Diploma Thesis

Multiple Energy Recovery from CO_2 Reservoirs by Smart Technical Solutions

of

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submitted to the

University of Leoben

created on the

Chair of Conveying Technology

in collaboration with the

Chair of Petroleum Production and Processing

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Leoben, June 2013

Diploma Thesis

for Julia KOPPLER

Subject: Multiple Energy Recovery from CO_2 Reservoirs by Smart Technical Solutions

Task:

It is to examine if energy production in other than hydrocarbon reservoirs could be useful and possibly bring an economic advantage. The reservoir should be supplied with supercritical CO_2, whereby a CO_2-methane mixture is produced from the production well. For the energy generation itself, different technologies should be used and considered in more detail.

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Affidavit

I declare in lieu of oath, that I wrote this thesis and performed the associated research myself, using only literature cited in this volume.

Leoben, June 2013 Julia Koppler

Acknowledgements

My special thanks go to the directorate of the Chair of Conveying Technology Univ.-Prof. Dr.mont. Dipl.-Ing. Franz Kessler and the directorate of the Chair of Petroleum Production and Processing Univ.-Prof. Dr.mont. Dipl.-Ing. Herbert Hofstätter for the opportunity of creating this thesis, as well as for reviewing the result.

My friends and fellow students, I would like to thank for their many years of help and support.

Last but not least I want to thank with all my heart my family. I thank my parents for the exceptional love and selfless support in every situation I can remember.

In the middle of difficulty

lies the opportunity.

(Albert Einstein)

With great gratitude, I dedicate this work to my parents Franz und Hildegard Koppler.

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P	Pressure	[bar] / [Pa]
∇p	Pressure gradient	-
η	Efficiency	-
η_C	Efficiency of the Carnot cycle	-
T	Temperature	[°C] / [K]
γ	Heat capacity ratio	-
P	Performance	[J/s] / [W]
Q_{sup}	Supplied quantity of heat	[J/s] / [W]
V	Volume	[m^3]
κ	Isentropic exponent	-
W_t	Technical work	[J/s] / [W]
$\dot{m}$	Mass flow	[kg/s]
c_p	Specific heat	[kJ/kgK]
ΔT	Temperature difference	[°C] / [K]
M	Molar mass	[g/mol]
w	Substance by weight	-
ρ	Density	[kg/m^3]
P_T	Turbine performance	[J/s] / [W]
ΔH	Calorific value	[MJ/kg]
η_{th}	Thermal efficiency	-
η_{mech}	Mechanical efficiency	-
η_{el}	Electric efficiency	-
QG	Quality grade	-
$Q_{exhaust}$	Quantity of heat in the exhaust gas	[J/s] / [W]
$\dot{n}$	Amount of substance	[mol/s]
h	Enthalpy	[kJ/kg]
Δh	Enthalpy difference	[kJ/kg]
P_{pump}	Pump performance	[J/s] / [W]
Δp	Pressure difference	[bar] / [Pa]
$\dot{V}$	Volume flow	[m^3/s]
P_{stc}	Performance of the steam cycle	[J/s] / [W]

Abstract

This thesis deals with the possibility to use other than hydrocarbon reservoirs for power generation with various technologies. Fact is that there are reservoirs that are not really worthy for hydrocarbon production and the idea rises to use them to generate electricity. The aim of this thesis is the further consideration of these technologies in terms of their usefulness and the description of a possible process cycle.

We inject supercritical CO_2 into the reservoir to produce in our case a gaseous CO_2-methane mixture. Supercritical CO_2 has certain advantages over water because of its special physical state, which is between liquid and gaseous. The produced mixture can then further be used by a variety of different energy cycles.

In this case, we relax the mixture first in an expander to reduce the high pressure while producing already a part of our desired electrical energy. However, it is important to ensure that the medium doesn't get too cool. We now separate our mixture. The gaseous CO_2 is then transformed again into its supercritical state by using a compressor, and if necessary a heat exchanger, so the circuit can be closed. The methane can then be used with the help of a combined cycle plant for the mean power generation. Combined cycle power plants consist basically of a gas turbine and a steam turbine process, which uses the exhaust gas temperatures of the gas turbine through a heat exchanger. Afterwards the remaining exhaust gases may be further used in a thermal power plant.

In this thesis, you can find accurate data and facts about the significance and magnitude of the potential energy to be recovered on a specific example. The amount of potential energy and the process flow may differ depending on the input parameters. The relatively high amount of power that can be generated depends of course on the input parameters. Although the investment costs will be slightly higher, since most machine components must be tailored according to the parameters.

1. Introduction

Since our current energy system depends very much on exhaustible energy resources, it is a major challenge to the 21st century to create a more sustainable energy supply. In order for environmental relief and contribute to resource protection, not only a more rational use of energy is needed, but also the increased use of renewable energies. Local energy sources in individual countries and regions need to be integrated into national and regional systems to ensure the best possible profit. Wind and solar power plants are already known and increasingly utilized, so it is also time to exploit the potential of geothermal energy, if we want to find solutions to regional and global energy problems.

Although geothermal energy has been used commercially since 1913 to generate electricity, the utilization has only increased rapidly in the last three decades. In over 80 countries spread all over the world geothermal resources have been found, while in 58 countries quantified records of geothermal utilization exist. There is still sufficient potential to accelerate the utilization of geothermal energy, because until now only a small fraction of the geothermal potential has been exploited. The future development of geothermal utilization is a question of economic and political competitiveness with other energy sources.[1]

In recent years, it is more and more considered to replace the current working fluid in the geothermal energy recovery, which is high pressure water, with supercritical CO_2. Those considerations were made because of the rising problems caused by the greenhouse effect and the need to reduce the greenhouse gas CO_2. There is the possibility to combine geothermal energy recovery with simultaneous geologic sequestration of CO_2 as an ancillary benefit. Other reasons for the use of supercritical CO_2 as working fluid are discovered gas reservoirs with high CO_2 content, where the economic hydrocarbon production was not purposeful until now.

Supercritical CO_2 as working fluid has some significant advantages over water, including larger heat extraction rates, better rock-fluid interactions, like porosity enhancement and reservoir growth, and more favorable wellbore hydraulics, due to its higher compressibility, higher expansivity and lower viscosity.[2]

Up to now there were no tests made on the field, but simple models predict gains in power generation by geothermal energy.[3]

[1] (Flidleifsson, 2001)
[2] (Pruess & Azaroual, 2006)
[3] (Dunn, 2009)

2. Advantages and Disadvantages of $SCCO_2$ versus Water for Geothermal Energy Recovery

If we use supercritical carbon dioxide as working fluid in Enhanced Geothermal Systems some advantages compared to water will possibly follow. These advantages include higher energy yield because of its lower viscosity, better wellbore hydraulics through larger expansivity and compressibility, higher heat extraction, more favorable chemical interaction with surrounding rock structure, CO_2 sequestration due to unavoidable fluid losses and higher temperature limit.

In order to achieve a higher energy yield, it is necessary to reduce the power consumption of the fluid circulation system. This is supported by the larger expansivity, which generates larger density differences between the cold CO_2 in the injection well and the hot CO_2 in the production well and provides by that larger buoyancy forces to reduce the power consumption. In addition, the lower viscosity leads to higher flow velocities for a given pressure gradient.

The favorable chemical interaction with the surrounding rock structure would lead on the one hand to fewer or no more scaling problems due to the fact that CO_2 is much less effective as a solvent than water, and on the other hand to porosity increase and reservoir growth initiated by mineral transformations which would be accompanied by a reduction of volume. Moreover the storage of CO_2 would be quite rapid at elevated temperatures, because CO_2 is an attractive heat transmission fluid with excellent thermophysical properties.

All these effects will help to compensate the lower mass heat capacity of CO_2.

CO_2 would be injected in the reservoir over a period of time to displace and finally remove the resident water at least from the central zone of the reservoir, first by immiscible displacement and later by dissolution into the CO_2 stream. During this procedure different fluids will be produced, among them a single aqueous phase, some water-CO_2 mixtures and eventually dry CO_2. Three clearly defined zones are formed, an inner zone with dry CO_2 because all water is completely removed, surrounding zones with two-phase water-CO_2 mixtures and an outer zone with a single aqueous phase with some dissolved CO_2.[4]

2.1. Physical Properties of CO_2

Supercritical carbon dioxide is a fluid state of carbon dioxide at or above its critical point, which is defined by a critical temperature of 31.1°C, a critical pressure of 73.8 bar and a critical density of 464 kg/m^3.

[4] (Pruess K. , 2007)

The phase diagrams of carbon dioxide and water are presented in Figure 1[5] and 2[6], showing the different physical states of the substances.

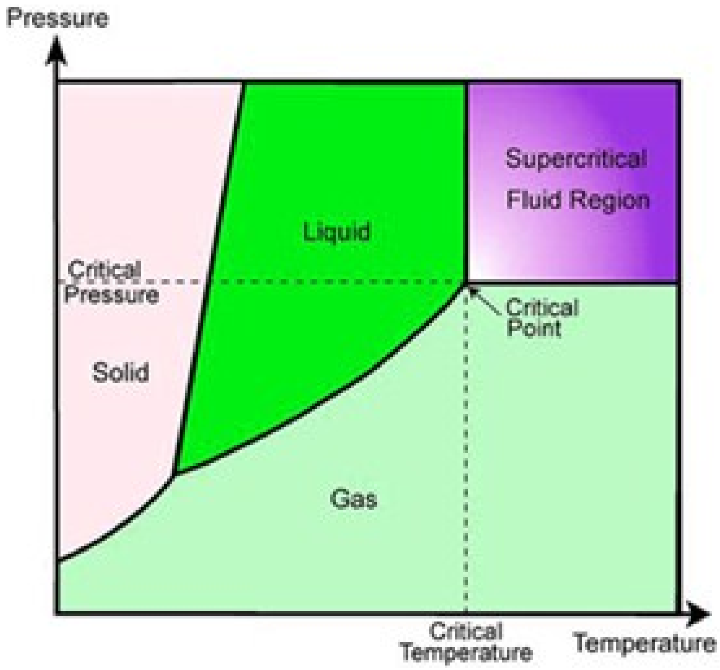


Figure 1: Phase Diagram of CO_2

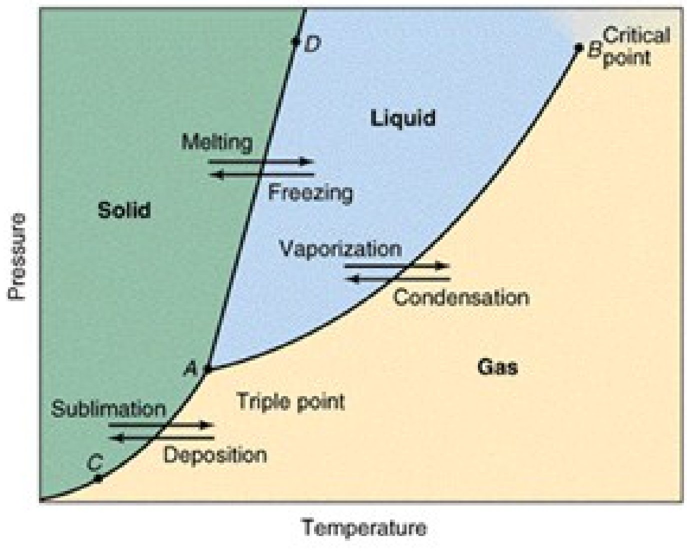


Figure 2: Phase Diagram of Water

As you can see carbon dioxide possesses in contrast to water a supercritical region above the critical point mentioned before. In this region the so called supercritical carbon dioxide can

5 (http://www.separex.fr/processDevelopment.php, 2011)
6 (http://crescentok.com/staff/jaskew/ISR/chemistry/class16.htm, 2011)

change continuously into gaseous or liquid CO_2 without phase boundaries, for example, it fills a container with the viscosity of a gas but with the density of a liquid.

The triple point of CO_2 at a temperature of -56.6°C and a pressure of 5.185 bar is the point at which gas, liquid and solid phases coexist in thermodynamic equilibrium.

Table 1[7] below shows the physical properties of solid CO_2.

Property	Value	Unit
Density	1562	kg/m³
Latent heat of fusion at triple point and 1.013 bar	196.104	kJ/kg

Table 1: Physical Properties of Solid CO_2

Table 2[7] lists the physical properties of liquid CO_2.

Property	Value	Unit
Density at -20°C and 19.7 bar	1032	kg/m³
Boiling point	-57	°C
Latent heat of vaporization at boiling point and 1.013 bar	571.08	kJ/kg
Vapor pressure at 20°C and 58.5 bar	58.5	bar
Liquid/Gas equivalent at 15°C and 1.013 bar	845	vol/vol

Table 2: Physical Properties of Liquid CO_2

In Table 3[7] the physical properties of gaseous CO_2 are presented.

Property	Value	Unit
Sublimation point	−78.5	°C
Gas density at sublimation point and 1.013 bar	2.814	kg/m³
Gas density at 15°C and 1.013 bar	1.87	kg/m³
Viscosity at 0°C and 1.013 bar	0.0001372	P
Compressibility factor at 15°C and 1.013 bar	0.9942	-
Thermal conductivity at 0°C and 1.013 bar	14.65	mW/(m.K)
Heat capacity at constant pressure, 25°C and 1.013 bar	0.037	kJ/(mol.K)
Heat capacity at constant volume, 25°C and 1.013 bar	0.028	kJ/(mol.K)
Ratio of specific heats at 25°C and 1.013 bar	1.293759	-
Specific gravity at 21°C and 1.013 bar	1.521	-
Specific volume at 21°C and 1.013 bar	0.547	m³/kg

Table 3: Physical Properties of Gaseous CO_2

[7] (http://encyclopedia.airliquide.com/, 2011)

2.2. Thermophysical Properties of CO_2

Fluid mass flow rates for a given driving force by a pressure gradient are proportional to the ratio of density to viscosity ($m = \rho/\mu$) and the carried heat by the mass flow is proportional to the specific enthalpy of the fluid. Other important parameters for the mass flow and the heat transfer behavior are compressibility ($c = (1/\rho)\cdot(\partial\rho/\partial P)$) and thermal expansivity ($\varepsilon = -(1/\rho)\cdot(\partial\rho/\partial T)$).[8]

The ratio of density to viscosity, which is called mobility, depends very much on temperature and pressure conditions, but differs for CO_2 and water as you can see in the figures 3 and 4[8]. For water, this ratio is predominantly a function of temperature and only a weak function of pressure due to the dependence of viscosity and density on temperature. For CO_2, this ratio is a function of temperature and pressure, because viscosity and density depend very much on both, temperature and pressure.

The mobility is in general larger for CO_2 than for water. The region with the maximum values is emanating from the CO_2 saturation line and becomes smaller for liquid and gaseous carbon dioxide. For fluid injections, which are mostly conducted at temperatures lower 50°C, CO_2 has the larger mobility than water by factors of 4-10. Also for temperatures near 200°C and high pressures the mobility is higher for CO_2 by a factor about 2, but for temperatures lower 150°C water has the larger value.

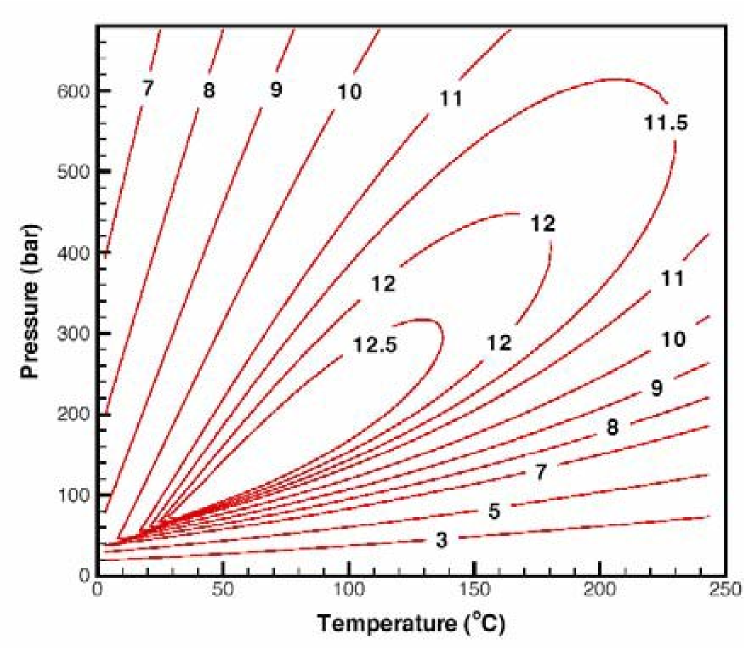


Figure 3: Mobility of CO_2 in Units of 10^6 s/m^2

[8] (Pruess & Azaroual, 2006)

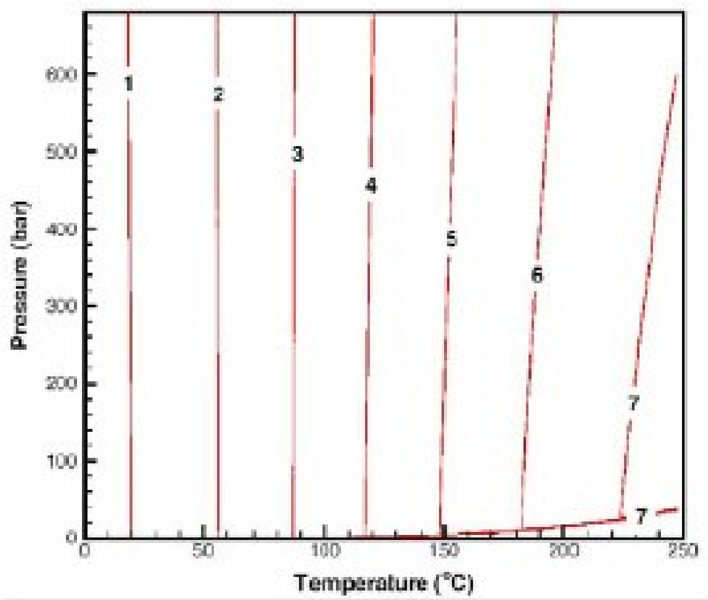


Figure 4: Mobility of Water in Units of 10^6 s/m^2

The specific enthalpies of water and CO_2 are shown in the figures 5 and 6 at a reference state of 20°C and 100 bar.[9] It is apparent that the specific enthalpy of liquid CO_2 depends rather small on pressure, however, this pressure dependence increases sharply at lower pressures and higher temperatures. In contrast, the specific enthalpy of water depends mainly on temperature and only slightly on pressure.

