

The simulation of a commercial-scale, hybrid solar and heat pump water heating system

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Dissertation accepted in fulfilment of the requirements for the degree *Master of Engineering in Mechanical Engineering* at the North West University

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MAGISTER DISSERTATION

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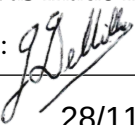
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November 28,2020

Declaration of Authorship

I, Jean-Pierre de Villiers, declare that this dissertation titled, "The simulation of a commercial-scale, hybrid solar and heat pump water heating system" and the work presented in it are my own. I confirm that:

- This work was done wholly while part-time in candidature for a research degree at this University.
- Where I have consulted the published work of others, this is always clearly attributed.
- Where I have quoted from the work of others, the source is always given. With the exception of such quotations, this dissertation is entirely my own work.
- I have acknowledged all main sources of help.
- Where the dissertation is based on work done by myself jointly with others, I have made clear exactly what was done by others and what I have contributed myself.
- The developed computer code of this research study is that of my own, and if a derivation was made from other sources, this has been clearly indicated.

Signed: 

Date: 28/11/2020

“Except the Lord build the house, they labour in vain that build it: except the Lord keep the city, the watchman waketh but in vain.”

Psalm 127:1

Abstract

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School of Mechanical and Nuclear Engineering

Master of Engineering in Mechanical Engineering

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by Jean-Pierre de Villiers

Due to the rotation of the earth and climatic conditions, the available solar radiation varies, not only in the 24-hour period but also daily. The latter poses a significant challenge to the designer of solar water heating systems. In order to ensure the constant availability of sufficiently hot water, large commercial scale or industrial installations use heat pumps. The combination of solar and heat pump technology has been introduced in various configurations to the commercial market to curb electricity usage. Several factors influence the performance of such systems: The size and performance of the solar collectors, the capacity of the storage vessels, the hot water demand profile, the size and performance of the heat pump, control systems and climatic conditions. The design and optimisation of such systems are, therefore, not straightforward.

Simulation programs are often used for the design and optimisation of such systems. Researchers and designers use simulations to design the system for maximum efficiency, user comfort or even a combination thereof. The problem, however, with most component-based simulations is that they often require specialist knowledge not only for the use and input requirements but also for the realistic calibration of the program. There exists a need to implement a simulation that uses readily available manufacturer performance curves and matrices of the various sub-components that make up the system.

Parametric studies and optimisation methods can often take long periods of time on simulation packages. Optimisation in a limited number of independent parameters becomes near impossible when a single simulation can take days to complete. This problem becomes even more evident when extended periods are simulated. For optimisation of such systems, a quick and accurate solver is required that can manage continually evolving boundary conditions. The conducting of parametric studies design simulations and stochastic optimisation studies on SWH/ASHP (Solar Water Heater and Air Source Heat Pump) systems, therefore, mandates the use of simulations programs with swift simulation times.

To this end, the study developed two simulations in EES®. Firstly, a reduced simulation model for the integrated, long-term, transient thermal simulation of a hybrid SWH/ASHP system. Secondly, a detailed model was developed to serve as a comparison to the reduced simulation. The simulations were verified using mathematical principles and a simulation built-in MATLAB® Simulink®. Consequently, three case studies were set up in MATLAB Simulink. The reduced simulation produced steady-state errors between 2.13 and -4.13 [%] ($\Rightarrow$ 0.49[°C] to -1.53[°C]) and a maximum error of -30.24 [%] ($\Rightarrow$ -3.88[°C]) in the most transient region after system start-up. The use of the reduced simulation was thus limited to non-startup conditions. The reduced simulation was able to simulate the most detailed case study nearly 31 times quicker than the Simulink equivalent. The simulation was validated against a real-world application of a commercial SWH/ASHP installed on a hospital in Potchefstroom, South-Africa.

For the reduced and detailed simulations, the average rate of energy error between the various energy fluxes was 2.54 [%] and 2.22 [%], respectively.

Subsequently, the simulation was used to run parametric studies on the components of the SWH/ASHP system. The reduced simulation was able to simulate six design-year simulations with a 60 [s] time-step simultaneously on a standard notebook and took roughly 13 hours to complete. The influence of different climatic regions in Southern-Africa, the aspect ratio and capacity of the storage vessels, ASHP capacity, SWH controller and the number of SWH manifolds on system performance was investigated. A notable finding was that additional SWH banks decrease ASHP electrical consumption at a lesser rate than the rate at which thermal energy addition increases from the SWH banks. The finding can be explained since the ASHP is forced to add the remainder of the required thermal energy under less favourable COP conditions. The reduced simulation can form the foundation for further studies by taking note of the computational expense and simulation accuracy. Future work can include the multi-objective optimisation of SWH/HP hybrid systems and their controllers.

Keywords: Reduced-order models, Long-term simulation, Solar heat pump systems, Parametric study, Fast simulation, Simulation development

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First and foremost, I would like to thank the Lord for giving me this opportunity and the insight and strength to have completed the research study. Conditions at times might have been less than ideal; but He carried my every step of the way. In the most pressing technical problems, I was saved by His grace. In Thy light.

To do research on a topic and write your findings in a dissertation amounts to the personal equivalent of a space program. There are no subjects with fixed outcomes; no curriculum that guides your every step. Only a goal. Along with a deadline. No roadmap to it. A goal that will demand your utmost best. An enduring onslaught on your mental, social and physical needs. Some might ask: why then? There is no universal answer to this question. Everyone will have their own answer. Or at least aim to seek it.

However, in the end, the goal of landing the proverbial spacecraft on a distant asteroid; that is what keeps you going. This journey cannot be conducted alone. Like the thrusters on a spaceship, occasional guidance is required to reach the final destination. The brief bursts of these various thrusters ensure a successful mission. These burst ensure that your energies do not miss the target. The persons to follow was my 'thrusters' throughout the journey and without their occasional help; this study would have been painstakingly difficult.

I would like to start by thanking my study leader, Dr Willem van Niekerk for his willingness to take on a part-time student. Dr Willem was always only a phone call away and willing to give his advice and wisdom in the form of various metaphors and analogies. These metaphors and analogies would touch a range of topics, from technical questions to writing style and the flow of transferring your ideas. Some analogies had to be repeated to me, but in the end, it sank in. Apart from this, he always promptly replied to questions and was keen on helping with administrative issues and instrumentation related questions. Finally, I would like to thank Dr Willem for lending the occasional mental support when I needed it most.

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The actual list of people is rather significant. A simple technical question from a friend; at the right time, goes a long way. Explaining the topic to an interested family member or colleague helps process your thoughts. To all of these people who unknowingly steered me in the right

direction: I thank you. Although the work and research product are that of my own; these encounters kept me on course.

Why do the research then? I believe it builds character. It forces you to manage your expectations and emotional well-being. It forces you to think critically, deductively and sometimes inductively, to discover your strength and weaknesses, to embrace the deficiencies and the contributions of others' work. To acknowledge their contributions and make it your own. If done correctly, it will engrave life-long tenacity. All of the above; are good qualities to forge.

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List of Abbreviations

ASHP	Air Source Heat Pump
CFD	Computational Fluid Dynamics
CFL	Courant Friedrichs Lewy
COP	Coefficient Of Performance
CSIR	Council for Scientific and Industrial Research
CT	Current Transformer
DB	Dry Bulb
DE	Differential Equation
DWH	Domestic Water Heating
EES	Engineering Equation Solver
GDP	Gross Domestic Product
GHG	Green House Gasses
GHI	Global Horizontal Irradiance
GSHP	Ground Source Heat Pump
HVAC	Heating Ventilation Aireat Conditioning
HP	Heat Pump
IMEX	IMplicit EXplicit
KPI	Key Performance Indicator
kWh	killoWatt-hours
LMTD	Log Mean Temperature Difference
MME	Mass Momentum Energy
MOL	Method Of Lines
MR	Multi Regression
ODE	Ordinary Differential Equation
OEM	Original Equipment Manufacturers
PBMR	Pebble Bed Modular Reactor
PDE	Partial Differential Equation
RH	Relative Humidity
RK	Runge Kutta
RMS	Root Mean Squared
RS	Response Surface
SCADA	Supervisory Control And Data Acquisition
SIMPLE	Semi-Implicit Method for Pressure Linked Equations
ST	Solar Technology
SWH	Solar Water Heater
VSD	Variable Speed Drive
WB	Wet Bulb

List of Symbols

$\dot{m}$	mass flow rate	kg s^{-1}
$\dot{Q}$	rate of energy/heat	W
$\dot{W}$	hydraulic/rate of work/electrical power	W
A	area	m^2
a	distance	m
C_p	specific heat at constant pressure	$\text{J}/(\text{kg} \times \text{K})$
d	diameter	m
D	diameter	m
E	approximation of specific energy	J kg^{-1}
e	specific total energy	J kg^{-1}
F	force	N
f	friction coefficient	dim
g	gravitational acceleration	m/s^2
G	irradiance	W/m^2
h	enthalpy	$\text{J kg}^{-1} \text{K}^{-1}$
H	hydraulic head	m
h	spatial distance between components	m
I	electrical current	A
I	irradiance	W/m^2
j	time step interval	dim
k	finite number of time-steps	s
k	thermal conductivity	$\text{W m}^{-1} \text{K}^{-1}$
k	time step size	s
L	length	m
l	length of pipe	m
m	cell mass	kg
M	fixed lapse of time	s
M	number of discrete components	m
MW	molecular weight	dim
n	normal vector, storage vessel identifier	dim
N	number of components or observations	dim
P	power	$\text{W (J s}^{-1})$
p	pressure	kPa
Q	energy	J
q	heat energy	J
q	specific heat	J kg^{-1}
Q	volumetric flow	m^3/s
R	friction force	N
R	thermal resistance	K W^{-1}
R, r	radius	m

T	temperature	K
t	time	s
u	axial velocity in the x-direction	m s^{-1}
u, U	internal energy	J kg^{-1}
UA	heat loss coefficient	W K^{-1}
V	axial speed	m s^{-1}
v	axial velocity in the y-direction	m s^{-1}
V	electrical potential	V
w	axial velocity in the z-direction	m s^{-1}
w	specific work	J kg^{-1}
W	work energy	J
x	quality	W
z	elevation	m
α	statistical coefficient	-
β	coefficient of performance	dim
β	coefficient of thermal expansion	K^{-1}
δ	difference	-
Δ	difference	-
ϵ	error	-
η	efficiency	dim
ω	humidity ratio	dim
ω	rotational speed	s^{-1}
ϕ	relative humidity	dim
ρ	density	kg m^{-3}
dV	differential control volume	m^3
a	dry air	-
atm	atmosphere	-
c	collector	-
cs	control surface	-
d	differential	-
db	dry bulb	-
g	saturation	-
i	inside	-
i	spatial identifier	-
j	time-step index	-
o	initial	-
o	outside	-
o	total/static	-
rms	root mean square	-
v	vapor	-
w	wetted area	-
wb	dry bulb	-
x	x direction	-
y	y direction	-
z	z direction	-

IN U LIG

Chapter 1

Introduction

1.1 Background

1.1.1 Humanity, our environment, and the world economy

The industrialised world consumes vast amounts of energy. Studies have shown that the planet takes a year and six months to restore our yearly consumption of some natural resources (Solarin and Bello, 2018). Expectations are that our electricity needs alone will amount to 26 trillion [kWh] in 2020 (Právělie, Patriche, and Bandoc, 2019). This enormous consumption of energy has an inevitable outcome: climate change due to the continued use of fossil fuels (Rahman and Miah, 2017; Martins, Felgueiras, and Smitková, 2018; Wang and Dong, 2019). There is a direct relationship between greenhouse gas emissions (GHG) and the gross domestic product (GDP) of a country. With the growth of the worlds' economies, more emissions are generated, accelerating the deterioration of the ecosystem. The utilisation of alternative sources and advances in renewable technologies are crucial to minimising GHG emissions and its environmental footprint (Arent, Wise, and Gelman, 2011; Uddin et al., 2017).

Recent studies of 151 countries have shown that there might exist an inverted U-shaped relationship (energy mix '*Kuznets Curve*') between real income and fossil fuel share of the energy mix (Kibria, Akhundjanov, and Oladi, 2019; Charfeddine and Mrabet, 2017; Bello, Solarin, and Yen, 2018). It has been found that as an economy matures, people increasingly value their environmental impact and thus are willing to pay more for a cleaner environment and energy supply. Renewable energy stands out as a promising replacement for fossil fuels. However, transitioning to renewable energy has the potential to restrict economic growth in the long term due to the higher initial capital investment (Barreto, 2018). Nevertheless, with no transition, a total collapse of the global economy will occur by exhausting the planets' fossil fuel reserves (Afonso, Marques, and Fuinhas, 2017). Reducing the productivity deficit between fossil fuels and renewable energy is vital in transitioning to a sustainable future (Barreto, 2018; Gielen et al., 2019). To accelerate the transition, it is the responsibility of researchers and engineers to increase the potency of renewable energy for the sake of the planet and humanity.

1.1.2 Solar and heat pump technologies

It has recently been suggested that a 100 [%] renewable energy system could be achievable in some European countries by 2050 (Zappa et al., 2019; Hansen, Mathiesen, and Skov, 2019). The transition would only be possible with considerable investment in storage technologies and a diversified renewable energy mix. A recent study showed that heating amounts to 47 [%] of the world's final energy utilisation and that there is a call to improve the sector's efficiency (Abas

et al., 2017). Thus, research and development in the heating sector can lead to meaningful reductions in the world's energy needs.

Of all the renewable energy sources, solar stands out as the largest for its global availability. The solar energy potential in some regions of the world is more than the current total energy demand in those regions (Timilsina, Kurdgelashvili, and Narbel, 2012). Nearly 40 [%] of the earth's landmass has excellent to superb ($>1800 \text{ [W/m}^2\text{]}$) GHI (Gross Horizontal Irradiance) solar energy potential (Prăvălie, Patriche, and Bandoc, 2019) as supported by Figure 1.1. The supply of solar thermal energy is dependent on a range of environmental and demand-side factors. The intermittent nature of solar energy requires auxiliary systems to be integrated with storage systems in household and industrial applications (Leckner and Zmeureanu, 2011; Farjana et al., 2018; Ge et al., 2018; Wu et al., 2018).

The provision of hot potable water is one of the basic requirements of modern society. The SWH (solar water heater) is the most significant contributor in the thermal applications sector and the most typical application of solar energy utilisation (Gautam et al., 2017). For a sustainable future, research in renewable energy modelling places both solar energy and heat pump technology as vital contributors (Pursiheimo, Holttinen, and Koljonen, 2019; Gielen et al., 2019; Hansen, Mathiesen, and Skov, 2019; Zappa et al., 2019). These studies suggest the primary use of heat pump technology combined with the smart charging of storage tanks to reduce peak loads. With the increased electrification of the heating sector, the higher electricity demand can be reduced by the use of heat pump technology (Ravestein et al., 2018; Chua, Chou, and Yang, 2010; Kåberger, 2018).

A water heating solution extensively researched in recent times is the combination of heat pump (HP) technology with solar water heating (SWH) (Lerch, Heinz, and Heimrath, 2015; Kong et al., 2018; Vega and Cuevas, 2018). This combination, along with sensible heat storage and auxiliary heating, is becoming increasingly popular in the hot water industry. The challenge remains to size the main components of such a system for maximum cost efficiency, taking into account aspects such as hot water usage variation and the climate.

1.1.3 Combined HP and SWH system design

To size and evaluate the performance of these combined systems, also referred to as hybrid systems, the use of numerical simulations is often employed (Allouhi et al., 2017; Wu et al., 2018; Rey and Zmeureanu, 2016; Li and Kao, 2018; Carbonell et al., 2014). The performance of such systems is dependent on a range of factors, including the choice of sub-components, user profile, irradiance, dry-bulb (DB) temperatures, and wet-bulb (WB) temperatures. These detailed simulations can give an accurate representation of the systems and have aided researchers and designers by increasing the efficiency of these hybrid systems.

There are, however, challenges to develop such a comprehensive simulation model that produces accurate and realistic results promptly. On a sub-component level, the manufacturer information is not always readily obtainable, and a highly detailed system model is computationally expensive. With the lack of manufacturer data, the calibration of these simulations can become a challenge.

In terms of modelling a system over an extended period, the constant changes in input variables such as the hot water demand need to be taken into account. Varying the input parameters leads to numerically changing boundary conditions for a simulation at each time-step. This requires numerical solvers and system formulations with a high order of convergence. The

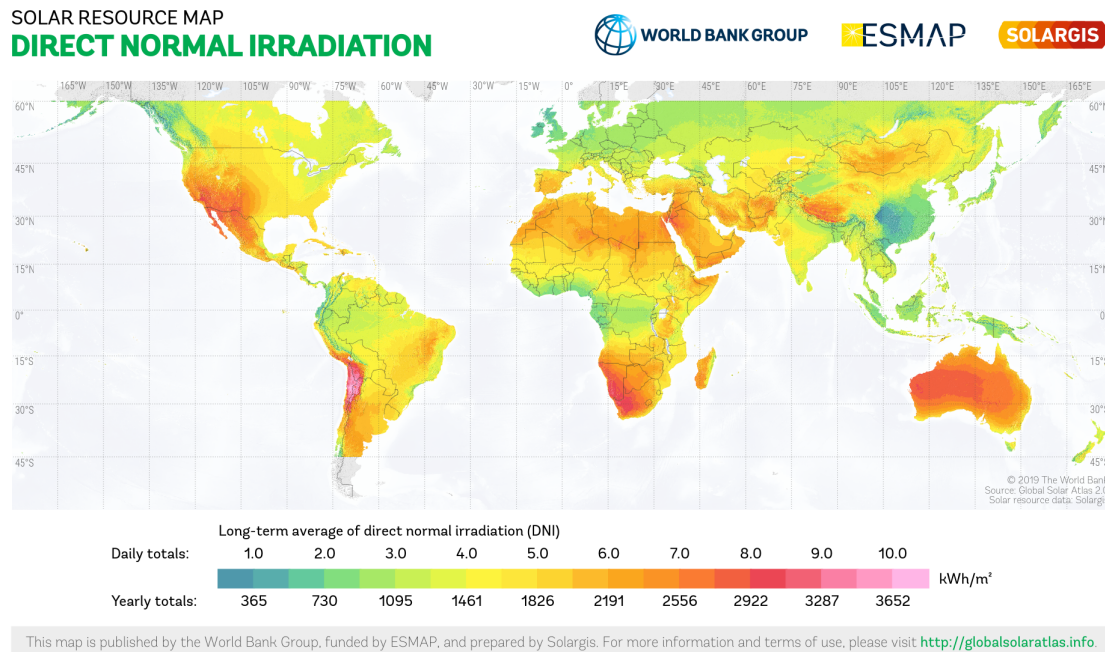


FIGURE 1.1: Global direct normal irradiance (Solargis World Bank Group ESMAP, 2019).

mass, -momentum, and energy equations (MME), as well as the equations of state that need to be solved, leads to high computational cost (Guelpa and Verda, 2019; De Lucia et al., 2017).

Optimisation and parametric studies on solar thermal systems have been successfully undertaken by using stochastic/evolutionary methods (Starke, Cardemil, and Colle, 2018; Araújo and Pereira, 2017; Rey and Zmeureanu, 2018). Recently researchers have started using surrogate models to simplify numerical simulations (Beykal et al., 2018; De Lucia et al., 2017; Tran and Georgakis, 2018; Bhosekar and Ierapetritou, 2018; Guelpa and Verda, 2019). These surrogate models are statistical or black-box methods that reduce the various partial differential equations (PDE) and differential equations (DE) into an elegant pure mathematical representation.

To make the hybrid HP/SWH technology and system integration economically viable, it is of utmost importance to the designer to ensure that an attractive return on investment can be attained.

1.2 Problem Statement

To better understand the synthesis of the various system components of a hybrid HP/SWH installation. There exists a need for a simulation model that uses readily available manufacturer data of solar, heat pump, and storage vessel sub-components for integrated, long-term, transient thermal system simulation. Furthermore, such a validated simulation model needs to be computationally inexpensive. Nevertheless, also simple to calibrate to enable extensive parametric and optimisation studies to be conducted.

1.3 Research Goals

The purpose of this study is to develop a simulation model for larger commercial-scale hybrid hot water systems making use of solar water heaters and heat pumps. The developed simulation model would enable one to run various parametric studies on the system using design year and user profile data in different climatic conditions. Furthermore, the developed simulation would be tested using a case study on a hospital system for the proof of methodology and validation of the developed simulation model.

1.4 Scope of the study

To achieve the research goals set out above, the methodology outlined in the sections below was followed:

1.4.1 Literature review

A comprehensive literature study was undertaken in the heat pump and solar water heater application fields. The literature study was divided into six sections. Each section focused on a distinct aspect of the problem statement. These sections included: heat pump-, solar water heaters-, water storage vessel-, combined SWH-HP simulation models. Finally, methods to reduce the computational effort of these simulations models and the application in optimisation and parametric studies were conducted. A list of concluding remarks on each section was given. Highlighting the essential findings and some deficiencies from literature.

1.4.2 Experimental setup and validation

The hybrid solar-heat pump heated water system of a hospital in Potchefstroom, South-Africa, was analysed. The system is of a parallel configuration with a ring-main feed and bulk water storage vessels. The system was monitored for two months using the systems' SCADA (Supervisory Control and Data Acquisition system) and some additional sensors. The captured data was used to draw up conclusions on the systems' dynamics and design qualities. Consequently, an energy balance was calculated on the various energy fluxes of the system. Validation of the developed simulation model could take place using the energy flux data gathered from this system and served as a case study. The system layout, with the specifications on the main components, was used as input for the developed simulation model.

1.4.3 Numerical modelling and verification

The governing equations for fluid flow were derived based on the finite difference method. Various simplifying assumptions and governing equations were used to establish the fundamentals for the simulation model. The assumptions that were made include the use of only the mass and energy conservation equations. The mathematical approach of the Method of Lines (MOL) was used to simplify the partial differential equations into differential equations. Using Runge-Kutta, the integration of these systems of differential equations could take place. The simulation model was developed in the EES® (Engineering-Equation-Solver) computer package. EES® allows for the quick scripting of mathematical statements and automatically orders the equations to obtain the correct order of calculation. As verification for the developed method, the mathematical concepts of stability, consistency, and convergence was used. Additionally, a model was developed in Matlab® Simulink® to serve as a verification of the

developed simulation model. Simple cases were tested in Simulink® as the benchmark for the RK-MOL (Runge-Kutta Method of Lines) method.

1.4.4 Parametric study and concluding remarks

A parametric study was undertaken on the main components of the case study. This included the capacity of the storage tanks, the rating of the heat pump and finally, the number of solar banks in parallel on the system. From this, a general discussion on the influence of various components on the system performance could be conducted. The case study system was further analysed in various South-African cities. Remarks on the influence of weather conditions were also made on the system performance. The time that it took to run these parametric studies were also noted to determine the effectiveness of the reducing measures on the simulation model.

1.5 Outline of the study

An extensive literature survey is discussed in Chapter 2. Chapter 3 outlines the experimental case study used to validate the simulation model. The available sensors and hospital network configuration are discussed. Chapter 3.4 outlines the methodology for the calculation of the various energy fluxes on the experimental setup. The results that were obtained are outlined near the end of this chapter. Chapter 4 discusses the computer code and simulation structure. Verification of the mathematical model from theory and a commercial code is also presented in this chapter. Chapter 4.4 serves to compare the accuracy of the simulation predictions to that of the experimental results. Chapter 5 utilises the validated simulation model to run various parametric studies for analyses. Chapter 6 concludes with the most relevant findings of the study and the recommendation for further study. Figure 1.2 illustrates the outline of the study graphically.

1.6 Features of this study

To the best of the authors' knowledge from the literature survey, the study contributed in the following ways:

- The use of heat pumps regression models in a validated dynamic SWH/HP simulation.
- A dynamic energy simulation that can be used for optimisation is scarce in the commercial scale hybrid SWH/HP field.
- A parametric study was presented varying the storage vessel capacity, the number of solar banks and heat pump size on an SWH/HP system.
- The influence of weather conditions in various Southern-African cities on SWH/HP system performance is studied.
- Aspects of the study were presented at the 6th SA-UNESCO Engineering, Science and Technology Conference held at Mahikeng on the 25th of September 2019.

