SOLAR-DRIVEN DISTRIBUTED HEATING SYSTEM

Upgrading a 200 kW Solar-driven Organic Rankine Cycle Unit for Distributed Heating

TOBIAS JAHN

In Cooperation with
The Key Laboratory of Efficient Utilization of Low and Medium Grade Energy
The University of Tianjin, China
ABSTRACT

The University of Tianjin, China is working on a 200 kW solar-driven Organic Rankine Cycle (ORC) plant. Due to difficulties with Chinese energy regulations and legislation, the plant will not be connected to the grid for electricity generation. The university intends therefore to use the solar system for distributed heating at times without ongoing experiments. Since no heat consumer was designated initially, the heating purpose resulting in the most cost-effective usage of the already purchased components was sought. In this context, the plant’s performance in four different heating scenarios was assessed to determine the necessary upgrades, which led to the optimized Levelized Costs of Energy (LCOE). The upgrades considered a thermal energy storage system (TESS), extension of heat exchanger (HE) capacity and redesign of the HE’s hot fluid outlet temperature (HFOT). Scenario 1 (S1) represents the current system on-site for space heating and cooling. The system has been upgraded with a TESS and the HFOT was lowered. Scenario 2 (S2) differs from S1 by also considering upgrading of the HE capacity. In S1, the LCOE for the optimized system based on the original HE capacity and heat demand are shown, whereas the LCOE for S2 indicate the minimum LCOE possible with an optimized system for an increased space heating and cooling demand. In scenario 3 (S3) and 4 (S4), the optimized LCOE of the system used for industrial heat loads was studied. The industrial heat load was assumed to be constant. Two durations were chosen: 24 hours and 7 days a week (24/7) and 16 hours and 7 days a week (16/7) according to a factory working in a three-shift and two-shift system, respectively. In order to obtain the optimized LCOE, the parabolic trough collector (PTC) field and TESS were modeled and simulated in TRNSYS and MATLAB, respectively. For S1 the minimum LCOE of 0.187 $/kWh is achieved providing that none of the analyzed upgrades are made to the current system. In S2 the minimum LCOE of 0.145 $/kWh is obtained at 750 kW HE capacity, 10 m³ TESS and 50°C HFOT. In this setup, the HE capacity is large enough to utilize nearly all solar energy immediately. For S3, the lowest LCOE of 0.106 $/kWh was obtained at 80 kW HE and 40 m³ TESS and for S4, it was 0.098 $/kWh at 130 kW HE and 30 m³ TESS. Based on those results, the following main conclusions are drawn: (1) the low degrees of utilization of the plant in S1 and S2 led to high LCOE which are not competitive with those for traditional heating with air conditioners (0.112 $/kWh), (2) the LCOE can be optimized when the system provides heat continually throughout the year as required, for instance, by industrial processes and (3) for a system with optimized LCOE, the CO₂ reductions associated to the upgrades are below 6% for space heating and cooling and 55 – 65% for industrial process heat.

Keywords: Concentrating solar power (CSP)  
Parabolic trough collector (PTC)  
Thermal energy storage system (TESS)  
Levelized costs of energy (LCOE)  
Distributed energy system (DES)  
Space heating and cooling  
Industrial process heat
PREFACE

I, Tobias Jahn, hereby certify that I have elaborated and conducted this thesis independently and that all used literature has been cited. It has not been submitted in any form of another degree or diploma to any other university or other institute of education.

This thesis is the outcome of a cooperation work between the School of Business, Society and Technology of The Mälardalen University in Västerås, Sweden and the Key Laboratory of Efficient Utilization of Low and Medium Grade Energy of The University of Tianjin, China.

Tianjin University is currently focusing on completing a concentrating solar power plant for the new university research institute in Binhai District. At time of my arrival, the plant was partially built and designed for research purposes. Within this framework, it was my task to identify a suitable heating purpose for times without ongoing experiments to increase the plant’s degree of utilization.

I am therefore pleased to express my gratitude and appreciation to Prof. Dr. Zi Zhao for giving me this opportunity by inviting me into his research team.

I would also like to thank my supervisors Dr. Hailong Li for his valuable suggestions and comments and Dr. Shuai Deng for his great support throughout my stay in China. Their experience, knowledge and dedication were of substantial importance for completing my research work.

Last but not least, I thank the foundation of Gustaf Dahl for granting me a scholarship. Without their financial support, this study would not have been possible.

2015-09-14, Leipzig

Date and place

Signature

Tobias Jahn
SUMMARY

The global energy mix, nowadays, relies heavily on fossil fuel based technologies causing not only climate change, but also economic uncertainties in terms of fossil fuel price developments. These problems can be tackled by using renewable energy, such as solar energy. Nonetheless, such technologies need to be continuously developed and optimized to remain cost-effective and competitive with conventional fossil fuel technologies.

To this end, the University of Tianjin, China, is working on a 200 kW solar-driven Organic Rankine Cycle (ORC) plant for demonstration and research purposes. At this current project status, the 1096 m² solar field, 2 MW auxiliary heater and 250 kW heat exchanger (HE), for the distributed heating cycle, as well as the piping and vessels have already been purchased and installed. Upon completion of the entire plant, it will not be connected to the grid due to licensing difficulties that arise with the current energy regulations and legislation. In order to use the plant during times without ongoing experiments, a heating purpose is sought to utilize the harvested solar energy.

Since no specific heating purpose was designed initially, four scenarios with different heat loads were identified and investigated to determine the necessary upgrades to the already purchased components, which can lead to a better technical coordination of the main components and minimize the Levelized Costs of Energy (LCOE). The considered upgrades were: (1) an additional thermal energy storage system (TESS) up to 70 m³, (2) extension of the 250 kW HE capacity and (3) redesign of the original 100°C HE's hot fluid outlet temperature (HFOT). In the first and second scenario (S1 and S2), the system covered a space heating and cooling load. The cooling demand has been simplified to represent the heat that needs to be transferred from the HE. While in S1 only the HFOT was lowered and an additional TESS was considered, S2 also included an extension of the HE capacity. In the third and fourth scenario (S3 and S4), all possible upgrades were applied to provide industrial process heat loads for 24 hours, 7 days a week (24/7) and 16 hours, 7 days a week (16/7), respectively.

This work started with a literature review about the fundamentals of solar radiation, the key components of a solar-driven distributed heating system, the theory on heating load estimation for space heating and cooling as well as industrial processes and, last but not least, the programs suitable for modeling solar thermal systems. Then, the structure of the solar system was defined and subsequently modeled in TRNSYS and MATLAB, respectively and verified by means of results of other studies. With the definition of economic, technical and environmental performance indicators, the basis for the subsequent assessment of the results was established.

In S1, the current system setup results in the lowest possible LCOE of 0.187 $/kWh for space heating and cooling meaning that no additional TESS, change in HE capacity or lowering of HFOT is advisable. Consequently, the CO₂ emissions are not further reduced. The solar fraction, which is the share of energy covered by the solar field to the total heat demand, is ~45%. In S2, the HE capacity has been continuously increased until 750 kW. The supplied heat from the HE increased from 375 MWh in S1 to 1120 MWh in S2. With each extension of
HE capacity, the corresponding LCOE dropped further mainly because more energy from the solar field can be used directly with a larger HE. Ultimately, a minimum LCOE of 0.145 $/kWh is obtained at 750 kW HE capacity, 10 m³ TESS and 50°C HFOT. A total investment of about $88500 is required. The optimized system reached a solar fraction of ~36% and upgrade related CO₂ savings of ~5%. The LCOE of the first two scenarios are not competitive compared with those for traditional heating with air conditioners in China of 0.112 $/kWh and thus a utilization of the system for space heating and cooling purposes is not reasonable.

In S3 and S4 the system is used to provide heat to industrial processes. The setup in S3 with 80 kW HE capacity, 40 m³ TESS and 50°C HFOT led to the optimized LCOE of 0.106 $/kWh and provides 700 MWh heat. A total investment of $175’000 is necessary. In S4, the LCOE of 0.098 $/kWh is obtained with 130 kW HE capacity, 30 m³ TESS and 50°C HFOT requiring an investment of $135’000. This setup delivers 760 MWh of heat. The solar fraction and upgrade related CO₂ savings for S3 are ~79% and ~67% and for S4 ~78% and ~54%, respectively. The high degree of utilization of the system each month, especially during the summer period, results in LCOEs which are below those for diesel-based auxiliary heating in China of 0.178 $/kWh.

Concluding, the utilization of the solar system for industrial process heat is recommended. For this purpose, a heat consumer in the close vicinity of the plant must be found to evaluate the cooperation potential and the further course of action. If the plant is to be used for space heating and cooling then the heat demand must be increased with, for instance, solar water heating. This measure improves the productivity by utilizing more solar energy and hence lowering the LCOE.
# TABLE OF CONTENT

ABSTRACT ............................................................................................................. II
PREFACE ............................................................................................................... III
SUMMARY .............................................................................................................. IV
TABLE OF CONTENT .......................................................................................... VI
LIST OF FIGURES ............................................................................................... VIII
LIST OF TABLES ................................................................................................... IX
ABBREVIATIONS ................................................................................................. X
NOMENCLATURE .................................................................................................. X

1 INTRODUCTION .............................................................................................. 1
   1.1 Background ................................................................................................... 2
      1.1.1 Solar Thermal Collectors ........................................................................ 2
      1.1.2 Steam Rankine cycle vs. Organic Rankine Cycle ................................. 2
      1.1.3 Current Energy Regulation and Legislation in China ........................... 4
      1.1.4 Project Status and Plant Layout ......................................................... 5
   1.2 Objective and Research Questions .............................................................. 6
   1.3 Scope and Limitations .................................................................................. 7
   1.4 Thesis Outline ............................................................................................. 7

2 METHODOLOGY .............................................................................................. 8

3 LITERATURE REVIEW .................................................................................... 9
   3.1 Fundamentals of Solar Radiation ................................................................. 9
   3.2 Key Components in Solar-driven Distributed Heating Units .................... 10
      3.2.1 Parabolic Trough Collectors ................................................................. 11
      3.2.2 Thermal Energy Storage Systems ....................................................... 12
      3.2.3 Auxiliary Heater .................................................................................... 14
      3.2.4 Solar-Driven Distributed Heating Unit ............................................... 14
   3.3 Heating Load Estimation for Space Heating and Cooling ......................... 15
      3.3.1 Heat Balance Method .......................................................................... 15
      3.3.2 Degree-Day Method ............................................................................ 15
      3.3.3 Chinese Building Standard ................................................................. 16
   3.4 Heating Load Estimation for Industrial Processes ....................................... 18
   3.5 Modeling Solar Thermal Systems ............................................................... 19
      3.5.1 TRNSYS ............................................................................................... 20
      3.5.2 WATSUN .............................................................................................. 20
      3.5.3 POLYSUN ............................................................................................ 20
      3.5.4 MATLAB ............................................................................................ 21
4 SYSTEM MODELING AND VERIFICATION ................................................................. 22
  4.1 Structure of the Model .................................................................................. 22
  4.2 Meteorological Data .................................................................................... 24
  4.3 Heating Load Estimation ............................................................................. 25
    4.3.1 Space Heating and Cooling ................................................................. 25
    4.3.2 Industrial Processes .......................................................................... 27
  4.4 Solar Field .................................................................................................... 27
  4.5 Thermal Energy Storage .............................................................................. 30
    4.5.1 Charging and Discharging ................................................................. 30
    4.5.2 Run Through ..................................................................................... 32
    4.5.3 Bypass ............................................................................................... 33
    4.5.4 Storage Reheat .................................................................................... 33
    4.5.5 Heat Losses from Return Tank ........................................................... 34
    4.5.6 TESS Parameters .............................................................................. 34
  4.6 Auxiliary Heater .......................................................................................... 35
  4.7 Verification of the Model .............................................................................. 36
5 PERFORMANCE INDICATORS ........................................................................... 38
  5.1 Economic Indicator .................................................................................... 38
  5.2 Technical Indicators ................................................................................... 39
  5.3 Environmental Indicator ........................................................................... 40
6 RESULTS ........................................................................................................... 41
  6.1 S1: Current System for Space Heating and Cooling ..................................... 41
  6.2 S2: Optimized System for Space Heating and Cooling .............................. 43
  6.3 S3: Optimized System for Industrial Process Heat – 24/7 ....................... 45
  6.4 S4: Optimized System for Industrial Process Heat – 16/7 ....................... 46
7 DISCUSSION ...................................................................................................... 48
  7.1 Optimal Systems ......................................................................................... 48
  7.2 Impact of Assumptions .............................................................................. 49
  7.3 Optimization Potential .............................................................................. 50
8 CONCLUSION ..................................................................................................... 52
9 SUGGESTION FOR FURTHER WORK ............................................................... 54
REFERENCES ........................................................................................................ XII
APPENDICES ......................................................................................................... XVIII
Appendix A: MATLAB CODE – Solar-driven DES – Main.m .............................. xviii
Appendix B: MATLAB CODE – TESS – TES_VT_wCS.m ..................................... xx
Appendix C: MATLAB CODE – Degree-Days – DegreeDay_wC.m .................. xxv
LIST OF FIGURES

Figure 1: Scheme of an ORC unit ................................................................. 3
Figure 2: TS-diagram for water (a) and an organic fluid (b) ............................. 3
Figure 3: Layout of the 200 kW solar-driven ORC demonstration plant of Tianjin University ................................................................. 5
Figure 4: Zenith angle, slope surface azimuth and solar azimuth angle for tilted surface ......................................................... 10
Figure 5: Schematic of a parabolic trough collector ........................................ 11
Figure 6: Simplified schemes of the two-tank in- & direct and thermoclone TESS ................................................................. 13
Figure 7: Cooling (blue) and heating (orange) set point temperatures ................. 18
Figure 8: Temperature levels in industrial processes ....................................... 19
Figure 9: Customized scheme of the solar-driven ORC unit for distributed heating ................................................................. 22
Figure 10: Information flow diagram for the distributed heating system .............. 23
Figure 11: Solar radiation for Tianjin (TJ) and Beijing (BEJ) from different data sources ................................................................. 25
Figure 12: Heat demand based on the load profile and HE capacity .................. 27
Figure 13: TRNSYS - Solar field model .......................................................... 28
Figure 14: Collector efficiency and frequency of occurrence ............................ 29
Figure 15: Mass balance of the model for one year ........................................ 36
Figure 16: Heat balance of the model for one year ......................................... 36
Figure 17: Storage tank filling levels for one year ........................................... 37
Figure 18: System behavior on March 1st ...................................................... 37
Figure 19: Correlation between system efficiency and solar fraction .................. 40
Figure 20: LCOE for current system and different TESS sizes (S1) .................... 42
Figure 21: HE capacity and corresponding heated floor area for different HFOT (S1) ........................................................................ 42
Figure 22: Annual solar fraction for current system and different TESS sizes (S1) ........................................................................ 42
Figure 23: Upgrade related CO₂ savings for current system and different TESS size (S1) ........................................................................ 42
Figure 24: Efficiency for different TESS sizes (S1) .......................................... 43
Figure 25: LCOE for different HE capacities and corresponding optimized TESS size (S2) ................................................................. 43
Figure 26: Optimized TESS sizes for different HE capacities (S2) ...................... 43
Figure 27: LCOE for different TESS sizes and 750 kW HE capacity (S2) ............. 44
Figure 28: Annual solar fraction for different TESS sizes and 750 kW HE capacity (S2) ........................................................................ 44
Figure 29: Upgrade related CO₂ savings for different TESS and 750 kW HE capacity (S2) ........................................................................ 44
Figure 30: Efficiency for different TESS sizes and 750 kW HE capacity (S2) ........ 44
Figure 31: LCOE for different HE capacities and corresponding optimized TESS sizes (S3) ................................................................. 45
Figure 32: LCOE for different TESS sizes and 80 kW HE capacity (S3) .............. 45
Figure 33: Annual solar fraction for different TESS sizes and 80 kW HE capacity (S3) ........................................................................ 45
Figure 34: Upgrade related CO₂ savings for different TESS and 80 kW HE capacity (S3) ........................................................................ 45
Figure 35: Efficiency for different solar thermal systems and 80 kW HE capacity (S3) ........................................................................ 46
Figure 36: LCOE for different HE capacities and corresponding optimized TESS sizes (S4). ................................................................. 46
Figure 37: LCOE for different TESS sizes and 130 kW HE capacity (S4) .............. 46
Figure 38: Annual solar fraction for different TESS sizes and 130 kW HE capacity (S4) ........................................................................ 46
Figure 39: Upgrade related CO₂ savings for different TESS and 130 kW HE capacity (S4) ........................................................................ 47
Figure 40: Efficiency for different solar thermal systems and 130 kW HE capacity (S4) ........................................................................ 47
Figure 41: Heat demand and direct solar radiation .......................................... 49
Figure 42: Wasted (unexploited) energy from solar field .................................. 49
Figure 43: LCOE for different TESS sizes and SWH loads at 50°C lower set temperature ................................................................. 54
Figure 44: Heat load and wasted energy for S1 with and without 80 kW SWH ........ 54
# LIST OF TABLES

Table 1: Identified and investigated scenarios ................................................................. 6  
Table 2: Comparison of north-south and east-west oriented tracking systems ..................12  
Table 3: Maximum heat transfer coefficients (U-values) .....................................................17  
Table 4: Characteristics of the Chinese reference building ..................................................17  
Table 5: Degree-Days and heating and cooling load for a reference building in China ........ 25  
Table 6: Parameters of a U.S. reference office building and corresponding heat load .......... 26  
Table 7: Industrial heat demand based on HE capacity ....................................................... 27  
Table 8: Solar field parameters for TRNSYS ........................................................................ 28  
Table 9: Correlation between incidence angle of beam radiation and the IAM .................. 29  
Table 10: Operating modes of the TESS ............................................................................ 30  
Table 11: Charging and discharging conditions .................................................................. 30  
Table 12: TESS and working fluid properties ...................................................................... 34  
Table 13: Data and assumptions for the economic analysis ................................................. 39  
Table 14: Summary of scenarios, corresponding optimal system and results ....................... 41
# ABBREVIATIONS

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>16/7</td>
<td>16 hours, 7 days a week</td>
</tr>
<tr>
<td>24/7</td>
<td>24 hours, 7 days a week</td>
</tr>
<tr>
<td>A/C</td>
<td>Air conditioning</td>
</tr>
<tr>
<td>ASHRAE</td>
<td>American Society of Heating, Refrigerating and Air-Conditioning Engineers</td>
</tr>
<tr>
<td>CH₄</td>
<td>Methane</td>
</tr>
<tr>
<td>CO₂</td>
<td>Carbon dioxide</td>
</tr>
<tr>
<td>COP</td>
<td>Coefficient of performance</td>
</tr>
<tr>
<td>CSP</td>
<td>Concentrating solar power</td>
</tr>
<tr>
<td>CSWD</td>
<td>Chinese standard weather data</td>
</tr>
<tr>
<td>DES</td>
<td>Distributed energy system</td>
</tr>
<tr>
<td>GWP</td>
<td>Global warming potential</td>
</tr>
<tr>
<td>DD</td>
<td>Degree-Days</td>
</tr>
<tr>
<td>HE</td>
<td>Heat exchanger</td>
</tr>
<tr>
<td>HFOT</td>
<td>Heat exchanger hot fluid outlet temperature</td>
</tr>
<tr>
<td>IAM</td>
<td>Incident angle modifiers</td>
</tr>
<tr>
<td>IWEC</td>
<td>International weather for energy calculation</td>
</tr>
<tr>
<td>LCOE</td>
<td>Life cycle costs of energy</td>
</tr>
<tr>
<td>MoHURD</td>
<td>Ministry of Housing and Urban-rural Development</td>
</tr>
<tr>
<td>N₂O</td>
<td>Dinitrogen monoxide</td>
</tr>
<tr>
<td>ORC</td>
<td>Organic Rankine cycle</td>
</tr>
<tr>
<td>PTC</td>
<td>Parabolic trough collector</td>
</tr>
<tr>
<td>PV</td>
<td>Photovoltaic</td>
</tr>
<tr>
<td>S₁ to S₄</td>
<td>Scenario 1 to 4</td>
</tr>
<tr>
<td>SF</td>
<td>Solar fraction</td>
</tr>
<tr>
<td>SWERA</td>
<td>The solar and wind energy resource assessment</td>
</tr>
<tr>
<td>SWH</td>
<td>Solar water heating</td>
</tr>
<tr>
<td>TESS</td>
<td>Thermal energy storage system</td>
</tr>
<tr>
<td>WWR</td>
<td>Window to wall ratio</td>
</tr>
</tbody>
</table>

# NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>𝐴ₐ</td>
<td>Aperture area</td>
</tr>
<tr>
<td>𝐴ᵣ</td>
<td>Surface of building envelope</td>
</tr>
<tr>
<td>𝐴ᵢ</td>
<td>Receiver area</td>
</tr>
<tr>
<td>𝐶</td>
<td>Concentration ratio</td>
</tr>
<tr>
<td>𝐶ᵢₐ</td>
<td>Carbon content coefficient of the fuel type</td>
</tr>
<tr>
<td>𝑐ₚ</td>
<td>Specific heat capacity</td>
</tr>
<tr>
<td>𝑑</td>
<td>Diameter of the tank</td>
</tr>
<tr>
<td>𝑑ᵢₐ</td>
<td>Degradation rate</td>
</tr>
<tr>
<td>𝐹</td>
<td>Interest expenditure</td>
</tr>
<tr>
<td>̇𝐠</td>
<td>Internal heat generation</td>
</tr>
<tr>
<td>𝑡</td>
<td>Height of fluid level in TESS</td>
</tr>
<tr>
<td>𝑡ₘₐₓ</td>
<td>Maximum fluid level in TESS</td>
</tr>
<tr>
<td>𝑡ₘₐₜ</td>
<td>Minimum fluid level in TESS</td>
</tr>
</tbody>
</table>
\( H_{V_f} \) Heating value of the fuel type
\( H/D \) Height to diameter ratio
\( i \) Time step
\( I \) Initial investment costs
\( L \) Required heating rate
\( LCOE \) Life cycle costs of energy
\( M_C \) Molar mass of Carbon
\( M_{CO_2} \) Molar mass of \( CO_2 \)
\( m_F \) Combusted amount of fuel
\( m_i \) Mass flow entering TESS
\( M_{max} \) Maximum possible fluid mass in TESS
\( M_{min} \) Minimum fluid mass in TESS
\( m_o \) Mass flow leaving TESS
\( M_t \) Maintenance costs for \( t \)
\( M_{tank} \) Fluid mass in TESS
\( N \) Amount of time steps per day
\( O \) Operating costs for \( t \)
\( Q_{aux} \) Heat supplied by auxiliary heater
\( Q_{in} \) Heat flow entering TESS
\( Q_{loss} \) Heat losses from TESS
\( Q_{need} \) Heat demand
\( Q_{out} \) Heat flow leaving TESS
\( Q_{sol} \) Solar beam radiation
\( Q_{supply} \) Heat supplied by solar thermal system
\( Q_{tank} \) Heat stored in TESS
\( Q_{tank_{max}} \) Maximum possible heat stored in TESS
\( Q_{tank_{min}} \) Minimum possible heat stored in TESS
\( Q_{useful} \) Useful heat from solar field
\( r \) Discount rate
\( SF \) Solar fraction
\( S_t \) Yearly rated energy output
\( T \) Lifespan of the project in years
\( T_{amb} \) Ambient temperature
\( T_b \) Indoor base temperature
\( T_c \) Solar field inlet temperature
\( T_h \) Building indoor temperature
\( T_{in} \) TESS inlet/solar field outlet temperature
\( T_{set} \) System base temperature
\( T_{tank} \) TESS tank temperature
\( U_{dry} \) Dry heat transfer coefficient
\( U_h \) Heat transfer coefficient of building envelope
\( U_{wet} \) Wet heat transfer coefficient
\( V \) Tank volume

**Greek**

\( \beta \) Surface azimuth angle
\( \gamma_s \) Solar azimuth angle
\( \delta \) Solar declination
\( \eta_{th} \) Thermal efficiency
\( \theta_i \) Angle of incidence
\( \theta_z \) Zenith angle
\( \rho_{fluid} \) Density of the heat transfer fluid
\( \phi \) Latitude
\( \omega \) Hour angle
1 INTRODUCTION

The major developments in the energy industry in the last century have been one of the most substantial driving forces for the world’s economic progression. In the past, the constantly growing demand for energy was mostly fulfilled by increasing the amount of affordable fossil fuel based technologies at the expense of climate change. Today, the world’s energy system relies heavily on these energy carries causing not only further destruction of the ozone layer and therefore global temperature rise but also uncertainties such as unpredictable fossil fuel price developments and economic crises (Tchanche et al. 2011). It is inevitable that the earth’s fossil fuel resources will be depleted at some point making the transition towards a renewable and sustainable energy system, in order to ensure energy supply security, indispensable.

Nowadays, there are many different technologies which utilize renewable energies. Among others, solar thermal energy accounts for the most promising resource due to its wide field of different applications to meet the world’s energy demands. While Photovoltaic (PV) cells convert the incident sun rays directly into electricity, solar collectors gather the solar energy by heating a heat transfer fluid. The stored heat can then be used in small scale appliances, such as domestic water and living area heating in private households or as process heat for industrial purposes. Furthermore, the solar collector technologies can be implemented into conventional fossil fuel driven Rankine Cycles in power plants to save fuel and therefore increase efficiency. Conventional Rankine Cycles are operated at 500 – 600°C. A solar system reaching these temperatures is fairly expensive due to the relative large size and collectors with high concentration ratios. As a matter of fact, the trend is more towards smaller systems having benefits in terms of usage flexibility, site selection and financial feasibility. The temperatures of smaller system are in the range of 60 - 300°C (Kalogirou 2004) and therefore not high enough to run a conventional stand-alone solar-driven Rankine Cycle.

The so called Organic Rankine Cycle (ORC) has been successfully applied for low and medium grade heat utilization from 80 - 400°C (Georges et al. 2013) such as waste heat recovery, biomass energy and geothermal power generation as well as for the above mentioned solar energy in recent years. These solar-driven ORC units up to a few hundred kWe have significant advantages, for instance easier energy collection, storage and supply of energy locally compared to large scale solar collector power plants. Additionally, heat storage is more lucrative than battery storage which is implied by PV-technologies (Jing 2015).
1.1 Background

The following sections briefly present the technical background of the underlying initial situation of this project. The solar thermal collector technologies, the differences between the Organic and Steam Rankine Cycle are addressed. Further, the current energy regulation and legislation in China are explained in order to justify the upgrading of the solar-driven ORC unit for distributed heating purposes.

1.1.1 Solar Thermal Collectors

The idea of utilizing solar energy dates back all the way to the year 212 BC and the Greek scientist Archimedes (Anderson 1977). However, in modern times solar energy exploitation appeared in the mid-1930s with the purpose of hot water and housing heating. It gained more and more interest throughout the 40s and 50s until its final breakthrough in the early 60s when industry for manufacturing solar water heaters expanded quickly around the globe (Kalogirou 2004).

In general, a solar energy collector is nothing but a heat exchanger that converts incident solar radiation into heat which is absorbed by a circulating fluid like air, water or oil. The fluid is subsequently either directly distributed to the hot water and house heating equipment or led to a heat exchanger placed in a thermal storage tank to facilitate on demand withdrawals.

Solar collectors are categorized in stationary (non-concentrating) and concentrating types. The main characteristic of stationary collectors is that the area is the same for intercepting and absorbing the solar radiation, while sun-tracking concentrating collectors have larger intercepting and reflecting areas shaped concave in order to concentrate the incident radiation in a smaller receiving area either nearby as practiced with parabolic trough collectors (PTC) or in further distance as practiced with solar tower systems (Kalogirou 2004).

1.1.2 Steam Rankine cycle vs. Organic Rankine Cycle

Since 1985 a lot of solar plants have been built, with the most significant being in the Mojave Desert in California, USA, with a total installed power of ~354 MW (U.S. Department of Energy 2010). Those, however, rely on a rather large solar collector area in order to achieve high temperatures which are required to run a parabolic trough steam Rankine Cycle. The limitations are caused by the used working fluid water requiring steam temperatures as high as 600°C in order to prevent condensation towards the end of the expansion process. Additionally, steam turbines operating at inlet temperatures below 370°C achieve lower efficiencies and therefore become less economical. Another drawback of using solar energy is the high temperature difference being above 400°C during daytime when the system is under operation and 30°C at night. The high fluctuation puts the system under pressure due to many material expansions and contractions (Jing 2015). In order to reach high temperatures, collectors need to track the sun to increase the solar energy yield, high concentration
collectors make less use of diffuse radiation and energy storage at high temperatures is not commercially available (Pei et al. 2010). Moreover, the commonly used molten salt storage fluids for high operating temperatures have to be kept above a minimum temperature, for instance 277°C at Solana Generation Station in Arizona, to avoid solidification (Wang 2011). Consequently, the TESS needs to be equipped with an auxiliary heater causing additional expenditures.

The ORC overcomes most of the aforementioned disadvantages with its ability to utilize low grade heat with temperatures as low as 100°C and thus allowing smaller PTC apertures. In general, ORC and conventional Rankine Cycles are quite similar with some minor differences. The most significant one is the replacement of water (and steam) with an organic fluid boiling at low temperatures (Hattiangadi 2013). In contrast to Rankine Cycles, ORC units are smaller in size, are immune against freezing due to the substitution of a water- with air-condensers and require only few or no presence of personnel. Also, ORC units have lower working temperatures and volume flow ratios of the working fluid at the turbines inlet and outlet permitting the use of simpler and thus lower-priced turbines. The viability of ORC units is enhanced by the maturity of its components since being widely used in refrigeration applications (Jing et al. 2010). Further, heat storage systems at lower operating temperatures are easier to handle, consequently less complex and cheaper (Jing 2015). Overall, the combination of solar thermal collectors with the ORC technology permits smaller power and heat generation plants and therefore a better adjustment of the complete process to the requirements at the desired installation site.

**Figure 1**: Scheme of an ORC unit

**Figure 2**: TS-diagram for water (a) and an organic fluid (b)
A scheme of the ORC process is depicted in figure 1. The energy transfer from the heat source occurs in a single heat exchanger, the evaporator, where preheating, vaporization and superheating, provided that superheat is needed, take place. The vapor is then led to the turbine for expansion. Afterwards, it can still be in superheated condition due to the curve characteristics of organic fluids, according to figure 2. The fluids are categorized as wet, dry or isentropic depending on the slope, which can be positive, negative or isentropic of the corresponding saturation curve (Chen n.d.). The recuperator cools the vapor further down close to the condensation condition. In the final step, the vapor condensates in an air-cooled condenser from which the organic fluid is recirculated to the evaporator through the recuperator for preheating purposes (Hattiangadi 2013).

1.1.3 Current Energy Regulation and Legislation in China

The current energy regulation and legislation in China have a long history of development. With the establishment of the People’s Republic of China, the energy sector was subject to the planned economy model meaning that the monthly energy production was controlled and allocated. As of the 1980s, the government has passed many reforms to make a transition from a ‘government controlled’ to a ‘market plus government regulated’ industrial model. Among others, the Renewable Energy Law has been enacted in 2005 and revised in 2009 as a reaction of the increased political attention caused by high pollution, climate change and high energy consumption to promote the utilization of renewable energies (Qiu & Li 2012). These measures led China to become the world largest photovoltaic and solar thermal industries. The latter’s main products are solar thermal collectors for water heaters with an annual growth rate of about 20%, accounting for more than half of the world’s output. By 2008, nearly 60% of the total global installations were made in China meaning that China’s solar thermal industry is mainly driven by its domestic market (UNEP.org 2010; Li 2012).

The solar thermal industry benefited from guidance by the government, which set the direction of development for renewable energies. This measure provided the necessary safety for enterprises to invest in the field. Nevertheless, the renewable energy goals specified in the first Renewable Energy Law in 2005 were low with little support effort and insufficient promotion. The revise of the law in 2009 changed this and the Chinese government released several policies to enhance the use of renewable energy. From 2009 on, companies dealing with solar thermal technologies were accepted to become a member of China’s Consumer Electronics Association. Further, solar water heaters were included for the first time in subsidy policies. Aside from financial policies, China’s government has set out the goals of providing 15% energy from renewable sources as well as reducing carbon dioxide emissions by 40 – 45% by 2020, compared to 2005 (Li 2012).

Despite these efforts, the country’s energy sector is still undergoing many reforms resulting in lack of awareness and enforcement of these laws (Wang 2006; Li 2012). The regulatory authority is still not acting independently and influenced by the political authority. Moreover, about seven departments are in charge of energy related topics leading to uncertainties concerning the areas of responsibility and therefore more complicated licensing procedures (Qiu & Li 2009). Xiaohua Li suggests in his study Development characteristics of the solar...
energy industry and related policies in China several adjustments to policies to promote China’s solar industry. The key recommendations are as follows:

- Improve goals for solar energy utilization
- Speed up the relevant legislation
- Strengthen the incentives for solar energy utilization
- Provide financial and tax support for solar energy utilization

1.1.4 Project Status and Plant Layout

Tianjin University in China is working on a 200 kW solar-driven ORC demonstration plant for the new university research institute in Binhai District. The goal of this project is to establish a distributed energy system (DES) which can be operated for research purposes involving the ORC unit and electricity generation. At this early stage, parts of the power plant, in particular the solar thermal system, which includes a 1096 m² solar field, 2.4 MW auxiliary heater and 250 kW heat exchanger (HE), for the distributed heating cycle, as well as the piping and vessels, have been purchased and installed according to the layout depicted in figure 3. The design conditions for the 250 kW counter flow HE are 45°C inlet and 55°C outlet temperature for the cold fluid and 120°C inlet and 100°C outlet temperature for the hot fluid. The ORC cannot be operated at this current stage due to a missing turbine and generator.

The ORC can either be operated with or without the solar field depending on the availability and amount of solar radiation. If the latter is the case, then the solar field is simply bypassed and the working fluid is pumped directly to the auxiliary heater, where it is heated and subsequently transferred to the ORC HE. At times with sufficient solar radiation, the working fluid is pumped through the solar field to reduce the auxiliary heater’s fossil-fuel consumption. In the current setup, the distributed heating cycle can only be operated in conjunction with the solar field, the auxiliary heater is bypassed.

![Figure 3: Layout of the 200 kW solar-driven ORC demonstration plant of Tianjin University](image-url)
1.2 Objective and Research Questions

Upon completion of the DES, it will not be connected to the grid due to restrictions by Chinese law and difficulties with the licensing procedure as mentioned in 1.1.3 Current Energy Regulation and Legislation in China. For that reason, Tianjin University intends to upgrade the solar-driven ORC unit in order to supply heat to either the local scholar community or nearby industry whenever the plant is not used for power generation. This measure will utilize the harvested solar energy effectively.

Within this framework, the DES is to be upgraded by including a properly sized thermal energy storage system (TESS) to provide heat and cooling to the scholar community during the winter and summer or heat to some industry process throughout the year. In common commercial projects, the solar power plant should be designed to fit the beforehand evaluated and designated utilization purpose in the most cost-effective way before it gets constructed. This study differs from the common procedure in that the plant has been built for other purposes in respect with the ORC unit and should now be upgraded for heating purposes for times without ongoing experiments. Therefore a heating purpose is sought to fit the plant’s specifications most cost-effectively. The considered upgrades were (1) the implementation of a TESS and (2) resizing of HE capacity for the distributed heating cycle. Further, the hot fluid outlet temperature (HFOT) of the HE was reviewed. For this reason, four utilization scenarios have been defined (table 1).

<table>
<thead>
<tr>
<th>Table 1: Identified and investigated scenarios</th>
</tr>
</thead>
<tbody>
<tr>
<td>TESS size</td>
</tr>
<tr>
<td>TESS size (5 m³ interval)</td>
</tr>
<tr>
<td>HE resizing</td>
</tr>
<tr>
<td>HE HFOT</td>
</tr>
<tr>
<td>Heating purpose</td>
</tr>
</tbody>
</table>

Scenario 1 (S1) represents the current system setup. None of the purchased components has been upgraded. Solely the HFOT of the HE has been altered with different TESS sizes to study the performance of the system for the current and original design when used for space heating and cooling of the scholar community. Scenario 2 (S2) is similar to S1 with the difference that the heat supply is increased by resizing the HE to utilize more of the harvested solar energy. The heat demand in S1 and S2 still remains very low during spring and autumn season when neither heating nor cooling is required leading to large amounts of not utilized solar energy. For this reason, an industrial process heat demand according to the HE size was applied for scenario 3 (S3) and 4 (S4). Many industries produce in a three working shift system 24 hours, 7 days a week (24/7) and thus a constant heat demand throughout the day and year according to the HE capacity has been applied in S3. A possible candidate in the near vicinity of the plant, which could use the harvested solar energy, works in a two shift system 16 hours, 7 days a week (16/7). To this end, a constant heat demand 16/7 has been used in S4 to investigate the cooperation potential.
The purpose of this thesis, documented in this report, is to conduct a performance analysis of the upgraded system for different utilization possibilities.

Finally, this implies the following concrete subtask:

1) Assessment of the meteorological data for the location Tianjin, China
2) Estimation of space heating and cooling loads
3) Estimation of industrial heating loads
4) Modeling and verification of the system
5) Definition of the performance indicators

1.3 Scope and Limitations

Prior to this thesis work, decisions had been made in terms of component sizing and design parameters such as temperature levels and fluids used in the system. As a matter of fact, most components have been purchased and installed already. Therefore, the investigated options for upgrading the system only involve the TESS and HE. Extending the solar field would require bigger pumps to cycle the heat transfer fluid. This in turn implies higher volume flows and therefore a replacement of all pipes, vessels etc. with larger ones. The latter is seen as not practical and hence neither investigated nor considered in this study. In the current layout of the plant, the auxiliary heater is bypassed when operating the HE for the distributed heating cycle, see figure 3. In the further course of this work, it is assumed that the auxiliary heater can also supply heat to the distributed heating cycle. For that purpose, only minor changes to the piping layout have to be made. The power plant’s site itself has enough space for additional components. However, the space inside the current boiler building is not sufficient. Still, in order to save ground area it is assumed that the TESS tank is standing upright. This study focuses on the preliminary sizing of the TESS, HE and HFOT for the four scenarios. Specific design parameters as well as placement, installation, additional buildings etc. are thus not subject of the report. A fixed exchange rate from July 1\textsuperscript{st}, 2015 of 6.08 for the Chinese Yuan Renminbi against the US dollar was used throughout this thesis work.

