because of the higher volume. Incomplete intercooling only matters with stage compression. Figure 3.16 shows the increased workneed in the T-s Diagram. Because the temperature range of compression is usually in the region between 27 and 300 $°C$, around 3 $°C$ less back cooling means around 1 % more volume work in the next stage, vis versa. More work for the same amount of pressurised air means reduced compression efficiency. Because all stages make almost the same work, in a two stage compression 6 $°C$ more or less back cooling lead to 1 % more or less work [14, p.37] .

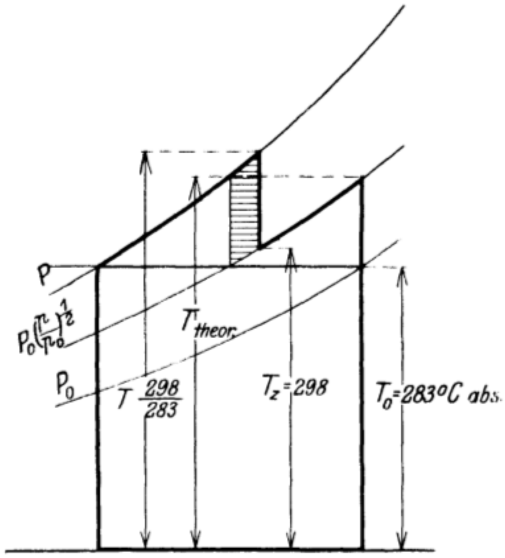

Figure 3.16.: Incomplete intercooling in the T-s diagram
[14, p.37]

3.6. Expansion

The provided work at adiabatic expansion of a gas from pressure p to p_0 has the same amount which is required to compress adiabatic from pressure p_0 to p. The end volume at the expansion is in accordance to the initial volume of compression. [14, p.12]

$$L_a = p \cdot \dot{V} \frac{\kappa}{\kappa - 1} \left[\left(1 - \frac{p_0}{p} \right)^{\frac{\kappa-1}{\kappa}} \right] mkg \tag{3.44}$$

To calculate the volume flow which is required for a given turbine power in adiabatic expansion, the formula is

$$\dot{V} = \frac{L_a}{p \frac{\kappa}{\kappa-1} \left[\left(1 - \frac{p_0}{p} \right)^{\frac{\kappa-1}{\kappa}} \right] mkg} \tag{3.45}$$

The T-s diagram for isothermal, adiabatic and polytropic expansion are shown in figure 3.17.

3.7. Energy Needs

Some basic equations are required to know how many compressors are required to cover the electricity needs, or how much electricity can be produced during one average day, see the following equations.

$$E_{el} = n_{pers} \cdot E_{el,capita} \tag{3.46}$$

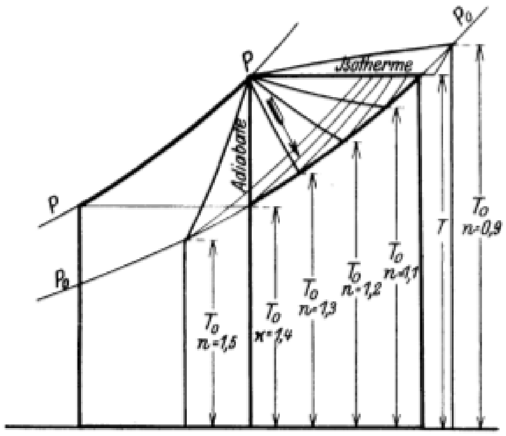

Figure 3.17.: Expansion in the T-s diagram
[14, p.29]

$$n_{compressors} = \frac{\dot{V}_{hp\ storage\ out}}{\dot{V}_{hp\ storage\ in}} \tag{3.47}$$

$$W_{total} = n_{compressors} \cdot W_{compressor} \tag{3.48}$$

$$W_{day} = h_{solar} \cdot W_{total} \tag{3.49}$$

3.8. Mixtures

Mean average temperature of three different mass-flows of the same substance for example in warm water storages can be calculated with the following formula.

$$T_2 = \frac{T_{1a} \cdot m_{1a} + T_{1b} \cdot m_{1b} + T_{1c} \cdot m_{1c}}{m_{1a} + m_{1b} + m_{1c}} \tag{3.50}$$

Including heat energy.

$$T_2 = T_1 + \frac{\dot{Q}_{1-2}}{\dot{m}_1 \cdot c_p} \tag{3.51}$$

4. Study of Heat Storages for CAES Applications

For the adiabatic CAES energy storage it is very important to find a good heat storage. Good means

high energy density to store much heat in a small volume

high thermal conductivity inside for fast charging and discharging

fast heat transfer means a good heat transfer coefficient and the possibility to form lamellas

cycle stability to use the storage over a long time

harmless material for the whole life cycle

good insulation to store the heat energy for long time

cheap material to be economic

high temperature differences to store heat in all required temperatures

Heat can be stored in different materials and with different techniques. It is possible to store heat in solid, liquid or gaseous materials. E.g. stone is a good heat storer. One of it's good characteristics is that it can store very high temperatures. Water for example is a very cheap heat storage that is totally harmless and perfectly adopts itself to any geometry to transfer heat. Figure 4.1 gives an overview over typical heat storages. All the previously counted materials store heat in one phase, which means latent. There is also the possibility to store sensible heat in a phase change of a material. Normally the phase change is from solid to liquid (endothermic process) and liquid to solid (exothermic process). The advantage of phase changing heat storages is that the material doesn't heat up during the heating process. all of the heat is stored in the phase change. This means that we don't have hight temperature differences compared to our environment, and according to the thermodynamic laws, only a few losses. A disadvantage of these materials is the price. Also the poisonousness, phase change temperature and cycle stability have to be proofed. A new way of storing thermal energy are metal organic frameworks (MFOs), which are highly porous and can store more than 1.4 times it's own weight with water. This water than can be released in form of vapour to cool, and in the opposite direction to bound humidity from the surrounding to heat [12]. This material seems to be made for building climatisation. The energy density and maximum temperature seem to be very low for technical systems.

State	Substance	$c_p \left[\frac{kJ}{kgK}\right]$
Gaseous	Air	1.012
	Argon	0.520
	Helium	5.193
	Methane (at 2 $°C$)	2.191
	Nitrogen	1.040
Liquid	Ethanol	2.44 0
	Gasoline (Octane)	2.22 0
	Mercury	0.140
	Methanol	2.140
	Oil	~ 2
	Water	4.181
Solid	Aluminium	0.897
	Concrete	~ 1
	Copper	0.385
	Gold	0.129
	Granite	0.790
	Lead	0.129
	Polyethylene	2.303
	Steel	0.466
	Uranium	0.116

Table 4.1.: Specific isobaric and volumetric heat capacities at 25 ° C for selected materials, data from wikipedia

4.1. Sensible Heat Storage

Thermal energy or heat that changes directly the temperature is called sensible heat. For example water between 0 $°C$ and 100 $°C$ in a closed tank will directly change the temperature when thermal energy is injected. This is shown by the equation 3.3 for heat capacity. With constant heat energy input, the temperature will increase constant, until a phase change occurs. If one material e.g. water has a specific heat capacity of 4,182 $\frac{kJ}{kgK}$, the temperature will increase less than a material with a lower specific heat capacity. For example concrete wit a specific heat capacity of around 1 $\frac{kJ}{kgK}$. The temperature of the water will increase around four times less than the temperature of concrete. Typically, solid materials have a lower specific heat capacity than liquids, see Table 4.1

Gaseous materials have a wide value range from low to high numbers. Because gasses have the characteristic of a very low density, they not recommendable for cheap and easy solutions in heat storage applications. Low density means that the system has to be very tight. Leaking connections are also hard to determine because the gas is not visible after the exit. As written before, the energy density of solid materials is very high, which makes these materials interesting to store heat. Also the normally very high phase change into liquid guarantees high maximum storage temperatures. The characteristic that solid materials are not leaking (because they are solid) is a good for heat storage applications because materials don't disappear or contaminate the environment anyhow. A problem that solid materials have is how to transfer the heat into them. If a concrete heat storage cube is positioned near the compressor, it is not possible to transfer heat as typically done with water or air - forced convection around the outer shelf of the compressor, because for this it is necessary to have a moving material. There are theoretically two options how heat can be transferred. First with very long blades from the compressor into the depth of the store with a lot of surface, that only heat conductivity is high enough for the application. Second is a circulating heat transport medium like water or oil. Because of the toxically harmlessness of water, an acceptable heat capacity, good circulating systems, accessibility , known characteristics and good price compared to other material prices, water is a good compromise. There are definitely better materials according to the technical requirements, but they might be more expensive, more toxic or harder to handle. If high energy density of a storage is a must, solid material might have better properties, which could be combined with water as transferring media.

4.2. Latent Heat Storage

Latent heat is the amount of heat, which is hidden or latent in the phase changing process of a material, body or system with constant temperature as shown in Figure 4.2. The quasi vertical red lines "Δ_{PCM}", temperature in the x-axis doesn't increase but the specific enthalpy does. The amount of heat can be written with a positive or negative prefix, depending on the direction of the process. For example boiling water is an endothermic process which means cold is generated, heat is required and water temperature remains constant at 100 $°C$ under normal conditions. Figure 4.2 sows a PCM material schematic with a phase change temperature of 40 $°C$. The specific enthalpy difference, which mirrors the thermal energy amount, is much higher with the latent part where the temperature is not increasing than the water line in the same temperature range between 30 and 60 °C. To be suited as a PCM (phase changing material) for heat storage applications in domestic houses, a phase changing temperature should be very roughly between 40 and 60 $°C$. If it is for a thermal solar panel or for a CAES system, the boundaries are comparable. The phase change should not be initiated from high surrounding temperatures, but close to the maximum surrounding temperatures so that only a small delta T is required to run such a system.

4.3. Thermo Chemical (and Sorption) Heat Storage

Thermochemical heat storages are storing thermal energy in reversible chemical reactions. They are charged or loaded with heat by an endothermic reaction, and discharged by exothermal reactions. There are function principles based on van der Waals forces and chemical reactions. For example Sorption storages are based on van der Waals forces. A

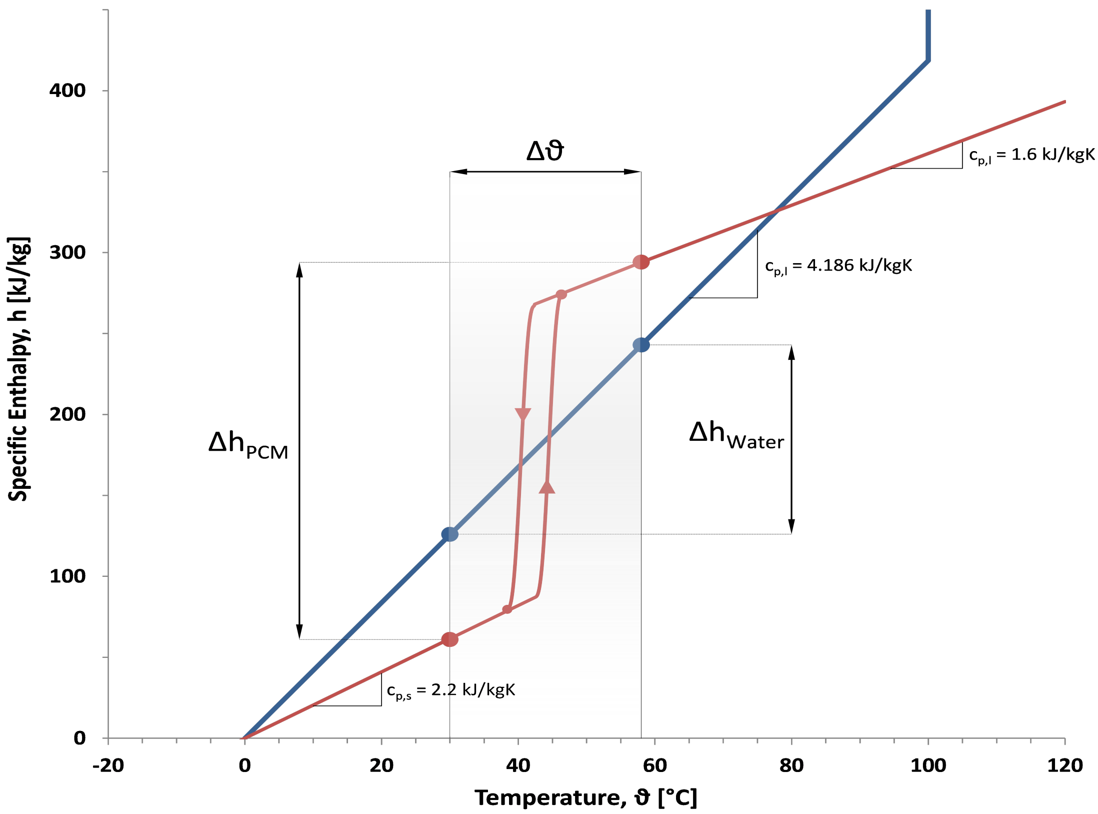

Figure 4.1.: Scheme of the storage capacity of a phase changing material compared with water

[22, p.4]

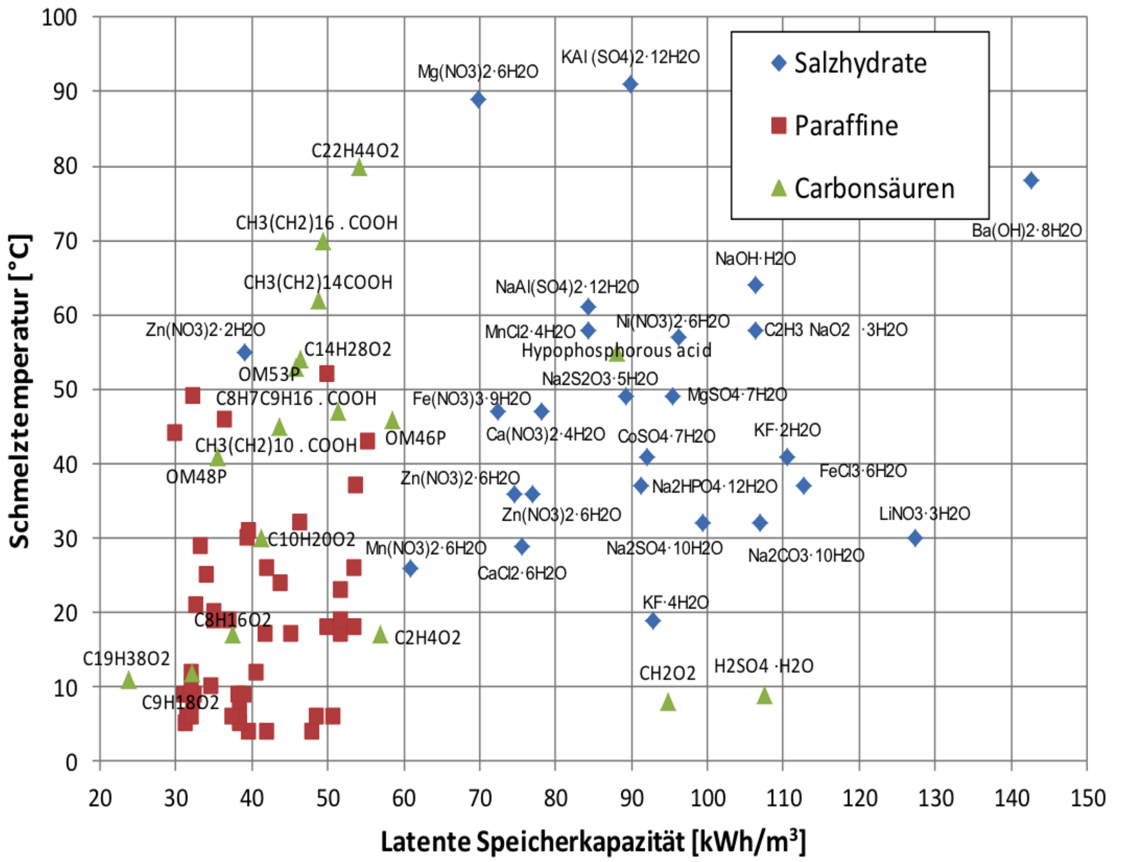

Figure 4.2.: Different storage materials - salt hydrates, paraffins and carboxylic acids; x - latent starage capacity, y - melting temperature
[23, p.29]

hygroscopic silica gel based granulate which is highly porous has a big inner surface (1 g has around 600 m^3 of surface). This material has the characteristic to attract water vapour which is adsorbed on the surface while heat is set free. In the opposite direction heat has to be used to dry silica gel, while cold is set free. A real example is tested by the German Research Centre for Air and Space Travel: "If calcium hydroxide, known as slaked lime, is heated, this creates calcium oxide at a temperature of approximately 450 degrees Celsius, because water is removed from the calcium hydroxide. During this reaction, around 80 percent of the input energy is stored in the form of chemical energy." These types of thermal storages are really good for long time storage. The chemically energy can be stored indefinitely without losses. To release the energy, water vapour is added and the strong exothermic reaction happens. For CAES systems, this storage might be too high-tech because of the required machinery like a vaporiser and the temperatures are very high, but further developments might bring more opportunities in the future. For example de-and rehydration of a Ca(OH)2 at high H2O partial pressures has high storage capacities at high temperatures like [11].

5. Theoretical Analysis of an AA-CAES System for Domestic Photovoltaic Installations

This chapter shows the thermodynamic states of one possible CAES system. After the boundary conditions, the calculated system is presented before discussing the single parts of it. A very important number for the whole system regarding energy density, capacity, costs and dangers is the maximum pressure. With the previous basic thermodynamic knowledge and information about state of the art commercialised pressurised air systems, a machine with 200 to 300 bar seems realistic. Scuba compressors produce around 300 bar maximum pressure. The PE 100 compressor from Bauer with 220 V voltage, 2.2 kW nominal power, 225 or 330 bar and around 2700 EUR seems to be a good machine for a domestic CAES system. The higher starting power has to be analysed and provided for example with a bypass system for the compressor, star-delta switch, a battery or capacitor. The heat transfer from the three inter coolers of the compressor to the warm water cycle have to be realised for example with three heat exchangers, which could be mounted instead of the metal coils used as standard. Standard gas cylinders can be used as pressure storage. Depending on the size and number, different storage volumes can be arranged. The storage system is also flexible regarding its volume. If one needs more capacity than the installed compressed air bottles allow, it is possible to install more bottles. The water volume must also be enlarged then because of the required amount of heat storage for the expansion process. The turbine can be an extra part, or the compressor working backwards, depending on the type. Every expansion reduces the temperature so that inter-heating could be an option to increase the efficiency.

5.1. Boundary Conditions

The following system is designed for one average Portuguese person (in matters of electricity use) in the Lisbon area. The data of one person is good to scale. In Table 5.1 some estimations are made to calculate the dimensions of the system.