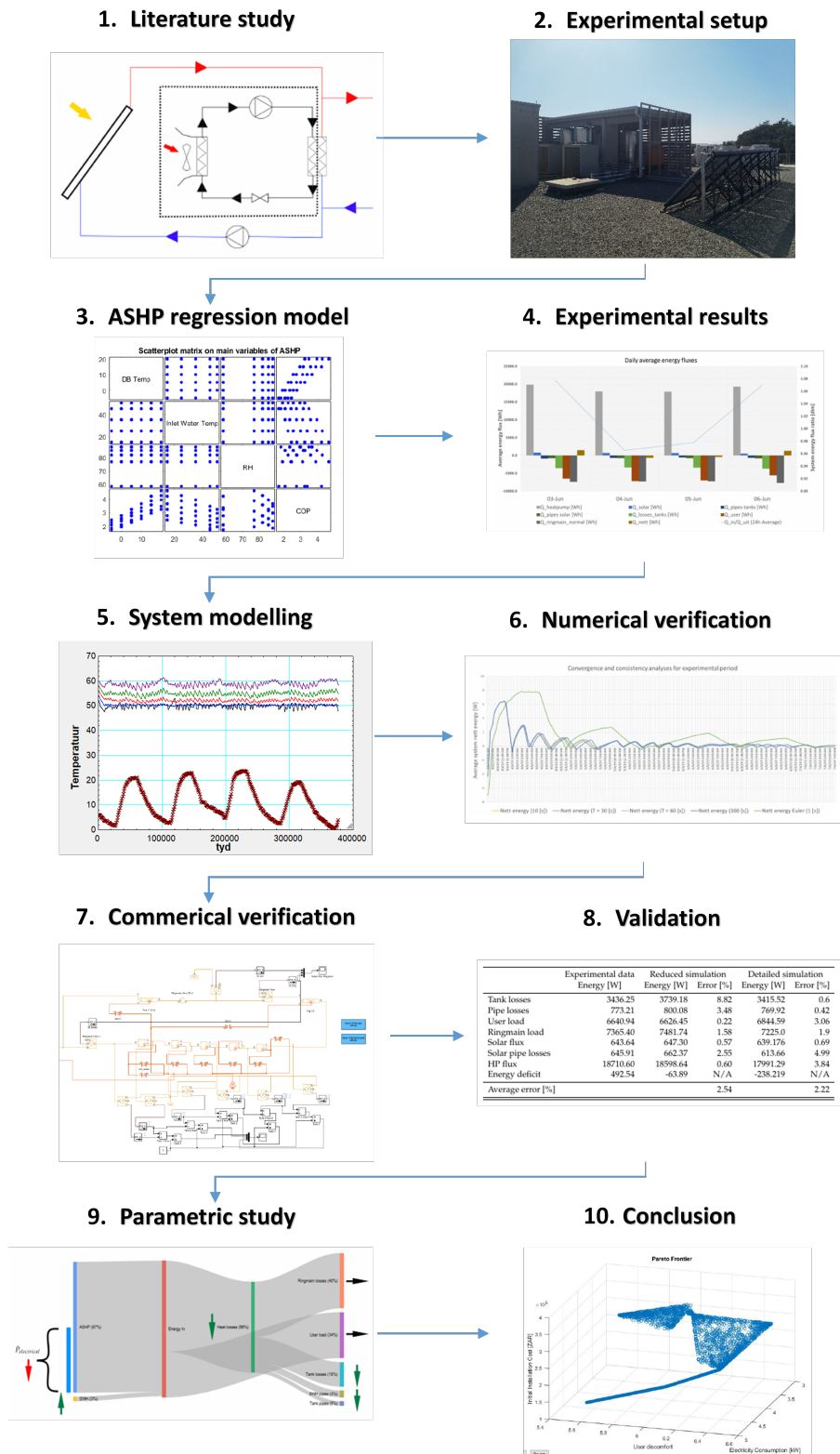


FIGURE 1.2: 1. Literature study (Chapter 2) 2. Experimental setup (Chapter 3) 3. ASHP regression model (Chapter 3.5) 4. Experimental results (Chapter 3.6.2) 5. System modelling (Chapter 4) 6. Numerical verification (Chapter 4.3) 7. Commercial verification (Chapter 4.3.2) 8. Validation (Chapter 4.4) 9. Parametric study (Chapter 5) 10. Conclusion (Chapter 6)

Chapter 2

Literature Survey

2.1 Introduction

In the previous section, the need to create a simplified but accurate modelling application for use in commercial water heater installations was addressed. To fully appreciate the scope of the problem and to build a workable solution, an extensive literature study was undertaken.

The literature study has the following sections:

- Combined solar water heater and heat pump applications and the modelling thereof
- Water storage vessel modelling
- Heat pump modelling
- Solar water heater modelling
- Reduced-order models of thermal systems
- Numerical methods of thermal-fluid problems
- Parametric studies of thermal systems

A list of concluding remarks and a summary will be given at the end of each of these sections. At the end of this chapter, a table will follow that summarises key concepts from literature and those used in this study Table 2.2.

2.2 Combined solar water heater and heat pump applications and the modelling thereof

The following section serves as an introduction to some recent research on combined solar water heater and heat pump applications.

Lerch et al. (Lerch, Heinz, and Heimrath, 2015) investigated two different configurations of combining solar and heat pump systems. The first system, the air source heat pump, is used as a benchmark and is shown in Figure 2.1 below. In Figure 2.2, solar energy is used in parallel, and in Figure 2.3, solar energy is used directly in the evaporator. Figure 2.4 illustrates a system where the HP can source energy from both ambient conditions or solar energy. This system can also operate the solar energy source in parallel to the ASHP. The latter system would be more appropriate where low ambient conditions adversely affect the coefficient of performance

(COP) of the heat pump. The boundary conditions were the same for the various configurations under consideration. It was found that parallel air source heat pump (HP) and solar energy systems can improve the performance and heating capacity compared to conventional air-source HP systems by 43 [%]. The use of solar energy directly on the evaporator of the HP had a marginal increase in system performance. Even with the added costs, conventional solar energy and heat pump technology is sufficient for the supply of adequate heating. The use of a combination of lumped and distributed models to simulate the various components is a useful alternative to solve heat-pump systems effectively and was employed in this study.

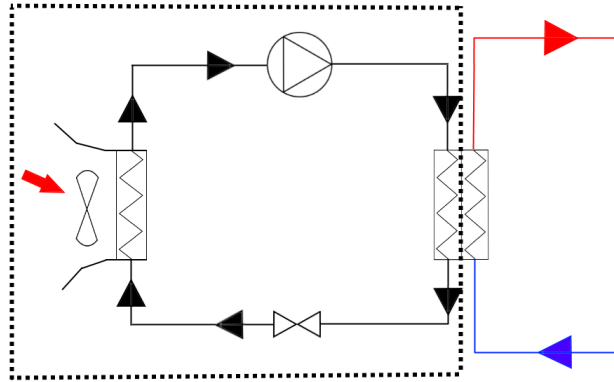


FIGURE 2.1: Air source heat pump configuration (ASHP).

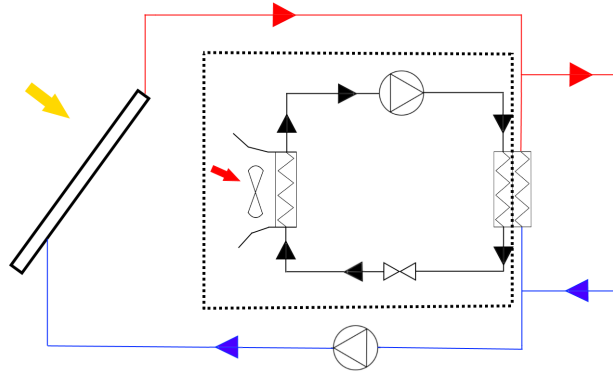


FIGURE 2.2: Solar-assisted air source heat pump configuration (SA-ASHP).

Recently a trend has emerged to investigate the use of various low-temperature heat sources for heat pump technology. The study of Deng and Yu (Deng and Yu, 2016) simulated the concept of utilising a dual-source air and solar heat pump against a conventional solar direct expansion heat pump. A controller with electric valves distributes the refrigerant flow between the solar and air sources depending on environmental factors. As a simulation-based study, the code was written in Fortran, and a time step of 150 seconds was used. The solar component made use of the Hottel and Whiller equation (Duffie and Beckman, 2013). The effectiveness-NTU method, along with a two-phase heat transfer coefficient, was used for the finned-tube evaporator. For the compressor, a semi-empirical method based on manufacturer data was used. The condenser and water tank were modelled by an energy method. Thermal capacitance was neglected for all components except the unstratified water tank. The COP of the hybrid system

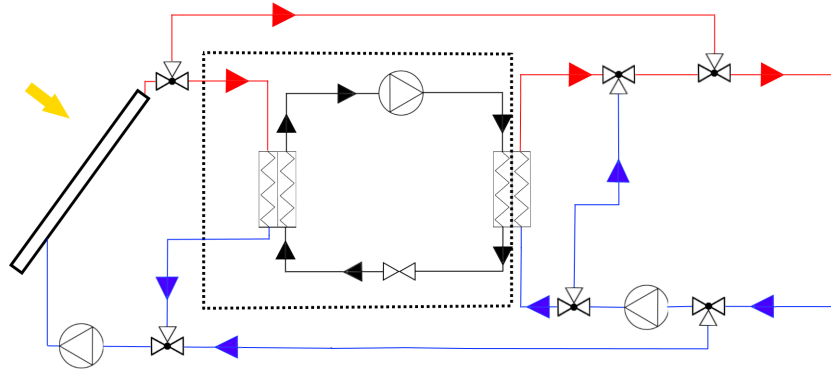


FIGURE 2.3: Solar-assisted parallel water source heat pump configuration (SAP-WSHP).

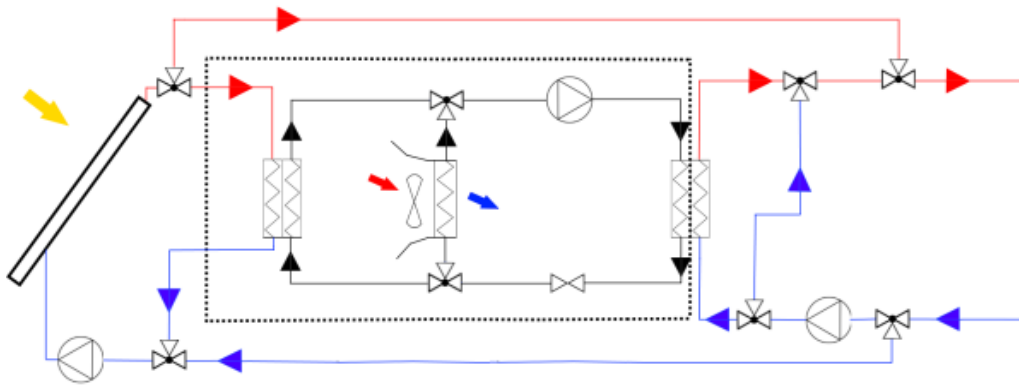


FIGURE 2.4: Solar-assisted dual-source air-water heat pump configuration (SA-DSHP).

increases by 19.8[%] on average to that of the solar sourced system. Based on the theoretical results, the hybrid sourced system has superior performance at low solar radiation to that of conventional sources.

Vega and Cuevas (Vega and Cuevas, 2018) conducted a simulation study on the direct heated water (DHW) and heating needs of a residential building in Concepción, Chile. TRNSYS 17 was utilised to study the application of parallel solar (SWH) and heat pump technology (ASHP) on a residential building. The thermal inertia of the walls and the various apartments were taken into account. The ASHP-SWH sub-simulation used a quadratic efficiency curve for the SWH and performance catalogue table for the ASHP component, respectively. However, due to numerical convergence problems, the whole building simulation was split into sub-simulations. Three simplified and one detailed simulation were coupled with the ASHP-SWH sub-simulation. One hour time-steps was used to study the entire building. Hereafter, a detailed simulation study was conducted for each floor. A final simulation using a five-minute time-step was used for the simulation of components with quick response to the system like the SWH. The SWH used variable speed drives (VSD) to supply the correct mass flow for optimal efficiency. These simulations were run for the winter and summer period, and the systems

energy usage was analysed. It was shown that the solar-assisted systems had a seasonal performance factor of 4.61 ($SPF_{system} = 4.61$) for flat plate collectors and a SPF_{system} of 4.86 with excavated tube collectors. It was found that the performance was mainly dependent on the demand profile of the individual apartments.

Bastos et al. (Bastos, Torres, and Castilla Álvarez, 2018) developed a simulation for the modelling of heat pumps with solar assistance and validated its results against experimental data. The study used lumped semi-empirical (mass and energy equations with manufacturer data) models for the expansion valve and compressor. The evaporator and condenser were modelled using mass, momentum and energy equations. The solution procedure started by choosing a random discharge pressure on the compressor. Hereafter, the mass flow was calculated throughout the system. If the mass flow does not converge, then an alternative pressure is specified using the Newton-Raphson method aided by the secant method. The Runge-Kutta method was used to solve the enthalpy, pressure and mass flow rate throughout the system. If the energy balance condition was met, then the simulation terminated. Matlab was used for the solving of the equations. The study further observed an oscillatory use of power by the compressor as the environmental conditions changed ever so slightly.

Kim et al. aimed to investigate the concept of a hybrid solar collector which can supply energy either to a water-to-water heat pump or directly to the storage tanks (Kim et al., 2018). A simple thermodynamic model was showcased and experimentally validated with the prototype. A maximum error of 8.8 [%] was attained between the predicted model results and experimental data. The concept was compared to conventional serial and parallel type solar assisted heat pump technologies. For the SWH component, the log-mean-temperature-difference (LMTD) method and Hottel and Whiller equations were used. For the ASHP, a bi-quadratic regression model was employed. The energy components of the various pumps and components were taken into account. The parallel configuration offered the best seasonal performance with compressor heating capacity (Q_c) at 9.91 [kW] and seasonal COP at 4.56. The serial and proposed hybrid system offered a (Q_c) of 2.7 [kW] and 7.82 [kW], respectively. The former and latter's annual COP values amounted to 2.5 and 3.67, respectively.

The study of Rekstad et al. conducted experimental energy analyses on two detached passive houses in Norway (Rekstad et al., 2015). The houses were equally sized, of the same design and at the same location. One house uses solar thermal heating and the other air-source heat pump technology for space and hot water heating. The houses were extensively monitored with high-resolution interval measurement and various sensors. The raw data were used, along with assumptions and calculations, to determine the various energy source fluxes and loads. The results indicated that the house that is supplied by the solar thermal technology uses 15-20 [%] less electricity than the house supplied by heat pump technology. These results indicated that solar thermal heating could be competitive in the harsh Nordic climate, even at those high latitudes.

2.2.1 Concluding remarks - Solar and ASHP combinations

Recent literature has numerous simulation studies focusing on the combination of solar and heat pump technology. It was evident from the literature that a comparison between experimental results and simulation programs are of standard practice. The comparison with large non-residential applications is, however, scarce. The following aspects from the literature that will be incorporated in the development of the solution are highlighted below.

Similar to the studies of Bastos, Terres and Castilla Alvarez, this study will aim to implement a lumped and distributed component approach on the reduced simulation program (Bastos, Torres, and Castilla Álvarez, 2018). This study will attempt to neglect (in the reduced computer program) the thermal capacitance of most of the components except the water storage vessel and SWH as with Deng and Yu (Deng and Yu, 2016).

The energy analyses to be conducted on the case study shares aspects in approach to the study of Rekstad et al. (Rekstad et al., 2015). The reduced simulation program that will be developed conducts an annual energy analysis with a similar time-step to the study of Guo and Goumba (Guo and Goumba, 2018). The study will aim to use regression to model the heat pump as with the study of Kim et al. (Kim et al., 2018).

From this section, the following conclusions can be made from the literature:

- In most of the studies, the ASHP was modelled by a regression model based on manufacturer catalogue data.
- Most of the studies used manufacturer solar efficiency curves to model the SWH component.
- The integration of solar energy with heat pump technology leads to higher seasonal system efficiency.
- Experimentally validated apartment-scale combined SWH-ASHP models seems to be scarce.
- As novel configurations are proposed with the integration of solar and heat pump technology; new simulation methods and control methods are required as these systems are becoming increasingly complicated.
- Smart control of these systems will lead to higher performance but will require validated simulations to test these control schemes.

2.3 Component modelling

With the combined simulations discussed, the following section will investigate studies that focus on component-based modelling of sensible water storage vessels, air-source heat pumps (ASHP) and solar water heaters (SWH). Useful features and ideas were listed from literature in the concluding remarks of each section.

2.3.1 Water storage vessel modelling

Inaccurate stratified tank models can be degrading towards energy predictions.

2.3.1.1 1D and 2D modelling approaches

Although an earlier study, the work by Kleinbach showcased a detailed approach to the various simulation techniques for stratification in solar water heating systems (Kleinbach, 1990). The study found that temperature stratification in domestic solar water heaters storage tanks have a direct impact on the performance of the system. The plugged flow and multi-node 1D tank models that existed in TRNSYS was compared and analysed. The field of applicability of these methods was determined. The use of a uniform tank temperature leads to an under prediction of useful energy and deliverable energy for all experimental measurements. Furthermore, it

was not recommended to use plume entrainment with multi-node models. The reasons being that the model predictions are not improved and were computationally expensive.

Kamal et al. did a comparative study where they compared experimental results to a 2D pure conduction model and a fully coupled 2D mass-momentum and energy system of equations (Ismail, Leal, and Zanardi, 1997). The pure conductive model neglected natural convection but only took into account the forced axial convection ($Re = 350$). The pure conductive model exclusively showed good agreement with the experimental data during the first few instances of charging/discharging. As soon as convection currents originate (20 min), the predictions between the coupled and uncoupled models diverge as time advances. The coupled moment and energy equation model is adequate to describe the thermocline in the tank model. It is of note that the uncoupled conduction model had a run-time of 5 minutes with the fully coupled model at 200 minutes, respectively.

Han et al. did a review on various mathematical models for thermal tank stratification (Han, Wang, and Dai, 2009). Researchers have often implemented 1D models, and the addition of empirical-based mixing parameters have enhanced their usability and accuracy. These methods are, however, not that accurate amidst problem that include: intricate tank geometries, backflow due to mixing and problems where the flow rate is too high. Numerous 2D model has been developed using the Navier-Stokes equations with the Boussinesq approximation to describe the mixing due to buoyancy and convective mixing. These methods tend to describe the mixing between layers more accurately. Various studies on 2D Navier-Stokes based solvers found that it is safe to model these water systems as incompressible as the viscous dissipation is negligible. The latest 3D CFD simulations have the highest accuracy and allow one to model complex fluid problems. These methods tend to focus on the optimisation of the tank geometry and the study of fluid properties. These methods, however, have longer simulation times in comparison to 1D and 2D simulation models. Depending on the application at hand, these solvers have their respective strength and weaknesses.

2.3.1.2 Computational fluid dynamic correlations

Fan et al. studied the influence of natural convection on thermal stratification and heat loss (Fan and Furbo, 2012). The intention was to establish a simple equation that correlates the heat loss between the various layers of the storage tank. A simple regression equation from CFD results was obtained that takes into account: the influence of tank volume, height to diameter ratio, tank insulation and the initial conditions of the tank. A few noteworthy findings were made: Heat loss to the sides of the tank helps build up thermal stratification in the tank. Temperature gradients oppose buoyancy-driven flow. A reduction in heat loss of 20-55 [%] was observed if the temperature gradient was between $0.2 - 1.5 [W K^{-1}]$.

Beaten et al. identified a need for an accurate low order 1D storage tank model that takes mixing and buoyancy effects into account (Baeten et al., 2016). One dimensional storage tank models are utilised regularly in the building industry. The problem with these models is that water mixing and buoyancy effects are three-dimensional. The study conducted various CFD simulations on storage tanks and captured stratification correlation parameters. Consequently, the 1D model made use of these correlation parameters. Thus, the accuracy of CFD buoyancy modelling was obtained at a drastically reduced computational effort.

2.3.1.3 Computational fluid dynamic modelling

Levers and Lin ran multiple CFD simulations to investigate the influences of operating conditions, inlet geometry and aspect ratio on the thermal stratification of storage tanks (Levers and Lin, 2009). Adiabatic tank insulation was assumed. The simulations revealed that an aspect ratio increase from 2.5 to 5 could lead to a 30.69 [%] improvement in thermal stratification. When the mass flow was increased from 0.05 to 0.15 [kg s⁻¹], the stratification contracted by 77 [%]. Improper placement of the inlet and outlet relative to the tank's top and bottom can decrease stratification by 72 [%]. These results indicate that to lessen mixing, the hot water inlet should always be at the top.

A CFD study was conducted by Kursun and Okten to analyse the effect of aspect ratio and tank position on thermal stratification in a rectangular tank (Kursun and Ökten, 2018). The storage tank inclination was varied from 0 to 60 [°], and aspect ratios from 0.5 to 1. The resulting stratification and second law efficiency were determined. For a vertically positioned tank, by reducing the aspect ratio from 1 to 0.5, the stratification was improved by 14 [%]. The thermal stratification increased with time for all aspect ratios and angles. For vertical tanks, the aspect ratio has a noticeable effect on thermal stratification.

2.3.1.4 Concluding remarks - Water storage vessel modelling

For the water tank components, various techniques were investigated. Ranging from the use and development of empirical (CFD) values in 1D models (Fan and Furbo, 2012; Baeten et al., 2016), to various 1D and 2D models (Kleinbach, 1990; Ismail, Leal, and Zanardi, 1997; Han, Wang, and Dai, 2009). The use of CFD in the analyses of storage tanks in the solar application field was also investigated (Levers and Lin, 2009; Fan and Furbo, 2012; Kursun and Ökten, 2018). The goal is to aid in the design of solar heat pump hybrid systems, not components. CFD and 2D models offer the most comprehensive results but are mostly used to design and study effects on component level. Especially on storage vessels and their respective inlet manifolds. In the case study, the impinging jet at the inlet of the tanks is of a low enough velocity to not disturb the stratification significantly. Stratification is enhanced with the pipe connections between the five discrete storage vessels. The above reduces the need for a 2D or CFD model. To model, the various layers of the tanks will take up too many computational resources. Therefore, from the rationale above a 1D heat balance model seems like the most likely candidate for implementation.

From this section, the following conclusions can be made from the literature:

- The inclusion of buoyancy and stratification in simulations are essential for the accurate performance modelling of solar water heating systems.
- The use of 1D multi-node models (heat balance models) is more suitable for scenarios where there is more than one inlet.
- 1D multi-node models showcase a reduction in efficiency if the plume of the incoming fluid is modelled explicitly.
- 1D models that only take conduction and axial convection into account are not as accurate when the buoyancy-driven flow is dominant in the simulation of storage tank stratification.
- Viscous dissipation is negligible in the modelling of hot water storage tanks during the use of 2D-models.

- Empirical based correlations can improve 1D and 2D simulations in the prediction of flow in complex tank geometries and high mass flow scenarios.
- CFD can model complex 3D transient flow tanks the best but do not lend themselves for long-term energy simulations due to their high computational cost.
- CFD simulations tend to be used often for the design and optimisation of storage tanks and not holistic system simulation.

2.3.2 Heat pump modelling

Guo and Goumba investigated the effect of individual (single apartment) and bulk storage strategies (apartment block) on the energy efficiency of DHW supply (Guo and Goumba, 2018). From TRNSYS simulations, it was determined that individual storage units have a combined higher efficiency than the collective bulk storage mediums. The ASHP was modelled from catalogue data and not a full physics model. For the circulating losses, the 'dead-end' legs of the centralised system were also taken into account. Stratified simulations modelled the storage vessels. The time step was taken as ten minutes, and a whole year was simulated. Although ASHP is more efficient with bulk storage, the pumping and heat loss associated with bulk storage make this configuration less efficient than individual storage units.

Manufacturers of ground-source heat pumps (GSHP) usually express the power consumed to operating conditions for some discrete values. Dynamic simulations often require values other than those that are listed. The study of Reddy et al. intended to use a simplified methodology where multiple regression models are employed to predict the performance of GSHP under a broad range of operating conditions (Simon et al., 2016). The study was able to use second-order multiple regression (MR) models with eight variables to predict the performance of the ground source heat pump. The model correlated COP source-, load flow rate, and the inlet temperatures of the condenser and evaporator. For the different heat pump models, COP correlation errors were 2.62 [%], 4.91 [%] and 3.21 [%], respectively. It was noted that proprietary regression models might have generated the manufacturer data of the studied GSHP. The study suggested that as the quality of these models cannot be verified, one should use these models cautiously.

Underwood, Royapoor and Sturm proposed a heat pump model which could be parametrised from either experimental-, manufacturer- or field data (Underwood, Royapoor, and Sturm, 2017). The model utilises a refrigerant-based regression model rather than the more common system side approach. In the past, studies have often used deterministic methods to model heat pump behaviour (Van Eldik, 1998; Kinab et al., 2010). These models required comprehensive knowledge and detailed information to deem them useful. The study aimed to develop a model of intermediate complexity that can be fitted reliably from limited field data. A non-linear regression model with ten input variables was used. A multi-objective optimisation routine in MATLAB was used to adjust the coefficients so that the root-mean-squared (RMS) error between the measured and predicted values were minimised. It was observed, as with the study by Reddy et al. that manufacturers usually over-predict the performance of their heat pumps (Simon et al., 2016). Predicted power consumption and heat output delivered an RMS error below 10 [%] for the mean test data values. This study differs from the former in that a grey box model was proposed rather than a purely statistical model.

