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# INVESTIGATION OF THE THERMAL STORAGE SYSTEM FOR A 5 MW $_{\ensuremath{\textit{EL}}}$ CONCENTRATED SOLAR POWER (CSP) PILOT PLANT

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## ABSTRACT

The use of thermal energy storages is seen to greatly improve the financial and energetic benefits of a CSP power plant. In order to use the storage system effectively its type, design and dimensions have to meet the requirements of the plant. In this study, the modeling of a thermal energy storage and the attached plant is developed and a sensitivity analysis is executed. A packed bed of rocks with air as the HTF was chosen as a very low cost solution for the storage of a combined cycle power plant. Correlations for heat transfer, destratification, pressure drop and heat losses from the literature have been analyzed and eventually a one-dimensional model was build. This was then validated against experimental data from several studies with satisfactory overall outcome. A simplified plant control, following the proposed SUNSPOT cycle, has been implemented into the model and simulations were run to find major sensitivities to the storage's geometry, particle size and plant layout.

The found optimum dimensions of the storage are a diameter of 19 m, a height of 12 m, a particle size of 4 cm and gas and steam turbines with nominal ratings of 5.0 and 3.5  $MW_{el}$ , respectively. The calculated levelized cost of electricity reached a value of approximately 1.12 ZAR/kWh which is as expected for a plant of this size.

Further work has to be done on new plant control mechanisms, a more reliable cost calculation and a mechanically detailed design.

## CONFIRMATION

I confirm that I independently prepared the thesis and that I used only the references and auxiliary means indicated in the thesis.

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# NOMENCLATURE

	Abbreviations
CAPEX	Capital Expenditure
CFD	Computational Fluid Dynamics
CSP	Concentrating Solar Power
DNI	Direct Normal Irradiation
DSG	Direct Steam Generation
DEM	Discrete Element Method
GT	gas turbine
HX	heat exchanger
HTF	Heat Transfer Fluid
LCoE	Levelized Cost of Electricity
LNG	Liquefied Natural Gas
(E-)NTU	(Effectiveness-) Number of Transfer Units
OPEX	Operational Expenditure
OECD	Organisation for Economic Co-operation and Development
PV	Photovoltaic
SM	solar multiple
ST	steam turbine
TES	Thermal Energy Storage
GLE	Generalized Lévêque Equation

Subscripts		
0	no flow	
а	annual	
amb	ambient	
В	fossil burner	
С	charging cycle	
CN	Chambadal-Novikov efficiency	
ch	chemical	
СО	co-firing	
con	conduction	
CS	cross section of the storage perpendicular to flow	
dc	discharging cycle	
destrat	destratification	
dis	dispersion	
eff	effective	
el	electric	
ex	exergetic	
f	fluid	
GC	gas cycle	
GT	gas turbine	
HX	heat exchanger	
Ins	Insurance	
i	storage layer i	
id	idling mode	
in	incoming	
ins	inside	
insul	Insulation	
i	time step j	
lining	wall lining	
m	mean	
nom	at nominal load	
out	outgoing	
outs	outside	
a	isobaric	
la	part-load	
RE	receiver	
rad	radiation	
SC	steam cycle	
ST	steam turbine	
St	storage	
s	solid	
sol	solar	
t	turbulent	
th	thermal	
V	volumetric	
v \//	wall	
vv vv/	wall loss	
VVI	waii 1055	

	Symbols	
A	area	[m <sup>2</sup> ]
Bi	Biot number	[-]
С	specific thermal capacity	[kJ / (kg K)]
$C_a$	annual costs	[ZAR]
D	storage diameter	[ m ]
$d_h$	hydraulic particle diameter	[ m ]
$d_{ ho}$	particle diameter	[ m ]
Ε	exergy	[ kJ/K ]
e <sub>fuel</sub>	specific chemical energy in fuel	[ J <sub><i>ch</i></sub> / kg ]
f	correction factor	[-]
f <sub>fossil</sub>	specific fuel consuption per amount of generated electricity	[ J <sub>ch</sub> / J <sub>el</sub> ]
$f_s$	friction factor	[m <sup>-2</sup> ]
G	fluid mass flux	[ kg / (m² s) ]
h	specific enthalpy	[J/(kg K)]
H <sub>Tower</sub>	height of the receiver tower	[m]
Hg	Hagen number	[-]
L	total storage length in flow direction	[m]
Lf	distance between two equivalent spheres	
LCOE	Levelized Lost of Electricity	[ZAR/KVVN <sub>e/</sub> ]
LI		[a]
m m	mass flow rate	[Kg]
III NI	mass now rate	[KG/S]
n n	lube-lo-particle-diameter ratio	[-]
NITLI	Number of Transfor Units	[-]
NIU	Nusselt number	[-]
Nu Р	nusseit number	[\/\/]
n n	pressure	$\left[ N \right] (m^2)$
р Pr	Prandtl number	[ ]
nr.	specific price of fuel	[7ΔR / ]]
, O	boat flow rate	$[\Delta/(17) O_{Cn}]$
D R	thermal resistance in wall heat transfer	[vv]
n Ro	narticle Reynolds number	
r r	interest rate	[%]
' SM	solar multiple	[-]
Sh	Sherwood number	[-]
s	specific entropy	[ J / (ka K) ]
T/t	temperature	[K] / [°C]
V	storage volume	$[m^3]$
V	volume flow rate	$[m^{3}/s]$
V	velocity	[m/s]
W	work	[]]
X	vertical position inside storage	[m]
Xf	friction fraction	[-]
vear	time after commissioning	[a]
$\alpha$	heat transfer coefficient	[W/(m <sup>2</sup> K)]
$\Delta$	difference	[-]
ε	void fraction	[-]
ζ	capacity factor	[-]
$\eta$	efficiency	[-]
$\lambda$	thermal conductivity	[ W/(m K) ]
$\mu$	dynamic viscosity	[ kg/(m s) ]
$\pi$	pressure ratio	[-]
$\rho$	density	[ kg / m <sup>3</sup> ]
au	time	[s]
$\psi$	particle sphericity	[-]

# **1** INTRODUCTION

The necessity to shift from conventional power generation - mostly from fossil fuels - towards the use of renewable energy sources is widely acknowledged. The reasons for this are manifold: Firstly, the resources of fossil fuels and uranium are limited. To meet rising demands due to population growth and higher per capita consumption in developing countries - according to IEA (2011) - more difficult, risky, environmentally harmful and expensive methods have to be used to access the remaining oil, natural gas, coal or uranium. Secondly, the burning of fossil fuels results in vast amounts of CO<sub>2</sub> being unleashed into the atmosphere. This man-made CO<sub>2</sub> release is thought to cause global warming and thus change the weather dramatically (UN General Assembly 1994). Thirdly, many countries seek independence from energy carrier imports. Most of the world's oil and natural gas reserves are located in countries that are considered 'unstable'. Although most of the facts stated above have been known for a long time and have been brought to the public's attention during the 1970's oil crisis, according to IEA (2011) more than 90% of the OECD's (Organisation for Economic Co-operation and Development) primary energy supply is still by fossil or nuclear fuels.

#### 1.1 RENEWABLE ENERGY IN THE GRID

Besides an increase in energy efficiency, using renewable energy for heating, power generation and transportation is the most promising way of decreasing usage of resources. It is understood, that the change will be gradually and not one kind of renewable energy (hydro, wind, solar, geothermal, biomass) will be the sole replacement. This thesis will be focused on power generation through concentrating solar power (CSP) as part of a big power grid. As compared to wind power or photovoltaic (PV) where other forms of energy are converted directly into electricity, in CSP the sun's radiation is first converted into thermal energy of a heat transfer fluid (HTF). In a second step the heat is turned into electricity in a turbine (or other engine). The huge advantage of thermal energy as opposed to electricity is that it's much easier to store. Since wind load and solar irradiation never exactly match the demand profile and are not precisely predictable, they can only be added to the energy mix in considerable amounts once storage or power supply with extremely short ramp-up are added as well (e.g. gas turbines). The only viable technology to store big amounts of electricity is hydro power via pump storages which require very specific conditions. Gas turbines, on the other hand, have much higher running costs than base-load plants due to the fuel used (natural gas or oil) and require access to the fuel in question. CSP plants with thermal energy storage (TES) can dispatch the incoming solar power when needed, thus acting as a grid stabilizer and decreasing the demand for back-up power plants especially when non-dispatchable renewables are used as well.

## 1.2 THE SUNSPOT CYCLE

There are several different kinds of CSP plants, each with its unique characteristics. Most commercially run plants are parabolic throughs which have been built since the late 1980's in sizes up to 80 MW<sub>el</sub>. Their solar fields consist of troughs made of mirrors in a parabolic form around a receiver pipe in the middle. Thus, its a line focusing technology as is the Linear Fresnel Reflector. Another approach promises higher efficiencies due to a higher concentration of the suns light: point focusing. There are two prominent representatives of it: "Dish Stirling" and "Power Tower" designs. The first contains of dishes each equipped with its own Stirling engine (usually in the  $kW_{el}$  range), the latter of a field of mirrors (so called heliostats) that all focus on one central receiver on top of a tower.

The central receiver design allows for efficiency maximization through higher temperatures, reduction of losses because of shorter piping and cost reductions on many parts. There are several companies and research institutes building power tower plants with different philosophies regarding HTFs (molten salts, direct steam generation, air), unit size and storage concepts. A technology overview can be found on the NREL (2011) web site. Kröger (2011) proposed the SUNSPOT cycle (Stellenbosch University Solar Power Thermodynamic cycle), consisting of a Brayton cycle with air as the working fluid and central solar receiver and/or fuel combustor as a heat source, a Rankine cycle that feeds off the exhaust gas of the gas turbine and a storage system (see Figure 1.1). The storage is charged by the gas turbine's exhaust gas too and, when discharged with ambient air, powers the steam cycle's boiler. The layout solves some of the problems that CSP plants still have and has advantages over other designs:



Figure 1.1: The SUNSPOT cycle (Kröger 2011)

The combined cycle potentially raises the electrical efficiency as compared to single cycle plants. The gas turbine makes use of the higher Carnot efficiency at high temperatures that are possible because of point focusing and gas as an HTF. Using the still hot exhaust gas for running a steam cycle increases the efficiency further.

Hybridization, as in most CSP designs, is part of the design. This means co-firing of fossil fuel or biofuel when energy demand is higher than solar intake. This enables the plant to (a) run even when the sun is not shining, (b) make full use of solar irradiation even when it's lower than the minimum amount necessary to run the gas turbine and create a smoother ramp-up and (c) be able to react quickly to changing supply or demand. On the other hand, for environmental and

economic reasons, the amount of back-up fuel used should be limited. The storage serves similar purposes to hybridization without the drawbacks of additional fuel costs (only little power is needed for the pumps/fans). A big enough storage enables a solar power plant to have 24 hours of uninterrupted power generation, as the Gemasolar (2011) power plant demonstrated. It also protects the boiler and the steam turbine from temperature drops and can shift the Rankine cycle's cooling demands to the night time, increasing its efficiency due to lower ambient temperatures. The size of the storage can vary between less than an hour of full load for pure levelization of the supply profile to the boiler, some hours of full load for shifting some of the output to the evening and morning peaks or many days of full loads for being able to truly offer base-load characteristics. In the end, the 'best' configuration is a trade-off of storage size, back-up fuel consumption, solar field size and power output.

The SUNSPOT cycle is not finalized and therefore leaves room for different designs and plant control methods. A pilot plant with the following characteristics has been proposed by Gauché (2011):

- located in South Africa
- 5 MW<sub>el</sub> net gas turbine rating
- the gas turbine's outlet temperature and therefore the heat exchanger's and storage's inlet temperature is approximately 530 °C
- the receiver will heat the air up to approximately 800 1,000 °C
- to demonstrate CSP's ability to deliver base load, the plants power output should match South Africa's demand curve
- considerable storage size, allowing 24 hour continuous power generation
- cheap storage system design

## 1.3 OBJECTIVE

In the course of this study, a storage-centric model of the proposed SUNSPOT pilot plant has to be developed and full-year simulations with a South Africa's grid demand curve and weather data for a South African location have to be run. Performance indicators need to be calculated and a sensitivity analysis for some of the main parameters is to be executed, resulting in optimal configurations of these.

## 1.4 METHODOLOGY

Developing a model of the storage will require the following:

- decide on a TES design
- obtain adequate formula for pressure drop, heat transfer and temperature distribution during charging, discharging and idling mode of the TES
- approximate heat losses through the walls of the tank
- build new or adapt existing model for the remaining parts of the plant
- run simulation with different plant operation modi and component sizes, find sensitivities and optima
- verify the (partial) simulations

# 2 PREVIOUS WORKS

### 2.1 GENERAL DEFINITIONS

**Equivalent particle diameter**  $d_p$ : The equivalent diameter of the particles if they were spheres. It can be determined in different ways, e.g. as the side length of a box surrounding the particle. A good summary on the subject can be found in (Allen 2010).

**Particle Reynolds number**  $Re_p$ : The Reynolds number for the gas flow through the pores of a packed bed.  $Re_p = (\dot{m}_f d_p)/(A_{cs} \mu_f)$ 

**Void fraction**  $\varepsilon$ : The part of the total volume that's filled with air. It depends on particle shape, storage shape and dimensions, whether the packing is random and differs inside the bed. It's an important parameter in all fluid flow calculations and determines the thermal capacity of the storage.

**Thermocline:** A region in a fluid in which the temperature changes rapidly. It separates the hot part of a one-tank storage system from the cold. The steeper the temperature curve in this region the better the thermocline and the more of the storage's theoretical thermal capacity is technically usable. Influencing parameters for the quality of a thermocline in a packed bed are particle size, fluid mass flux and flow homogeneity. Figure 2.1 shows thermoclines with short and long thermocline zones.