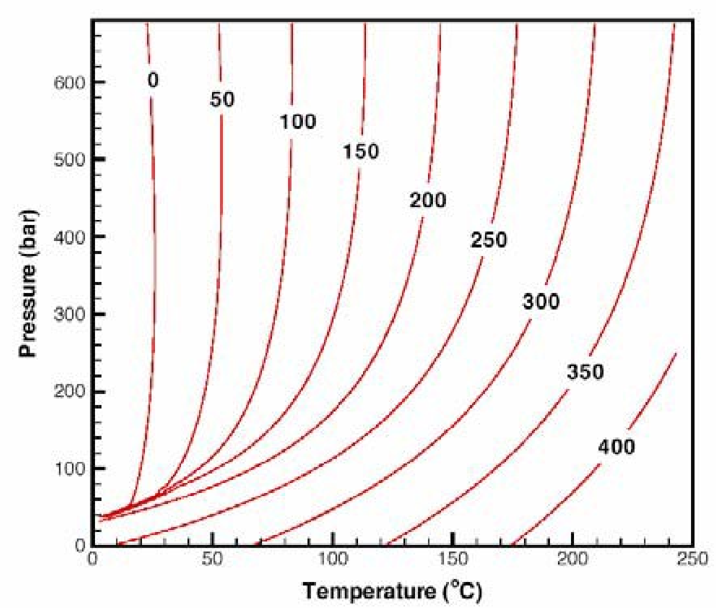


Figure 5: Specific Enthalpy of CO_2 in [kJ/kg] at 20°C

[9] (Pruess & Azaroual, 2006)

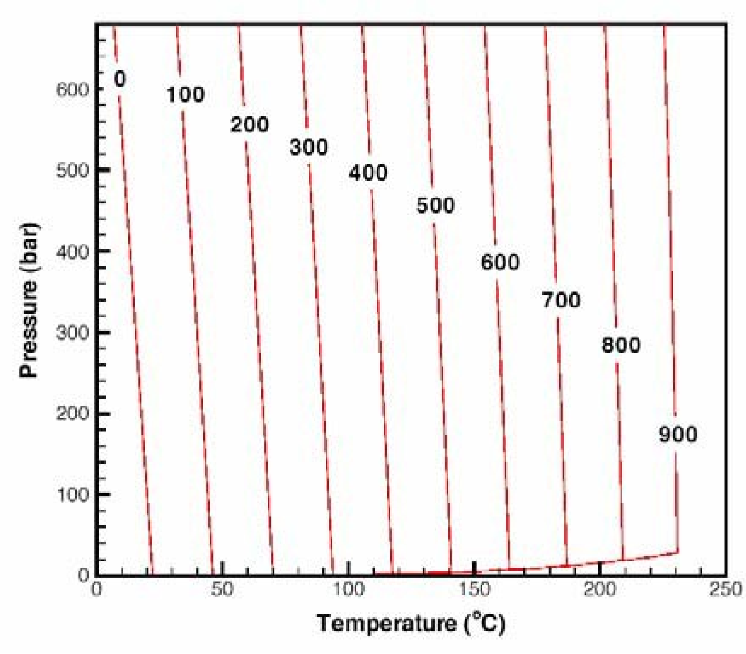


Figure 6: Specific Enthalpy of Water in [kJ/kg] at 20°C

At high pressures near 500 bar, the increase of specific enthalpy with temperature for CO_2 is less than half of the increase for water. This fact indicates that more than twice as much mass flow rate for CO_2 would be needed to achieve the same rate of heat transfer as in water.

For adiabatic decompression, which is used to avoid heat losses with thermal insulation, thermodynamic conditions will move along isenthalps, lines of constant specific enthalpy. Decompression of hot, high-pressure CO_2 will be accompanied by substantial temperature decrease, while for liquid water a small temperature increase would be expected. For compression of hot, modest-pressure CO_2 strong temperature increase will follow, while for compression lower 50°C temperature decline will arise.[10]

Advantages of CO_2 as heat transmission fluid are the larger heat extraction rates, which compensate the smaller specific heat and the smaller density, and the significant dependence on temperature and pressure. Simulations have indicated that the advantage of CO_2 over water as heat transmission fluid becomes larger for decreasing reservoir temperature as a result of the strong increase of water viscosity under these circumstances.[11]

[10] (Pruess & Azaroual, 2006)
[11] (Pruess K. , 2007)

2.3. Wellbore Flow of CO_2

The driving force of the wellbore flow is determined by the pressure gradient, which can by represented by a superposition of gravity, frictional and acceleration terms as you can see in equation 1 below.

$$\nabla p = (\nabla p)_{grav} + (\nabla p)_{fric} + (\nabla p)_{acc}$$

Equation 1: Pressure Gradient in a Flowing Well

In most applications, the contribution of the gravitational pressure gradient is the most influential and the other two are rather of secondary importance.[10]

In an injection well, the temperatures increase with depth, on the one hand because of the heat transfer from the surrounding rock structure and on the other hand, due to the fluid compression with increasing pressure. In a production well the opposite will happen and the temperatures will decrease due to the heat loss to the surrounding rock structure and the decompression. Since CO_2 is a highly compressible medium, the effect of increasing temperature with pressure has more influence in CO_2 than in water. The temperature effects from heat transfer with the surrounding rock structure are of a transient nature and will diminish over time. A significant buoyancy drive due to the striking difference in density between the hot rising fluid in the production well and the cold more dense fluid in the injection well is formed and is of great advantage. The density differences are much larger for CO_2 than for water.[12]

An adequate long-term flow behavior would be adiabatic, which ignores the heat transfer between the wellbore fluid and the surroundings. We use isenthalpic fluid flow to approximate such an adiabatic flow behavior. The isenthalpic flow approximation accounts for temperature changes that arise from compression or decompression of fluids, the so-called Joule-Thomson effect. Figures 7 and 8 show the temperature and pressure conditions for isenthalpic flow of CO_2 in an injection and a production well.[11]

Due to the charts we can see that for injection wells the difference between downhole and wellhead temperatures is larger for smaller wellhead pressures and increases with wellhead temperatures. At lower wellhead temperature, temperature changes versus depth are non-monotonic and decrease significant at greater depths, especially when wellhead temperature is low and/or wellhead pressure is large. These effects arise because of the dependence of the specific enthalpy of CO_2 on temperature and pressure.

[12] (Pruess & Azaroual, 2006)

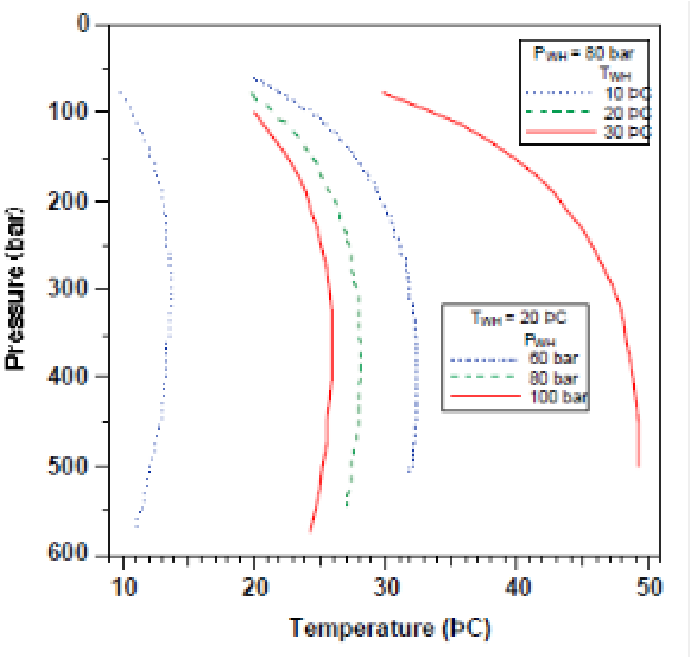


Figure 7: Temperature Pressure Conditions in an Injection Well for Isenthalpic Flow of CO_2

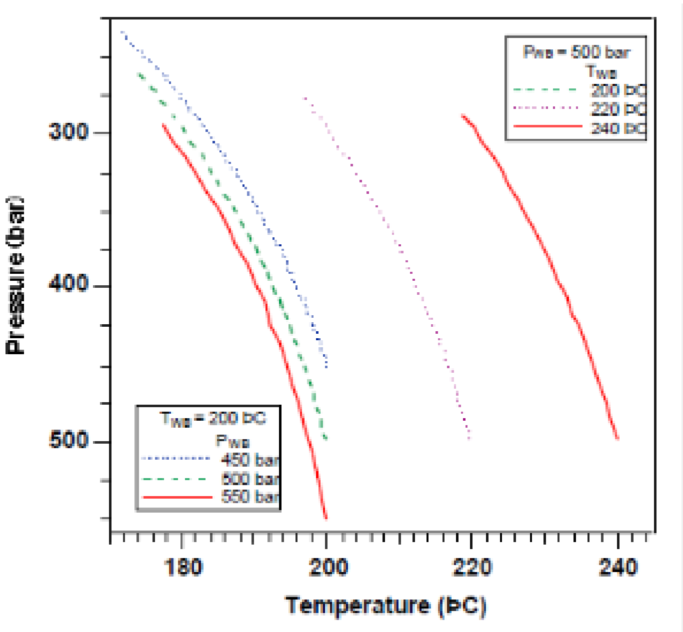


Figure 8: Temperature Pressure Conditions in a Production Well for Isenthalpic Flow of CO_2

With increasing wellhead temperatures downhole pressures will decrease, especially for adiabatic temperature conditions. This is because for adiabatic conditions wellbore temperatures are larger, and accordingly fluid densities are smaller.[13]

[13] (Pruess & Azaroual, 2006)

Increased temperatures with depth in an injection well would be favorable for the heat extraction, but unfavorable for the increase of pressure with depth. A reduction of the buoyant pressure drive, which is responsible for pumping CO_2 through the reservoir and reduces the power requirements for fluid circulation, will follow. In a production well as much as possible temperature increase during fluid upflow would be desirable and can be achieved by increasing downhole pressures and temperatures. However, increasing pressures lead unfortunately to increasing power consumption in the fluid circulation system. The temperature decline during the upflow in the well is a result of the decompression of CO_2. There must be a reasonable compromise between heat extraction and power consumption in the fluid circulation system found.

The case of laminar flow regime in reservoirs would more than double the production flow rate compared to water-based systems with the same reservoir flow impedance and injection pressure. A larger thermal power potential exceeding that of water-based systems would be provided.[14]

2.4. Chemical reaction of CO_2 with the surrounding rock structure

Even though the reactions between minerals and CO_2 should be quite rapid at elevated temperatures, supercritical CO_2 is much less solvent than water. If we consider the chemical reactions with in situ rock minerals, dissolution and subsequent precipitation of minerals would be reduced. Furthermore problems of scaling and formation plugging would be avoided.[15]

Water-based geothermal systems have a geochemically determined temperature limit, which occurs with approaching the critical point of water. At or above the critical point, the dissolution of silica and the following retrograde precipitation represent a massive obstacle for the operation of a water-based geothermal system at temperatures higher than the critical temperature of water. Supercritical CO_2 would avoid the mentioned silica effect, because it possesses a very low dissolving power for minerals deep in the crystalline basement. For this reason, supercritical CO_2 would allow to operate at temperatures near 400°C or even higher and increase thereby the thermodynamic efficiency.[14]

The poor dissolving power of supercritical CO_2 provides other advantages over water. The dissolved minerals in water, including especially silica and carbonates prevent the direct use of the working fluid in expansion turbines due to the precipitation of minerals. In addition, other trace elements in the solution, such as arsenic, fluoride, boron and hydrogen sulfide, could

[14] (Brown, 2000)
[15] (Pruess & Azaroual, 2006)

possibly cause environmental problems if they leak. The small amounts of minerals dissolved in supercritical CO_2 on the other hand would not represent a major problem.[16]

Supercritical CO_2 would slowly diffuse outward the pressurized reservoir, where the already existing water-filled network of interconnected microcracks would be filled with $SCCO_2$. An ancillary benefit of the dissolved minerals in the supercritical CO_2 would be the precipitates, which would tend to plug off the microcrack porosity and seal the reservoir boundaries by leaving a small amount of precipitates in the microcrack pore structure.[15] Although prolonged exposure of certain minerals like wairakit to CO_2 can lead to an increase in porosity and permeability, and possibly even to a growth of the reservoir by producing carbonate. This increase occurs because of dehydration reactions, which remove the loosely bound water and reduce the molar volume of the involved minerals.[16]

In addition, the content of swellable clay minerals is important for estimating the reaction with $SCCO_2$, because a secondary swelling of these clay minerals can lead to a reduction in the fluid effectiveness.[17]

As already mentioned at the beginning of this chapter, three different zones with different compositions of CO_2 and water are formed while injecting CO_2 over a period of time in the reservoir. Chemical interactions between the working fluid and the surrounding rock structure are expected to be different in these three different zones. It can cause problems due to the corrosiveness of aqueous solutions of CO_2, which can dissolve rock minerals, as well as attacking steel liners and casings used in the well construction. The intention would be to create a CO_2-stream as dry as possible to avoid such corrosion problems.[16]

[16] (Pruess & Azaroual, 2006)
[17] (Merkel, 2005)

3. Possible Procedures for Electrical Power Generation

To convert geothermal energy into electrical energy, various processes and systems are available. All those include a turbine, in which the working fluid expands and drives a generator via a shaft. The extracted fluid from the bore hole is used either directly as a working medium (single loop cycles) or transfers its heat through a heat exchanger to a secondary fluid (binary cycles).

Reservoir temperature, vapor content, pressure, amount of non-condensable gases and mineralization determine what type of power generation we should use. Plants for direct use of the reservoir fluid can be used with or without previous flash at reservoir temperatures of about 150°C, possible low level of non-condensable gases and low mineralization. In contrast, binary systems are used at temperatures of 80°C, high level of non-condensable gases and high mineralization, also higher temperatures are possible and would significantly improve the efficiency.[18]

By choosing a suitable working medium, the respective temperature levels are exploited best. Of great interest is supercritical CO_2 as cycle fluid, which assumes a binary plant or a single loop system. The supercritical CO_2 can be run through the reservoir first and then captured in a sequestration site, this would have the advantage of removing the condenser step and so the parasitic losses.[19]

To become a competitive electrical power generation we need to improve the efficiency of converting geothermal heat into mechanical torque on the generator shaft. Efficiency is especially determined by the brine inlet temperature, the brine rejection temperature and the energy conversion cycle. The thermal efficiency is defined as the produced mechanical power divided by the rate of heat input into the working fluid and is limited by the efficiency of the Carnot cycle, which represents ideal power conversion conditions. The quality of the source determines the hot fluid temperature T_H, while the cold temperature T_C is set by the ambient temperature and the temperature of the cooling fluid.

$$\eta_C = 1 - \frac{T_C}{T_H}$$

Equation 2: Thermal efficiency of the Carnot cycle

Compared to steam power plants, only a small fraction usually about 40% of the theoretical limit is exploited. This shows us that there is a lot of room to improve the electrical power generation from reservoirs, therefore new cycle options, cycle fluids and turbines should be optimized.[19]

[18] (GeoForschungsZentrum Potsdam, 2001)
[19] (Gurgenci, Rudolph, Saha, & Lu, 2008)

3.1. Single Loop Cycles

A single loop system is a geothermal power plant cycle, where the extracted fluid from the borehole is used as both, heat exchange and power cycle fluid. Usually the brine is re-injected into the reservoir by an injection well, or it is flashed again at a lower pressure. Those systems are used when brine is present at sufficient temperature and pressure in sufficient quantity, for that reason they are used especially at higher well enthalpies and higher well temperatures than binary systems.

The more the fluid expands in the turbine the higher would be the delivered power to the turbine, this requires low temperatures and pressures behind the turbine. These conditions behind the turbine are determined by the prevailing conditions in the condenser. The condenser temperature is limited downwards by the coolant temperature and the coolant quantity, while the condenser pressure is determined by the amount of non-condensable gases. The higher the amount of non-condensable gases, the more effort is required to remove those gases from the condenser and thus reduce the pressure.