5.2. Example User Type

To tailor a good system, it is important to know the behaviour of the user.

- If the user is not at home most of the working week and doesn't need a lot of electricity from Monday to Friday, but has many friends at home during the week ends

State	Assumption
Location	Portugal, Lisbon
Average sun radiation per year	1750 $\left[\frac{kWh}{m^2a}\right]$
Average solar hours	8 h
PV collector efficiency	20 %
Average ambient temperature	15 $°C$
Domestic Electricity consumption 2014 Portugal [4, p.36]	11'908 $[GWh]$
Portuguese population 2014 [19]	$10.401 \cdot 10^6$ people
Domestic Electricity consumption 2014 per capita	3.137 $\left[\frac{kWh}{d}\right]$
Compressor	Baur PE 100-T
System pressure	200/ 300 bar
CAES System electricity cover	100 %
Maximum outlet turbine power	2 kW

Table 5.1.: Assumed environment conditions and example configuration

that he reaches the average consumption Sunday evening, the storage capacities for heat and air have to be bigger because they have to store more air and heat energy, than the storage capacities of somebody who needs the same amount of electricity every day. Also the turbine has to produce more electricity than the average. The PV cell area must be the same.

- If a user is absent all week, and only needs the average daily electricity at the weekends, the compressor, the storage and the PV cell area can be smaller but the turbine needs the average dimension.

For the following system design we assume an average electricity consumption with equal balances all over the week. Our "client" has the same behaviour from Monday to Sunday.

5.3. System

Figure 5.1 shows the system scheme consisting of the three stage compressor (stage 0-1) Bauer PE 100-T, the legend is shown in Table 5.3 . Each of the three compression stages transfers the heat into the heat exchanger for inter cooling (stage 0.1-0.2 0.3-0.4 0.5-1) which cools down the compressed air and stores the heat in the water heat storage. The high pressure storage (stage 2) stores the pressurised air at ambient temperature. Until this stage, half of the system can be calculated and designed with the data from Bauer, see figure 5.11. When electricity is required, pressurised air is released (stage 3) through the heat exchanger to be pre heated (stage 4). The warmer air has more volume and can do more volume work in the turbine, and is not getting so cold in the following high pressure regulation valve (stags 4-5) that it can freeze and block its function. The low

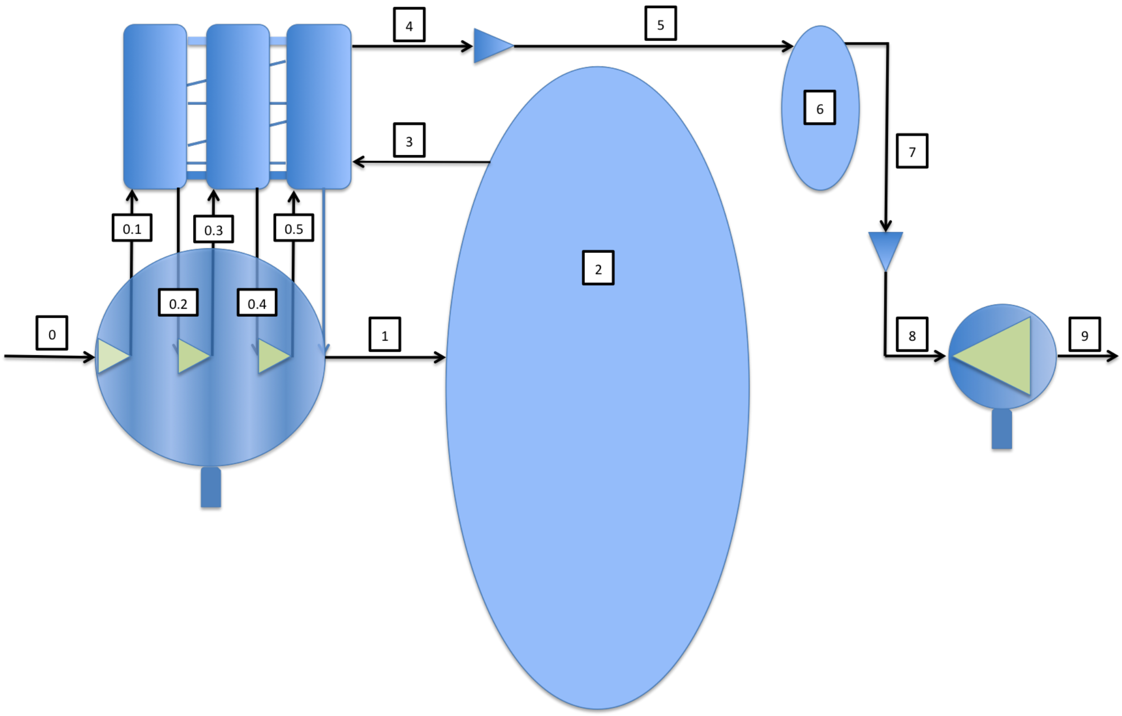

Figure 5.1.: The CAES System scheme, component description in table 5.2

pressure storage (stage 6) has more flow in (stage 5) than flow out (stage 7), that the high losses in the high pressure reduction valve which occur because of the geometry of the valve for the big pressure and temperature drop can be excluded from a efficiency calculation. The low pressure valve (stage 7-8) reduces the pressure to optimal turbine pressure (stage 8-9) and is released back into the surroundings at ambient temperature and pressure. Table 5.3 explains the Numbers for the diagrams, Figure 5.2 shows the referring h-s diagram.

System state	Diagram state	CAES System scheme
0	1	inlet
0.1	2	1st compression stage
0.2	3	1st inter-cooling
0.3	4	2nd compression stage
0.4	5	2nd inter-cooling
0.5	6	3rd compression stage
1	7	3rd inter-cooling and outlet compressor
2	8	main high pressure storage
3	9	hp storage exit
4	10	warm water HX outlet
5	11	high pressure reduction
6	12	low pressure storage
7	13	low pressure reduction
8	14	turbine
9	15	outlet

Table 5.2.: Explanation of state numbers "air system" in the System Diagrams

Following are shown the System diagrams which could be made after the complete system calculation. To speak about the different parts, the diagrams are first shown and explained because this knowledge has to be used for the specific components. The diagrams have a slightly different numbering of the stages compared to the system scheme, beginning with one and continuing straight. Figure 5.3 shows the pressure progression, Figure 5.4 the temperature progression for 15 $°C$ surrounding temperature. The temperature rises to 229 $°C$ in the first compression stage at 7 bar, 232 $°C$ and 50 bar in the second compression stage, and 155 $°C$ in the third compression stage at 200 bar. Operating at 300 bar outlet pressure, the third compression stage reaches 208 $°C$. The temperature in the compressor increase with higher surrounding temperature. With 25 $°C$ surrounding temperature, the temperature in the second compression stage reaches 250 $°C$. With 50 $°C$ surrounding temperature, the compression temperature in the second compression stage reaches 294 $°C$. Compared with 0 $°C$ surrounding temperature, second stage compression temperature reaches 206 $°C$. The high pressure storage works with ambient temperature. The required pre heating temperature for diagram state 10 is the theoretically required temperature, to reach surrounding temperature at the exit

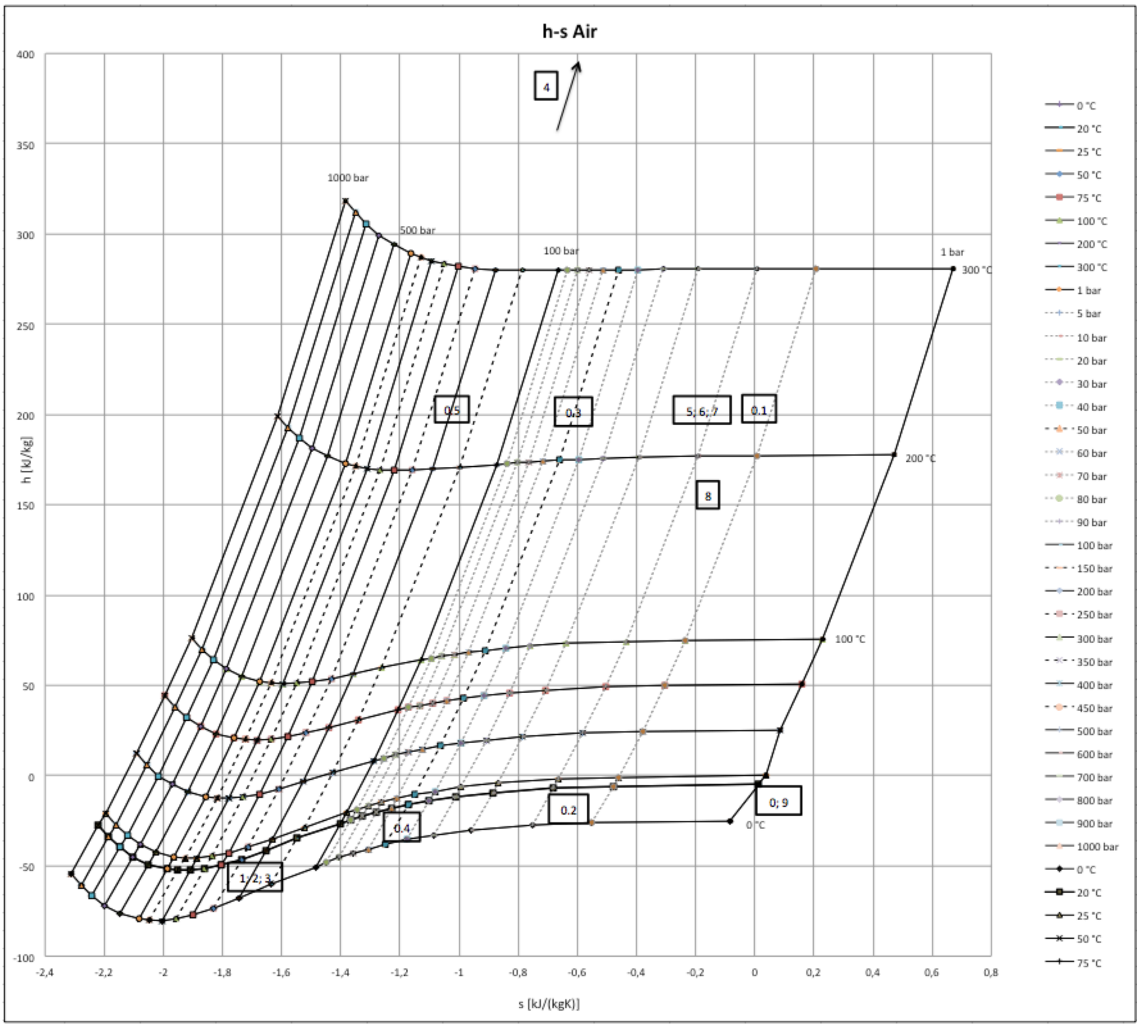

Figure 5.2.: The h-s diagram of air for the chosen system, data from VDI [15]

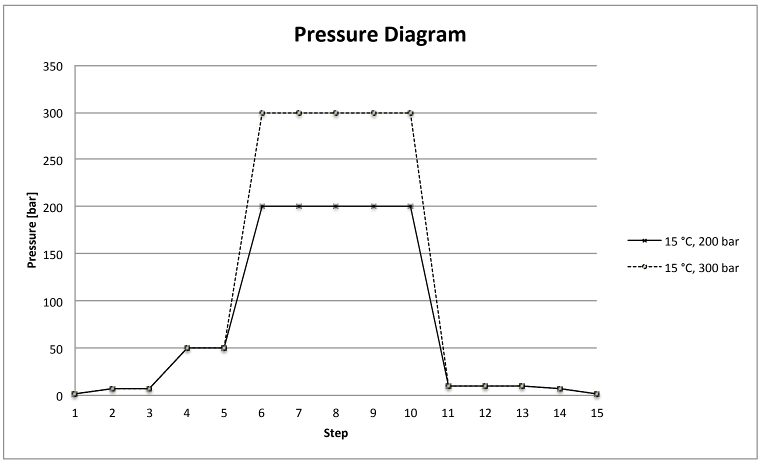

Figure 5.3.: Pressure Diagram

of the turbine. Because it is only calculated with one heat exchanger at the very beginning of the expansion process, the temperature must be very high. 893 $°C$ for the 200 bar system and 1036 $°C$ for the 300 bar system. These temperatures are not reachable with the water storage system, nor with the compression temperatures. But as we know from formula 3.3, the amount of heat is constant, delta Temperature is smaller in reality, which leads to more mass. More mass at lower temperature has to exchange its heat with the air in more steps. Therefore it is also possible to heat the pressure reduction valves. The most important point for the efficiency of the system is that the air is as hot as possible in the turbine. Going on with point 5 after the high pressure reduction valve, temperature then drops to 222 $°C$ and 10 bar in the small pressure tank. This heat must stay constant with insulation of the storage. This storage should not lose too much heat because it is only refilled when air flows out, and because of the small volume the air doesn't stay very long in the tank, that it does not have the time to cool down after standing for too long. The low pressure reduction valve reduces the pressure to 7 bar turbine operation pressure and 174 $°C$. In the turbine the pressure is reduced to 1.5 bar (assumption) and surrounding temperature. Very important and used as calculation control is the continuum of mass, the mass-flow, which is constant from inlet to outlet. Because of different temperatures and pressures, the volume flow varies as shown in 5.5. Interesting is the increase of Volume-flow with decreasing temperature. To reach 200 bar at 15 $°C$ ambient temperature, 3.45 $\frac{m^3}{min}$ are required. At 10 $°C$, 4.5 $\frac{m^3}{min}$ are required, and at 0 $°C$ 7.7 $\frac{m^3}{min}$. To reach 300 bar at 0 $°C$, 18.26 $\frac{m^3}{min}$ are required. Figure 5.6 shows the enthalpy diagram for the adiabatic CAES cycle. Enthalpy exceeds 200 $\frac{kJ}{kg}$ in the first and second compression stage. The third stage 119 $\frac{kJ}{kg}$ for 200 bar and 178 $\frac{kJ}{kg}$ for 300 bar. After heating the enthalpy level reaches 955 $\frac{kJ}{kg}$ for 200 bar and 1125 $\frac{kJ}{kg}$ for 300 bar. The high pressure reduction valve leads to an enthalpy level of 200 $\frac{kJ}{kg}$ at 10 bar. The inlet enthalpy of the turbine is 151 $\frac{kJ}{kg}$ and -10 $\frac{kJ}{kg}$ at the outlet. The density increases

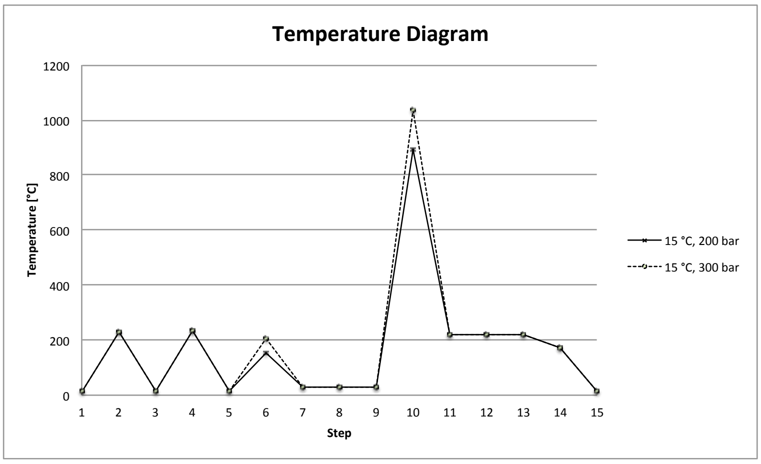

Figure 5.4.: Temperature Diagram

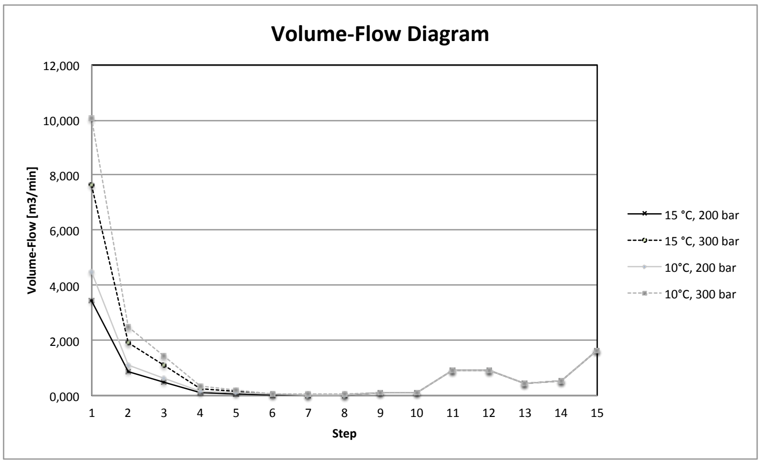

Figure 5.5.: Volume Flow Diagram

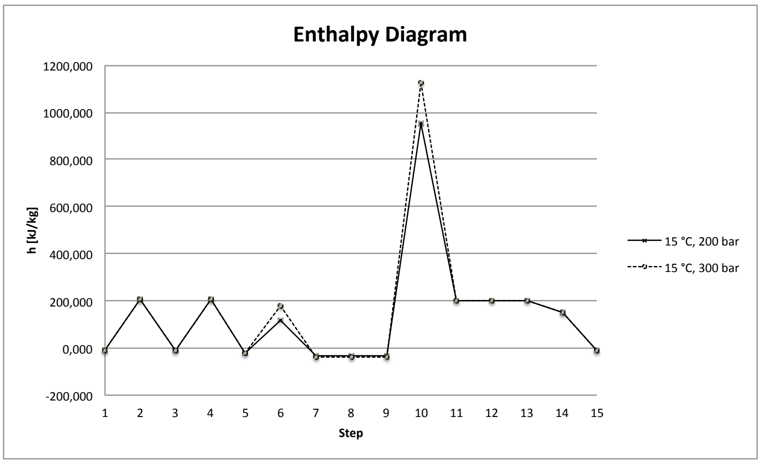

Figure 5.6.: Enthalpy Diagram

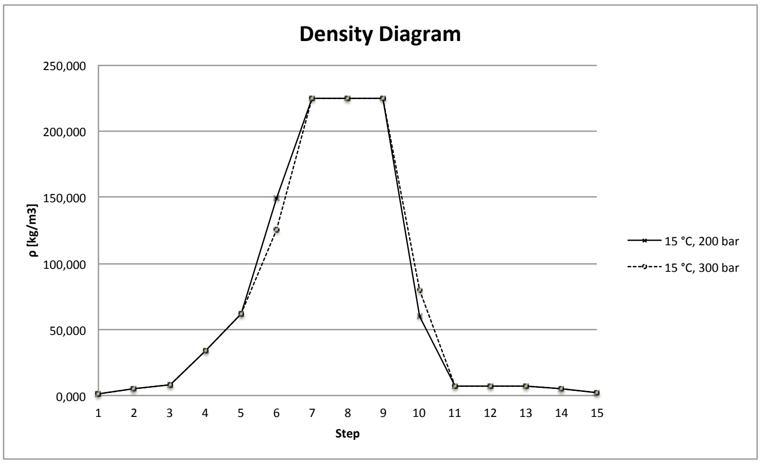

Figure 5.7.: Density Diagram

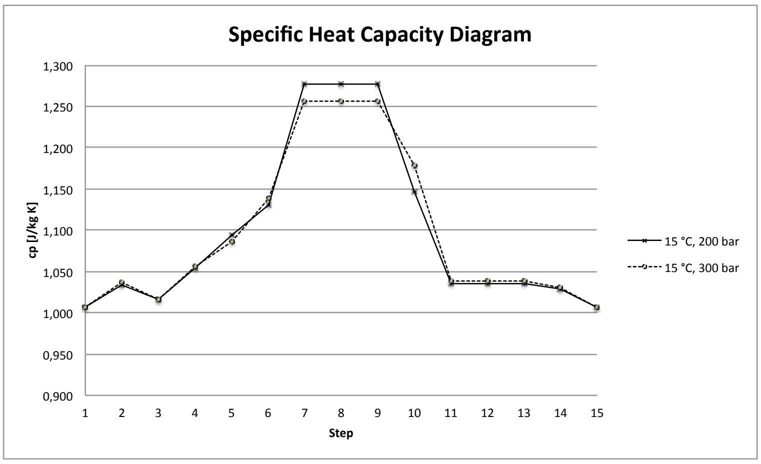

Figure 5.8.: Heat Capacity Diagram

with pressure as shown in Figure 5.7. The maximum is 225 $\frac{kg}{m^3}$ at 200 bar. The heat capacity increases with pressure and temperature as shown in Figure 5.8.