1.4 Thesis Outline

Chapter 2, methodology, describes and justifies the applied methods to accomplish this thesis. Chapter 3 presents a literature review on the fundamentals of solar radiation, the key components in solar-driven distributed heating units and the theory on heating load estimation as well as an overview about different simulation software used in connection with solar systems. In Chapter 4 the modeling and validation of the model is explained. The structure of the entire model is given first. The chapter proceeds with evaluation of the meteorological data, heating load estimation, the modeling of the solar field, TESS and auxiliary heater before it concludes with the validation of the entire model. The report continues onto Chapter 5 in which the economic, technical and environmental performance indicators are defined. Based on these, the simulation results are presented in Chapter 6 and discussed in Chapter 7. Conclusions are drawn in Chapter 8 and suggestions for further work are given in Chapter 9.
2 METHODOLOGY

The goal of this thesis has been to identify a heating purpose and corresponding system upgrades leading to the most cost-effective use of the already purchased components. In this context associated literature was studied in order to reflect (1) the state of the art technology regarding the key components of such a system, (2) the methods to estimate heating and cooling loads of office buildings and industrial processes and (3) common used software for modeling and simulation of solar thermal systems. The study for heating and cooling load estimation in office buildings included the Heat Balance and Degree-Day (DD) method. The Heat Balance method requires detailed information about the buildings properties such as insolation type, floor area, occupation etc. This information was not available meaning that the heating and cooling load estimation for office buildings has been determined with the DD method, which requires solely the ambient air temperature and the current Chinese building standard. The obtained heat and cooling profile from this method was simplified to represent the heat that needs to be transferred from the HE. Additionally, the profile was scaled so that the annual maximum heat demand corresponded to the HE capacity. In this way, the solar system could supply as much heat as possible. Finally, the programs TRNSYS, WATSUN and POLYSUN, for simulating solar thermal systems, and MATLAB, for solving general mathematical problems, have been studied in terms of their applicability for the underlying research questions.

Subsequent to the literature review, the system modeling and validation have been carried out. The modeling and simulation of the meteorological data and PTC have been performed with the simulation software TRNSYS due to its ability to process different weather data types and the availability of a detailed component for PTC given through the TESS extension library. The modeling and simulation of the heating load with the DD method, the TESS and auxiliary heater have been realized with MATLAB. It has decisive advantages in respect of flexibility and stability of the calculation compared to TRNSYS.

For the four studied scenarios, the different parameters have been varied for the design evaluation and the results have been subsequently saved into EXCEL for assessment. The results were analyzed in terms of economic, technical and environmental performance. The economic performance is paramount for the utilization of the solar system. Therefore, the setup, leading to the best economic performance, is presented in the results. The associated trends of the technical and environmental performance for the determined setup are also depicted.
3 LITERATURE REVIEW

The utilization of solar radiation provides the foundation for all solar applications. A brief review of the fundamentals is essential for further understanding and therefore subject of the first section. The following section covers the theoretical background of key components in a solar-driven distributed heating unit, in particular solar collectors, TESS, auxiliary heater and heating cycle. For the latter, a reflection on the theory about heating load estimations is of substantial importance. The chapter ends with an overview of programs suitable for modeling and simulating solar thermal systems.

3.1 Fundamentals of Solar Radiation

Simply speaking, solar radiation is a result of a permanently ongoing fusion reaction in the core of the sun. The released energy in the center is transferred by conduction and convection to the outer layer, the so called photosphere, from which it is finally emitted in a continuous spectrum of radiation (Duffie & Beckman 2013).

The extraterrestrial radiation is the part of solar radiation intercepted by the earth’s outer atmosphere and is on average about 1367 W/m². While the radiation passes through the atmosphere, some parts get absorbed by molecules or reflected back to space, others reach the earth’s surface after being scattered by molecules and particles, whereas the remaining part is not scattered at all. The latter two are referred to as diffuse radiation and direct solar radiation or beam radiation, respectively. The amount of radiation reaching the surface depends also on the latitude, longitude and height of the location as well as other weather factors such as season of the year, cloud cover, air pollution etc (Duffie & Beckman 2013; Wald 2009).

For the use of solar collectors the understanding of different sun-earth angles is of importance. These are defined by Tiwari & Dubey (2010) as follows and depicted in figure 4.

- The zenith angle $\theta_z$ is the angle between the sun’s rays and the zenith direction. The angle becomes smallest during noon and is 90° at sunrise and sunset.
- The solar azimuth angle $\gamma_s$ describes the location of the sun in respect to south. When it becomes 0° the sun is shining from geographic south in the northern hemisphere.
- The solar declination $\delta$ (not shown in figure 4) is the angle between the sun and the equator.
- The hour angle $\omega$ describes the rotation of the earth to bring a location directly under the sun. It is maximum negative at sunrise, zero at noon and maximum positive at sunset.
• The surface azimuth angle $\beta$ is a measure for the incline of a surface compared to the horizontal. It is of major importance when the solar energy yield by a surface is to be increased throughout the day. The angle can be fixed or variable.

The angle of incidence $\theta_i$ is calculated by means of the latitude $\phi$ of the location with equation 1.

$$
\cos \theta_i = (\cos \phi \cos \beta + \sin \phi \sin \beta \cos \gamma) \cos \delta \cos \omega + \cos \delta \sin \omega \sin \beta \sin \gamma + (\sin \phi \cos \beta - \cos \phi \sin \beta \cos \gamma) \sin \delta
$$

Equation 1

For this thesis work meteorological data was available which provides recorded values for the above mentioned angles. Further explanations of the underlying calculation method are therefore not expedient and reference is made for more information on the topic to Tiwari & Dubey (2010) and Kalogirou (2009).

3.2 Key Components in Solar-driven Distributed Heating Units

The solar collector field is the centerpiece of any solar-driven application. The decision for PTC has already been made prior to this study. As a consequence, only a literature review on these will be presented below. For additional information on other collector types such as flat-plate, compound parabolic, evacuated tube collectors and linear Fresnel or parabolic dish reflectors refer to Duffie & Beckman’s comprehensive synopsis Solar Engineering of Thermal Processes (2013) and Kalogirou’s summarized study Solar thermal collectors and applications (2004).

The two-tank thermal and thermocline energy storage systems as well as the auxiliary heater are studied in section 3.2.2 and 3.2.3, respectively.
3.2.1 Parabolic Trough Collectors

Parabolic trough collectors (PTC) belong to the group of concentrating collectors. Unlike stationary collectors, concentrating collectors use an optical device to focus solar radiation on a smaller absorption surface which results in delivering energy at higher temperatures. This implies that processes with concentrating collectors work at higher thermodynamic efficiency than processes with stationary, non-concentrating, collectors, while both have the same solar absorption surface. Further, less material is needed due to large reflecting areas but only small absorption surfaces. At the same time, however, the production of reflecting components requires higher precision to ensure the focus of solar radiation in one line. Moreover, the higher the concentration ratio is the less diffuse radiation can be gathered. Since focusing of solar radiation requires a correct orientation towards the sun the collectors need to be equipped with a sun tracking system which could throw shadows onto the reflecting area. The tracking system raises the investment costs for concentrating collectors compared to stationary collectors. Last but not least, the reflecting area needs to be cleaned regularly from dirt in order to guarantee good reflectance (Duffie & Beckman 2013; Kalogirou 2004).

PTCs usually reach temperatures between 50 and 400°C (Kalogirou 2009). The sun rays are focused by shaping the reflective surface, the so called parabola, parabolic as depicted in figure 5.

![Figure 5: Schematic of a parabolic trough collector (Kalogirou 2004)](image)

The receiver is composed of a metal tube in the middle with black coding to increase the absorption. A glass tube surrounds the metal tube to reduce heat loss, however, it also reflects some sun rays coming from the parabola thus adding some transmission losses. The relationship of the aperture’s area $A_a$ to the receiver’s area $A_r$ is defined as the concentration ratio $C$ and is one of the essential characteristic of PTCs (equation 2). This value determines, to a great extent, the temperature of the produced heat. The upper theoretical ratio for concentrators, in general, is limited due to the fact that the incident solar beam is not entirely parallel but rather forms a cone with an angle of about 0.5°. Hence, the receiver must be of a certain size, since the reflected beam is also a cone, to gather most of the reflected solar radiation, see Rabl's (1976) *Comparison of Solar Concentrators* for detailed information about the topic.

$$C = \frac{A_a}{A_r}$$  \hspace{1cm} Equation 2
In practice, collector arrays are quite long and therefore the usage of a single axis tracking system is sufficient enough, because solely minor concentrating losses occur at the end of the collector array. The orientation of the system highly depends on the application. Table 2 summarizes the advantages and disadvantages of a north-south and east-west oriented tracking system. Furthermore, aside from tracking the sun itself, the system needs to be able to orient the collectors at the end of the day back to its morning position and to continuously track the sun even during periods of cloud cover. A safety mechanism prevents the collector field from operating in dangerous working conditions such as wind gust, overheating etc. by turning the collectors out of focus (Kalogirou 2004).

Table 2: Comparison of north-south and east-west oriented tracking systems

<table>
<thead>
<tr>
<th>Advantages</th>
<th>East-West</th>
</tr>
</thead>
<tbody>
<tr>
<td>Higher collection during morning and evening hours</td>
<td>Little collector adjustment throughout the day</td>
</tr>
<tr>
<td>Overall energy collection in one year is higher than with an east-west tracking system</td>
<td>Collectors always fully face the sun at midday</td>
</tr>
<tr>
<td>Collects more energy during winter, this results in less annual fluctuation</td>
<td>Collects more energy during summer</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Disadvantages</th>
</tr>
</thead>
<tbody>
<tr>
<td>At midday the collectors performance is the lowest due to unfavorable orientation towards the sun</td>
</tr>
<tr>
<td></td>
</tr>
</tbody>
</table>

3.2.2 Thermal Energy Storage Systems

Thermal energy storage systems (TESS) have a decisive advantage when it becomes a matter of overcoming fluctuation in energy production due to sunset and clouds. In general, solar energy is stored in and discharged from the tank to ensure an as uniform as possible electricity production or heat availability (SkyFuel 2012). The charging process in detail depends on the TESS technology, but the overall principle is the same. During the day, excessive heat from the collectors which cannot be used for electricity generation or heat applications is used to charge the tank. The stored heat is then discharged during cloudy periods or evening hours for continuously electricity generation and heat supply (Ravaghi-Ardebili et al. 2013).

The two-tank indirect system, two-tank direct system and the single-tank thermocline system use sensible heat storage and have emerged as the most significant TESS technologies over the past few decades. The storage medium is circulated actively through all the mentioned systems (Libby 2010; U.S. Department of Energy 2013). The former two-tank indirect system distinguishes itself from two-tank direct system by having a separated, independent heat storage circuit, see figure 6. For this reason, the indirect system is deemed to be the technology with lowest risk. However, this system entails also a non-negligible disadvantage. Two heat transfers, one for charging and one for discharging the storage tank, are necessary. Both require a certain temperature difference in the heat exchanger, meaning that the fluid temperature is 10-20°C lower when the system is operated from the TESS compared to being
operated from the solar collectors. This leads to a storage cycle efficiency of about 93% which, in turn, results in a lower overall system efficiency when operated from the two-tank indirect TESS (Libby 2010).

![Diagram](image)

*Figure 6: Simplified schemes of the two-tank in- & direct and thermocline TESS, adapted from Cocco & Serra (2015)*

The two-tank direct TESS, depicted in the middle of figure 6, uses the same fluid in the storage tank and collector field and does not require an additional heat exchanger. The storage is charged by pumping fluid from the cold tank through the solar collector field to the hot tank and discharged by circulating the fluid from the hot tank through the heat consumer, namely ORC or distributed heating unit, and finally back to the cold tank. With such a setup it is possible to charge and discharge at the same time when the available solar energy is higher than the required energy of the heat consumer. Two-tank direct TESS stipulate a relatively large amount of working fluid in order to also ensure heat storage. The usage of molten salt as a heat transfer and storage medium is inexpensive compared to synthetic oils hence vital to maintain the financial competitiveness. The disadvantage of molten salt is that it needs to retain a relative high temperature, around 270°C, to avoid solidification of the fluid in any part of the system. Both two-tank TESS are ultimately better suitable for larger applications where solar collectors reach temperatures above the molten salt freezing point (Libby 2010).

The thermocline TESS can either be designed as an indirect or direct single-tank system, see figure 6. The hot and cold fluids are both present in one tank and separated by the thermocline, hot in the top and cold in the bottom part. Further, most of the storage fluid is substituted with less expensive filling material. The charging simply takes place by introducing hot fluid at the top of the tank. It then transfers heat to the filling material by flowing through it to the bottom of the tank where it eventually is returned to the cycle for reheating. Discharging takes place in the opposite way. This method forms a thermal gradient throughout the tank. The temperature of the cold fluid leaving the tank rises with progressive charging which lowers the performance of the solar collector. Likewise, the temperature of the hot fluid, leaving the tank in discharging mode, declines which affects the heat consumer's performance (Libby 2010).
3.2.3 Auxiliary Heater

Auxiliary heaters operated with biomass, fossil fuels, electricity etc. are usually integrated into solar-driven power plants to guarantee operation at the desired capacity even during periods with low solar radiation and at night as well as when the TESS is discharged. These plants are typically designed to run 100% on solar energy, 100% on the auxiliary heater or any combination of both. However, the achieved efficiency in ORC units with fossil-fuel driven auxiliary heaters is lower than in conventional steam cycles and is therefore only used to cover the demands. The lower heat grade available from, for instance, waste heat recovery is more suitable in combination with ORC units. The type of auxiliary heater in distributed heating applications is from a technical perspective rather insignificant and depends solely on the availability of fossil-fuels and pricing (U.S. Department of Energy 2010).

3.2.4 Solar-Driven Distributed Heating Unit

Solar collectors are used in a variety of applications including heating. A pump circulates the heat transfer fluid through the collectors when there is enough solar energy available. Subsequently, and depending on the unit design, the heated fluid is stored in a tank or supplied to the heat exchanger of the heating cycle. It should be noted that heat can be supplied directly from the collector to the distributed heating unit by using the same heat transfer fluid or indirectly by using a different heat transfer fluid and separated collector cycle. Since the latter is the case for the studied system, direct systems are set aside and are no longer subject of the further review. For more information on that matter, reference is made to Kalogirou's Solar thermal collectors and applications (2004).

The heating load is specified by many factors such as the size of the building and its purpose of the underlying industrial process. Although, the heating load could be satisfied with solar energy only, it would result in an oversized system for most parts of the year, making the system less viable hence the use of auxiliary heating inevitable. The correct sizing of the system is mainly determined by the LCOE described in section 5.1 Economic Indicator.

When combining the solar heating system with a TESS and auxiliary equipment equal levels of comfort, temperature stability and reliability can be reached compared to conventional systems (Kalogirou 2004). In such a system, five basic operation modes, depending on the system conditions at a certain time, are considered. Duffie & Beckman (2013) specify them as follows:

1. If solar energy is available and heat is not needed, energy gain from the collector is added to TESS.
2. If solar energy is available and heat is needed, energy gain from the collector is used to cover the heat demand.
3. If solar energy is not available, heat is needed, and the TESS has stored energy in it, the stored energy is used to cover the heat demand.
4. If solar energy is not available, heat is needed, and the TESS has been depleted, auxiliary energy is used to cover the heat demand.
5. If the TESS is fully charged, there are no heat demands to meet, and the collector is absorbing heat.
For the last case, a safety system, such as pressure relief valves or a tracking mechanism that rotates the collectors out of focus, needs to be employed to prevent the system from getting damaged. It is also possible that a combination of two basic modes is applied when, for instance, heat from the collector is available but not sufficient to cover the energy needs of a building or process, meaning that additional auxiliary heating is required. In addition to solar heating, solar cooling for buildings further extends the utilization of the solar-driven heating unit and consequently lowers its LCOE. The utilization is favored even for systems without any storage technology due to the solar cooling loads being in phase with the availability of solar radiation (Kalogirou 2004).

3.3 Heating Load Estimation for Space Heating and Cooling

The load of a distributed heating system is influenced to a great extent by the climate, the building properties such as the heat transfer coefficient from the building to the surrounding but also by the human behavior and temperature comfort levels etc. The lack of this information makes the thermal load calculations complex. Various calculation methods have therefore been developed to estimate heating loads. The following subsections outline the approach.

3.3.1 Heat Balance Method

Results with sufficient precision can be obtained by using heat source, heat gain and heat loss calculation concepts based on transient heat transfer analysis, commonly referred to as ‘The Heat Balance Method’ (Kalogirou 2009; ASHRAE 2001). The method is based on the fundamentals of thermodynamics meaning, that all energy flows must be balanced. Heat gains, losses and transfers due lights, computers, humans, solar radiation, outside temperature etc. are therefore considered. However, this method requires a great deal of information about the building’s design and purpose. A detailed explanation of the method is not expedient due to lack of this information and, moreover, would be beyond the scope of this thesis. For further discussion of the heat balance method the reader is referred to Kalogirou’s Solar Energy Engineering: Process and Systems (2009), in which equations and correlations are briefly described, and to the ASHRAE Handbook of Fundamentals (2001) which gives a detailed explanation about the method as well as tables and figures with reference data for heat emission of lights, computers, humans and other domestic appliances.

3.3.2 Degree-Day Method

The degree-day method, summarized by Duffie & Beckman in Solar Engineering of Thermal Processes (2013), can be used if the aforementioned details are not known, as it is the case for the current project status, in order to estimate the heating and cooling load of a building by taking its ambient temperature, purpose and the building standards into account. The method is based on the principle that energy loss from a building are considered to be proportional to the temperature difference between in- and outdoors, see right hand side of
Further, it considers the rate of internal heat generation $\dot{g}$ from humans, lights and other appliances leading to the required heating rate $\dot{L}$ at which the building needs to be heated to maintain its indoor temperature. The following equations were compiled from Duffie & Beckman (2013) and Dubin & Gamponia (2007), and reference is made to the corresponding literature for a complete derivation.

$$\dot{L} + \dot{g} = (UA)_h * (T_h - T_{amb})$$  \hspace{1cm} \text{Equation 3}

If the building is heated to a desired indoor temperature $T_h$ then there exists an indoor base temperature $T_b$ at which the internal heat generation compensates the losses and no heating is required. Consequently,

$$\dot{L} = (UA)_h * (T_b - T_{amb}) .$$  \hspace{1cm} \text{Equation 4}

Finally, the heating load for a desired time span is obtained by integrating all cases where the ambient temperature is lower than the base temperature thus contributing to the heating load. The superscript ($^+$) denotes that only the positive part of the integral is considered.

$$L = (UA)_h * \int (T_b - T_{amb})^{+} \, dt$$  \hspace{1cm} \text{Equation 5}

Since temperature data is usually given as hourly averages, the load at a specific time step $L_i$ can be approximated with sufficient precision by using

$$L_i = (UA)_h * DD_i$$  \hspace{1cm} \text{Equation 6}

with

$$DD_i = \frac{1}{N} \sum(T_b - T_{amb_i})^{+}$$  \hspace{1cm} \text{Equation 7}

where $DD_i$ is the degree-days in °C-day, $T_{amb_i}$ is the average ambient temperature of the time step $i$ and $N$ the total amount of time steps per day.

The aforementioned equations are also suited to estimate cooling loads if the ambient temperature stays mainly above room temperature. These loads arise as a consequence of energy entering the building through its envelope, as solar gains through windows and ventilation of warmer outside air (Duffie & Beckman 2013).

### 3.3.3 Chinese Building Standard

The Chinese government has focused more and more on the buildings energy efficiency since the mid-1980s. The Ministry of Housing and Urban-Rural Development (MoHURD) is responsible for developing the Chinese national building energy codes. The MoHURD revised the GB50189 - Design Standard for Energy Efficiency of Public Buildings in 2005 mandating an efficiency improvement by 50% compared to the underlying baseline defined by the 1980s building characteristics (MoHURD 2005). It must be mentioned that the
MoHURD is currently updating the 2005 standard with the ambition of enhancing the energy efficiency by 30% compared to 2005 (Feng et al. 2014). The updated 2014-standard is not active at the current stage of the thesis but it is suspected to become valid without any further changes and will therefore be applied for the further estimation of the heating and cooling load.