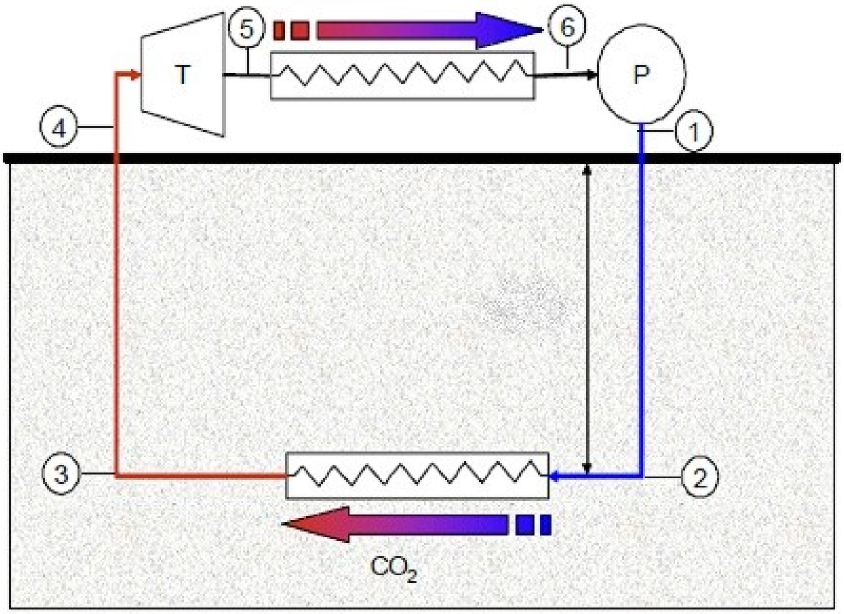


Figure 9: Single Loop System with CO_2

As you can see in figure 9 [20] at point 1 the working fluid is pressed by a pump through an injection well into the reservoir. As it flows through the reservoir from point 2-3 to the production well, it is heated and then rises in the production well. After separation of the entrained solids and the droplets, the gas mixture reaches the turbine at point 4, where it is relaxed far below the ambient

[20] (Gurgenci, Rudolph, Saha, & Lu, 2008)

pressure. For that reason the turbine operates a generator and power is produced. Behind the turbine at point 5 a condenser is placed, from which the non-condensable gases are removed through a so-called ejector. Afterwards our working fluid reaches the pump again and is anew pressed into the reservoir through the injection well. The cooling of the condenser can be performed either by a coolant cooling unit or a dry cooling.

3.1.1. Single Flash System

In reservoirs that do not provide sufficient gaseous brine through the production well, so-called Flash-systems can be used. The fluid is therefore in a first flash tank partially expanded, whereby a part of the brine is evaporated. Liquid and gaseous phases are separated in the separator, the liquid phase is fed directly to the injection well and the gaseous phase to the turbine. In figure 10[21] the schematic of such a single flash cycle is shown.

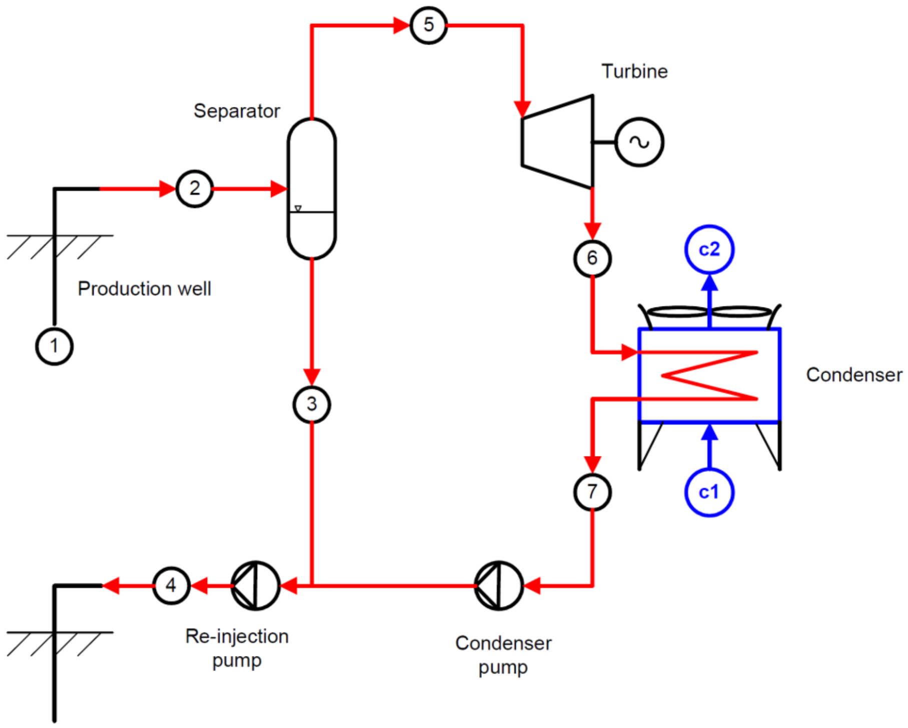


Figure 10: Single Flash Cycle Schematic

The geothermal fluid enters with source inlet temperature at station 1. Afterwards when it enters the separator at station 2, it already has started to boil due to the well pressure loss. The separated brine from the separator at station 3 is re-injected into the injection well with geothermal fluid return conditions at station 4. The gas from station 5 enters the turbine, through

[21] (Valdimarsson, 2011)

which it is going to be expanded down until station 6, where the temperature and pressure conditions are set through the condenser. After the condenser the again liquid geothermal fluid is re-injected into the reservoir at station 4.[22]

The condenser shown in this schematic is air cooled, where the cooling air enters the condenser at station c1 and leaves at c2. The condenser hot well is positioned at station 7.[22]

3.1.2. Double Flash System

Figure 11[22] shows the schematic of a double flash cycle.

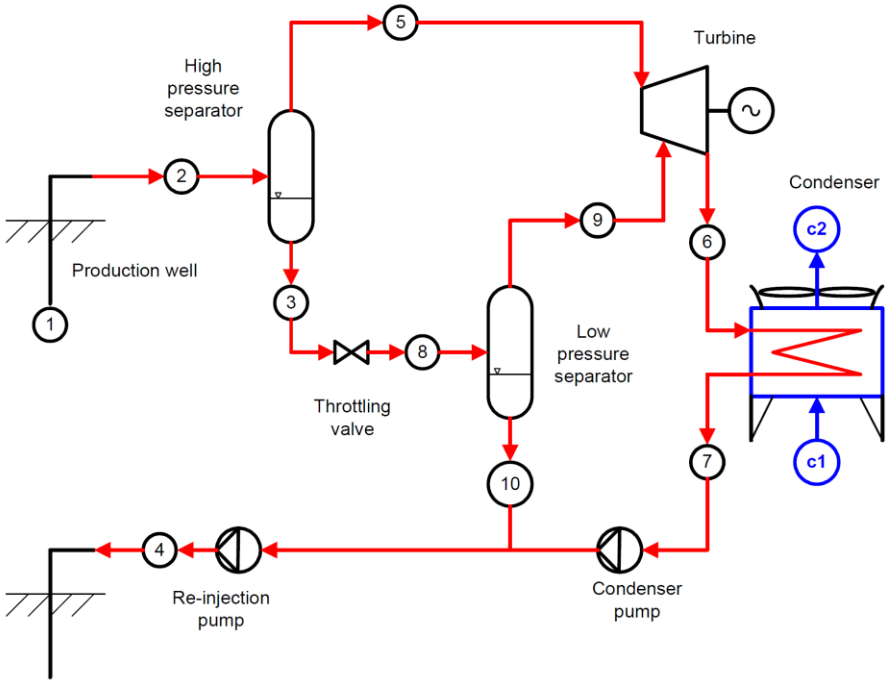


Figure 11: Double Flash Cycle Schematic

The difference to a single flash system is that the separated brine from the separator in station 3 is not directly re-injected into the injection well. Instead the brine is throttled down to a lower pressure level at station 8 and then led again to a low pressure separator. The brine from the low pressure separator at station 10 is re-injected into the injection well at geothermal fluid return conditions at station 4, while the gas from the low pressure separator at station 9 is led to the turbine. The gas from the low pressure separator enters the turbine a few stages later than the gas from the high pressure separator. The pressure difference over the first stages in the turbine is the same as the pressure difference between the high and the low pressure separators. The

[22] (Valdimarsson, 2011)

mass flow in the lower pressure stages of the turbine is then higher than in the high pressure stages. The remaining cycle is similar to the single flash cycle.[23]

3.2. Binary Cycles

Binary cycles have advantages over single loop cycles at low moderate temperature heat sources. They are used, if it is not possible to produce vapor from the fluid due to pressure and temperature, if there are too aggressive thermal fluids, if there is still too much enthalpy available after a single loop system or if there are excessive quantities of non-condensable gases.

In a binary system, the geothermal fluid is passed through a heat exchanger, where its heat is transferred into a second liquid. This liquid should have a lower boiling point than the geothermal fluid. After being heated the binary liquid flashes to vapor and expands across the turbine to drive the generator. The vapor is then re-condensed to liquid status and is reused repeatedly in a closed loop cycle without emissions to the air or to the environment. Such a scheme of a binary cycle is shown in figure 12. [24]

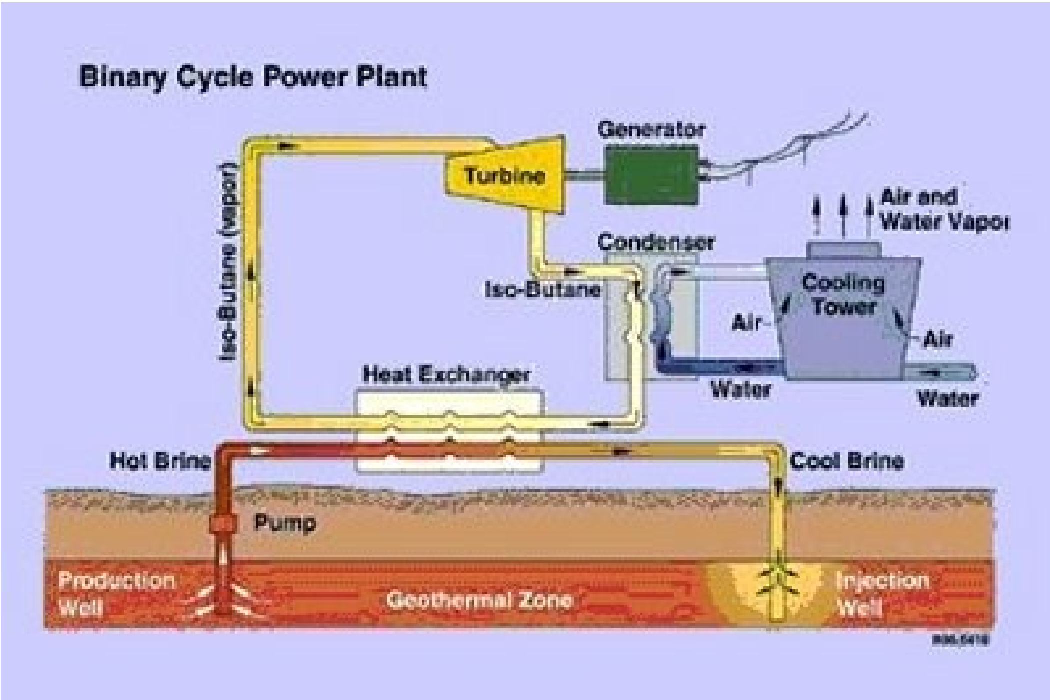


Figure 12: Scheme of a Binary Cycle

[23] (Valdimarsson, 2011)
[24] (http://www.daviddarling.info/encyclopedia, 2012)

3.2.1. Organic Rankine Cycle

Organic Rankine Cycles can produce electricity over a wide range of temperatures normally from about 110 - 180°C and offer many advantages due to the simplicity of the turbine, the control system and the balance of plant. Because of the fluid density differences the turbine and piping sizes are smaller and thus less costly. Comparing to the Kalina cycle those sizes are bigger due to the higher volume flow and also the turbine cost is thus higher. The condensing pressure in an Organic Rankine Cycle is in general above atmospheric pressure, which eliminates the need for complex vacuum and gas purging equipment.[25]

Thermal energy in the waste heat stream is transferred to the vaporizer to vaporize the ORC working fluid by a nonflammable heat transfer fluid flowing through the heat recovery unit. The ORC working fluid should be selected to optimize the power output. The resulting vapor after the vaporizer drives the turbine, which is coupled to a generator, or an additional compressor. When organic vapor expands in the turbine it becomes superheated or dryer, unlike steam which becomes wetter during expansion. For that reason preheating of the vapor is not required. After the turbine the exhausted vapor flows through the recuperator for being condensed and recycled by the ORC working fluid. Due to the low freezing temperature of organic fluids, there is no freezing in the condenser, even at extremely low ambient temperatures.[25]

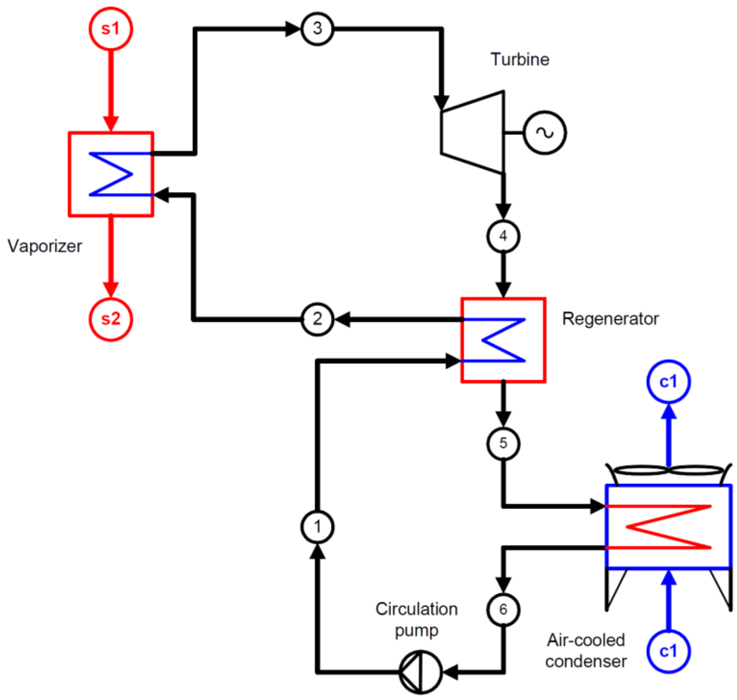


Figure 13: Scheme of an Organic Rankine Cylce with Regeneration

[25] (Bronicki)

In figure 13 the flow diagram of an Organic Rankine Cycle with regeneration is presented. At station s1 the geothermal fluid enters the well at source inlet temperature. If the pressure is kept high enough, a gas extraction system is not necessary, because no non-condensable gases would be separated from the liquid. The vapor enters the turbine after leaving the vaporizer at station 3. At station 4 the vapor from the turbine enters the regenerator, where the superheat in the steam can be used to pre-heat the condensed fluid before entering the vaporizer. After the regenerator the cooled vapor enters the condenser at station 5 to be condensed down to a saturated liquid, which leaves the condenser at station 6. The condenser shown in this figure is air cooled, with the cooling air entering at c1 and the heated air leaving at c2. For entering the regenerator, a circulation pump raises the pressure after the condenser to a high pressure level in station 1. After the regenerator the pre-heated fluid enters the vaporizer at station 2. The fluid is heated to each saturation in the vaporizer. Here it is cooled down and then sent to re-injection at station s2.[26]

Regeneration increases the efficiency by recovering a part of the rejected heat. The temperature of the working fluid is raised at the vaporizer entry, and leads thus to higher geothermal fluid temperatures at the vaporizer exit. Organic Rankine Cycles with regeneration improve the lower temperature limit on the geothermal fluid temperature, which is imposed by chemistry or requirements of a secondary process.[26]

The used working media is determined by the temperature of the available heat source and should be non-toxic, have no climate-damaging effect and should evaporate at lower temperatures compared to the geothermal fluid. Through the use of organic working fluids different technical requirements occur. Due to the often aggressive working fluids, the surfaces of the turbines and heat exchangers have to be protected against corrosion. In addition, the seal of the circuits is more complicated and the turbines are usually special versions because of the different properties of organic working fluids.[27]

In an ideal Organic Rankine Cycle, the expansion would be isentropic, while the evaporation and the condensation would be isobaric. In the real Organic Rankine Cycle on the other hand, the efficiency is lowered by the presence of irreversibilities, which occur mainly during the expansion or in the heat exchanger. During the expansion only a part of the recoverable energy from the pressure difference is transformed into useful work, while the rest is converted into lost heat. The efficiency of the turbine is defined by the comparison with an isentropic expansion. In the heat exchanger the working fluid takes a long sinuous path, which causes pressure drops that lower

[26] (Valdimarsson, 2011)
[27] (Koehler & Saadat)

the recoverable energy. Furthermore the temperature difference between the heat source and the working fluid generates exergy destruction and reduces therefor the cycle performance.