5.4. Compression

The electricity needs are, as shown in figure 5.1, 3.137 $\frac{kWh}{d}$ and the CAES system should cover the needs with 100 %. In this boundary conditions, if the compressor runs the whole day with 8 h sunshine it produces 5.192 $\left[\frac{kWh}{d}\right]$. This means that the users needs are covered with 165 %. One could think that it would be possible to use a smaller compressor, which is possible, but the situation changes with the surrounding temperature. The warmer it gets, the smaller is the pressurised air production is. Between 20 and 25 $°C$ ambient temperature is the equalisation point for this installation. With higher temperatures, one single compressor can't enable the electricity production anymore. If the energy of one compressor is not sufficient, a second or a bigger one has to be installed. So it is important to also keep in mind the possible changing situations of an installation that also in the worst case will be enough air to feed the turbine. Figure 5.9 shows the power the compressor needs related to ambient temperature on the abscissa, and because calculated ideal, the usable energy.

The continuous curve shows the characteristic for our model user with 200 bar compressor pressure. The broken curve represents the characteristic for 300 bar compressor pressure. It shows that operate the compressor on 300 bar has a great effect on the workload in the lower temperature levels. At 0 $°C$ it brings more than the double performance or 18.4 $\frac{kWh}{d}$ more absolute. The curves also shows the logarithmical decrease with higher ambient temperatures. One compressor can fulfil higher energy needs with lower ambient temperatures which means a cheaper installation. The pressure ratio in stage one is 7.0, stage two 7.1 and stage three 200 bar 4.0, stage three 300 bar 6.0.

Figure 5.11 shows the final exit conditions of the Bauer scuba compressor PE 100-T

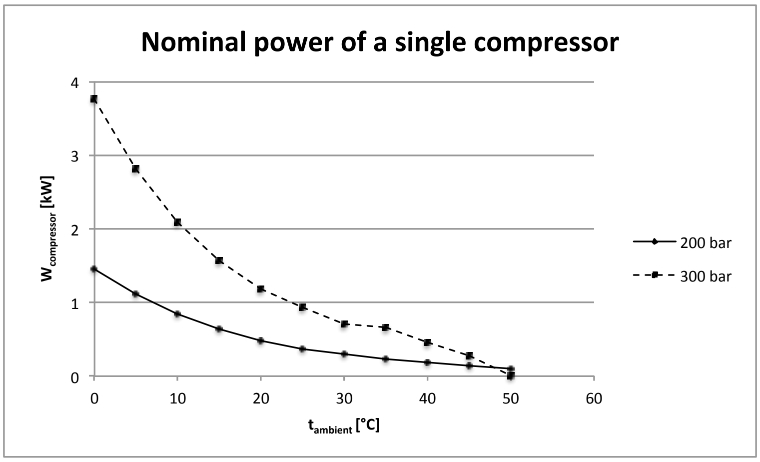

Figure 5.9.: Nominal power per day of PE 100-T compressor related to ambient temperature

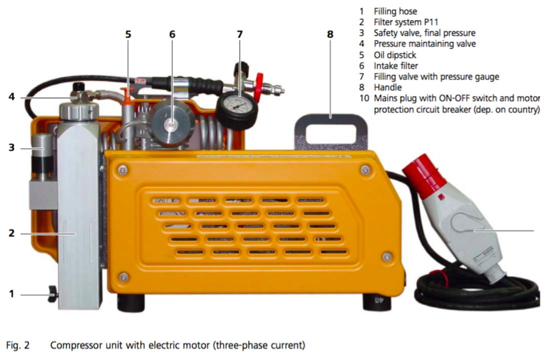

Figure 5.10.: Intercoolers of Baur compressor
[13, p.2]

with 2,2 kW nominal power, which is the basis for the calculation.

Figure 5.12 shows the T-s diagram of three stage compression.

5.5. Pressure Storage

The pressurised air can be stored in standard bottles. The example system stores 8.6 m^3 air at 200 bar and outlet temperature of the compressor. The system is equipped with steel bottles made for 300 bar with 50 l volume each, 1.5 m height and 23 cm diameter. The weight is 75 kg. 173 such bottles are required to store the compressed air. 10 x 17 bottles need an area of 2.3 x 3.9 m.

1. Filter cartridge 057679: lifetime [hours]		
Filling pressure p = 200 bar		PE 100-T
Ambient temperature tU [°C]	Final separator temperature tAb [°C]	Delivery Q [l/min] 100
10	20 - 24	26 - 21
15	25 - 29	20 - 16
20	30 - 34	15 - 12
25	35 - 39	11 - 9
30	40 - 44	9 - 7
35	45 - 49	7 - 6
40	50 - 54	5 - 5
Filling pressure p = 300 bar		PE 100-T
Ambient temperature tU [°C]	Final separator temperature tAb [°C]	Delivery Q [l/min] 100
10	20 - 24	39 - 31
15	25 - 29	29 - 24
20	30 - 34	22 - 18
25	35 - 39	17 - 14
30	40 - 44	13 - 11
35	45 - 49	10 - 9
40	50 - 54	8 - 7

Figure 5.11.: Bauer PE-100T outlet conditions
[13, p.20]

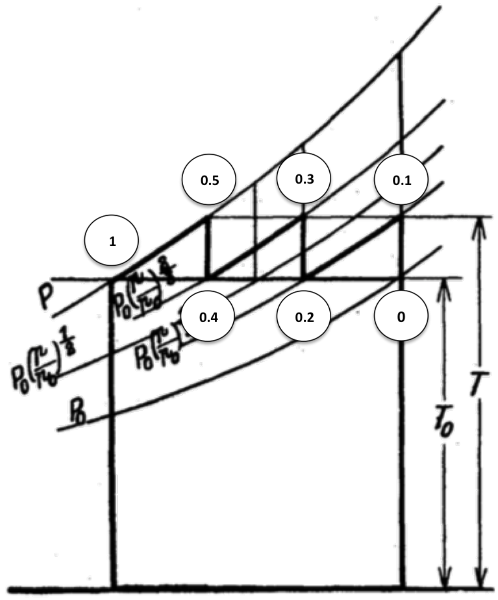

Figure 5.12.: Three stage compression in the T-s diagram
[14, p.34]

5.6. Heat Storage

Water is chosen as cheap, non-toxic and universally available heat storage medium. Storage mediums are discussed in the previous chapter. As said before, the cooling coils of the Bauer compressor see figure 5.10 have to be changed into a heat exchanger. The water is stored in one tank which feeds all three inter cooling heat exchangers with the same temperature, see figure 5.13. Because the compression stages have small divergent temperature outlets (stage 0.1, 0.3 and 0.5) the mass-flow in the three heat exchangers has to be different to ensure the same cooling effect, back to surrounding temperature. The water will heat up by system working time. Because $\dot{Q}$ must be constant to ensure the same cooling effect, the mass-flow of the water pump must increase it's flow rate. The formulas in chapter 3.8 show how the temperature mix and mass-flow is calculated after the three heat exchangers (stage T_{3b}). The volume flow in the three heat exchangers could also be equal, if the compression stage with the highest cooling needs would dictate the system. Then the energy consumption of the circulating pump would be higher, but the cooling effect in the other stages would also be better. The best system efficiency point has to be determined with different tests. The system capacity is physically reached when the water in the tank reaches the temperature of the compressor. The CAES system is limited by safety reason at less than 100 $°C$. A leakage in a pipe or system part with water inside over 100 $°C$ can be really dangerous because of the danger of explosion danger. The more water is stored in a tank, the slower the system will heat up. It is a question of system design how much heat must be stored to satisfy the user's needs. With increasing storage temperature, ΔT decreases, and so the mass flow of the pump must increase to transport the same amount of heat to the outside. Figure **??** shows the system behaviour with 58 l storage water. Warmer storage temperature means increasing compressor inlet temperature, which also leads to decreasing system efficiency. Otherwise, warmer storage temperature means warmer air before the expansion which leads to a better system efficiency. When the system pressure is reached, the warm water heat storage should have 100 $°C$. Figure **??** shows that the 58 l tank matches perfect in the system needs. After eight hours, the heat storage has 100 $°C$ and the system switches of because it is totally loaded. Also the pump to circulate the water is still working in normal conditions. Starting with 0,124 $\left[\frac{kg}{min}\right]$ mass-flow, it has a maximum mass-flow of 0.245 $\left[\frac{kg}{min}\right]$. More volume flow means more electricity need for the pump and so less system efficiency for the whole system.

Figure 5.15 shows how a system with under dimensioned heat storage is collapsing. The mass-flow of the pump increases exponentially. The pump tries to carrie away the amount of heat with higher revolution speed, but as the cooling water strives for the compression temperature, Δ T is getting smaller and smaller the the mass-flow has to strive for infinity. It would be also possible to have a separate cold and warm water tank and not a circulating system. The disadvantage here is double of the installation volume and tanks, and the abrupt inability of the system to work if one tank is totally full or empty.

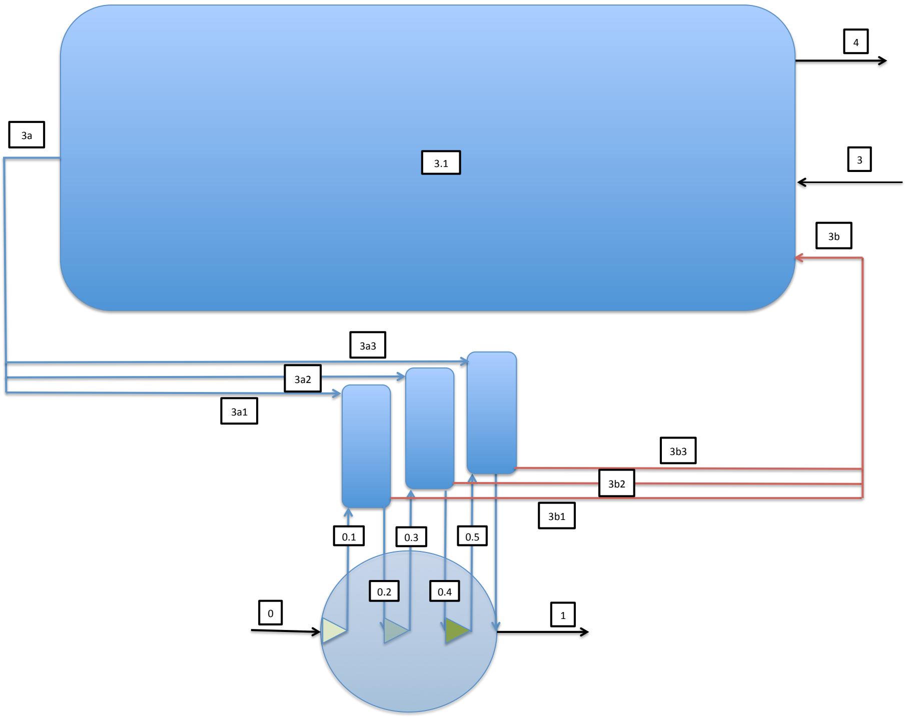

Figure 5.13.: Scheme of a circulating water storage tank in combination with the CAES system, sections named in table 5.3

5.7. Pressure Reduction

Minor losses in piping systems are a result of different influences. As formula 3.16 shows, minor losses are a result of pipe geometry, fluid density and velocity squared. To minimise these losses, a small low pressure storage (no. 6 in Figure 5.1) of 10 bars is included into the system. Minor losses only appear when a fluid is flowing continuously or has a velocity. If the flow from the main pressure storage is divided into two sections before it reaches the turbine, by a secondary pressure storage, the first sections is not counted in the minor loss equation. To reach quasi static pressure in point 7, low pressure storage outlet, more mass flow has to flow into the tank from the pressure side 5, than it flows out to the turbine side 7. To calculate the dimensions of the system, the maximum flow at the turbine has to be known. Thus it is decided to have a maximum electricity generation of k

5.8. Expansion

Calculated as average daily electricity needs per person in domestic homes are 3.137 $\left[\frac{kWh}{d}\right]$. 3.137 $\left[\frac{kWh}{d}\right]$ is the average electricity need, but this won't be used constantly within 24 hours. A maximum outlet power of 2 kW is assumed and therefore the expan-

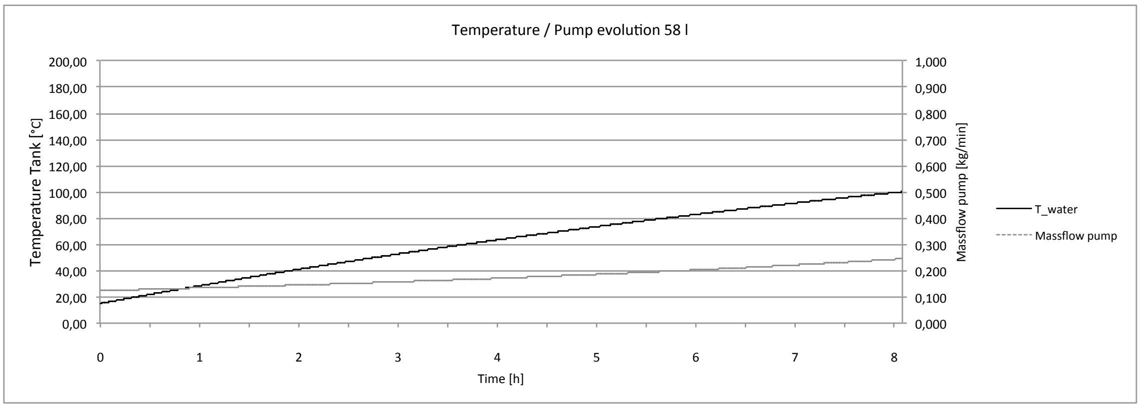

Figure 5.14.: Temperature evolution of a circulating water storage tank with 58 l capacity

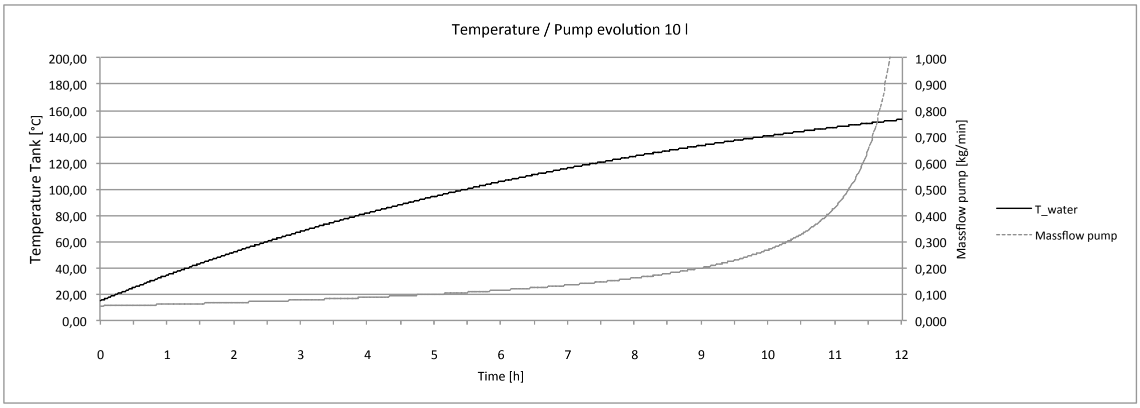

Figure 5.15.: Qualitative example of temperature evolution of a circulating water storage tank with 10 l capacity until collapse

System state	Scheme number
0	inlet compressor air
0.1	1st compression stage
0.2	1st inter-cooling
0.3	2nd compression stage
0.4	2nd inter-cooling
0.5	3rd compression stage
1	3rd inter-cooling and outlet compressor
3	hp storage exit
3.1	water tank
3a	water tank outlet
3a1	inlet HX compression stage 1
3a2	inlet HX compression stage 2
3a3	inlet HX compression stage 3
3b1	outlet HX compression stage 1
3b2	outlet HX compression stage 2
3b3	outlet HX compression stage 3
3b	water tank inlet
4	warm water HX outlet

Table 5.3.: Explanation of state numbers "water system" in the System Diagrams

sion side of the system is calculated.

5.9. PV Cells

The required PV cell area can be calculated wit formula 3.1. In this case, 5.4 m^2 are sufficient when the sun is shining the average time with average intensity. It has to be proved how the system behaves when there are clouds or shadows on the cells. A bigger cell surface as safety factor seems to make sense, this could be 50 % estimated, but has to be calculated in detail. To reach the nominal power of the compressor, 2.2 kW, 18.4 m^2 PV cells are required.

5.10. Summary

The Table 5.4 gives the overview of the CAES system designed for the example user. The full calculation is in the appendix B for compression to 200 and 300 *bar* with surrounding temperatures from 0 to 50 $°C$. Around 95% of the compression energy is converted into heat [10]. If this heat would not be used, a big part of the system energy would be lost to the surrounding which leads to less efficiency. The calculated temperatures to pre heat the air in one step before the expansion are really high, and not to reach with the sensible water heat storage. The following chapter simulates this expansion problem numerically.