An extract of the maximum allowed heat transfer coefficients for the different parts of a building located in China’s cold zone, as it is the case for Tianjin, which will be designed based on GB50180-2014 code, is depicted in table 3.

| Table 3: Maximum heat transfer coefficients (U-values), in extracts from MoHURD (2014) |
|---|---|---|
| Tianjin, China (Cold Zone) | $SC^{1} \leq 0.3$ | $0.3 < SC^{1} \leq 0.4$ |
| Roof | $\leq 0.45$ | $\leq 0.4$ |
| Wall | $\leq 0.5$ | $\leq 0.45$ |
| WWR | $WWR \leq 0.2$ | $\leq 3.0$ | $\leq 2.8$ |
| | $0.2 < WWR \leq 0.3$ | $\leq 2.7$ | $\leq 2.5$ |
| | $0.3 < WWR \leq 0.4$ | $\leq 2.4$ | $\leq 2.2$ |
| | $0.4 < WWR \leq 0.5$ | $\leq 2.2$ | $\leq 2.0$ |
| | $0.5 < WWR \leq 0.6$ | $\leq 2.0$ | $\leq 1.8$ |
| | $0.6 < WWR \leq 0.7$ | $\leq 2.0$ | $\leq 1.8$ |
| | $0.7 < WWR \leq 0.8$ | $\leq 1.8$ | $\leq 1.6$ |
| | $0.8 < WWR \leq 1.0$ | $\leq 1.5$ | $\leq 1.4$ |
| Floor | | $\leq 1.5$ |

1) The shading coefficient describes the amount of heat passing through a material compared to a sheet of clear float glass

The lack of typical parameters such as the building’s shape, number of floors, window-to-wall ratio (WWR) etc. in China had induced Feng et al. (2014) to survey these by examining existing buildings as well as design drawings of new constructions from Chinese design institutes. A reference building based on the Chinese standard is crucial to compare simulation results. The findings and additional information are shown in table 4.

| Table 4: Characteristics of the Chinese reference building |
|---|---|
| Layout | $30 \text{ m} \times 50 \text{ m}^{1}$ |
| Height | $85 \text{ m}^{2}$ |
| WWR | $0.4^{3}$ |
| Floors | $18^{4}$ |

1) Feng et al. (2014) 2) CTBUH (2015)

Further, the GB50180-2014 code advises a set of building operation conditions and schedules such as the heating and cooling set points throughout the day, presented in figure 7.
3.4 Heating Load Estimation for Industrial Processes

In industrial applications the heat demand depends to great extent on the process itself. Still, most industrial processes are continuous, meaning that the demand is quite constant or, at least, assessable throughout the day thus making the heating load estimation more accurate and reliable compared to load estimations for space heating and cooling, see 3.3 Heating Load Estimation for Space Heating.

Figure 8 shows the frequency of occurrence for temperature levels in different sectors. Food and tobacco, chemical and pulp and paper industries are the largest energy consumers with a high share of low and medium grade heat demands (Werner 2006). For most applications below 100°C flat plate collectors are the better option. However, in the temperature range of 130 – 200°C lie many industrial processes. To take one example, the pasteurization process in the dairy and food industry requires heating to kill most bacteria in order to avoid spoilage of products. This pasteurization takes place at temperatures between 72°C and 140°C (U.S. Department of Health and Human Services 2009; Tortora et al. 2012). For these types of processes, concentrating collectors are able to supply heat in the same quality as industrial boilers and therefore often used in parallel or serial to backup boilers (Minder 2013; ESTIF 2006).

The determination of heating loads for industrial processes is quite different compared to those for space heating. The heat demand can simply be determined by knowing the underlying industrial process and the heat requirements for the certain process steps for which the solar heating unit is to be designed. If an existing factory is to be upgraded with a solar system then preliminary measurements can be performed to increase accuracy. In general, the load estimation for industrial processes is more accurate and reliable than for space heating and cooling due to factors such as outside temperatures, building insulation, human occupation etc. having less influence.
3.5 Modeling Solar Thermal Systems

Obtaining the optimal setup of a system for the best performance is nowadays more important than ever before. When modeling a system many factors such as, component properties (predictable) and weather data (unpredictable) are considered. Kalogirou summarized in Solar thermal collectors and applications (2004) the advantages in conjunction with modeling of solar thermal systems as follows:

- Elimination of expenses for building prototypes
- Organization of complex systems in an understandable format
- Comprehensive understanding of system operation and component interactions
- Optimization of system components
- Estimation of energy delivered by system
- Change of design variables and investigation of their influence on the system performance

The modeling process starts with describing a simplified structure of the studied system. The outcome of this step strongly depends on the underlying purpose of the study. For instance, optimizing a component requires much greater detail of the physical properties described in the model than performing a preliminary design study. Therefore the knowledge about the simulation purpose allows for the definition of the system boundaries and the structure of the system. The sections hereinafter briefly outline the three commonly used programs TRNSYS, WATSUN and Polysun for simulations with solar thermal systems and MATLAB for solving general mathematical problems. For a deeper insight into the various tools suited for solar thermal system simulations reference is made to Kalogirou's book Solar Energy Engineering: Process and Systems (2009).
3.5.1 TRNSYS

The program TRNSYS (transient system simulation) is developed by the University of Wisconsin by the Solar Energy Laboratory. The advantages of the program come with the many supplied mathematical descriptions given as ordinary differential or algebraic equations for different components of solar heating systems. These subroutines are called proformas or types in TRNSYS and linked in any desired way to represent the studied system. The overall problem is simplified to a problem where only the components that are included in a system have to be identified and subsequently added to the model. Finally, the information flows (links) from one component to another, the parameters of the components as well as simulation parameters such as step size are specified (Ayompe et al. 2011).

The degree of accuracy has been studied by creating several models representing real physical systems in TRNSYS. The result of this validation was that the simulation provides a degree of accuracy with a mean error below 10% compared to actual measurements from the operated system (Kreider & Kreith 1981).

3.5.2 WATSUN

The program WATSUN is developed by the Watsun Simulation Laboratory of the University of Waterloo in Canada. The user provides hourly based weather data for the studied location and information about the collection and storage system to calculate the systems state for every hour. The program is beneficial when it comes to long-term performance and economic analysis to assess the viability of the used solar energy system. It is placed right between the spreadsheet-tools for a quick assessment and the complex simulation tools with a high degree of flexibility (Kalogirou 2009).

The program itself has been validated by comparing the results obtained in WATSUN to the ones from TRNSYS for several test cases. The differences in prediction of the yearly energy gain for all studied systems were less than 1.2% (Kalogirou 2009).

3.5.3 POLYSUN

The POLYSUN program helps to optimize dynamic annual simulations of solar thermal systems. A comprehensive user interface allows the clear and comfortable setup of the model as well as input of all system parameters. The description of the simulation relies entirely on physical models and no empirical correlation terms are therefore required. Further, the software provides templates for all components and allows the user to edit in order to fit the requirements of a specific product (Kalogirou 2009). The presentation of results in POLYSUN is clear, structured and gives a professional impression. It is, therefore, suited and favored in businesses which provide solar system solutions to customers rather than in research departments with general performance analysis.

An accuracy within 5 – 10% for results obtained with the POLYSUN program has been found by Gantner (2000).
3.5.4 MATLAB

The program MATLAB (matrix laboratory) is developed by the company *The MathWorks, Inc.* for solving primarily numerical mathematical problems and visualizing the corresponding results. The possibility of manipulating matrices, plotting functions and data, implementation of algorithms and compatibility with other programming languages makes it very flexible and suitable for different applications.
4 SYSTEM MODELING AND VERIFICATION

The following subsections explain the structure of the distributed heating model, its key components and, where applicable, the underlying physical and mathematical foundations. It also covers the validation of the results from the sub-models as well as the entire solar distributed heating model.

4.1 Structure of the Model

The original layout of the solar-driven ORC plant (figure 3) needs adjustment in order to use the available auxiliary heater also for distributed heating. It is therefore necessary to relocate the junction for the HE piping of the distributed heating cycle from before to after the auxiliary heater, figure 9. This measure will improve the quality and secure the availability of heat. Further, a two tank TESS has been included into the scheme as a possible upgrade. The underlying operation procedure of the entire plant remained nearly the same. At times with sufficient solar radiation, the working fluid is pumped through the solar field to the hot tank. From there it is pumped to the HE by either passing through the auxiliary heater if auxiliary heating is required, or by bypassing it if the supplied heat by the solar field and TESS is sufficient. Finally, the fluid is circulated back to the cold tank.

Note: Figure 9 only depicts the principle layout of the investigated and upgraded plant. Security feature such as overflow protection, pipes for component bypassing etc., are not shown. The ORC unit can still be operated, but is not subject of this study and will therefore not be included in the model.

The distributed heating model has been divided into two sub-models. The results were processed in Microsoft EXCEL, see figure 10. The PTC was modeled in TRNSYS since the
software already provides its mathematical description (Type 536) as well as the necessary proforma for processing weather data (Type 15-3). The results, in particular the useful energy production from the PTC and ambient air temperature, were saved to a txt-file (Type 56a) and subsequently imported manually to MATLAB. This procedure is to be favored over an automatic import due to the fact that the same weather file and PTC parameters yield the same useful energy production each time the simulation is carried out thus saving calculation time. A detailed description of the solar field model is given in section 4.4 Solar Field.

The results from TRNSYS are imported to MATLAB by using the implemented function ‘import.data’. All variables are available globally thus being accessible to any function. The heat demand depends on the studied scenario and is either defined by the Degree-Day method (function ‘DegreeDay_wC’) for space heating or set as a fixed value for process heat. The determination of the space heating and cooling load is explained in section 4.3.1. The maximum capacity of the HE is used for the process heat demand. The heat demand along with the ambient air temperature and useful energy from the PTC are passed on to the TESS (function ‘TES_VT_wCS’). This subroutine calculates the energy which is supplied from the solar field, TESS and, if necessary, the auxiliary heater to the HE for covering the heat demand for each time step. These results are saved to an EXCEL template which summarizes the values and further determines the performance indicators. All of these can also be realized in TRNSYS but the main disadvantage is that the mass flows in and out of the TESS cannot be calculated within the TESS component even when all temperatures are known. For this reason the model in TRNSYS requires another component (pump) leading to additional iterations and ultimately to an increased vulnerability to converging failures. The underlying equations for the TESS are specified in section 4.5 Thermal Energy Storage and the performance indicators in chapter 5.

Figure 10: Information flow diagram for the distributed heating system
4.2 Meteorological Data

Tianjin is located in northern China near the capital Beijing. Its temperate continental climate is mostly characterized by the four seasons being dry and windy in spring, hot and monsoon influenced in summer followed by cooler and comfortable autumns and dry, cold winters.

For the model different weather data sources have been studied in terms of data integrity and plausibility. Those are:

- The Chinese Standard Weather Data (CSWD) which has been developed by Dr. Jiang Yi, Department of Building Science and Technology at Tsinghua University and the China Meteorological Bureau. The CSWD has been developed for renewable energy utilization in building heating, cooling and energy use. It comprises, among others, annual design and typical year data generated from typically 30 years of record.

- The International Weather for Energy Calculations (IWEC) is the result from a research project of the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE). The data is licensed by the U.S. Department of Energy and normally used in building energy simulations. The files have been generated from 18 years of hourly weather record but do not include solar radiation. These have been added by taking the earth-sun geometry, hourly weather elements and particularly cloud amount information into account.

- The Solar and Wind Energy Resource Assessment (SWERA) which includes high quality information on solar and wind energy resources for developing countries. The project is funded by United Nations Environment Program. The datasets are derived from models, satellite and global weather observations and are commonly used in geo-referenced information systems (U.S. Department of Energy 2015).

The CSWD dataset is available for Tianjin and Beijing, whereas the IWEC and SWERA datasets are only available for Beijing, resulting in a total of four analyzed weather files. In terms of annual direct solar radiation only the SWERA weather file showed a value of about 14% higher than the other three datasets, see figure 11. However, CSWD weather files lack continuity of solar radiation, especially throughout the day, leading to high solar radiation during one hour and zero during the next hour. This causes problems in the PTC component and creates unreasonable output temperatures of two consecutive time steps. The IWEC and SWERA datasets do not show this behavior.

The applicability of SWERA data has been proven successful for numerous of projects in developing countries over the past 20 years (United Nations Environment Program 2011). As stated earlier, the data has been conducted exclusively for developing countries like China to improve the quality of information. The annual direct solar radiation for Beijing is 830 kWh/m² and in the range of the values 800 - 1000 kWh/m² provided by SolarGIS (2013) and Heimiller (2005). Further, Tianjin and Beijing are within a 120 km distance and a comparison of the direct solar radiation has shown that weather files are similar. For these reasons, the author believes that the SWERA weather data for Beijing is to be favored over IWEC data and Tianjin CSWD data in the simulation.
4.3 Heating Load Estimation

Heat load for space heating and cooling as well as for industrial processes are determined in the following two subsections.

4.3.1 Space Heating and Cooling

A weather file of Beijing has been validated in 4.2 Meteorological Data to represent the weather in Tianjin with reasonable degree of accuracy, meaning that it can also be used for determining the heating and cooling load for a building in Tianjin. The heating and cooling load has been estimated by applying the Degree-Day Method and Chinese Building Standard described in 3.3.2 and 3.3.3, respectively.

Since the average hourly ambient temperature was given, the Degree-Days and subsequently the heat and cooling load has been calculated for each hour throughout the year. Table 5 constitutes due to reason of simplification only a monthly summary.

\[
\begin{array}{|c|c|c|c|c|c|c|c|c|c|c|c|}
\hline
\text{Jan} & \text{Feb} & \text{Mar} & \text{Apr} & \text{May} & \text{Jun} & \text{Jul} & \text{Aug} & \text{Sep} & \text{Oct} & \text{Nov} & \text{Dec} & \text{Sum} \\
\hline
\text{Degree Days [°C-day]} & 499 & 339 & 230 & 33 & 36 & 66 & 78 & 55 & 25 & 50 & 224 & 404 & 2039 \\
\hline
\text{Heating and Cooling Load [kWh/m²]} & 8.5 & 5.8 & 3.9 & 0.6 & 0.6 & 1.1 & 1.3 & 0.9 & 0.4 & 10.9 & 3.8 & 6.9 & 34.9 \\
\hline
\end{array}
\]

The calculation has shown that an annual load of 34.9 kWh/m² is necessary to maintain the set point temperatures of a reference office building in China. Feng et al. (2014) used the simulation software EnergyPlus and a model with greater detail of the building properties to carry out the analysis and came to a comparable result, namely \( \sim 35 \) kWh/m² annual load. Additionally, the same calculation as for Tianjin, China, has also been performed for Chicago, USA, in order to validate the applicability of the underlying method for load estimations in this thesis. Both cities are located in alike climate zones (Feng et al. 2014). The ASHRAE 90.1-2013 standard, valid for designed and constructed buildings in the U.S. after 2013, has
been applied for the office building’s set point temperatures as well as heat transfer coefficients, see table 6.

Table 6: Parameters of a U.S. reference office building and corresponding heat load (U.S. Department of Energy 2014)

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Chicago, USA (Zone 5) / SC =~0.44</td>
<td>0.35</td>
<td>0.70</td>
<td>3.24</td>
<td>1.86</td>
<td>25</td>
<td>3974</td>
<td>94.0</td>
</tr>
</tbody>
</table>

A heating and cooling demand of 94.0 kWh/m² was determined after applying the U.S. parameters on the Chinese reference building. Feng et al. (2014) has obtained a similar value of ~100 kWh/m². However, these values are not comparable with those found from heat load calculations for a reference office building in the U.S. due to two main differences, the different comfort criteria and operation conditions. In China the temperature comfort level for heating and cooling suggested by the corresponding standard is around 20°C and 25°C, respectively. Whereas in the U.S., the heating and cooling comfort level is around 25°C and 22°C (MoHURD 2014; ASHRAE 2013). It should be noted that real applied values in practice might be different. Moreover, in U.S. office buildings the temperatures are usually controlled with thermostats allowing temperature setbacks at night, whereas controlling in Chinese office buildings is discontinuous and rather a matter of on/off function (Duffie & Beckman 2013; Feng et al. 2014). For these reason, the hourly heating demand evaluated with the aforementioned procedure is seen as appropriate for the model. Furthermore, the fact that the ambient temperature tends to follow the availability of direct solar radiation, meaning that higher values of direct solar radiation cause higher ambient temperatures compared to days without direct solar radiation during the same period of the year, strengthens the previous statement.

In the further course of this work, the heating and cooling demand for the building has been reduced to a heat load which needs to be covered by the HE for the subsequent distributed heating and cooling cycles.

Only the load profile, regardless of the square meter, of the calculated specific heating and cooling demand was of importance. In that way, the same profile could be applied in all cases for S1 and S2. The heat demand was determined by scaling the profile so that the annual maximum value was equal to the HE capacity. The annual maximum occurred on February 1st, 8 o’clock. The resulting heat demand based on the profile and HE capacity is exemplary depicted in figure 12.
4.3.2 Industrial Processes

As mentioned in 3.4 Heating Load Estimation for Industrial Processes, the underlying process needs to be known to optimally customize the solar energy system. For the purpose of this performance analysis, a constant heat demand based to the maximum capacity of the investigated heat exchanger in S3 and S4 was assumed, see table 7. This assumption is seen as appropriate since steady ongoing production cycles are common practice, which is to say a continuous heat demand. The performance analysis for a three working shift schedule, meaning 24 hours a day, 7 days a week, and a two working shift schedule meaning 16 hours a day, 7 days a week, was conducted for S3 and S4, respectively.

<table>
<thead>
<tr>
<th>HE capacity [kW]</th>
<th>70</th>
<th>80</th>
<th>90</th>
<th>100</th>
<th>110</th>
<th>120</th>
<th>130</th>
<th>140</th>
<th>150</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat demand</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>[MWh]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>S3 (24/7)</td>
<td>613</td>
<td>700</td>
<td>788</td>
<td>876</td>
<td>964</td>
<td>1051</td>
<td>1139</td>
<td>1226</td>
<td>1314</td>
</tr>
<tr>
<td>S4 (16/7)</td>
<td>-</td>
<td>467</td>
<td>526</td>
<td>584</td>
<td>642</td>
<td>701</td>
<td>760</td>
<td>818</td>
<td>876</td>
</tr>
</tbody>
</table>

4.4 Solar Field

The centerpiece of the solar field is the PTC which is separated into two collector arrays with an area of 548 m² each resulting in a combined area of 1096 m². The slope of the surface where the field is built on is considered flat. The arrays’ axes are oriented north-south.

The solar collector field has been modeled with the TRNSYS simulation software since a detailed PTC model is already given in the add-on TESS component library as well as the possibility to read and process meteorological data. Further, the useful energy production through the entire solar field is primarily dependent on the incident beam radiation and ambient temperature which are both given for one year in the SWERA weather file. These reasons promote the use of a stand-alone sub-model in TRNSYS to determine the useful energy production prior to the entire system simulation.
Figure 13 shows the setup of the solar field model in TRNSYS and table 8 the applied parameters. The component SF Weather (Type 15-3) reads the weather file at regular time intervals. The user specifies tracking mode, azimuth and slope of surface in this component to subsequently calculate, among other things, the incident beam radiation onto a surface (see 3.1 Fundamentals of Solar Radiation). This data and the ambient temperature are passed on to the SF PTC (Type 536) component, which is based on the theoretical equations summarized in Duffie & Beckman's Solar Engineering of Thermal Processes (2013). The useful energy gain is calculated, saved with component Type 65a and later used in MATLAB.