3.2.2. Kalina Cycle

The Kalina power generation cycle is a modified Clausius-Rankine cycle with the difference that an ammonia-water mixture is used as working fluid. The main advantages of this mixture are the variable temperatures for vaporization and condensation. It mains that there are no simple boiling or condensation points, instead exist a boiling and condensation temperature range. Those ranges are caused by the fact, that the phase change process is a combined process of the phase change of the substances and the absorption of ammonia from water.[28]

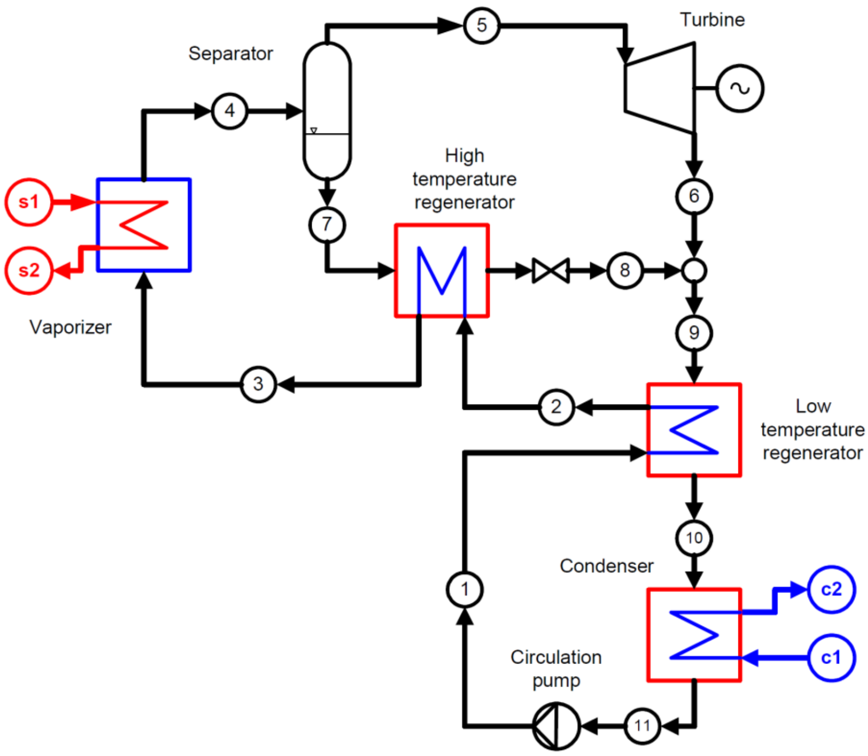


Figure 14: Scheme of a Kalina Cycle

As you can see in figure 14 above, the Kalina Cycle works almost similar to the Organic Rankine Cycle. Differences occur for example in the vaporizer, where the ammonia-water mixture is notentirely boiled and a liquid-vapor mixture leaves the vaporizer at station 4 to enter the separator. This is done to maximize the vapor temperature at the vaporizer outlet. After the separator the liquid leaves the separator and flows into the high temperature regenerator at station 7. Afterwards it’s throttled down to the condenser pressure in station 8 and mixed with

[28] (Valdimarsson, 2011)

the turbine exit vapor from station 6. The separated vapor passes the same path like in the ORC. Another difference exists in the condenser, where an absorption process is going on. That means that the ammonia rich vapor is absorbed into the leaner liquid and the temperature is lowered. The rate of absorption is determined by the kinetics of the absorption, whereas the condensation process is controlled by the heat transfer and the heat capacity. Finally all the mixture is in saturated liquid phase in the condenser.[29]

The benefit of the Kalina cycle lies in the more favorable heat transfer conditions during evaporation and condensation. In contrast to the ORC those are not isothermal, rather the temperature changes due to changes in concentration at a constant total concentration and a constant pressure. Due to the non-isothermal evaporation and condensation, the temperature differences in the heat transfer are lowered and thus the heat losses during the heat transfer. Furthermore, the mean temperature of the heat transfer will raise and the one of the condensation will be lowered. Both effects result in an improvement of the Carnot efficiency, the desired process efficiency.[30]

The Kalina cycle is superior to the ORC if the geothermal fluid is only water and the temperatures get lower than 150-160°C. Another technical difference between these binary cycles is the pressure level, which is higher for the Kalina cycle. It does not seem that there are any major differences in requirements of the piping and equipment material except the turbine, where corrosion has to be avoided in the Kalina cycle.[29]

3.2.3. Brayton Cycle

The principle of a Brayton cycle shown in figure 15[31], is to vaporize the geothermal fluid before compressing the gas via a compressor linked to a turbine. Heat is added to increase the gas enthalpy. Afterwards at station 4 the hot, high-pressure gas is sent through a turbine, which is linked to a generator to create electricity. The warm fluid leaving the turbine is sent through a heat exchanger at station 5 to preheat the fluid just entering the combustion chamber at station 3. The combustion products could be re-injected to the reservoir.

In an ideal Brayton cycle the compression would be isentropic process, where the gas is drawn into the compressor to be pressurized. The heat addition and the heat rejection on the other hand would be isobaric due to the constant pressure in the combustion chamber, where the compressed gas runs through. In the combustion chamber fuel is burned to heat the gas and since the chamber is open to flow in and out constant pressure would be reached. Also the expansion

[29] (Valdimarsson, 2011)
[30] (Koehler & Saadat)
[31] (Gurgenci, 2010)

would be isentropic, where the heated and pressurized gas gives up its energy while expanding in the turbine. Though in a real Brayton cycle expansion and compression are adiabatic processes, which mean that there is no heat exchange with the environment.[32]

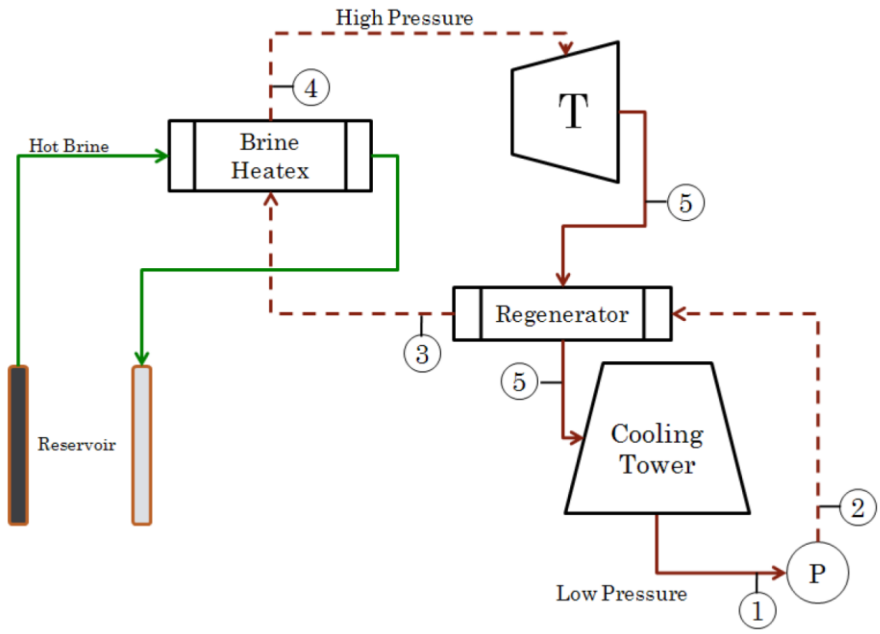


Figure 15: Scheme of a Brayton Cycle

To increase the overall power output of a Brayton cycle system the most direct way is to increase the compression ratio. The efficiency of an ideal Brayton cycle is defined in equation 3 below, in which ϒ represents the heat capacity ratio.

$$\eta_C = 1 - \frac{T_1}{T_2} = 1 - \left(\frac{P_1}{P_2}\right)^{\frac{(\gamma-1)}{\gamma}}$$

Equation 3: Efficiency of an ideal Brayton Cycle

There are different possibilities to increase the power of a Brayton cycle. One of them is called Reheat, where the working fluid expands through a series of turbines and is then passed through a second combustion chamber. This increases the power output for a given compression ratio without exceeding any metallurgical constraints, but is often associated with an efficiency reduction. The other method is named Overspray, where water is injected into the compressor after a first compressor stage. This leads to a higher mass-flow inside the compressor, a higher turbine output power and a reduction of the compressor outlet temperature. Afterwards in a

[32] (www.wikipedia.com, 2012)

second compressor stage the water is completely vaporized and offers some intercooling due to its latent heat of vaporization.[33]

Furthermore there exist methods to improve the efficiency of a Brayton cycle. As already mentioned increasing the pressure ratio would increase the efficiency, but there are practical limits. First of all increasing the pressure ratio will increase the compressor discharge temperature and may further cause the temperature of the gasses leaving the combustor to exceed the metallurgical limits of the turbine. Second reason is that the gap between the blades and the engine casing increases in size as a percentage of the compressor blade height. As the blades get smaller in diameter at higher pressure stages, a greater percentage of the compressed air is able to leak back past the blades and causes therefore a drop in the compressor efficiency. The other method is called Regeneration, where the warm fluid leaving the turbine is sent through a heat exchanger to preheat the fluid just entering the combustion chamber. This results directly in less fuel consumption, higher efficiency and less power loss as waste heat for the same operating conditions. Regeneration is only an option when the pressure ratio is sufficiently low to ensure that the exhaust temperature is higher than the compressor discharge temperature.[33]

By using the Brayton cycle, there is some flexibility in selecting the operating pressure independently of the cooling medium temperature. However the efficiency is highest when the heat reduction occurs almost at the critical pressure. Also the compressor work would be highest at this point, but is compensated by the higher turbine work.[33]

Benefit is the simplicity of the Brayton cycle decreases the likelihood of mechanical failure of the implemented machines. Moreover, the compactness of those system guarantees less capital costs and less required space due to the smaller turbines and the missing condenser.

Supercritical fluids would increase efficiency of Brayton cycles by performing compression around the critical point due to the fact that they have properties of liquids and gases. Furthermore the higher fluid density of the supercritical fluids will reduce the required turbine size. On the other hand supercritical fluids as working fluids would require a much greater heat exchange area, because of the higher flow rate needed in supercritical cycles. The normal operating temperature of such a supercritical Brayton cycle is about 650 K, which equals 377°C.[34]

3.3. Combined Cycles

The previously treated cycles are frequently used in combination. Usually a binary cycle is then used as the bottoming cycle for a flash cycle. The binary cycle serves to increase the total power

[33] (Gurgenci, Rudolph, Saha, & Lu, 2008)
[34] (Rousseau, 2007)

plant efficiency, while the flash cycle has the benefit of low investment. As a result the total efficiency is raised at the cost of complexity.

4. Chosen Procedure for Electrical Power Generation

Our elected reservoir possesses the following data and features:

- Reservoir temperature: ~ 150 °C
- Reservoir pressure: ~ 300 bar
- Temperature of reservoir media at the top of the production well: ~ 120 °C
- Pressure of reservoir media at the top of the production well: ~ 150 bar
- Amount of non-condensable gases: low
- Mineralization: low
- Well enthalpy: low

Due to these facts, we can exclude some of the possible cycles of energy generation and select the most appropriate one for our scenario.

At the simplest, we can rule out the binary Brayton cycle. In order to operate a useful Brayton cycle, the reservoir temperature should be above 377 °C.[35] This requirement is not fulfilled by our reservoir because the reservoir temperature is around 150 °C.

Next, we consider the binary Kalina cycle. For this cycle our reservoir would indeed have the right reservoir temperature, but the reservoir pressure may be too low. In addition the most appropriate geothermal fluid for the Kalina cycle would be water, while there is a gas mixture in our reservoir. [36]

Regarding the Organic Rankine cycle, the reservoir temperature would as well be sufficient, because the Organic Rankine cycle works above 80 °C. Higher reservoir temperatures would simply help to increase the cycle efficiency. Although Organic Rankine cycles are preferred used at higher amounts of non-condensable gases, higher mineralization and if it’s not possible to produce gas from the fluid due to pressure and temperature differences. [37]

Single Loop cycles are operated at temperatures above 150 °C and relatively high pressure levels. Our reservoir meets with 150 °C, reservoir temperature, and 300 bar, reservoir pressure, both of these requirements and furthermore provides the advantage of low mineralization and low amount of non-condensable gases.[37]

For these reasons we decided to try a Single Loop cycle, even though the Organic Rankine cycle would also be an option. Only a closer look at the potential electrical power generating cycles combined with our scenario will show the usefulness of the selected cycle.

[35] (Rousseau, 2007)
[36] (Valdimarsson, 2011)
[37] (GeoForschungsZentrum Potsdam, 2001)

4.1. First Idea of our Process Flow

The following figure 16 shows the schematic process flow of our energy cycle, which we imagine for our specific problem:

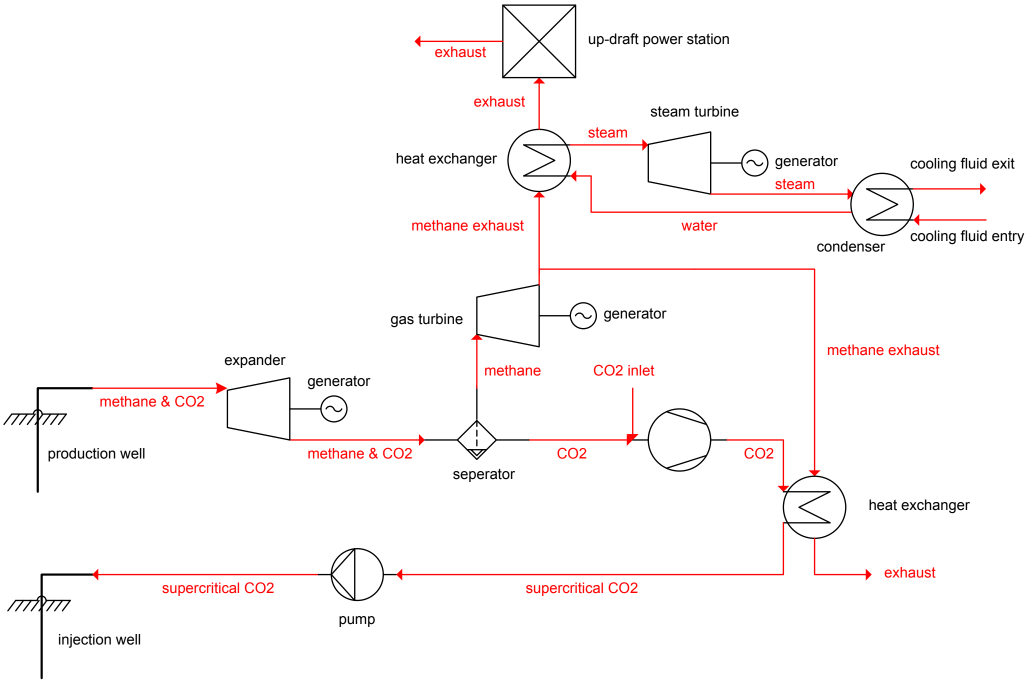


Figure 16: First Scheme of our Process Flow for our specific Problem

There is a high volume flow of the gas mixture consisting of CO_2 and methane, which we produce from the production well, available.

This high flow rate might be useful to relax the gas mixture in a turbo-expander and thereby to generate electricity via a shaft, which drives a generator. Since drastic temperature drops occur with the relaxation of gases and our exit temperature at the production well is only about 120 ° C, it is very important to ensure that the temperature of the gas mixture does not drop too much. Solidification would lead to big problems.

After depressurization, we split the methane from CO_2 to achieve a greater benefit. The separated methane is burned in a gas turbine to operate a further generator to produce electricity. Manufacturers offer individual gas turbines, some even entire combined gas and steam plants. In a gas and steam plan, the gas turbine exhaust is used to evaporate water in a heat exchanger. The steam again is used to drive a steam turbine with a generator for generating more electricity. After the turbine, the steam is liquefied by a condenser and the evaporation cycle starts again. The ORC process represents an alternative to the steam cycle and differs essentially by the used working fluid. More detailed considerations will show which cycle is advantageous in our case.

If there is enough heat left after the heat exchanger for the steam/ORC process, there is the question whether it would be useful to install an up-draft power plant.

The CO_2 should be injected into the injection well after the separator in order to utilize it again. Since we want to inject supercritical CO_2, we must perform a pressure and temperature increase on the previously relaxed CO_2. The pressure increase is provided by a compressor and the temperature increase by a heat exchanger, which can also be operated with the waste heat of the gas turbine.

If the pressure is not sufficient enough to press the supercritical CO_2 back into the reservoir, a pump is required.

5. Required Machinery Components for the selected Energy Cycle

A brief explanation of the machinery components for our power generation cycle in order to be able to understand how they work and for what they are needed.

5.1. Turboexpander

The figure 17 [38] below shows a picture of a turboexpander.

Figure 17: Picture of a Turboexpander

Expanders belong to turbomachines and are in the narrow sense turbines where pressurized gases are expanded to produce work to drive compressors, pumps or generators. So the energy of the gas is used.

They consist of the actual turbine and have neighter a compressor, nor a combustion chamber, which means that the expanding gas is not generated by the machine itself, but exists already.

An expander consists of a single-stage or multistage drive shaft of axial (bladed stages) or radial (impellers) design, guide stages (vanes or diffuser) and an outer housing.

The expansion is assumed to be approximately isentropic, so the low pressure exhaust gas from the turbine reaches a very low temperature level, depending on the operating pressure and the gas properties. Partial liquefaction of the expanded gas is not uncommon.

Turboexpanders operate in a range of about 750 W up to 7.5 MW. Their efficiency is the higher, the higher inlet temperature of the gas and pressure ratio of the turbine become.[39]

Because of the hot gas parameters, such as temperature and gas composition, high demands are placed on the materials of the expander components. In addition to high strength and temperature resistance, a high resistance to oxidation and corrosion is required. For this reason,

[38] (www.atlascopco.com, 2013)
[39] (www.wikipedia.com, 2012)

the most common used materials are high-alloy ferritic and austenitic steels and nickel-based alloys.

Expanders can be classified by their loading device or bearings. The main loading devices of turboexpanders are centrifugal compressors, electrical generators or hydraulic brakes. With centrifugal compressors and electrical generators the shaft power is recovered either to recompress the process gas or to generate electrical energy. Hydraulic brakes are used when the expander is very small and harvesting the shaft power is not economically justifiable. Bearings used are either oil bearings or magnetic bearings.[40]

5.2. Gas Separation

Gas separation is the technology to separate different gases combined in a gas mixture from each other.