Multiple heat exchangers could be a solution to bring the same amount oh heat into the system, but this has to be proofed in reality.

State	T $[°C]$	V $\left[\frac{m^3}{min}\right]$	p $[bar]$	ρ $\left[\frac{kg}{m^3}\right]$	c_p $\left[\frac{kJ}{kgK}\right]$	$P_a/\frac{\delta Q}{\delta t}$ $[kW]$
0	15	3.456	1	1.21	1.0063	
0.1	229	0.861	7	4.8774	1.0340	0.216
0.2	15	0.494	7	8.5043	1.0168	-0.254
0.3	232	0.121	50	34.13	1.0542	0.216
0.4	15	0.069	50	61.48	1.0948	-0.281
0.5	155	0.026	200	149.04	1.1304	0.216
1	27	0.018	200	225.03	1.2771	-0.184
2	27	0.018	200	225.03	1.2771	
3	27	0.106	200	225.03	1.2771	
4	893	0.106	200	59.9	1.1465	1.972
5	222	0.900	10	7.1	1.0357	
6	222	0.900	10	7.1	1.0357	
7	222	0.425	10	7.1	1.0357	0.133
8	174	0.548	7	5.5	0.0284	0.517
9	15	1.648	1.5	1.82	1.0071	2.0

Table 5.4.: CAES System Calculation at 15 ° C Ambient Temperature

The theoretical isothermal compressor power $P_{is,15°C}$ is 0.5 kW. The adiabatic compressor of the example is 0.65 kW. After formula 3.30 η_{is} is 77 %. Compressing to 300 bar, η_{is} is 75 %. In theory the isothermal compressor is 25 % more efficient than the adiabatic of our system. It has to be proved in reality, which efficiency an isothermal compressor can reach, and if the expected more costs justify such a system. The amount of pressurised air bottles is relative high which leads to high initial investments. Also enough space has to be given for the installation. The amount of water to store the heat is really small compared with the air volume. In the example, a maximum storage capacity of 5.2 $\frac{kWh}{d}$ is possible from the compressor in eight hours of work which leads to 173 bottles. The system can be reduced regarding installation volume to around 130 pressurised air bottles, if also in lower ambient temperatures, where more nominal power is possible to produce, the goal of the system is not to store as much as possible but only as much as required, which is 3.137 $\frac{kWh}{d}$. In the characteristic curves of the circulating water heat storage the volume flow line shows that below 100 $°C$ a system collapse because of overcharging the water pump is not an option because the system is not allowed to exceed 100 $°C$. Also in this case if the pump is for example designed for double the maximum volume flow, safe operation until 130 $°C$ is secured.

Figure 5.16 shows the related system dimensions for different ambient temperatures. The nominal power is increasing with colder ambient temperature. To feed increasing nominal power, the cell area has to be bigger, as well as the number of pressurised air storage bottles and heat storage water volume. One compressor produces enough electricity for two persons until 10 $^\circ C$ ambient temperature. Over 35 $^\circ C$, also one person needs multiple compressors to produce enough pressurised air. With 50 $^\circ C$ ambient temperature and storage pressure of 300 bar, one needs 46 compressors which seems definitely uneconomic. The amount of cooling water is little compared with the amount of required air bottles. Because the compressor looses efficiency with incomplete intercooling (around 1 % each stage each 3 $^\circ C$), and the amount of water volume is very little, the usage of a two storage system should be considered. As $2.3 \cdot 3.9$ m^2 the required area for the pressurised air vessels should be the biggest part regarding installation area. It is assumed that the whole system can be installed in $3 \cdot 4 \cdot 2$ m or 24 m^3.

CAES System dimensioning

0	average	no of	daily	CAES System	daily	one comp. for	needed	nominal	daily	daily nominal	sun	sun	PV cell	Needed	needed	needed	no. of air
T	electr. needs	persons	electr. needs	cover of needs	system needs	daily needs	compressors	power comp	sunshours	power comp	irradiation	irradiation	efficiency	$A_{PV\ cell}$	bottle vol.	bottle vol.	bottles
[°C]	[kWh/pp d]	[p]	[kWh/d]	[%]	[kWh/d]	[%]	[n]	[kW]	[h]	[kWh/d]	[kWh/m² a]	[kWh/m² d]	[%]	[m²]	[m³]	[l]	[pcs]
0	3,137	1	3,137	1	3,137	5,607	1	1,451	8	11,605	1750	4,795	0,200	12,1	19,4	19354,3	388
5	3,137	1	3,137	1	3,137	4,316	1	1,116	8	8,929	1750	4,795	0,200	9,3	14,9	14880,0	298
10	3,137	1	3,137	1	3,137	3,273	1	0,847	8	6,774	1750	4,795	0,200	7,1	11,3	11280,0	226
15	3,137	1	3,137	1	3,137	2,509	1	0,649	8	5,192	1750	4,795	0,200	5,4	8,6	8640,0	173
20	3,137	1	3,137	1	3,137	1,883	1	0,487	8	3,897	1750	4,795	0,200	4,1	6,5	6480,0	130
25	3,137	1	3,137	1	3,137	1,395	2	0,722	8	5,777	1750	4,795	0,200	6,0	4,8	4800,0	96
30	3,137	1	3,137	1	3,137	1,117	2	0,578	8	4,624	1750	4,795	0,200	4,8	3,8	3840,0	77
35	3,137	1	3,137	1	3,137	0,908	2	0,470	8	3,760	1750	4,795	0,200	3,9	3,1	3120,0	63
40	3,137	1	3,137	1	3,137	0,699	3	0,543	8	4,341	1750	4,795	0,200	4,5	2,4	2400,0	48
45	3,137	1	3,137	1	3,137	0,560	3	0,434	8	3,475	1750	4,795	0,200	3,6	1,9	1920,0	39
50	3,137	1	3,137	1	3,137	0,395	4	0,409	8	3,270	1750	4,795	0,200	3,4	1,4	1354,3	28
0	3,137	1	3,137	1	3,137	13,253	1	3,769	8	30,156	1750	4,795	0,200	31,4	30,5	30497,1	610
5	3,137	1	3,137	1	3,137	9,885	1	2,812	8	22,494	1750	4,795	0,200	23,5	22,7	22731,4	455
10	3,137	1	3,137	1	3,137	7,311	1	2,080	8	16,637	1750	4,795	0,200	17,3	16,8	16800,0	336
15	3,137	1	3,137	1	3,137	5,540	1	1,576	8	12,605	1750	4,795	0,200	13,1	12,7	12720,0	255
20	3,137	1	3,137	1	3,137	4,184	1	1,190	8	9,520	1750	4,795	0,200	9,9	9,6	9600,0	192
25	3,137	1	3,137	1	3,137	3,245	1	0,923	8	7,383	1750	4,795	0,200	7,7	7,4	7440,0	149
30	3,137	1	3,137	1	3,137	2,513	1	0,715	8	5,719	1750	4,795	0,200	6,0	5,8	5760,0	116
35	3,137	1	3,137	1	3,137	2,305	1	0,656	8	5,246	1750	4,795	0,200	5,5	5,3	5280,0	106
40	3,137	1	3,137	1	3,137	1,573	1	0,447	8	3,579	1750	4,795	0,200	3,7	3,6	3600,0	72
45	3,137	1	3,137	1	3,137	0,974	2	0,554	8	4,434	1750	4,795	0,200	4,6	2,2	2228,6	45
50	3,137	1	3,137	1	3,137	0,030	46	0,392	8	3,139	1750	4,795	0,200	3,3	0,1	68,6	2

Figure 5.16.: Equipment size for different ambient temperatures

6. Numerical Study of the CAES Pressure Control Valve

The high gradient pressure reduction from 200 or 300 *bar* storage pressure to 10 *bar* low pressure pressure of the small pressure tank is interesting to analyse because the valve has an important effect on the system‘s functionality and also efficiency. As seen in chapter 5 the theoretically required temperature for pre heating of the air of around 800 $°C$ is not reachable. In this case positive is the fact that the real system is not absolute adiabatic. Initialising the pressure and thus temperature drop from 50 $°C$, temperatures in the gap of the valve will drop into the minus region. The temperature difference with the surroundings then will lead to temperature flow into the fluid with a heating effect. According Mr. Scheffner from Samson who is introduced in the next passage, this pressure reduction can definitely be handled within two steps, which means two PRVs and less pressure drop. The functionality with just one PRV has to be tested in reality. A smaller pressure drop means smaller temperature differences and the possibility, to heat again between the two steps. This simulation is made with constant valve body temperature of 50 $°C$. Fluent flow simulation programs calculate given problems by iterative proceeding from knot to knot of the grid, modelled into the flow area or volume. The essential kinematic and state quantities or variables of fluid mechanics which describe a flow field are

- pressure p
- flow velocity in all directions v_x, v_y and v_z
- temperature T

The essential fluid properties are

- density ρ
- kinematic viscosity μ
- dynamic viscosity ν

Fundamentals of computational fluid dynamics are

- mass conservation equation, for a newtonian incompressible fluid with constant density and viscosity

$$\frac{\partial \rho}{\partial t} + \nabla_{x,y,z}(\rho \cdot v_{x,y,z}) = \frac{\partial \rho}{\partial t} + \frac{\partial(\rho \cdot v_x)}{\partial x} + \frac{\partial(\rho \cdot v_y)}{\partial y} + \frac{\partial(\rho \cdot v_z)}{\partial z} = 0 \tag{6.1}$$

- momentum conservation on base of the Navier-Stokes equation

$$\rho g_x - \frac{\partial p}{\partial x} + \mu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right) = \rho \frac{du}{dt} \tag{6.2}$$

$$\rho g_y - \frac{\partial p}{\partial y} + \mu \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) = \rho \frac{dv}{dt} \tag{6.3}$$

$$\rho g_z - \frac{\partial p}{\partial z} + \mu \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right) = \rho \frac{dw}{dt} \tag{6.4}$$

- energy conservation

$$\frac{\partial h}{\partial t} + \nabla_{x,y,z}(h \cdot v_{x,y,z}) = \lambda \Delta T = \frac{\partial h}{\partial t} + \frac{\partial (h \cdot v_x)}{\partial x} + \frac{\partial (h \cdot v_y)}{\partial y} + \frac{\partial (h \cdot v_z)}{\partial z} = \lambda \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) \tag{6.5}$$

- turbulence conservation with Reynolds averaging of the Navier-Stokes equation, Z represents the Reynolds stress which can be interpreted as an additional apparant viscosity due to turbulence

$$\frac{\partial v_{meanx,y,z}}{\partial t} + (v_{meanx,y,z} \cdot \nabla_{x,y,z}) v_{meanx,y,z} = \rho g_{x,y,z} - \frac{1}{\rho} \nabla_{x,y,z} p_{mean} + \mu \Delta v_{meanx,y,z} + Z_{x,y,z} \tag{6.6}$$

A numerical simulation is made based on the calculated theoretical system values. Dipl. Ing. Manfred Scheffner from Samson AG, a company based in Germany specialised in metal valves, chose a pressure reduction valve for the previously calculated System. Asked was to select a valve for air (gaseous) with initial pressure of 200 bars reduced to 10 bars. According to the theoretical calculation with 2 kW output on the turbine, the mass flow must be 0,141 $\frac{kg}{s}$ or 8,5 $\frac{kg}{min}$. The dimensioning of Samson is shown in the appendix see figure **??**. The result is a two step pressure reduction each with $\delta_p = 95$ bar. The valve can be heated with a secondary connection by the warm water, which flows around the valve body. The dimension is DN 25 with 15 mm hub. It may be possible to use only one PRV because it can handle these pressure differences but the real temperatures have to be proved in practice. The valve function (shaft movement up and down to reduce the flow) can be affected negatively by icing when the air is cooling down too much during expansion. The valve is version 250 Type 3251. It is available from PN16 to 400 and from -196 to +550 ° C. This valve is regulated by a pneumatic actuator. Figure 6.1 shows the cross section of the valve. There exists also medium pressure actuated PRVs but Samson prefers the actuated version in these extreme pressure drops. Figure 6.2 shows another PRV from Samson where the valve body as well as the shaft connection to the actuator is heated by a secondary cycle. The shaft heating can be necessary in extreme conditions when gases expand with big pressure drops with the danger of icing inside the valve which can lead to shaft blockage. This simulation is done on the second PRV with maximum reduction from 105 to 10 bar. To get a "feeling" for the system, it is better to perform a simulation not with the extreme numbers, where convergence is harder to achieve, especially with compressible flow. Also in reality, the plug is moving to control the outlet pressure, which can't be simulated in this relatively simple simulation. Because of this, a pressure drop in the middle range (70 to 10 bar) and therefore pre heating in the middle dimension (50 $°C$) is done. If the system is only half full, also the heat storage has only around half of the temperature of the maximum values.

6.1. Geometry

The dimensions of the Samson PRV Type 3251 are known from the data sheet from Samson. Inside measures must be estimated by proportion from the cross section because they are not published by the manufacturer. The maximal shaft hub is 15 mm according Mr. Schaeffner. To simplify the simulation, the PRV is constructed in 2d. The simulation in 3d would take more time for the computer to calculate each iteration and reach convergence. The gap between the plug and the housing is the most interesting spot where the gradients change the most. Because the plug is round, the model is built as axis symmetric so as not to lose the expanding effects into radial direction. The cross section can not be used directly to setup the construction. Therefore, the vertical shaft and plug of figure 6.1 have to be turned clockwise into horizontal position. The mirrored S-shape, the air has to pass through the valve, is lost in the 2D axisymmetric model. Also the exact wall contact surface areas which heat the air with assumed 50 ° C constant wall temperature are not included exactly because some area is lost during the transformation. Figure 6.3 shows the dimensions used for the simulation and figure 6.4 shows the 2d model with the named sections. The valve is 1 % or 0,15 mm opened. 1 % opening is the estimated value to reach a relative high pressure step to show different effects like velocity increase and temperature drop in the simulation. Different input conditions showed that with the inlet pressure of 70 bar results an outlet pressure of 10 bar with suitable volume flow. This figure also shows that the outlet of the valve is elongated to get a more homogeneous outlet without backflow effects, like they appeared without elongation.

6.2. Meshing and Simulation Setup

The valve geometry is divided into three parts to refine the mesh with high gradients. Behind the gap is the finest mesh with element size of 0,006 mm. The highest gradients appear directly after the gap between the wall and the plug. Before the gap, velocities can't exceed mach numbers 1 because of the stagnation pressure. After this gap area, extreme turbulences are expected and therefore the element size is still small with 0,05 mm. The inlet and outlet has elements between 0,3 and 1,2 mm. To calculate thermal wall effects accurate, the walls are dissolved with 15 to 17 layers. The gap is dissolved only by 7 layers because the gap is very small with 0,15 mm and there is not enough space for many knots. Expansion effects in the middle of the flow are more important in the gap than wall effects. Figure 6.5 shows the whole net, Figure 6.6 shows the gap situation.

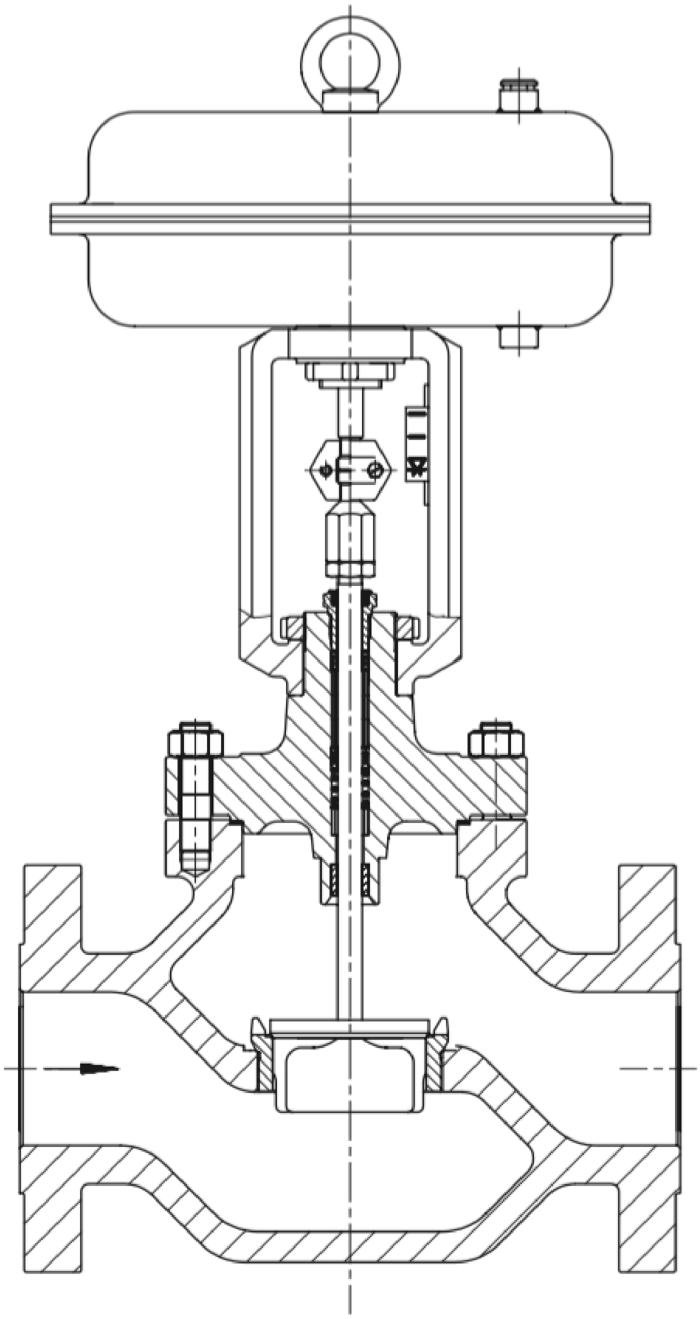

Figure 6.1.: Cross section of Samson's PRV type 3251 with pneumatic actuator typ 3271 [6, p.2]

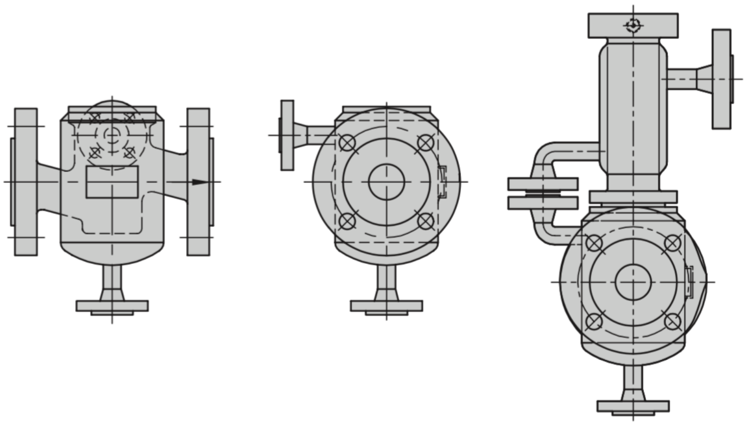

Figure 6.2.: Samson's PRV type 3241 with heating jacket and boot heating [5, p.16]