![Figure 13: TRNSYS - Solar field model](image)

In addition to known parameters of the PTC, it was necessary to specify intercept efficiency, efficiency slope and incidence angle modifiers (IAM). Losses in thermal collectors are categorized into optical and thermal origin. Optical losses, usually around 20% for linear parabolic concentrators, are caused, to a large extent, by the receiver shading parts of the reflector. This, in turn, means that the intercept efficiency for PTCs is commonly in the range of 70 to 80% (Stynes & Ihas 2012; Kalogirou 2004). The supplier Hi-Min Solar Co. Ltd of the solar collectors specifies an intercept efficiency of 78%. Due to a single evacuated receiver tube, thermal losses in PTCs are smaller than, for instance, with flat plate collectors, meaning that PTCs are less affected by temperature difference between collector inlet and ambient temperature thus having small efficiency slopes. The efficiency slope has been set to 8.0 kJ/hr,m²K according to typical values presented by Stine & Geyer (2001).

<table>
<thead>
<tr>
<th>SF Weather</th>
<th>SF PTC</th>
<th>Online plotter and output file</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type 15-3</td>
<td>Type 536</td>
<td>Type 56a</td>
</tr>
<tr>
<td><strong>Tracking mode</strong></td>
<td>the surface rotates about a fixed axis</td>
<td><strong>Aperture area</strong></td>
</tr>
<tr>
<td><strong>Slope of surface</strong></td>
<td>0 °</td>
<td><strong>Concentration ratio</strong></td>
</tr>
<tr>
<td><strong>Azimuth of surface</strong></td>
<td>0 °</td>
<td><strong>Intercept efficiency</strong></td>
</tr>
<tr>
<td><strong>Efficiency slope</strong></td>
<td></td>
<td>8.0 kJ/(hr m² K)</td>
</tr>
</tbody>
</table>

The frequency of occurrence of different collector efficiencies has been investigated in order to demonstrate the little influence of the assumed efficiency slope. As it can be seen in figure 14, even for the worst case, meaning collector inlet temperature equal to 100°C, more than 85% of the yearly solar field operations take place at collector efficiencies above 70%. The frequency of occurrence further increases to 94% at inlet temperatures of 50°C.
The optical efficiency is described reasonably well by equations for solar radiation direct normal onto the collector plane. However, this situation rarely occurs with one-axis tracking systems, making it difficult to describe the optical efficiency analytically due to complex concentrator and receiver geometries and optics. The IAM is a function of the incidence angle between the direct solar beam and the normal of the collector plane. It is used to describe the change of the optical efficiency related to the incidence angle. The IAMs are usually supplied by the manufacturer but had to be assumed in this study due to lack of information. Common IAMs for PTCs were extracted from Kovacs’ et al. Solar Rating and Certification Procedures (2013), Valenzuela’s STE plants with parabolic trough collectors (2012) and Larcher’s et al. Characterization of a Parabolic Trough Collector for Process Heat Applications (2014). A summary is shown in table 9. Incidence angles above 80° are neglected since the collector is shading itself due to its geometry and therefore no concentrated radiation is reaching the receiver (Larcher et al. 2014).

Table 9: Correlation between incidence angle of beam radiation and the IAM

<table>
<thead>
<tr>
<th>Incidence Angle</th>
<th>0°</th>
<th>10°</th>
<th>20°</th>
<th>30°</th>
<th>40°</th>
<th>50°</th>
<th>60°</th>
<th>70°</th>
</tr>
</thead>
<tbody>
<tr>
<td>Incidence Angle Modifiers</td>
<td>1.00</td>
<td>0.98</td>
<td>0.96</td>
<td>0.93</td>
<td>0.90</td>
<td>0.80</td>
<td>0.65</td>
<td>0.38</td>
</tr>
</tbody>
</table>

As a reason of missing experimental data from the solar field, in terms of useful energy gain in correlation with solar radiation, the simulation results could not be verified with actual measurements from the plant. However, the PTC model used in TRNSYS is based on broadly established equations (Duffie & Beckman 2013). Further, the parameters of the actual collector, when available, were used in the simulation. Common values for PTCs were used from literature, as described earlier in this section, when no data was available. Moreover, Kalogirou states in Solar thermal collectors and applications (2004) that the efficiency of PTCs for beam radiations between 500 to 1000 W/m² is between 73 to 77%. For the purpose of validation, the collector’s efficiency has been investigated illustratively on January 18th at 2.00 p.m. with an incidence beam radiation of 505 W/m² and a corresponding useful energy gain of 383 W/m² leading to an efficiency of 76%. In addition, the supplier of the solar field Hi-Min Solar Co. Ltd states an optical efficiency of 78% in the technical specifications.
the system is operated at relatively low temperatures, heat losses from the evacuated receiver tubes are also low, about 2%. The impact of different values for the efficiency slope on useful energy gain is seen as insignificant as validated above. For these reasons, the annual useful energy gain of ~790 kWh/m² obtained from the solar field submodel in TRNSYS is seen as reasonable and sufficiently verified.

### 4.5 Thermal Energy Storage

A direct two-tank energy storage system has been modeled with MATLAB. The operating mode of the TESS at each time step is a function of useful energy gain from the solar field, energy demand by the heating system and stored energy in the TESS. Simplified, the modes are described in table 10.

<table>
<thead>
<tr>
<th>Operating mode</th>
<th>Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Charging</td>
<td>$Q_{\text{useful}} &gt; Q_{\text{need}}$</td>
</tr>
</tbody>
</table>
<pre><code>                 | $Q_{\text{tank}} &lt; Q_{\text{tank max}}$         |
</code></pre>
<p>| 2. Discharging  | $Q_{\text{useful}} &lt; Q_{\text{need}}$         |
| $Q_{\text{tank}} &gt; Q_{\text{tank min}}$         |
| 3. Run through  | $Q_{\text{useful}} &lt; Q_{\text{need}}$         |
| $Q_{\text{tank}} = Q_{\text{tank min}}$         |
| 4. Bypass       | $Q_{\text{useful}} = 0$                       |
| $Q_{\text{tank}} = Q_{\text{tank min}}$         |</p>

The following sub-sections describe the underlying equations for each operating mode. The current time step is denoted with $i$ and the previous one with $i-1$. The fluid temperature in the tank is varying which also affects the maximum possible storable energy. It is therefore more convenient to monitor the mass and temperature of the fluid in the tank in order to determine which operating mode applies. Also, verification of the thermal storage tank is part of the entire system validation conducted in 4.7 Verification of the Model.

#### 4.5.1 Charging and Discharging

In the model, charging and discharging operating modes can be combined for cases where the storage tank stays within its tank boundaries. Separate operating modes are necessary for cases where either the upper storage or the lower storage level is reached.

<table>
<thead>
<tr>
<th>Charging and discharging</th>
<th>Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Charging to maximum level</td>
<td>$M_{\text{tank}} + m_i - m_o &gt; M_{\text{max}}$</td>
</tr>
<tr>
<td>2. Charging and discharging within tank boundaries</td>
<td>$M_{\text{min}} &lt; M_{\text{tank}} + m_i - m_o &lt; M_{\text{max}}$</td>
</tr>
<tr>
<td>3. Discharging to safety level</td>
<td>$M_{\text{tank}} + m_i - m_o &lt; M_{\text{min}}$</td>
</tr>
</tbody>
</table>

The heat and mass balance equations for all three conditions is given with equation 8 and equation 9.
\[ M_{\text{tank}}(i) = M_{\text{tank}}(i - 1) + \frac{m_i(i)}{\Delta t} - \frac{m_o(i)}{\Delta t} \]  
\[ \dot{Q}_{\text{in}}(i) = \frac{\Delta Q_{\text{tank}}(i)}{\Delta t} + \dot{Q}_{\text{out}}(i) + \dot{Q}_{\text{loss}}(i) \]

where
- \( M_{\text{tank}} \) is the mass of the fluid in the tank
- \( m_i \) is the mass flow entering the tank
- \( m_o \) is the mass flow leaving the tank
- \( \dot{Q}_{\text{in}} \) is the heat entering the tank from the solar field
- \( \dot{Q}_{\text{out}} \) is the heat leaving the tank to the heat exchanger
- \( \Delta Q_{\text{tank}} \) is the change of heat in the tank
- \( \dot{Q}_{\text{loss}} \) is the heat loss from the tank to the surroundings

The change of heat stored in the tank depends on the chosen length of the time step \( \Delta t \) but since weather data and heat demand is given on hourly bases, it is advisable to use a time step of one hour length. For convenience, \( \Delta t \) will not be written out in the further course of this work.

The useful energy gain \( \dot{Q}_{\text{in}} \) from the solar field has already been determined with TRNSYS. With the design outlet temperature \( T_{\text{in}} \) of the solar field and the inlet temperature \( T_c \), which is the return temperature of the system, the fluid mass flow through the solar field to the storage tank is calculated, see equation 10.

\[ \dot{Q}_{\text{in}}(i) = m_i(i)c_p(T_{\text{in}} - T_c(i)) \]  
\[ \dot{Q}_{\text{out}}(i) = m_o(i)c_p(T_{\text{tank}}(i) - T_{\text{set}}) \]

The change of heat stored in the tank is simply the difference of heat in the tank between the current and the previous time step.

\[ \Delta Q_{\text{tank}}(i) = M_{\text{tank}}(i)c_p(T_{\text{tank}}(i) - T_{\text{set}}) - M_{\text{tank}}(i - 1)c_p(T_{\text{tank}}(i - 1) - T_{\text{set}}) \]

Last but not least, heat losses in the tank are calculated with equation 13 where \( UA(i) \) denotes the heat transfer coefficient for the tank and \( T_{\text{amb}} \) refers to the temperature around the tank.

\[ \dot{Q}_{\text{loss}}(i) = UA(i)(T_{\text{tank}}(i) - T_{\text{amb}}(i)) \]

The \( UA(i) \) value is a function of the fluid level in the tank. The difference in wet and dry heat transfer coefficients is quite small (see 4.5.6 TESS Parameters) much like the change of the \( UA(i) \) value between two adjacent time step. It is therefore assumed sufficiently accurate to calculate the heat transfer coefficient with the fluid level from the previous time step prior to the calculation of the heat balance in order to save computational time.

\[ UA(i) = \pi d \left[ U_{\text{wet}} \left(h(i - 1) + \frac{d}{4}\right) + U_{\text{dry}} \left(h_{\text{max}} - h + \frac{d}{4}\right) \right] \]  

Equation 14
Finally, the mixing temperature in the tank for the current time step can be determined as follows

\[
T_{tank}(i) = \frac{m_i(i)T_{in} + T_{set}(M_{tank}(i) - M_{tank}(i-1) - m_i(i) + m_o(i)) + M_{tank}(i-1)T_{tank}(i-1) + \frac{UA(i)}{c_p}T_{amb}(i)}{M_{tank}(i) + \frac{UA(i)}{c_p} + m_o(i)}. \tag{Equation 16}
\]

Since the outgoing mass flow is a function of the temperature in the tank at the current time step, the temperature is calculated by initially guessing the outgoing mass flow for the first iteration. A good first guess can be obtained when using the temperature from the previous time step and the current heat demand. With the new determined temperature at each iteration, a more accurate outgoing mass flow is calculated letting the tank temperature eventually converge to its final value.

Two special cases occur when the tank reaches the maximum or minimum fluid level. When it reaches the maximum level the excessive mass flow bypasses the storage tank. In the model this excessive mass flow is saved in a variable \(m_{waste}\). In reality some collectors are turned out of focus to lower the useful heat gain and avoid overheating of the system. However, the discussion of safety measures is not part of this thesis and is only briefly mentioned here. When the tank reaches the minimum level then the outgoing mass flow is limited to the amount entering the tank plus the amount left over for discharge.

See Appendix B: MATLAB CODE – TESS – mode 1 to 3 for detailed information on determining the current operating mode and iteration procedure.

\subsection*{4.5.2 Run Through}

The storage tank is in run through mode when energy from the collector field can be used but partial auxiliary heating is still required. This is the case when the tank temperature is above the system set temperature. The mass flow into the tank is then equal to the outgoing mass flow meaning that the mass in the tank stays at the safety level.

The heat balance and energy equations from 4.5.1 Charging and Discharging also apply for this mode. With the mass balance for the tank (equation 17) where \(m_i(i) = m_o(i) \to 0\), the tank temperature is defined as follows:

\[
M_{tank}(i) = M_{tank}(i-1) \tag{Equation 17}
\]

\[ h = \frac{4M_{tank}(i - 1)}{\rho_{fluid} \pi d^2} \]  

Equation 15

where

- \(U_{wet}\) is the wet heat transfer coefficient
- \(U_{dry}\) is the dry heat transfer coefficient
- \(h\) is the height of the fluid level in the tank
- \(d\) is the diameter of the tank
- \(\rho_{fluid}\) is the density of the heat transfer fluid
\[ T_{\text{tank}}(i) = \frac{m_i T_{\text{in}} + M_{\text{tank}}(i - 1) + \frac{UA(i)}{c_p} T_{\text{amb}}(i)}{M_{\text{tank}}(i) + \frac{UA(i)}{c_p} + m_i(i)} \]  

Equation 18

See Appendix B: MATLAB CODE – TESS – mode 6 for detailed information on determining the current operating mode and iteration procedure.

### 4.5.3 Bypass

The storage tank is bypassed whenever no usable energy is available from the tank and solar field. This case occurs at times with no solar radiation and after complete discharge of the storage tank. Heat is continuously being dissipated from the tank due to a safety level that needs to be maintained in the tank \( M_{\text{min}} \). This results in a temperature drop of the remaining fluid. Applying the mass balance

\[ M_{\text{tank}}(i) = M_{\text{tank}}(i - 1) = M_{\text{min}} \]  

Equation 19

and the heat balance

\[ Q_{\text{tank}}(i) = Q_{\text{tank}}(i - 1) - Q_{\text{loss}}(i) \]  

Equation 20

yields the tank temperature in bypass mode

\[ T_{\text{tank}}(i) = T_{\text{tank}}(i - 1) - \frac{Q_{\text{loss}}(i)}{c_p M_{\text{min}}} \]  

Equation 21

The variable \( Q_{\text{loss}} \) is a function of the tank temperature at the current time step, iteration is therefore required to establish the correct tank temperature.

See Appendix B: MATLAB CODE – TESS – mode 4 for detailed information on determining the current operating mode and iteration procedure.

### 4.5.4 Storage Reheat

For safety reasons, large storage tanks usually have an own auxiliary heater to keep the tank at a certain minimum temperature level to prevent solidification of the storage medium (Libby 2010). This is of considerable importance when using, for instance, the molten salt \( \text{NaNO}_3 - \text{NaNO}_2 \) compound which has a melting point of 221°C (Wang 2011). However, the plant, subject of this study, uses Therminol 55 with a melting temperature of -40°C (Eastman Chemical Company 2015) as heat transfer and storage medium in the solar field and TESS, respectively. Auxiliary heating is thereby not required as a reason of the low melting temperature. Nevertheless, reheating is required to raise the tank temperature above the HFOT. This is accomplished by only charging the tank with heat from the solar field. The following mass balance and heat balance are valid.

\[ M_{\text{tank}}(i) = M_{\text{tank}}(i - 1) + m_i(i) \]  

Equation 22
\[ Q_{\text{tank}}(i) = Q_{\text{tank}}(i-1) + Q_{\text{in}}(i) - Q_{\text{loss}}(i) \]  

Equation 23

Leading to the mixing tank temperature:

\[
T_{\text{tank}}(i) = \frac{m_i(T_{\text{in}} - T_{\text{set}}) + M_{\text{tank}}(i-1)(T_{\text{tank}}(i-1) - T_{\text{set}})}{M_{\text{tank}}(i) + \frac{UA(i)}{c_p}T_{\text{amb}}(i) + \frac{UA(i)}{c_p}T_{\text{set}}} + \frac{M_{\text{tank}}(i)T_{\text{set}}}{M_{\text{tank}}(i) + \frac{UA(i)}{c_p}} \]

Equation 24

This mode only applies when the tank temperature after mixing is still below the HFOT. Otherwise, the mode will be switched to \textit{run through} in order to lower the auxiliary heating demand. In reality, the system rarely operates in the \textit{storage reheat} mode since the remaining mass in the storage tank is relatively low compared to the mass flow coming from the solar field at 120°C. This results, generally, in mixing temperatures above the HFOT and therefore \textit{run through} mode.

See Appendix B: \textit{MATLAB CODE – TESS – mode 5} for detailed information on determining the current operating mode and iteration procedure.

### 4.5.5 Heat Losses from Return Tank

Operating modes including tank temperature, mass flows and heat losses to the environment has been determined in sections 4.5.1 to 4.5.4. Once the fluid has been circulated through the HE it is returned to the cold storage tank at HFOT before it is pumped through the solar field again. While remaining in the cold storage tank, further heat losses occur and have to be considered. The temperature in the cold tank is calculated analog to section 4.5.2 \textit{Run Through} and used in the next time step to determine the required mass flow through the solar field to obtain an outlet temperature of 125°C.

### 4.5.6 TESS Parameters

TESS can be found on the market in any imaginable shape and technical design. The most common are horizontal and vertical cylindrical, flat bottom and spherical tanks (Bradley 2004). A vertical cylindrical tank has been selected in this model due to space restrictions. This way the tank volume can still be quite large compared to the occupied ground space. The properties of the tank and working fluid have been summarized in table 12 below.

<table>
<thead>
<tr>
<th>Table 12: TESS and working fluid properties</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Tank and Fluid Properties</strong></td>
</tr>
<tr>
<td>( T_{\text{in}} )</td>
</tr>
<tr>
<td>( T_{\text{set}} )</td>
</tr>
<tr>
<td>( V )</td>
</tr>
<tr>
<td>( H/D )</td>
</tr>
<tr>
<td>( h_{\text{safety}} )</td>
</tr>
<tr>
<td>( U_{\text{dry}} )</td>
</tr>
</tbody>
</table>
Table 12 (continued): TESS and working fluid properties

<table>
<thead>
<tr>
<th>$U_{\text{wet}}$</th>
<th>4.0 kJ/hr,m²K</th>
<th>Wet heat transfer coefficient</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluid</td>
<td>Therminol 55</td>
<td>Working fluid</td>
</tr>
<tr>
<td>$c_p$</td>
<td>2.01 kJ/kgK</td>
<td>Heat capacity at 120°C</td>
</tr>
<tr>
<td>$\rho_{\text{fluid}}$</td>
<td>852 kg/m³</td>
<td>Density at 120°C</td>
</tr>
</tbody>
</table>

The original design inlet and outlet temperature ($T_{\text{set}}$) of the HE is 120°C and 100°C, respectively. The inlet temperature of the TESS has been raised by 5°C to 125°C since the withdrawal temperature from the TESS will be lower because of heat losses. The tank height to diameter ratio has been set to 3 and the minimum oil storage level ($h_{\text{safety}}$) to 0.2 m. The safety level prevents the pump from pumping air into the system thus avoiding damage. For the sake of simplicity it has been assumed that the bottom of the tank is flat. In reality, however, the bottom of the tank is most likely funnel-shaped to further reduce the residual safety volume. The wet and dry heat transfer coefficients are usually between 2.0 to 6.0 kJ/hr,m²K for temperature differences up to 100°C (Spirax Sarco Ltd. 2015). These values depend highly on the chosen insulation which is a trade-off between cost and heat loss. However, the optimization of these parameters is not subject of this study. Nevertheless, the chosen heat transfer coefficients are seen as reasonable due to an hourly temperature drop in a fully filled tank of about 1.5°C, similar to the one stated by Libby in Solar Thermocline Storage Systems - Preliminary Design Study (2010). Finally, the heat capacity and density of the working fluid Therminol 55 at 120°C have been used throughout the simulation.