For this purpose, various methods can be used. These include, for example, the Linde process, where the gas mixture is pressurized, cooled and relaxed until the individual gases liquefy sorted according to their boiling point. Also gas centrifuges, which use the centripetal force due to their mass differences, can be used for gas separation. Their biggest problem is however the low throughput.[40]

The most interesting method is the gas separation with membranes. Here the separation principle is based on the different permeability of the gases through the used membrane materials. The driving force for this process is the partial pressure difference across the membrane. Figure 14 shows the basic structure of a membrane gas unit.[41]

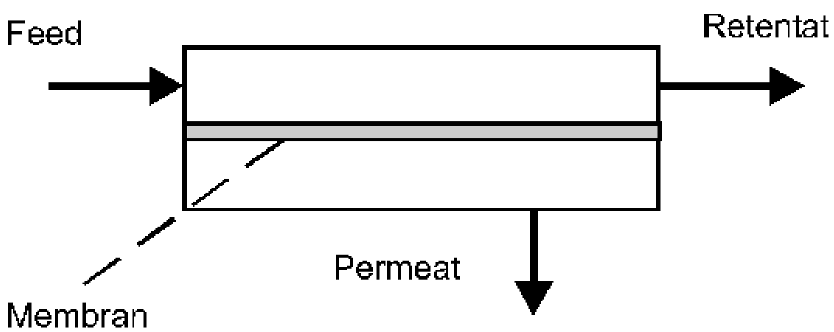


Figure 18: Basic Structure of a Membrane Gas Unit

A gas mixture, the so-called feed, is fed to the separating device and the material flow is separated there. The stream passing through the membrane is called permeate, while the part of

[40] (www.wikipedia.com, 2012)
[41] (www.pca-gmbh.com, 2013)

the stream leaving the separation unit depleted is called retentate. Besides, different flow guides are available for gas management.[42]

If due to the non-ideal selectivity of the membranes the desired separating effect is not achieved in one stage, a plurality of membrane modules are connected in series. In the simplest case, the permeate of the first stage serves as feed for the second stage, while the retentate of the second stage is mixed with the feed of the first stage.

Particular advantages of membrane processes are the relative immunity to impurities, the modular design for easy expansion, the relative lightness and compactness, and the fact that they work without moving parts. In addition, they have almost no minimum size and can also be run with variable flow rates.[42]

5.3. Gas Turbine

Figure 19 [43] below illustrates the principle of a gas turbine.

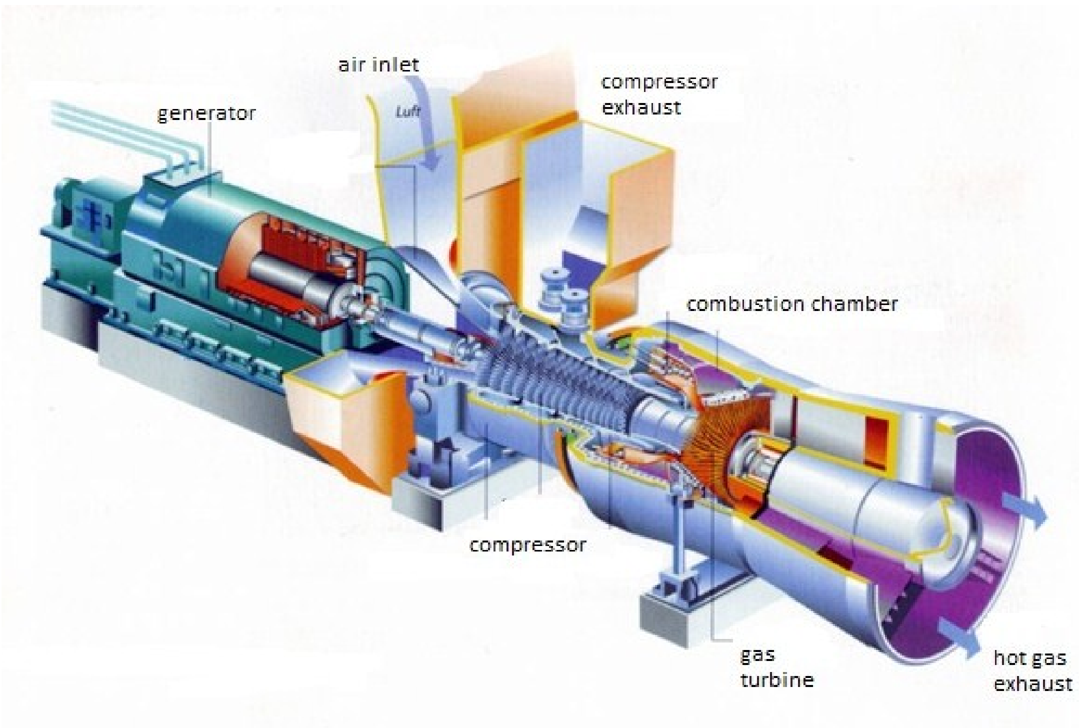


Figure 19: Principle of a Gas Turbine

Gas turbines belong to the internal combustion engines and are based on the Joule process. Their efficiency is the higher, the higher the turbine inlet temperature of the gases and the pressure ratio of the turbine are.[44]

[42] (www.pca-gmbh.com, 2013)
[43] (http://www.hagelstein-consult.de/technischedokumentation/turbinenundabhitze/index.html, 2013)
[44] (www.wikipedia.com, 2012)

They consist of an inlet, a compressor, a combustion chamber, an expander, a diffuser and an output shaft. The compressor compresses with blades of one or more compressor stages air, is then mixed, ignited and burned in the combustion chamber with a gaseous or liquid fuel. This creates a hot gas mixture consisting of the combustion gases and the air, which is expanded in the subsequent turbine part.

This converts thermal energy into mechanical energy and thus drives both, the compressor and a generator, rotor, compressor or pump.

Except the inlet and the diffuser, all components are coupled by one or several shafts. One-, two- or three-shaft machines, are distinguished here. In the single-shaft gas turbine all compressor stages and all turbine stages are placed in one row on the same shaft. That means, the entire machine runs at the same speed. The output can either be on the compressor side or at the turbine end of the shaft. For the operation of generators, it is mostly at the compressor side of the shaft, because it makes it possible to install a better exhaust diffuser. [45]

In the two-shaft arrangement, the turbine part can be differed in gas generator and power turbine. Thereby, the first turbine stages drive the first compressor stages and form together the gas generator unit. In the same housing, right after, the power turbine runs at a speed independent of the gas generator. The output is usually on the turbine side, since mainly pumps or compressors are driven.[45]

The inlet serves as fluidic adaption between the inlet environment and the airflow in the compressors. Goal is an air duct with no turbulence and the prevention of flow separation. In particular, at high speeds the inlet has an important function, because the incoming air mass is already there pre-compressed and slowed down.[45]

After the air inlet the compressor complex follows, which may consist of axial or radial compressors. Axial compressors usually consist of multiple impellers with blades in an axial assembly, which are typically divided into low-pressure and high-pressure compressor stages. The air mass flowing receives through it pressure energy by supplied kinetic energy in the diffuser-shaped spaces between the compressor blade. According to the law of Bernoulli the increasing cross-sectional area of the duct decreases the flow rate. A complete compressor stage of an axial compressor consists of a rotor, where pressure and temperature as well as the speed increase, and a stator, in which the pressure increases at the expense of speed. The rotor stages are connected to a common drum and the stator stages are built into the inside of the compressor housing.[45]

The high compression of the air causes a sharp rise in temperature. The thus heated air then flows into the combustion chamber where fuel is supplied to it. The fuel-air mixture is then ignited.

[45] (www.wikipedia.com, 2012)

Subsequently, the combustion takes place continuously. By the exothermic reaction, the temperature rises again and the gas expands.[46]

Then escaping gases of the combustion chamber meet a turbine and this drives via a shaft the generator.

5.4. Steam Power Cycle

Figure 20 [46] illustrates a steam power cycle for power generation.

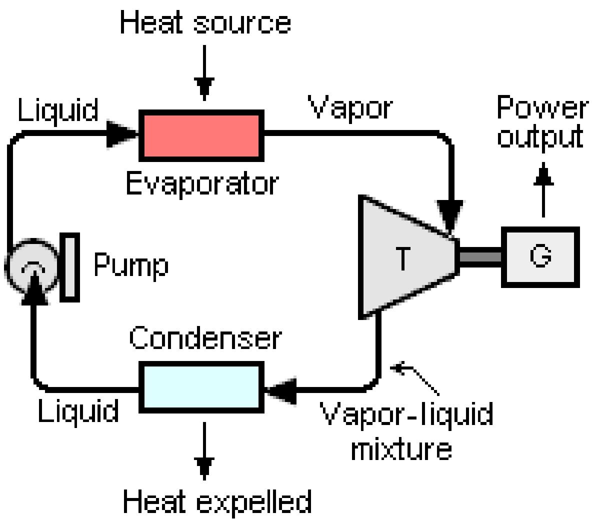


Figure 20: Steam Power Cycle for Power Generation

A steam power cycle is an electric power generation system that uses a heat source, a cooling medium (air, water or other), a circulating working fluid and a turboexpander. The system can accommodate a wide variety of heat sources, in our case the exhaust of a gas turbine.[46]

The circulating working fluid is pumped to a high pressure and then vaporized in the heat exchanger with the available heat source. The resulting vapor flows with its high pressure to the expansion turbine, where it is isentropic expanded. A vapor-liquid mixture leaves the turboexpander and is then condensed into a liquid by another heat exchanger with the available cooling medium. The condensed liquid is then pumped back to the evaporator to complete the cycle.[46]

If water is the working fluid to generate a high-pressure steam, the cycle is called Rankine Cycle. But when an organic working fluid is used, we talk about an Organic Rankine Cycle. Organic liquids are mainly composed of carbon and hydrogen and aren't resistant to high temperatures. They are more qualified than water if the temperature difference between heat source and heat sink is

[46] (www.wikipedia.com, 2012)

small and if the existing heat source is at a low temperature level. The working fluid used has an impact on the working process and thus affect the efficiency. [47]

The efficiency depends essentially on the temperature level of the supplied heat and the temperature level of dissipated heat. It therefore follows from the ratio of benefit, which means the efficiency of the turbine, to cost, the supplied amount of heat. How to calculate the efficiency is shown in the equation 4 below.[48]

$$\eta = \frac{P_{turbine}}{Q_{sup}}$$

Equation 4: Efficiency of a Steam Cycle

The efficiency increases the higher the temperature of the supplied heat and the lower the temperature of the dissipated heat. If the dissipated heat is no longer needed, it can be cooled as much as possible and the efficiency would increase. But if it is used for other processes, the dissipated heat must be set to a higher temperature level and the efficiency would be somewhat lower.[48]

5.5. Gas and Steam Power Plant

A gas and steam power plant combines a gas turbine with a steam power cycle to achieve a higher performance and is shown in figure 21 [49] below.

After a conventional gas turbine, their exhaust gases are used to evaporate water via a heat exchanger and to operate a steam power process with the created steam.

In a gas and steam power plant, electricity is produced by one to four gas turbines and one steam turbine, wherein either each turbine drives a generator or a gas turbine and the steam turbine drive together one generator.[48]

Gas and steam power plants have a high efficiency because of the high process temperatures at the heat inlet of the gas turbine and the low temperatures of the heat transfer in the steam power cycle.[50] More advantages are flexibility, low emissions and low fuel consumption.[51]

Further developments will go toward increasing efficiency, reducing emissions and increasing availability. This in turn requires a material optimization to increase the turbine inlet

[47] (Rosanelli, 2009)
[48] (www.wikipedia.com, 2012)
[49] (http://cogeneration.net/combined-cycle-power-plants/, 2013)
[50] (TU-Dresden, 2007)
[51] (Elsen, 2003)

temperatures, combustion optimization, improved quality assurance, optimized maintenance and robust system components.[52]

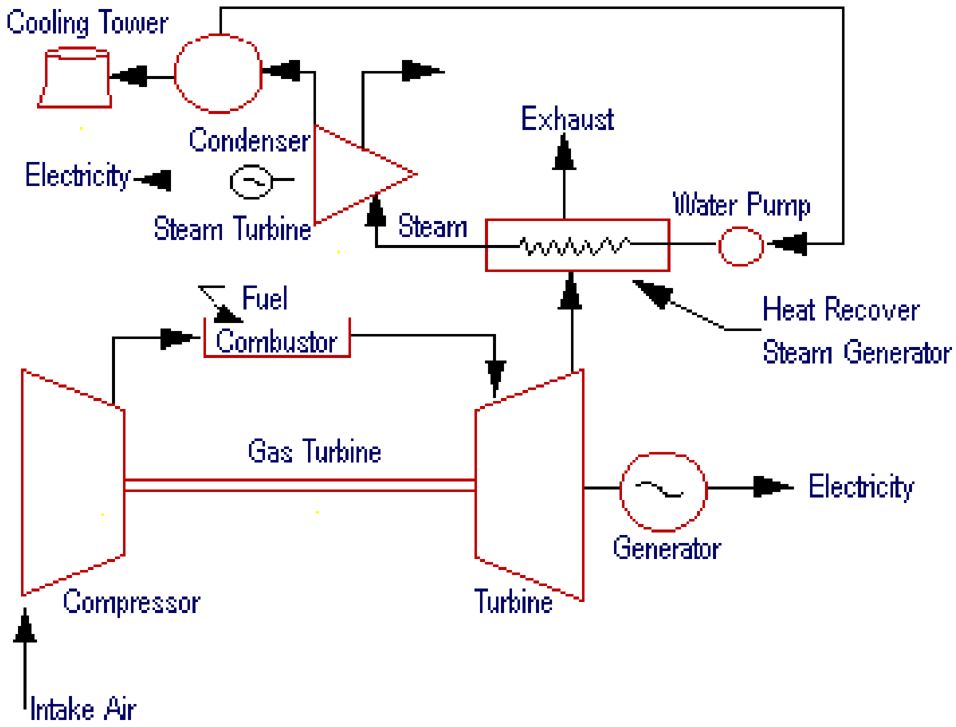


Figure 21: Scheme of a Gas and Steam Power Plant

5.6. Thermal Power Station

In a thermal power plant, also called solar chimney power plant, gas (usually air) is heated by the sun and rises in a chimney. One or more turbines generate electricity from the flow. Figure 22 [53] shows the operating principle.

The operation principle is actually very simple. Mostly the sun shines through a large glass or a transparent plastic roof (collector) and heats the ground and the air below like in a greenhouse. The warm air rises and flows under the glass roof to a chimney in the center of the plant. The result is an updraft (thermal), which is converted into electricity using turbines. In our case, we would not warm air, but continue to use our existing exhaust gases of the gas turbine. It can be used either horizontal axis turbines, at the annular foot of the tower, which means more exactly at the transition point from the collector to the tower, or a vertical axis turbine at the lower end of the tube.[54]

[52] (TU-Dresden, 2007)
[53] (http://www.ibfranetzki.de/aufwindkraftwerk.html, 2013)
[54] (http://www.aufwindkraftwerk.org/aufwindkraftwerk.php, 2013)

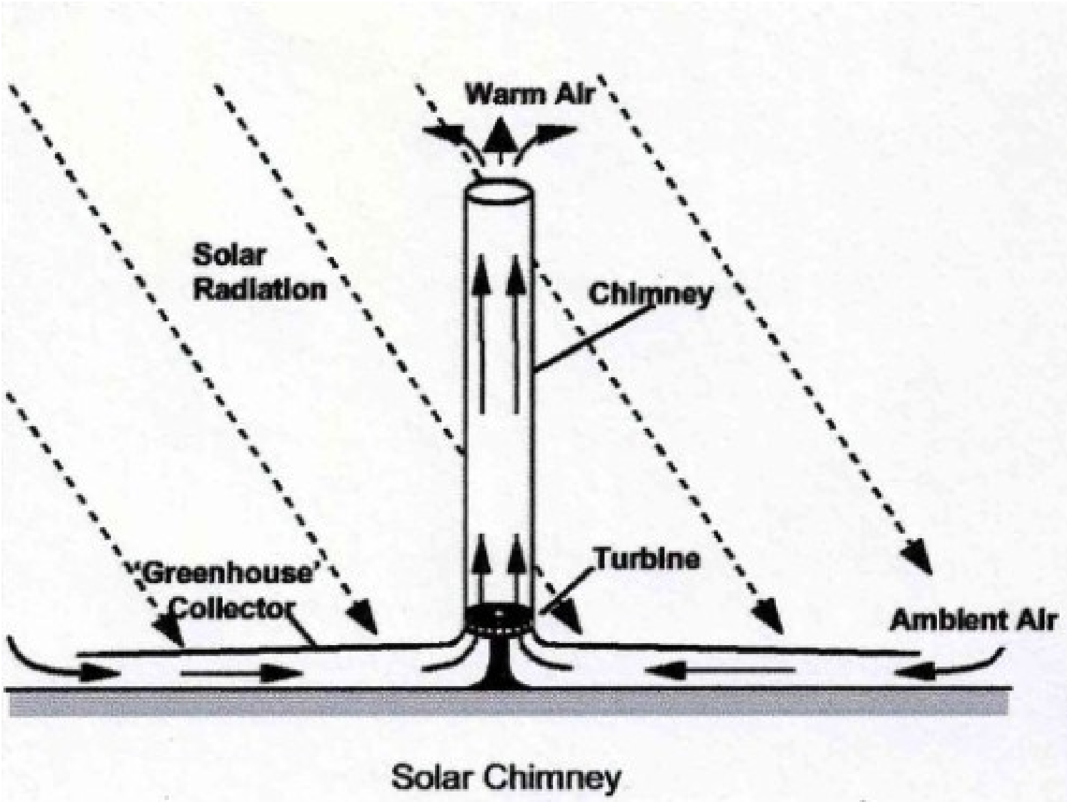


Figure 22: Operating Principle of a Thermal Power Station

Apart from the intensity of solar radiation, the performance of such a power plant depends on the collector area and the chimney height. The larger the covered area, the more air is heated and the faster rises the air in the chimney. The higher the chimney, the greater is the pressure difference to the ground and the faster rises the air. So in our case the most important influence factor would be the chimney height. Therefore lead both an increase of the collector area and of the chimney height to a greater capacity of the system.[55]

Generally thermal power stations have only a low energy density. The reason lies in the low temperature and pressure gradient that drives the flow in respect to the other power plants. The efficiency of the flow is obtained from the ratio of the flow of power to the power on the shaft of the rotor. The total efficiency of electric power generation with such power plants is with obtainable values less than 1% very low.[55]