Tabatabaei, Treur and Waumans; compared computational models using climate conditions as input to predict ASHP performance (Tabatabaei, Treur, and Waumans, 2016). Linear, quadratic,

exponential and higher-order models were tested and tuned to match the experimental data. The fourth-degree polynomial was the most accurate, followed by the cubic model in the prediction of the heat pump performance. The regression models were tested for the climatic conditions of eight different cities. It was shown that a model could approximate seasonal heat pump performance reasonably by only using dry bulb temperatures. Higher-order models worked for specific circumstances but did not in all of the cases. The latter suggests regression over-fitting. The study highlighted the point that the use of simple equations could be used to define annual heat pump performance.

2.3.2.1 Concluding remarks - HP modelling

For the ASHP component, a statistical regression strategy as used by the studies of (Simon et al., 2016; Underwood, Royapoor, and Sturm, 2017; Shafieian, Khiadani, and Nosrati, 2019; Guo and Goumba, 2018) appears to offer a good compromise between accuracy and computational resources. As noted from Chapter 1, the aim is not to design the ASHP and therefore, component-based techniques will not be used. Although the studies of (Simon et al., 2016; Underwood, Royapoor, and Sturm, 2017) opted for component-based inputs; this study will opt for system side regression. This information would be more accessible during the experimental setup and from the manufacturer catalogue data.

From this section, the following conclusions can be made from the literature:

- In recent times, there has been an initiative from academia to investigate the use of regression methods to characterise the performance of heat pumps.
- Manufacturers tend to over-predict the performance of their heat pumps.
- From the literature, the use of these regression models in a validated dynamic energy simulation is rare.
- Simple regression models can accurately predict heat pump performance.
- The use of higher-order models tends to over-fit the regression data.

2.3.3 Solar water heater modelling

Allouhi et al. conducted a simulation and optimisation study in TRANSOL on the supply of hot water to four different industrial processes (Allouhi et al., 2017). The financial feasibility of solar energy supplying a portion of heating was investigated by means of energy analysis. The case study was implemented on a milk processing company situated in Morocco. The three main parameters that were altered was the volume of the storage tanks, the aperture area and the tilt angle of the solar system. For the storage tank, a stratified model was used. For the SWH, a modified polynomial efficiency curve sensitive to tilt angle was used. The proposed system had a solar fraction of 41 [%] with an annual reduction of 77 tons in CO₂ emissions. The latter also had a life cycle savings of 179 [kUSD] over a period of 20 years. The study showcases the fact that solar energy can lead to substantial savings in countries with an abundance of solar energy and the need for heat in industrial processes. The study was also unique as it optimises a design that supplies four different setpoint temperatures.

The study of Shafieian, Khiadani and Nosrati, evaluated the performance of an evacuated solar tube system experimentally and utilising simulations (Shafieian, Khiadani, and Nosrati, 2019). The solar system was tested to meet the hot water consumption pattern of a real-world residence under non-ideal climatic conditions in Pert, Australia. The study presented a validated

mathematical model with a maximum absolute error of 5.6 [%] to that of the experimental data. The simulation was based on a complete heat pipe thermal resistance circuit based on the condenser and evaporator of the evacuated solar tube. The second part was based on the effectiveness-NTU method to determine the thermal properties at the outlet of each evacuated tube. Combined with the validated method, a parametric study on the number of tubes and collector mass flow rate was conducted. A regression equation and efficiency graph for the collector was derived from experimental data. The study concluded that mass flow had a direct impact on supply temperature and collector efficiency.

Martínez-Rodríguez, Fuentes-silva and Picó; developed a design approach for solar thermal energy to low energy-intensive applications (Martínez-Rodríguez, Fuentes-Silva, and Picón-Núñez, 2018). The developed thermal model was compared to experimental results. A maximum deviation of 8 [%] was reported between the model and experiments. Furthermore, the mathematical coupling between thermal and hydraulic equations was used. The solar thermal equation resembles that of Hottel and Whiller (Duffie and Beckman, 2013). The complete thermal circuit, as with the previous study, was replaced with a single resistance variable. The former allowed the authors to run a parametric study and to change the number of collectors in series and parallel and other environmental conditions. Environmental conditions heavily influence the number of solar collectors required to supply the targeted thermal behaviour. The study also developed graphical parametric plots to compare the main design parameters.

The study of Daghigh and Shafieian experimentally and theoretically determined the performance of an SWH system using evacuated solar tubes (Daghigh and Shafieian, 2016). An SWH was designed and constructed. The system was tested under the conditions of Sanandaj city where the hot water consumption pattern of a dormitory hall was simulated. The study attempted to ascertain the effects of various parameters, such as the number of heat pipes and mass flow, on the efficiency of the SWH. The developed model had a maximum error of 8.4 [%] to that of the experimental results. A thermal circuit similar to the study of Shafieian, Khiadani and Nosrati were employed (Shafieian, Khiadani, and Nosrati, 2019). The effectiveness-NTU method was used, and it was found that the impact of collector mass flow rate was low during the beginning and end of each day. The user profile had a significant effect on the thermal efficiency of the solar water heating system.

The design of solar banks for thermal processes require that hydraulic and thermal requirements have to be met. The reasons being that for non-residential applications, the working fluid through the collectors have to be forced using pumps. The latter is due to the significant pressure drop through these solar collectors. The study of Picón-Núñez, Martínez-Rodríguez and Fuentes-silva; developed a simulation with industrial applications in mind (Picón-Núñez, Martínez-Rodríguez, and Fuentes-Silva, 2014). The simulation had a maximum error of 1.35 [%]. Thermal and hydraulic coupling was used. The resistance circuit, however, focused on the header of the SWH circuit and not the entire collector tube. It was observed that the number of collectors in series determines the maximum temperature. In contrast, the number in parallel determines the mass flow and the heat load that can be supplied.

2.3.3.1 Concluding remarks - SWH models

For the SWH component, the study will aim to take the thermal capacitance of the SWH into account due to variable flow. A steady-state thermal circuit and effectiveness-NTU method have been widely implemented in domestic SWH systems (Shafieian, Khiadani, and Nosrati,

2019; Daghighi and Shafieian, 2016). Additionally, these studies used a steady-state continuous mass flow through the SWH manifold. Finally, the studies neglected the heat losses and thermal capacitance of the manifold. The reduced and detailed modelling programs that will be developed for this study will make use of an experimental/manufacturer graph. Additionally, the thermal mass of the SWH and heat loss of the manifold will be taken into account. The rationale is followed as the mass flows varies according to a controller (*on-off* control), and the flow through the SWH is on occasion zero. Furthermore, the SWH manufacturer graph will be used as boundary input to the manifold control volume as the effectiveness-NTU method from literature requires extensive calibration and details on the heat pipes. The detailed model will additionally strive to take the hydraulic losses into account, similar to the studies of Martínez-Rodríguez and Picón-Núñez (Martínez-Rodríguez, Fuentes-Silva, and Picón-Núñez, 2018; Picón-Núñez, Martínez-Rodríguez, and Fuentes-Silva, 2014).

From this section, the following conclusions can be made from the literature:

- User consumption and weather patterns should be taken into account when hot potable water systems are designed.
- The mass flow rate heavily influences solar water heater performance.
- For the effective use of solar energy, the network configuration should be taken into account to meet the hydraulic and thermal requirements.
- Researchers often use regression from experimental results to derive an equation for SWH efficiency.
- Most studies did not take thermal mass and heat loss of the manifold into account.
- Some studies make use of a steady-state thermal model and the effectiveness-NTU method to conduct parametric studies.
- The Hottel-Whiller equation is common in the application of header focused simulations.

2.4 Simulation structure

The following section focuses on the techniques for the simulation of thermo-fluid problems from the literature. Reducing simulation complexity is showcased as well as the numerical methods employed in these problems. The section concludes with the use of these developed computer programs in parametric studies and the usefulness thereof.

2.4.1 Reduced-order thermal-fluid models

Recently researchers have started applying surrogate models to imitate the dynamics of advanced simulations and experiments (Bhosekar and Ierapetritou, 2018; San and Maulik, 2018; Chang and Dinh, 2019; Rahman and Smith, 2018; Tran and Georgakis, 2018). A surrogate model in the thermo-modelling domain uses simplifying methods to mimic the complex characteristics of comprehensive physics-driven simulations. Table 2.1 contains a simple example of the methodology in constructing a surrogate model.

Bornatico et al. presented a study on various grid structures for surrogate models in the building field (Bornatico et al., 2013). The focus was on the effective distribution of nodes and the use of interpolation methods to build the surrogate models. The nodes represent the simulation output, and the grid is the domain under which the parameters are varied. Polysun as

simulation program was used to determine the outputs on the grid. The problem was to create a high dimensional grid to capture the system dynamics but not statistically over-fit the model. For the 12-D case, a model with a relative error of less than 5 [%] was presented. The Poisson disk offers the smallest interpolation error for a low number of nodes compared to the cartesian and hexagonal (uniform) distributions. The developed surrogate and grid converge 150 times faster to optimum solutions than conventional techniques.

Guelpa and Verda developed a reduced-order physical model for the quick simulation of large thermal networks (Guelpa and Verda, 2019). The case study was the Turin network in Italy (56 million [m³]). The number of iterative calculations was reduced by simplifying the mathematical coupling of the momentum equation with that of the energy and mass equations. The study used a combination of black-box regression methods and physical equations to model the system. An equivalent tree network was constructed, eliminating the various looped networks. Linear regression captures the mass flow dynamics of the loops of the original network. These regression coefficients can be obtained from measurement or full physical simulation. These regression models impose boundary conditions on the tree-network. The continuity of mass is then solved over the entire network, followed by the continuity of energy equations. The methods outlined in the study reduced the computational cost of simulating this network by two orders of magnitude. The error between the reduced-order and full physical simulation was under 5 [%] for all scenarios.

The study of Edwards et al. studied the use of surrogate models in generating comparable outcomes to that of the US department of energy's building simulation program, EnergyPlus (Edwards et al., 2017). The program is used for the retrofit of existing buildings, design of new energy-efficient buildings, development of building codes as well as energy efficiency strategies. The study used Lasso regression and feed-forward neural networks (FFNN) as surrogate models to improve the simulation time. The study used 5 million simulations for the input parameter space of the surrogate models. The most important inputs for the building simulation amounted to 156 input variables and 90 output variables obtained from the 8 million simulations—the former amounts to 5×10^{52} simulations on the defined distribution grid. The latter highlights the computational infeasibility and the need for lightweight simulations. The developed surrogate models were able to calculate most of the essential outputs defined by domain experts within a 5 [%] accuracy. The surrogate models were achieving this level of accuracy within 3 seconds where the equivalent EnergyPlus simulation amounted to 5 [min] of computational run time. Whole building energy consumption could be predicted within 0.07 [%] accuracy. The techniques and methods showcased in this study illustrate the trade-off between simulation time and accuracy.

2.4.1.1 Concluding remarks - Reduced thermal-fluid modelling

Guelpa and Verda were able to simplify the mathematical coupling between the momentum equations and that of the energy and mass equations on a sizeable thermal network (Guelpa and Verda, 2019). This study will apply a comparative approach by attempting to enforce known mass flows in individual branches depending on the boundary conditions. The continuity of energy and mass equations will be calculated accordingly; on the rest of the system. The studies of Bonaticco et al. and Bhosekar and Ierapetritou demonstrated the use of various regression methods to characterise a system (Bhosekar and Ierapetritou, 2018; Bonaticco et al., 2013). This study will attempt to use manufacturer data for subcomponents. The former entails the use of multivariate linear regression and polynomial regression to capture the ASHP and SWH performance, respectively. The study of Edwards et al. serves as an illustration in the

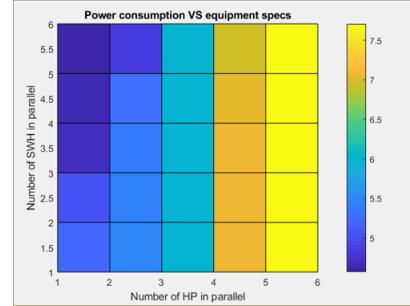
use of regressional methods derived from detailed simulations and the computational benefit thereof.

From this section, the following conclusions can be made from the literature:

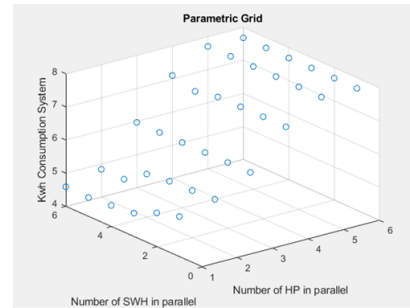
- In specific thermal networks, the momentum equation can be simplified to decrease the computational cost without sacrificing accuracy.
- Local regression is beneficial if a high dimensional grid is to be fitted.
- Regression and other surrogate modelling techniques offer the designers of complex systems to run conventional computationally expensive simulations in drastically lessened time-frames.
- Black and grey box methods on the interfacing between components can lead to a decrease in computational cost.

TABLE 2.1: Basic methodology in constructing reduced models (surrogates) of complex systems

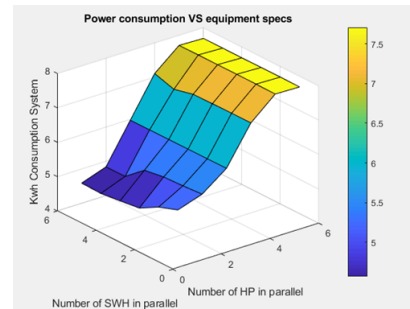
- Determine independent system variables.
- Determine the practical domain of the variables.
- Determine a parametric grid/mesh that encompasses the domain of all of the independent variables of the system.
- The example to the right shows a cartesian grid with two independent variables that are varied.



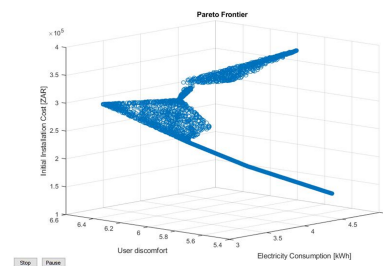
- List the dependant variables in the form of key performance indicators (KPI's) or system outputs.
- Calculate these outputs at the grid/mesh points determined in the first step using experiments or physics-driven simulations.
- Conduct a preliminary model analysis between dependant and independent variables.
- The example to the right illustrates electricity consumption to two independent variables.



- Choose a suitable regression model or surrogate to fit through the system outputs.
- Verify the accuracy of the surrogate by comparing its output to the source (simulation or experiment) at a point other than the grid/mesh nodes.
- The example to the right is a response surface (RS) fitted through the independent variables.
- This concludes the methodology to construct a surrogate model.



- The surrogate derived in the previous step can now be used to imitate the physics-driven simulation at a reduced computational cost.
- The surrogate/regression model can be used for multi-objective optimisation problems.
- The example to the right shows a typical Pareto frontier of a three-dimensional cost function.



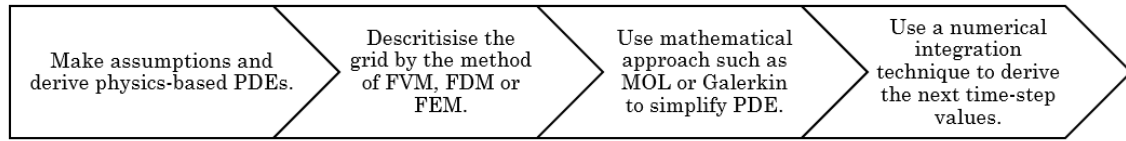


FIGURE 2.5: The mathematical approach to producing a numerical time marching algorithm.

2.4.2 Numerical methods of thermal-fluid problems

Figure 2.5 illustrates the mathematical approach required to develop a time marching simulation. The following section briefly discusses each of these aspects.

Kriel developed a 1D unified total pressure formulation for pipe networks that could be used for both compressible and incompressible flow regimes (Kriel, 2012). This unified pressure formulation allows computer codes to use one set of equations for all flow problems. The numerical method used is of the IMplicit-EXplicit (IMEX) type numerical solver (Chattot and Hafez, 2015). The study uses a weight factor (α -method) to allow the developed code to shift between fully explicit ($\alpha = 0$) and fully implicit ($\alpha = 1$) numerical solutions. An adaptive method was developed that analyses the flow and chooses the corresponding weight factor to suit the prevailing flow conditions. Finally, the study used adaptive time-stepping to improve computational efficiency.

Botha implemented an explicit method for the simulation of transient flow in compressible pipe networks (Botha, 2003). The code was used as verification for the then recently developed Flownet used by the PBMR company (Pebble Bed Modular Reactor). The conservation equations were solved by the two-step Lax-Wendroff method (Tu, Yeoh, and Liu, 2008; Leveque, 1999). Total and static pressure formulations were used to create an elementary pipe network solver in C++. Specific components such as that of the turbines and orifices were modelled as non-inertial components. The numerical solution was compared to Flownet and RELAP5. Various slow and fast transients scenarios were used as case studies. In all of the studies, Lax-Wendroff method compared well with that of Flownet. Anomalies were detected with the isothermal flow at high Mach numbers. Notable differences were also detected at boundary conditions with fast pressure transients and no-flow conditions. With the end application in mind, the simulation compared adequately.

De Villiers derived a 1D network code for the simulation of the main arteries of the cardiovascular system (De Villiers, 2012). The derived simulation discretisation is of finite volume method on a staggered grid. The conservation equations were derived from the Reynolds Transport theorem. The momentum equation is rewritten in the total pressure formulation similar as also used in the studies of (Fick, 2014; Botha, 2003; Kriel, 2012). Although the blood can be assumed to be incompressible, the change in arterial wall geometry leads to a somewhat compressible phenomenon. The former is added as an external pressure term (empirical based), and this consequently leads to conservation in mass, momentum and arterial wall dynamics. Energy conservation was neglected. The SIMPLE numerical technique was used to solve the conservation equations. Similar results to that of CFD was obtained.

Izzom demonstrated an implicit-explicit (IMEX) Runge-Kutta (RK) method for the solution of stiff and non-stiff differential systems (Izzo and Jackiewicz, 2017). The IMEX-RK was derived by three different methods and was derived for the fourth to second-order. The conservation form of the 1D MME equations of the previous studies has two processes evolving on two different time scales. In the advection-diffusion PDE equations, there are elements of stiff coupling. In order to circumvent this, the study uses an implicit solution of the stiff part (diffusion) and an explicit solution for the non-stiff part (advection). The area of stability was maximised, and the coefficients of the explicit RK methods were optimised. This poses the possibility of creating stable methods that can handle stiff PDE problems.

PDE equations represent most thermo-fluid diffusion-advection problems. These PDE usually is solved numerically to solve numerous engineering problems. The Method of Lines is a technique where PDE equations are converted into ODE equations by using spatial approximations. These methods have been proven capable of handling stiff differential equations with modification, and in specific problems, no modification (Jackson, 1996; Shaw et al., 2017; Vande Wouwer et al., 2005; Ahmad and Berzins, 2001; Sarker and Chakravarty, 2020). Although a field of active development, the method proves useful as it allows PDE problems to be solved with the numerous ODE solvers that are available.

The Runge-Kutta method is one of the most well-known numerical methods for solving ODE problems, and its application and modifications are still an active field of study to this day (Bernardini and Pirozzoli, 2009; Vermeire, 2019; Caplan and Carretero-González, 2013; Eremin, 2019). In the past, this method was famous with Navier-Stokes flow codes used in the European aircraft design industry. The same codes using structured grids almost unanimously used explicit RK methods as their steady-state solver (Vos et al., 2002). Recent use of this method in IMEX methods enhances the usability of the method for diffusion and advection related problems (Izzo and Jackiewicz, 2017).

2.4.2.1 Concluding remarks - Numerical methods

First off, the well-known MOL strategy will be applied to reduce the conservation PDE equations into DE form. The MOL strategy is chosen from literature because of its ease of implementation. Finite differencing will be used due to its simplicity and the structured grid that will be applied. The former will further be adopted due to the study of a rigid pipe network, unlike the study of De Villiers (De Villiers, 2012). The studies of Izzom and Kriel made use of IMEX type solvers to hold the inherent benefit of a single flexible diffusion and advection solver (Izzo and Jackiewicz, 2017; Kriel, 2012). The domain, as used in this study, is advection dominated. Because of the former and the decoupling of the momentum equation, the study will use a fully explicit method. The relevance of the Runge-Kutta method has been demonstrated, as seen by several studies (Bernardini and Pirozzoli, 2009; Vermeire, 2019; Caplan and Carretero-González, 2013; Eremin, 2019). For this reason, the classical fourth-order Runge-Kutta method will be used to solve the MOL discretised problem.

From this section, the following conclusions can be made from the literature:

- IMEX numerical codes are becoming increasingly popular to handle both advection (non-stiff) and diffusion (stiff) related problems with one unified solver
- The RK method is to this day a relevant and adapting numerical method for modern applied sciences
- The MOL numerical approach is popular for structured grids

- The total pressure formulation is famous in 1D CFD codes and in-network simulations

2.4.3 Parametric studies and optimisation

Jonas, Frey and Theis by using TRNSYS simulations, studied the performance of parallel HP and ST (solar technology) systems and compared these to where no ST collectors are present (Jonas, Frey, and Theis, 2017). This simulation studied was conducted on four different configurations and two climatic regions. It was found that adding solar energy to HP systems can significantly improve system performance. For each of these configurations, a parametric study was conducted where storage tank size was varied from 500 - 2000 [L] and the solar collector areas from 5 - 20 [m²]. The study found that air source HP in combination with solar energy has the potential to achieve the same or even higher levels of seasonal performance than ground source HP systems for moderate climates. This becomes even more evident for buildings with low heating demand. Ground source HP systems in combination with solar energy have higher seasonal performance than air sourced HP systems. The difference in performance between these systems increases as the ambient temperatures decrease and the heat load demand increases.

Frangopoulos wrote a manuscript recently on the syntheses, design and optimisation (SDO) of energy systems. Optimisation, as end-use, commands the development of computer models (Frangopoulos, 2018). The different methods in parametric design, namely: search, calculus and stochastic methods were showcased. It was stated that if steady-state assumptions cannot be made on energy systems, then dynamic optimisation and simulation is often required. From this review; publications exerting this strategy for complex systems are rare in the literature. There is a renewed push to develop efficient but accurate methods for the optimisation of energy systems.

The study of Starke, Cardemil, and Colle; demonstrates an innovative approach for the multi-objective optimisation of solar assisted heat pumps systems (Starke, Cardemil, and Colle, 2018). Two design variables were changed, namely the heat pump scale factor as well as a solar field area parameter. The combined goal function was to minimise cost and user discomfort. A rectangular domain was constructed by varying the design parameters and calculating the sequential goal function value using TRNSYS. A response surface surrogate model was constructed in MATLAB. The genetic algorithm (GA) algorithms of MATLAB was used to construct the Pareto frontiers. All of the solar-assisted configurations showed annual life cycle savings (ALCS). The solar-assisted, water sourced heat pump had an ALCS between 395 and 887 [US\$].

The study of Beykal et al. evaluates a data-driven approach to surrogate model identification for optimum system design (Beykal et al., 2018). Beykal et al. developed the p-ARGONAUT system or parallel AlgoRithms for Global Optimisation of coNstrAined gre-box compUTational systems (Boukouvala and Floudas, 2017). The software uses the input-output of a system and the constraints for optimisation. Surrogate models are automatically identified. An optimisation was conducted on the derived surrogate models. Finally, the system was tested on multi-objective optimisation problems and was compared to that in the literature. The study was able to consistently and efficiently identify the Pareto frontier, using the η -constraint method along with p-ARGONAUT. The novel method out-performed all other derivative-free methods.

It is common among studies to make use of single-objective optimisation algorithms to design complex solar thermal systems. Rey and Zmeureanu presented the design of a combi-system against two conflicting objective functions (Rey and Zmeureanu, 2016). Eight design variables were varied, including the storage tank size and collector slope. MATLAB as an optimiser

was used in tandem with the system simulation package, TRNSYS. Three methodologies were used to design the system. The weighted sum method (WSM) using a hybrid particle swarm optimisation/Hooke-Jeeves algorithm, a multi-objective particle swarm algorithm and a hybrid multi-objective particle swarm optimisation/Hooke Jeeves (MOPSO/HJ). The execution time and the final design proposed by the different methods were evaluated. Using the MOPSO/HJ method, the life cycle cost and life cycle energy could be reduced by up to 88.6 [%] and 63.9 [%], respectively. The new MOPSO/HJ method also reduced the computing time to that of the WSM method by six times. Computing time for one optimisation run was around 86 [h]. Thus, to create the Pareto frontier, 39 days were required.