### 4.6 Auxiliary Heater

The plant on site features a 2.4 MW diesel-driven auxiliary heater mainly acquired for experimental purposes in conjunction with the ORC unit. The boiler is oversized for the distributed heating unit but will still be used for that purpose. Its efficiency of 92% is used to calculate the fuel consumption which is necessary to cover the heat demand. The auxiliary heat demand is calculated with equation 25

$$Q_{\text{aux}}(i) = Q_{\text{need}}(i) - Q_{\text{supply}}(i)$$  \hspace{1cm} \text{Equation 25}

where $Q_{\text{need}}$ is the heating demand and $Q_{\text{supply}}$ is the supplied heat from the TESS (equation 26).

$$Q_{\text{supply}}(i) = m_o(i) c_p(T_{\text{tank}}(i) - T_{\text{set}})$$  \hspace{1cm} \text{Equation 26}
4.7 Verification of the Model

The plausibility of the meteorological data, heat demand estimation and solar field heat gain have been validated in sections 4.2 through 4.4. The verification of the TESS and subsequently the entire model is subject of this section.

The proper operation of the TESS can be proven by assuring the fulfilment of heat and mass balances throughout the year. For this purpose, a storage tank size of 35 m³ and HFOT of 70°C were chosen exemplarily. The mass balance is given by equation 8 in section 4.5.1 Charging and Discharging. The incoming mass flow is only a function of the tank temperature in the return tank and the design inlet temperature for the storage tank and can be determined. The outgoing mass flow is calculated with the storage tank temperature simultaneously within the operation mode.

The mass balance equations is fulfilled whenever the mass in the tank from the previous time step plus the mass flowing into and out of the tank in the current time step equals the mass remaining in the tank at the end of the current time step. Consequently, the difference between both sites of the mass balance equation becomes zero which means it is fulfilled. The heat balance equation is fulfilled with the same analogy as applied to the mass balance equation. Meaning, whenever the heat in the tank from the previous time step plus the heat entering and leaving the tank equals the heat remaining in the tank at the end of the current time step, the difference between both sides of the heat balance equation become zero. Figure 15 and figure 16 show that the differences are mostly zero throughout the year with negligible differences and thus both balance equations fulfilled. The difference results mainly from the assumption that the overall heat transfer coefficient for the current time step is calculated by using the filling level of the previous time step. This leads to a deviation of heat losses. Mass and heat balance are closely interrelated and therefore the same behavior can be observed in both figures.

Figure 15 depicts the TESS filling levels throughout the year. The variable $H_{\text{max}}$ indicates the maximum height of fluid in the tank and, vice versa, $H_{\text{min}}$ represents the minimum fluid level. The filling levels stay within the tank boundaries. It should be noted that the plant is only in operation in this model during heating seasons which is before April 11th and after October 19th. The vast surplus of solar energy during the cooling season results in 100% demand coverage and is simply covered within the EXCEL template.
Last but not least, the system behavior on March 1st has been investigated exemplarily, see figure 18. At times without solar radiation ($Q_{useful} = 0$) and empty tank ($Q_{storage} = 0$), for instance in the morning, the heat demand is covered completely by the auxiliary heater ($Q_{aux}$). At about 8 o’clock enough solar radiation is available. Some parts of the useful energy from the solar field is used to reheat the remaining fluid in the tank to above HFOT while some parts are lost to the environment, leading to the actual supplied energy ($Q_{supply}$). Note that $Q_{loss}$ is not depicted in figure 18. Throughout the remaining day the tank is further charged at times with high solar radiation before it is discharged again to minimum level.

**Figure 17: Storage tank filling levels for one year**

**Figure 18: System behavior on March 1st**
5 PERFORMANCE INDICATORS

The four studied scenarios are evaluated in terms of their performance. The key indicators to characterize the performance of the solar distributed heating system are the Levelized Costs of Energy (LCOE) explained in 5.1 Economic Indicator, the thermal efficiency and solar fraction of the solar system, including the TESS, as outlined in section 5.2 Technical Indicators, and the emission of the greenhouse gas CO₂ as presented in 5.3 Environmental Indicator. The definition of indicators has been limited to describe solely the overall system performance since this study investigated the general applicability of the solar system in Tianjin, China, for space heating and cooling or industrial process heat purposes.

5.1 Economic Indicator

The method Levelized Costs of Energy (LCOE) is commonly used to benchmark the cost-effectiveness of different energy generation technologies and system configurations. The generated energy and all the corresponding costs, for instance investment or maintenance expenditures, over the technology’s lifetime are taken into account to estimate a price per unit energy generated (Branker et al. 2011).

The LCOE has been determined according to the guidelines compiled in Branker's et al. (2011) journal article A review of solar photovoltaic levelized cost of electricity. The following equation was applied:

\[
\text{LCOE} = \frac{\sum_{t=0}^{T} (I_t + O_t + M_t + F_t)/(1 + r)^t}{\sum_{t=0}^{T} S_t (1 - d_r)^t / (1 + r)^t}
\]

Equation 27

where

- \( T \) is the lifespan of the project in years
- \( t \) is the year
- \( I_t \) is the initial investment costs
- \( O_t \) is the operating costs for \( t \)
- \( M_t \) is the maintenance costs for \( t \)
- \( F_t \) is the interest expenditure for \( t \)
- \( S_t \) is the yearly rated energy output for \( t \)
- \( r \) is the discount rate for \( t \)
- \( d_r \) is the degradation rate
The main assumptions used for the economic analysis are summarized in table 13.

### Table 13: Data and assumptions for the economic analysis

<table>
<thead>
<tr>
<th>Specific costs</th>
<th>Cost per unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tank</td>
<td>$700/m³</td>
</tr>
<tr>
<td>Heat exchanger</td>
<td>$333/m²</td>
</tr>
<tr>
<td>Therminol 55</td>
<td>$3476/m³</td>
</tr>
<tr>
<td>Fuel</td>
<td>$0.093/kWh</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Fix costs</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Total investment</td>
<td>$490,000</td>
</tr>
<tr>
<td>O&amp;M costs</td>
<td>1.0 % of total investment</td>
</tr>
<tr>
<td>O&amp;M escalator</td>
<td>5.0 %</td>
</tr>
<tr>
<td>Discount rate</td>
<td>5.0 %</td>
</tr>
<tr>
<td>Annual degradation</td>
<td>0.5 %</td>
</tr>
<tr>
<td>Operating lifetime</td>
<td>20 years</td>
</tr>
</tbody>
</table>

1) from Cocco & Serra (2015)
2) from Binhai project documents
3) calculated with 1.01 $/l fuel price (June 22nd, 2015 - GlobalPetrolPrices.com) and 39 MJ/l heating value
4) assumption

### 5.2 Technical Indicators

Two common measures for evaluating the system performance of the solar system, including the TESS, are the thermal efficiency and the solar fraction.

The thermal efficiency ($\eta_{th}$) of a solar system is determined by the energy supplied ($Q_{supply}$) to the heat exchanger compared to the total available solar beam radiation ($Q_{sol}$).

$$\eta_{th} = \frac{Q_{supply}}{Q_{sol}}$$  \hspace{1cm} \text{Equation 28}

The solar fraction ($SF$) states the amount of heat supplied by the solar system ($Q_{supply}$) compared to the overall heat requirements ($Q_{need}$). It is usually presented on monthly and annual basis (Deutsche Gesellschaft für Sonnenenergie 2010).

$$SF = \frac{Q_{supply}}{Q_{need}}$$  \hspace{1cm} \text{Equation 29}

The thermal efficiency in solar systems, that also including a TESS, is closely connected to the solar fraction and vice versa (Deutsche Gesellschaft für Sonnenenergie 2010). Figure 19 is valid for the general assumption that a larger collector field requires a larger TESS to avoid overheating of the system. The figure shows that the efficiency for the system decreases with a larger collector area and storage tank mainly due to higher heat losses from the TESS. At the same time an increase of solar fraction occurs since more energy is gained through the solar field. This makes it possible to cover additional heat demand. As a whole, it means that every extra kilowatt-hour gained by the system becomes more expensive.
Note: Figure 19 is not true-to-scale and intended merely for illustrative purposes.

![Figure 19](image)

**Figure 19: Correlation between system efficiency and solar fraction, adapted from Deutsche Gesellschaft für Sonnenenergie (2010)**

5.3 Environmental Indicator

During the combustion of fossil fuels greenhouse gases such as CO$_2$, CH$_4$ and N$_2$O are emitted. Weighted on their global warming potential (GWP), which rates the gases ability to trap heat in the atmosphere, CO$_2$ is responsible for 99% of all greenhouse gas emissions. Methods for CO$_2$ estimation are widely established since appropriate emission factors for the consumed fuel type are known. Aside from fuel characteristics, CH$_4$ and N$_2$O emissions also depend on the combustion technology and process, flue gas cleaning equipment, maintenance, operation practices etc. (Greenhouse Gas Protocol 2007). A much too high calculation effort arises with CH$_4$ and N$_2$O estimations, mainly due to the relatively small share in greenhouse gas emissions, compared to CO$_2$, and a large number of uncertainties. The method for estimating CO$_2$ emissions is therefore only covered in this section.

The fuel analysis approach determines the carbon content of combusted fuel and correlates it to the amount of burned fuel. The CO$_2$ emissions are determined with equation 30 according to the guidance *Stationary Combustion Sources* by U.S. Environmental Protection Agency (2008).

\[
CO_2 = m_F \times HV_F \times C_F \times \frac{M_{CO_2}}{M_C}
\]

Equation 30

where
- $CO_2$ is the CO$_2$ emission
- $m_F$ is the combusted amount of fuel
- $HV_F$ is the heating value of the fuel type
- $C_F$ is the carbon content coefficient of the fuel type
- $M_{CO_2}$ is the molar mass of CO$_2$ (44 kg/mol)
- $M_C$ is the molar mass of Carbon (12 kg/mol)

In this study, a heating value of 39 MJ/l and a carbon content coefficient of 0.72 kgC/l for the diesel fuel were used (Biomass Energy Centre). The additional savings of CO$_2$ emissions, induced by the optimized compared to the original system, have been presented in the results. It is referred to as upgrade related CO$_2$ savings.
6 RESULTS

Usage capabilities of the solar-driven distributed heating system have been investigated by means of the four scenarios, described in section 1.2 Objective and Research Questions, and the Performance Indicators, mentioned in chapter 5. The following subsections present the results in detail of each studied scenario. An overview of the systems, leading to the best economic performance in each scenario, is given below:

Table 14: Summary of scenarios, corresponding optimal system and results

<table>
<thead>
<tr>
<th>Identified scenarios</th>
<th>S1</th>
<th>S2</th>
<th>S3</th>
<th>S4</th>
</tr>
</thead>
<tbody>
<tr>
<td>TESS size</td>
<td>0 - 35 m³ (5 m³ interval)</td>
<td>0 - 70 m³ (10 m³ interval)</td>
<td>0 - 70 m³ (10 m³ interval)</td>
<td>0 - 70 m³ (10 m³ interval)</td>
</tr>
<tr>
<td>HE resizing</td>
<td>No</td>
<td>Yes</td>
<td>Yes</td>
<td>Yes</td>
</tr>
<tr>
<td>HE HFOT</td>
<td>50 - 100°C</td>
<td>50 - 100°C</td>
<td>50 - 100°C</td>
<td>50 - 100°C</td>
</tr>
<tr>
<td>Heating purpose</td>
<td>Space heating and cooling</td>
<td>Space heating and cooling</td>
<td>Industrial process heat (24/7)</td>
<td>Industrial process heat (16/7)</td>
</tr>
<tr>
<td>Systems with optimized LCOE</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>TESS size</td>
<td>0 m³</td>
<td>10 m³</td>
<td>40 m³</td>
<td>30 m³</td>
</tr>
<tr>
<td>HE capacity</td>
<td>250 kW</td>
<td>750 kW</td>
<td>80 kW</td>
<td>130 kW</td>
</tr>
<tr>
<td>HE HFOT</td>
<td>100°C</td>
<td>50°C</td>
<td>50°C</td>
<td>50°C</td>
</tr>
<tr>
<td>Performance</td>
<td>Heat supplied</td>
<td>375 MWh</td>
<td>1120 MWh</td>
<td>700 MWh</td>
</tr>
<tr>
<td>LCOE</td>
<td>0.187 $/kWh</td>
<td>0.146 $/kWh</td>
<td>0.106 $/kWh</td>
<td>0.098 $/kWh</td>
</tr>
<tr>
<td>Solar fraction</td>
<td>44.8%</td>
<td>36.4%</td>
<td>78.9%</td>
<td>77.5%</td>
</tr>
<tr>
<td>Thermal efficiency</td>
<td>14.5%</td>
<td>35.3%</td>
<td>47.8%</td>
<td>50.9%</td>
</tr>
<tr>
<td>Upgrade related CO2 savings</td>
<td>0%</td>
<td>5.4%</td>
<td>66.9%</td>
<td>54.3%</td>
</tr>
<tr>
<td>Investment</td>
<td>$0</td>
<td>$88,500</td>
<td>$175,000</td>
<td>$135,000</td>
</tr>
</tbody>
</table>

6.1 S1: Current System for Space Heating and Cooling

The current system for space heating and cooling has been studied to find the optimal parameters which lead to the best economic performance in terms of LCOE in S1. The scenario was chosen to study the performance of the present on-site setup in Tianjin. There have been no additional changes made to the entire system aside from varying the HFOT from 50 to 100°C and installing a TESS with a corresponding tank size that ranges from 5 to 35 m³.
The different LCOEs for all temperature levels and TESS sizes are depicted in figure 20. A lower HFOT results in a higher LCOE mainly due to the fact that a lower HFOT also decreases the HE capacity, see figure 21. This, in turn, implies lower productivity of the entire plant and less floor area to be heated.

From an economic perspective, a system without any TESS and a HFOT of 100°C, as the current system setup, leads to the lowest LCOE of 0.187 $/kWh. No investment is required and an annual heating and cooling load of about 375 MWh can be supplied. The solar fraction of such a system is 44.8% (figure 22) and CO₂ savings related to the solar field are 54.6 tons/year. However, the system is not upgraded with any TESS, meaning that the upgrade related CO₂ savings remain at 0% (figure 23).

The thermal energy system has an efficiency of 14.5% (figure 24). The low efficiency is the result of large amounts of solar energy available during summer month and simultaneous relative low heat demand. The thermal efficiency improves with an increase in TESS size, because more heat from the solar field can be utilized and stored for the later usage. An increase in TESS size also causes additional heat losses due to a larger surface for heat transfer to the environment. Consequently, a maximum efficiency is reached when the tradeoff between additional energy utilized equals the heat losses (figure 24, trend of blue curve T<sub>set</sub>=50°C). Despite the fact that the thermal efficiency improves with an increase in TESS size, the system with the highest thermal efficiency does not obtain the lowest LCOE, because an increase in TESS size allows the storage of more solar energy only for a couple of hours, while extra heat losses occur for 8760 hours a year. This means that each extra
kilowatt-hour useful energy gain becomes more expensive since it provokes additional heat losses.

6.2 S2: Optimized System for Space Heating and Cooling

The optimized parameters for a system with the purpose of space heating and cooling which lead to the optimized LCOE were sought in S2. The capacity of the heat exchanger has been varied from 250 to 800 kW, the HFOT from 50 - 100°C and the TESS size was increased up to 70 m³.

Figure 25 shows the minimum LCOE for each temperature level obtained at different HE capacities. The LCOE decrease with larger HE since more heat from the solar field can be used directly. This reduces the required TESS size (figure 26). The minimum LCOE of 0.145 $/kWh is reached at 750 kW HE capacity, 10 m³ TESS and 50°C HFOT. A total investment of about $88'500 has to be made of which $50'000 account for the TESS and $38'500 for the upgrade of the HE. This system covers a heat demand of 1120 MWh, which is about 3 times higher than in S1.

In general, the heat losses from all components, especially from the storage tanks, decrease with lower HFOT due to smaller temperature differences to the surroundings. This, in turn, implies a better utilization of the solar energy leading to higher productivity of the system and ultimately lower LCOE within the same setup (figure 27). In S2, the LCOEs for HFOT 50 – 70°C decrease first for a 10 m³ TESS and then rise again for 20 m³ TESS and above mainly because the system harvests more solar energy than is required. This unused energy is stored.
in a TESS for later usage. The LCOE fall as long as the price for the extra stored kilowatt-hour is below the LCOE, but with more energy stored, increase also the heat losses making every extra stored kilowatt-hour more and more expensive. This also indirectly explains the fact that the maximum solar fraction of 38.1%, upgrade related CO₂ savings of 7.9% and efficiency of 37.8% is obtained with a 40 m³ TESS, see figure 28, 29 and 30. Above this TESS size, more energy is lost to the surroundings than made available with extra storage.

The system in S2, with optimized LCOE for space heating and cooling, has a solar fraction of 36.4%, upgrade related CO₂ savings of 5.4% and an efficiency of 35.3%. Compared to S1, the LCOE were lowered by 0.041 $/kWh while a three times higher heat demand was covered (table 14). However, the higher heat demand also increased the auxiliary heating demand at times without available solar radiation causing a lower solar fraction. But at the same time, less solar energy was wasted which led to an improvement of the thermal efficiency from 14.5% in S1 to 35.3% in S2.
6.3 S3: Optimized System for Industrial Process Heat – 24/7

The optimal parameters for the solar system used in an industrial process requiring heat 24/7 were obtained in S3. The capacity of the HE has been varied from 70 to 150 kW, the HFOT from 50 to 100°C and the TESS size has been increased up to 70 m³. Since no specific heat demand was provided, a heat demand was sought at which the system operates the most cost-effective. This heat demand determined the HE capacity, refer to 4.3.2 Industrial Processes.

In the investigated setup, the minimum LCOE of 0.106 $/kWh were found at a HE capacity of 80 kW (figure 31), HFOT of 50°C and a TESS size of 40 m³ (figure 32). The system reaches a solar fraction of 78.9% (figure 33), an efficiency of 47.8% (figure 35) and upgrade related CO₂ savings of 66.9% (figure 34). A total investment of $175,000 for the TESS is required, whereas no additional upgrading is necessary for the HE. The system covered a heat load of 700 MWh.

In this setup, about 48% of the available solar energy is delivered to the heat consumer. The remaining 52% account for 24% optical and thermal losses caused by the solar field (4.4 Solar Field), 19% wasted solar energy and 9% heat losses from the TESS. Utilizing more of the harvested solar energy to reduce the wasted energy increases the LCOE due to two main reasons. First, it requires a larger TESS which increases the heat losses from the tanks and, second, a larger HE capacity to provide more heat. Since the industrial heat demand was assumed constant, higher fossil-fuel consumptions arise at time without solar radiation and, therefore, increase the LCOE.
The extensive performance improvements are the result of a 40 m³ TESS providing 80 kW of heat for about 14 hours after being fully charged. This covers large amounts of the heat demand during night time in winter and nearly all of it during summer.