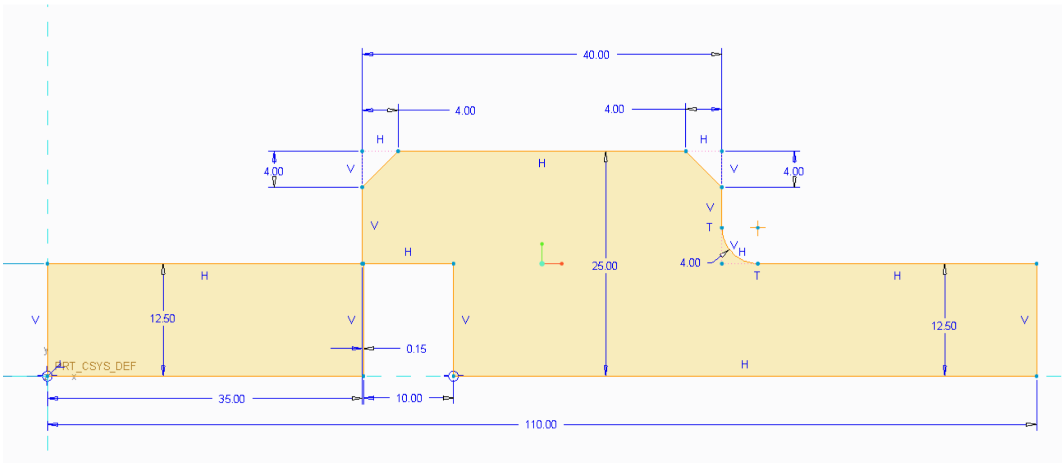

Figure 6.3.: Dimension of Samson Typ 3251 in 2d transferred to axissymmetric

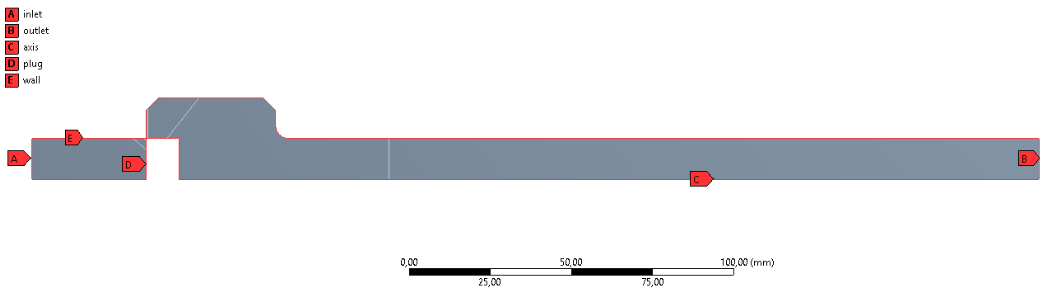

Figure 6.4.: Named sections for the 2d simulation

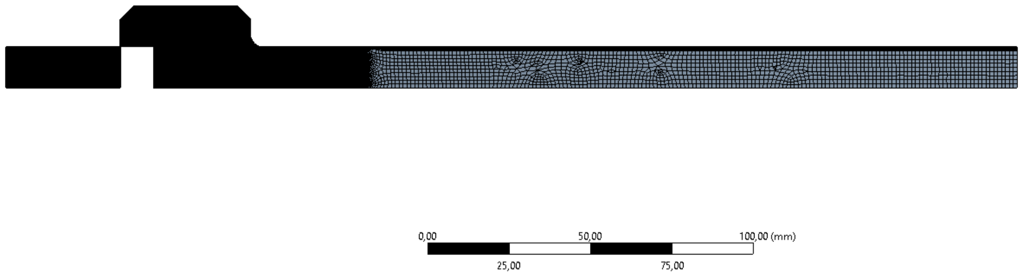

Figure 6.5.: Meshed PRV

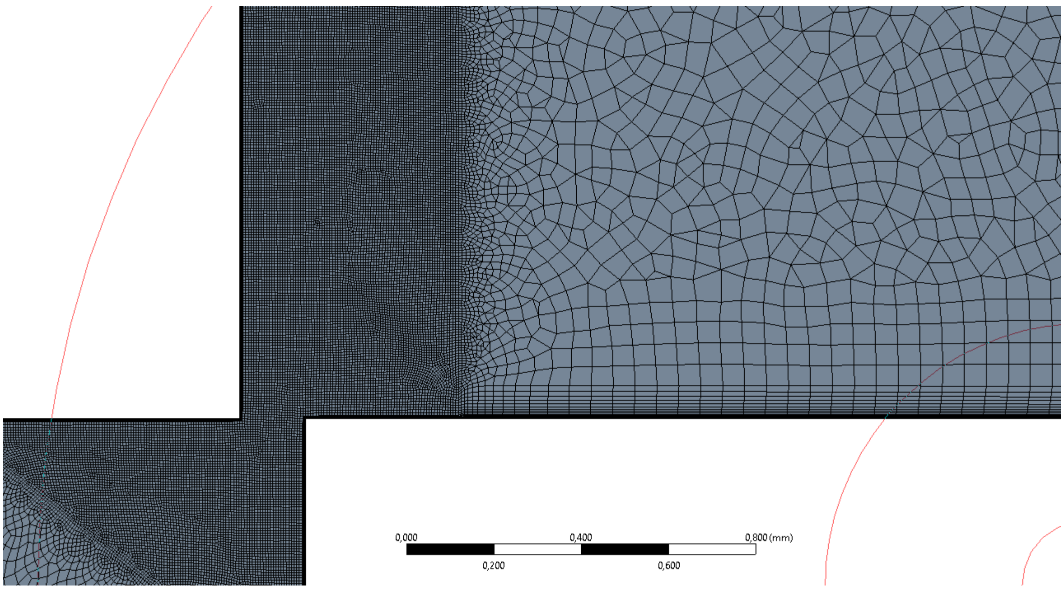

Figure 6.6.: Mesh detail of the gap, element size 0,006 mm

6.3. Numerical Settings in Ansys Fluent

The simulation in calculated as steady state conditions. Double precision is activated to minimise the decimal number errors. The pressure based solver is used. The density based solver is better with compressible overall high speed systems. The PRV has partial high speed areas.

Turbulence Model

The $k - \epsilon$ model with SST and enhanced wall treatment is used.

Materials

The PRV body walls and plug is out of steel. The walls have constant temperature of 50 °C . The fluid is air as ideal gas with constant viscosity.

Boundary Conditions

- Inlet
  Pressure inlet with 70 bar, initial pressure 60 bar. Temperature 50 °C
- Walls
  Constant temperature 50 °C
- Plug
  No heat flux
- Outlet
  Pressure outlet 10 bar

Solution Controls

The Courant number is set to 1,5 as advised in the Ansys Fluent handbook. The time step is finally set to 0,1 after trying different settings.

Run Calculation

The residuals had the default value of 10^{-3}. As they reached this convergence criteria, the calculation was continued because the results were not yet stationary.

6.4. Solution

6.4.1. Verification

Qualitative Evaluation

The Plots can show a possible solution by judging the flow direction. If the solution would have been calculated without a rotation axis, the stream direction would be more

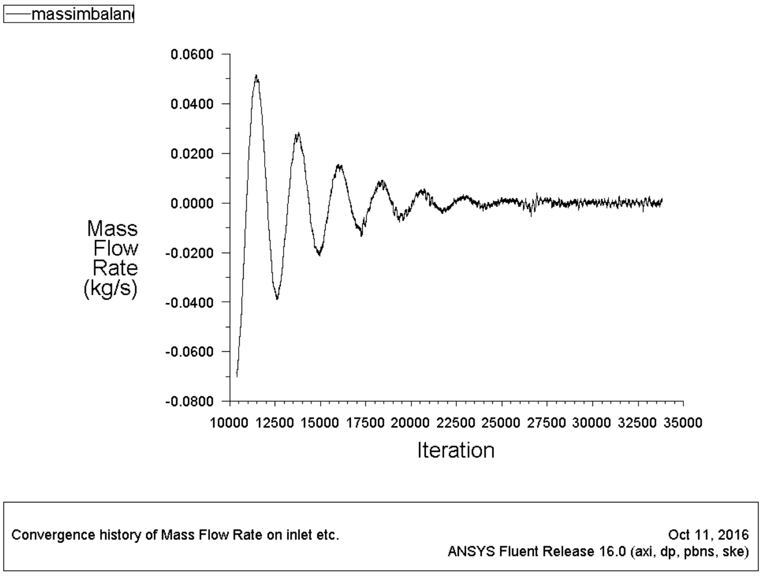

Figure 6.7.: Monitor of massimbalance

or less 45 °C. Also the calculated mass-flow of 0,167 $\frac{kg}{s}$ seems realistic. Mr. Schaeffner chose this valve for 0,141 $\frac{kg}{s}$ which is required for a 2 kW turbine.

Imbalances Strive for 0

The mass imbalance is shown in figure 6.7. The value is oscillating striving for 0 and reaches this value with a difference from inlet to outlet of 0,0015 $\left[\frac{kg}{s}\right]$ which is equivalent to 0,1 %. This value is acceptable.

Stationary Solution

The convergence history is stationary which means that the values are not changing with more iterations. Stationary convergence is reached with constant small oscillations. The Convergence history is shown in appendix C.2. The mass imbalance is shown in 6.7, the static temperature is shown in the appendix in Figure C.3 and the maximum velocity is shown in Figure C.4. The values are oscillating in a straight range which remains constant.

Residuals Decline 10^{-3}

The residuals all declined minimum 10^{-3} and then stayed constantly oscillating at one level as you can see in the appendix Figure C.1.

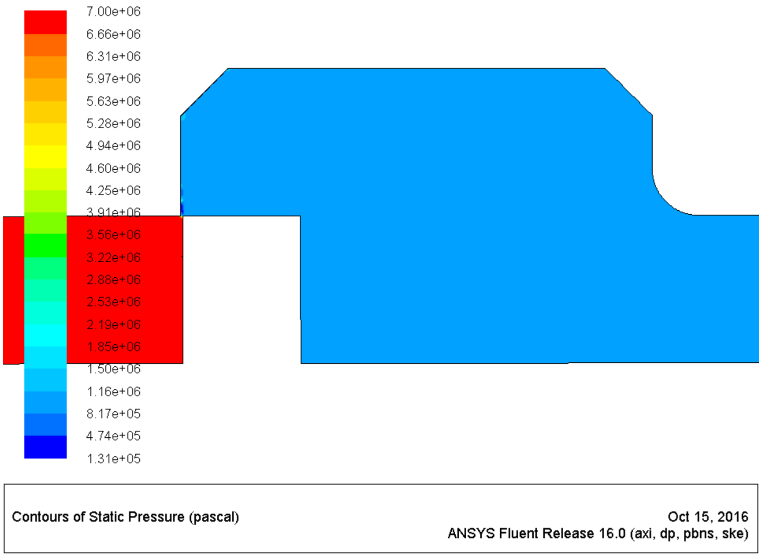

Figure 6.8.: Simulation of PRV DN25 - static pressure

Inflation Layers

The Y+ value is reduced to maximum 98,7 which means that the wall resolution is accurate.

6.4.2. Plots of the Solutions

Static pressure

The static pressure is reduced from 70 to 10 bar as you can see in figure 6.8.

Total Pressure

The total pressure drops from 75 to 15 bar.

Static Temperature

The static temperature drops from 65 $°C$ inlet temperature to minimum -142 $°C$ directly after the gap. Already after some millimetres, temperature reaches 0 $°C$. With the extreme turbulences which force the convective heat exchange and bring warmer air into the colder regions, the exit temperature is around 40 $°C$ which is only 10 $°C$ below inlet temperature. Figure 6.10 shows that the wall after the pressure drop has a big influence in temperature. The air is heated from the warmer wall at a constant 50 °C until it reaches wall temperature at the valve outlet in the wall close boundary layers. This is a very important finding because the theoretical calculation for the system design is done adiabatic. The heat load is known from the system without losses. The heat that goes

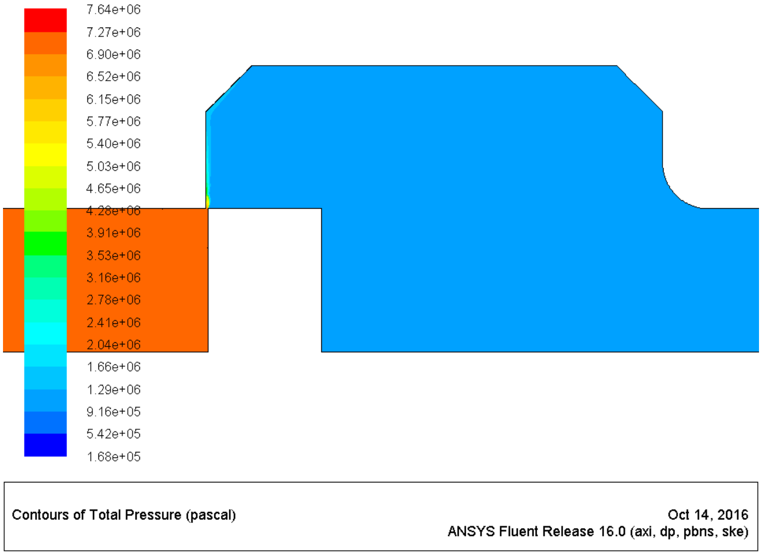

Figure 6.9.: Simulation of PRV DN25 - total pressure

out during compression is the same that comes in during expansion. But it is not known if the exchange area in the valve is enough to bring in enough heat because the absolute temperature of the warm water which heats the valve is not very high, compared with the very low expanding temperatures in the gap (around -142 °C). This simulation shows that the area is sufficient for this pressure drop and warm water temperature, because the real valve has even more contact surface. Figure 6.11 shows an enlargement of the temperatures in the gap. The heat transfer rate which is transported from the walls into the fluid is 33,4 W.

Velocity

The maximum velocity in the valve reaches 614 $[\frac{m}{s}]$ or a mach number of 2,67. Figure 6.12 shows the velocity vectors and figure 6.13 the velocity in the valve. Figure 6.14 shows the highest velocities appearing after the gap enlarged. The vectors show that in the section after the gap and before the constant outlet geometry are strong turbulences. One vortex is in the middle of the room, rotating clockwise with the same direction as the main flow close to the wall. A second vortex close to the plug turns anti clockwise and seems to block the main flow after the gap. Because this vortex has higher temperature than the expanding air after the gap, it could explain the strong increase of temperature after the first cold temperature field shown in figure 6.11 and the velocity decrease and increase after the gap in flow direction.