2.4.3.1 Concluding remarks - Parametric studies and optimisation

Frangopoulos noticed that the use of computer programs in optimisation or parametric studies defines the formers' development. It was furthermore noted that the design of energy systems requires computationally inexpensive computer programs (Frangopoulos, 2018). Recent literature also made use of model-reducing measures to create programs that can assist with the optimisation of heat pump solar hybrids (Starke, Cardemil, and Colle, 2018; Poppi et al., 2016; Rey and Zmeureanu, 2016; Calise et al., 2016; Jonas, Frey, and Theis, 2017). Contemporary research trends also indicate the escalating use of surrogate models in simulation studies to aid in the parametric design of thermal systems (Chang and Dinh, 2019; Beykal et al., 2018; Rey and Zmeureanu, 2016; Blackman et al., 2019; San and Maulik, 2018; Tran and Georgakis, 2018). The latest trends focus on reducing methods and the newest machine learning techniques. These findings illustrate the relevance of developing reduced computer methods for the parametric design of thermal systems. To this end, the study will aim to develop a validated model for solar heat pump systems for non-residential purposes.

From this section, the following conclusions can be made from the literature:

- The use of machine learning methods, along with surrogate models, are becoming increasingly useful design tools for energy and solar systems.
- It was noted that the development of a computer model is mostly dependant on whether it will be used for optimisation.
- Dynamic energy simulation (in the field of the study) with the goal of optimisation is scarce in the open literature.
- A recent trend has emerged where accurate but efficient models are required for the optimisation of energy systems.
- The use of multi-objective optimisation for the evaluation of energy systems are becoming increasingly important.

2.5 Concluding remarks

This chapter illustrated the critical concepts used from literature to develop a reduced computer model for the simulation of hybrid solar and heat pump systems. Table 2.2 illustrates the key concepts and those that were used. The central theme is the use of reducing measures to enable one to run various design simulations in quickened time frames. Central to this idea is the use of 'surrogate' models that imitate more complex modelling techniques. Table 2.1 illustrates the basic methodology that leads to the derivation of a response surface. The last step in

the table illustrates the implementation of these methods in a parametric study. The following chapter will deal with the experimental setup used as a case study and the calculation of the energy fluxes from the experimental data.

TABLE 2.2: Key concepts from literature and those used in the study.

Literature	Study
<i>Water tank storage</i>	
Plug flow (Kleinbach, 1990; Alizadeh, 1999; Yoo and Pak, 1996)	1D Heat Balance (Detailed and reduced simulation)
1D Heat balance (Yoo and Pak, 1996; Han, Wang, and Dai, 2009; Kleinbach, 1990; Alizadeh, 1999)	Mass-Energy (Reduced simulation)
1D Model with CFD correlations (Fan and Furbo, 2012; Baeten et al., 2016)	Mass-Momentum-Energy (Detailed simulation)
2D Heat balance (Ismail, Leal, and Zanardi, 1997; Han, Wang, and Dai, 2009)	
CFD analyses (Ievers and Lin, 2009; Fan and Furbo, 2012; Kurşun and Ökten, 2018)	
Mass-Momentum-Energy (Han, Wang, and Dai, 2009; Ievers and Lin, 2009; Fan and Furbo, 2012; Kurşun and Ökten, 2018)	
Mass-Energy (Ismail, Leal, and Zanardi, 1997; Guelpa and Verda, 2019)	
<i>Heat pump simulations</i>	
System-side regression (Shafieian, Khiadani, and Nosrati, 2019)	System-side regression (Detailed and reduced simulation)
Component-side regression (Ruschenburg 2014; Simon et al., 2016; Underwood, Royapoor, and Sturm, 2017)	
Component models (Chen and Yu, 2017; Bastos, Torres, and Castilla Álvarez, 2018; Çakır and Çomaklı, 2016; Van Eldik, 1998)	
<i>Solar water heater component</i>	
Hottel and Whillier (Shrivastava 2017; Rey and Zmeureanu, 2016)	Regression Graphs (Detailed and reduced simulation)
(Duffie and Beckman, 2013; Calise, Libertini, and Vicidomini, 2018)	Hydraulic effects (Detailed simulation)
Regression Graphs (Moss et al., 2018; Shafieian, Khiadani, and Nosrati, 2019; Daghigh and Shafieian, 2016)	
Effectiveness-NTU (Shafieian, Khiadani, and Nosrati, 2019; Daghigh and Shafieian, 2016)	
Hydraulic effects (Martínez-Rodríguez, Fuentes-Silva, and Picón-Núñez, 2018; Picón-Núñez, Martínez-Rodríguez, and Fuentes-Silva, 2014)	

Literature	Study
<i>Reduced-order models</i>	
Mass-Energy equations (Guelpa and Verda, 2019)	Mass-Energy equations (Reduced simulation)
Regressional polynomials (Ruschenburg2014; Mudunuru et al., 2017)	Regressional polynomials (Reduced and detailed simulation)
(Guelpa and Verda, 2019)	Simpler models (Reduced simulation)
Response-surface (Tran and Georgakis, 2018; Starke, Cardemil, and Colle, 2018; Bornatico et al., 2013)	
Neural Networks (Vázquez-Canteli et al., 2019; Elsheikh et al., 2019; San and Maulik, 2018)	
Simpler models (Sharma2019 ; Bornatico et al., 2013)	
Grey-box models (Sharma et al., 2019; Beykal et al., 2018; Boukouvala and Floudas, 2017)	
<i>Numerical Methods</i>	
Explicit-Implicit (IMEX) (Izzo and Jackiewicz, 2017; Chattot and Hafez, 2015; Kriel, 2012)	Mass-Energy (Reduced model)
Total pressure formulation (Kriel, 2012; Fick, 2014; Botha, 2003; De Villiers, 2012)	Runge–Kutta (Reduced model)
Runge–Kutta (Bernardini and Pirozzoli, 2009; Vermeire, 2019; Caplan and Carretero-González, 2013; Eremin, 2019)	Finite difference (Reduced simulation)
Method-of-Lines (Jackson, 1996; Shaw et al., 2017; Vande Wouwer et al., 2005; Ahmad and Berzins, 2001; Sarker and Chakravarty, 2020)	Finite difference (Detailed simulation)
Finite Difference (Chattot and Hafez, 2015; Vande Wouwer et al., 2005; Sarker and Chakravarty, 2020)	Method-of-Lines (Reduced simulation)
Finite Volume (Shaw et al., 2017; De Villiers, 2012; Tu, Yeoh, and Liu, 2008; Vos et al., 2002; Baeten et al., 2016)	

Chapter 3

Experimental

The hot water supply installation of a hospital was monitored to serve as the experimental setup to the study. The objective of the experimental measurements was to calculate the various energy fluxes from the experimental data and compare these to the results of the simulation program. It will become further evident from the chapter that the data was also used to derive a regression equation for the heat pump. Some of the experimental data would also serve as input to the computer model for the parametric studies conducted in the following chapters.

3.1 Experimental background and methodology

The measurements were conducted over a two month period that included some of the coldest months of the South-African winter (June - July). This period was chosen as it is the most challenging time of the year for hot water supply and would possibly strain the system within its design limits. Additionally, the extended test period allowed room for the calibration of the sub-component models used in the software and the validation thereof.

Monitoring a dynamic system with no control over its operation posed some challenges and opportunities to the study. The operational characteristics, such as user demand and weather patterns, changed daily. Furthermore, the performance of the ASHP and SWH bank was profoundly dependant on the ever-changing environmental conditions. As it was not a controlled environment, it did present the study with an opportunity to analyse a real-world system. Challenges included the deployment of the limited amount of sensors available to the study. These would have to be placed optimally to accommodate the transient boundary conditions of the system. Furthermore, due to the uncontrolled conditions imposed on the system; the calibration of consequent simulations was complicated. This chapter was divided into the following sections:

- Setup (3.2) - Prelude to the experimental layout of the case study and the main system components.
- Instrumentation (3.3) - Discussion on the several sensors available to the study.
- Energy balance (3.4) - The methodology for calculating the energy fluxes using the available instrumentation is presented.
- Regression-based fluxes (3.5) - The methodology used for the energy fluxes that requires a regression-based calculation is showcased.
- Experimental results (3.6) - The comprehensive discussion on results obtained from the case study and the analysis thereof.



FIGURE 3.1: *The system.*

3.2 Setup

The hot potable water supply system of a hospital in Potchefstroom, South Africa, was studied (Figure 3.2 and Figure 3.1). From here on, the experimental setup of the hospital will generally be referred to as '*the system*'. The system employs a hybrid energy supply scheme with inputs from air source heat pumps and extra energy supplied by an evacuated solar tube bank. Furthermore, the ASHPs can supply maximum heat input of 90 [kW] at 60 [°C] each, depending on ambient conditions. The solar bank consists of 96 evacuated tubes with an on-off control philosophy for the circulating pump. The designers implemented a ring main supply configuration with daily water circulation of up to 30 [kL] ¹ and a target supply temperature range of 55 - 60 [°C]. The former is common in large commercial systems and commonly referred to as '*hot water on tap*'. The ring-main circulates approximately the equivalent of two times the total storage capacity of the system ² Five insulated storage tanks with a combined capacity of 15 [kL] were installed. The hospital technical team uses a Supervisory Control and Data Acquisition (SCADA) system to monitor the system's daily performance. The system-level sub-components will be discussed in the following sections.

¹As obtained from measurement during the experimental test period.

²The total volume, in this case, refers to **only** the total capacity of the storage vessels.

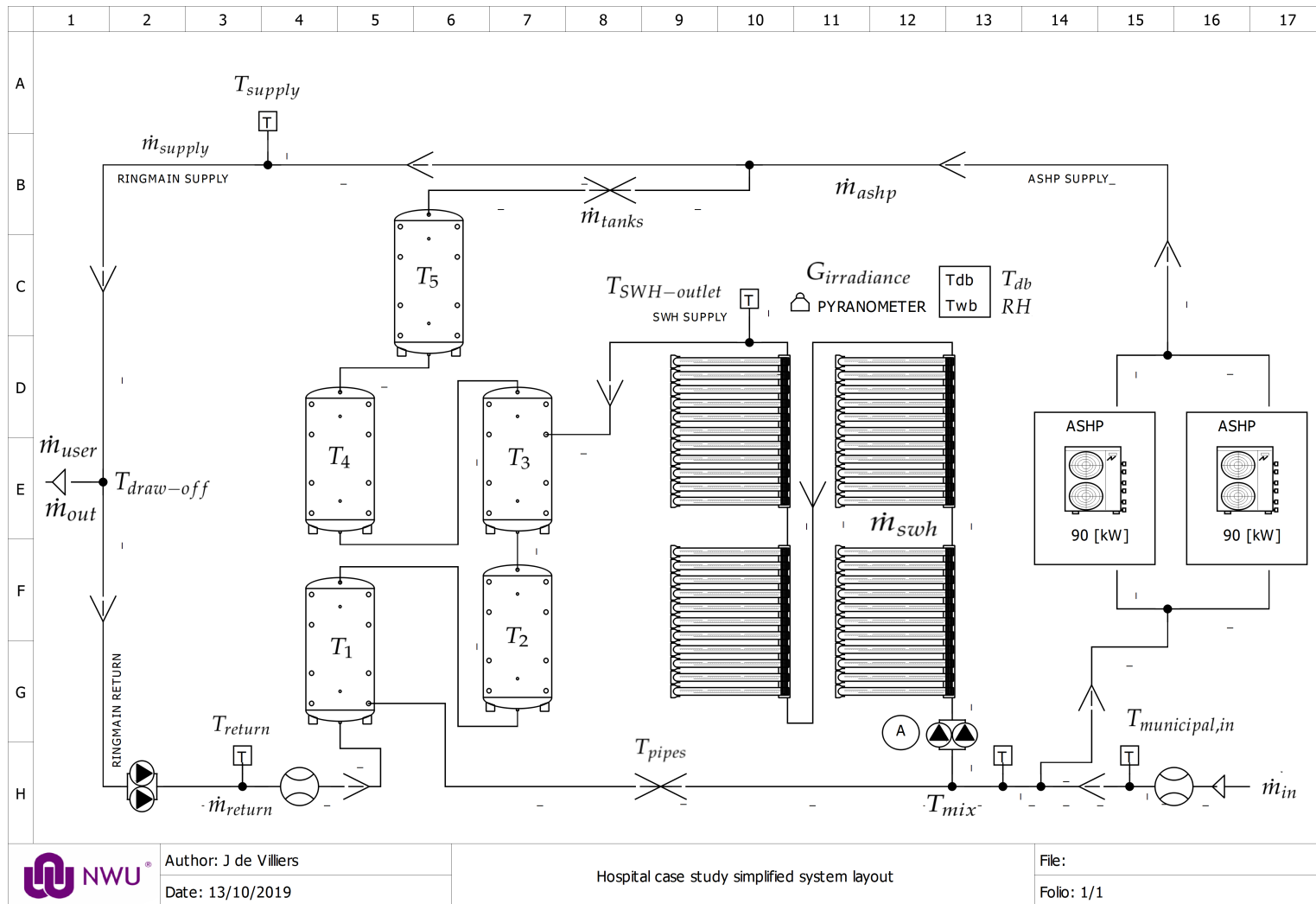


FIGURE 3.2: Simplified diagram of system layout.

3.2.1 Air-source heat pump

The ASHP is the primary energy source of the hot potable water system (Figure 3.3). The hospital has two ASHP units configured in parallel. Only one of the units are used at any given time to preserve the other machine for maintenance purposes. In winter, ambient temperatures in South-Africa do not plummet as low compared to other regions, such as Europe. The warmer climate is thus one of the reasons why ASHP technology is still a viable option with favourable performance characteristics in South-Africa.

Refer to Figure 3.2 for the system level diagram on the interfacing of the sub-components. The ASHP has been connected so that the inlet of the machine is the closest to the coldest storage vessel (diagram position *H14*, Figure 3.2). However, depending on user demand, the ASHP can use either source water entirely from the municipal supply, the storage tanks or a combination thereof. This is of noticeable significance as it determines how the system is analysed. Consequently, the outlet of the machine is directly connected to the hot water feed of the building and the top storage tank (*B11*, Figure 3.2). This dual ability permits the heat pumps to either charge the storage vessels or supply warm water directly to the building.

In this study a 'discharge' cycle will refer to a state where the user demand is greater than the heat pump supply mass flow ($\dot{m}_{user} > \dot{m}_{ashp} \mid \dot{m}_{tanks} > 0$ [kg s⁻¹]). Consequently, a 'charge' cycle will refer to the system state where the user demand is less than the mass flow supplied by the heat pump ($\dot{m}_{user} < \dot{m}_{ashp} \mid \dot{m}_{tanks} < 0$ [kg s⁻¹]). The study will follow the convention where a positive vessel mass flow ($\dot{m}_{tanks} > 0$) follows the direction of flow from the coldest tank (T_1 , *F6*, Figure 3.2) to the warmest tank (T_5 , *C7*, Figure 3.2) and *vice versa* for negative mass flow ($\dot{m}_{tanks} < 0$).

The ASHP employs a variable flow control philosophy with on-off control to maintain a fixed outlet temperature. Apart from the variable flow control, the machine uses a sensor in the second coldest storage vessel (T_2 , Figure 3.2) to determine if the machine needs to be activated or not. The user specifies a setpoint temperature (50 [°C]) pertaining to the mentioned sensor. Once the temperature falls below the setpoint; the machine waits for 15 minutes and then activates. The machine will deactivate once the setpoint temperature has been achieved. Once T_2 in Figure 3.2 falls below the setpoint, the machine will reactivate. Additional controls such as electrical overloading protection and frost prevention were also implemented by the original equipment manufacturers (OEM). The ASHP specifications can be found in Table 3.1.

TABLE 3.1: System-level ASHP specifications.

Specification	Value
Rated outlet temperature [°C]	55
Maximum outlet temperature [°C]	60
Maximum COP [°C]	4.6
Maximum rated heated water output [L h ⁻¹]	1882
Maximum heating capacity [kW]	87.4
Maximum heat energy [kW]	90
Ambient operating range [°C]	-7 to 43
Maximum power input [kW]	28.2



FIGURE 3.3: One of the 90 [kW] ASHP units of the system.

3.2.2 Solar water heater bank

The SWH bank serves as a supplemental energy source to that of the ASHP (Figure 3.4). The SWH bank consists of four banks of evacuated solar water heaters connected in series. Each bank consists of 24 evacuated tubes, with a combined area of $7.6 \text{ [m}^2\text{]}$. Furthermore, the circulation pump circulates at a flow rate of 32 [L/h] . Further information on the SWH bank can be sourced from Table 3.2.

The SWH works with an *on-off* control philosophy. The circulating pump and control system has been configured for a high-temperature differential before the pump is activated. The control strategy causes the SWH to charge and discharge periodically. The temperature control activates the SWH pump once the outlet temperature of the SWH is $12 \text{ [}^\circ\text{C]}$ warmer than the middle tank temperature (T_3 , E8, Figure 3.2) of the stratified tanks. Finally, the circulating pump deactivates when the temperature of the outlet of the solar is $4 \text{ [}^\circ\text{C]}$ warmer than the middle storage tank temperature (T_3 , E8, Figure 3.2). This ensures that the SWH control system operates with a control band to ensure smooth operation. Finally, the SWH has a temperature probe that ensures that the SWH will not activate once a user-specified maximum temperature in the storage vessels is exceeded.

TABLE 3.2: Evacuated solar bank specifications

<i>Specification</i>	<i>Value</i>
Total number of evacuated tubes [dim]	96
Number of evacuated tubes for a single bank [dim]	24
Glass thickness [mm]	2.2 - 2.3
Single bank aperture area [m ²]	1.9
Manifold tube diameter [mm]	48
Inner evacuated tube diameter [mm]	47
Outer evacuated tube diameter [mm]	58
Outer evacuated tube glass material [dim]	High Borosilicate glass
Metal tube heat pipe material [dim]	Red copper
Manifold maximum pressure [kPa]	8
Manifold insulation material [dim]	Rock Wool



FIGURE 3.4: The evacuated solar tube bank of the system.

3.2.3 Storage vessels

The system consists of five storage vessels (Figure 3.5), each with a capacity of 3 [kL]. The storage vessels were configured in a cascading stratified temperature arrangement, as seen in Figure 3.2. The average temperature difference between the five storage vessels is roughly two degrees Celcius during the "charge" cycles of the system. The warmest storage vessel's outlet is directly coupled to the building hot water supply and outlet of the ASHP (*B11*, Figure 3.2). The coldest storage vessel is connected to the cold municipal inlet feed, SWH inlet and building hot water return (*H14*, Figure 3.2). This allows the SWH and ASHP to work with the coldest possible water allowing the former and latter to work with maximum efficiency. The SWH outlet is connected to the 3rd storage vessels (*D8*, Figure 3.2) in the stratified storage vessel arrangement. The SWH has been configured in this manner as the temperature output of the

SWH is variable and may have a decremental effect on the hot water supply of the building if connected directly to the ring-main system. The storage vessels and all connected piping have been insulated with thin aluminium sheeting and insulation foam. The main characteristics of the storage vessels are listed in Table 3.3.

TABLE 3.3: Storage vessel specifications

Specification	Value
Single tank volume [L]	3000
Height [m]	2.72
Diameter [m]	1.22
Total number of tanks [dim]	5
Insulation material [dim]	Glasswool with resin
Thermal conductivity of insulating material [$\text{W}/(\text{K} \times \text{m}^2)$]	0.036



FIGURE 3.5: The sensible heat storage vessels of the system.

3.2.4 Ring-main

The ring-main system circulates water from 0.72 to 1.25 [kL/h] depending on the user load ³. The average temperature drop between the supply and return of the ring-main is roughly 6.6 [°C] ⁴. In Figure 3.2, the ring-main system is simplified to only a single draw-off point. The true ring-main consists of the main artery with numerous draw off points along the length of the main supply pipe. The draw-off pipes will be referred to in this study as 'dead-end' branches. Figure 3.6 depicts a simplified view indicating the configuration. A single draw-off point was

³The resistance curve of the ring-main changes according to the number of taps that are open. Furthermore, once the pumps of the ASHP activates; the flow through the ring-main changes.

⁴This temperature difference represents the temperature drop measured on the ring-main during the experimental period. Furthermore, the temperature drops that were measured varied between 4 - 9 [°C].

assumed to simplify the calculations on energy balance and simulation. The main characteristics of the building hot water supply are showcased in Table 3.4. The instrumentation installed at the installation and the additional sensors will now be discussed.

TABLE 3.4: Ring-main system-level characteristics

Description	Symbol	Value
Main supply line length [m]	L_{ring}	163
Coefficient of pipe heat loss [$W/m \times K$]	K_d	0.33
Average length of dead-end pipes [m]	$L_{dead-pipes}$	8.5
Outer diameter of main supply pipes [m]	d_o	0.05
Inner diameter of dead-end pipes [m]	d_i	0.02
Number of dead end pipes [dim]	$N_{dead-pipes}$	10
Time constant [m]	δ_t	900

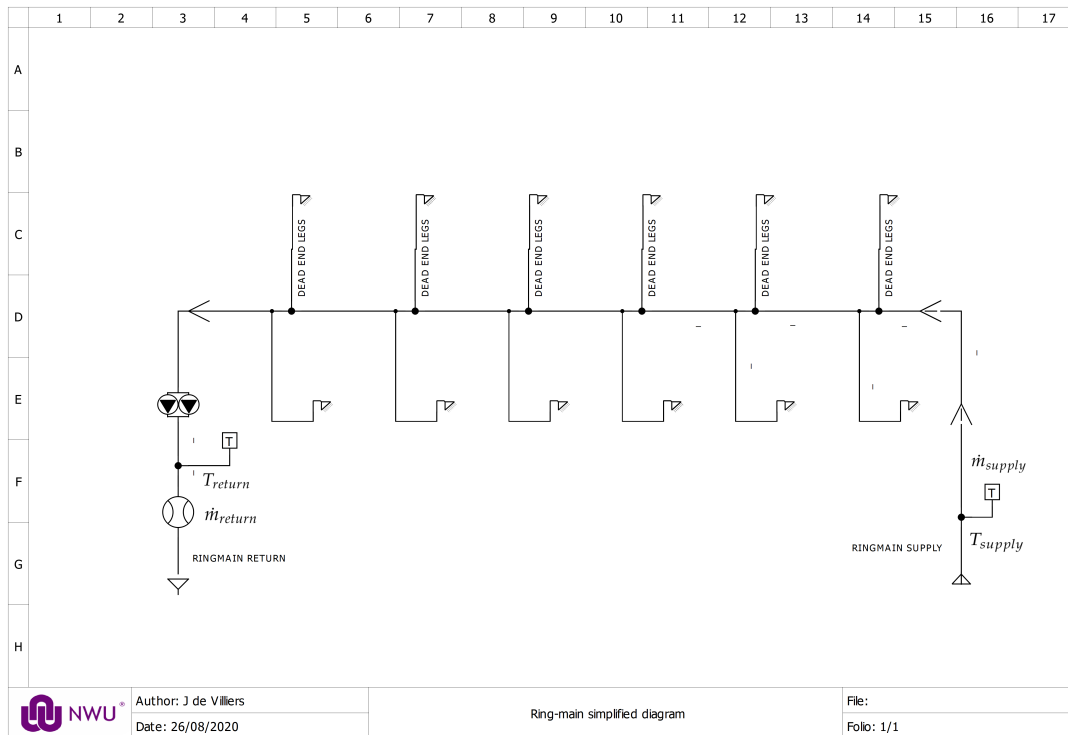


FIGURE 3.6: Simplified diagram on the ring-main system.

3.3 Instrumentation

The SCADA system used by the hospital has instrumentation that measures the most critical key performance indicators (KPI's). Of utmost importance is to ensure that the patients receive the targeted warm water, especially during periods of high user demand. The former entails continuous flow- and temperature measurement. The SCADA system, however, does not have sufficient sensors to measure all of the relevant terms pertaining to the energy fluxes for a 1st law energy balance. Thus additional sensors and HOBO loggers were installed to measure the applicable terms for the calculation of these energy fluxes. The sensors employed by the

SCADA system and the additional sensors are discussed in the sections to follow. Table 3.5 list all of the instrumentation used on the system and their positions in Figure 3.2.

The ring-main distributes the warm water of the system to the hospital and is well instrumented. The ring-main return flow rate ($\dot{m}_{return}$, H5, Figure 3.2), return (T_{return} , H4, Figure 3.2) and supply (T_{supply} , B4, Figure 3.2) temperatures were measured. By measuring the temperature difference; insight could be gained into the heat losses of the ring-main. The inlet municipal flow-rate ($\dot{m}_{in}$, H17, Figure 3.2) and temperature ($T_{municipal,in}$, H16, Figure 3.2) was measured to determine the user load. With these measurements, the ring-main characteristics, user consumption and supply temperatures could be monitored -Figure 3.6 aids in illustrating the ring-main in isolation from the main system.

To determine the performance and energy flux of the ASHP, SWH and heat loss characteristics in some of the pipes; the inlet conditions had to be known. The most economical placement for an additional sensor had to be determined to gain the inlet temperature for multiple components. A particular T-section where the municipal water supply, heat pump inlet, and coldest storage vessel connections meet (T_{mix} , H14, Figure 3.2) was chosen and termed 'mixing' point during the course of the study. This intersection can contain purely cold municipal water if the demand is high ($\dot{m}_{user} > (\dot{m}_{ashp} + \dot{m}_{swh})$). Alternatively, it can be a mixture of cold and warmer water obtained from the storage vessels ($\dot{m}_{user} < (\dot{m}_{ashp} + \dot{m}_{swh})$). Finally, it can be solely water obtained from the storage vessels ($\dot{m}_{user} = 0$). The mixing point temperature will later aid in determining if the system is in a 'charge' or 'discharge' energy state⁵.