6.4 S4: Optimized System for Industrial Process Heat – 16/7

As well as in S3, the heat demand was sought in S4 that led to the most cost-effective use of the plant. To that end, the HE capacity determined the maximum heat demand, which was required 16/7 from 6 a.m. to 10 p.m., e.g. two-shift factory. The remaining simulation parameters stayed the same as in S3.

The LCOE dropped to 0.098 $/kWh when using a HE with 130 kW capacity at 50°C HFOT and a TESS size of 30 m³, see figure 36 and 37. The annual heat demand increases slightly to 760 MWh compared to S3. The solar fraction in this setup is 77.5% (figure 38), the efficiency reaches 50.9% (figure 40) and the upgrade related CO₂ savings result in 54.3% (figure 39). A total investment of $135'000, $133'500 for the TESS and $1'500 for the HE, is necessary.

The 30 m³ TESS provides 130 kW of heat for about 7.5 hours. Similar as in S3, the TESS covers most of the heat demand after sunset and before sunrise. A larger heat demand can be satisfied since no heat is required from 10 p.m. to 6 a.m. The larger heat demand in S4, compared to S3, causes a higher volume flow of the heat transfer fluid and hence a duration difference in supply hours for the TESS in S3 and S4.
Figure 38: Annual solar fraction for different TESS sizes and 130 kW HE capacity (S4)

Figure 39: Upgrade related CO$_2$ savings for different TESS and 130 kW HE capacity (S4)

Figure 40: Efficiency for different solar thermal systems and 130 kW HE capacity (S4)


7 DISCUSSION

In this chapter, the Results from chapter 6 for the four scenarios are discussed in sections 7.1 Optimal Systems, 7.2 Impact of Assumptions and 7.3 Optimization Potential.

7.1 Optimal Systems

The economic performance evaluations of the first scenarios have shown that neither upgrading with a TESS nor changing design parameters of the current system is advisable, meaning that the systems LCOE of 0.187 $/kWh is already the lowest for space heating and cooling purposes. Further, upgrading this system in the second scenario with a TESS and HE while simultaneously lowering the HFOT led only to the most optimal LCOE for the associated HE capacity. However, the further the HE capacity is increased the lower becomes the LCOE. The minimum LCOE of 0.145 $/kWh (figure 25) is reached at 750 kW HE capacity, 10 m³ TESS and 50°C HFOT. The required TESS size decreases since more and more energy from the solar field is used directly without the necessity of being stored (figure 26). In addition, the solar fraction decreases, more fossil fuel is consumed and consequently more CO₂ is emitted. Regardless of the better economic performance of the optimized system in S2 compared to the current one in S1, the optimized system is not seen as feasible in practice, because the HE in S1 with 100°C HFOT has a heat transfer surface of 15 m² while the HE in S2 with 50°C HFOT requires a surface of 115 m².

A published version 8.0 of the Levelized Cost of Energy Analysis in 2014 by Lazard states LCOE of 0.118 to 0.130 $/kWh for solar thermal energy systems with TESS for the United States. Libby (2010) determined in his Solar Thermocline Storage Systems – Preliminary Design Study a LCOE of 0.130 to 0.160 $/kWh without TESS and 0.080 to 0.110 $/kWh with TESS. In this context, the found LCOEs in this report are seen as reasonable. However, the underlying system has only been optimized regarding to the size of the TESS, HE and HFOT for the purpose of providing heat. The original system has been designed for experimental utilization related to the ORC unit. Some components, such as the auxiliary heater, that have not been analyzed may not be operating at optimal conditions. Consequently, the LCOE for providing heat found in this study only indicates the lowest LCOE possible in relation to the specific plant in Binhai district and the investigated upgrades. Hence, it is believed that a lower LCOE can be obtained in China with an entirely optimized system.

The LCOE for the first and second scenario can compete with an LCOE of 0.178 $/kWh for fossil fuel based auxiliary heating and cooling. On the other hand, Chinese traditional heating with air conditioners (A/C) for heating and cooling have an LCOE of 0.112 $/kWh when applying an electricity price for end consumers of 0.08 $/kWh (Holmes 2014), A/C unit specific cost of 140 $/kW installed capacity, based on average prices acquired from made-in-
China.com, and replacement of A/C units three times in 20 years. A further advantage is that the temperature of each room can be controlled individually with A/C units according to occupants needs. The relatively high LCOEs of S1 and S2, compared to traditional heating, are the results of poor exploitation during summer months especially during April, May, September and October (figure 41). The winter in Tianjin is rather cold which causes for a high heat demand and thereby leading to the least amount of unexploited energy during these months (figure 42). Furthermore, the heating and cooling seasons in Tianjin last from end-October to early-April and mid-May to end-August, respectively, resulting in about 80 to 100 days without any heating or cooling demand each year. During this period the solar radiation and output of the PTC are the highest but not utilized and therefore entirely wasted. The constant heat demand throughout the year as in S3 and S4 has decisive advantageous in terms of use of available solar energy. Nevertheless, about 23 to 25% of the solar energy is wasted. An extension of the HE capacity or the TESS size, in order to use more of the available solar energy, is still not advisable due to two main reasons. First, a higher HE capacity increases the diesel fuel consumption, at a price of 0.092 $/kWh, and CO₂ emissions during morning hours and at times without solar radiation and empty TESS. Second, a larger TESS size allows for storage of more solar energy but at the same time increases thermal losses from the tanks. Both upgrades involve extra investments which add to the diesel fuel price leading eventually to higher LCOE than presented for S3 and S4.

7.2 Impact of Assumptions

In this model, meteorological data from SWERA was used since it showed best behavior in terms of data integrity and plausibility. But it also had the highest annual direct solar radiation compared to the remaining studied data sources. Moreover, the vast increase of air pollution, especially in the Beijing-Tianjin region, over the past decade has led to less direct solar radiation and less productivity of CSP technologies. Even though stricter regulations, recently introduced by the Chinese government, for reducing air pollution have shown visible results (China.org.cn 2015), the relevance of the data for the current situation is to be questioned since the meteorological data is comprised of 30 years of recording.

The heating load for S1 and S2 has been determined by using the Degree-Day method which is suitable for a first assessment. This method solely considers the ambient temperature, but neither properties of the building nor the applied heating and cooling technology. Especially
the heat load required by the cooling process varies significantly with the technology. For instance, the coefficient of performance (COP), which is the ratio of cooling provided to electrical or thermal energy consumed, is for absorption refrigeration 0.7 – 1.3 and for compressor refrigeration commonly 3 – 4 (IBS Heizung 2012). This, in turn, implies that the heat demand by the cooling process can differ by factor 4. If the building and its properties are known then a more detailed heat-balance method must be applied to improve accuracy of the model. Simulation programs such as TRNSYS provide plugins with which a detailed description of the house can be implemented into the simulation model. The heating load for S3 and S4 was assumed to be constant and corresponded to the maximum capacity of the HE. In reality, depending on the underlying process, heat demand and productivity might not be constant and extra free capacities as backup are considered. In addition, the heat transfer coefficient was determined to be 278 W/m²K, based on the manufactures information given for the original design, and assumed constant throughout the simulation. This coefficient together with the HFOT was used to scale the HE according to the required heat demand in order to calculate its acquisition costs. Nevertheless, the heat transfer coefficient changes for forced convection with different flow velocities. This behavior gains importance when the flow rate changes due to different HFOT. All the aforementioned aspects ultimately affect the LCOE.

7.3 Optimization Potential

The biggest optimization potential for S1 and S2 is seen by increasing the heat demand with, for instance, additional solar water heating (SWH) for a better use of free capacities, especially during summer month. This measure decreases the unexploited solar energy and therefore increases productivity of the entire system.

The relative low outlet temperature of the solar field is already beneficial when it comes to its optimization potential. Due to the small receiver area, the collector’s efficiency is less affected by its operating temperature and heat losses compared to the following pipes and TESS. For these reasons, the optimization of the solar field is of secondary importance.

A direct two-tank TESS was chosen to be modeled in MATLAB due to less detailed heat balance equations compared to those of the thermocline tank technology. However, with the obtained size of the two-tank TESS, the size of the thermocline TESS can be deducted given that the heat storage capacity is known (Libby 2010). The thermocline tank systems are expected to be about 30% more cost-effective than two-tank systems thus lowering the LCOE even further. This assumption has been commonly accepted and used in research papers by, for instance, Cocco & Serra (2015), Angelini et al. (2013) and Libby (2010). The model of the TESS itself can be further optimized by finding the optimal tradeoff between upper temperature level, heat losses from the system and tank insulation levels.

The inlet and outlet temperatures from the solar circle were chosen to be 120°C and 100°C respectively, mainly because the installed HE was designed for utilization without TESS. This temperature setting influences the fluid flow through the HE and the entire system, meaning
that a lower HFOT also lowers the fluid flow and thus the required TESS size since more heat can be utilized from the same storage volume.

However, to proceed with this optimization, a detailed study of TESS and HE parameters, such as TESS placement, outside or inside a building, insulation material costs, HE temperature settings and sizing, is required to determine the best tradeoff between temperature levels and heat losses leading to the minimum LCOE.
8 CONCLUSION

In this thesis work, the 200 kW solar-driven ORC-unit of Tianjin University was to be evaluated for distributed heating. The current system has been designed for research purposes associated with the ORC unit and will not be connected to the grid upon completion for power generation due to restrictions by the Chinese law and difficulties with the licensing procedure. To this end, a heating purpose is sought to utilize the harvested solar energy at times without ongoing experiments. On this score, four heat utilization scenarios were defined and the influences of different TESS sizes from 0 to 70 m³, HE capacities from 70 to 800 kW and varying HFOT from 50 to 100°C were investigated. The first scenario represents the current system on-site for space heating and cooling. It has been assumed that no changes to the current components, aside from lowering the HFOT and installing a TESS, can be made. The second scenario differs from the first in that also the HE is subject of upgrading. Industrial process heat 24 hours, 7 days a week and 16 hours, 7 days a week were assumed in scenario 3 and 4, respectively. All the above mentioned upgrades were considered and investigated. The results have been analyzed in terms of economic (LCOE), technical (solar fraction and efficiency) and environmental (CO₂ emissions) performance among which the LCOE is considered as decisive to find the most cost-effective heating purpose. The following conclusions are drawn:

(1) The current system in S1 for space heating and cooling is most viable without upgrading or change of parameters. With an LCOE of 0.187 $/kWh it cannot compete with merely diesel-based auxiliary heating and cooling at about 0.178 $/kWh or with traditional heating with air conditioners at about 0.112 $/kWh.

(2) The upgraded system in S2, for space heating and cooling, becomes more cost-effective with larger HE capacities due to more utilization of the solar energy which results in less wasted energy. Nevertheless, the obtained LCOE of 0.145 $/kWh are still above that for traditional heating.

(3) The use of an upgraded system for industrial heat purposes in S3 and S4 have shown to be most viable due to its constant heat demand throughout the year. Both LCOE of 0.106 $/kWh for S3 and 0.098 $/kWh for S4 are below those obtained for diesel-based auxiliary and traditional heating.

(4) If the system is going to be used for space heating and cooling then the heat demand, during summer months, has to be increased by, for instance, a solar water heating system in order to improve productivity hence lowering the LCOE. If the system is going to be used for industrial heating purposes then a heat consumer must be sought in the close vicinity.
Concluding, the results presented in this paper for the preliminary design study suggest that the Binhai CSP plant should be used for industrial process heat. The obtained LCOE outline the magnitude of the upgrade in terms of costs and component sizing. To this end, further research work is required to obtain optimized settings of all components once the designated purpose has been agreed upon.
9 SUGGESTION FOR FURTHER WORK

The output of the solar field was not experimentally validated since operation under design conditions was not possible due to lack of heat consumers and cooling equipment. Still, results from the solar field sub-model are seen as reasonable on the basis of the consistency with values provided by the manufacturer Hi-Min Solar and reviewed literature, see 4.4 Solar Field. Nevertheless, once the solar field is ready for operation, the performance of the solar collectors needs to be evaluated by following, for instance, the testing procedure described in ASHRAE Standard 93-2010 Methods for Testing to Determine the Thermal Performance of Solar Collectors.

Using the system for space heating and cooling becomes only viable when increasing the heat demand with, for instance, a SWH base load throughout the year. For this reason, the impact of different SWH base loads on LCOE was studied to identify the range of the base load at which the system achieves the lowest LCOE. As expected from the similar setup in S3, a base load of 80 kW, TESS size of 40 m³, HE capacity of 250 kW and an HFOT of 50°C result in the lowest LCOE of 0.114 $/kWh (figure 43) which is marginally higher than that of traditional heating. In this case, the annual wasted energy is decreased from 68% to 21% (figure 44). It shall be noted, that in case of provided incentives for CSP plants and a more optimized system, the LCOE will fall below that of traditional heating thus being more cost-effective. Therefore, a further analysis of local conditions regarding space heating, cooling and SWH requirements is proposed.

![Figure 43: LCOE for different TESS sizes and SWH loads at 50°C lower set temperature](image1)

![Figure 44: Heat load and wasted energy for S1 with and without 80 kW SWH](image2)

In order to use the solar system for industrial process heat, a further study of potential heat consumers in the close vicinity is required. A brief examination of the area shows that the factory producing rocket fuels, located next to the power plant, might be suited. From observation it is known that this factory operates a two-shift day and uses a 250 kW diesel-driven boiler to supply heat to its manufacturing facilities. However, neither heat demands nor temperature levels of the underlying process are known hence making it difficult to assess the suitability of the solar plant for this application. For this reason, a meeting
between the University and the operating company of that factory is recommended to evaluate the cooperation potential and the further course of action.

Once a decision has been made to use the system for either space heating and cooling, including SWH, or industrial process heat, the model can be designed with a greater detail from which the results can be analyzed according to the studies: *Calculation of performance indicators for solar cooling, heating and domestic hot water systems* by Nowag et al. (2012) and *Definition of performance figures for solar and heat pump systems* by Malenković (2012).
REFERENCES


APPENDICES

Appendix A: MATLAB CODE – Solar-driven DES – Main.m

%--------------------------------------------------------------------------
% Master Thesis
% Solar-driven Distributed Heating System
% Tobias Jahn
%--------------------------------------------------------------------------
% Program to calculate energy balances throughout the system
% Last change: 2015/08/01

clear all;
close all;
c1c;

global Q_useful
global Q_need
global Q_tank
global Q_tankC
global Q_loss
global Q_lossC
global Q_storage
global Q_aux
global Q_supply
global Q_waste
global Q_reheat
global T_outside
global T_tank
global T_cold
global M_tank
global M_cold

%---------------------------------INPUT------------------------------------
%-------------------------------DESIGN DATA-------------------------------
Mode=1;  % Determines the Temperature Set points (TSP)
WE=0;   % Determines whether weekend is considered for TSP
time=8760;   %hr   length of simulation

%-------------------------------LOOP VALUES AND INTIAL GUESSES-----------------------------
t(1)=1;  %time vector required for plots

%-------------------------------DATA FROM TRNSYS-------------------------------
TRNSYS_data = importdata('Q.txt',' ');  %reads the result file from the
                                    %solar field
Q_useful=TRNSYS_data.data(:,2);  %kJ/hr
T_outside=TRNSYS_data.data(:,4);  %°C
%-------------------CALCULATIONS-------------------
%--------DETERMINATION OF THE HEAT DEMAND WITH THE DEGREE DAY METHOD--------
Qn_need=DegreeDay(Mode,WE); %kJ/m²hr required for MATLAB calc.
Qn_need_wC=DegreeDay_wC(Mode,WE); %kJ/m²hr required for EXCEL template
Qn_max=max(Qn_need); %kJ/m²hr

%-------------------Heating Season-------------------
for Tset=50:10:100 %Varying the system base temperature
  %comment either (1) or (2)
  Cap=250; %kW used for varying heat exchanger capacities (1)
  A=Cap*3600/Qn_max; %m² used to determine heat demand (1)
  % A=HE_Capacity(Tset,Qn_max); %m² used for fixed HE capacities (2)
  Q_need=Qn_need*A; %kJ/hr heat demand
  Q_need_wC=Qn_need_wC*A; %kJ/hr heat demand for EXCEL template

%-------------------EXCEL TEMPLATE PREPARATION-------------------
excelFileName = 'Results_template_HE.xlsx';
excelFilePath = pwd; % Current working directory.
% Open Excel file.
objExcel = actxserver('Excel.Application');
objExcel.Workbooks.Open(fullfile(excelFilePath, excelFileName));
C={'Results_','Tset',num2str(Tset,'%1d '),'.xlsx'};
newfilename=strrep(reshape(char(C)',1,[]),'
','
');
objExcel.ActiveWorkbook.SaveAs(fullfile(pwd,newfilename));
objExcel.ActiveWorkbook.Close;
objExcel.Quit;
objExcel.delete;

%-------------------DETERMINATION OF ALL HEAT FLOWS AND HEAT STORED IN TANK-------------------
k=1;
for V_tank=10:10:70 %change of TESS sizes from 10 to 70m³
  i=2;
  while i<=time
    %Subroutine for TESS
    [mfi(i),mfo(i),h_tank(i),state(i)]=TES_VT_wCS(i,V_tank,Tset);
    t(i)=i;
    Qp_useful(i)=Q_useful(i); % to plot Q_useful
    Qp_need(i)=Q_need(i); % to plot Q_need
    Q_in(i)=Qp_useful(i)-Q_reheat(i)-(Q_tank(i)-Q_tank(i-1))...
    -Q_loss(i)-Q_supply(i)-Q_waste(i);
    Q_out(i)=Qp_need(i)-Q_aux(i)-Q_supply(i);
    M(i)=mfi(i)-(M_tank(i)-M_tank(i-1))-mfo(i);
    i=i+1;
  end
%Preparation of results for and saving in EXCEL template

Summaryheader(k)=V_tank;
k=k+1;
sheetname=num2str(V_tank);
Ergheader={'Volume',sheetname,'','','','','','','','','','','','','','Heat and Mass Balance','','','','','','','','','','','','','','','Qneed','Quseful','Qaux','Qsupply','Qtank','Qstorage','Qloss','Qwaste','Ttank','T_outside','Mtank','mfi','mfo','Htank','state','Qin','Qout','M','QlossC','Tcold','Mcold'};
Erg=[t;Qp_need;Qp_useful;Q_aux;Q_supply;Q_tank;Q_storage;Q_loss;...Q_waste;T_tank;T_outside;M_tank;mfi;mfo;h_tank;state;Q_in;...Q_out;M;Q_lossC;T_cold;M_cold]';
xlswrite(newfilename,Ergheader,sheetname,'A1');
xlswrite(newfilename,Erg,sheetname,'A3');
end
xlswrite(newfilename,Summaryheader,'Summary','D2');
xlswrite(newfilename,Tset,'0','E1');
xlswrite(newfilename,Q_need_wC','0','B3');
xlswrite(newfilename,Q_use_excel','0','C3');
xlswrite(newfilename,Cap,'LCOE','O17');
end

%---------------------------------------------------------------
%---------------------------------------------------------------
%---------------------------------------------------------------
%

Appendix B: MATLAB CODE – TESS – TES_VT_wCS.m

% Master Thesis
% Solar-driven Distributed Heating System
% Tobias Jahn

% Last change: 2015/08/01

function [mfi,mfo,h_tank,state] = TES_VT_wCS(i,V,Tset)

%Fluid Therminol 55
cp=2.01; %kJ/kgK    heat capacity of therminol 55 @ 120°C
density=852; %kg/m³   density of therminol 55 @ 120°C
%TES tank