# 2.2 TECHNICAL REVIEW ON THERMAL ENERGY STORAGE SYSTEMS

This section gives an overview of the available types of TES systems for CSP applications and their characteristics. As summarized by Duffie and Beckman (1991) and Gil et al. (2010) the major requirements on thermal energy storages for CSP are:

• (volumetric) energy capacity



Figure 2.1: Quality of thermocline during charging

- charge and discharge heat rates
- maximum and minimum temperatures, sensible or latent heat storage
- safety and environmental impact
- thermal and chemical stability for thousands of cycles
- heat losses
- quality of the thermocline after charging
- degradation of the thermocline during idle mode
- power requirements for charge and discharge
- costs (for the whole storage system)

According to Gil et al. (2010), thermal energy can be stored in three different forms: as a temperature rise (sensible heat), a phase change (latent heat) or chemical potential (see Figure 2.2) and thermal storage concepts can be either active (direct or indirect) or passive. Chemical and phase change stores promise great opportunities but are still subject to research whereas all existing TES systems of CSP plants store sensible heat.

The NREL (2011) website shows which TES systems have been built into CSP plants already. The simplest way of storing heat in a CSP plant is to use the primary HTF as the storage medium as well. This works relatively well when synthetic oil (as in Luz Industry's Solar Energy Generation System (SEGS) I plant) or molten salts are used because they remain liquid at elevated temperatures but the high price, especially of synthetic oil, proves a big financial drawback. In direct steam generation (DSG) plants or when gas is the primary HTF, a gaseous medium would have to be stored, resulting in either very high pressure inside the tank or low volumetric energy capacities. A so called direct active storage is therefore not practical for the SUNSPOT cycle.

In indirect active storage systems, the storage medium is separated from the primary HTF by a heat exchanger. The advantage of this configuration is that no compromises have to be made in finding a medium that serves as both, an HTF and a sensible energy storage. Instead two 'specialized' media can be used. The trade-off is between the cost reduction or efficiency



Figure 2.2: Storable thermal energy forms. Derived from Gil et al. (2010)

improvement through the two media and the costs for the additional heat exchangers and plant control system. The Andasol<sup>1</sup> plants each have 1,100 MWh<sub>th</sub> two-tank molten salt storage systems while their primary HTF is synthetic oil.

In both kinds of active systems, it's possible to have solid "filler material" with high thermal capacity inside the storage. This can enhance the overall capacity, improve the thermocline quality in one-tank designs and save costs by substituting an expensive fluid storage medium with cheaper solid particles, e.g. ceramics or rocks.

Gil et al. (2010) defined passive storage systems as systems in which the storage medium doesn't move but is flown through by the HTF (traditionally such reversed flow heat exchangers are called 'regenerators'). The advantages deriving from the layout are a simple design and possibly cheap storage material. Laing et al. (2006) experimented with blocks of concrete, embedding pipes that the HTF passed through. The project mainly aimed at the development of a low cost storage due to the cheap storing medium concrete. However, a packed bed storage built of locally abundant rocks that are virtually free enables an even cheaper design. Such storages also have excellent environmental and safety properties: They can't explode, catch fire (except for insulation material perhaps) or leak toxic substances and almost no CO<sub>2</sub> is released during manufacturing. In these beds, the HTF actually gets in contact with the storage medium but due to the use of rocks and an open air cycle, no degradation, contamination or chemical reaction is to be expected. Although no CSP plant has been built using this storage concept yet, it's widely used in metallurgical and chemical processes, e.g. the Cowper stove in blast furnace processes.

Allen (2010) examined samples of different South African rocks for suitability in large packed bed thermal storages. Namely, he determined density, weighted particle size, specific heat capacity, bed void fraction and thermal cycling resistance of collected samples of sandstone, quartz, black shale, granite and dolerite and conducted pressure drop experiments for packed beds of these. Especially due to the thermal cycling tests he concludes that granite is the most appropriate rock for a high temperature TES application.

The design of the SUNSPOT cycle with ambient air in a primary open Brayton cycle makes a packed bed storage attractive. No expensive liquid, like molten salt or thermal oil, would be necessary, only one heat exchanger (HX) as the boiler of the Rankine cycle is needed and thus a low relative cost can be achieved for a big storage tank. The rock bed storage appears to best fit the requirements by the SUNSPOT storage system and, in compliance with Gauché (2011), has been chosen as the storage for this thesis' model.

<sup>&</sup>lt;sup>1</sup>The Andasol plants were the first commercial parabolic through plants to be build since the SEGS in the 1980's. They are located near Guadix, Spain.

## 2.3 LITERATURE REVIEW ON PACKED BED TES SYSTEMS AND CSP PLANT SIMULATIONS

There has been extended research during the last hundred years on packed beds as a thermal storage and as a chemical reactor/catalyst, too. However, these studies show disagreement in the fundamental dependencies of flow and thermal properties and produce results that sometimes differ in orders of magnitude. In the following, an overview of works on the subject is given.

#### 2.3.1 Heat Transfer

Heat transfer characteristics are the most important quality of a TES system. Charge/Discharge performances have to meet the plant's design so that no energy goes to waste; the same is true for the usable storage capacity. Yet, heat transfer inside a packed bed is a highly complex process especially if non-uniformly shaped and sized rocks form a random packing. The 'classical' analytical description of this problem was made by Schumann (1929). His main assumptions were: (a) High thermal conductivity inside the solids, (b) little conductivity between solids in axial direction<sup>2</sup>, (c) constant fluid properties, (d) no wall heat losses, (e) plug flow and (f) no temperature gradient in radial direction. One further simplification is neglecting the thermal capacitance of the fluid which leads to the Number of Transfer Units (NTU) method as described by Duffie and Beckman (1991). Hughes (1975) introduced the so-called Effectiveness-NTU (or E-NTU) method that simplifies the differential equations of the Schumann model and adds heat losses to the surroundings.

Many studies relax one or more of Schumann's assumptions (see Section 2.3.2). Assumption (a) leads to constant temperatures inside each solid element. This can be relaxed by using a corrected value for the NTU, taking into consideration the time until a surface temperature change reaches the center of a solid, as done by Jeffreson (1972) or Sagara and Nakahara (1991). The latter conducted tests with especially large solid bodies of different shapes.

Obtaining the heat transfer coefficient between solid and fluid is the NTU method's major challenge so there are numerous empirical equations for it. Löf and Hawley (1948), Coutier and Farber (1982) and others found relatively simple dependencies only on the particle size and the air mass flux through experiments. Wakao and Funazkri (1978), Gunn (1978), Nsofor and Adebiyi (2001) and others objected that the heat transfer coefficient mustn't approach zero at low flow speeds, but rather a constant value.

Martin (2005) proposed in his version of the Generalized Lévêque Equation (GLE) a dependency on the pressure drop over the bed. Allen (2010) verified this method with wind tunnel test of different rocks at low temperatures.

Authors of many studies tried to find expressions for 'effective' thermal conductivities in different spatial dimensions. Wen and Ding (2006) built a two-dimensional model assuming plug flow and derived terms for effective thermal conductivities in axial and radial directions through experiments with a tube diameter to particle diameter ratio of N = 8. Wakao (1976) calculated effective thermal conductivities in two dimensions for low Reynolds numbers and considered temperature gradients inside the solids. Chauk and Fan (1998) gave a good overview of published formulas for effective thermal conductivities.

Most authors neglect the effect of radiative heat transfer because their experiments are conducted at relatively low temperatures of less than 150 °C which is sufficient for solar thermal heating of buildings. Schröder, Class, and Krebs (2006) investigated local fluid-to-solid heat transfer at elevated temperatures of up to 700 °C at low to medium Reynolds numbers. According to them the radiative effect is negligible below 500 °C. Chauk and Fan (1998) stated without further explanation that the effect of radiative heat transfer is negligible at temperatures below 600 °C.

 $<sup>^{2}</sup>$ A vertical cylindrical tank is often assumed. Thus, 'axial' is used for the vertical length and 'radial' for the horizontal width of the packed bed.

Stamps and Clark (1986) reported that a one-dimensional simulation software, ROCKBED, has been developed but not much more information could be found. Beasley and Clark (1984) extended this program into a two-dimensional model (axial and radial) in order to account for inhomogeneous fluid flow and compared it to low temperature experiments. A different approach on simulating the several different phenomena occurring in a packed bed is the relatively new field of computational fluid dynamics (CFD). This makes it much easier to calculate three-dimensional problems and account for several forms of heat transfer (conduction, convection and radiation) separately instead of combining them to a less accurate effective conductivity term. Logtenberg, Nijemeisland, and Dixon (1999) conducted simulations for beds with a low tube-to-particle diameter ratio of N = 2.43 and only ten solid objects. Small configurations are especially interesting for CFD analysis because single phenomena have a big influence when the bed consists of only a few particles. Nijemeisland and Dixon (2004) investigated the local wall heat flux in a bed with N = 4 and, somewhat surprisingly, didn't find the wall heat losses to be proportional to the local air velocity. Ookawara et al. (2007) built a CFD model with  $N \leq 8$  but didn't choose a perfect packing of spheres. Instead they used the discrete element method (DEM) that simulates physical forces on the particles while falling into the tube and therefore creates a more realistic packing.

#### 2.3.2 Pressure Drop

The pressure drop in a thermal storage not only determines the power required for charging and discharging it. Heat transfer between fluid and solids as well as the axial and radial temperature distributions are also greatly influenced by it.

The benchmark used by most authors for their pressure drop predictions and experimental results is the Ergun equation. Ergun (1952) formulated the relative pressure drop per unit length as dependent on the sum of a constant term and a term dependent on the inverse Reynolds number in a one-dimensional model. This so-called friction factor is empirical and has been altered many times to fit different authors' measurements. Hollands and Sullivan (1984), for example, changed the factor slightly to match their experiments with rock samples of different diameters between 10 and 19 mm within  $\pm 20$  %. Their measurement range was N = 8...16 and  $Re_p = 125...350$ . Chandra and Willits (1981) kept Ergun's concept of a friction factor but found a different dependency on the bed's void fraction.

In real world packed beds, there will always be radial flows, perpendicular to the main flow direction. Combined with different flow speeds in the main flow direction this creates temperature gradients in the radial dimension. The latter is caused by the uneven contribution of void volume in the bed (called wall effect), friction at the outside walls and uneven flow profiles at in- and outlet. The magnitude of flow uniformity is not being agreed on but most studies concluded that the smaller density of solids next to the wall leads to a much higher superficial fluid velocity there. Hollands, Sullivan, and Shewen (1984) mentioned up to 150 % velocity increase for bigger rocks and up to 70 % increase for smaller samples. The used tank had a relatively big N-value of more than 30. Papageorgiou and Froment (1995) reported oscillating void fractions in the proximity of the walls, that level out within a few particle diameters distance to the walls. They suggested that void area induced velocity inconsistencies increase with higher Reynolds numbers and values of N. Subagyo, Standish, and Brooks (1998) also modeled an oscillating velocity profile with peaks of about 2.4-times as high superficial velocities close to the wall compared to the mean velocity. In contrary to Papageorgiou they calculated a smother profile for higher tube-to-particle ratios. Negrini et al. (1999) conducted experiments with N values ranging from 3 to 60 at different air flow rates and compared it to several formulas from the literature. They found a much smoother overall velocity profile in bigger tanks with the same particle size but still reported of flow speed maxima near the wall, although less distinct than in the smaller tanks. Nsofor and Adebiyi (2001) and Allen (2010) used wall linings to avoid high void volumes near the walls of the tanks. Eisfeld and Schnitzlein (2001) stated that at tube-to-particle ratios of more than 20 the pressure drop at the walls is virtually the same as in a bed with a ratio approaching infinity.

Singh, Saini, and Saini (2006) stated that if small particles are used, the pressure drop has to be increased in order to achieve a uniform flow distribution. They tested particles of different shapes and developed a pressure drop correlation considering the 'sphericity' of the particles. Zunft, Hahn, and Kammel (2011) did CFD simulations to optimize inlet and outlet designs for uniform flow profiles.

#### 2.3.3 CSP Plant Simulations

Optimal size and design of the thermal storage can immensely improve a CSP plant's energetic and economic efficiency. Meier and Winkler (1991) applied the software PACKBEDA (a modified version of PACKBED) on a large storage model for a proposed 5  $MW_{el}$  solar thermal power plant. However, they only modeled the rock bed storage and optimized the design for simple charge/discharge cycles, as did Allen (2010).

Zavattoni et al. (2011) built a 25 m<sup>2</sup> concrete test storage into the ground. They created a CFD model of it and simulated the effects of gravity on the porosity (and consequently temperature) distribution in a rock bed.

Tamme, Laing, and Steinmann (2003) compared different charging and discharging modes and connections of the TES in a CSP plant. Koll et al. (2011) developed concepts for a hybridized combined cycle CSP plant and conducted year-long simulations with hourly time steps. The HTF was air but, unlike in the SUNSPOT cycle, this only drove the boiler of the Rankine cycle and not the gas turbine which used fossil fuel only. The storage was a packed bed of ceramic particles of different sizes.

Grasse (1991) did a feasibility analysis for a 30 MW<sub>el</sub> central receiver plant with a 250 MWh<sub>th</sub> passive TES. He used measured direct normal irradiation (DNI) values and one aim was to sufficiently meet a specific peak demand through dispatching which seems to have been done at reasonable detail. However, the detail at which he simulated the storage thermodynamically is not clear. Luzzi et al. (1999) did economic calculations and design proposals for a 10 MW<sub>el</sub></sub> plant with Ammonia as an HTF and storage medium. Price (2003) developed a software tool to simulate Rankine cycle parabolic solar thermal power plants with or without TES. He validated his tool against hourly DNI and power output data from existing plants and added detailed economic calculations including an optimization algorithm for the Levelized Cost of Energy (LCoE). Sioshansi and Denholm (2010) suggested that CSP plants with big TES will prove more viable if they shift electricity generation to times of peak prices on the market. They ran simulations based on a relatively simple 110 MW<sub>el</sub> CSP plant model and used historic prices from wholesale electricity markets of regions close to potential CSP sites. Different assumptions were made regarding the demand forecast accuracy of the operating companies. They also proposed using such a plant to guarantee reserves for the grid and concluded that such a control mode could prove very viable.

# **3 THE STORAGE MODEL**

The storage type and specific design have to fit the requirements given by the power plant and external factors. In this chapter the TES system is described and the derivation of its thermal model is presented.