Apart from the outlet temperature of the SWH bank ($T_{SWH-outlet}$, D11, Figure 3.2), the SCADA does not monitor other variables on the solar water heater bank. This includes solar irradiance. Although not necessary from an operational standpoint, the data is necessary as input for the simulation study. Consequently, a HOBO logger with a pyranometer (Figure 3.7) was installed in the inclined plane of the solar tube bank to allow for the calculation of the SWH efficiency. The solar pyranometer (C12, Figure 3.2) under question was the SP-215 pyranometer with a UX-120-006M HOBO logger. The inlet to the SWH is measured with the 'mixing' temperature probe discussed in the previous paragraph. For the mass flow through the SWH, an additional sensor was required.

⁵With 'charge' and 'discharge' pertaining to the definition as presented in paragraph 3.2.1.



FIGURE 3.7: The pyranometer installed on the SWH bank.

No modifications were allowed on the hospital system. Thus no additional flow sensors could be installed. With this limitation, an alternative was devised to obtain the flow rate through the SWH bank. An Onset® CTV-B 20A CT-ammeter (current transformer) was used to measure the current of the single-phase circulation pump (*C11*, Figure 3.2). A UX120-006M Hobo logger was used for data capture. It will later be shown that this ammeter was only used to indicate if the pump was active or not. The SCADA actively measures the reactive and active current and voltage per phase on the ASHPs. The auxiliary electricity devices utilise the second phase of the ASHP supply for electrical requirements. Using the voltage value measured by the SCADA system and the current measurement of a CT along with an assumed power factor, one can determine the theoretical flow rate from the pump curve.

The environmental wet bulb and dry bulb temperatures (T_{db}, T_{wb} , *C14*, Figure 3.2) were measured to predict the heat pump performance and calculate the estimated heat losses of the various system components. The ASHP regression model used the ambient conditions and the electrical power consumption measured by the SCADA to establish the heat energy added to the system.

TABLE 3.5: Sensors and their designations. Refer to Figure 3.2 for sensor locations.

<i>Description</i>	<i>Sensor type</i>	<i>Source</i>	<i>Drawing location</i>
Ringmain supply temperature (T_{supply}) [$^{\circ}\text{C}$]	Thermocouple	SCADA	B4
Ringmain return temperature (T_{return}) [$^{\circ}\text{C}$]	Thermocouple	SCADA	H4
Ringmain return flow ($\dot{m}_{return}$) [L s^{-1}]	Flow meter	SCADA	H5
Cold municipal consumption ($\dot{m}_{in}$) [L s^{-1}]	Flow meter	SCADA	H16
Outlet temperature SWH ($T_{SWH-outlet}$) [$^{\circ}\text{C}$]	Thermocouple	SCADA	C11
Solar irradiance ($G_{irradiance}$) [W/m^2]	Pyranometer	Supplemental	C12
Dry and wet bulb temperatures (T_{db}, T_{wb}) [$^{\circ}\text{C}$]	Thermocouple	Supplemental	C14
Coldwater inlet temperature ($T_{municipal,in}$) [$^{\circ}\text{C}$]	Thermocouple	Supplemental	H16
Mixing temperature node (T_{mix}) [$^{\circ}\text{C}$]	Thermocouple	Supplemental	H14
Ammeter of the SWH pump (I_{SWH}) [mA]	Current transformer	Supplemental	G13

3.3.1 Supplemental HOBO loggers and logging intervals

Additional HOBO loggers were installed to log the data supplied by the various additional sensors. A total of four additional loggers were used and, their main measurement sensors, along with characteristics, can be obtained from Tables 3.6 and 3.7. These loggers were additionally set up with different logging intervals to match the behaviour of the measured parameter. For solar irradiance, the measurement interval was set to five minutes as a one-minute interval would not have been necessary. For the electrical current measurement on the SWH circulating pump, a measurement interval of one minute was used to be able to capture the irregular pumping patterns. For the SCADA system of the hospital, all sensor information was extracted in 15 minute averaged intervals.

TABLE 3.6: Supplemental sensors and HOBO loggers. Refer to Figure 3.2 for sensor locations.

<i>Measurement</i>	<i>Hobo Logger Model</i>	<i>Supplemental Sensor</i>	<i>Drawing location</i>
Solar irradiance [W/m^2]	U12-008	SP-215 Pyranometer	C12
Dry and wet bulb temperatures [$^{\circ}\text{C}$]	U14-001	Internal DB/WB	C14
Coldwater inlet temperature [$^{\circ}\text{C}$]	U12-006	Supplemental	H16
Ammeter of the SWH pump [mA]	UX120-006M	Supplemental	G13
Mixing point temperature [$^{\circ}\text{C}$]	UX120-006M	Supplemental	H14

TABLE 3.7: HOBO loggers time logging intervals.

<i>HOBO Model</i>	<i>Number of inputs Model</i>	<i>Measurement interval Sensor</i>
U12-008 Outdoor/Industrial	4 channels	5 min
U14-001	No external channels	5 min
U12-006	4 channels	1 min
UX120-006M	4 channels	1 min

For the 1st law energy balance, the instantaneous energy flux for all of the parameters was calculated with data from the smallest possible time-step. This was done due to the fast transients of the SWH and the ASHP. Especially the SWH, as the SWH will activate and deactivate numerous times during some of the averaged 15-minute time-steps. Mass flow measurements would have been missed by taking a courser time grid. The process is shown graphically in Figure 3.8.

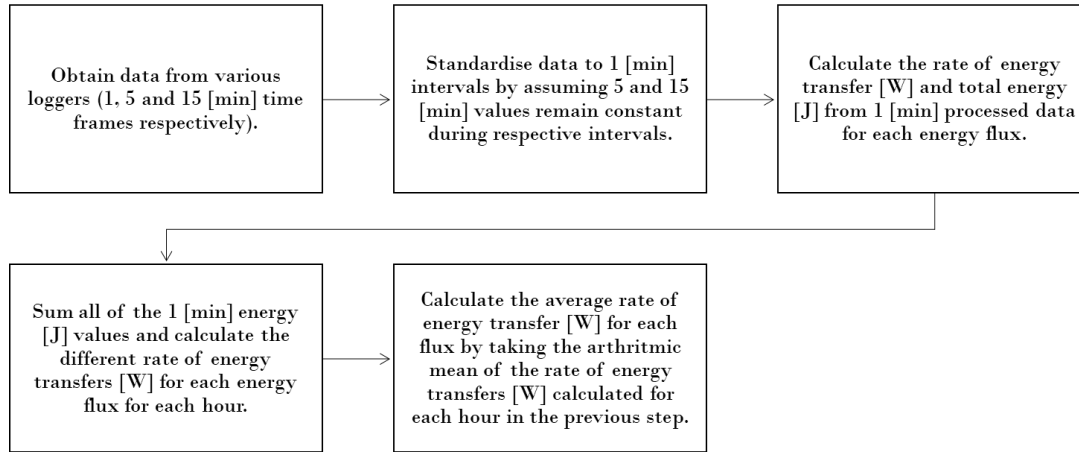


FIGURE 3.8: The methodology that was followed to process the experimental data on the most transient energy fluxes and the calculation of the hourly averaged values.

An example will present the process of calculating the energy fluxes. Assume the data to have been processed between all sensors to the smallest 1 [min] time step. Furthermore, assume the measured conditions to be constant within this minute. Figure 3.9 represents the instantaneous energy flux contributed by the SWH in a 15 [min] period. From Figure 3.9, it can be noted that the SWH pump activates in the second, fifth, ninth, and fourteenth minute. The average rate of energy transfer for the SWH in the 15 [min] period can be calculated by summing all of the energy transferred and dividing the sum by the elapsed time.

$$\begin{aligned} (3[\text{kW}] \times 60[\text{s}]) + (4[\text{kW}] \times 60[\text{s}]) + (5[\text{kW}] \times 60[\text{s}]) + (4[\text{kW}] \times 60[\text{s}]) &= 960[\text{J}] \\ 960[\text{J}] / (15 \times 60[\text{s}]) &= 1.066[\text{kW}] \end{aligned}$$

The same result can be obtained by taking the sum of the rate of energy transferred over the smallest time-steps (where the rate of energy transfer is assumed constant); and dividing the sum between the number of time-steps with constant conditions:

$$\begin{aligned} 3[\text{kW}] + 4[\text{kW}] + 5[\text{kW}] + 4[\text{kW}] &= 16[\text{kW}] \\ 16[\text{kW}] / 15 &= 1.066[\text{kW}] \end{aligned}$$

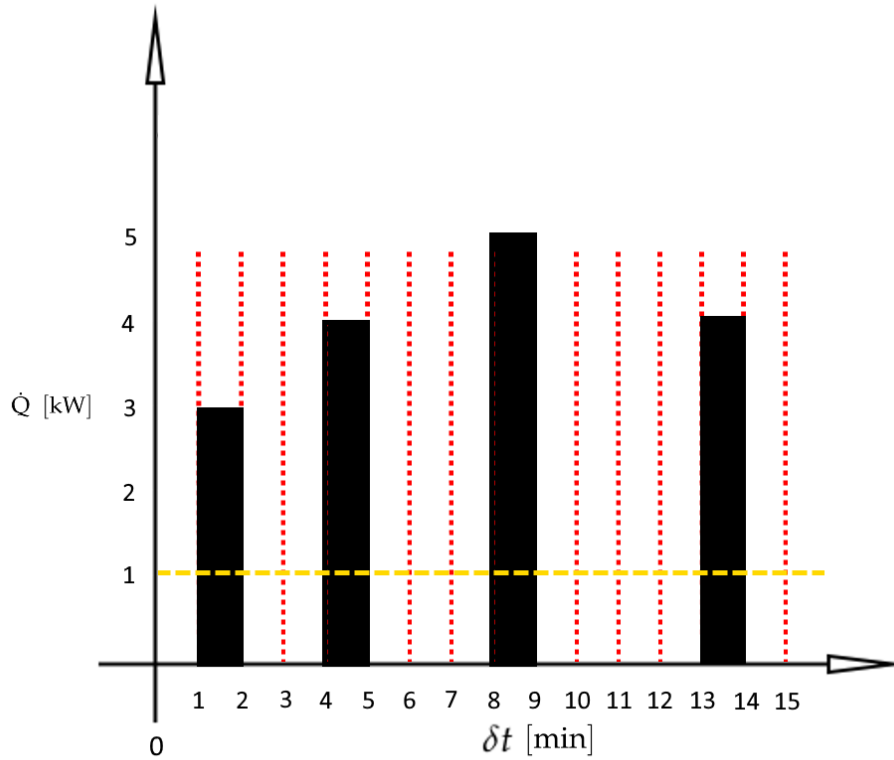


FIGURE 3.9: Energy calculation example.

In short, the raw data is standardised in 1 [min] time-steps. The energy transferred is calculated for each energy flux at each of the 1 [min] time-steps. The total energy is summed over the 24 [h] period. Hereafter the total energy is averaged over the 24 [h] period to obtain the averaged rate of energy exchange [W] for each energy flux (as tabulated at the end of the chapter). Additionally, the instantaneous rate of energy exchange [W] for each 1 [min] time-step was also calculated. This allowed the plotting of high-resolution graphs on the rate of energy transfer for each energy flux on the system. These graphs will be presented near the end of the chapter.

3.3.2 Sensor validation

The sensor validation on the additional sensors was undertaken before experimental measurements commenced. The sensor validation is discussed in Appendix A. Additional information on the sensors is also listed in the same appendix. The main system components, along with the various sensors and limitations, were discussed in the previous paragraphs. With this holistic view, the energy fluxes and the calculation thereof can be better understood.

3.4 1st Law Energy Balance

This section will focus on the calculation of the various energy fluxes from experimental data for the 1st law energy balance. For the study, a system boundary encompassing the whole system was chosen. The energy fluxes were:

Energy flux out:

- User energy load (3.4.1.1)
- Storage vessel heat losses (3.4.1.2)
- Ring-main energy losses (3.4.1.3.1)
- System pipe heat losses (3.4.1.3.2)
- SWH heat losses (3.4.1.3.3)

Energy flux in:

- Heat Pump (3.4.2.1)
- Solar Water Heater (3.4.2.2)

This subsection is divided into four sections—the first deals with the calculation method of the various energy fluxes mentioned above. Secondly, a preliminary energy balance is conducted. After the analyses on the results, the fluxes that required a regression component will be showcased. Finally, the final results obtained during the test period after alterations to the regression equation is showcased at the end of this section. Table 3.8 compiles the main equations of energy fluxes and measurable terms. Appendix C showcases an example of the calculation of the daily energy fluxes.

3.4.1 Energy Out

As with any thermodynamic system, there are inevitable heat losses. With the chosen delivery temperature, storage and circulations characteristics of the system, most of the energy was dissipated by heat losses. The remaining energy that leaves the system is that of the useful user energy flux.

3.4.1.1 User energy load

The user energy load compromises the energy necessary to heat the water consumed by the hospital. As the SCADA measures the total supplemental cold water supplied by the municipality ($\dot{m}_{in}$, B4 Figure 3.2), a mass balance could be calculated encompassing the whole system, including the ring-main.

The specific heat (at constant pressure) for water throughout the system is assumed constant⁶. Furthermore, the change in the density of water is neglected, and the system is pressurised, with the total system mass remaining constant. Thus from Reynolds transport theorem, the total mass that leaves the system ($\dot{m}_{user}$) must be supplemented by the total mass that enters the system ($\dot{m}_{in}$). As such, the flow meter, in essence, was measuring the water consumption of the system. Working with the previously mentioned assumption of a single draw-off point on the ring-main, the temperature of the extracted water could be approximated.

A temperature on the inlet ($T_{municipal,in}$, H16, Figure 3.2) of the system was measured. The draw-off temperature on the ring-main was taken as the average of the return (T_{return} , H4, Figure 3.2) and supply temperatures (T_{supply} , B4, Figure 3.2) on the warm water building supply:

⁶Appendix A.5 discusses the variation of the parameter with temperature.

TABLE 3.8: Equations for the various energy fluxes.

Terms	Equations
User energy load	
Measured: $T_{\text{supply}}, T_{\text{return}}, T_{\text{municipal,in}}, \dot{m}_{\text{out}}, \dot{m}_{\text{in}}$	$T_{\text{draw-off}} = \frac{T_{\text{supply}} + T_{\text{return}}}{2}$ $\dot{Q}_{\text{user,out}} = \dot{m}_{\text{out}} C_{p,\text{water}} T_{\text{draw-off}}$
Assumed: $C_{p,\text{water}}$	$\dot{m}_{\text{in}} = \dot{m}_{\text{out}}$ $\dot{Q}_{\text{municipal,in}} = \dot{m}_{\text{in}} C_{p,\text{water}} T_{\text{municipal,in}}$ $\dot{Q}_{\text{user}} = \dot{Q}_{\text{user,out}} - \dot{Q}_{\text{municipal,in}}$
Storage vessel heat losses	
Measured: T_{ambient}	$\dot{Q}_{\text{tank-losses}} = UA_{\text{system}} (T_{\text{tank}} - T_{\text{ambient}})$
Calculated: UA_{system} (Appendix B.1)	
Assumed and measured: T_{tank} depending on system state (Chapter 3.4.1.2)	
Ring-main energy losses	
Measured: $\dot{m}_{\text{return}}, \dot{m}_{\text{out}}, T_{\text{supply}}, T_{\text{return}}$	$\dot{Q}_{\text{ringmain,supply}} = \dot{m}_{\text{supply}} C_{p,\text{water}} T_{\text{supply}}$ $\dot{Q}_{\text{ringmain,return}} = \dot{m}_{\text{return}} C_{p,\text{water}} T_{\text{return}}$
Assumed: $C_{p,\text{water}}$	$\dot{Q}_{\text{ringmain,losses}} = \dot{Q}_{\text{ringmain,supply}} - \dot{Q}_{\text{ringmain,return}}$ $\dot{m}_{\text{supply}} = \dot{m}_{\text{return}} + \dot{m}_{\text{out}}$ $\dot{Q}_{\text{user,ringmain}} = \dot{m}_{\text{user}} C_{p,\text{water}} (T_{\text{supply}} - T_{\text{draw-off}})$ $\dot{Q}_{\text{ringmain,steady-state}} = \dot{Q}_{\text{ringmain,losses}} - \dot{Q}_{\text{user,ringmain}}$
System pipe heat losses	
Measured: $T_{\text{ambient}}, T_{\text{mix}}, T_{\text{supply}}$	$\dot{Q}_{\text{pipe-losses}} = UA_{\text{pipe-losses}} (T_{\text{pipes-tank}} - T_{\text{ambient}})$ $T_{\text{pipes-tank}} = \frac{T_{\text{mix}} + T_{\text{supply}}}{2}$
Calculated: $UA_{\text{pipe-losses}}$ (Appendix B.2)	
SWH heat losses	
Measured: $T_{\text{ambient}}, T_{\text{SWH-pipes}}$	$\dot{Q}_{\text{SWH-pipes}} = UA_{\text{SWH-pipes}} (T_{\text{SWH}} - T_{\text{ambient}})$
Calculated: $UA_{\text{SWH-pipes}}$ (Appendix B.2)	
Heat pump	
Measured: $T_{\text{db}}, RH, T_{\text{In-ASHP}}, \dot{P}_{\text{electrical}}$	$COP_{\text{ASHP}} = f(T_{\text{db}}, RH, T_{\text{In-ASHP}})$ $\dot{Q}_{\text{ASHP}} = \dot{P}_{\text{electrical}} COP_{\text{ASHP}}$
Assumed: $COP_{\text{ASHP}} = f()$ (Chapter 3.5.1)	
Solar water heater	
Measured: $T_{\text{SWH-outlet}}, T_{\text{inlet}}$	$\dot{Q}_{\text{swh-1}} = \dot{m}_{\text{swh}} C_{p,\text{water}} (T_{\text{SWH-outlet}} - T_{\text{inlet}})$
Assumed and measured: $\dot{m}_{\text{swh}}$ (Chapter 3.4.2.2)	
Assumed: $C_{p,\text{water}}$	

$$T_{draw-off} = \frac{T_{supply} + T_{return}}{2} \quad (3.1)$$

By combining the measurements and the temperature approximation; the user load flux could be calculated from:

$$\dot{Q}_{municipal,in} = \dot{m}_{in} \times C p_{water} \times T_{municipal,in} = \dot{m}_{municipal,in} \times h_{municipal,in} \quad (3.2)$$

$$\dot{Q}_{user,out} = \dot{m}_{out} \times C p_{water} \times T_{draw-off} = \dot{m}_{out} \times h_{supply,user} \quad (3.3)$$

$$\dot{Q}_{user} = \dot{Q}_{user,out} - \dot{Q}_{municipal,in} \quad (3.4)$$

The terms of the above equations, along with the sensors and their respective error bands, are tabulated in Table 3.9. Although sensor quality is measured with more parameters than the measurement accuracy; for simplicity, the latter is shown in the table below. Additional sensor information can be referenced in Appendix A. The propagation of uncertainty is discussed in Appendix A.5 and the error will be reported in the tables below throughout the chapter.

TABLE 3.9: User load energy flux term, source and general sensor accuracy. For detailed sensor information, refer to Appendix A. Refer to Figure 3.2 for drawing locations.

Term	Symbol	Source	Source Accuracy	Drawing location
System mass flow in [kg/s]	$\dot{m}_{in}$	MeiStream 50	1.8 [%]	H17
System mass flow out [kg/s]	$\dot{m}_{out}$	Calculated	1.8 [%]	E2
Ring-main supply temperature [°C]	T_{supply}	RS PT100	$\pm(0.3[^\circ\text{C}] + 0.5[\%])$	B4
Ring-main return temperature [°C]	T_{return}	RS PT100	$\pm(0.3[^\circ\text{C}] + 0.5[\%])$	H4
System inlet temperature [°C]	$T_{municipal,in}$	Hobo TMP probe	$\pm 0.15[^\circ\text{C}]$	H15
Specific heat at constant pressure [J/°C]	$C p_{water}$	Assumed	n/a	n/a
Propagation of measurement error [dim]	ϵ_c	Calculated	3.04 [%]	n/a

3.4.1.2 Heat loss storage vessels

Apart from the ring-main network, the heat losses of the storage vessels contributed to a significant energy expense on the system. One can, however, note the efforts from Table 3.3 and Figure 3.5 to combat these losses with significant insulation. The convective losses due to forced and natural convection is calculated in Appendix B. It could be seen that the natural convection heat losses dominate heat transfer. The reason for this being that the maximum flow of $1.2 [\text{kg s}^{-1}]$ is circulated over a cross-sectional diameter of $0.75 [\text{m}]$. This leads to a slow average (forced) velocity. The heat loss of the storage vessels can be calculated as:

$$\dot{Q}_{tank-losses} = UA_{system} \times \Delta T = UA_{system} \times (T_{tank} - T_{ambient}) \quad (3.5)$$

The derivation of the UA_{system} term and convective flow analyses is discussed in Appendix B. T_{tank} is assumed as the average fluid temperature of all of the vessels. The sensors that were used to represent this term was the supply temperature (T_{supply} , B4, Figure 3.2) of the ring-main and the mix temperature probe (T_{mix} , H14, Figure 3.2)⁷ at the coldest part of the storage vessels.

⁷With T_{mix} pertaining to the definition as introduced in paragraph 3.3.

The rest of the vessels temperatures were calculated based on a linear temperature profile. The assumptions on the various tank temperatures were as follows for the *charge cycle*:

$$\begin{aligned}
 T_{max} &= T_5 = T_{supply} \\
 T_n &= \left(\frac{T_{max} - T_{min}}{5} \times n \right) + T_{min} \\
 n &\in \{2, 3, 4\} \\
 \forall n &\neq \min, \max \\
 T_{min} &= T_1 = T_{mix}
 \end{aligned}$$

Where n defines the index of the storage tank vessels. The temperatures on each tank node could not be measured. The former was mainly due to sensors shortages and the restriction to change the hospital system. Thus, an alternative was devised for the *discharge cycle*. In the *discharge* configuration, the minimum temperature probe does not measure the coldest tank temperatures but mainly the cold water supplied by the municipal inlet. It was noted during the experiment from the analogue temperature probes that a two-degree temperature stratification per vessel was maintained (Figure 3.10). Therefore, for the *discharge* case, the outlet temperature of the warmest tank is used for the top node/tank, and a linear two-degree temperature stratification was assumed for the rest of the tank temperatures. Thus in equation format:

$$\begin{aligned}
 T_{max} &= T_5 \\
 T_n &= T_{max} - ((5 - n) \times 2) \\
 \forall n &\neq 5
 \end{aligned}$$

The terms in the equations above, along with the sensors and their respective error bands are tabulated in Table 3.10.

TABLE 3.10: Storage vessel heat loss flux term, source and general sensor accuracy. For detailed sensor information, refer to Appendix A. Refer to Figure 3.2 for drawing locations.

Term	Symbol	Source	Source Accuracy	Drawing location
Ambient dry bulb temperature [°C]	$T_{ambient}$	Hobo U14-001	$\pm 0.21[^\circ\text{C}]$	C13
Average tank temperature [°C]	T_{tank}	Calculated	n/a	n/a
Ring main supply temperature [°C]	T_{supply}	RS PT100	$\pm(0.3[^\circ\text{C}] + 0.5[\%])$	B4
Mixing temperature [°C]	T_{mix}	Hobo TMP Probe	$\pm 0.15[^\circ\text{C}]$	H14
Heat loss storage vessel [W/°C]	UA_{system}	Appendix B	n/a	n/a
Propagation of measurement error [dim]	ϵ_c	Calculated	2.45[%]	n/a

3.4.1.3 Piping losses

The user energy and storage vessel energy fluxes were discussed in the previous sections. The remaining energy expenses on the system is based on losses in the pipe network. The system's piping was divided into three main components for the definition of '*piping losses*':

- Ring-main (3.4.1.3.1)
- System pipes (3.4.1.3.2)
- SWH pipes (3.4.2.2)

The ring-main losses entail the energy expended (by heat losses) by pumping the water through the supply network of the hospital. These losses do not include the user consumption that is extracted from said network. The system pipes refer to all additional piping apart from the



FIGURE 3.10: Temperature stratification between warmest and coldest tank during a 'charge' cycle.

ring-main supply network and the SWH pipes. The SWH pipes are the pipes leading towards and from the solar bank. The following sections will discuss the calculation of the energy fluxes, as mentioned above.