T_in=125; % °C inlet tank temperature
HD_ratio=3; % height to diameter ratio
d=(4*V/(pi()*HD_ratio))^(1/3); % m diameter of tank
h_max=d*HD_ratio; % m height of tank
h_safe=0.2; % m minimum level of liquid in tank
Mt_min=density/4*pi()*d^2*h_safe; % kg min mass in tank
Mt_max=density/4*pi()*d^2*h_max; % kg max mass in tank
U_d=3; % kJ/hr,m²,K dry loss coefficient
U_w=4; % kJ/hr,m²,K wet loss coefficient

---------------------------------------------------------------------
CALCULATIONS
---------------------------------------------------------------------

% LOOP AND INITIAL VALUES

if i==2
    T_tank(i-1)=30; % °C start temp of tank
    T_cold(i-1)=80; % °C start temp of cold
    M_tank(i-1)=Mt_min; % kg minimum storage level
    M_cold(i-1)=Mt_max-Mt_min;
    Q_tank(i-1)=cp*M_tank(i-1)*(T_tank(i-1)-Tset); % heat at min storage lvl
elseif i==7002
    T_tank(i-1)=122;
    M_tank(i-1)=Mt_max;
    Q_tank(i-1)=cp*M_tank(i-1)*(T_tank(i-1)-Tset); % heat at max storage lvl
end

T_tank(i)=T_tank(i-1); % °C initial guess for iteration
T_t_temp=-100; % to enter the loop first time
TOL=0.0001; % °C temperature difference at which iteration stops
dt=1; % time step (do not change)

---------------------------------------------------------------------

DETERMINATION OF UA DEPENDING ON THE FLUID LEVEL IN TANK
---------------------------------------------------------------------

h_real=M_tank(i-1)/(density/4*pi()*d^2);
UA_hw=U_w*h_real*2*pi()*d/2+U_w*pi()*d^2/4;
UA_hd=U_d*(h_max-h_real)*2*pi()*d/2+U_d*pi()*d^2/4;
UA_cw=U_w*(h_max-h_real)*2*pi()*d/2+U_d*pi()*d^2/4;
UA_cd=U_d*h_real*2*pi()*d/2+U_d*pi()*d^2/4;
UA_h=UA_hw+UA_hd; % hot tank
UA_c=UA_cw+UA_cd; % cold tank

---------------------------------------------------------------------

DETERMINATION OF STORAGE OPERATION STATUS
---------------------------------------------------------------------

% NOTE: control value is needed for cases where boundary values are reached
if T_tank(i-1) < Tset
    control=0;
    Q_storge(i)=0;
    mfo=0;
else
    control=-1;
    Q_min=cp*Mt_min*(T_tank(i-1)-Tset);
    Q_storage(i)=Q_tank(i-1)-Q_min;
    mfo=Q_need(i)/(cp*(T_tank(i-1)-Tset)); % kg/hr guessed outlet mass flow
end

mfi=Q_useful(i)/(cp*(T_in-T_cold(i-1))); % kg/hr inlet mass flow

---------------------------------------------------------------------
%----------------HEAT AND MASS BALANCES FOR DIFFERENT OPERATION MODES----------------
%NOTE: While loop in case of wrong mode due to calculations close to the
%boundary values
while abs(T_tank(i)-Tt_temp) > TOL

%............................Mode 1 - Charging to Maximum Level..........................
%NOTE: Charging to maximum storage level, excessive heat is bypassed and
%wasted (amount of wasted heat saved in Q_waste)
if M_tank(i-1)+mfi-mfo > Mt_max && T_tank(i-1) > Tset
    state=1;
    mfi_temp=mfi;

%Determination of mixing temperature in tank---------------------------
while abs(T_tank(i)-Tt_temp) > TOL
    mfi=Mt_max-M_tank(i-1)+mfo;
    M_tank(i)=M_tank(i-1)+(mfi-mfo)*dt;
    Tt_temp=T_tank(i);
    T_tank(i-1)=mfi*Tin+Tset*(M_tank(i)-M_tank(i-1)-mfi+mfo)+...     
                  M_tank(i-1)*T_tank(i-1)+UA_h/cp*T_outside(i)/(M_tank(i)...  
                  +UA_h/cp+mfo);

    %Calculating a better value for outgoing mass flow
    mfo=Q_need(i)/(cp*(T_tank(i)-Tset));
end

%--------------------------------------------------------------------------
Q_supply(i)=Q_need(i);          %kJ/hr
Q_aux(i)=0;                     %kJ/hr
Q_storage(i)=cp*(T_tank(i)-Tset)*M_tank(i)-Mt_min;    %kJ/hr
m_waste=mfi_temp-mfi;           %kg/hr
Q_waste(i)=m_waste*cp*(Tin-Tset); %kJ/hr
Q_loss(i)=UA_h*(T_tank(i)-T_outside(i)); %kJ/hr
Q_tank(i)=cp*M_tank(i)*(T_tank(i)-Tset); %kJ/hr
%--------------------------------------------------------------------------

%............Mode 2 - Charging/Discharging within Tank boundaries ...........
%NOTE: Neither the upper nor the lower tank fluid level is reached
elseif M_tank(i-1)+mfi-mfo > Mt_min && T_tank(i-1) > Tset && control==1
    state=2;

%Determination of mixing temperature in tank---------------------------
while abs(T_tank(i)-Tt_temp) > TOL
    M_tank(i)=M_tank(i-1)+(mfi-mfo)*dt;
    Tt_temp=T_tank(i);
    T_tank(i)=mfi*Tin+Tset*(M_tank(i)-M_tank(i-1)-mfi+mfo)+...     
                  M_tank(i-1)*T_tank(i-1)+UA_h/cp*T_outside(i)/(M_tank(i)...  
                  +UA_h/cp+mfo);

    %Calculating a better value for outgoing mass flow
    mfo=Q_need(i)/(cp*(T_tank(i)-Tset));
end

%--------------------------------------------------------------------------
%check if real mfo would change operation mode and if so then set back
%of tank temperature to initial guess
if M_tank(i-1)+mfi-mfo < Mt_min
    T_tank(i)=T_tank(i-1);
else
    Q_supply(i)=Q_need(i);          %kJ/hr
    Q_aux(i)=0;                     %kJ/hr
    if T_tank(i) < Tset
        Q_storage(i)=0;
end

xxii
else
Q_storage(i)=cp*(T_tank(i)-Tset)*(M_tank(i)-Mt_min); %kJ/hr
end
Q_waste(i)=0; %kJ/hr
Q_loss(i)=UA_h*(T_tank(i)-T_outside(i)); %kJ/hr
Q_tank(i)=cp*M_tank(i)*(T_tank(i)-Tset); %kJ/hr
end
%.................................................................

%..................Mode 3 - Discharging to safety level..................
%NOTE: When there is heat for discharge left but reaches the safety level
elseif (M_tank(i-1) > Mt_min && control==-1) || (-Q_tank(i-1)+UA_h*... 
(T_tank(i-1)-T_outside(i)) < Q_useful(i) && control==0)
state=3;
M_tank(i)=Mt_min;

%Determination of mixing temperature in tank-----------------------------
while abs(T_tank(i)-Tt_temp) > TOL
T_t_temp=T_tank(i);
T_tank(i)=(mfi*Tin+Tset*(M_tank(i)-M_tank(i-1)-mfi+mfo)+... 
M_tank(i-1)*T_tank(i-1)+UA_h/cp*T_outside(i))/(M_tank(i)... 
+UA_h/cp+mfo);
Q_storage(i)=cp*(M_tank(i-1)+mfi-Mt_min)*(T_tank(i)-Tset);
end

%check if stored heat after mixing is higher then head demand to
%determine correct outgoing mass flow
if Q_storage(i) > Q_need(i)
  mfo=Q_need(i)/(cp*(T_tank(i)-Tset)); %kg/hr
  Q_supply(i)=Q_need(i); %kJ/hr
  M_tank(i)=M_tank(i-1)+mfi-mfo;
else
  mfo=Q_storage(i)/(cp*(T_tank(i)-Tset)); %kg/hr
  Q_supply(i)=Q_storage(i); %kJ/hr
  M_tank(i)=Mt_min; %kg
end

%.................................................................

%check if tank temperature would change operation mode and if so then
%set back of tank temperature to initial guess and change control value
if T_tank(i) < Tset
  control=1;
  T_tank(i)=T_tank(i-1);
else
  Q_aux(i)=Q_need(i)-Q_supply(i); %kJ/hr
  Q_storage(i)=Q_storage(i)-Q_supply(i); %kJ/hr
  Q_waste(i)=0; %kJ/hr
  Q_loss(i)=UA_h*(T_tank(i)-T_outside(i)); %kJ/hr
  Q_tank(i)=cp*M_tank(i)*(T_tank(i)-Tset); %kJ/hr
end
%.................................................................

%.........................Mode 4 - BYPASS..........................
%NOTE: No heat to / from the tank, only heat losses and complete auxiliary
%heating
elseif mfi == 0 && M_tank(i-1) <= Mt_min
state=4;

%Determination of mixing temperature in tank-----------------------------
while abs(T_tank(i)-Tt_temp) > TOL
T_t_temp=T_tank(i);
Q_loss(i)=UA_h*(T_tank(i)-T_outside(i)); %kJ/hr

xxiii
\[ T_{\text{tank}}(i) = T_{\text{tank}}(i-1) - \frac{Q_{\text{loss}}(i)}{cp \cdot M_{\text{tank}}(i-1)}; \quad ^\circ\text{C} \]

\[ Q_{\text{supply}}(i) = 0; \quad \%\text{kJ/hr} \]
\[ M_{\text{tank}}(i) = M_{\text{tank}}(i-1); \quad \%\text{kg} \]
\[ Q_{\text{aux}}(i) = Q_{\text{need}}(i); \quad \%\text{kJ/hr} \]
\[ Q_{\text{storage}}(i) = 0; \quad \%\text{kJ/hr} \]
\[ mfo = 0; \quad \%\text{kg} \]
\[ Q_{\text{waste}}(i) = 0; \quad \%\text{kJ/hr} \]
\[ Q_{\text{tank}}(i) = cp \cdot M_{\text{tank}}(i) \cdot (T_{\text{tank}}(i) - T_{\text{set}}); \quad \%\text{kJ/hr} \]

\[
%.........................\text{Mode 5 - HEAT TO STORAGE USEABLE}.........................
%NOTE: when in the morning T\_tank below T\_set
\]

\[
\text{elseif } -Q_{\text{tank}}(i-1)+UA_{\text{h}} \cdot (T_{\text{tank}}(i-1)-T_{\text{outside}}(i)) > Q_{\text{useful}}(i) \&\& ... \\
\quad mfi > 0 \quad || \quad \text{control}==1
\]
\[
\text{state}=5;
\]

\[
%\text{Determination of mixing temperature in tank}------------------------------------
M_{\text{tank}}(i) = M_{\text{tank}}(i-1)+mfi;
T_{\text{tank}}(i) = (mfi \cdot (T_{\text{in}}-T_{\text{set}}) + M_{\text{tank}}(i-1) \cdot (T_{\text{tank}}(i-1)-T_{\text{set}}) + M_{\text{tank}}(i) \cdot T_{\text{set}} + UA_{\text{h}}/cp \cdot T_{\text{outside}}(i))/(M_{\text{tank}}(i)+UA_{\text{h}}/cp);
\]

\[
%\text{check if tank temperature would result in heat storage}
if \quad T_{\text{tank}}(i) > T_{\text{set}}
\quad Q_{\text{storage}}(i) = Q_{\text{tank}}(i) - Q_{\text{min}}; \quad \%\text{kJ/hr}
else
\quad Q_{\text{storage}}(i) = 0; \quad \%\text{kJ/hr}
end
\]

\[
Tt_{\text{temp}}=T_{\text{tank}}(i); \quad \%\text{to leave main while loop}
\]
\[ Q_{\text{supply}}(i) = 0; \quad \%\text{kJ/hr} \]
\[ Q_{\text{waste}}(i) = 0; \quad \%\text{kJ/hr} \]
\[ Q_{\text{loss}}(i) = UA_{\text{h}} \cdot (T_{\text{tank}}(i)-T_{\text{outside}}(i)); \quad \%\text{kJ/hr} \]
\[ mfo = 0; \quad \%\text{kg} \]

\[
%.........................\text{Mode 6 - RUN THROUGH}..........................\n%NOTE: When mass flow in = mass flow out (M\_tank(i)=constant), it is the %case when tank temperature is higher than T\_set and only partial auxiliary %heating is required
\]

\[
\text{else}
\quad state=6;
\quad M_{\text{tank}}(i) = M_{\text{tank}}(i-1);
\quad mfo = mfi;
\]

\[
%\text{Determination of mixing temperature in tank}-------------------------------
T_{\text{tank}}(i) = (mfi \cdot T_{\text{in}} + M_{\text{tank}}(i) \cdot T_{\text{tank}}(i-1) + UA_{\text{h}}/cp \cdot T_{\text{outside}}(i))/(M_{\text{tank}}(i)+UA_{\text{h}}/cp+mfi); \\
\]

\[ Q_{\text{storage}}(i) = mfi \cdot cp \cdot (T_{\text{tank}}(i)-T_{\text{set}}); \quad \%\text{kJ/hr} \]

\[
%\text{check if stored heat after mixing is higher then head demand to} \%\text{determine correct outgoing mass flow}
if \quad Q_{\text{storage}}(i) > Q_{\text{need}}(i)
\]
mfo=Q_need(i)/(cp*(T_tank(i)-Tset));  \( \text{kg/hr} \)

else

Q_supply(i)=Q_storage(i);  \( \text{kJ/hr} \)
TT_temp=T_tank(i);
Q_aux(i)=Q_need(i)-Q_supply(i);  \( \text{kJ/hr} \)
Q_storage(i)=Q_storage(i)-Q_supply(i);  \( \text{kJ/hr} \)
Q_waste(i)=0;  \( \text{kJ/hr} \)
Q_loss(i)=UA_h*(T_tank(i)-T_outside(i));  \( \text{kJ/hr} \)
Q_tank(i)=cp*(M_tank(i-1)+(mfi-mfo))*(T_tank(i)-Tset);  \( \text{kJ/hr} \)
end

%..........................................................................
end
%--------------------------------------------------------------------------
end
%--------------------------------------------------------------------------
%--------------------------------------------------------------------------

%-----------------------------------------ADDITIONAL OUTPUTS-----------------------------------------
%Determining mixing temperature in Cold Water Tank
if state==1
mfi_c=mfi_temp;
else
mfi_c=mfi;
end

M_cold(i)=Mt_max-M_tank(i)+Mt_min;
T_cold(i)=(M_cold(i)*Tset+M_cold(i-1)*(T_cold(i-1)-Tset)+mfi_c*Tset+...%UA_c/cp*T_outside(i))/(M_cold(i)+mfi_c+UA_c/cp);
Q_lossC(i)=UA_c*(T_cold(i)-T_outside(i));
Q_reheat(i)=mfi_c*cp*(Tset-T_cold(i-1));
Q_tankC(i)=M_cold(i)*cp*(T_cold(i-1)-Tset);

h_tank=M_tank(i)/(density/4*pi()*d^2);
%--------------------------------------------------------------------------

Appendix C: MATLAB CODE – Degree-Days – DegreeDay_wC.m

%--------------------------------------------------------------------------
% Master Thesis
% Solar-driven Distributed Heating System
% Tobias Jahn
%--------------------------------------------------------------------------
% Subroutine: Estimation of Heat and Cooling Demand with Degree-Day-Method
% Last change: 2015/08/01

function [QDD] = DegreeDay_wC(Mode,WE)
global T_outside

%-----------------------------------------INPUT-----------------------------------------
%---------------------------------CHARACTERISTICS OF CHINESE REFERENCE BUILDING-----------------------------------------
%Shape
w=50;  \( \text{m} \)  \( \text{width} \)
l=30;  \( \text{m} \)  \( \text{length} \)
h=85;  \( \text{m} \)  \( \text{height} \)

%Mean Values from Feng's et al "Evaluation of Energy Savings..."
WWR=0.4;  \( \text{Window-to-Wall ratio} \)
floors=18;
Heat transfer coefficients from "Design Standard for Energy Efficiency..."
GB 50189-2014 (MoHURD)

$U_{\text{roof}}=0.425; \quad \text{W/m}^2\text{K}$
$U_{\text{wall}}=0.475; \quad \text{W/m}^2\text{K}$
$U_{\text{04WWR}}=2.3; \quad \text{W/m}^2\text{K}$
$U_{\text{floor}}=1.5; \quad \text{W/m}^2\text{K}$

% Calculated properties

\begin{align*}
A_{\text{wall}} &= 2 \times w \times h + 2 \times l \times h; \quad \text{m}^2 \\
A_{\text{roof}} &= w \times l; \quad \text{m}^2 \\
A_{\text{floor}} &= \text{floors} \times A_{\text{roof}}; \quad \text{m}^2 \\
U_{\text{A}} &= A_{\text{roof}} \times U_{\text{roof}} + \text{WWR} \times A_{\text{wall}} \times U_{\text{04WWR}} + (1 - \text{WWR}) \times A_{\text{wall}} \times U_{\text{wall}} + \ldots \\
&\quad A_{\text{roof}} \times U_{\text{floor}}; \quad \text{W/K}
\end{align*}

% Calculations

% Setup of temperature set point vector

T_setpt_H = [5 5 5 5 12 15 17 17 17 17 17 17 17 15 12 5 5 ... 5 5; 5 5 5 5 5 5 5 5 5 5 5 5 5 5 5 5 5 5 5 5; 22 22 22 22 22 22 22 22 22 22 22 22 22 22 22 22 22 22 22; 13 13 ... 13 13 13 18 18 18 18 18 18 18 18 18 18 18 18 13 13 13 13 13 13];

T_setpt_C = [35 35 35 35 35 35 27 22 22 22 22 22 22 22 22 22 22 22 22 22 22 22; 35 35 35 35 35 35 35 35 35 35 35 35 35 35 35 35 35 35 35 35 35 35];

% NOTE: Row one is typical Tianjin heating schedule from Feng et.al
% Row two is used if heating schedule distinguishes between weekend
% and week days
% Row three and for one and two for Chicago, USA as reference

hr=1; \quad \% initial value for loop (starting hour)
day=1; \quad \% initial value for loop (starting day)
for i=1:8760
    if i <= 2400 || i > 7008 \% then heating degree days (winter)
        if WE==1 \&\& (day==6 \|\| day==7)
            Ta_setpt(i)=T_setpt_H(Mode+1,hr); \% °C
        else
            Ta_setpt(i)=T_setpt_H(Mode,hr); \% °C
        end
    elseif i > 3216 \&\& i <= 5568 \% then cooling degree days (summer)
        if WE==1 \&\& (day==6 \|\| day==7)
            Ta_setpt(i)=T_setpt_C(Mode+1,hr); \% °C
        else
            Ta_setpt(i)=T_setpt_C(Mode,hr); \% °C
        end
    end
end
else
    hr=1;
    if day==7
        day=1;
    else
        day=day+1;
    end
end
else
    Ta_setpt(i)=T_outside(i);
end

%------------------------------------------------------------------------
%------------------------------------------------------------------------

%-----------------HEAT DEMAND BASED ON DEGREE DAY METHOD-----------------
for i=1:8760
    if i <= 2400 || i >7008
        QDD(i)=UA*(T_outside(i)-Ta_setpt(i))*3.6/A_floor*-1; %kJ/m²
    else
        QDD(i)=UA*(T_outside(i)-Ta_setpt(i))*3.6/A_floor; %kJ/m²
    end
    if QDD(i) < 0
        QDD(i)=0;
    end
end
%------------------------------------------------------------------------
%------------------------------------------------------------------------