### 3.1 STORAGE MEDIUM

The 5 MW<sub>el</sub> pilot plant is supposed to prove that CSP is able to meet South Africa's demand profile in terms of dispatchability and supplying base-load electricity. The SUNSPOT cycle offers a combination of two approaches to the problem: back-up fuel and thermal energy storage. In order to dispatch solar energy in longer times of no irradiation, a very big storage is needed. Because of this, the concept can only be viable if the specific storage costs are very low. Since the HTF is air, a direct active storage system would need to have an enormous volume and therefore is no option. An indirect TES system would require an additional heat exchanger in between the air and a storing fluid with a high boiling temperature to avoid needing a pressure tank. A passive system is the most viable option if the storage medium is cheap and available in big quantities. For this reason experiments have been made on high temperature concrete blocks and packed beds of ceramics but their costs are still believed to be much higher than rock beds.

South Africa has no shortage of rocks and Allen (2010) showed that at least some of them are well suitable for a TES. His thermal cycling tests led to the conclusion that granite from the Northern Cape province is the most suiting for the application. The Northern Cape also has one of the best solar properties in the world and is very sparsely populated. The granite's most important characteristics are given in Table 3.1.

Property	Abbreviation	Unit	Value
Density	$ ho_{s}$	[kg/m <sup>3</sup> ]	2 893
Void fraction	ε	[-]	0.38
Sphericity	$\psi$	[-]	0.54
Specific heat capacity	$C_S$	[J/(kg K)]	840
Equivalent diameter	$d_{ ho}$	[m]	0.0655

|--|

In general, small particles with high thermal conductivity make for a good thermocline (Fricker 2004). A good thermocline enables a high net storage capacities. This is defined by Fricker as

"Starting from an isothermal cold or hot core, and charging or discharging it with a constant air mass flow rate of constant temperature, the net capacity is defined as the total energy stored or extracted until the outlet temperature deviates by 1 % of the nominal temperature difference between hot and cold. To maintain this capacity, storage must be reconditioned to uniform temperature after each cycle. The product of this regeneration is degraded outlet air; in a power plant, it can be used to operate the steam system in part load and for preheating it." The influence of the rock size on the thermocline will be investigated during simulations. The recommended regeneration is not considered necessary in a rather oversized storage.

#### 3.2 PRELIMINARY ESTIMATES

Table 3.2 shows the assumptions and results of a first estimate of the plant's performance. The calculations can be found in APPENDIX A.

Parameter	Unit	Value
P <sub>GC,nom</sub>	[MW <sub>el</sub> ]	5.0
T <sub>f,St,in,max</sub>	[°C]	530
m <sub>f,St,in,max</sub>	[kg/s]	20.6
m <sub>f,St,out,max</sub>	[kg/s]	30.3
D <sub>min</sub>	[m]	7
$Re_{p,max}$	[-]	1 830

Table 3.2: Estimated values for some parameters

#### 3.3 TANK GEOMETRY

The tank geometry used by almost all proposed or built CSP TES systems is cylindrical to ensure homogenous flow properties and uniform mechanical stress distribution. Although information about design details is scarce this geometry will be assumed in this thesis' model as well. To avoid degradation by natural convection, the temperature inside the storage must decrease towards the bottom. Otherwise hotter fluid with a lower density will flow upwards and vice versa. This eventually lowers the maximum temperature inside the tank and therefore its usable energy (energy). The most effective way of ensuring an effective thermocline is by using a vertical tank that is charged from the top and discharged from the bottom (as recommended by Stamps and Clark (1986)). Degradation can still occur, e.g. when the highest layers cool down to a temperature lower than the ones beneath due to insufficient insulation. Of course, this heat loss not only effects the storage's exergy but also its energy losses. The hot ducts should be as short as possible and lined with insulation material, as should the storage itself. Zunft et al. (2011) suggested that a flap valve in pipes at the tank's top could decrease losses. The investment in insulation is generally small and it's being used in many large scale tank applications.

The air compressor for discharging in the described plant design should be installed at the cold side, namely at the storage's inlet. This not only increases its life expectancy but also decreases the necessary pumping power, which is being calculated according to Equation 3.1.

$$P_{fan} = \frac{\dot{V}_f \Delta p}{\eta_{fan}} \tag{3.1}$$

The problem of uneven flow distribution in proximity of the walls has been addressed by many authors (see Section 2.3.2). Higher flow speeds near the walls lead to central sections of the

storage not being used. There's agreement in the literature, e.g. in (Papageorgiou and Froment 1995) or (Negrini et al. 1999), that this is caused by a lower density of solids in the packing near the wall. Most of the experiments were conducted at low tube-to-particle ratios, however. Meier and Winkler (1991) and Eisfeld and Schnitzlein (2001) consider the wall effect negligible at ratios higher than 40 or 20, respectively. A resulting storage diameter of more than 2.6 m would be needed for the proposed amount of energy anyway. Nsofor and Adebiyi (2001) and Allen (2010) proposed the use of wall linings (ceramic fiber sheets) to create a smaller void fraction by 'embedding' the solids. This method is useful in experimental sized stores and might be advantageous in utility sized tanks as well. Besides minimizing the void volumes, they also thermally insulate and could even reduce mechanical stresses. These stresses, induced by different thermal expansion coefficients of rocks and the tank itself, are one of the very critical questions regarding the use of packed bed TES systems. They were investigated e.g. by Dreiß igacker, Müller-Steinhagen, and Zunft (2010) but are beyond the scope of this thesis. In- and outlet of the storage also play an important role in flow distribution. Zunft, Hahn, and Kammel (2011) carried out CFD analyses of a packed bed storage with different shapes of domes. They decided that a cone-shaped and a hemispherical dome are the best solutions for in- and outlet, respectively. According to their study, the flow uniformity mainly depends on the pressure drop inside the bed, the tank's diameter, the inlet pipes diameter and the cone's height. In order to achieve uniform flow, the cone has to have a very steep slope and therefore be considerably high and the inlet pipe should be wide. There will be a trade-off between these flow improving methods and capital costs for the tank system. Similarly, a high pressure drop helps in achieving uniform flow but leads to high pumping costs. Advanced flow distribution systems are also an option but might not suit the concept of a low cost system. For now, sufficient uniformity is assumed which will have to be proven once the pressure drop is calculated for a fixed and detailed design.

#### 3.4 THERMAL CORRELATIONS

The first decision in building the thermal model of the TES was how many spatial dimensions to consider. The early models, like Schumann's (1929), were one dimensional. Later on, a radial component was added to account for radial flows and temperature gradients. A third dimension is usually not necessary as long as a cylindrical tank is used. As mentioned in Section 3.3 void fraction differences and flow uniformity can be neglected in vessels with very large tube-to-particle ratios. Radial temperature differences due to wall-heat losses will assumed to not play a major role due to insulation and standstill times that usually don't exceed 10 hours unless the storage is empty.

#### 3.4.1 Charging and Discharging Modes

The used model will be based on the Schumann equations in one-dimensional form and the Effectiveness-NTU method proposed by Hughes (1975) for charging and discharging modes. This means neglecting the thermal capacity of the fluid. Due to the extreme density difference between fluid and solid, this is justifiable. Equation (3.2) also contains a term for thermal losses to the surroundings. The derivation of it can be found in Section 3.4.3.

$$T_{f,i+1} = T_{f,i} - (T_{f,i} - T_{s,i}) \left( \underbrace{1 - e^{-NTU_i \frac{\Delta x}{L}}}_{\eta_{NTU_i} \frac{\Delta x}{L}} \right) - \frac{\dot{Q}_{loss,i}}{\dot{m}_f c_{p,f,i}}$$
(3.2)

With this equation the temperature of the air entering the next section (i + 1th) of the bed can be calculated, assuming an exponential temperature distribution inside each section. Having done this, the rock temperature at the next *time*-step j+1 can be found by applying energy

conservation formulas (see APPENDIX B). The time constant  $\tau_{NTU}$  (Equation (3.4)) has been substituted into Equation (3.3) for simplification purposes.

$$T_{s,ij+1} = \frac{T_{s,ij} \left( 1 - \frac{1}{2} \left[ \frac{\Delta \tau L \eta_{NTU,ij}}{\Delta x \tau_{NTU,ij}} \right] \right) + T_{f,ij} \left[ \frac{\Delta \tau L \eta_{NTU,ij}}{\Delta x \tau_{NTU,ij}} \right]}{1 + \frac{1}{2} \left[ \frac{\Delta \tau L \eta_{NTU,ij}}{\Delta x \tau_{NTU,ij}} \right]}$$
(3.3)

$$\tau_{NTU} = \frac{\rho_s c_s (1 - \varepsilon) A_{cs} L}{\dot{m}_f c_{p,f}} = \frac{m_s c_s}{\dot{m}_f c_{p,f}}$$
(3.4)

Hughes (1975) suggested using the corrected NTU term  $NTU_c$  to account for finite thermal conductivity inside the solids, as done by Jeffreson (1972). This correction addresses the problem that big particles will have a temperature gradient between the center and the outside when in contact with a fluid of a different temperature because the heat doesn't conduct instantaneously. Jeffreson used Equation (3.5) which features the fluid particle Biot number for heat transfer of Equation (3.6).

$$NTU_c = \frac{NTU}{1 + \frac{Bi}{5}}$$
(3.5)

$$Bi = \frac{\alpha \, d_p}{2\lambda_s} \tag{3.6}$$

Sagara and Nakahara (1991) used a different correction term. Their 'apparent' NTU term  $NTU^*$  depends on the beds void fraction. They derived Equation (3.7) by experimenting with big sized particles of different shapes and materials.

$$NTU^* = NTU \frac{20}{3\frac{\alpha_{\nu}d_{\rho}^2}{4\lambda_{f}(1-\varepsilon)} + 20}$$
(3.7)

Because the latter agree well with the potential use in the SUNSPOT cycle, Sagara and Nakahara's correction factor for the *NTU* is used in the model. Therefore, in Equation (3.2) *NTU\** has to be inserted instead of *NTU*. However, the standard *NTU* value is still needed to calculate the apparent *NTU*. It can be calculated by Equation (3.8), as demonstrated for example by Hughes (1975).

$$NTU = \frac{\alpha_v L}{G c_{p,f}}$$
(3.8)

The volumetric heat transfer coefficient  $\alpha_v$  is equal to the areal heat transfer coefficient  $\alpha$  multiplied by the specific surface area of the bed per unit volume *a*. Equation (3.9) is technically only valid for spheres and cubes but is sufficiently accurate for other shapes as well since the calculations are already based on equivalent diameters.

$$\alpha_{\nu} = \alpha \, a = \alpha \frac{6 \, (1 - \varepsilon)}{d_{\rho}} \tag{3.9}$$

Finally, finding this areal heat transfer coefficient is probably the biggest challenge in modeling a packed bed thermal storage. There are dozens of different approaches to the topic and most of them seem to disagree with each other. The previous assumptions regarding plug flow, no radial temperature gradient and no axial dispersion limit the necessary modeling dimensions to one which makes the calculation much easier. The latter can be justified because axial dispersion only plays a major rule at low Reynolds numbers and flow speeds. During regular charging and discharging it is not an issue, as reported by Shen, Kaguei, and Wakao (1981) and Bauer (2001).

It might has to be considered in the model for the idle mode. Wall heat transfer was added separately as well (see Equation (3.2)) and is expected to not play a big role in overall heat transfer. The same applies for natural convection, i.e. heat transfer through fluid flow induced by density differences. This kind of convection can only occur when there's no forced flow and even then only when there are colder layers of fluid on top of warmer layers. According to Stamps and Clark (1986) this is an issue for thermal storages that get loaded at inconsistent temperatures (like low temperature solar TES systems). However, storages that get loaded at constant (or rising) temperatures and unloaded in reversed flow direction are "inherently stable". Having ruled out many not well comprehended effects, the simpler questions of a one-dimensional plug flow has to be solved. The usual approach is to either use a correlation for the Nusselt number of the flow system and calculate the heat transfer coefficient from it by use of Equation (3.10) or use a direct expression for a (effective) heat transfer coefficient.

$$Nu = \frac{\alpha d_p}{\lambda_f} \tag{3.10}$$

These expressions usually feature dependencies on one or several of the following parameters (e.g. by Chauk and Fan (1998), Gunn and De Souza (1974), Gunn (1978), Wakao and Funazkri (1978), Chandra and Willits (1981)):

- fluid mass flux G
- particle diameter d<sub>p</sub>
- particle Reynolds number Rep
- fluid Prandtl number  $Pr = \frac{c_{p,f}\mu_f}{\lambda_f}$
- bed void fraction  $\varepsilon$
- particle sphericity  $\psi$  by Singh, Saini, and Saini (2006)

Martin (2005) used the "Generalized Lévêque Equation" (GLE) to calculate heat transfer in external flows (single sphere or pipe) and internal flows (packed beds) dependent on their pressure drops. His Equation (3.11) also includes a so-called "friction fraction"  $x_f$  that accounts for particle shapes that differ from the sphere.

$$\frac{Nu}{Pr^{1/3}} = 0.4038 \left[ 2x_f Hg \frac{d_h}{L_f} \right]^{1/3}$$
(3.11)

The Hagen number Hg represents the pressure drop and is calculated according to Allen (2010):

$$Hg = \rho_f \frac{\Delta \rho}{\Delta x} \frac{d_\rho^3}{\mu_f^2} \tag{3.12}$$

 $d_h/L_f$  is the ratio of the hydraulic diameter and the distance between two "equivalent spheres". For a bed of spheres, Martin defines it as follows:

$$\frac{d_h}{L_f} = \frac{2}{3} \frac{\varepsilon}{(1-\varepsilon)^{2/3}} \tag{3.13}$$

Martin compared Equation (3.11) to 2,646 data points from 41 sources, with Reynolds number between 1 and 10<sup>12</sup> and states that heat transfer characteristics are represented correctly. The calculated root mean square for GLE predictions of packed beds is favourable to state of the art standard methodologies.

#### 3.4.2 Idling Mode

Heat losses and temperature changes are calculated in small time steps and not as a static loss. Thermal equilibrium between solid and fluid is assumed at the beginning of every idling mode. The starting temperatures are calculated according to energy conservation and can be found in APPENDIX C.