3.4.1.3.1 Ring-main heat losses Warm water is circulated throughout the ring-main network in a perpetual fashion. Continuous flow inevitably leads to a significant heat loss with the associated energy cost. One can delimit the average energy loss in the ring-main system as:

$$\dot{Q}_{\text{ringmain,supply}} = \dot{m}_{\text{supply}} \times C_{p_{\text{water}}} \times T_{\text{supply}} = \dot{m}_{\text{supply}} \times h_{\text{supply}} \quad (3.6)$$

$$\dot{Q}_{\text{ringmain,return}} = \dot{m}_{\text{return}} \times C_{p_{\text{water}}} \times T_{\text{return}} = \dot{m}_{\text{return}} \times h_{\text{return}} \quad (3.7)$$

$$\dot{Q}_{\text{ringmain,losses}} = \dot{Q}_{\text{ringmain,supply}} - \dot{Q}_{\text{ringmain,return}} \quad (3.8)$$

All of the above variables on the right-hand side were measured directly except for the supply mass flow ($\dot{m}_{\text{supply}}$) and the specific heat at constant pressure ($C_{p_{\text{water}}}$). Positions (T_{supply} , **B4**), (T_{return} , **H4**) and ($\dot{m}_{\text{return}}$, **H5**) is the terms of the above equation and their respective measurement locations in Figure 3.2.

The cold municipal flow is measured ($\dot{m}_{\text{in}}$, **H16** Figure 3.2) along with the return mass flow ($\dot{m}_{\text{return}}$, **H5** Figure 3.2) on the ring-main system. Therefore, the supply mass flow to the user can be calculated from:

$$\dot{m}_{\text{supply}} = \dot{m}_{\text{return}} + \dot{m}_{\text{in}} \quad (3.9)$$

The ring-main losses expressed in the above equation encompasses the user load that circulates through the ring-main and the heat losses in the ring-main:

$$\dot{Q}_{\text{ringmain,steady-state}} = \dot{Q}_{\text{ringmain,losses}} - \dot{Q}_{\text{user,ringmain}} \quad (3.10)$$

$$\dot{Q}_{ringmain,steady-state} = \dot{m}_{return}Cp_{water}(T_{supply} - T_{return}) - \dot{m}_{user}Cp_{water}(T_{supply} - T_{draw-off}) \quad (3.11)$$

Although the above was sufficient for the 1st law energy balance, the simulation requires additional characteristics to simulate the ring-main. These characteristics can be written in compact equation format as follows:

$$\dot{Q}_{ringmain,steady-state} = f(UA_{ring-main}, T_{ringmain,average}, T_{indoors}) \quad (3.12)$$

Accordingly, a special case on the ring-main was investigated and is showcased in Appendix B.3. The derived characteristics is tabulated in Table 3.4. The terms of the above equations, the sensors and their respective error bands are tabulated in Table 3.11 below.

TABLE 3.11: Ring-main flux term, source and general sensor accuracy. For detailed sensor information, refer to Appendix A. Refer to Figure 3.2 for drawing locations.

Term	Symbol	Source	Source Accuracy	Drawing location
Ring main supply mass flow [kg/s]	$\dot{m}_{supply}$	Calculated	2.54 [%]	B4
Ring main return mass flow [kg/s]	$\dot{m}_{return}$	MeiStream® 50	1.8 [%]	H4
Ring main supply temperature [°C]	T_{supply}	RS PT100	$\pm(0.3[^\circ\text{C}] + 0.5[\%])$	B4
Ring main return temperature [°C]	T_{return}	RS PT100	$\pm(0.3[^\circ\text{C}] + 0.5[\%])$	H4
Propagation of measurement error [dim]	ϵ_c	Calculated	3.19 [%]	n/a

3.4.1.3.2 Heat loss system pipes The system apart from the ring-main loop consists of various pipes connecting the storage tanks and heat sources, respectively. The length of these pipes was measured. With the measured lengths, the exposed area could be calculated. The internal temperature was assumed as the average between the mixing and maximum tank node measurements ($T_{pipes-tank} = \frac{T_{mix} + T_{supply}}{2}$). The calculation of the heat loss coefficient (UA_{pipes}) is discussed in Appendix B. The system pipe heat losses can be calculated using:

$$\dot{Q}_{pipe-losses} = UA_{pipes} \times \Delta T = UA_{pipe-losses} \times (T_{pipes-tank} - T_{ambient}) \quad (3.13)$$

The terms of the above equations, the sensors and their respective error bands are tabulated in Table 3.12 below.

TABLE 3.12: System pipes heat loss flux term, source and general sensor accuracy. For detailed sensor information, refer to Appendix A. Refer to Figure 3.2 for drawing locations.

Term	Symbol	Source	Source Accuracy	Drawing location
Ambient dry bulb temperature [°C]	$T_{ambient}$	Hobo U14-001	$\pm 0.21[^\circ\text{C}]$	C13
Mixing temperature probe [°C]	T_{mix}	Hobo TMP Probe	$\pm 0.15[^\circ\text{C}]$	D11
Ring main supply temperature [°C]	T_{supply}	RS PT100	$\pm(0.3[^\circ\text{C}] + 0.5[\%])$	B4
Heat loss system pipes [W/°C]	UA_{pipes}	Appendix B	n/a	n/a
Propagation of measurement error [dim]	ϵ_c	Calculated	2.45 [%]	n/a

3.4.1.3.3 Heat loss SWH pipes Leading from the *on-off* control philosophy of the SWH system; warm water accumulated frequently in the pipe network of the SWH. The pipes have a much higher exposed surface to volume ratio to that of the storage vessels. This ratio,

along with the comparatively higher temperatures, consequently leads to high heat losses. The SCADA system measures the output temperature of the SWH. The average temperature in the solar pipes is taken as the outlet temperature measured by the SCADA system. It should be noted that this energy loss does only take into account the supply pipes leading from the discharge of the SWH to the storage vessels. The heat losses through the pipes were calculated from:

$$\dot{Q}_{SWH-pipes} = UA_{SWH-pipes} \times \Delta T = UA_{SWH-pipes} \times (T_{SWH-pipes} - T_{ambient}) \quad (3.14)$$

TABLE 3.13: SWH heat loss flux term, source and general sensor accuracy. For detailed sensor information, refer to Appendix A. Refer to Figure 3.2 for drawing locations.

Term	Symbol	Source	Source Accuracy	Drawing location
Ambient dry bulb temperature [°C]	$T_{ambient}$	Hobo U14-001	$\pm 0.21[\%]$	C13
SWH output temperature [°C]	$T_{SWH-pipes}$	RS PT100	$\pm(0.3[^\circ\text{C}] + 0.5[\%])$	D11
SWH heat loss [W/°C]	$UA_{SWH-pipes}$	Appendix B	n/a	n/a
Propagation of measurement error [dim]	ϵ_c	Calculated	2.39 [%]	n/a

The derivation of the heat loss coefficient of the pipes $UA_{SWH-pipes}$ is discussed in Appendix B. Table 3.14 lists the system-level specification for the SWH pipes. The terms of the above equation, the sensors and their respective error bands are tabulated in Table 3.13. Therefore, this concludes the section on energy fluxes out of the system.

TABLE 3.14: SWH and system pipes specifications

Description	Symbol	Value
Pipe length SWH [m]	L_{SWH}	34.7
Pipe length tanks [m]	L_{SWH}	30
Inside diameter pipes [m]	d_i	0.03
Outside diameter pipes [m]	d_o	0.055
Area of pipes [m ²]	$A_{SWH-pipes}$	5.92
Volume of pipes [m ³]	$V_{SWH-pipes}$	10.02
Overall heat loss coefficient of SWH pipes [W K ⁻¹]	$UA_{SWH-pipes}$	21.12
Overall heat loss coefficient of tank pipes [W K ⁻¹]	$UA_{Tank-pipes}$	18.26

3.4.2 Energy In

The energy sources adding towards the net energy balance of the system consists mainly of the ASHP and the SWH to a lesser extent.

3.4.2.1 Heat pump

Due to the restriction on modifications to the system, the mass flow supplied by the heat pump could not be measured. Therefore, to determine the useful heat supplied by the HP, the manufacturers' performance matrix was utilised. A regression equation based on the manufacturer data is derived in Appendix H and will be discussed in the following sections. As mentioned in Chapter 3.3, the SCADA actively monitors the electrical supply and consumption on the ASHP. The dry-bulb temperature (T_{db}) and relative humidity (ϕ) were measured at the corresponding

time step to calculate the theoretical COP (β) from the derived regression equation. Using the electrical data measured by the SCADA and multiplying the former and latter, the theoretical heat input ($\dot{Q}_{ASHP}$) could be calculated.

$$\dot{P}_{electrical} = \dot{P}_{phase_1} + \dot{P}_{phase_2} + \dot{P}_{phase_3} \quad (3.15)$$

$$\dot{P}_{phase_i} = (I_{SCADA-i-Phase} \times V_{SCADA-i-Phase-Neutral}) \quad (3.16)$$

$$\beta = f(T_{db}, T_{inlet}, \phi) \quad (3.17)$$

$$\dot{Q}_{ASHP} = \dot{P}_{electrical} \times \beta \quad (3.18)$$

3.4.2.2 Solar water heater (SWH)

For this energy flux, two calculation approaches were used. In one method, the direct change in the energy of the water was determined. The other method entailed using a performance curve and was necessary for the simulation program. The direct energy added to the water was calculated using the following:

$$\dot{Q}_{swh} = \dot{m}_{swh} \times C_{p_{water}} \times (T_{SWH-outlet} - T_{inlet}) \quad (3.19)$$

The system boundary was assumed to start and stop at the SWH manifold inlet and outlet. The flux thus represents the useful energy after the SWH has dissipated heat-energy through the manifold. However, certain assumptions had to be made to ensure that the method could be used. As discussed in Chapter 3.3, the current measurement was used to determine the flow rate through the SWH flow pump. The problem, however, is that the ammeter used on the single-phase pump could not effectively measure [mA] values. It could, however, detect when the pump was active or not. Therefore a hand-held, high-resolution ammeter was used to do a spot check on the circulating pump. The steady-state measurement was then used as the steady-state electrical current measurement. Therefore the HOBO logger was used only to indicate when the circulating pump was active or not.

With the steady-state circulating electrical current known, the power consumed by the pump could be calculated. The auxiliary pumps use the second phase of the three-phase ASHP electrical supply. As the latter is actively monitored by the SCADA system, the voltage supplied to the circulating pump is known. The flow rate was calculated by assuming a power factor for the water pump:

$$\dot{P}_{pump} = V_{rms} \times I_{rms} \times \cos(\theta) \quad (3.20)$$

$$\dot{Q}_{flow} = f(\dot{P}_{pump}) \quad (3.21)$$

with

$$\frac{\partial \rho}{\partial t} = 0 \quad (3.22)$$

$$\dot{m}_{flow} = \dot{Q}_{flow} \times \rho \quad (3.23)$$

The theoretical solar energy contribution could be calculated by using solar measurement and the performance curve of the SWH (Figure 3.13). Using the steady-state mass flow and inlet

conditions, one could calculate the energy flux and outlet temperature of the SWH by neglecting thermal capacitance. The second method entails:

$$\eta_{solar} = f(G_{irradiance}, T_{inlet}) \quad (3.24)$$

$$\dot{Q}_{swh-in} = \eta_{solar} \times G_{irradiance} \quad (3.25)$$

$$\dot{Q}_{swh-out} = (1 - \eta_{solar}) \times G_{irradiance} \quad (3.26)$$

$$T_{SWH-outlet} = (\dot{Q}_{swh-in} - \dot{Q}_{swh-out}) / (\dot{m}_{swh} \times C_{p_{water}}) + T_{inlet} \quad (3.27)$$

3.5 Regression-based energy fluxes

The equations that were used to calculate the energy fluxes of the ASHP and SWH was showcased in Sections 3.4.2.1 and 3.4.2.2. The regression equations simplified the calculation of the energy balance and its use in the consequent simulation. Making use of regression methods had the added benefit that the use of complex physics simulation for the sub-components was circumvented. These models were derived from manufacturer data. The following section focuses on the derivation of these regression-based approximations.

3.5.1 HP regression model

The statistical method that was used to establish the regression for the ASHP will be showcased in the following sections. Regression requires the following steps to develop an equation:

- Identify independent variables.
- Establish a relationship between independent variables.
- Make use of statistical methods to establish the feasibility of the proposed relation using data.
- If unfeasible, adjust the relationship, add additional variables or obtain a more suitable dataset.

The above will be explained below. For a detailed discussion on the regression of the ASHP performance, refer to Appendix H. The ASHP used in this study was designed to deliver a fixed outlet temperature. By utilising a system-level approach, it can be noted that the heat pump performance will be highly determinate on the heat source and sink conditions (Carnot efficiency). This will depend on the mass flow and thermal properties of the refrigerant and ambient air.

Psychrometry helps to illustrate the impact of relative humidity on heat pump performance and its essential incorporation into the regression model. One can conclude that for a specific heat pump, the following factors will impact the performance of the machine:

- Psychrometrics at HP evaporator (T_{db}, RH)
- Inlet water temp at HP condenser (T_{inlet})⁸

⁸The outlet condenser temperature on the refrigerant side determines the condensing pressure. However, the outlet temperature is relatively constant and therefore, the inlet on the waterside is used as an independent parameter.

The dry bulb, wet bulb and total pressure conditions determine the available enthalpy at the source (evaporator). Furthermore, the inlet water conditions determine the system conditions at the sink (condenser). The following section will aid in explaining the impact of relative humidity on the available energy to an ASHP cycle. From psychrometry and Dalton's law, the following equations can be used:

$$P_v = P_g \left(\frac{\phi}{100} \right)^9 \quad (3.28)$$

$$P_a = P_{atm} - P_v \quad (3.29)$$

$$\omega = \frac{MW_{water} \times P_v}{MW_{air} \times P_a} \quad (3.30)$$

$$h_{mix} = \frac{(\omega \times h_v) + h_a}{\omega + 1} \quad (3.31)$$

From the above analyses, it can quickly be recognised that the enthalpy of the gas-vapour mixture increases as the dry bulb temperature (T_{db}) and consequently enthalpy of the dry air increases (h_a). With the increase in relative humidity (ϕ)¹⁰, the enthalpy of the gas-vapour mixture at the evaporator of the heat pump increases. Nevertheless, higher enthalpy at the evaporator does not guarantee increased performance as the energy has to be transferred to the evaporator by heat transfer. Three types of heat transfer conditions can manifest at the evaporator of an ASHP unit based on the ambient dry bulb temperature, relative humidity, atmospheric pressure and evaporator temperature:

- Frosting
- Condensation
- Dry air heat transfer

From literature, the following relations between the variables listed above and ASHP performance have been made:

- For frosting conditions, an increase in relative humidity leads to a reduction in the COP of an ASHP (Wu et al., 2020; Li et al., 2017; Wang et al., 2018; Zhang, Hihara, and Saikawa, 2012; Underwood, Royapoor, and Sturm, 2017).
- It has been shown that for non-frosting conditions, an increase in relative humidity increases the rate of dropwise condensation. Dropwise condensation, in turn, increases heat transfer under conditions outlined from literature (Danilo, Dominique, and Frédéric, 2016). It can thus be argued that higher relative humidity will increase ASHP performance under these conditions as relatively higher energy heat (opposed to dry-air) is available at lower thermal resistance (Eimann et al., 2020). However, condensation build-up can degrade heat transfer, and consequently, performance (Chung, Kadala, and Taylor, 2020). Thus, from heat transfer literature, and thermodynamics, one can argue that increased relative humidity will generally increase ASHP COP for ambient conditions where condensation occurs.

⁹ P_g refers to the saturation pressure of the vapor in the mixture at the same temperature of the mixture (in accordance with Dalton's law).

¹⁰At constant dry bulb temperature.

- Higher water inlet temperatures degrade COP performance (Guo and Goumba, 2018; Vega and Cuevas, 2018).
- Higher dry bulb temperatures, based on ASHP and heat transfer literature, increase COP performance for frosting, condensing and non-condensing regions of heat transfer (Tabatabaei, Treur, and Waumans, 2016; Underwood, Royapoor, and Sturm, 2017; Guo and Goumba, 2018; Kinab et al., 2010).

By taking the above into account, the study resumed by deriving a performance map based solely on OEM and experimental data. Deriving an expression that will accurately describe ASHP performance in all of the heat transfer conditions outlined above falls out of the scope of the study with the data and experimental setup that was available. The following section will describe the statistical analyses on the proposed regression formula for COP.

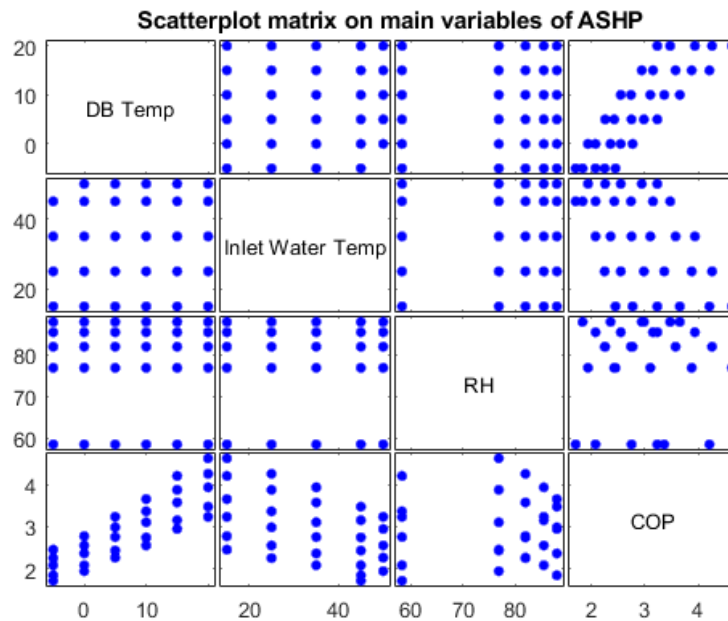


FIGURE 3.11: Scatter-plot matrix on ASHP variables from MATLAB.

With the independent variables identified above, the relationships between variables from manufacturer data will be discussed. The manufacturer tabulated the COP values of the HP at various wet (WB) and dry bulb (DB) temperatures and the corresponding water inlet temperatures (Appendix D). Under the same conditions, the electrical power consumption and the heat energy generated was listed. A scatter-plot on the relation between the different variables on said data is showcased in Figure 3.11. It can be seen that the dry bulb temperature has a somewhat positive linear relation to COP. The inlet water temperature has a negative linear correlation with COP. The relative humidity, however, was not varied enough by the manufacturer to assume its effects on the COP. However, it seems to have a 2nd order effects on the COP. To make an accurate fit, the RH has to be varied over the full extent of the performance space (0 – 100[%]) to avoid extrapolation. Therefore statistically, one cannot with confidence determine a correlation between RH and COP based on the OEM data.

From the analyses above it was assumed that the COP could be expressed as a linear multi-variate equation in the form of:

$$\beta \approx \alpha_0 + (\alpha_1 \times T_{db}) + (\alpha_2 \times T_{inlet}) + (\alpha_3 \times \phi) \quad (3.32)$$

Two regression equations were derived for ASHP performance. These equations predict the COP and electrical energy required by the ASHP, respectively. The statistical analyses on these equations are presented in Appendix H. Dry and inlet temperatures parameters (α_1, α_2) were fitted with OEM data (Appendix D). The relative humidity parameter (α_3) were fitted using infield data as the experimental results have a more substantial variance in RH. From this, a regression equation was derived with an accurate representation of COP over the whole performance space (Figure 3.12) based on the data of the experimental period.

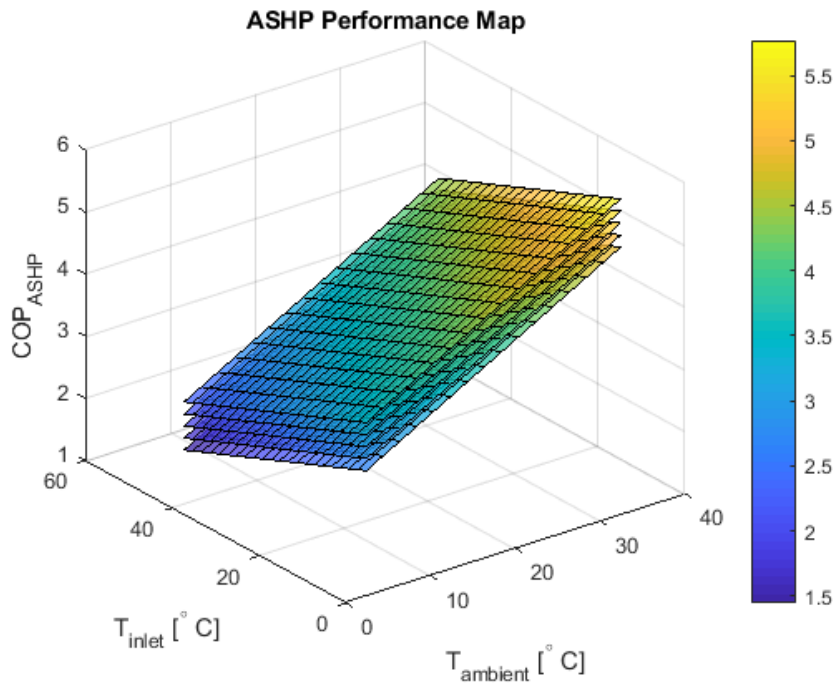


FIGURE 3.12: Regression performance map of ASHP as used in the study. Relative humidity surfaces were plotted for 20, 40, 60, 80 and 100 % respectively with the lowest Relative humidity represented by the uppermost surface.

3.5.2 Solar water heater regression

For the solar water heater, two types of regressions models were employed. The one is for the efficiency of the solar water heater and the other to determine the SWH pump's mass flow.

3.5.2.1 Efficiency curve

From standard solar engineering practice, the efficiency of an SWH is customarily expressed as:

$$\eta_{solar} \approx \alpha_0 \left(\frac{T_{inlet} - T_{amb}}{G} \right)^2 + \alpha_1 \left(\frac{T_{inlet} - T_{amb}}{G} \right) + \alpha_2 \quad (3.33)$$

For this study, the curves that were used is graphically represented in Figures 3.13 and 3.14. These curves demonstrate the efficiency of a single SWH bank used in the system simulation. These graphs cannot be used when more of these banks are added in series as the graph will look fundamentally different. Figure 3.13 illustrates that the SWH at high irradiance is less sensitive to temperature variations between the inlet and ambient conditions. The optical efficiency ($T_{inlet} - T_{ambient} = 0$) can be noted in the centre of Figure 3.13, with all of the irradiance plots intersecting this line. In Figure 3.14, the optical efficiency can be noted on the left-most part of the efficiency graph. From Figure 3.14, a significant drop in SWH efficiency can be noted at low solar irradiance and inlet temperatures that are 20 [°C] higher than ambient temperatures.

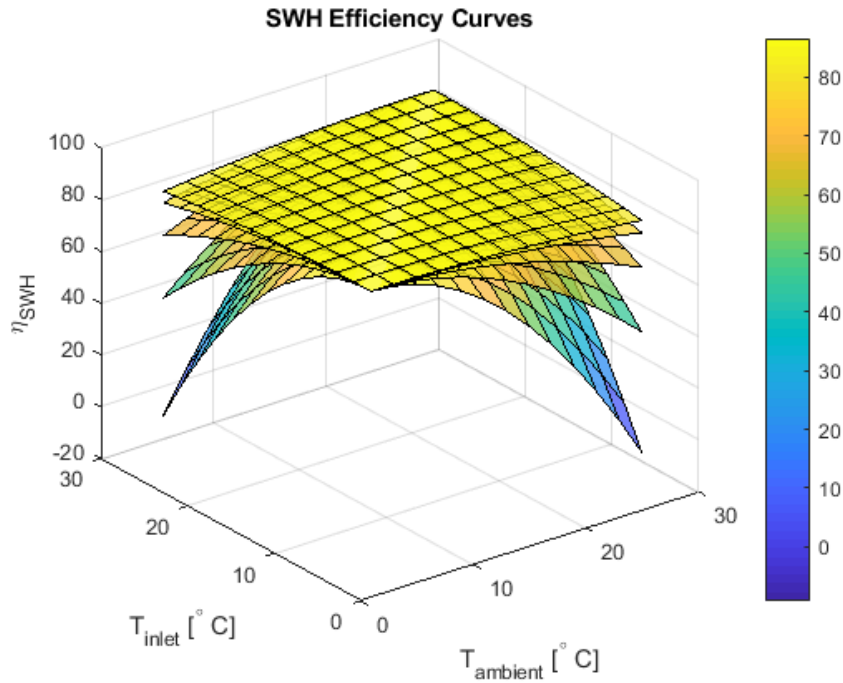


FIGURE 3.13: Regression performance map of SWH as used in the study. Irradiance surfaces were plotted for 250, 350, 500, 750 and 1000 W/m^2 respectively with the highest irradiance represented by the uppermost surface.