Advantages of such solar chimney power plants are their sustainable, environmentally friendly energy production and the fact that they are base load power plants. Disadvantages include the large space requirements, the architectural complexity and the high investment costs.[56]

[55] (http://www.aufwindkraftwerk.org/aufwindkraftwerk.php, 2013)
[56] (Michael Farrenkopf, 2013)

5.7. Heat Exchanger

A heat exchanger is used to transfer thermal energy from one material flow to another.
Heat exchangers can be classified in terms of heat transfer in direct, indirect and semi-indirect heat transfer. Direct heat transfer is based on the operation of combined heat and mass transfer with separable streams. Indirect heat transfer is characterized in that material flows are separated by a heat-permeable wall. Half Indirect heat transfer uses a heat accumulator. The heat accumulator is alternately heated by the hotter medium, and then cooled by the colder medium so as to transfer thermal energy from the hotter to the colder medium.[57]
The heat transfer is highly dependent on the geometric guide of both streams to each other and can be distinguished in three basic forms shown in figure 23.[57]

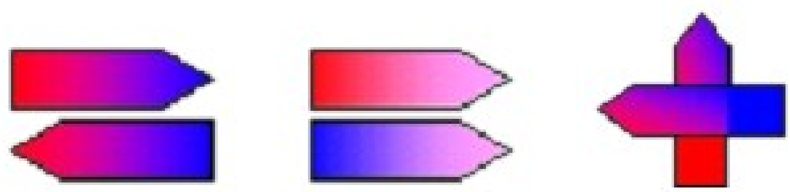


Figure 23: Counter Flow, Direct Flow and Cross Flow

Counter flow leads the substances so that they pass each other accommodating. Ideally, the temperature of the material flows are exchanged, which means that the initially hot fluid reaches the temperature of the cooler medium and the other way round. This ideal case requires equal heat capacities of the flows on both sides of the heat exchanger. In addition, the heat exchanger would have to have an efficiency of 100 percent, which is only approximately possible in reality. Direct flow means that the materials pass alongside each other in the same direction. Ideally, both material temperatures are adjusted and are always between the initial temperatures. If the streams cross their directions, it is called cross flow. In this case the resulting temperatures lie between the results from counter and direct flow. Also combinations of those three basic forms are possible.[57]
The efficiency is the ratio of thermal energy absorbed at the cold side to outgoing energy on the hot side. Since insulation reduces the heat loss to the environment, but is no prevention, a portion of the usable heat is lost. Depending on how large the temperature difference between the media and the environment is, this loss can be more or less.[57]

[57] (www.wikipedia.com, 2012)

The performance of a heat exchanger is large, if it is able to heat up one stream as much as possible and to cool the other down as much as possible. The heat always flows from hot to the cold stream.[58]

For a good efficiency the material that separates the media have to have a good thermal conductivity and a large surface. Besides the heat transfer between the surface and the flowing media must be quite good, which is favored by a turbulent flow. This means that the flow velocity should be high and the viscosity of the fluid low. However, an increased speed and a large wetted surface also increases the energy consumption for pumping the media through the heat exchanger.[58]

In heat exchangers, in which one medium is liquid and the other is a gas, the thermal capacity per volume of media is very different. Therefore, more gas must flow through the heat exchanger, and it is necessary to increase the area for heat transfer.[58]

There are many different types of heat exchangers, such as plate heat exchanger, spiral heat exchanger, tubular heat exchangers, etc.

5.8. Compressor

Compressors belong to the fluid energy machines and are used to compress gases. Figure 24 [59] illustrates a turbo compressor.

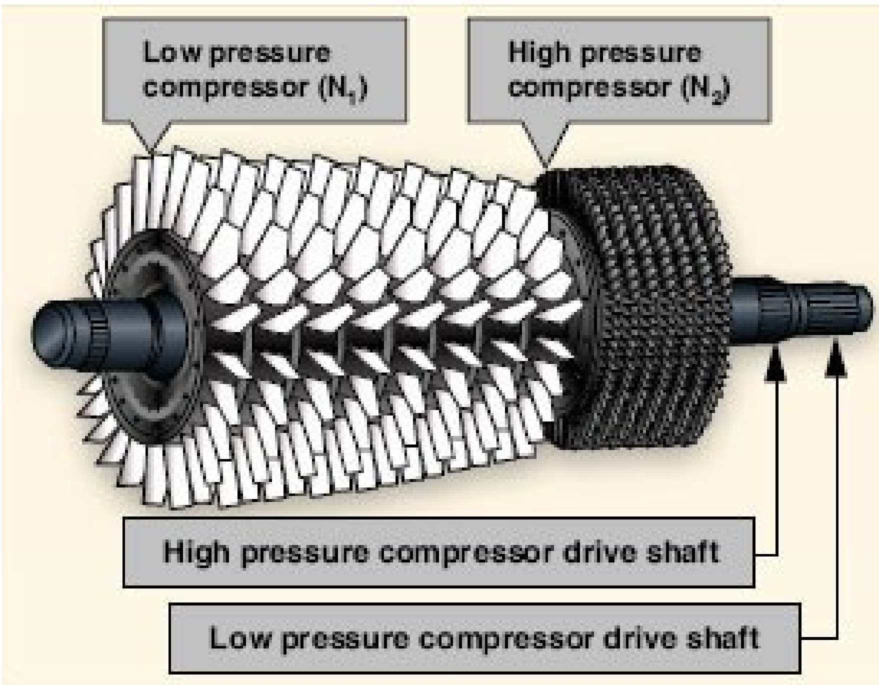


Figure 24: Depiction of a Turbo Compressor

[58] (www.wikipedia.com, 2012)
[59] (http://www.flightlearnings.com/2010/03/09/turbine-engine-instruments/, 2013)

Compacting or compressing means to decrease the volume of a gas. That means, that in compression processes an existing initial volume with a certain pressure is compressed to a smaller volume. After that the pressure in the smaller volume is higher and the gas heats up.

Basically there are two basic modes of operation of compressors. For one, the turbo compressor for high flow rates at low compaction pressures and the other displacement compressor for small volumes at high pressures. Furthermore, the different types of compressors are divided into oil-lubricated and oil-free compressors.[60]

Piston compressors use the principle of displacement. The gas is enclosed in a volume, compressed and discharged again. They work cyclically and have high pressure ratios, but only low flow rates.[60]

Screw compressors belong to the rotating, twin-screw displacement compressors with internal compression. They have a simple structure, small size, low mass and a uniform pulsation-free production. Two parallel, mechanically coupled shafts with intermeshing helical teeth in a case are the heart of this system. At the pitch line between the two shafts the passage for the fluid is mechanically locked. The medium is situated in the toothed gears is held by the housing therein. Flow direction is the axial direction. There are control slots located at the two ends of the axles in the housing for input (suction) and outlet (pressure side). The medium forms a spiral hose around the shaft and the medium is conveyed towards the pressure side by further rotation. The length of the waves, the slope of the spirals and the control slots must be adjusted so that there is no direct passage from the pressure side to the suction side, so that there is no reflux possible. The delivered volume of the medium is apart from the losses speed dependent.[61]

Turbo compressors add through a rotor according to the laws of fluid mechanics energy to the flowing fluid. This design works continuously and is characterized by a low pressure increase per level and a high volume throughput. Radial and axial compressors are the two main types of turbo compressors. In the axial compressor the compressing gas flows in the direction parallel to the axis of the compressor. In the radial compressor, the gas flows axial in the impeller of the compressor and is then deflected to the outside. Here, the pressure does not increase by the tapered channel cross section, rather by the fact that the space between the blades has the form of a diffuser. Pressure and temperature increase while the speed decreases.[61]

The most important characteristics of compressors include the delivery quantity (volume of fluid delivered per unit of time), the operating pressure (pressure achievable), the pressure ratio (discharge pressure / suction pressure) and the volumetric efficiency (ratio of extracted to the theoretical volume flow).[61]

[60] (http://www.flightlearnings.com/2010/03/09/turbine-engine-instruments/, 2013)
[61] (www.wikipedia.com, 2012)

5.9. Pump

Pumps belong to the fluid energy machines and are machines with which liquids are moved. For this the drive work is converted into the kinetic energy of the medium. Pumps are classified according to their operating principle in two significant main groups, displacement and flow pumps.

For displacement pumps, the medium is conveyed by self-contained volumes, prevention of back flow is achieved by valves or dampers, other media or gravity. Except through design-related leakage the medium can’t flow through the pump in the reverse direction. Displacement pumps are self-priming usually. The maximum suction height is limited by the achievable vacuum, the local air pressure, the density of the medium and the flow resistance to be overcome. It is additionally distinguishes between constant and variable displacement pumps. Constant displacement pumps always displace the same volume with each rotation, while variable displacement pumps on the other hand can adjust the displacement volume.[62]

In turbo machinery, the energy transfer is effected exclusively by fluid mechanical processes. The medium flows through the machine freely without dampers and valves. At standstill, the medium could flow through the pump backwards, therefore valves must be used. Flow pumps are not self-priming, so the suction lines must always be filled with fluid, or a sufficiently large volume of liquid before the actual impeller inlet must be present. The maximum suction height is here also limited by the local air pressure and the occurring flow resistance. Flow pumps can be divided into axial, diagonal and centrifugal pumps.[62]

[62] (www.wikipedia.com, 2012)

6. Thermal Fundamentals and Performance Calculations of the Machine Components

Here are the basics of thermodynamics and performance calculations of the individual machine components calculated with our specified reservoir parameters. The calculations and especially the results will vary more or less with the input parameters. Thus, the process flow has to be considered according to the reservoir conditions always new and modified.
These data should provide an overview of performance and help to select the suitable machines. The investment cost will be slightly higher because it will be necessary for most components to make them customize according to the given parameters. For some it may also be that there is still need for development.

6.1. Expander

For the relaxation in the expander, we assume the ideal case of a purely isentropic process. Figure 25 shows the p-V and the T-S diagram of such an isentropic expansion.

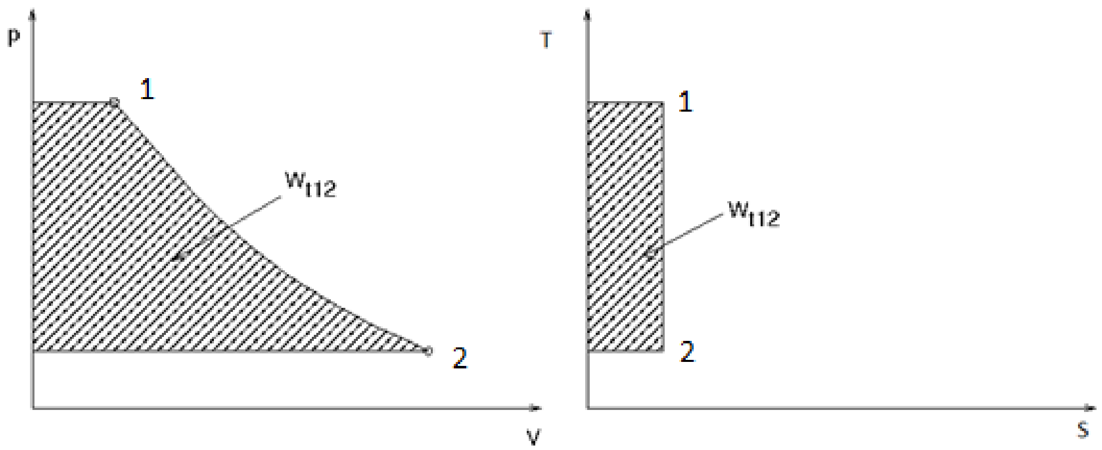


Figure 25: p-V and T-S Diagram of an Isentropic Expansion

Isentropic equation[63]:

$$p_2 \cdot {V_2}^{\kappa} = p_1 \cdot {V_1}^{\kappa} = const.$$

Equation 5: Isentropic Equation

$$\frac{p_2}{p_1} = \left(\frac{V_2}{V_1}\right)^{-\kappa}$$

Thermal equation of state [62]:

$$\frac{p_2 \cdot V_2}{T_2} = \frac{p_1 \cdot V_1}{T_1}$$

Equation 6: Thermal Equation of State

[63] (Oberwinkler, 2011)

The combination of these two equations:

$$\frac{p_2}{p_1} = \frac{T_2}{T_1} \cdot \frac{V_1}{V_2}$$

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

$$\frac{T_2}{T_1} = \left(\frac{V_2}{V_1}\right)^{1-\kappa} = \left(\frac{p_2}{p_1}\right)^{\frac{\kappa-1}{\kappa}}$$

Equation 7: Combination of the Isentropic and the Thermal Equation

1. Let's try the case to relax the CO_2-methane mixture from 150 bar after the production well to 20 bar. 20 bar so because subsequent gas turbines work with such pressure.

   Information: $p_1 = 150\ bar$ $p_2 = 20\ bar$

   $T_1 = 120\ °C = 393\ K$ $T_2 = ?$

   $\kappa_{methane} = 1.316$ ...isentropic exponent for methane

   $\kappa_{CO_2} = 1.301$ ...isentropic exponent for CO_2

   $\Rightarrow\ \kappa_{mixture} = 1.3$

   From equation 7 follows:

   $T_2 = T_1 \cdot \left(\frac{p_2}{p_1}\right)^{\frac{\kappa-1}{\kappa}} = 393 \cdot \left(\frac{20}{150}\right)^{\frac{1.3-1}{1.3}}$ = 246.9 K = -26 °C

   Since we are already here in the negative temperature range, this is not the optimum output range for the operation of our subsequent gas turbine.

2. So now we try to reduce the temperature of 120 ° C to 15 ° C.

   Information: $p_1 = 150\ bar$ $p_2 = ?$

   $T_1 = 120\ °C = 393\ K$ $T_2 = 15\ °C = 288\ K$

   $\kappa_{methane} = 1.316$

   $\kappa_{CO_2} = 1.301$

   $\Rightarrow\ \kappa_{mixture} = 1.3$

   From equation 7 follows:

   $$p_2 = p_1 \cdot \left(\frac{T_2}{T_1}\right)^{\frac{\kappa}{\kappa-1}} = 150 \cdot \left(\frac{288}{393}\right)^{\frac{1.3}{1.3-1}} = 39\ bar$$

   Now we have the appropriate parameters for the operation of a subsequent gas turbine.

We now want to determine the technical work and thus the expander performance.[64]

$$W_t = \dot{m} \cdot c_{p_{mixture}} \cdot \Delta T$$

Equation 8 Determination of the Technical Work

$\Delta T = 120\,°C - 15\,°C = 105\,°C = 105\,K$

$c_{p_{methane}} = 2.1562\; kJ/kgK$

$c_{p_{CO_2}} = 0.8169\; kJ/kgK$

$\Rightarrow\; c_{p_{mixture}} = 0.1 \cdot 2.1562 + 0.9 \cdot 0.8169 = 0.9508\; kJ/kgK$

$M_{methane} = 16.04\; g/mol$

$M_{CO_2} = 44.01\; g/mol$

$\Rightarrow M = \sum_i x_i \cdot M_i = 0.9 \cdot 44.01 + 0.1 \cdot 16.04 = 41.213\; g/mol$

$w_{methane} = \frac{x_{methane} \cdot M_{methane}}{M} = \frac{0.1 \cdot 16.04}{41.213} = 0.039$

$w_{CO_2} = \frac{x_{CO_2} \cdot M_{CO_2}}{M} = \frac{0.9 \cdot 44.01}{41.213} = 0.961$

$\rho_{methane} = 0.72\; kg/m^3$

$\rho_{CO_2} = 1.98\; kg/m^3$

$\Rightarrow\; \rho_{mixture} = \left(\sum_i \frac{w_i}{\rho_i}\right)^{-1} = \left(\frac{0.039}{0.72} + \frac{0.961}{1.98}\right)^{-1} = 1.85\; kg/m^3$

$\dot{V} = 1\,000\,000 \frac{m^3}{d} = \frac{1\,000\,000}{24 \cdot 60 \cdot 60}\; s/d = 11.574\; s/d$

$\dot{m} = \rho_{mixture} \cdot \dot{V} = 1.85\; kg/m^3 \cdot 11.574\; s/d = 21.4\; kg/s$

$W_t = \dot{m} \cdot c_{p_{mixture}} \cdot \Delta T = 21.4\; kg/s \cdot 0.9508\; kJ/kgK \cdot 105\; K = 2135.7\; kJ/s = 2135.7\; kW$

We can gain a maximum energy of 2 135.7 kW with the expansion of our gas mixture and the specified flow rate after the production well. Since a machine never achieves an efficiency of 100%, we are only able to actually use some of that energy. The efficiency is higher, when the inlet temperature of the gas and the pressure ratio of the turbine become higher.[65]

[64] (Oberwinkler, 2011)
[65] (www.wikipedia.com, 2012)

In order to use the highest possible energy, it is recommended to inform a manufacturer on the given parameters to find the most appropriate expander. It may also be that certain modifications or developments may cause the machine to a higher power potential.

6.2. Separator

Gas separators must be configured according to the individual applications and adapted to the needs and circumstances.
The energy requirement of gas separators depends on various factors and can therefore not be determined without detailed consideration of the individual case. However, the design of such a gas separator is a complex matter and requires a lot of effort.