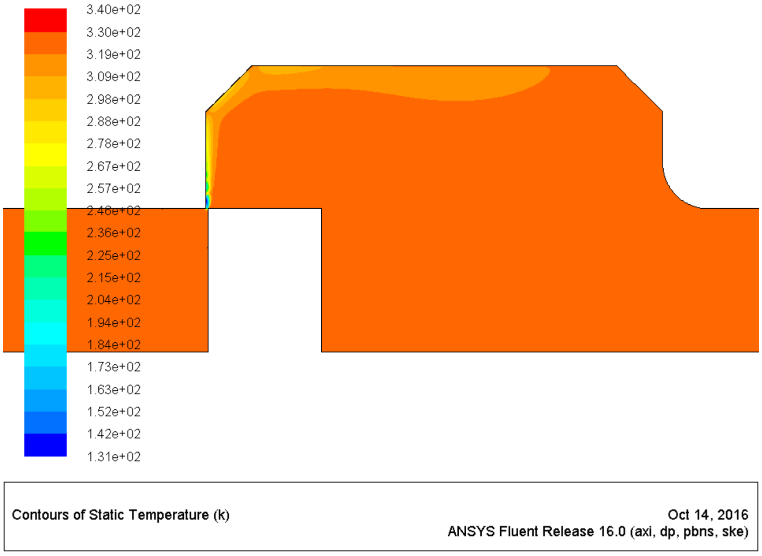

Figure 6.10.: Simulation of PRV DN25 - static temperature

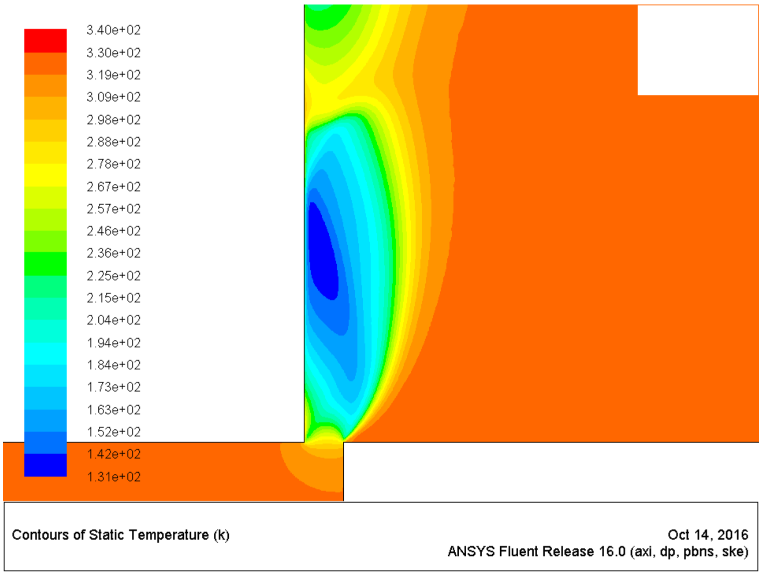

Figure 6.11.: Simulation of PRV DN25 - static temperature in the gap

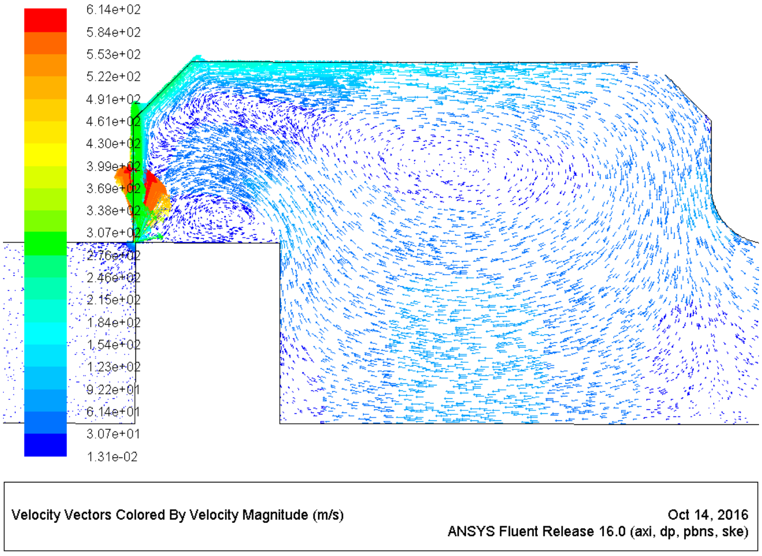

Figure 6.12.: Simulation of PRV DN25 - velocity magnitude vectors

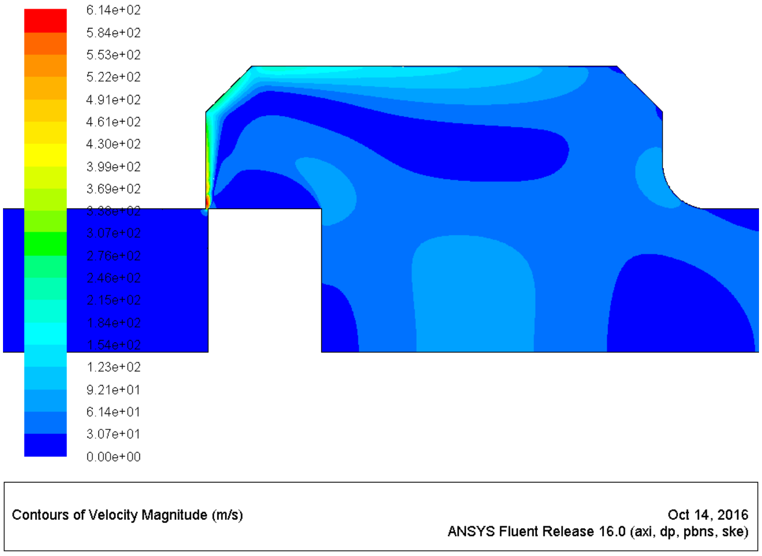

Figure 6.13.: Simulation of PRV DN25 - velocity magnitude

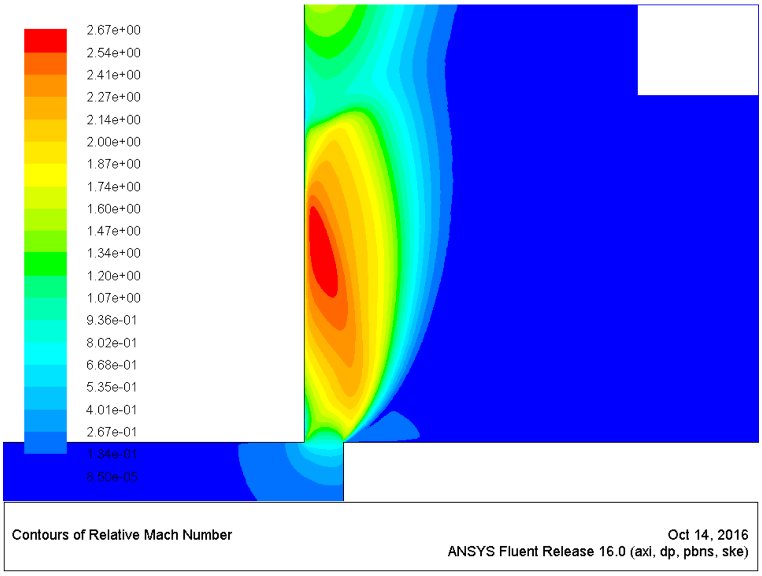

Figure 6.14.: Simulation of PRV DN25 - velocity magnitude gap

6.5. Critical Statement

The values of the calculated solution by simulation seem to be realistic. It is which was expected from the theoretical calculation. The convergence history could be better without the oscillating curves. It seems like changing one value has an influence on the other value in the other direction and vice versa. This could be a result of the solver system of Ansys Fluent. The kinematic and state equations are calculated after each other. This could explain the oscillating system behaviour. This could be proved by simulating the same setup in Ansys CFX because the kinematic and state equations are calculated in the same time. The real values of momentum, temperature, energy, mass and pressure seem to be somewhere between the oscillating curve range, but it can not be totally eliminated.

7. CAES System Improvements

Technical systems can very often be optimised for efficiency, rentability, space, lifetime, and many more. Here are two examples specifically chosen with good optimisation options for a CAES system.

7.1. Pre-Cooling the Air on the Suction Side of the Compressor

As shown in figure 5.9 one compressor can bring much more performance in lower ambient temperatures. One possibility to use this characteristic is to cool down the air before the compressor inlet. This could be done with the cold in the expansion cycles. The temperature in the turbine exit should not be decreased too much, because this would mean the same decrease of performance of the turbine. The characteristic shows that already small temperature decreases lead to high performance increases. This performance increase is an increase of the system economics, not an efficiency increase. To use this approach, a cold storage should be used, because if there is no turbine work (electricity use) during the compression, no cold is produced. If cold could be stored whenever the user generates electricity it could be used to pre-cool whenever the compressor runs. Special storage techniques should be considered in exchange for water, because when temperatures drop below 0 $°C$ water is freezing and heat won't be transported anymore.

7.2. Combined Cycles for Overall Low Energy Houses

Nowadays, in times of engagement for low energy demand and high efficiency installations, it is common to use waste energy from one process for another where it is required. For example cogeneration in thermal power plants like fossil power plants. Heat is emitted during electricity generation because some parts and processes have to be cooled. Instead of "wasting" this heat to the environment via cooling towers for example, it is possible to use this heat with a heat exchanger and a transportation system like pipes to a nearby place like a neighbourhood, school or hospital where heat is required. It is possible to do the same overall energy effective combination of different systems with a CAES installation. Due to predictable heat losses of the warm water storage over time with not optimal insulation, the outflow of the used air will be colder than the surrounding temperature. The colder outlet temperature could be used in this case to cool for example the house in which the CAES system is installed, instead of using an extra AVAC installation. In cold winter months it could also be possible to use the compression heat for heating the building, depending on the CAES system. Especially the PRVs and turbine have to manage low temperatures without freezing. In this case the efficiency of the CAES system would decrease because of lower volume work in the turbine of colder air, but the whole house as focused system could profit from combinations like this. Further analyses have to be done to find out the optimum user points for each installation.

7.3. Liquid Air for Less Storage Volume

Air can change its state as other substances from gaseous, as it is in normal conditions (20 ° C, ambient pressure), to liquid by cooling or extreme isothermal compression. The advantage of liquid air compared to gaseous air as storing media is obvious. The volume decreases considerably with the state change from gaseous to liquid. Less volume means less space for storage and higher energy density. As air consists of different gases, they have different boiling points. The boiling point of oxygen lies at -183 ° C, the boiling point of nitrogen lies at -196 ° C at ambient pressure. In 1895 Carl von Linde developed a technical method to separate gases which enables the separation of atmospheric gases like oxygen, nitrogen and argon in large amounts. The Linde process is based on the Joule-Thomson-Effect of real gases. With this process it is possible to liquify air constantly.

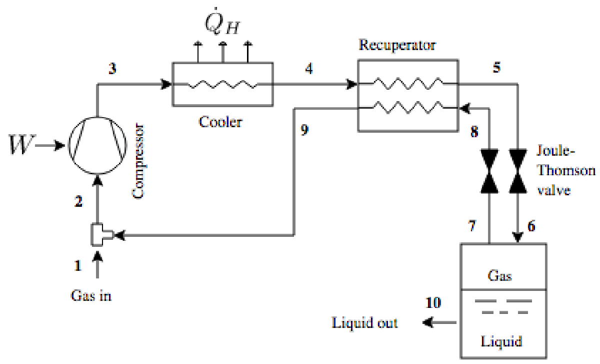

Figure 7.1.: Scheme of the Linde-Hampson cycle
[8]

Figure 7.1 shows a simplified scheme of the process. Surrounding air (20 $°C$, 0 bar) is soaked in and compressed to around 200 bar. The increased temperature is afterwards cooled down back to surrounding temperature with a heat exchanger to the surroundings. The pressurised air is cleaned by a molecular sieve from water vapour, dust, hydrocarbon, nitrous oxide and carbon dioxide. Hydrocarbon and nitrous oxide can lead to deflagration and explosion. Because of the decreased temperature δT it is afterwards possible to expand the pressurised air through a turbine, which cools the air to little above the point of liquification. Then pressure is reduced by a Joule Thomson valve, which decreases the air temperature according the Joule-Thomson effect. If the temperature difference is big enough, air will change its state from gaseous to liquid. If the pressure step was too small and therefore the temperature drop, the exiting cold air is used to cool afterwards the incoming air before expanding in the expansion valve with a counter-flow heat exchanger. The returned air goes through the cycle again. The temperature decrease may now be big enough to reach the liquid state. In an opened vessel, temperature of liquid water stays constant at around -190 $°C$ because it is boiling in these conditions. The vessel is not allowed to be totally closed because of the gas which is generated by boiling. The pressure would increase until some part of the storage would break. It has to be proved, depending on the surrounding conditions, if it is possible to store this liquid several hours or several days. Around the year 1900 an American English joint project produced and demonstrated an automobile driven by liquid air with the claim of a range of 160 kilometres.

8. Conclusion

Nowadays a CAES system becomes meaningful. In the first years of commercial Photovoltaic installations it was highly rentable to sell electricity to the grid. Compensations were artificially high because politics wanted to push the green energy production. For one sold kWh could be bought far more than one kWh which means an economic efficiency > 100 %. The actual development shows that these guaranteed compensations decreased extreme. Nowadays much more kWh electricity have to be sold to buy back one kWh back. In Germany around three kWh for one, which means 33 % economic efficiency, see chapter 3.2. It is impossible to see where this development is going to in the future, but the tendency that it is economically reasonable to store the energy instead of selling it is clear. 33 % efficiency should be beatable by a CAES system. Because around 95 % of the compressor power are transferred into heat [10] it is not very promising to build a CAES system without heat storage, see chapter 2.2. Not only that heat is lost during normal air compression to the surrounding, the air must be heated before and during the expansion in a turbine, because warmer air means more volume means more turbine work see equation 3.23. According the first law of thermodynamic, energy can't be lost in a closed system, the heat which is required to reach initial conditions after the turbine, is exactly the same amount of heat that has to be removed during compression. There are two "good" possibilities to compress air efficiently, adiabatic or isotherm. Isothermal compression is theoretically the best way to compress air, because the compression volume does not increase during compression - which is an exothermal state change of ideal gases, see equation 3.6. Isothermal means, that the heat has to be carried away as soon as it appears, which makes it impossible to cool over the surrounding surfaces. After the second law of thermodynamics therefor temperature gradients are required which are not allowed in the definition of isothermal. The most promising way is to spray water mist directly into the compression chamber, because water will then evaporate, which is an endothermic process. This technique has already been discussed and tested in 1927, but has been dismissed because of technical problems during the grinding parts [14, p.32]. Today there are two examples of isothermal or quasi isothermal processes, the compressor of LightSail chapter 2.6 and the carmaker BMW which injects water in the combustion chamber in it's M4 safety car. One indirect system is also presented from the company CAEstorage chapter 2.4 which reaches a quasi isothermal process because it works extreme slow. These systems seem technically very good, but the complexity is relative high. Also the costs of such an installation should be relative high. It is always important to have a good balanced between technological advance, robustness and price. The future will show if they can establish successful in the market. The second "good" possibility is Advanced Adiabatic - CAES which is thermodynamically researched in this work, chapter 5. Adiabatic means that no heat is exchanged with the surrounding. Because the compressor has to be cooled, the waste heat has to be transferred into a special heat storage, see chapter 4. A standard scuba compressor with 2,2 kW nominal power, three compression stages and inter-cooling is used to compress air up to 300 bar. A circulating water heat storage stores heat until 100 $°C$. Pressurised air is stored in standard gas bottles. A heat exchanger pre heats the high pressurised air before the heated pressure reduction valve. This PRV is chosen by a specialist of the company Samson and also numerically simulated in chapter 6. A second small low pressure air storage equalised the flow losses in the previous piping system. A small low pressure PRV reduces the air for a turbine to produce electricity. The technical conclusion of the example system, based in the great Lisbon area, is written in chapter 5.10. Key statements are

- The 2,2 kW scuba compressor is sufficient for one average domestic person in Portugal who needs
- around 3.137 $\frac{kWh}{d}$
- Circulating water heat storage with 60 l needs very little space and could be replaced ba a continuous heat storage with two tanks
- The amount of pressurised-air-bottles is high with 173 is relative high and needs an assumed total installation space of 24 m^3
- The pneumatic actuated PRV seems relative complex
- Theoretically required temperatures before PRV unreachable, around 1000 $°C$
- Simulations shows good function of PRV with constant jacket temperature of only 50 $°C$
- Isentropic efficiency around 77 %
- According to actual studies, the efficiency of AA-CAES systems is between 50 and 70 % [16, p.541]
- PV cell area 5 - 20 m^2 for the example user

There are still many questions opened because the topic is extensive with many different specialisation possibilities, that future works could be

- Exact system design incl. pipes etc. and drawing, maybe scaled for a prototype
- Total costs of installation of this system
- Search of financial sources for a prototype
- Build a prototype and research the system like start behaviour, noise or efficiency
- Further Flow simulations

- Feasibility of a liquid air storage
- Combined minimum energy houses
- System improvements

A. PRV Selection by Company Samson

Angeb./Auftr. Nr.:		Kunde:	
Projekt:		Datum: 27.04.2016 bearbeitet von:	sfn
Positionsnummer:		Meßstellennummer:	
Durchflußmedium:	Luft	Mediumszustand am Eingang:	gasförmig

SAMSON Ventilauslegung Version 7.09e

Prozeß und Mediumsdaten

			Ventil 1	Ventil 2	Fall 3
Durchfluß	Q	[m³/h]	8.46	8.46	
Eingangsdruck	p1	[bar(a)]	200	105	
Ausgangsdruck	p2	[bar(a)]	105	10	
Eingangstemperatur	t1	[°C]	200	200	
Normdichte	rhon	[kg/m³(N)]	1.29	1.29	
Isentropenexponent	gamma		1.59	1.49	
Realgasfaktor	Z		1.07	1.03	
Viskosität	eta	[mPas]	0.0288	0.0273	

Ergebnisse und Faktoren

Ventilkenngr. errechnet	***Kv***		***0.389***	***0.366***
min erf. Nennweite	DN erf.	[mm]	6.17	15.0
Austrittsgeschw.	w	[Mach]	0.0183	0.108
SDP VDMA 24422 mod.	***LA***	***[dB(A)]***	***78***	***81***
Strömungszustand				Begrenzung
relativer Hub	h	[%]	89.1	87.4
Differenzdruckverhältnis	x		0.47	0.90
FL-Wert	FL		0.97	0.97
xT-Wert	xT		0.80	0.80
Ventilformfaktor	Fd		0.32	0.30
Niveauexponent	G1		-3.97	-3.98
Neigungsexponent	G2		1.50	1.50

Ventildaten

Bauform Ventil			***Durchgangsv.***	***Baureihe***	***250***
Ventilkenngröße	***Kvs***		***0.63***	***Typ***	***3251***
Nennweite	***DN***	***[mm]***	***25***	***Gehäusewerkstoff***	***WN 1.0619***
Nenndruckstufe	***PN***		***250***	***Geräuschminderung***	***keine***
Hub	***H***	***[mm]***	***15***	***Kennlinie***	***gleichproz.***
Sitzbohrung	***SB***	***[mm]***	***6***	***Fließrichtung***	***öffnend***
Kegelstangend.	***Kd***	***[mm]***	***12***	***Druckentlastung***	***ohne (0.0)***
Werkstoff Innenteile	***4404 / 316L***			***Leckageklasse***	***IV***
Stopfbuchse	***PTFE (3.2)***			***Oberteil***	***standard***
Dichtkante	***metallisch (2.0)***				

Rohrdaten

Rohrleitungstyp Stahlrohr	Rohrleitungsisolation keine	D1 [mm] 25	D2 [mm] 25
cR [m/s] 5100	rho [kg/m³] 7800	di [mm] 26.5	s [mm] 3.6

Antriebsdaten

Typ			**3277**	**Sicherheitsst.**	**ausfahrend**		
Membranfläche	**A**	**[cm²]**	**350**	**Stelldruckbereich**	**pstell**	**[bar]**	**0.8 ... 2.4**
				Zuluft	**pzu**	**[bar]**	**2.60**

(Vorgaben: p1max [bar(a)] 200 p2min [bar(a)] 1.01 t1max [°C] 200)

Antriebsergebnisse

erf. Stellkraft	Fo erf.	[kN]	2.45	erf. Diff. pzu-ps100	d ps	[bar]	0.04
zul. Stellkraft	Fzul.	[kN]	27.01	Antriebskraft	Fa	[kN]	2.80
zul. dp am Kegel	d pzul.	[bar]	230.24	Sicherheitsf. schließen	Fa/Fo (SF)		1.14
erf. Anfangswert Stb.	ps0erf.	[bar]	0.77				