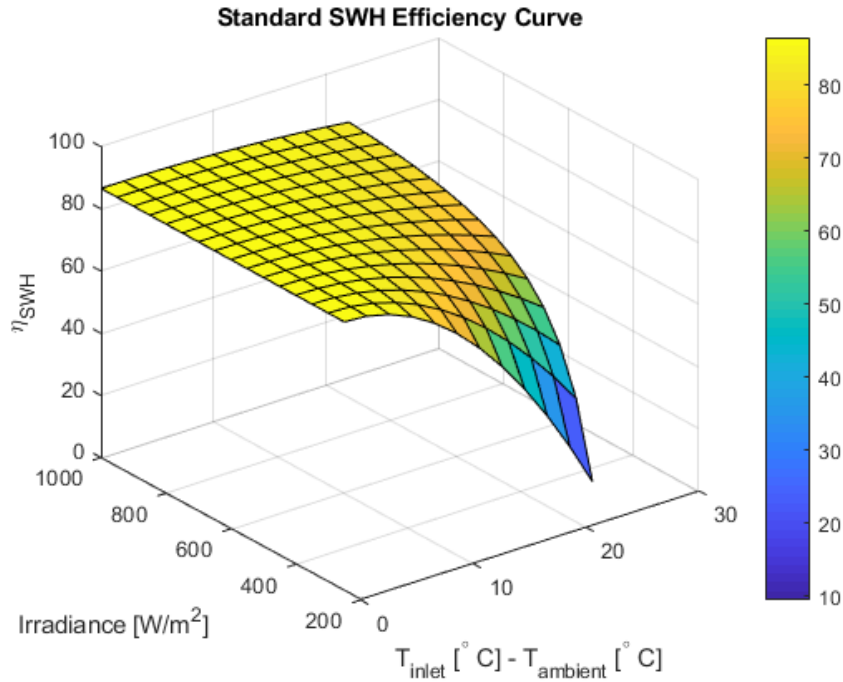


FIGURE 3.14: Regression performance map of SWH as used in the study. The curve illustrates the sensitivity of the SWH efficiency to temperature differentials and irradiance. The optical efficiency can be noted to the left.

3.5.2.2 Pump curve

Due to the limitation that no adjustments were allowed on the system, no flow sensors could be installed on the SWH loop. With this limitation, the pump curves were utilized. A CCG 05-25C pump with a maximum head of 8 [m] and a flow rate of 3.9 [m³/h] is used to circulate water through the SWH loop. Appendix J contains the graphs and information on the pump. The pump has three-speed settings. The OEM, however, only supplies the power curve for the high-speed setting. The hospital technical team had set the pump to a low-speed setting. To obtain the equivalent power curve for the lower speed setting, dimensional analyses was employed. The geometric properties of the pump, along with the fluid properties, remain the same. Incompressible effects are assumed along with the same surface roughness. By using the special case, the following equations can be used to scale between the speed setting on the pump:

$$\frac{Q_{low}}{Q_{high}} = \left(\frac{\omega_{low}}{\omega_{high}}\right) \times \left(\frac{D_{low}}{D_{high}}\right)^2 \quad (3.34)$$

$$\frac{H_{low}}{H_{high}} = \left(\frac{\omega_{low}}{\omega_{high}}\right)^2 \times \left(\frac{D_{low}}{D_{high}}\right)^2 \quad (3.35)$$

$$\frac{P_{high}}{P_{low}} = \left(\frac{\omega_{high}}{\omega_{low}}\right)^3 \times \left(\frac{D_{high}}{D_{low}}\right)^5 \quad (3.36)$$

Using the above, along with the Euler pump equation and the definition of efficiency, the lower energy curve could be obtained:

$$P_{Euler} = \rho \times g \times Q \times H \quad (3.37)$$

$$\eta = \frac{P_{Euler}}{P_{solar,pump}} \quad (3.38)$$

With the low-speed power and efficiency curve known, the theoretical flow rate could be calculated. The second phase voltage of the HP was actively monitored along with a Hobo logger that measures the current on the circulating pump. Finally, a power factor was assumed for the pump. By using the above rationale, the theoretical flow rate could be calculated using the following equations:

$$P_{solar,pump} = I_{solar,pump} \times V_{2nd,phase} \times \cos(\theta) \quad (3.39)$$

$$P_{solar,pump} = f(\dot{Q}_{solar}) \quad (3.40)$$

$$\dot{m}_{solar} = \dot{Q}_{solar} \times \rho \quad (3.41)$$

3.6 Experimental Results and Analyses

This section will focus on the experimental results that were obtained and commentaries on the energy flux distribution of the system. Detailed analyses of the system operation during the test period will be presented near the end of the chapter.

3.6.1 Preliminary energy balance

The formulas to calculate the various energy fluxes was illustrated. This section will showcase the 1st law energy balance with the relative humidity coefficient (α_3) of the ASHP solely based on manufacturer data, as reported in Appendix D and H and discussed in Section 3.5.1.

Experiments were conducted from May to July 2019. A four-day dataset from the 4th - 7th of June was used for analyses and the comparison to the developed simulation as discussed in the following chapter. Following the methodology, as outlined in Section 3.4, the comparative calculations could be conducted. Two methods were used to quantify the 1st law energy balance that was obtained. The net energy rate and the energy flux ratio quantified as follows:

$$\dot{Q}_{nett} = \dot{Q}_{in} - \dot{Q}_{out} \quad (3.42)$$

$$\dot{Q}_{ratio} = \frac{\dot{Q}_{in}}{\dot{Q}_{out}} \quad (3.43)$$

Furthermore, the input and output fluxes as defined in Section 3.4 were expressed as:

$$\dot{Q}_{in} = \dot{Q}_{ASHP} + \dot{Q}_{Solar} \quad (3.44)$$

$$\dot{Q}_{out} = \dot{Q}_{Ringmain-steady-state} + \dot{Q}_{User} + \dot{Q}_{Storage-tank-losses} + \dot{Q}_{System-pipe-losses} + \dot{Q}_{SWH-losses} \quad (3.45)$$

Preliminary results from Table 3.15 clearly shows that the nett energy flux ($\dot{Q}_{nett}$) is not equal to zero. By the energy flux ratio, one can conclude that the energy that enters the system appears to be more than the total energy expenditure. Referring to Table 3.8, the various measured, calculated and assumed variables of the energy fluxes can be noted. The relative errors of the

sensors are presented in various tables throughout this chapter and Appendix A. The former and latter were taken into account to determine which energy flux was most probably responsible for the violation of the energy balance.

TABLE 3.15: Hourly energy flux values for the test period, prior to developing the coefficient from experimental data.

	03/06/2019	04/06/2019	05/06/2019	06/06/2019
$Q_{ASHP}[\text{W}]$	24 978.5	22 063.5	22 496.2	24 546.9
$Q_{Solar}[\text{W}]$	808.8	715.2	689.3	526.4
$Q_{Pipes-Tank}[\text{W}]$	784.8	746.8	572.0	822.1
$Q_{Pipes-Solar}[\text{W}]$	906.2	697.0	750.7	668.7
$Q_{Losses-Tank}[\text{W}]$	3 487.7	3 318.8	3 336.4	3 653.6
$Q_{User-Load}[\text{W}]$	6 157.9	7 560.9	6 915.1	5 340.7
$Q_{Ringmain}[\text{W}]$	7 414.6	7 167.2	7 229.4	7 699.6
$Q_{Nett}[\text{W}]$	7 036.1	3 287.9	4 381.9	6 888.6
$\frac{Q_{In}}{Q_{Out}}[dim]$	1.38	1.17	1.23	1.39

It was concluded that the energy fluxes based on regression equations would be the most probable candidates. The ASHP is the single largest energy contributor. Therefore this specific regression was revisited. The ASHP regression was compared to various input conditions to try and determine the large unaccounted energy surplus. Considering the above-mentioned, along with the results reported in Figure 3.15, one can assume that there might exist a correlation between the relative humidity and the energy flux ratio of the system. Indicating the relative humidity (α_3) derived from the manufacturer data might be to blame. Therefore the regression was altered to be based on the combination of OEM and in-field data. Figure 3.16 illustrates the energy balance when the relative humidity is based on in-field data. The following section will discuss the energy balance results obtained based on the hybrid regression equation ¹¹.

¹¹Hybrid in this context, describing the equation using data obtained from the OEM and in-field data to obtain the regression coefficients.

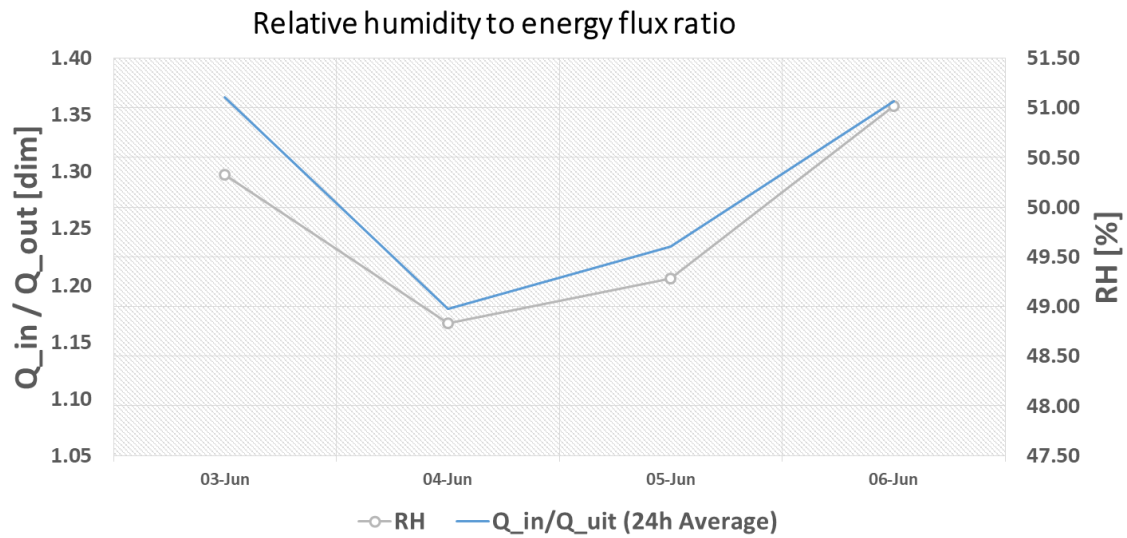


FIGURE 3.15: Relative humidity to energy flux ratio without using in-field data on the relative humidity coefficient.

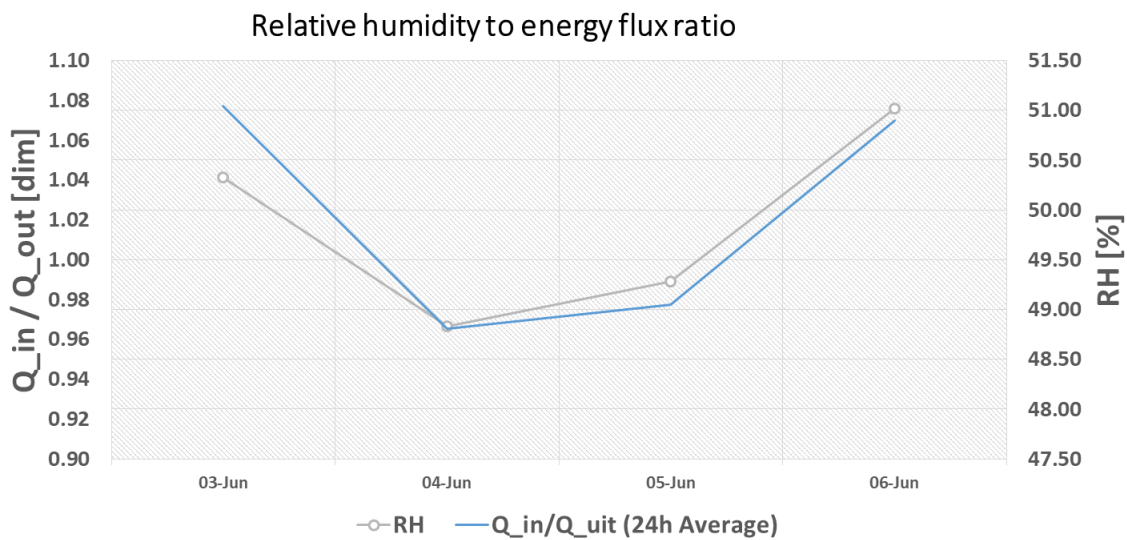


FIGURE 3.16: Relative humidity to energy flux ratio by using in-field data for the relative humidity coefficient.

3.6.2 Holistic system discussion

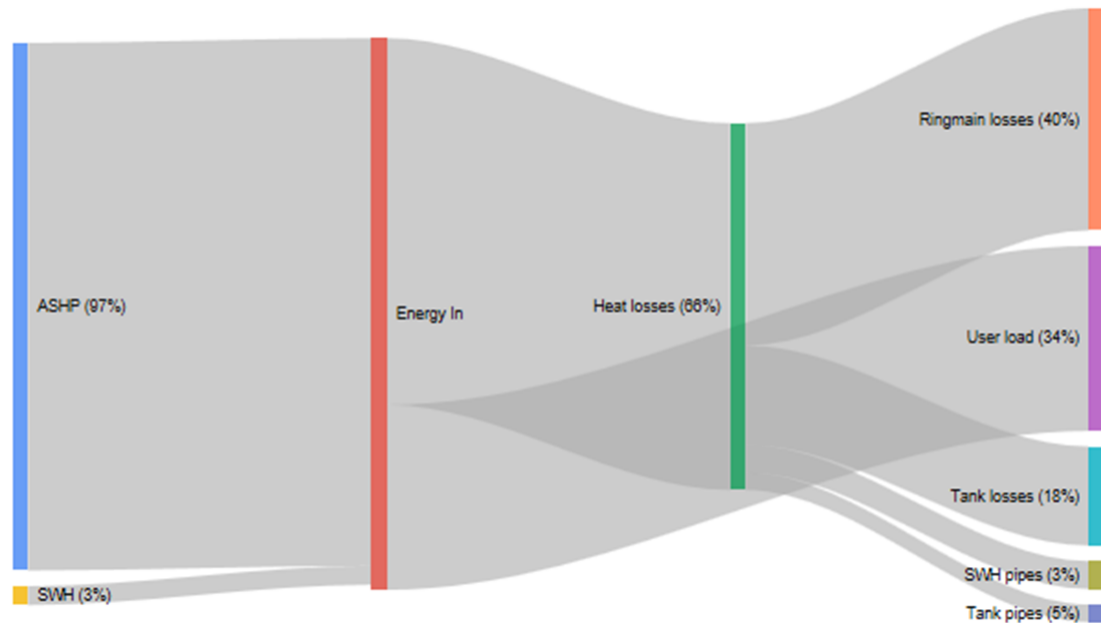


FIGURE 3.17: Sankey diagram illustrating energy flux distribution.

From Figure 3.17, the Sankey-diagram for the system is illustrated. Focusing on system energy input, the ASHP contributes significant amounts of energy (97 [%]) to the system compared to that of the SWH (3 [%]). The energy distribution can be ascribed to the control systems employed by the heat pump and the solar water heater system, and the number of SWH banks. Apart from the control system, a simple comparison validates the analyses. The heat pump is rated at 87.4 [kW]¹², with the solar banks peak energy delivery at 5 [kW]. Thus it can quickly be concluded that apart from the useable daylight hours, the SWH contribution would be small in comparison to that of the ASHP. With the ring-main extracting energy mainly in the form of heat losses perpetually, the ASHP will have to supply the system with energy during night-time operation.

By focusing on energy expenditure, the most striking figure is that heat losses account for nearly two-thirds of all energy input. One has to perceive the distribution of these losses to conclude if the system has been over-designed to the user requirements or not. It can be seen that the ring-main contributes to a significant energy expense at nearly 40 [%], followed by the user load at 34 [%]. One can presume that the ring-main is an inevitable heat loss due to the high temperature and relatively high mass flow supplied to the pipe network within the hospital building. There are numerous stagnant bodies of water in the pipe connections leading to the draw-off points throughout the hospital. Of particular note is that the end-user uses only a third of total energy expenditure.

The losses in the storage vessels can be attributed to the relatively high temperatures of the thermal mass that are maintained in these vessels. The vessels serve as a buffer when the

¹²One can note that by even using an average COP of three and taking note of the solar energy contributed during the useable daylight hours, that the SWH contribution is small in comparison to that of the ASHP.

TABLE 3.16: Hourly energy flux values for the test period.

	03/06/2019	04/06/2019	05/06/2019	06/06/2019
$Q_{ASHP}[W]$	19 595.3	17 934.9	17 741.8	19 246.2
$Q_{Solar}[W]$	808.8	715.2	689.3	526.43
$Q_{Pipes-Tank}[W]$	784.8	746.8	572.0	822.1
$Q_{Pipes-Solar}[W]$	906.2	697.0	750.7	668.7
$Q_{Losses-Tank}[W]$	3 487.7	3 318.8	3 336.4	3 653.6
$Q_{User-Load}[W]$	6 157.9	7 560.9	6 915.1	5 34.7
$Q_{Ringmain}[W]$	7 414.6	7 167.2	7 229.4	7 699.6
$Q_{Nett}[W]$	1 765.4	51.0	152.6	1 685.2
$\frac{Q_{in}}{Q_{out}}[dim]$	1.09	1.00	0.99	1.09

heat pumps are not able to meet user demand during periods of peak demand. The ring-main distributes roughly two times the total storage vessel volume per day (30 [kL]). Daily user consumption is, on average, 3 [kL]. Taking the former and latter into account can conclude that the system has too many storage vessels for the current user load (total capacity of 15 [kL]). To support this comment, one can consider a situation where total daily energy consumption is extracted in a single hour ($0.83[L s^{-1}]$). Even with the ASHP working at a fraction of its maximum flow rate ($0.55[L s^{-1}]$), the total volume of the second coldest storage vessel will not be displaced to that of the warmest vessel. System pipe losses between the ASHP, storage vessels and ring-main are comparatively small. The length of the pipes in the hospital is noticeably longer than those between the main components, explaining the smaller heat loss.

Furthermore, considering the mass in the pipes relative to that of the storage vessels, the heat loss in the pipes appear to be large. This is the case due to the fact that the surface area to volume ratio of the storage vessels is lower than that of the pipes. Naturally, the heat transfer coefficient and flow regime in the pipes also plays a role in the comparatively higher heat losses.

The SWH is of particular concern as the losses in the pipe network roughly equates to the energy addition of the SWH. The high heat losses can be explained due to the high-temperature build up in the solar pipes as well as the active hours of energy addition. Furthermore, the SWH pipes contribute to heat loss when the daylight hours have passed. It can be reasoned that the component currently serves a load-shift purpose as when viewed in isolation, the former has roughly a net-zero energy benefaction to the system.

In Table 3.16, the daily averaged energy flux values are tabulated. Noticeable is that the average useful energy flux of the ASHP (Q_{ASHP}) is roughly 19 [kW]. Referring to Table 3.4, the maximum power input and heat energy are reported as 28.2 [kW] and 87.4 [kW], respectively (corresponding to a COP of roughly 3). Thus it can be concluded that even under low COP conditions, the HP system is underutilised in this application. One must also remember that these tests were conducted during the winter period where an ASHP have to work relatively hard in comparison to summer conditions due to lower dry bulb temperatures.

The energy flux ratio is small enough to assume that energy conservation is maintained. The average net energy transfer that is unaccounted for varies from roughly 51 [W] to 1765 [W]. By taking the methodology and errors of the various sensors into account (Appendix 3.5.2.2), this deficit can be assumed to be relatively small to that of the maximum energy flux. An example of the calculation regarding the energy fluxes with daily averaged inputs is demonstrated in Appendix C. The example uses the daily averages for all of the inputs. For the simulations and

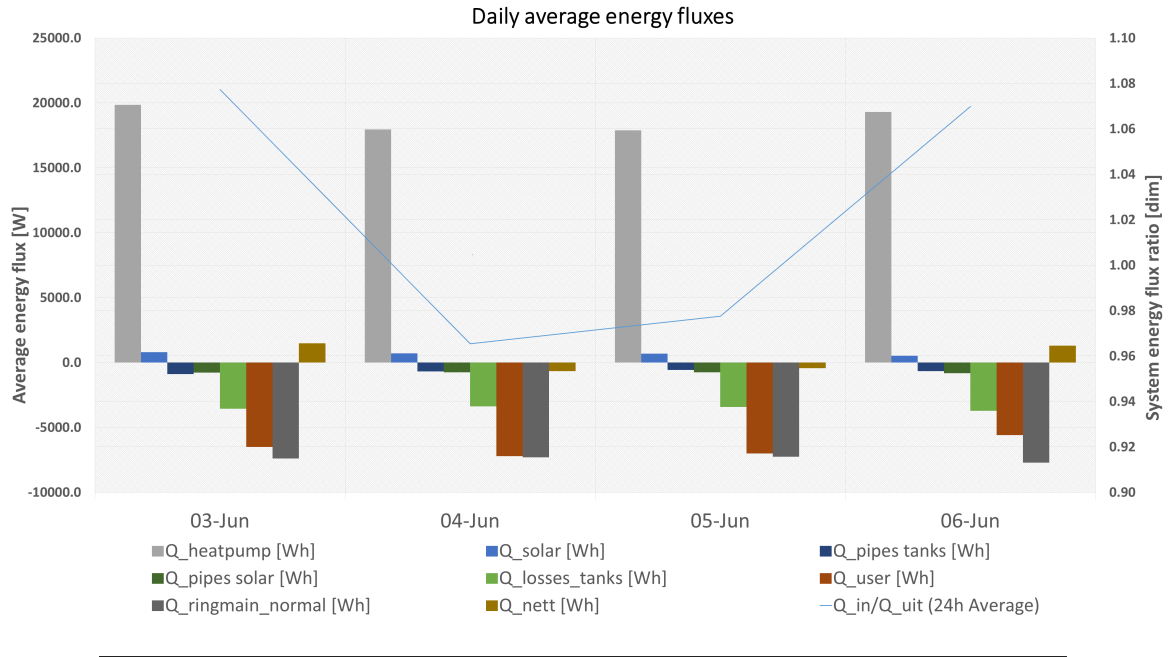


FIGURE 3.18: 1st Law Energy balance on system.

comparison with the experimental results, 15 [min] and 1 [min] dataset time intervals are used as per the methodology explained in the previous sections.

The data of Table 3.16 is graphically represented in Figure 3.18. The waterfall graph distinguishes the highest energy contributions and losses in the system for each of the four days studied. The energy flux ratio was also overlayed to illustrate the energy balance error. It should be noted that this energy balance was calculated with the finer non-averaged datasets obtained directly from the SCADA and HOBO sensors. In contrast, the example in Appendix C uses daily averages with simplifying assumptions.

3.6.3 Detailed discussion

The previous sections made commentaries and discussed the sizes of the various energy fluxes. The following section will aim to analyse the distribution of these energy fluxes in depth to draw correlations.

Figure 3.19 compiles and summarises the experimental data that was obtained. Most of the commentaries in the following paragraphs centre around the previous mentioned Figure 3.19. It can be seen that the supply temperature was sustained at approximately 60 [°C] during the test period. The measured temperature helped to illustrate that the design temperature of the system can be maintained even in the cold winter periods. Furthermore, the net energy flux indicates the constant 'charge' and 'discharge' of the system storage vessels.