During idling mode there's no fluid flow to the outside, so the only energy losses are via conduction to the surroundings. For all sections except the most top and bottom ones where losses through the air ducts have to be added this is simply the wall heat loss (see Section 3.4.3).

These energy losses are accompanied by, and sometimes cause of, exergetic losses through destratification, i.e. temperature equalization inside the storage and therefore degradation of the thermocline. Although Sullivan, Hollands, and Shewen (1984) found that "natural convection is not a major contributor to the destratification in a rock bed,[...]" this might be caused by their limited experiments at low temperatures and it has to be shown in the simulation that no 'upside-down' thermoclines are created by big heat losses in the top-most regions. The model will therefore not feature free convection at first. The validity will be checked in Section 4.2.2. Two different correlations are implemented and compared to experiments in Section 4.2.2. The first correlations are derived from Mohamad, Ramadhyani, and Viskanta (1994) and feature a collective effective conductivity term  $\lambda_{eff}$  for conduction  $\lambda_{con}$ , dispersion  $\lambda_{dis}$  and radiation  $\lambda_{rad}$  (Equations (3.14) - (3.17)). Equation (3.16) is set to zero because of the assumption of no fluid flow. The heat flow rate at a given time *j* from layer *i* to layer *i* + 1 is then calculated by Fourie's law in one-dimensional form (Equation (3.18)).

$$\lambda_{eff} = \lambda_{con} + \lambda_{dis} + \lambda_{rad} \tag{3.14}$$

$$\lambda_{con} = \frac{2\lambda_f}{1 - \frac{\lambda_f}{\lambda_s}} \left( \frac{\ln \frac{\lambda_s}{\lambda_f}}{1 - \frac{\lambda_f}{\lambda_s}} - 1 \right)$$
(3.15)

$$\lambda_{dis} = 0.0895 Pr \, Re_p \, \lambda_f = 0 \tag{3.16}$$

$$\lambda_{rad} = 0.707\lambda_f \left[\frac{\lambda_s}{\lambda_f}\right]^{1.11} \left[\frac{4\sigma T_{m,i-i+1}}{2\frac{1}{\varepsilon_s - 1} + \frac{1}{F_{i-i+1}}}\frac{d_p}{\lambda_s}\right]^{0.96}$$
(3.17)

$$\dot{O}_{destrat,i} = -\lambda_{eff} A_{cs} (T_{i+1} - T_i)$$
(3.18)

Where  $\sigma = 5.67 \cdot 10^8 \frac{W}{m^2 K^4}$  is the Stefan-Boltzmann constant and  $\varepsilon_s$  is the emissivity of the solid that's approximated to be 0.5. Exact information couldn't be retrieved on this parameter but a sensitivity analysis proved negligible consequences of false estimation.  $F_{i-i+1}$  is the radiation view factor between the layers. It will be taken as equal to 0.5 which means that half of each layer's surface faces the next layer towards the (cooler) bottom and the other half towards the (warmer) top. The area towards the wall is neglected since thin layers are assumed. The mean temperature  $T_m$  of the layers is calculated arithmetically by Equation (3.19).

$$T_{m,i-i+1} = \frac{T_{f,i+1} + T_{f,i}}{2}$$
(3.19)

Radiation effects are often not accounted for in thermal equations describing packed beds. This is mainly due to the relatively low temperatures of many applications and the big values for convective heat transfer so that neglecting radiation seems to lead to marginal errors. During
standstill on the other hand, when there's no forced convection, radiation could play a role in a high temperature TES. Chauk and Fan (1998) state without further explanation that radiative effects are 'significant' above 600 °C. Schröder, Class, and Krebs (2006) found in their tests that "Above 300 °C the consideration of radiation in beds of porous slate is useful." At another place, they state that radiation is negligible below 500 °C.

Tsotsas and Martin (1987) compared many approaches to calculating destratification in a packed bed and recommended a model which features the following correlations

(Equations (3.20)-(3.23)) for the effective thermal conductivity without radiative terms:

$$\lambda_{eff} = \left(1 - \sqrt{1 - \varepsilon} + \sqrt{1 - \varepsilon} \,\frac{\lambda_{con}}{\lambda_f}\right) \lambda_f \tag{3.20}$$

$$\lambda_{con} = \frac{2\lambda_f}{N_{destrat}} \left( \frac{B}{N_{destrat}^2} \frac{\frac{\lambda_s}{\lambda_f} - 1}{\lambda_s} \ln\left[\frac{\lambda_s}{\lambda_f B}\right] - \frac{B+1}{2} - \frac{B-1}{N_{destrat}} \right)$$
(3.21)

$$N_{destrat} = 1 - \frac{B\lambda_f}{\lambda_s}$$
(3.22)

$$B = 1.25 \left(\frac{1-\varepsilon}{\varepsilon}\right)^{10/9} \tag{3.23}$$

#### 3.4.3 Wall Heat Losses

Wall heat losses mustn't play a major rule in utility sized TES systems for them to remain economical. Between the energy loss and the exergy loss of the storage, the latter is more critical because a TES that powers a Rankine cycle has to supply heat above a certain threshold temperature. If the store's insulation is not good enough to keep the elevated temperatures, the remaining energy is worthless anergy.

For the evaluation of the local wall heat losses, Fourier's law in one-dimensional form featuring overall resistance *R* is used. Equation (3.24) additionally includes the simplification of a constant wall perimeter between inside and outside which is justified because of the small thickness compared to the diameter. The resistance is calculated by adding the reciprocates of the involved heat transfer coefficients (as explained by Mills (1999)): (a) Heat transfer from the bed/fluid to the wall. This is mostly driven by convection in charging/discharging mode and driven by conduction and radiation in idling mode (if no free convection is assumed). (b) Conduction within the containment. It's mostly determined by quality and thickness of the insulation. The actual (steel) structure and the wall lining (if applicable) will most likely not play a major role. (c) Heat transfer to the surrounding. All three transfer mechanisms can play a role here but in the case of a free standing tank conduction is neglected and in the case of a ground-buried storage, convection can be neglected. Unless the insulation fails, the containment's outer temperature shouldn't cause any measurable radiative heat losses, additionally TES tanks in use at CSP plants have a highly reflective surface. These assumptions lead to Equation (3.25) for the containment's heat resistance.

$$\dot{Q}_{wl,i} = \frac{1}{R_i} A_w (T_{f,i} - T_{amb}) = \frac{1}{R_i} \pi D \Delta x (T_{f,i} - T_{amb})$$
(3.24)

$$R_{i} = \frac{1}{\alpha_{w,ins,i}} + \frac{\Delta r_{lining}}{\lambda_{lining,i}} + \frac{\Delta r_{tank}}{\lambda_{tank,i}} + \frac{\Delta r_{insul}}{\lambda_{insul}} + \frac{1}{\alpha_{w,outs,i}}$$
(3.25)

The correlations for temperature depending heat transfer coefficients of linings and insulations can be found in APPENDIX D. The inner heat transfer coefficient can be calculated from the wall

Nusselt number  $Nu_w$  using Equation (3.10).  $Nu_w$  is best divided into two parts (see Equation (3.26)), as done by Bauer (2001).

$$Nu_{w} = Nu_{w,0} + Nu_{w,t}$$
(3.26)

 $Nu_{w,0}$  is applicable for stagnant flow, i.e. in idle mode without natural convection. According to Bauer the use of it is subject to discussions and it's values in the literature differ in the wide range of  $0 \le Nu_{w,0} \le 80$ . Bey and Eigenberger (2001) found  $\lim_{Re_p\to 0} Nu_{w,0} = 20$  in their measurements with spheres at  $N \approx 8$ . This value was chosen for the model as well (Equation (3.27)).

$$Nu_{w,0} = 20$$
 (3.27)

 $Nu_{w,t}$  is the turbulent wall Nusselt number and often given in dependency of  $Re_p$  and Pr, e.g. by Bey and Eigenberger (2001) or Dixon and Labua (1985). Hennecke and Schlünder (1973) found a much more complex correlation which according to them shows better agreement to measuring results but this was for small N values. In this study Equation (3.28) is used in charging/discharging mode. It derives from Dixon and Labua's findings, see APPENDIX D.

$$Nu_{w,t} = Re_p^{0.61} Pr^{1/3} ag{3.28}$$

The heat transfer coefficient on the outside of the tank should be calculated dependent on wind speeds if they're part of the weather data. In this study,  $\alpha_{w,outs}$  is assumed to be constant at 5 W/(m<sup>2</sup> K) as done by Stamps and Clark (1992) as the influence of the coefficient is believed to not be drastic.

#### 3.4.4 Storage System Efficiencies

An ideal storage stays for unlimited time in exactly the same state and is able to deliver the same quantity and quality of energy that's been given to it. Such an ideal storage doesn't exist and in the real world losses have to be taken into account. The three cycles (charging, discharging and idling) are accounted for in separate equations. Each with one equation for first law and one collective equation for the yearly averaged second law efficiency (Equations (3.29)-(3.32)).

$$\eta_{c,j} = \frac{\frac{m_s}{n} c_s \sum_{i=1}^{n} (T_{s,i,j+1} - T_{s,i,j}) + \sum_{i=1}^{n} \left( m_{f,i,j} c_{\rho,f,i,j} (T_{f,i,j+1} - T_{f,i,j}) \right)}{(h_{f,c,i,n,j} - h_{amb,j}) \dot{m}_f \Delta \tau}$$
(3.29)

$$\eta_{dc,j} = \frac{(h_{f,dc,out,j} - h_{f,dc,in,j})\dot{m}_{f}\Delta\tau}{\frac{m_{s}}{n}c_{s}\sum_{i=1}^{n}(T_{s,i,j+1} - T_{s,i,j}) + \sum_{i=1}^{n}\left(m_{f,i,j}c_{p,f,i,j}(T_{f,i,j+1} - T_{f,i,j})\right)}$$
(3.30)

$$\eta_{id,j} = \frac{\frac{m_s}{n} c_s \sum_{i=1}^{n} T_{s,i,j+1} + \sum_{i=1}^{n} \left( m_{f,i,j} c_{p,f,i,j} T_{f,i,j+1} \right)}{\frac{m_s}{n} c_s \sum_{i=1}^{n} T_{s,i,j} + \sum_{i=1}^{n} \left( m_{f,i,j} c_{p,f,i,j} T_{f,i,j} \right)}$$
(3.31)

$$\eta_{ex,St,a} = \frac{E_{f,St,out,a}}{E_{f,St,in,a}}$$
(3.32)

Theses equations are derived from the definition of efficiencies. The only energetic losses are wall losses (see Section 3.4.3) and exiting hot air in charging mode that can't be used in the bed because of the storage being too hot already or the heat transfer being to small. The energy gained or lost by the fluid can not simply be calculated from a specific heat capacity  $c_{p,f}$  because this coefficient is influenced by the temperature changes. Hence, the specific enthalpy  $h_f$  is used here and calculated in APPENDIX E. The difference between the enthalpy difference approach and adding the specific heat capacities of all layers together has been calculated and found negligible.

The exergetic efficiency is equal to the quality and quantity of the energy taken out of the storage divided by the quality and quantity of energy input, as demonstrated by Rosen and Dincer (2003). In both cases, the carrier for the thermal energy is air, the energy consumption of the fan is added on the input side. The storage medium's exergy is not taken into account. In the course of a full year, this should be negligible. A more detailed explanation can be found in APPENDIX E.

#### 3.5 PRESSURE DROP CORRELATIONS

The following section shows formulae used in the literature for predicting the pressure drop as well as derivation and legitimation of the ones that were used for the model. As mentioned in Section 2.3.2 the classical equation for pressure drop prediction derives from Ergun (1952). The Ergun equation (3.33) is an attempt to combine linear and non-linear pressure drop dependencies on fluid velocities at low and high Reynolds numbers, respectively. The strong influence of beds void fractions is apparent. Ergun uses the equation in the particle Reynolds number range of approximately 0 to 3 000.

$$-\frac{\Delta \rho}{L} = 150 \frac{(1-\varepsilon)^2}{\varepsilon^3} \frac{\mu_f v_f}{D^2} + 1.75 \frac{1-\varepsilon}{\varepsilon^3} \frac{\rho_f v_f^2}{D}$$
(3.33)

Chandra and Willits (1981) fitted their data with a similar expression. In Equation (3.34) the dependency on the bed's void fraction and other empirical values have been changed as compared to Equation (3.33). They give the validity range of their equation to be  $0.33 < \varepsilon < 0.46$  and  $1 < Re_p < 1000$ .

$$\frac{\Delta p \rho_f d_p^3}{\mu_f^2} = \varepsilon^{-2.6} (185 \, Re_p + 1.75 \, Re_p^2) \tag{3.34}$$

Chandra and Willits called the left hand side of this equation "pressure drop number". The right hand side is the friction factor  $f_s$ .

Singh, Saini, and Saini (2006) experimented with different shapes of particles and tried to find a new formula also taking into account the sphericity  $\psi$  of the bed material. Equation (3.35) is empirical and was developed using experimental data and comparing it to existing correlations.

$$\frac{\Delta \rho \rho_f d_p}{LG^2} = 4.466 R e_p^{-0.2} \psi^{0.696} \varepsilon^{-2.945} e^{11.85(\log_{10}\psi)^2}$$
(3.35)

The experiments to acquire this correlation were conducted within the following limits:

- 0.306 < ε < 0.63
- $0.55 < \psi < 1.00$
- $1\,000 < Re_p < 3\,000$

They found that Nusselt number and friction factor have a minimum at a sphericity of approximately 0.80. Both increase when the sphericity is increased towards spheres ( $\psi$ =1) or highly non-spherical particles ( $\psi$  = 0.55). The given explanation is that the available surface for heat transfer decreases when the shape has more flat parts. On the other hand, when the solids are very 'edgy' they create turbulances which increases friction and heat transfer. Compared to other works, they find deviations of less than ± 25 % in their results. Allen (2010) found this equation to best fit his experimental data of packed beds of South African rocks compared to other correlations. A sphericity of  $\psi$ =0.54 was found to represents the friction factor curves most accurately. Since Allen's work actually used the same kind and sizes of rocks that are now proposed for the use in the SUNSPOT cycle, it is assumed that Singh, Saini, and Saini's (2006) equation will also represent the pilot storage.