6.3. Compressor

As for the relaxation, we assume the ideal case of a purely isentropic process for the compression, which is shown in figure 22 below.[66]

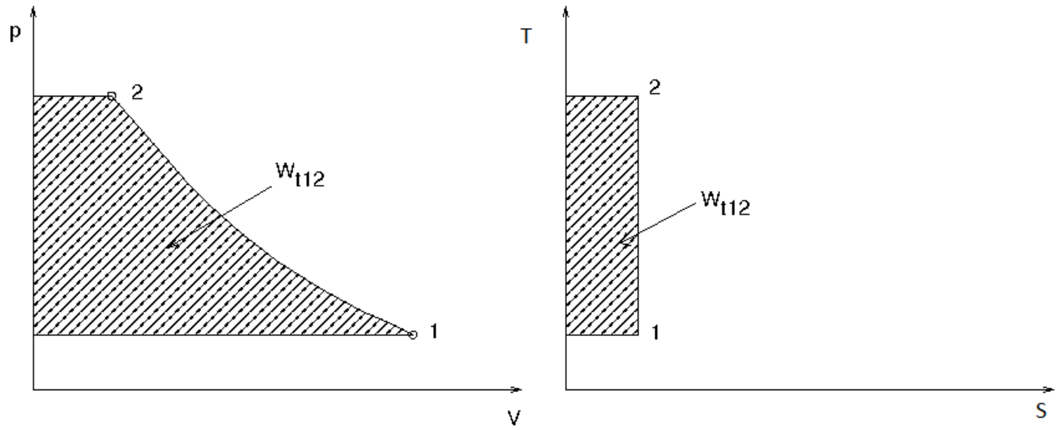


Figure 26: p-V and T-S Diagram of an Isentropic Compression

We want to compress the gaseous CO_2 from 39 bar after the relaxation in the expander to 80 bar. The reason for this is the fact that this pressure is required to reach again the supercritical state of CO_2.

Information: $p_1 = 39\ bar$ $\quad$ $p_2 = 80\ bar$

$T_1 = 15\ °C = 288\ K$ $\quad$ $T_2 = ?$

$\kappa_{CO_2} = 1.301$

[66] (Oberwinkler, 2011)

From equation 7 follows:

$$T_2 = T_1 \cdot \left(\frac{p_2}{p_1}\right)^{\frac{\kappa-1}{\kappa}} = 288 \cdot \left(\frac{80}{39}\right)^{\frac{1.301-1}{1.301}} = 340\ \text{K} = 67\ °\text{C}$$

CO_2 is then available at a pressure of 80 bar and a temperature of 67 °C and reaches therefore its critical state. The supercritical CO_2 can thus be injected again through the injection well into the reservoir. Here for we do not need any pump, because the pressure which is required for the injection is only about 50 bar. We also do not need a heat exchanger to bring the CO_2 to its supercritical temperature, since the temperature of the gas increases during the compression by more than 50 °C. Thus the isentropic compression provides us the desired supercritical state of CO_2.

To determine the technical work of the compressor we use again equation 8:[67]

$$P_t = \dot{m} \cdot c_{p_{CO_2}} \cdot \Delta T$$

$$\dot{m} = 21.4\ kg/s \cdot 0.9 = 19.26\ kg/s$$

$$c_{p_{CO_2}} = 0.8169\ kJ/kgK$$

$$\Delta T = 67\ °C - 15\ °C = 52\ °C = 52\ K$$

$$P_t = \dot{m} \cdot c_{p_{mixture}} \cdot \Delta T = 19.26\ kg/s \cdot 0.8169\ kJ/kgK \cdot 52K = 818.1\ kW$$

For the compression of the gaseous CO_2 we need a minimum energy amount of 818.1 kW, which is relatively small in relation to the gained energy during the expansion.

The isentropic efficiency of a compressor rises with the design size and achieves top values of 87-92% for axial compressors. The efficiency is lower for centrifugal compressor.[68]

$$P = \frac{P_t}{\eta} = \frac{818.1\ kW}{0.9} = 909\ kW$$

The efficiency increases our needed energy amount to 909 kW.

[67] (Oberwinkler, 2011)
[68] (Groten & Feldhusen, 2007)

6.4. Gas Turbine

A gas turbine operates normally according to the Joule-process, which is shown in Figure 27.[69]

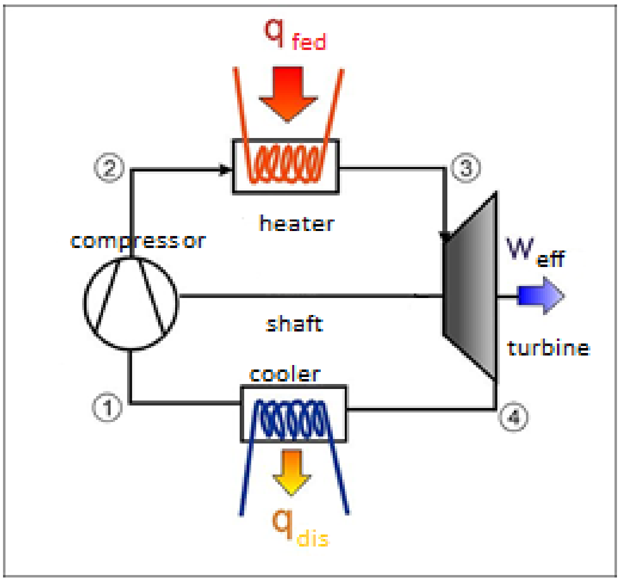


Figure 27: Representation of the Joule Process

It is a comparison process for the processes occurring in a gas turbine during operation and consists of two isentropic and two isobaric state changes. These state changes are highly visible in the p-V and T-S diagram illustrated in figure 24.[69]

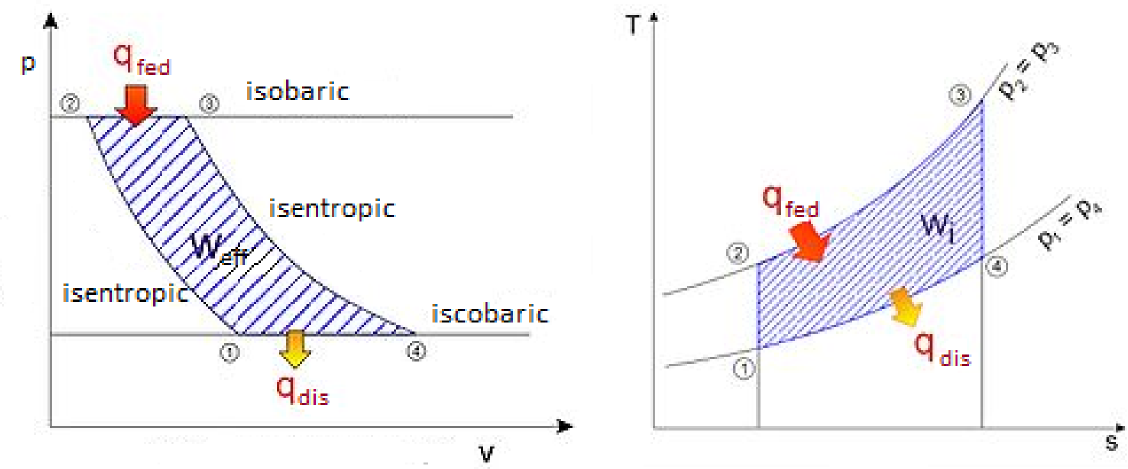


Figure 28: p-V and T-S Diagram of the Joule Process

[69] (www.wikipedia.com, 2012)

The Joule process essentially consists of four process steps. 1 - 2 is an isentropic compression by an adiabatic compressor, wherein work is supplied. 2-3 shows an isobaric intake of heat through a heat exchanger. At 3 –4 occurs an isentropic expansion through an adiabatic turbine, which means that the turbine work is removed. And from 4-1 it comes to an isobaric heat transfer, also through a heat exchanger. The area enclosed by the lines corresponds to the specific process work w.[70]
In our case, the cooling does not apply because of the continuous suction and compression of cold gas, what is called an open process.

The performance of a gas turbine results from multiplying the amount of heat supply with the overall efficiency, shown in equation 9.[71]

$$P = \eta_{ges} \cdot Q_{sup}$$

Equation 9: Performance Calculation of a Gas Turbine

The amount of fed heat in this case is generated from the combustion of methane, and can be calculated as follows. [68]

$$Q_{sup} = \Delta H_{CH_4} \cdot \dot{m} = 50\ MJ/kg \cdot 21.4\ kg/s \cdot 0.1 = 107\ MJ/s = 107\ MW$$

ΔH_{CH_4} ... Calorific value of CH_4

The total efficiency in a gas turbine is a composition of the quality grade, the thermal efficiency, the mechanical efficiency and the electrical efficiency. Equation 10 illustrates the overall efficiency.[71]

$$\eta_{total} = QG \cdot \eta_{th} \cdot \eta_{mech} \cdot \eta_{el}$$

Equation 10: Calculation of the Overall Efficiency of a Gas Turbine

The pressure ratio of heavy industrial gas turbines is between 10 to 16 bar and by blade cooling gas turbine inlet temperatures up to 1200 °C can be reached. With these parameters, we can now assume the effective efficiency from figure 29.[71]

[70] (www.wikipedia.com, 2012)
[71] (Groten & Feldhusen, 2007)

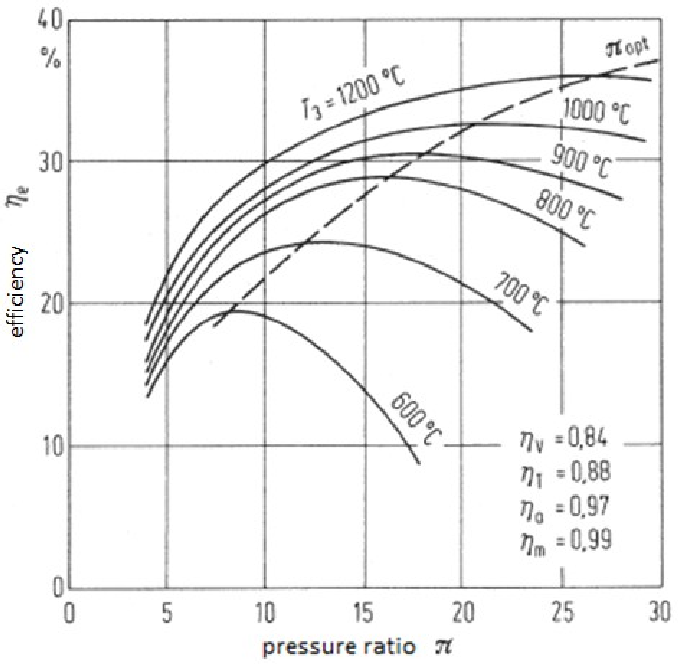


Figure 29: Effective Efficiency of a Simple Gas Turbine Cycle on the Pressure Ratio and the Turbine Inlet Temperatures

The effective efficiency, which consists of the quality grade, the thermal efficiency and the mechanical efficiency, is so to about 32%. The electrical efficiency of generators has values between 95-98 %.[72] This results in an overall efficiency of about 30 %.

$$P = \eta_{total} \cdot Q_{sup} = 0.3 \cdot 107\ MW = 32.1\ MW$$

Assuming an efficiency of 30 %, the gas turbine provides us with a capacity of 32.1 MW.

6.4.1. Combustion Process in the Gas Turbine

Figure 30 shows the energy and mass balance in an adiabatic combustion chamber.[73]

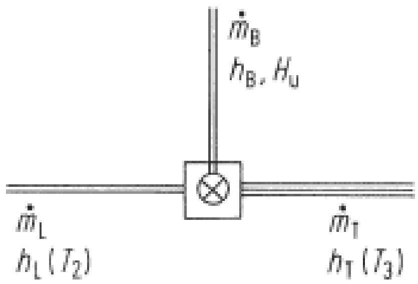


Figure 30: Energy and Mass Balance in an adiabatic Combustion Chamber

[72] (www.wikipedia.com, 2012)
[73] (Groten & Feldhusen, 2007)

Wanted is the energy that is still in the exhaust gases after passing through the gas turbine. For this purpose we compute in equation 11, the amount of heat that is still in the exhaust gases.[74]

$$Q_{exhaust} = \dot{m}_{H_2O} \cdot c_{p_{H_2O}} \cdot \Delta T + \dot{m}_{CO_2} \cdot c_{p_{CO_2}} \cdot \Delta T + \dot{m}_{residual} \cdot c_{p_{residual}} \cdot \Delta T$$

Equation 11: Calculation of the Heat in the Exhaust Gases of the Gas Turbine

The specific heat capacity of the remaining residue after combustion will not differ much from the specific heat capacity of air.

$c_{p_{H_2O\ gaseous}} = 1.8\ kJ/kgK$

$c_{p_{CO_2}} = 0.8169\ kJ/kgK$

$c_{p_{residual}} \approx c_{p_{air}} = 1.005\ kJ/kgK$

The temperature difference represents the difference between the ambient temperature and the exhaust gas temperature. Looking at the values of exhaust temperatures of various gas turbines, it can be seen that 550 °C is a realistic value.[75]

$\Delta T = 550\ °C - 15\ °C = 535\ °C = 535\ K$

Methane is burned with air, creating CO_2, water and a residual part. The stoichiometric equation is as follows.

$$CH_4 + 2 \cdot (O_2 + residual\ part) \rightarrow CO_2 + 2 \cdot H_2O + residual\ part$$

$M_{CH_4} = 16.04\ g/mol$

$M_{CO_2} = 44.01\ g/mol$

$M_{O_2} = 6.36\ g/mol$

$M_{H_2O} = 36\ g/mol$

$\dot{m}_{CH_4} = 21.4\ kg/s \cdot 0.1 \cdot 10^3 = 2140\ g/s$

To determine the amount of substance n, the equation for the determination of the molar mass has to be transformed.

$$M = \frac{m}{n}$$

Equation 12: Determination of the Molar Mass

[74] (Oberwinkler, 2011)
[75] (Siemens Power Generation, 2013)

$$\dot{n} = \frac{\dot{m}}{M}$$

$\dot{n}_{CH_4} = \frac{\dot{m}_{CH_4}}{M_{CH_4}} = \frac{2140\ g/s}{16.04\ g/mol} = 133.4\ mol/s$

$\dot{n}_{CO_2} = 133.4\ mol/s$

$\dot{n}_{O_2} = 2 \cdot 133.4\ mol/s = 266.8\ mol/s$

$\dot{n}_{H_2O} = 2 \cdot 133.4\ mol/s = 266.8\ mol/s$

$\dot{m}_{CO_2} = \dot{n}_{CO_2} \cdot M_{CO_2} = 133.4\ mol/s \cdot 44.01\ g/mol = 5870\ g/s = 5{,}87\ kg/s$

$\dot{m}_{O_2} = \dot{n}_{O_2} \cdot M_{O_2} = 266.8\ mol/s \cdot 16.04\ g/mol = 4279\ g/s = 4{,}28\ kg/s$

$\dot{m}_{H_2O} = \dot{n}_{H_2O} \cdot M_{H_2O} = 266.8\ mol/s \cdot 36\ g/mol = 9604\ g/s = 9.60\ kg/s$

The mass fraction of oxygen in air is about 22%, thus follows:[76]

$\dot{m}_{air} = \frac{\dot{m}_{O_2}}{0.22} = \frac{4.28\ kg/s}{0.22} = 19.45\ kg/s$

$\dot{m}_{residual\ part} = \dot{m}_{air} - \dot{m}_{O_2} = 19.45\ kg/s - 4.28\ kg/s = 15.17\ kg/s$

$\dot{m}_{exhaust} = \dot{m}_{CO_2} + \dot{m}_{H_2O} + \dot{m}_{residual\ part} = 5.87 + 9.60 + 15.17 = 30.64\ kg/s$

$$Q_{exhaust} = 9.60\ kg/s \cdot 1.8\ kJ/kgK \cdot 535\ K + 5.87\ kg/s \cdot 0.8169\ kJ/kgK \cdot 535\ K + 15.17\ kg/s \cdot 1.005\ kJ/kgK \cdot 535\ K = 19966.8\ kJ/s = 19966.8\ kW$$

To calculate the specific heat capacity of the gas turbine exhaust gas we use equation 13:[76]

$$Q = \dot{m} \cdot c_p \cdot \Delta T$$

Equation 13: Calcualtion of the Quantity of Heat

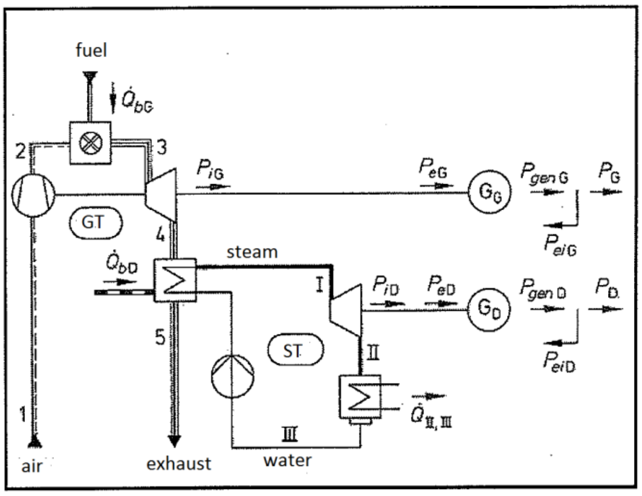
$$c_{p_{ex}} = \frac{Q_{ex}}{\dot{m}_{ex} \cdot \Delta T} = \frac{19966.8\ kJ/s}{30.64\ \mathrm{kg/s} \cdot 535\ \mathrm{K}} = 1.218\ kJ/kgK$$

6.5. Heat Exchanger

The heat exchanger is used to evaporate the water for the steam turbine process. He makes use of the exhaust gas temperature of the gas turbine, and thus represents the link between gas

[76] (Groten & Feldhusen, 2007)

turbine and steam turbine. Figure 31 shows the diagram of a combined gas and steam turbine process, where the heat exchanger can be seen between 4 – 5.[77]



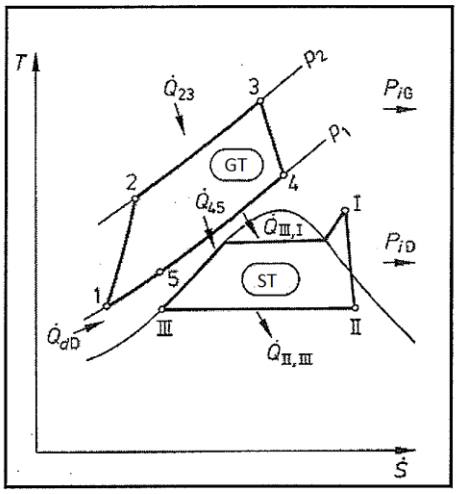


Figure 31: Combined Gas and Steam Turbine Process

We want our water to heat up to 270 °C, since we can assume that it will be a good average temperature for a steam turbine process. After the 2nd law of thermodynamics in the formulation of Clausius, it is for a body of lower temperature not possible to transfer heat to a body of higher temperature. Thus our exhaust temperature after the heat exchanger has as well 270 °C.[77]

Our parameters of the exhaust gas, which is supplied to the heat exchanger then result into following:

Input material:	exhaust gas
Mass flow:	30.64 kg/s
Inlet temperature:	550 °C
Output temperature:	270 °C

$c_{p_{H_2O\ liquid}} = 4.19\ kJ/kgK$

$c_{p_{H_2O\ gaseous}} = 1.8\ kJ/kgK$

$c_{p_{ex}} = 1.218\ kJ/kgK$

$H_{V_{H_2O}} = 333.5\ \mathrm{kJ/kg}$

$h" = 2800\ kJ/kg$ …from the Mollier diagram in the appendix

$h_0 = 200\ kJ/kg$ …from the Mollier diagram in the appendix

[77] (Oberwinkler, 2011)

By equating the energies follows:

$$\Delta Q_{exhaust} = \Delta Q_{water}$$

$$\dot{m}_{ex} \cdot c_{p_{ex}} \cdot (T_{ex\,in} - T_{ex\,out}) = \dot{m}_{H_2O} \cdot (h" - h_0)$$

$$\dot{m}_{H_2O} = \frac{\dot{m}_{ex} \cdot c_{p_{ex}} \cdot (T_{ex\,in} - T_{ex\,out})}{(h" - h_0)} = \frac{30.64\ kg/s \cdot 1.218\ kJ/kgK \cdot (550-270)\ K}{(2800-200)\ kJ/kg} = 4.02\ kg/s$$

Our parameters of the water, which is heated up by the heat exchanger then result into following:

Input material:	water
Mass flow:	4.02 kg/s
Inlet temperature:	20 °C
Output temperature:	270 °C

Thus all parameters for a custom-made heat exchanger are given.