B. AA-CAES System Dimensioning

stage	0					0.1						
T_0	V	p	ρ	c_p	h	T	V	p	ρ	c_p	h	P
[°C]	[m^3/min]	[bar]	[kg/m^3]	[kJ/kgK]	[kJ/kgK]	[°C]	[m^3/min]	[bar]	[kg/m^3]	[kJ/kgK]	[kJ/kgK]	[kW]
0	7,725	1	1,28	1,0059	-25,15	203	1,924	7	5,1122	1,0291	180,27	0,484
5	5,944	1	1,25	1,0060	-20,12	212	1,480	7	5,0339	1,0307	189,32	0,372
10	4,509	1	1,23	1,0061	-15,09	221	1,123	7	4,9557	1,0324	198,36	0,282
15	3,456	1	1,21	1,0063	-10,06	229	0,861	7	4,8774	1,0340	207,41	0,216
20	2,594	1	1,19	1,0064	-5,03	238	0,646	7	4,7992	1,0357	216,45	0,162
25	1,923	1	1,17	1,0065	0,00	247	0,479	7	4,7209	1,0373	225,50	0,120
30	1,539	1	1,15	1,0067	5,0360	255	0,383	7	4,6427	1,0390	234,54	0,096
35	1,251	1	1,13	1,0070	10,0720	264	0,312	7	4,5644	1,0406	243,59	0,078
40	0,963	1	1,11	1,0072	15,1080	273	0,240	7	4,4862	1,0423	252,63	0,060
45	0,771	1	1,10	1,0075	20,1440	282	0,192	7	4,4079	1,0439	261,68	0,048
50	0,544	1	1,08	1,0077	25,18	290	0,136	7	4,3297	1,0456	270,72	0,034
0	18,259	1	1,28	1,0059	-25,15	203	4,548	7	5,1122	1,0291	180,27	1,256
5	13,620	1	1,25	1,0060	-20,1200	212	3,393	7	5,0339	1,0307	189,32	0,937
10	10,073	1	1,23	1,0061	-15,0900	221	2,509	7	4,9557	1,0324	198,36	0,693
15	7,632	1	1,21	1,0063	-10,0600	229	1,901	7	4,8774	1,0340	207,41	0,525
20	5,764	1	1,19	1,0064	-5,0300	238	1,436	7	4,7992	1,0357	216,45	0,397
25	4,470	1	1,17	1,0065	0,00	247	1,113	7	4,7209	1,0373	225,50	0,308
30	3,463	1	1,15	1,0067	5,0360	255	0,863	7	4,6427	1,0390	234,54	0,238
35	3,176	1	1,13	1,0070	10,0720	264	0,791	7	4,5644	1,0406	243,59	0,219
40	2,167	1	1,11	1,0072	15,1080	273	0,540	7	4,4862	1,0423	252,63	0,149
45	1,342	1	1,10	1,0075	20,1440	282	0,334	7	4,4079	1,0439	261,68	0,092
50	0,041	1	1,08	1,0077	25,18	290	0,010	7	4,3297	1,0456	270,72	0,003

stage	0.2							0.3						
T_0	T	V	p	ρ	c_p	h	δQ/δt	T	V	p	ρ	c_p	h	P
[°C]	[°C]	[m^3/min]	[bar]	[kg/m^3]	[kJ/kgK]	[kJ/kgK]	[kW]	[°C]	[m^3/min]	[bar]	[kg/m^3]	[kJ/kgK]	[kJ/kgK]	[kW]
0	0	1,104	7	8,9656	1,0179	-26,78	-0,568	206	0,271	50	35,81	1,0513	180,79	0,484
5	5	0,849	7	8,8118	1,0175	-21,6996	-0,437	215	0,208	50	35,25	1,0523	190,04	0,372
10	10	0,644	7	8,6581	1,0172	-16,6152	-0,332	223	0,158	50	34,69	1,0533	199,29	0,282
15	15	0,494	7	8,5043	1,0168	-11,5308	-0,254	232	0,121	50	34,13	1,0542	208,55	0,216
20	20	0,371	7	8,3506	1,0165	-6,4464	-0,190	241	0,091	50	33,58	1,0552	217,80	0,162
25	25	0,275	7	8,1968	1,0161	-1,36	-0,141	250	0,067	50	33,02	1,0561	227,06	0,120
30	30	0,220	7	8,0677	1,0160	3,7164	-0,113	258	0,054	50	32,46	1,0571	236,31	0,096
35	35	0,179	7	7,9386	1,0159	8,7948	-0,092	267	0,044	50	31,90	1,0580	245,56	0,078
40	40	0,138	7	7,8095	1,0158	13,8732	-0,071	276	0,034	50	31,34	1,0590	254,82	0,060
45	45	0,110	7	7,6805	1,0157	18,9516	-0,056	285	0,027	50	30,78	1,0599	264,07	0,048
50	50	0,078	7	7,5514	1,0156	24,03	-0,040	294	0,019	50	30,22	1,0609	273,32	0,034
0	0	2,608	7	8,9656	1,0179	-26,78	-1,343	206	0,640	50	35,81	1,0513	180,79	1,256
5	5	1,946	7	8,8118	1,0175	-21,6996	-1,002	215	0,478	50	35,25	1,0523	190,04	0,937
10	10	1,439	7	8,6581	1,0172	-16,6152	-0,741	223	0,353	50	34,69	1,0533	199,29	0,693
15	15	1,090	7	8,5043	1,0168	-11,5308	-0,561	232	0,268	50	34,13	1,0542	208,55	0,525
20	20	0,823	7	8,3506	1,0165	-6,4464	-0,423	241	0,202	50	33,58	1,0552	217,80	0,397
25	25	0,639	7	8,1968	1,0161	-1,36	-0,328	250	0,157	50	33,02	1,0561	227,06	0,308
30	30	0,495	7	8,0677	1,0160	3,7164	-0,254	258	0,121	50	32,46	1,0571	236,31	0,238
35	35	0,454	7	7,9386	1,0159	8,7948	-0,233	267	0,111	50	31,90	1,0580	245,56	0,219
40	40	0,310	7	7,8095	1,0158	13,8732	-0,159	276	0,076	50	31,34	1,0590	254,82	0,149
45	45	0,192	7	7,6805	1,0157	18,9516	-0,098	285	0,047	50	30,78	1,0599	264,07	0,092
50	50	0,006	7	7,5514	1,0156	24,03	-0,003	294	0,001	50	30,22	1,0609	273,32	0,003

stage	0.4							0.5						
T_0	T	V	p	ρ	c_p	h	δQ/δt	T	V	p	ρ	c_p	h	P
[°C]	[°C]	[m^3/min]	[bar]	[kg/m^3]	[kJ/kgK]	[kJ/kgK]	[kW]	[°C]	[m^3/min]	[bar]	[kg/m^3]	[kJ/kgK]	[kJ/kgK]	[kW]
0	0	0,154	50	65,21	1,1075	-38,10	-0,638	133	0,057	200	159,03	1,1424	93,9180	0,484
5	5	0,119	50	63,97	1,1033	-32,62	-0,489	140	0,044	200	155,70	1,1381	102,3823	0,372
10	10	0,090	50	62,72	1,0990	-27,14	-0,369	148	0,034	200	152,37	1,1337	110,8465	0,282
15	15	0,069	50	61,48	1,0948	-21,66	-0,281	155	0,026	200	149,04	1,1304	119,2251	0,216
20	20	0,052	50	60,24	1,0905	-16,18	-0,209	162	0,019	200	145,71	1,1277	127,5630	0,162
25	25	0,038	50	59,00	1,0863	-10,70	-0,154	170	0,014	200	142,39	1,1249	135,9009	0,120
30	30	0,031	50	57,99	1,0835	-5,31	-0,123	177	0,011	200	139,06	1,1222	144,2388	0,096
35	35	0,025	50	56,99	1,0807	0,09	-0,099	185	0,009	200	135,73	1,1194	152,5767	0,078
40	40	0,019	50	55,98	1,0778	5,48	-0,076	192	0,007	200	132,40	1,1167	160,9146	0,060
45	45	0,015	50	54,97	1,0750	10,88	-0,061	200	0,006	200	129,07	1,1138	169,68	0,048
50	50	0,011	50	53,97	1,0722	16,27	-0,043	207	0,004	200	125,74	1,1130	177,4770	0,034
0	0	0,365	50	65,21	1,1075	-38,10	-1,508	183	0,102	300	136,69	1,1517	150,1578	1,256
5	5	0,272	50	63,97	1,1033	-32,62	-1,119	191	0,076	300	132,96	1,1476	159,5198	0,937
10	10	0,201	50	62,72	1,0990	-27,14	-0,823	199	0,056	300	129,22	1,1435	168,8818	0,693
15	15	0,153	50	61,48	1,0948	-21,66	-0,620	208	0,042	300	125,48	1,1416	178,1210	0,525
20	20	0,115	50	60,24	1,0905	-16,18	-0,465	216	0,032	300	121,74	1,1399	187,3487	0,397
25	25	0,089	50	59,00	1,0863	-10,70	-0,358	224	0,025	300	118,01	1,1383	196,5764	0,308
30	30	0,069	50	57,99	1,0835	-5,31	-0,276	233	0,019	300	114,27	1,1366	205,8041	0,238
35	35	0,064	50	56,99	1,0807	0,09	-0,252	241	0,018	300	110,53	1,1349	215,0318	0,219
40	40	0,043	50	55,98	1,0778	5,48	-0,171	249	0,012	300	106,79	1,1333	224,2594	0,149
45	45	0,027	50	54,97	1,0750	10,88	-0,106	258	0,007	300	103,06	1,1316	233,4871	0,092
50	50	0,001	50	53,97	1,0722	16,27	-0,003	266	0,000	300	99,32	1,1300	242,7148	0,003

stage	1							Σ 0-1		0-1			
T_0	T	V	p	ρ	cp	h	δQ/δt	δQ/δt	p	L a	P a	P is	η is
[°C]	[°C]	[m^3/min]	[bar]	[kg/m^3]	[kJ/kgK]	[kJ/kgK]	[kW]	[kW]	[kW]	[mkg]	[kW]	[kW]	[%]
0	12	0,0403	200	240,83	1,3191	-52,0376	-0,430	-1,636	1,451	532356	1,451	1,115	0,769
5	17	0,0310	200	235,40	1,3045	-45,4616	-0,326	-1,252	1,116	409597	1,116	0,858	0,769
10	22	0,0235	200	229,98	1,2899	-38,8856	-0,243	-0,944	0,847	310728	0,847	0,651	0,769
15	27	0,0180	200	225,03	1,2771	-32,4312	-0,184	-0,719	0,649	238172	0,649	0,499	0,769
20	32	0,0135	200	220,79	1,2669	-26,1592	-0,137	-0,536	0,487	178751	0,487	0,374	0,769
25	37	0,0100	200	216,56	1,2567	-19,8872	-0,100	-0,395	0,361	132496	0,361	0,278	0,769
30	42	0,0080	200	212,32	1,2465	-13,6152	-0,080	-0,315	0,289	106064	0,289	0,222	0,769
35	47	0,0065	200	208,09	1,2363	-7,3432	-0,064	-0,255	0,235	86230	0,235	0,181	0,769
40	52	0,0050	200	204,19	1,2273	-1,1584	-0,049	-0,196	0,181	66371	0,181	0,139	0,769
45	57	0,0040	200	200,77	1,2200	4,8956	-0,039	-0,156	0,145	53127	0,145	0,111	0,769
50	62	0,0028	200	197,36	1,2126	10,9496	-0,027	-0,110	0,102	37495	0,102	0,079	0,769
0	12	0,0635	300	240,83	1,3191	-60,5636	-0,956	-3,808	3,769	1383295	3,769	2,838	0,753
5	17	0,0474	300	235,40	1,3045	-53,7276	-0,703	-2,824	2,812	1031837	2,812	2,117	0,753
10	22	0,0350	300	229,98	1,2899	-46,8916	-0,511	-2,076	2,080	763152	2,080	1,566	0,753
15	27	0,0265	300	225,03	1,2771	-40,1764	-0,382	-1,563	1,576	578223	1,576	1,186	0,753
20	32	0,0200	300	220,79	1,2669	-33,6424	-0,286	-1,174	1,190	436692	1,190	0,896	0,753
25	37	0,0155	300	216,56	1,2567	-27,1084	-0,219	-0,905	0,923	338660	0,923	0,695	0,753
30	42	0,0120	300	212,32	1,2465	-20,5744	-0,168	-0,698	0,715	262356	0,715	0,538	0,753
35	47	0,0110	300	208,09	1,2363	-14,0404	-0,153	-0,638	0,656	240641	0,656	0,494	0,753
40	52	0,0075	300	204,19	1,2273	-7,5992	-0,103	-0,433	0,447	164172	0,447	0,337	0,753
45	57	0,0046	300	200,77	1,2200	-1,2972	-0,063	-0,267	0,277	101689	0,277	0,209	0,753
50	62	0,0001	300	197,36	1,2126	5,0048	-0,002	-0,008	0,009	3131	0,009	0,006	0,753

stage	2						3					
T_0	T	V	p	ρ	c_p	h	T	V	p	ρ	c_p	h
[°C]	[°C]	[m^3/min]	[bar]	[kg/m^3]	[kJ/kgK]	[kJ/kgK]	[°C]	[m^3/min]	[bar]	[kg/m^3]	[kJ/kgK]	[kJ/kgK]
0	12	0,0403	200	240,83	1,3191	-52,0376	12	0,037	200	240,83	1,3191	-52,0376
5	17	0,0310	200	235,40	1,3045	-45,4616	17	0,037	200	235,40	1,3045	-45,4616
10	22	0,0235	200	229,98	1,2899	-38,8856	22	0,037	200	229,98	1,2899	-38,8856
15	27	0,0180	200	225,03	1,2771	-32,4312	27	0,037	200	225,03	1,2771	-32,4312
20	32	0,0135	200	220,79	1,2669	-26,1592	32	0,037	200	220,79	1,2669	-26,1592
25	37	0,0100	200	216,56	1,2567	-19,8872	37	0,037	200	216,56	1,2567	-19,8872
30	42	0,0080	200	212,32	1,2465	-13,6152	42	0,037	200	212,32	1,2465	-13,6152
35	47	0,0065	200	208,09	1,2363	-7,3432	47	0,036	200	208,09	1,2363	-7,3432
40	52	0,0050	200	204,19	1,2273	-1,1584	52	0,036	200	204,19	1,2273	-1,1584
45	57	0,0040	200	200,77	1,2200	4,8956	57	0,036	200	200,77	1,2200	4,8956
50	62	0,0028	200	197,36	1,2126	10,9496	62	0,036	200	197,36	1,2126	10,9496
0	12	0,0635	300	240,83	1,3191	-60,5636	12	0,024	300	240,83	1,3191	-60,5636
5	17	0,0474	300	235,40	1,3045	-53,7276	17	0,024	300	235,40	1,3045	-53,7276
10	22	0,0350	300	229,98	1,2899	-46,8916	22	0,024	300	229,98	1,2899	-46,8916
15	27	0,0265	300	225,03	1,2771	-40,1764	27	0,024	300	225,03	1,2771	-40,1764
20	32	0,0200	300	220,79	1,2669	-33,6424	32	0,024	300	220,79	1,2669	-33,6424
25	37	0,0155	300	216,56	1,2567	-27,1084	37	0,024	300	216,56	1,2567	-27,1084
30	42	0,0120	300	212,32	1,2465	-20,5744	42	0,024	300	212,32	1,2465	-20,5744
35	47	0,0110	300	208,09	1,2363	-14,0404	47	0,024	300	208,09	1,2363	-14,0404
40	52	0,0075	300	204,19	1,2273	-7,5992	52	0,024	300	204,19	1,2273	-7,5992
45	57	0,0046	300	200,77	1,2200	-1,2972	57	0,024	300	200,77	1,2200	-1,2972
50	62	0,0001	300	197,36	1,2126	5,0048	62	0,024	300	197,36	1,2126	5,0048

stage	4							5					
T_0	T	V	p	ρ	c_p	h	δQ/δt	T	V	p	ρ	c_p	h
[°C]	[°C]	[m^3/min]	[bar]	[kg/m^3]	[kJ/kgK]	[kJ/kgK]	[kW]	[°C]	[m^3/min]	[bar]	[kg/m^3]	[kJ/kgK]	[kJ/kgK]
0	968	0,160	200	56,2	1,1589	1043,30	1,954	254	1,356	10	6,6	1,0409	232,5
5	991	0,160	200	55,2	1,1628	1070,82	1,963	264	1,356	10	6,5	1,0426	243,5
10	1013	0,160	200	54,3	1,1665	1096,69	1,967	274	1,356	10	6,4	1,0441	253,2
15	1036	0,160	200	53,3	1,1702	1123,39	1,972	283	1,356	10	6,3	1,0457	263,2
20	1059	0,160	200	52,4	1,1740	1152,07	1,975	293	1,356	10	6,2	1,0473	273,3
25	1082	0,160	200	51,4	1,1777	1179,05	1,977	302	1,356	10	6,1	1,0489	283,3
30	1104	0,160	200	50,6	1,1815	1206,03	1,984	312	1,356	10	6,0	1,0505	293,3
35	1127	0,160	200	49,8	1,1852	1233,01	1,990	322	1,356	10	5,9	1,0521	303,4
40	1150	0,160	200	49,0	1,1890	1259,99	1,995	331	1,356	10	5,8	1,0537	313,4
45	1173	0,160	200	48,2	1,1928	1286,97	1,998	341	1,356	10	5,7	1,0553	323,4
50	1195	0,160	200	47,4	1,1965	1313,95	2,001	351	1,356	10	5,6	1,0569	333,4
0	1120	0,119	300	75,0	1,1842	1225,18	2,399	254	1,356	10	6,6	1,0409	232,5
5	1146	0,119	300	73,8	1,1884	1255,47	2,408	264	1,356	10	6,5	1,0425	243,2
10	1172	0,119	300	72,5	1,1926	1285,77	2,416	274	1,356	10	6,4	1,0441	253,2
15	1197	0,119	300	71,2	1,1968	1316,06	2,422	283	1,356	10	6,3	1,0457	263,2
20	1223	0,119	300	70,0	1,2010	1346,35	2,426	293	1,356	10	6,2	1,0473	273,3
25	1248	0,119	300	68,7	1,2052	1376,65	2,429	302	1,356	10	6,1	1,0489	283,3
30	1274	0,119	300	67,6	1,2095	1406,94	2,438	312	1,356	10	6,0	1,0505	293,3
35	1299	0,119	300	66,6	1,2137	1437,24	2,446	322	1,356	10	5,9	1,0521	303,4
40	1325	0,119	300	65,5	1,2179	1467,53	2,452	331	1,356	10	5,8	1,0537	313,4
45	1350	0,119	300	64,4	1,2221	1497,82	2,457	341	1,356	10	5,7	1,0553	323,4
50	1376	0,119	300	63,4	1,2263	1528,12	2,460	351	1,356	10	5,6	1,0569	333,4