# 4 VALIDATION OF THE STORAGE MODEL

The validation of this thesis' simulation model is especially critical because of the many assumptions and diverging opinion in many fields of pressure drop and heat transfer correlations. In the following, the model will be applied on experimental setups and its outcomes compared to empirical data.

## 4.1 PRESSURE DROP

Since the pressure drop is the basis for the chosen heat transfer correlations, its validation has to be executed first. The first comparison made is rather a verification than a validation. The graph generated by the model is compared to a graph of Singh, Saini, and Saini's (2006) formula and some experimental values, extracted from their work. The two lines in Figure 4.1 representing the friction factor as a function of the particle Reynolds number are exactly the same and lay close to the empirical data. But it should be noted that even this verification shows a mean error of approximately 8 % to the actual measuring result. The main parameters of this and the following authors' experiments and/or the stated validity range of their correlations are given in Table 4.1. All particles (except for Allen's (2010) rocks) are spheres and therefore given a sphericity of  $\psi$ =1 and a friction fraction of  $x_f$ =0.45.

Table 4.1: Parameters of pressure drop validation experiments	
(* equivalent diameter for rectangular bed)	

Study	<i>d</i> <sub>p</sub> [m]	<i>D</i> [m]	<i>L</i> [m]	Rep	ε
Singh, Saini, and Saini (2006)	0.136	0.6	0.7	10002000	0.4
Allen (2010)	0.0655	0.25*	0.5	1 000 7 000	0.395
Hollands and Sullivan (1984)	0.0105	0.46	1.8	40100	0.39
Chandra and Willits (1981)	0.0163	0.35*	0.4	11000	0.38

Allen advocated the same equation because it best fitted his experimental results in a rock bed wind tunnel test. His plots for an inverse dimensionless pressure drop are well reproduced by the model in Figure 4.2.



Figure 4.1: Verification of pressure drop correlations with Singh, Saini, and Saini's (2006) data



Figure 4.2: Validation of pressure drop correlations with Allen's (2010) data



Figure 4.3: Validation of pressure drop correlations with Hollands and Sullivan's (1984) data

The experimental data of Hollands and Sullivan (1984) does not show the same tendency as the model's prediction. However, the predicted values are in the same order of magnitude in the small range applied (see Figure 4.3).

Figure 4.4 shows the calculated pressure drop per unit length over  $Re_p$  as calculated by Equation (3.34) of Chandra and Willits (1981) and by this model. The curves agree well up to  $Re_p = 150$  and then start diverging more strongly. Chandra and Willit's equation is applicable up to  $Re_p = 1000$  in which it matches *their* experimental data very well. The maximum error of this study's model at this point is less than 25 %. A possible explanation is that at their N value, wall channeling has a more dominant effect due to the relatively large area with big void volumes. Allen's (2010) experiments without wall linings show pressure drops 35 to 45 % lower than with them installed.



Figure 4.4: Validation of pressure drop correlations with Chandra and Willits's (1981) data

## 4.2 HEAT TRANSFER

Some of the many - often only slightly - differing correlations in the literature for Nusselt numbers over particle Reynolds numbers are compared to the present model in Figure 4.5. All of them were derived from experimental data by the authors and compared favorably. Chandra and Willits (1981) e.g. stated a mean deviation of only 3.4 % in their best fit curve. The input values for the plot model and calculations are taken from Coutier and Farber (1982) and given in Table 4.2. They lie in the stated areas of validity.



Figure 4.5: Validation of heat transfer correlations with Coutier and Farber (1982), Ranz and Marshall (1952) and Wakao and Funazkri's (1978) data

<i>d</i> <sub>p</sub> [m]	<i>D</i> [m]	<i>L</i> [m]	<i>Re</i> <sub>p</sub> [-]	ε[-]	ψ [-]
0.0177	0.57	0.84	150 900	0.4	1.0

Table 4.2: Parameters of heat transfer validation experiments

The curves agree quite well in this area of Reynolds numbers that's also technically realistic. There are experimental results in other sources that show completely different outcomes (e.g. Singh, Saini, and Saini (2006)). The correlations in (Sagara and Nakahara 1991) even seem to be faulty since they don't match the graphs that they're supposedly derived from. Now that the fundamental heat transfer equations are validated, the next step is to compare temperature changes in experimental setups to the simulated reproduction.

#### 4.2.1 Charging and Discharging

Many temperature curves for charging or discharging a packed bed can be found in literature. Unfortunately, almost all of them lack some information needed for reproduction. For this reason, validation for example with data from Coutier and Farber (1982), Sorour (1988), Beasley and Clark (1984) or Jones and Golshekan (1989) was not possible. The experimental parameters of the three studies that contain complete information are given in Table 4.3.

Meier and Winkler (1991)+	0.10	14	25	3 700	0.36	1030	2 800	2.50	550	25
Meier and Winkler (1991)	0.02	0.15	1.2	220	0.40	1030	2 800	2.50	550	25
Sullivan, Hollands, and Shewen (1984)	0.0181	2.031*	ω	80 / 110	0.41	920	2 711	0.49	62	22
Allen (2010)	0.0655	0.25*	0.5	1 000	0.395	845	2 893	3.00	70	25
eter Unit	[m]	[ɯ]	[m]	Ξ	Ξ	[J/(kg K)]	[kg/m <sup>3</sup> ]	[W/(m K)]	[°C]	[0°]
Param	$d_p$	D	L	$Re_p$	ω	$c_{\rm s}$	$\rho_{s}$	$\lambda_s$	t <sub>in</sub>	$t_{out}$

# Table 4.3: Parameters of charging/discharging experiments (\* equivalent diameter for rectangular bed) (<sup>+</sup> calculated)



Figure 4.6: Validation of charging cycle simulation. Plot by Allen (2010) with own data

Once again, Allen (2010) used very similar correlations to the ones the model is based on for his calculations. Therefore, it's not surprising that his curves for the fluid temperature inside the bed are very closely approximated by this model in Figure 4.6.

Sullivan, Hollands, and Shewen (1984) conducted experiments on a large-size rock storage of 5.8 m<sup>3</sup>. The shape of the thermocline in Figures 4.7 and 4.8 is reproduces well for several hours but the storage appears to be more filled in the model. This is true for both fluid mass fluxes examined. The reason for this might lie in not very high precision at the low temperatures the experiments have been conducted at. Noticeably, the higher temperatures at the top of the tank agree the least. This could be due to heat losses in the experiment that haven't been accounted for to the right extend in the model. Sullivan, Hollands, and Shewen (1984) also states that the temperature difference in one layer can be up to 7 K. The shown values of their study are either averaged over a layer or central temperatures.

Lastly, Meier and Winkler's (1991) setup featured a small tube-to-particle ratio but elevated temperatures. The simulation predicts a better stratification than the actual measurements do (see Figure 4.9). At an *N* value of only 7, the wall effect could have a noticeable effect on the outcome of the experiments. Also, Meier and Winkler state that the thermal losses through the wall are 'considerable' but could be reproduced well by their model. Due to a lack of detailed information on their insulation, these losses can only be approximated very roughly in this thesis' model.

They also ran a simulation of a utility-size TES for a 30  $MW_{el}$  CSP plant. Compared to this, the model shows more conservative temperatures but again a better thermocline (see Figure 4.10). Nonetheless, the overall agreement even after a simulation time of 8 h is sufficient. The losses they calculate for the big storage are probably much smaller because of the decreased surface-area-to-volume ratio of the tank.

To sum up, the model is far from exactly reproducing every measured charging or discharging temperature curve. This seems impossible anyway, because the proposed correlations in the literature usually can't describe the experiments in the same paper within a margin of  $\pm 10$  %. When applied on other author's experimental outcome, the agreement is almost always very poor. Keeping this in mind, the accuracy of the analyzed model is satisfactory.



Figure 4.7: Validation of charging cycle simulation. Plot by Sullivan, Hollands, and Shewen (1984) with own data



Figure 4.8: Validation of charging cycle simulation. Plot by Sullivan, Hollands, and Shewen (1984) with own data



Figure 4.9: Validation of charging cycle simulation. Plot with empirical data points by Meier and Winkler (1991) with own data



Figure 4.10: Validation of charging cycle simulation. Plot with simulated temperature curves by Meier and Winkler (1991) with own data

#### 4.2.2 Wall Heat Losses and Destratification

Detailed experiments for wall heat losses in big packed bed storages could not be found in the literature. In the following, the overall heat loss will be compared to the few sources found and the heat transfer from the bed to the wall will be checked in more detail. There have been some experiments on destratification of packed beds but they focused on solar heater applications where the storages are charged at inconsistent temperatures. This, of course, creates an unstable thermocline due to natural convection (see Section 3.4.2) Most wall heat transfer experiments have been conducted in very small apparatus. Bey and Eigenberger (2001) ran tests on beds of ceramic spheres up to an N-value of 13. Since no details about the thermal conductivity of the used ceramic spheres are given, a value of  $\lambda_s$  = 1.2 W/(m<sup>2</sup> K) is assumed as used by Logtenberg (1998) for his ceramic particles. This leads to the Nusselt number over Reynolds number correlations seen in Figure 4.11. The no-flow Nusselt number  $Nu_{w0}$  of approximately 20 fits the data well and the slope of the lines at slightly higher velocities is similar. However, the calculated heat transfer is higher than measured. The model should be more correct for the intended purposes at high tube-to-particle diameter ratios (which is assumed in the derivation of the turbulent Nusselt number, see APPENDIX D). Dixon, DiCostanzo, and Soucy (1984) supported the view that higher wall Nusselt numbers are

observed at these higher ratios when keeping the Reynolds number constant (see Figure 4.12). Besides, the wall heat loss is not supposed to play a major role. Given sufficient insulation these differences in inner wall heat transfers should not be significant.



Figure 4.11: Validation of wall Nusselt number correlations. Plot with simulated and experimental outcome by Bey and Eigenberger (2001) with own data

Heat losses at the top of the tank depend on the design of the TES. HTF-based tanks in use at CSP plants usually have almost flat covers with similar insulation to the walls. Because of this, the heat transfer correlations should be similar as well except for the different flow characteristics in an almost horizontal plane. A tank charged by gas, on the other hand, needs a big duct which should slowly expand the air as described in Section 3.3. For calculating the losses of this duct the surface of the design will be estimated and the same heat transfer characteristics are assumed. This is not correct because of the different flow properties but it can only be calculated correctly in a later stage when the TES design is looked at more closely. The heat losses towards the bottom of the tank are negligible unless the bottom temperature rises significantly. The same uncertainty as with the top duct applies to the bottom design. Top and bottom losses will be added to the wall heat losses of the top and bottom layer, respectively.



Figure 4.12: Nusselt number dependency on value of *N*. Plot by Dixon, DiCostanzo, and Soucy (1984).  $d_t = D$ 

Both investigated correlations for destratification in a packed bed are compared to the experiments of Jones and Golshekan (1989) in Figures 4.13 and 4.14. Mohamad, Ramadhyani, and Viskanta's (1994) correlation can't correctly reproduce the temperature changes inside the bed, Tsotsas and Martin's (1987) recommended approach is much closer to the experiments. Simply for comparison, a best fit for the curves has been used and  $\lambda_{eff} = 25 \text{ W/(m}^2 \text{ K})$  has been found to be the closest (see Table 4.5). The differences in temperature curves in Figure 4.15 are mainly in the lower layers of the storage which get heated up more in the experiment then according to the simulation. Since the TES in a CSP plant will only supply heat over a certain threshold temperature, this region will most likely not be of importance.

|--|

Experiment	Correlation	$\lambda_{eff} \left[ \frac{W}{m^2 K} \right]$
Jones and Golshekan (1989)	Mohamad, Ramadhyani, and Viskanta (1994)	0.13
	Tsotsas and Martin (1987)	2.6
	best fit	25
Zunft et al. (2011)	Mohamad, Ramadhyani, and Viskanta (1994)	0.5
	Tsotsas and Martin (1987)	4.5
	best fit	30
Sullivan, Hollands, and Shewen (1984)	Mohamad, Ramadhyani, and Viskanta (1994)	0.12
	Tsotsas and Martin (1987)	2.3
	best fit	37

Zunft et al. (2011) investigated the heat losses and destratification of a 120 m<sup>3</sup> packed bed storage installed in a central air receiver CSP plant. However, they didn't state important parameters of their storage medium, tank dimensions and thermal isolation. Comparing of the simulation with their temperature profiles after standstill in Figure 4.16 is therefore rather a best fit for the missing data then a proper validation. Also, their storage is divided into four parallel



Figure 4.13: Validation of charging, idling and heat loss simulation. Plot by Jones and Golshekan (1989) with own simulation according to Mohamad, Ramadhyani, and Viskanta (1994)



Figure 4.14: Validation of charging, idling and heat loss simulation. Plot by Jones and Golshekan (1989) with own simulation according to Tsotsas and Martin (1987)



Figure 4.15: Validation of charging, idling and heat loss simulation. Plot by Jones and Golshekan (1989) with own simulation,  $\lambda_{eff} = 25 \frac{W}{m^2 K}$ 

chambers inside the housing. The heat losses are therefore even harder to approximate. They also state that there has been back flow from the cold pipes into the storage due to the lack of a flap valve which explains the very high temperature drop at the hot end of the storage. This flow of cold air probably cools down several of the top regions which can't be simulated in the models. The informative value of this validation is therefore not high.

Reproduction of Sullivan, Hollands, and Shewen's (1984) destratification could not be achieved. The simulation shows much higher heat losses at the top of the storage when their described insulation is implemented unless destratification is increased to such an extend that the temperature drop is distributed all over the storage. The lower layers don't reach the high temperatures suggested by their study. This is true for Tsotsas and Martin's (1987) correlations in Figure 4.17 as well as when higher effective thermal conductivities are assumed, as in Figure 4.18.