The temperature difference represents the difference between the ambient temperature and the exhaust gas temperature.

$\Delta T = 270\ °C - 15\ °C = 255\ °C = 255\ K$

$Q_{exhaust} = \dot{m}_{H_2O} \cdot c_{p_{H_2O}} \cdot \Delta T + \dot{m}_{CO_2} \cdot c_{p_{CO_2}} \cdot \Delta T + \dot{m}_{residual} \cdot c_{p_{residual}} \cdot \Delta T = 9.60\ kg/s \cdot 1.8\ kJ/kgK \cdot 255\ K + 5.87\ kg/s \cdot 0.8169\ kJ/kgK \cdot 255\ K + 15.17\ kg/s \cdot 1.005\ kJ/kgK \cdot 255\ K = 9516.9\ kJ/s = 9516.9\ kW$

The remaining energy in the exhaust gas is thus still 9516.9 kW.

6.6. Steam Turbine Process

To determine the power requirement of the turbine, and the change of state, we use the Mollier diagram from the appendix. The variables of temperature, humidity, enthalpy and density are determined graphically. In thermodynamics, you can find far more complex diagrams, which apply the aforementioned states over various pressures.[78]

The feed water pump increases the pressure of the water to 50 bar before it is heated up by the heat exchanger to 270 ° C, thus the water is completely transformed to saturated steam with the enthalpy h'' = 2800 kJ/kg. Then we relax the saturated vapor isentropic in the turbine at a back pressure of 0.1 bar, which corresponds to a practical value. A low back pressure has a positive effect on the turbine performance. The enthalpy is thereby lowered to a value of h' = 1900 kJ/kg.

[78] (www.wikipedia.com, 2012)

The amount of heat supplied through the heat exchanger to the steam turbine cycle can be calculated as follows.

$$Q_{sup} = 19966.8\ kW - 9516.9\ kW = 10449.9\ kW$$

The turbine power is calculated by the following equation 13.[79]

$$P_T = \Delta h \cdot \dot{m}$$

Equation 14: Calculation of the Turbine Performance

$\dot{m}_{H_2O} = 4.02\ kg/s$

$\Delta h = h'' - h' = 2800\ kJ/kg - 1900\ kJg/kg = 900\ kJ/kg$

$$P_T = \Delta h \cdot \dot{m} = 900\ kJ/kg \cdot 4.02\ kg/s = 3618\ kJ/s = 3618\ kW$$

In this case, the performance that can be tapped at the turbine results in 9648 kW.

The power requirement of the feed water pump is given by equation 14.[76]

$$P_{pump} = \Delta p \cdot \dot{V}$$

Equation 15: Power Requirement of the Feed Water Pump

$\Delta p \approx 50\ bar = 50 \cdot 10^5\ Pa$

$\rho_{H_2O} = 1000\ kg/m^3$

$\dot{V}_{H_2O} = \dfrac{\dot{m}_{H_2O}}{\rho_{H_2O}} = \dfrac{10.72\ kg/s}{1000\ kg/m^3} = 0.01072\ m^3/s$

$$P_{pump} = \Delta p \cdot \dot{V} = 50 \cdot 10^5\ Pa \cdot 0.01072\ m^3/s = 53600\ W = 53.6\ kW$$

The power required by the feed pump is found to be 53.6 kW.

The total efficiency of the steam turbine process is given by equation 16.[79]

$$\eta = QG \cdot \eta_{th} \cdot \eta_{mech} \cdot \eta_{el}$$

Equation 16: Total Efficiency of the Steam Turbine Process

[79] (Oberwinkler, 2011)

$$QG = \frac{P_T}{Q_{sup} + P_{pump}} = \frac{3618\ kW}{10449.9\ kW + 53.6\ kW} = 0.35$$

For the thermal, mechanical and electrical efficiency, we assume realistic values based on experience.

$$\eta = QG \cdot \eta_{th} \cdot \eta_{mech} \cdot \eta_{el} = 0.35 \cdot 0.9 \cdot 0.95 \cdot 0.99 = 0.3$$

The performance of the steam turbine cycle results from equation 17.[80]

$$P_{stc} = \eta \cdot Q_{sup}$$

Equation 17: Performance of the Steam Turbine Cycle

$$P_{stc} = \eta \cdot Q_{sup} = 0.3 \cdot 10449.9\ kW = 3135\ kW$$

By the steam process, we could achieve a power of about 3135 kW. The use of an organic working fluid instead of water could improve the efficiency, but is much more complex in the selection of the working fluid.

6.7. Thermal Power Station

After heating up the water for the steam turbine cycle in the heat exchanger the gas turbine exhaust gas still has an energy amount of 9516.9 kW.

The efficiency of such thermal power plants is unfortunately only around 1%.[81]

$$P_{TPS} = \eta_{TPS} \cdot Q_{exhaust} = 0.01 \cdot 9516.9\ kW = 95.17\ kW$$

[80] (Oberwinkler, 2011)
[81] (http://www.aufwindkraftwerk.org/aufwindkraftwerk.php, 2013)

7. Process Flow and Concrete Description

The following figure 32 shows the schematic process flow of our energy cycle after specific observation of the thermal fundamentals and the performance calculations of the machine components.

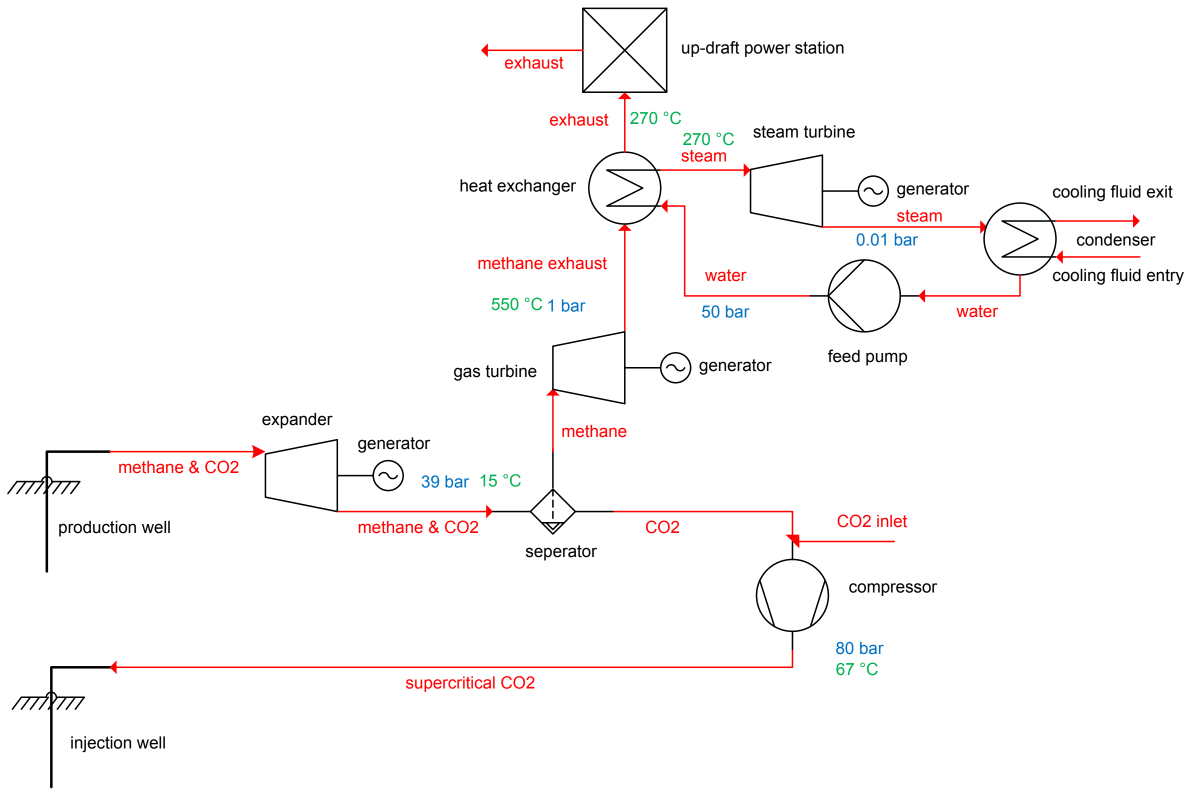


Figure 32: Depiction of the Process Flow

The flow of the CO_2-methane mixture is relaxed in a turbo expander after producing it from the production well with 120 °C and 150 bar. We relax our gas mixture at 15 ° C, since no solidification occurs and thus it is a suitable input temperature for the gas turbine. In the isentropic expansion to 15 ° C., a pressure of 39 bar is established and provides us with a theoretical output of 2135.7 kW. The theoretical output does not include the efficiency, which will slightly reduce the achievable energy.

After expansion, we split the methane from the CO_2 by using a membrane separator. Since such separators are designed for each special task and are very complex, it is not possible to determine their energy consumption.

In order to bring back the separated CO_2 in its supercritical state before injecting it into the injection well, we use a compressor. This compresses our CO_2 to 80 bar, resulting in a temperature increase to 67 °C. Because CO_2 has already reached its supercritical state at this value, no further heat exchanger is necessary and also a pump will not be needful, since only 50 bar are needed for the injection. Thus, the energy consumption of the compressor is about 909 kW, already including the efficiency factor of 0.9.

The separated methane is burned in a gas turbine and achieves a performance of 32.1 MW by including the efficiency factor, which is about 0.3. Manufacturers offer individual gas turbines, some even entire combined gas and steam plants.

The gas turbine exhaust is used to evaporate water in a heat exchanger for the steam turbine cycle. The heating up in the heat exchanger extracts the exhaust gas energy and feeds it to the water to evaporate it. The power output of the steam turbine cycle will be around 3135 kW the efficiency with factored. The ORC process represents an alternative to the steam cycle and differs essentially by the used working fluid and the increased efficiency.

Due to the left heat in the exhaust gas after the heat exchanger, it is possible to install a thermic power plant. The efficiency of such thermal power plants is, however, very low with 1%, so the amount of energy production will only be about 95.17 kW.

On the whole, you will receive a energy production of 30 MW, considering all energy producers and energy consumers.

8. Conclusion

To use other than hydrocarbon reservoir, which could be supplied with supercritical CO_2 and produce a CO_2-methane mixture out of the production well, could be used for energy generation.

By utilization of a turbo expander for the first expansion of the gas mixture already energy may be produced. Although the downstream connected separator for the separation of the methane and from the CO_2 and the compressor for the pressure and temperature increase of the CO_2 will consume energy. The compressor is necessary in order to bring the gaseous CO_2 back in its supercritical state and to press it in turn through the injection well into the reservoir.

The separated methane is then used to produce by the combustion in a gas turbine the greatest part of the possible energy generation. With the exhaust gases from the gas turbine, it is now possible to operate a steam turbine or ORC process via a heat exchanger and provide more power output.

In a steam turbine process a feed pump increases the pressure of the working medium to evaporate it in the heat exchanger, what further requires energy. The operated steam turbine on the other hand, produces a much higher amount of energy than what is needed for the feed pump. Due to this fact the steam turbine process and the ORC process can be used for further power generation. Thereafter, the operation of a thermal power plant with the remaining energy in the exhaust gases is possible, but doesn't provide that high amount of the desired energy generation.

On the whole, we can assume an energy production of approximately 30 MW, where is still great potential for improvement of the individual machine components and the use of ORC processes exists.

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Appendix

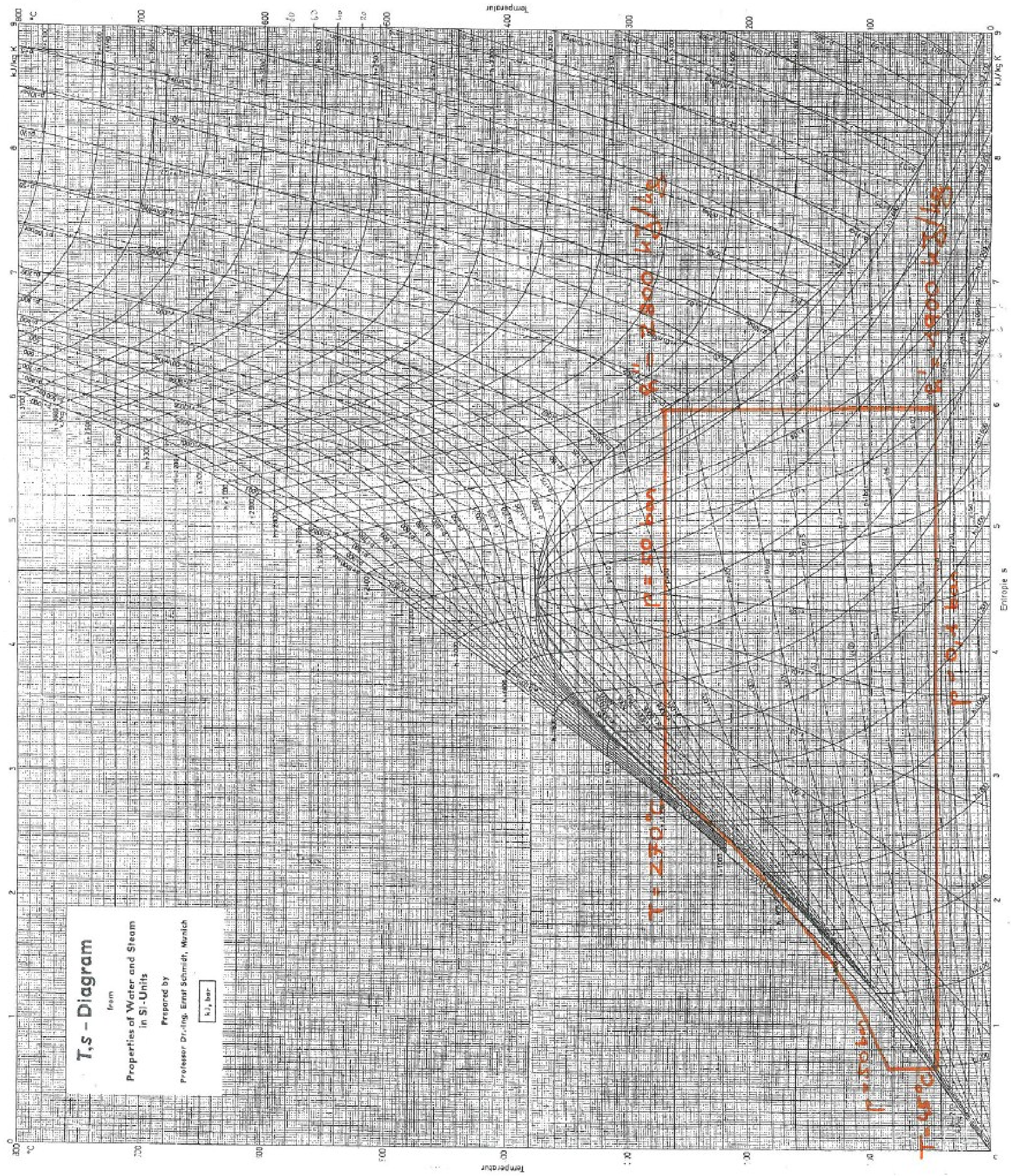

Figure 33: Mollier Diagram