stage	6						7						
T_0	T	V	p	ρ	c_p	h	T	V	p	ρ	c_p	h	δQ/δt
[°C]	[°C]	[m^3/min]	[bar]	[kg/m^3]	[kJ/kgK]	[kJ/kgK]	[°C]	[m^3/min]	[bar]	[kg/m^3]	[kJ/kgK]	[kJ/kgK]	[kW]
0	254	1,356	10	6,6	1,0409	232,5	254	1,355	10	6,6	1,0409	232,5	0,132
5	264	1,356	10	6,5	1,0426	243,5	264	1,355	10	6,5	1,0426	243,5	0,132
10	274	1,356	10	6,4	1,0441	253,2	274	1,355	10	6,4	1,0441	253,2	0,132
15	283	1,356	10	6,3	1,0457	263,2	283	1,355	10	6,3	1,0457	263,2	0,133
20	293	1,356	10	6,2	1,0473	273,3	293	1,355	10	6,2	1,0473	273,3	0,133
25	302	1,356	10	6,1	1,0489	283,3	302	1,355	10	6,1	1,0489	283,3	0,133
30	312	1,356	10	6,0	1,0505	293,3	312	1,355	10	6,0	1,0505	293,3	0,133
35	322	1,356	10	5,9	1,0521	303,4	322	1,355	10	5,9	1,0521	303,4	0,133
40	331	1,356	10	5,8	1,0537	313,4	331	1,355	10	5,8	1,0537	313,4	0,133
45	341	1,356	10	5,7	1,0553	323,4	341	1,355	10	5,7	1,0553	323,4	0,134
50	351	1,356	10	5,6	1,0569	333,4	351	1,355	10	5,6	1,0569	333,4	0,134
0	254	1,356	10	6,6	1,0409	232,5	254	1,355	10	6,6	1,0409	232,5	0,132
5	264	1,356	10	6,5	1,0425	243,2	264	1,355	10	6,5	1,0425	243,2	0,132
10	274	1,356	10	6,4	1,0441	253,2	274	1,355	10	6,4	1,0441	253,2	0,132
15	283	1,356	10	6,3	1,0457	263,2	283	1,355	10	6,3	1,0457	263,2	0,133
20	293	1,356	10	6,2	1,0473	273,3	293	1,355	10	6,2	1,0473	273,3	0,133
25	302	1,356	10	6,1	1,0489	283,3	302	1,355	10	6,1	1,0489	283,3	0,133
30	312	1,356	10	6,0	1,0505	293,3	312	1,355	10	6,0	1,0505	293,3	0,133
35	322	1,356	10	5,9	1,0521	303,4	322	1,355	10	5,9	1,0521	303,4	0,133
40	331	1,356	10	5,8	1,0537	313,4	331	1,355	10	5,8	1,0537	313,4	0,133
45	341	1,356	10	5,7	1,0553	323,4	341	1,355	10	5,7	1,0553	323,4	0,134
50	351	1,356	10	5,6	1,0569	333,4	351	1,355	10	5,6	1,0569	333,4	0,134

stage	8							9						
T_0	T	V	p	ρ	c_p	h	δQ/δt	T	V	p	ρ	c_p	h	P
[°C]	[°C]	[m^3/min]	[bar]	[kg/m^3]	[kJ/kgK]	[kJ/kgK]	[kW]	[°C]	[m^3/min]	[bar]	[kg/m^3]	[kJ/kgK]	[kJ/kgK]	[kW]
0	203	1,748	7	5,1	1,0327	180,23	0,515	0	7,019	1	1,28	1,0059	-25,1500	2,000
5	212	1,748	7	5,0	1,0341	189,45	0,517	5	7,019	1	1,25	1,0060	-20,1200	2,000
10	221	1,748	7	4,9	1,0354	198,10	0,517	10	7,019	1	1,23	1,0061	-15,0900	2,000
15	229	1,748	7	4,9	1,0367	207,04	0,517	15	7,019	1	1,21	1,0063	-10,0600	2,000
20	238	1,748	7	4,8	1,0380	215,97	0,517	20	7,019	1	1,19	1,0064	-5,0300	2,000
25	247	1,748	7	4,7	1,0393	224,91	0,517	25	7,019	1	1,17	1,0065	0,0000	2,000
30	255	1,748	7	4,6	1,0407	233,84	0,518	30	7,019	1	1,15	1,0067	5,0360	2,000
35	264	1,748	7	4,5	1,0420	243,59	0,518	35	7,019	1	1,13	1,0070	10,0720	2,000
40	273	1,748	7	4,5	1,0433	252,63	0,519	40	7,019	1	1,11	1,0072	15,1080	2,000
45	282	1,748	7	4,4	1,0446	261,68	0,519	45	7,019	1	1,10	1,0075	20,1440	2,000
50	290	1,748	7	4,3	1,0459	270,72	0,519	50	7,019	1	1,08	1,0077	25,1800	2,000
0	203	1,748	7	5,1	1,0327	180,23	0,515	0	7,019	1	1,28	1,0059	-25,1500	2,000
5	212	1,748	7	5,0	1,0340	189,17	0,516	5	7,019	1	1,25	1,0060	-20,1200	2,000
10	221	1,748	7	4,9	1,0354	198,10	0,517	10	7,019	1	1,23	1,0061	-15,0900	2,000
15	229	1,748	7	4,9	1,0367	207,04	0,517	15	7,019	1	1,21	1,0063	-10,0600	2,000
20	238	1,748	7	4,8	1,0380	215,97	0,517	20	7,019	1	1,19	1,0064	-5,0300	2,000
25	247	1,748	7	4,7	1,0393	224,91	0,517	25	7,019	1	1,17	1,0065	0,0000	2,000
30	255	1,748	7	4,6	1,0407	233,84	0,518	30	7,019	1	1,15	1,0067	5,0360	2,000
35	264	1,748	7	4,5	1,0420	243,59	0,518	35	7,019	1	1,13	1,0070	10,0720	2,000
40	273	1,748	7	4,5	1,0433	252,63	0,519	40	7,019	1	1,11	1,0072	15,1080	2,000
45	282	1,748	7	4,4	1,0446	261,68	0,519	45	7,019	1	1,10	1,0075	20,1440	2,000
50	290	1,748	7	4,3	1,0459	270,72	0,519	50	7,019	1	1,08	1,0077	25,1800	2,000

C. Ansys Fluent Simulation

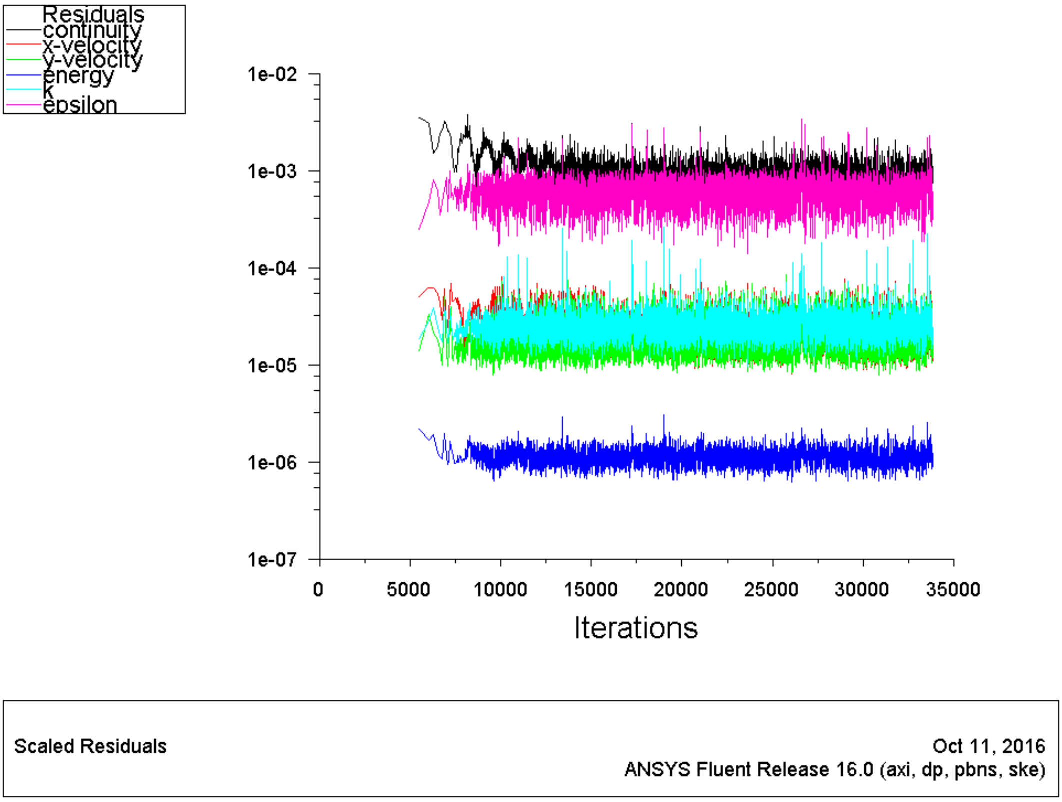

Figure C.1.: Simulation residuals

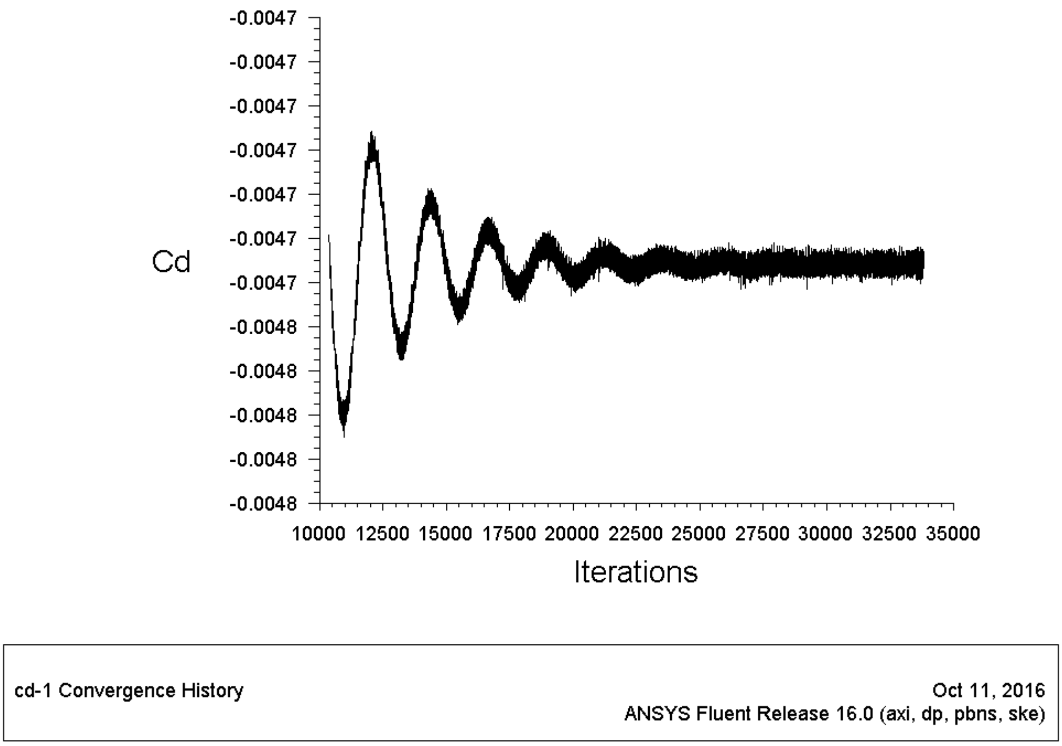

Figure C.2.: Simulation convergence history

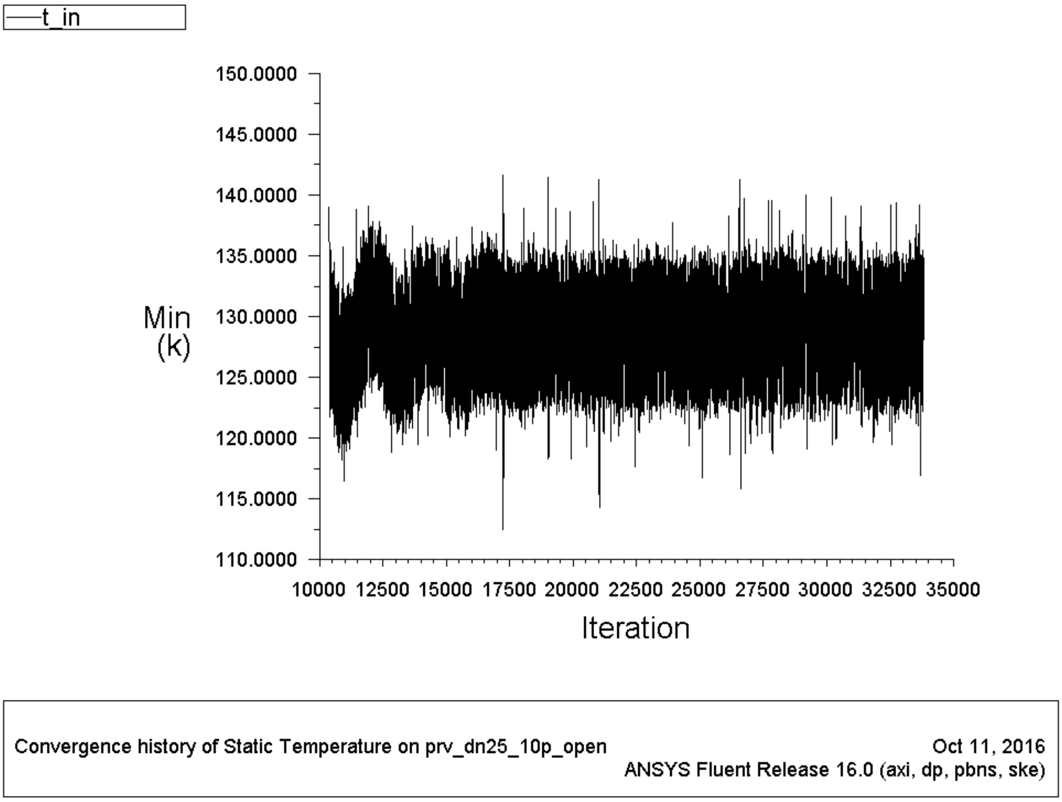

Figure C.3.: Simulation temperature convergence

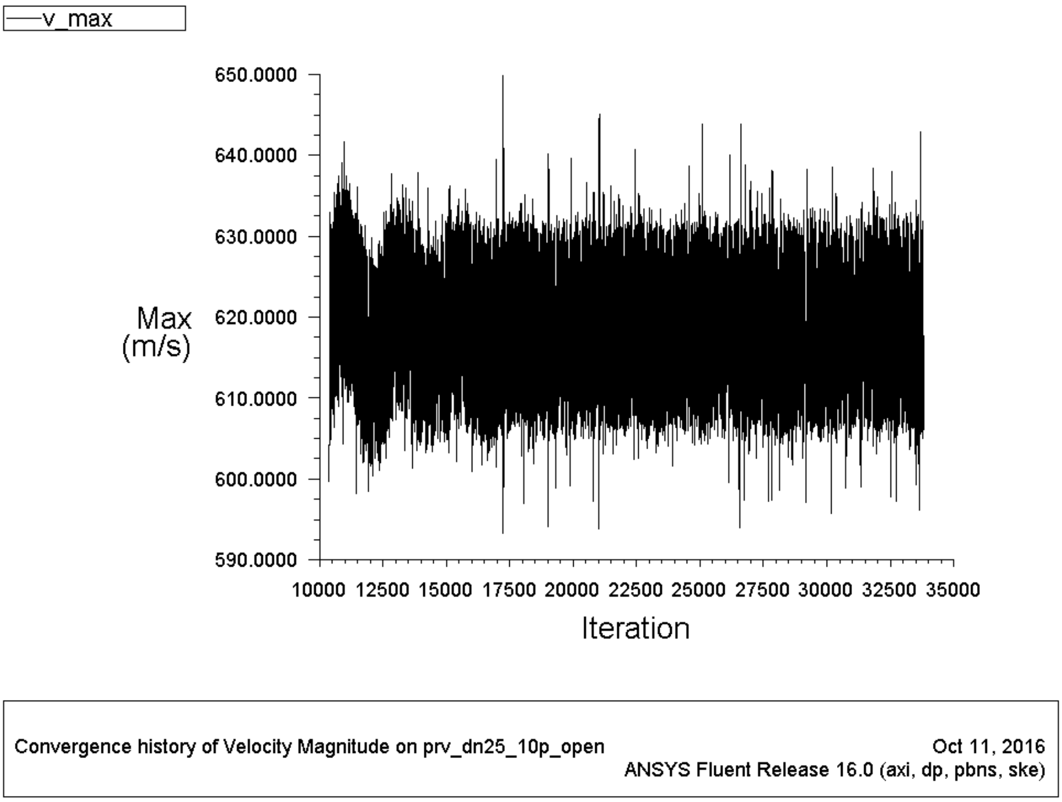

Figure C.4.: Simulation maximum velocity convergence

Bibliography

[1] Lightsail energy.

[2] Principle of electricity generation by photovoltaic cells.

[3] *Illustrated Experiments in Fluid Mechanics.* National Committee for Fluid Mechanics Films, Education Development Center, Inc., 1972.

[4] Energia em portugal. Technical report, Direção-Geral de Energia e Geologia, March 2016.

[5] Samson AG. Bauart 240 . 250 . 280 . 290 . baureihe v2001 . pfeiffer. Weismüllerstraße 3, 60314 Frankfurt am Maiun, Germany, 02 2015.

[6] Samson AG. Bauart 250 - pneumatische stellventile typ 3251-1 und typ 3251 - 7. Weismüllerstraße 3, 60314 Frankfurt am Maiun, Germany, 11 2015.

[7] BBC. Energieversorgung. *Brown Boveri Mitteilungen*, 67(8):465–471, 1980.

[8] SmoWeb by SysMo. Linde-hampson cycle.

[9] Oliver Romberg Dirk Labuhn. *Keine Panik vor Thermodynamik!* Vieweg, Wiesbaden, 2007.

[10] Krinner Drucklufttechnik.

[11] A. Wörner H. Müller-Steinhagen F. Schaube, L. Koch. A thermodynamic and kinetic study of the de- and rehydration of ca(oh)2 at high h2o partial pressures for thermochemical heat storage. *Thermochimica Acta*, 538:9–20, June 2012.

[12] Mike Blicker Fraunhofer. Sorption energy storage and conversion for cooling and heating, March 2014.

[13] Bauer Kompressoren GmbH. Instruction manual - pe 100-t, 2010. [Online; accessed 01.06.2015].

[14] Adolf Hinz. *Thermodynamische Grundlagen der Kolben- und Turbokompressoren.* Julius Springer, Berlin, 1927.

[15] Verein Deutscher Ingenieure. *VDI Wärmeatlas.* Springer, Berlin, Heidelberg, 2006.

[16] C. Kruck L. Eltrop Niklas Hartmann, O. Vöhringer. Simulation and analysis of different adiabatic compressed air energy storage plant configurations. *Applied Energy*, (93):541–548, 2012.

[17] Helmut Schaefer. *VDI-Lexikon Energietechnik*. Helmut Schaefer, 1994.

[18] F. White. *Fluid Mechanics*. McGraw-Hill, 1998.

[19] Wikipedia. Demographics of portugal.

[20] Wikipedia. Photovoltaics.

[21] Dr. Harry Wirth. Aktuelle fakten zur photovoltaik in deutschland. Fraunhofer ISE, Heidenhofstraße 2, 79110 Freiburg, 12 2015.

[22] Dr. Jörg Worlitschek. Kti antrag. 10 2015.

[23] Dr. Jörg Worlitschek. Thermische nenergiespeicher - projekt: Latent. page 42. Lucerne - School of Engineering and Architecture, 10 2015.

[24] Richard A. Zahoransky. *Energietechnik*. Vieweg+Teubner, Wiesbaden, 2009.