A destratification model in a one-dimensional approach is very likely to not describe the temperature waves well. On the other hand, destratification shouldn't be a ruling part of the overall plant performance because the idling times of the storage should be short (see Figure 5.3). Thus, the correlations proposed by Tsotsas and Martin (1987) are implemented into the model knowing that there might be very large differences to reality. This is a matter which should be investigated thoroughly in an experimental study.



Figure 4.16: Validation of charging, idling and heat loss simulation. Plot by Zunft et al. (2011) with own simulation according to Tsotsas and Martin (1987)



Figure 4.17: Validation of charging, idling and heat loss simulation. Plot by Sullivan, Hollands, and Shewen (1984) with own simulation according to Tsotsas and Martin (1987)



Figure 4.18: Validation of charging, idling and heat loss simulation. Plot by Sullivan, Hollands, and Shewen (1984) with own simulation,  $\lambda_{eff} = 37 \frac{W}{m^2 K}$ 

# 5 THE PLANT MODEL

The thermal storage model has to be integrated into a power plant model in order to simulate realistic working modes and to find optimal dimensions of the plant's parameters for year-long operation. In the following, the modeling of the plant, the underlying assumptions and the controlling are described.

# 5.1 SOLAR ENERGY SOURCE

The solar heat source of the SUNSPOT cycle is a central receiver with a heliostat field. The optics of the heliostat field and thermal behavior of the receiver are calculated according to a model proposed by Gauché, Backström, and Brent (2011), assuming a checker-board pattern as in ESolar's (2011) design. Input data are several efficiencies, geometrical properties, thermal correlations and weather data. Some of these are:

- solar field aperture area
- receiver dimensions
- absorptivity, emissivity and view factor of receiver
- receiver operating temperature
- latitude and longitude of location
- hourly data for direct normal irradiation (DNI), ambient temperature and wind load at location

From these, the exact position of the sun at every hour of the year is calculated and an energy balance for the receiver is made. In it, the radiative energy input into the receiver is compared to approximated convective and radiative heat losses. The (positive) difference is the heat flow supplied to the Brayton cycle.

#### 5.2 DEMAND PROFILE

The simulation is supposed to prove that CSP plants with TES can replace fossil power plants by showing that the proposed pilot plant can meet the electricity demand of the national South African grid. This, of course, is not a realistic assumption because there will always be different kinds of power plants or at least several plants in a generation system. These can assist each other so that not all of them have to be able to deliver output at every point in time. The used demand profile features hourly data of the national grid for the year 2010. For a more reliable simulation, profiles of several years should be studied but these could not be retrieved. The South African demand profile is determined by morning and usually bigger evening peaks after sunset. In contrast to countries with wide-spread air conditioning installations, like the U.S., the winter demand is considerably higher than that of the summer (see Figure 5.1). The first makes short-term storages necessary to shift energy from times of high solar irradiation to the evening, the latter calls for big energy storages in combination with either oversized solar fields or high-fossil fuel usage. In order to have a filled TES for winter nights, the demand profile is scaled in such a way that the GT on it's own can supply the maximum winter evening peak.



Figure 5.1: Weather and demand curves for Jan-01 and Jul-06 (not to scale)

#### 5.3 BRAYTON CYCLE

The Brayton cycle has a nominal power rating of 5  $MW_{el}$ . However, the actual maximum power output depends on the ambient temperature, as does the overall cycle efficiency (see Equations (5.1) and (5.2)). Another efficiency factor has to be introduced for part-load behavior in Equation (5.3). These correlations have been obtained by linearization of plots by Li and Priddy (1985).

$$f_{GT,amb} = 3.234 - 0.007692 \frac{T_{amb}}{K}$$
(5.1)

$$\eta_{GT,amb} = 1.472 - 0.001641 \frac{T_{amb}}{K}$$
(5.2)

$$\eta_{GC,pl} = 0.74 + 0.26 \frac{P_{GC}}{P_{GC,nom}}$$
(5.3)

All of the efficiencies are multiplied with the Brayton cycle's Chambadal-Novikov efficiency of Equation (5.4) in order to obtain an overall efficiency factor. This modification of the Carnot efficiency describes real world heat engines much better than the original and has been recommended by Lubkoll, Brent, and Gauché (2011) and Gauché (2012).

$$\eta_{GC,CN} = 1 - \sqrt{\frac{T_{f,GT,out}}{T_{f,GT,in}}}$$
(5.4)

Heide (2012) stated that state of the art gas turbines' exhaust temperatures can be kept constant up until their load sinks below approximately 60 % of the nominal load. Since the part-load efficiency factor also decreases rapidly at this operating mode, the gas turbine will not be used at a lower part-load.

Some parameters of the Brayton cycle are given in Table 5.1. The gas turbine (GT) is defined closely to the SGT-100 turbine's data sheet by SIEMENS AG (2010).

Parameter	Abbreviation	Unit	Quantity
GT rating	P <sub>GT,nom</sub>	[MW <sub>el</sub> ]	5.0
Minimum load		[%]	60
GT exhaust temperature	T <sub>f,GT,out</sub>	[°C]	530
GT inlet temperature	T <sub>f,GT,in</sub>	[°C]	1 000
Compressor pressure ratio	$\pi$	[-]	15:1
Compressor temperature increase	$\Delta T_C$	[K]	385 EC (2005)
Chambadal-Novikov efficiency	$\eta_{GC,CN}$	[-]	0.21

Table 5.1: Parameters of the Brayton cycle

Current South African gas turbine stations run on Diesel fuel. However, for international comparability the use of a natural gas combustor is assumed. The high Diesel prices would result in a cost optimum with a very small co-firing rate but a very high LCoE, making it non-viable as compared to the big base-load coal power plants common in South Africa. Pressure drops of receiver and combustor are neglected because consideration of them would require a much more detailed modeling of the Brayton cycle. Pressure drops at the outlet of gas turbines cause lower efficiencies and maximum power outputs of it. In the SUNSPOT cycle, these originate in the HX and/or the storage system. Since both of these losses depend on the gas mass flows, incorporating them in the enthalpy demand of the GT means an additional loop in each calculation. For the sake of computational time, these losses are therefore just handled as additional parasitic losses and the demand will not be met completely. The storage system's pressure drop during charging will be overcome by a fan at its cold side.

## 5.4 RANKINE CYCLE

The Rankine cycle is modeled very roughly by its Chambadal-Novikov efficiency and an additional part load efficiency. The high temperature is kept constant unless the storage outlet temperature drops. This is tolerated until a pre-defined threshold temperature is reached. The low temperature is determined by the (dry bulb or wet bulb) ambient temperature, the cooling mechanism (dry or wet) and the condenser design. Because of the dry climate in most regions with good solar irradiation, dry cooling is used in the following.

The steam turbine is assumed to have the suiting requirements for the deliverable energy quality, i.e. the nominal ST inlet temperature is equal to the nominal GT outlet temperature minus losses in ducts, HX and piping. The power rating is handled as a matter of optimization in Section 7. An existing example is the SST-600 turbine by SIEMENS AG (2011) that operates between 0.4 and 6 MW<sub>el</sub>.

# 5.5 PLANT CONTROL

Controlling the plant model is a non-trivial problem because of the number of degrees of freedom ( $P_{GC,sol}$ ,  $P_{GC,co}$ ,  $P_{SC}$ ) and certain requirements that can't be met at the same time. A detailed, universal plant control is out of the scope of this project and should be created in a following study but during a first optimization phase it has been decided which ones of the rules will not be obeyed in which situations. The preliminary outcome is shown in Table 5.2. Figure 5.2 shows two selected days' simulated curves for power demand, supplied electricity from the GT through the receiver, through the combustor and from the ST.

Requirement	Implementation
The demand is always met	Fulfilled
GT and ST run between 60 and 100% load or at no-load	Fulfilled
As much solar energy as possible is used	If the storage is full enough, the GT will only be turned on once the theoretical power generation of 'solar-only' reaches 1 MW <sub>el</sub> .
The ST runs steadily for several hours and is not constantly turned on and off	<ul> <li>When the storage cannot supply the HX's threshold temperature, the plant will generate power only with the GT for three hours.</li> <li>The ST strictly follows the demand profile as long as possible. It therefore runs at a different rating every hour.</li> </ul>
Âğ	



Figure 5.2: Supply and demand curves for Jan-02 and Jul-06

It can be seen that both turbines never run at the same time. This is due to the scaling of the demand profile to the maximum power output of the (bigger) GT. The ST therefore only generates power during night time or time of very low irradiation. The GT has to jump in for the ST whenever the demand exceeds it's maximum power rating because the two combined minimum power ratings are higher than the GT's maximum rating.

The summer day's profile shows that the fossil combustor supports the GT when there's fluctuations in the receiver's output at a mediocre level as it's supposed to. The shown day (January 2<sup>nd</sup>) has been chosen because it has demanding DNI characteristics: A regular summer day wouldn't need back-up fuel during midday. The winter day shows three phases: (a) night-time with power supplied by the ST, (b) midday when the GT can be fired by solar energy and (c) late afternoon and evening, when there's no more solar energy available, the demand peak is too high for the ST and the fossil combustor generates the needed heat. The exact energy source distribution is, of course, dependent on the turbine sizes, the storage dimensions, the demand scaling and power plant controlling.

Figure 5.3 shows the air mass flows through the storage. It almost exclusively runs at nominal (dis-)charge flows so the idling mode doesn't occur. The big uncertainty concerning the modeling at no-flow (see Section 4.2.2) is therefore not problematic for the overall storage simulation. It has been decided that the Rankine cycle will always be kept warm in order to minimize start-up times and thermal stresses on HX and the ST. For this, 10 % of the nominal heat requirement of the cycle is assessed. Therefore, whenever the storage is charged, there will also be a air mass flow from the GT to the HX. This practice will be re-evaluated for technical and economical feasibility in further investigations of the SUNSPOT cycle.



Figure 5.3: Fluid mass flow through storage (negative values indicate discharging)

# 6 COST ANALYSIS

To achieve economic benefits beside the obvious ecological goals, detailed optimization of the main parameters is of particular importance. In the following, correlations for capital and operational expenditures as well as the Levelized Cost of Energy are presented.

#### 6.1 CAPITAL EXPENDITURE

The high initial investments are the biggest draw-back of CSP power plants and most other renewable power generation systems. The capital expenditure (CAPEX) is usually the determining element of final energy costs and the early need for such significant funds a major risk. Most investment costs incur for the power block, the solar field and - if applicable - the TES system. Estimates for the specific costs of the described cycle are given in Table 6.1.

Table 6.1: Investment costs of the described cycle without storage system (EUR 1 = ZAR 10, USD 1 = ZAR 7.50) derived from Gemasolar (2011) and EC (2005)

Element	Unit	Dependent on	Value
Power Block			
Brayton Cycle	[ZAR/kW <sub>e/</sub> ]	P <sub>GC,nom</sub>	4 170.00
Adaption of GT for solarization	[ZAR]	[-]	4 000 000.00
ST + Balance of Plant + Control	[ZAR/kW <sub>e/</sub> ]	P <sub>SC,max</sub>	6 900.00
Trafo + Cables	[ZAR/kW <sub>e/</sub> ]	$P_{GC,nom} + P_{SC,max}$	380.00
Engineering + Installation	[ZAR]	[-]	24 000 000.00
Solar			
Receiver	[ZAR/m <sup>2</sup> ]	$A_{RE}$	288000.00
Heliostats	[ZAR/m <sup>2</sup> ]	$A_{aperture}$	1 320.00
Tower	[ZAR/m]	H <sub>Tower</sub>	110 000.00
100061	[ZAR]	[-]	14 000 000.00
Land use	[ZAR/m <sup>2</sup> ]	$A_{aperture}$	121.00

The costs of the storage system are a very important part of the optimization later on. However, because of the novelty of the system's use for CSP plants, reliable sources couldn't be found. In the following, two different designs are being evaluated economically.

#### 6.1.1 Tank Storage

A cylindrical tank system has been assumed for the thermal storage to allow the use of relatively simple one-dimensional equations. Herrmann, Kelly, and Price (2004) gives estimates for some cost elements of molten salt 2-tank storage systems in dependence of their size, namely costs for the tank itself, insulation, foundation, balance of system, HTF-salt heat exchangers, salt pumps and the salt. The latter three don't apply to a rock bed storage. The calculated costs for the tank itself have to be increased because of the thermal stresses induced by the rock bed. It will be assumed that these are covered by a cost correction of +50 %. The correlations for the rock bed tank system are given in Table 6.2.

#### 6.1.2 Structureless Storage

An open ambient air cycle and a rock bed TES allow for the use of an extremely simple design because of the environmental harmlessness and the infinite availability of the HTF. Therefore, designs without any steel structure or any structure at all can be realized. An example for the first would be an old coal mine, filled with rocks and only lightly covered. The latter could be build in the form of a pile of rocks, held by foundation and covered with a ductile, yet thermo-resistent layer, e.g. insulation wool. The problem of thermal stresses could be evaded in both cases. The drawback of these simple designs is the uncertainty in flow and heat distribution. On the other hand, they would most likely prove so cheap that hugely oversized storage systems are still economically viable.

In the course of this study, the LCoE of a structureless storage design and of a tank system will be calculated. However, the packed bed model will remain unchanged so that the cone-like shape of a rock pile can't be depicted. The foundation is assumed to be half as expensive as in the tank design, the tank costs are omitted, the insulated area is 30 % bigger and the balance of system costs are the same (see Table 6.2).

The costs for the rocks in questions are taken from quotes by AfriSam (2010) and other local quarries, as reported by Allen (2012). Therein, the costs of the rocks itself are negligible while the crushing of them into smaller particles generates some, still low, costs. The finer the grain, the more expensive are the rocks. An optimization for the particle size has been conducted in Section 7.

## 6.2 OPERATIONAL EXPENDITURE

The operational expenditure (OPEX) of a CSP plant has to be much lower than in conventional power plants. Only then is the high CAPEX justifiable. In a hybridized plant this means keeping the fossil co-firing at a minimum while assuring that the demand is still met. In general, higher CAPEX in a TES, a bigger heliostat field or a bigger steam turbine are supposed to lower fuel costs during the years of use.

Since detailed data for OPEX couldn't be found, the bulk specific value in Equation (6.1) from Turchi et al. (2010) has been used for yearly operation and maintenance costs. In Equation (6.2), fuel costs are calculated from the heat delivered by the combustor and predicted fuel prices. Because it's not a common fuel in South Africa, the local price for natural gas could not be identified. Instead, (historical) world market prices from the energy statistics of the IEA (2011) have been used to generate a prediction for the trend in the future. The derivation can be found in APPENDIX F.

$$OPEX_{O\&M} = 487.50 \frac{ZAR}{kW_{el}a} (P_{SC,nom} + P_{GC,nom})$$
(6.1)

Element	Unit	Dependent on	Value
Rocks	[ZAR/kg]	m <sub>s</sub> ,d	0.08 - 0.43*
Ducts	[ZAR]	[-]	200 000.00
Ralance of System	[ZAR/m <sup>3</sup> ]	$V_{St}$	552.70
Dalarice of System	[ZAR]	[-]	3 043 000.00
Tank Design			
Tank	[ZAR/m <sup>3</sup> ]	$V_{St}$	1 102.00
Idlik	[ZAR]	[-]	797 100.00
Inculation	[ZAR/m <sup>2</sup> ]	$A_{St,surface}$	2 640.00
Insulation	[ZAR]	[-]	46 000.00
Foundation	[ZAR/m <sup>2</sup> ]	$A_{cs}$	9 761.00
Touridation	[ZAR]	[-]	1 758 000.00
Structureless Design			
Tank	[ZAR]	[-]	0.00
Inculation	[ZAR/m <sup>2</sup> ]	$A_{St,surface}$	3432.00
Insulation	[ZAR]	[-]	59800.00
Foundation	[ZAR/m <sup>2</sup> ]	$A_{cs}$	4880.50
FUUIIUation	[ZAR]	[-]	879000.00

Table 6.2: Investment costs of the different storage designs (EUR 1 = ZAR 10, USD 1 = ZAR 7.50) derived from Herrmann, Kelly, and Price (2004), Allen (2012) and AfriSam (2010) \* inversely proportional to particle size

$$OPEX_{fuel} = \frac{\sum Q_B}{\eta_B e_{fuel}} pr_{fuel}$$
(6.2)

Insurance costs are assumed to be constant for every year which might be realistic in spite of inflation because of depreciation of the plant. The derivation from the total CAPEX as well as the quantitative value of 1.5 % in Equation (6.3) are taken from the SOLGATE project's final report EC (2005).

$$OPEX_{lns} = 0.015 CAPEX \tag{6.3}$$

#### 6.3 LEVELIZED COST OF ENERGY

One of the most common financial tools to compare different power generation technologies is the Levelized Cost of Energy. In it all CAPEX and OPEX are allocated to the total amount of generated electricity. Since the costs as well as the energy output occur at different times within a period of several decades they have to be shifted to one point in time. The formula given for the problem by Short, Packey, and Holt (1995) is Equation (6.4).

$$LCOE = \frac{\sum_{y ear=0}^{LI} \frac{C_{year}}{(1+r)^{year}}}{\sum_{y ear=1}^{LT} \frac{P_{el}}{(1+r)^{year}}}$$
(6.4)

With the values given in Table 6.3,  $C_{year}$  being the combined costs in the period *year* and  $P_{el}$  being the accumulated yearly energy output. Investment costs are being distributed over the life-time by creating net present values.

Parameter	Abbreviation	Unit	Value
Anticipated plant life-time	LT	[a]	30
Annual discount rate	r	[%]	8.00

Table 6.3: Financial parameters used in LCoE calculations, as by Turchi et al. (2010)

# 7 OPTIMIZATION AND SENSITIVITY ANALYSIS

Once build and validated, full-year simulations are run with the plant model. At first, many parameters are varied in a wide range to narrow the scope to viable combinations and find the major sensitivities. Since the inputs, like weather data and the demand profile, are changed on an hourly basis, the plant model runs in these steps as well. For every hour's plant mode the storage model has to run in a specific mode (charge, discharge, idle) as well but due to the higher sensibility to temperature changes it runs in steps of 30 seconds. The drawback of this detailed simulation, as well as of the loops that are necessary to re-evaluate the turbines' outputs, is a relatively long computation time of approximately 30 minutes for one full-year run. All LCoE values shown are for the tank solution and the location Upington, Northern Cape. The structureless design usually turns out slightly cheaper.

# 7.1 STORAGE VOLUME, SOLAR MULTIPLE AND TURBINE RATING

The first runs are made with varying storage volumes and solar multiples (SM). The latter is the quotient of the installed aperture area to the one needed to supply enough heat to run the GT at nominal load at a sunny summer day's noon. The receiver area and tower height are changed according to linear dependencies on the aperture area which have been derived from EC (2005). Diameter and length of the storage unit are changed at the same rate.

In the second round, the ST's power rating and the SM are varied. This is supposed to help find a good rating ratio between ST and GT. The ranges in which the parameters are changed are given in Table 7.1.

In the given range, the outcomes of these simulations in Figures 7.1 and 7.2 show not a very strong dependency of the required amount of fuel per generated amount of electricity on the SMs. The dependency on the storage size is high up to a volume of approximately 1 500 m<sup>3</sup>. When increasing its size even further, the additional fuel saving is negligible. The same is true for the steam turbine's power rating: when increasing it to over approximately 4.4 MW<sub>el</sub>, there is no further decrease in specific fuel consumption. This is because of the limitation of the ST to run at a minimum load of 60 %. When the demand, mostly at night, drops below this threshold, the ST cannot meet the demand any more. The trend of the LCoE is very similar to these energetic findings.

Parameter	Unit	Range	
		Set 1	Set 2
SM	[-]	1.75 3.05	
$A_{aperture}$	[m²]	4500093000	
V <sub>St</sub>	[m <sup>3</sup> ]	230 2 300	785
D	[m]	5.50 14.5	10.0
L	[m]	4.50 13.5	10.0
P <sub>SC,nom</sub>	[MW <sub>e</sub> /]	4.3	3.6 6.0

Table 7.1: Ranges of parameters in first runs.



Figure 7.1: Sensitivity to storage volume and solar multiple



Figure 7.2: Sensitivity to steam turbine rating and solar multiple

## 7.2 STORAGE LENGTH AND SOLAR MULTIPLE

The preliminary storage size of 1 500 m<sup>3</sup> makes a big tank diameter necessary if typical ratios of height and width are aspired. For the following simulation a diameter of 19 m and a ST rating of 4.4 MW<sub>el</sub> are fixed and solar multiple as well as the storage length in flow direction are varied. Figure 7.3 shows that the LCoE of this plant's configuration reaches a minimum at a storage height of 17 m and a solar multiple of 2.5. This minimum is relatively insensitive to changes of the SM in the range of 1.8...3.0 or the storage height above approximately 10 m. The costs of the storage are obviously not a determining issue. A small storage, on the other hand, leads to the inability to store the energy in the exhaust gases.





## 7.3 PARTICLE DIAMETER, STORAGE LENGTH AND DIAMETER

The thermocline in a packed bed can be improved by decreasing the particle size, making more efficient use of the storage length (see Figure 7.4). The energetic downside of this is that the pressure drop and therefore the parasitic losses increase. Also, the rocks' specific price increases when they have to be crushed to finer particles. Figures 7.5 and 7.6 show LCoE optimizations of particle size and bed length for two different storage diameters. The remaining parameters are kept constant at the optima found above. This applies to the void fractions as well because it is assumed that the change in particle size will not affect it noticeably. Figure 7.4 shows the thermoclines for two different particle diameters for the same storage size and point in time. It can clearly be seen that the storage with bigger particles cannot store as much energy as the one with small particles.

Figure 7.5 shows that the particle diameter used was the most economic for the layout the simulation was based on, namely a bed length of 19 m. Yet, according to the outcome, a setup featuring a smaller storage with a length of 12 m but also a smaller particle diameter of 0.04 m would mean an even lower LCoE. Since the LCoE rises more strongly when the storage is smaller than the optimum as compared to a bigger storage, some oversizing of it is advisable. The simulation with a considerably smaller storage *diameter* (Figure 7.6) shows increased costs. To sum up, none of the small changes demonstrated in this chapter had a huge impact on the final electricity costs. Mostly the LCoE stayed between 1.60 and 1.70 ZAR/kWh. A change in



Figure 7.4: Thermocline at different particle diameters



Figure 7.5: Sensitivity to particle diameter and bed length for D = 19 m



Figure 7.6: Sensitivity to particle diameter and bed length for D = 14 m

plant control on the other hand would most likely have a profound impact. If the plant was allowed to feed its output into the grid whenever possible and didn't have to follow the demand curve exactly, the plant control would became much easier, the turbines could run on their design point most of the times and the use of fossil fuel would decrease noticeably. To prove this, another simulation was run with different rules of plant control.

## 7.4 PLANT CONTROL

The turbine control is a very simple one: Whenever the DNI exceeds 100 W/m<sup>2</sup> the gas turbine will deliver as much power as possible from the heat passed on by the receiver. Only if the output is less than 3 MW<sub>el</sub> will the fossil fuel combustor jump in and keep it at this threshold. If the DNI is less then 100 W/m<sup>2</sup>, the ST runs at its nominal load. Therefore, the demand will not be met at night and might be exceeded or fallen below during day time. The chosen parameters can be found in Table 7.2.

Parameter	Unit	Value
P <sub>GC,nom</sub>	[MW <sub>el</sub> ]	5.0
P <sub>SC,nom</sub>	[MW <sub>e</sub> /]	3.5
D	[m]	19
$d_{ ho}$	[m]	0.04
Ĺ	[m]	12
SM	[-]	2.3

Table 7.2: Parameters for simulation with changed plant control

The chosen plant control makes very little use of the installed fossil fuel combustor. It's only used in the mornings and evenings when the DNI is not strong enought to deliver the chosen minimum power output of 3.0 MW<sub>th</sub> (Figure 7.7). Most of the time, the ST runs very steadily for many hours at its nominal load. The annual delivered electricity doesn't differ much from the plant with demand-control but evening peaks can't be met. The decreasing GT output is due to the correction factor for ambient temperature, which increases during the day. Some performance indicators are compared in Table 7.3.



Figure 7.7: Supply and demand curves for turbine controlled plant

Indicator		Unit	turbine controlled	demand controlled
f <sub>fossil</sub>		[J <sub>ch</sub> /J <sub>el</sub> ]	0.078	0.2119
$W_{el,a}$		[GWh <sub>el</sub> ]	32.3	31.4
$\zeta_a$		[%]	43.4	38.1
LCOE	tank	[ZAR/kWh <sub>e/</sub> ]	1.14	1.62
	structureless	[ZAR/kWh <sub>e/</sub> ]	1.12	1.60
$\eta_{c,a}$		[%]	69.4	83.8
$\eta_{dc,a}$		[%]	99.3	99.5
$\Delta p_{\textit{f,m}}$		[N/m <sup>2</sup> ]	169	166

Table 7.3: Performance indicators for different kinds of plant control

# 7.5 COMPARISON

The found LCoE values should be compared to prices for CSP-generated electricity found in other studies. Turchi et al. (2010) analyzed molten-salt TES systems and predicts an LCoE of 1.03 ZAR/kWh for a 100 MW<sub>el,net</sub> power tower with 6 h of thermal storage capacity. The EC (2005) calculates LCoEs between 1.00 and 1.50 ZAR/kWh for combined cycle tower systems with a rating of 16 MW<sub>el,net</sub> and more than 50 % fossil co-firing rate.

To put the economic findings into perspective, the current average over different residential electricity tariffs charged by ESKOM (2012) is approximately 0.92 ZAR/kWh. These prices have undergone dramatic changes during the last years and will continue to do so. The average price adjustments for the years 2008 to 2011 were in the order of +25 % p.a. and the next years are prospected to bring changes of similar magnitude (see ESKOM (2011)). This means that, according to the simulation with adjusted plant control, the LCoE could reach parity to charged tariffs as early as 2014. Naturally, this doesn't exactly reflect the prices of electricity generation in South Africa.
# 8 CONCLUSION AND FURTHER WORK REQUIRED

### 8.1 CONCLUSION

A power plant model of the SUNSPOT cycle with focus on the thermal energy storage system has been built and validated. It has been used to predict a 5  $MW_{el}$  pilot plant's behavior in year-long hourly simulations while the rock bed storage was analyzed in a much finer solution. A life-time analysis of capital and operational expenditure was used to find sensitivities to the storage's major parameters and the turbine rating and eventually optimize the plant for the levelized cost of electricity.

The chosen storage system is directly flown through by the HTF air. It consists of a packed bed of rocks in a tank or lose pile, a discharging blower and the connecting ducts. The model is one-dimensional and based on the "Effectivness-Number of Transfer Units" analogy as presented by Hughes (1975) with the correction of Sagara and Nakahara (1991) and the GLE approach of Martin (2005). The lack of a radial dimension means neglecting temperature gradients in it. These are often caused by wall channeling which has been subject to many studies. However, several of them state that this kind of flow uniformity is negligible in beds with sufficiently big tube-to-particle diameter ratios. Big losses through the walls would cause radial temperature gradients as well but in accordance to studies on existing TES systems of CSP plants, energetic losses to the outside are considered small. The discharging efficiency, which directly derives from these losses, has been calculated to be at very high levels of above 99 %. The charging cycle's efficiency, that's influenced by the lost heat in the outgoing fluid, is calculated at much lower values. This could be solved by adding more length to the storage but the economic evaluation proved it not viable.

The pressure drop of the rock bed storage, that is also used in the GLE, has been calculated according to correlations proposed by Singh, Saini, and Saini (2006) for non-spherical particles. The energetic impact of the required power for blowing the air is completely insignificant due to an average pressure drop of only little more than 100 Pa at the found optimum. An artificial increase of this parameter, e.g. by adding a flow distribution system, could enhance the thermocline quality while increasing the blower's power consumption.

The idling mode and destratification during it were analyzed and modeled as well. Validation with other studies' data didn't show very good agreement which might be caused by the fact that a one-dimensional model is used. However, when only short stand-still times apply, the impact of destratification should not be determining. With the developed preliminary plant control mechanisms, the idle mode didn't occur at all because the storage was always either

being charged or discharged. Only when gas turbine and steam both don't run or both generate electricity at the same time would there be no flow inside the storage.

The plant control has been developed under the primacy of always supplying exactly the amount that the national grid needs (skaled down to a maximum of approximately 5  $MW_{el}$ ). This results in many changes of the turbines' loads and a relatively high usage of back-up fuel burning but proves that CSP with a sufficiently big storage is able to meet almost any given demand profile. A different, still not optimal, controlling rule was tested and showed much more favourable results in terms of turbine load behavior and cost efficiency.

The cost calculations are based on values from the literature but had to be completed with estimated costs. Especially the storage's costs are hard to determine because of the novelty of the concept at utility scale in CSP technology. Nevertheless, the calculations for the LCoE output comparable results to the literature.

Sensitivities to some of the main parameters could be analyzed through the conducted simulations. The required heliostat aperture area was found to be relatively big at a solar multiple of 2.5. However, the sensitivity of the LCoE to the solar multiple in a certain area was not very big. Most parameters showed a decently flat optimum which means that small changes from the optimum don't strongly affect the viability. According to another simulation, the storage volume should be at least 1 500 m<sup>3</sup>, while the cost optimum is at much bigger volumes. The longer (in direction of flow) the storage is, the more energy can be stored by heating up more stones. The smaller the particle size, the more of the thermal capacity can be used through a steeper thermocline and thus less losses by exiting fluid flow. These dependencies were reproduced by the model as well.

### 8.2 FURTHER WORKS REQUIRED

If a highly detailed flow investigation is wished for, there's probably no way around a three-dimensional CFD analysis. However, with today's models and computational capacity this is not feasible. Another way of considering at least two-dimensional temperature profiles, is by working with (effective) radial conductivities, but just as with a CFD analysis, it 's questionable whether this would noticeably improve the simulation quality when a full year is analyzed. In the opinion of the author, the simplest and yet most effective way of increasing the model's significance is by implementing more realistic and more favourable plant control mechanisms, which wasn't in the scope of this study. There are some possibilities in the SUNSPOT cycle that were not accounted for as well. In the current model, all heat that can't be used in the gas turbine is dumped or wasted by defocussing heliostats. Because of the amount of exergy that has to be allocated to compressing air, it wouldn't be advisable to use this air to fill the storage but if a low-pressure part of the receiver uses spare heat to charge the storage via a bypass, it could increase the output of the plant considerably.

A more reliable cost calculation would reduce the financial risk and improve predictability. However, this requires a detailed design of the storage containment which, when using a tank, requires taking into account the thermally induces mechanical stresses in such a vessel. The proposed low-cost design featuring no mechanical containment at all, needs some more detailed planning and most likely experimenting as well.

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## **A PRELIMINARY CALCULATIONS**

The following is a preliminary estimate for some parameters:

 $P_{GC} = 5 \text{ MW}_{el}$ 

A gas turbine that falls in this range is SIEMENS AG's (2010) SGT-100. Its parameters are:

 $\begin{array}{l} P_{GC,max} = 5.4 \; \text{MW}_{el} \\ \eta_{GC,el} = 0.31 \\ T_{GT,out,max} = 531 \; ^{\circ}\text{C} \\ \dot{m}_{f,GT,out,nom} = 20.6 \; \frac{kg}{s} \end{array}$ 

Properties of air:

$$\begin{split} c_{p,f}(t = 500) &= 1\,100\,\frac{kJ}{kg\,K} \\ \rho_f(t = 500) &= 0.44\,\frac{kg}{m^3} \\ \mu_f(t = 500) &= 2.19\cdot10^{-5}\,\frac{kg}{m\,s} \\ t_{amb} &= 25 \\ \eta_{SC} &= 0.30 \end{split}$$

 $\dot{Q}_{GT,out,max} = \dot{m}_{f,GT,out} c_{p,f} \left( T_{GT,out} - T_{amb} \right) = 20.6 \frac{kg}{s} \, 1 \, 100 \frac{J}{kg \, K} \, (531 - 25) \, K = 11.47 \, MW_{th}$ 

A typical SC/GC-rating ratio is 0.5:

 $P_{SC} = 0.5 \, P_{GC} = 2.5 \, MW_{el}$ 

A steam turbine with a suiting maximum inlet temperature is SIEMENS AG's (2011) SST-060:  $t_{ST,in,max} = 530$  °C

$$\dot{Q}_{HX,max} = rac{P_{SC,nom}}{\eta_{SC}} = 8.33 \; MW_{th}$$

 $c_{p,f}(t=25...500 \text{ °C}) = 1\ 100\ \frac{kJ}{kgK}$ 

$$\dot{m}_{f,HX,max} = \frac{\dot{Q}_{HX,max}}{c_{p,f} (T_{f,HX,in} - T_{amb})} = 15.15 \frac{kg}{s}$$

If a maximum fluid mass flux of  $G = 0.4 \frac{kg}{m^2 s}$  is assumed:

$$A_{cs,min} = rac{m_{f,HX,max}}{G} = 38 \ m^2$$

$$D_{St,min} = \sqrt{A_{cs,min} \frac{4}{\pi}} = 7 m$$

If the steam turbine rating is increased, e.g. doubled, the cross-sectional area has to be increased accordingly.

 $P_{SC} = P_{GC} = 5.0 \ MW_{el}$ 

$$\dot{Q}_{HX,max} = \frac{P_{SC,nom}}{\eta_{SC}} = 16.66 \ MW_{th}$$
$$\dot{m}_{f,HX,max} = \frac{\dot{Q}_{HX,max}}{c_{p,f} \left(T_{f,HX,in} - T_{amb}\right)} = 30.30 \ \frac{kg}{s}$$
$$A_{cs,min} = \frac{\dot{m}_{f,HX,max}}{G} = 76 \ m^2$$

$$D_{St,min} = \sqrt{A_{cs,min} \frac{4}{\pi}} = 10 m$$

For particle sizes between  $d_{\rho} = 0.01...0.1$  m the Reynolds number will be in the range of:

$$Re_{p,max} = \frac{d_p G}{\mu_f} = \frac{0.01...0.1 \cdot 0.4}{2.19 \cdot 10^{-5}} = 182...1830$$

# B DERIVATION OF E-NTU TEMPERATURES

This appendix shows the derivation of the solid temperature as given in Equation (3.3). It was found in (Allen 2010).

The fluid temperature after passing one section of the packed bed is given by Equation (3.2). If outside heat losses are neglected, the following equation evolves:

$$T_{f,i+1} = T_{f,i} - (T_{f,i} - T_{s,i}) \left(1 - e^{-NTU_i \frac{\Delta x}{L}}\right)$$

Due to energy conservation, all the energy has to be passed on from the fluid to the solids or the other way around. After regrouping this differential equation can be found:

$$\frac{dT_s}{d\tau} = \frac{L \dot{m}_{f,j} c_{p,f}}{\Delta x m_s c_s} (T_{f,i} - T_{s,i}) \eta_{NTU,i,j}$$

With linearization of the solid's temperature difference

$$\frac{dT_s}{d\tau} \approx \frac{T_{s,i,j+1} - T_{s,i,j}}{\Delta \tau}$$

and approximating the average solid temperature

$$T_{s,i,m} = \frac{T_{s,i,j+1} + T_{s,i,j}}{2}$$

the former differential equation can be expressed as

$$\frac{T_{s,i,j+1} - T_{s,i,j}}{\Delta \tau} = \frac{L \,\dot{m}_{f,j} \, c_{p,f}}{\Delta x \, m_s \, c_s} \left( T_{f,i} - \frac{T_{s,i,j+1} + T_{s,i,j}}{2} \right)$$

which then leads to Equation (3.3).

# C EQUILIBRIUM TEMPERATURE BEFORE IDLING

Energy conservation at start of idling mode:

$$T_{f,i,j} c_{p,f,i,j} m_{f,i,j} + T_{s,i,j} c_s \frac{m_s}{n} = T_{f,i,j+1} c_{p,f,i,j+1} m_{f,i,j+1} + T_{s,i,j+1} c_s \frac{m_s}{n}$$

With the following assumptions

 $T_{f,i,j+1} = T_{s,i,j+1}$   $c_{\rho,f,i,j+1} = c_{\rho,f,i,j}$   $m_{f,i,j+1} = m_{f,i,j} = \varepsilon A_{cs} \Delta x \rho_{f,i,j}$ 

it becomes

$$T_{f,i,j+1} = \frac{T_{s,i,j} c_s \frac{m_s}{n} + T_{f,i,j} c_{\rho,f,i,j} \varepsilon A_{cs} \Delta x \rho_{f,i,j}}{c_s \frac{m_s}{n} + c_{\rho,f,i,j} \varepsilon A_{cs} \Delta x \rho_{f,i,j}}$$

## D CALCULATIONS FOR WALL HEAT TRANSFER

#### **Turbulent Wall Nusselt Number**

These formulas are given by Dixon and Labua (1985):

$$Nu_{t}=Sh_{w,0}\left(\frac{Pr}{Sc}\right)^{1/3}$$

$$Sh_{w,0} = (1 - \frac{d_p}{D}) Sc^{1/3} Re_p^{0.61}$$
,  $Re > 50$ 

since  $d_p \ll D$ , this can be simplified to

$$Nu_t = Re_p^{0.61} Pr^{1/3}$$

#### **Details of Insulation**

Heat transfer coefficients are dependent on the linings temperature  $T_{ins}$  and are calculated by retrieved polynomial approximations. The following equation has been retrieved from ET (The Engineering ToolBox) (*Mineral Wool Insulation*) as a general correlation for mineral wool insulation and validated with data from IIG (2012). The overall agreement at mean temperatures up to 400 °C is satisfactory.

$$\lambda_{ins} = \left(8.7 \cdot 10^{-8} \left[\frac{T_{ins}}{K}\right]^2 + 4.9 \cdot 10^{-5} \left[\frac{T_{ins}}{K}\right] + 0.035\right) \frac{W}{mK}$$

The equation below is the correlation Unifrax (2009) gives for their *Fibrefrax®Durablanket Hot Face* lining product that is durable up to 1 400 °C. Since the insulation will make up for most of the temperature gradient,  $T_{ins}$  is approximated as the mean temperature between the storage and the surroundings for determining the heat transfer coefficient. Because of this, the temperatures of the tank and linings must logically be equal to the bed temperature.

$$\lambda_{lining} = (1.2 \cdot 10^{-7} \left[ \frac{T_{fi}}{K} \right]^2 + 2.5 \cdot 10^{-5} \left[ \frac{T_{fi}}{K} \right] + 0.03) \frac{W}{mK}$$
$$T_{ins} = \frac{T_{fi} + T_{amb}}{2}$$

# E THERMODYNAMIC PROPERTIES OF AIR

### **Thermal Conductivity of Air**

Derived from Lemmon and Jacobsen (2004):

$$\lambda_{f} = (1.5207 \cdot 10^{-11} \left[\frac{T_{f}}{K}\right]^{3} - 4.8574 \cdot 10^{-8} \left[\frac{T_{f}}{K}\right]^{2} + 1.0184 \cdot 10^{-4} \frac{T_{f}}{K} - 3.9333 \cdot 10^{-4} \frac{W}{m^{2} K}$$

### **Dynamic Viscosity of Air**

Derived from Lemmon and Jacobsen (2004):

$$\mu_f = \left(8.0 \cdot 10^{-14} \left[\frac{T_f}{K}\right]^2 + 4.0 \cdot 10^{-9} \frac{T_f}{K} - 1.9 \cdot 10^{-5}\right) \frac{kg}{ms}$$

### Thermal Capacity of Air

Valid for  $0^{\circ}C < T_f < 1000^{\circ}C$ . Derived from Bauer (2001):

$$c_{p,f} = \left(1007 - 7.453 \cdot 10^{-2} \frac{T_f}{K} + 2.431 \cdot 10^{-4} \left[\frac{T_f}{K}\right]^2\right) \frac{J}{kg \, K}$$

#### **Enthalpy Difference between Two Temperatures**

$$\Delta h_{f} = \int_{T_{f1}}^{T_{f2}} c_{p,f} dT$$

$$= \int_{T_{f1}}^{T_{f2}} \left(1\ 007 - 7.453 \cdot 10^{-2} \frac{T_{f}}{K} + 2.431 \cdot 10^{-4} \left[\frac{T_{f}}{K}\right]^{2}\right) \frac{J}{kg \, K} dT$$

$$= \left(1\ 007 \frac{T_{f2} - T_{f1}}{K} - 3.7265 \cdot 10^{-2} \frac{T_{f2}^{2} - T_{f1}^{2}}{K^{2}} + 8.1027 \cdot 10^{-5} \frac{T_{f2}^{3} - T_{f1}^{3}}{K^{3}}\right) \frac{J}{kg}$$

### **Exergy of Fluid**

Exergy of heat is derived from the standard equation:

$$E = m(h - h_{amb} - T_{amb}(s - s_{amb}))$$

The enthalpy difference is calculated according to the correlations described above for hot and ambient air temperatures.

The exergy difference  $\Delta s$  can be determined as shown in the following equation if the ambience is defined as the reference. In it  $c_{p,f}$  is the specific heat capacity as calculated before.

$$\Delta s = c_{p,f} \ln(\frac{T}{T_{amb}}) + \ln(\frac{p}{p_{amb}})$$

## F GAS PRICE

The figure below shows the development of natural gas prices on different markets between 1985 and 2010 as seen in (IEA 2011).





#### Natural gas prices in USD/MBtu, IEA (2011)

A polynomial fit has been executed with the mean value of the data of the USA and (when possible) Europe. Japan's gas prices are much higher because of the transport in the form of liquefied natural gas (LNG). The used correlation after changing the units to the "International System of Units" and ZAR then consists of a starting price in the first year and a yearly price increase:

$$pr_{fuel,0} = 51.54 \cdot 10^{-9} \frac{ZAR}{J}$$
$$\Delta pr_{fuel} = 1.693 \cdot 10^{-9} \frac{ZAR}{Ja}$$