Of particular note is that the ASHP was operational throughout most of the early mornings past midnight. Taking note of the net energy flux of the system, the ASHP seems to charge the cycle throughout the first half of the morning cycle. The remaining energy is extracted throughout the morning, and the ASHP runs relatively laborious to maintain these temperatures before about 9:00 [AM]. This statement is made since although the ASHP was active, the net energy

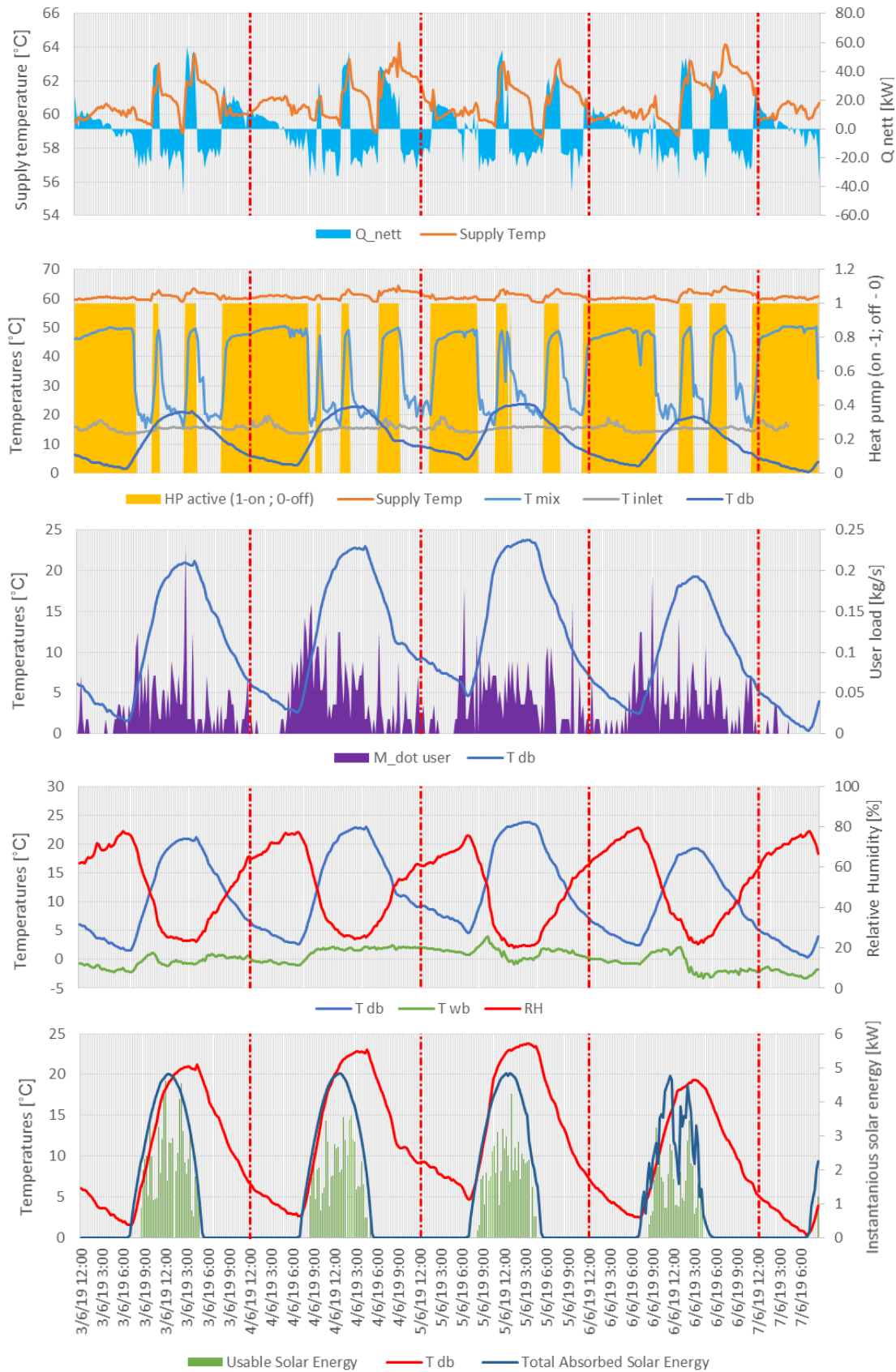


FIGURE 3.19: Experimental test period analysis graphs.

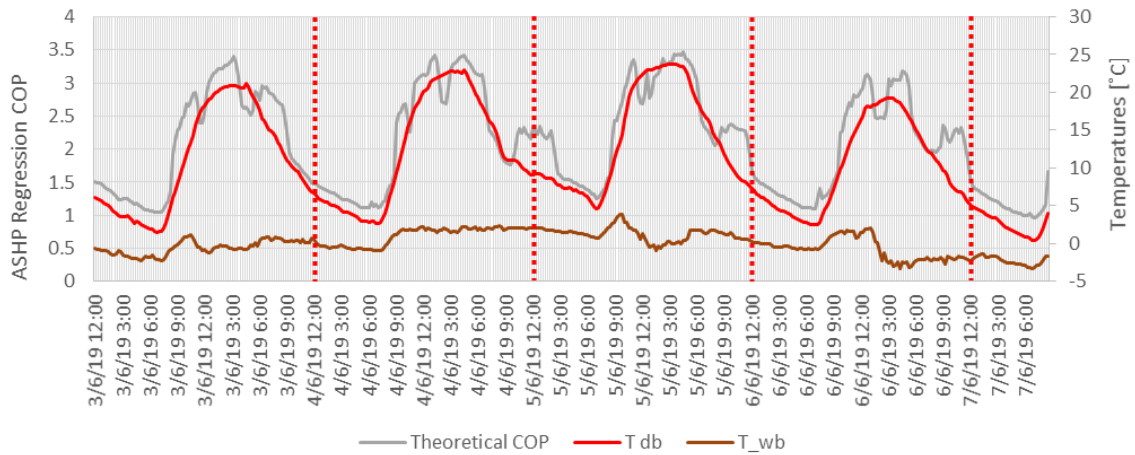


FIGURE 3.20: ASHP performance prediction with measured environmental conditions.

flux was negative near the end of the operational periods. Moreover, the heat pump seems to be active for lengthier periods as opposed to the rest of the day.

From Figure 3.20, the predicted COP was calculated from the environmental conditions. During early morning periods, the RH is at a peak, and the DB temperatures are at a minimum. It should be noted that for a dry bulb temperature of 5 [°C] and relative humidity of 80 [%], the dew point temperature is roughly 1.8 [°C]. The surface temperature of the evaporator is usually 5 - 10 [°C] below dry bulb temperatures. Thus, frost formation is possible in these early morning periods. These environmental conditions limit the amount of energy the ASHP can extract, and therefore, low COP values were consequently achieved. Additionally, from Figure 3.19, it can be noted that the ASHP was mostly active during these unfavourable conditions. Promoting the basis of the previous chapter that relative humidity can impact ASHP performance, as observed from the experimental results on the system.

It can moreover be seen that the ASHP is switched on about two times during the daytime hours. In the early mornings and at an evening round. During all of these daytime running periods, the ASHP was able to convert the net energy flux to a charge cycle initially. The time that the ASHP was operational during these daytime hours was reputedly shorter than in the morning period.

User demand seems to follow no apparent profile during the period in which the system was monitored. Notable peaks in user demand seemed to be occurring in the 6:00 - 9:00 [AM] morning periods. Particularly during the period in which the ASHP is active, but the net energy flux indicates a 'discharge' state predominantly.

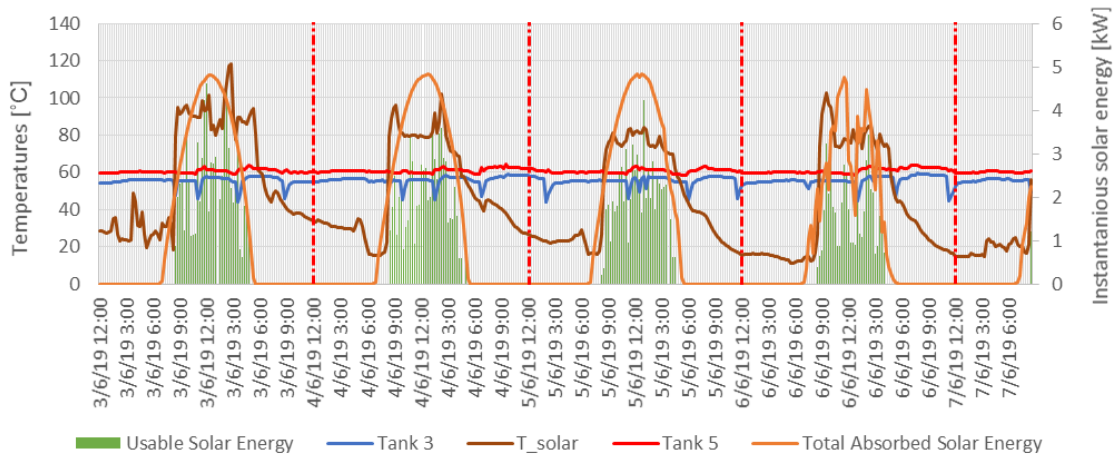


FIGURE 3.21: SWH energy flux and temperature data.

Figure 3.21 illustrates the middle- (T_3), top storage vessel (T_5) and SWH outlet temperatures. Additionally, the instantaneous usable and total absorbed solar irradiance was overlaid to understand the underperformance of the SWH better. Although absorbed instantaneous solar flux can peak at nearly 5 [kW], the usable energy was much lower.

The control philosophy exercised by the SWH and the flow rate through the SWH system was mostly responsible. The required temperature differential by the control system is 12 [°C] higher than the middle storage vessel temperature (T_3). Figure 3.21 illustrates data obtained from the SCADA system. It can be seen that the solar temperatures in the manifold during daylight hours, is on average, close to 80 [°C]. Furthermore, temperatures were measured as high as 120 [°C]. These figures lead to three possibilities explaining the underutilisation: The first is that the SWH pump is not removing the energy build-up at a high enough rate. The second is that the differential temperature was even higher than 12 [°C]. Finally, the maximum temperature probe was set too low. The SWH control deactivates if the warmest tank temperature is at the maximum setpoint. The ASHP maintains the temperature of the warmest tank relatively well at this temperature (as it is the same as the maximum rating of the ASHP).

Consequently, even though the 12 [°C] set point is attained, with the warmest vessel average temperature at 61.5 [°C], the SWH will not activate. Thus a combination of the flow rate and maximum temperature set point leads to the irregular addition of solar energy. Apart from this, one can note that the energy is accumulated during the day declines steadily throughout the night leading to high heat losses. This can be noted as the available energy in the SWH manifold is not extracted before daylight hours pass.

One must, however, take into account the active hours of the SWH as its' contribution is averaged over the 24 [h] period. During observation of the system, it was noted that the SWH pumps did not switch on for periods longer than 60 [s]. The former can be motivated due to the control system, mass flow through the SWH, the 4.8 [kW]¹³ peak energy and the size of the buffer vessels (15[kL]). Furthermore, there can be seen that cloud cover limited the amount of useful solar energy on the last day of the experimental period.

¹³As a simple calculation verification: assume 6 hours of sunlight. Assume during these hours; the SWH works continuously at 2.8 [kW]. Daily energy contribution is thus $2.4[\text{kW}] \times 6[\text{h}] = 14.4[\text{kWh}]$. By averaging the contribution over the 24 hour period a 600 [W] equivalent energy contribution is obtained.

One can comment that the system energy utilisation can be increased by lowering the maximum and differential temperatures on the SWH. The flow rate through the SWH can be increased to aid the removal of energy in the SWH. A third possibility is to lower the differential temperatures and lower the flow rate so that the SWH works continuously without erratic temperature build-up and discharge. Additionally, more SWH banks can be added in parallel to increase SWH contribution. Furthermore, if the system was redesigned from scratch, the number of storage vessels could be lowered to three. This would enable the system to hold sufficient thermal capacitance buffer as well as lower the heat losses and the load on the ASHP to maintain the storage vessels at the setpoint temperature.

The next chapter will deal with the development of the detailed and reduced computer model as well as the verification of the developed simulation model.

Chapter 4

Simulation development

This chapter intends to showcase the fundamental functioning of the developed simulations and the comparison to the experimental data of the previous Chapter 3. This study produced three simulation models. One of which attempts to use reducing measures to streamline computational effort without compromising accuracy. The remaining simulations use comparatively few additional constraints that lead to significantly longer simulation times. The simulations are as follows:

- Runge-Kutta method of lines solver (referred to in this study as the '*reduced model*' or '*RK-MOL*')
- Euler model (referred to as '*detailed model*' or '*Euler model*')
- MATLAB® Simulink® model

Both the detailed and reduced models were programmed in EES®. Using the parametric table feature of EES®, dynamic simulations can be developed. The chapter is divided into the following sections that intend to portray the development of the simulation:

- Fundamentals (4.1) - The fundamental equations that govern the simulation programs are presented along with the calculation of the simulation originated energy fluxes.
- Algorithm (4.2) - The approach that was used to deploy the fundamental equations introduced in Section 4.1 is addressed.
- Verification (4.3) - The numerical tests for convergence, stability and consistency is showcased, and a comparison to a MATLAB® Simulink® model is presented.
- Validation (4.4) - Comparison of the simulation's 1st law energy results to that of the experimental data.
- Conclusions (4.5) - Concluding remarks is given on the simulations accuracies, solver times, restrictions and possible applications.

4.1 Fundamental conservation equations

The objective of the simulations is to predict the evolving internal energy of the system (as presented in Chapter 3.2) and its' components under varying boundary conditions. The simulations, therefore, are dynamic as the boundary conditions are dynamic. From these predictions, a 1st law energy balance can be conducted for the system corresponding to the fluxes as presented in Chapter 3. The rate of energy exchange can be calculated for each time-step, and the

average of these fluxes would represent the values that can be compared to the experimental data. Throughout this chapter, unless otherwise stated, *the system* will involve the fixed control volume as presented in Figure 4.1. The control volume was chosen as such to exclude the SWH and ASHP; as these components' internal conditions are represented by regression equations.

The section was divided into the following subsections:

- 1st law energy balance (4.1.1) - A system-level¹ discussion on the methodology used by the simulations to calculate the internal energy evolution on the system.
- Sub-components (4.1.2) - On a more comprehensive level, the sub-component models that the simulation comprises of are discussed that are required for the 1st law energy balance.

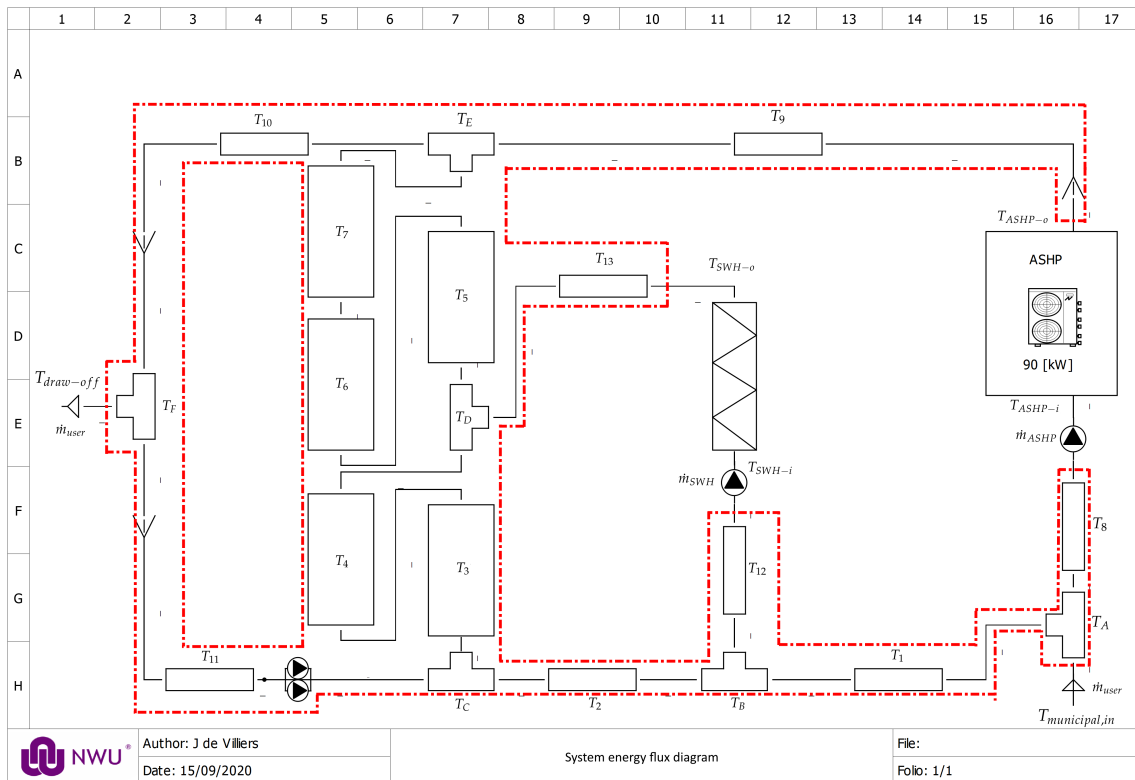


FIGURE 4.1: Total system control volume.

4.1.1 1st law energy balance

Consider a stationary, fixed, non deforming control volume (dV) that encompasses the fluid of the system throughout the various pipes and components (Figure 4.1). An incompressible fluid is observed, and the system is pressurised; thus the fluid mass in the control volume remains constant. Furthermore, assume this control volume to not include the ASHP and SWH but to only begin and end at the outlet and inlet of these respective components. The change in system energy from an initial ($U_{initial}$), to a final energy state (U_{final}) after a fixed lapse of time (M), can be expressed as:

¹System-level in this context entails the fixed control volume that has been chosen for the 1st law energy balance as presented in Figure 4.1.

$$U_{initial} + Q_{in} + W_{in} = Q_{out} + W_{out} + U_{final} \quad (4.1)$$

Equation 4.1 only asserts that energy is added ($Q_{in} + W_{in}$) or lost ($Q_{out} + W_{out}$) to reach the final energy state. Mechanical work (W) is path dependant, and heat energy (Q) is therefore, also path dependant². The path dependence of these terms is important to quantify the transition from the initial energy state ($U_{initial}$), to the final energy state (U_{final}) as defined in Equation 4.1. Therefore, the rate of energy addition or expenditure has to be evaluated at multiple time-steps if the process is to be predicted accurately.

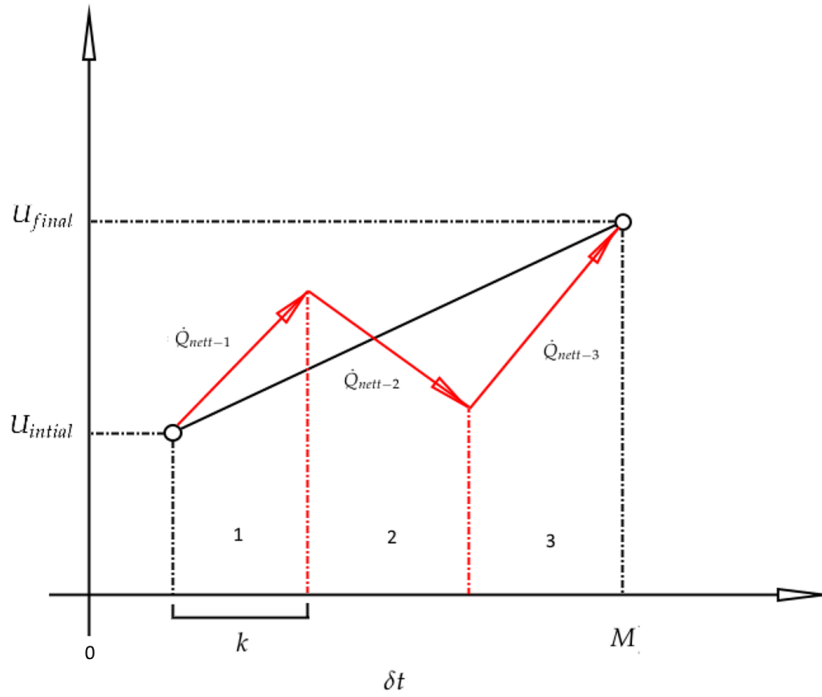


FIGURE 4.2: System energy state evolution example.

As an example, observe Figure 4.2. The graph illustrates internal energy to simulation time for a simple example of the system. Assume that due to the storage vessel temperatures being too low, the ASHP activates in the first time-step. Furthermore, the ASHP is able to convert the net system energy flux to a positive value ($\dot{Q}_{net-1} > 0$). Furthermore, assume that in the following time-step, the hospital consumes a vast amount of hot water, where the ASHP struggles to maintain the net energy flux as positive, leading to a negative net energy flux ($\dot{Q}_{net-2} < 0$). However, as the user drains hot water from the storage vessels, the ASHP have favourable COP conditions due to the lower inlet temperatures to the ASHP. In the last time step, the user load is zero, and the ASHP can change the net energy flux positive with a higher gradient ($\dot{Q}_{net-3} > \dot{Q}_{net-1} > 0$). The higher gradient is due to colder inlet water and the energy expenditure of the system that has been reduced.

²According to the principle of the 'mechanical equivalent of heat' ($q = \Delta U + w$).

In the scenario explained above, the gradient of the ASHP energy flux was a function of the controller, ambient conditions, and internal energy state. With the output of the controller determined by the internal energy state at the beginning of the time-step. Therefore, to simulate the system and its' control systems effectively, a suitable time-step has to be used. The controller will have a direct effect on the path taken by the heat energy added and removed from the system control volume.

It can be noted that the average energy gradient of the three time-steps ($\dot{Q}_{net-1}$, $\dot{Q}_{net-2}$, $\dot{Q}_{net-3}$) are the same as that of the averaged energy gradient ($\frac{U_{final}-U_{initial}}{M}$) between the initial ($U_{initial}$) and final (U_{final}) energy conditions. There are infinite scenarios to transition from the initial to the final energy state. However, the boundary conditions, fixed input parameters of the various components and the characteristics of the controllers, ensure that only a subset of these infinite amounts of solutions is plausible.

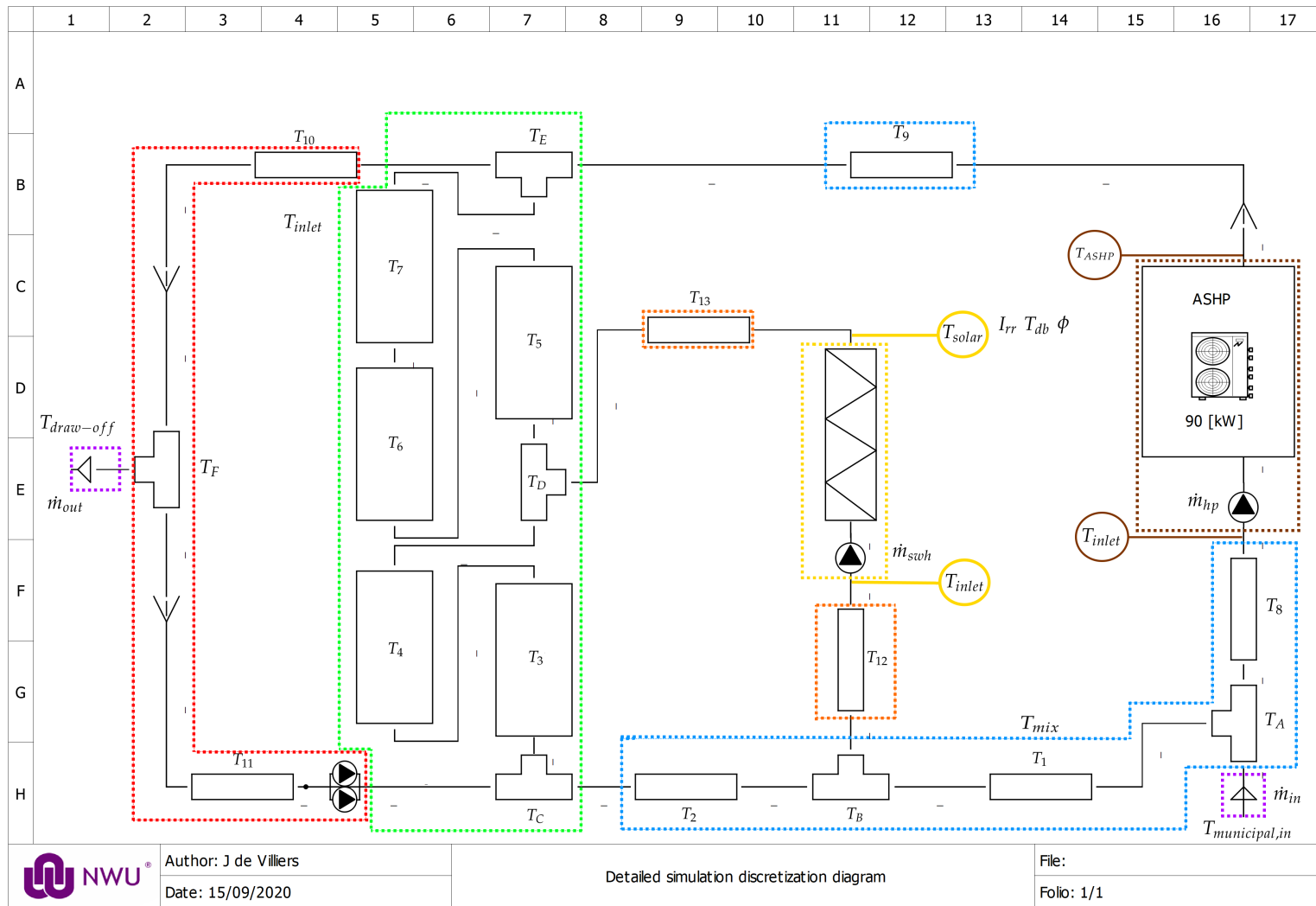


FIGURE 4.3: Discretization diagram and sub component control volumes demarcation.

From the rationale above, an equation is required that links the energy accumulation in the control volume, energy entering the control volume and change in the energy state of coincident control volumes. Consequently, Reynolds theorem is used and can be written as:

$$\frac{D}{Dt} \int e \rho d\forall_{cv} = \frac{\partial}{\partial t} \int e \rho d\forall_{cv} + \int e \rho \mathbf{V} \cdot \mathbf{n} dA_{cs} \quad (4.2)$$

From the first law of Thermodynamics, energy can only change form and have to be conserved. Equation 4.2 states that the rate of energy exchange to and from the coincident control volumes ($\frac{D}{Dt} \int e \rho d\forall_{cv}$)³, must be equal to the time rate of change of the stored content of the control volume ($\frac{\partial}{\partial t} \int e \rho d\forall_{cv}$), and the net rate at which energy leaves and enters the control volume by mass fluxes through its' surfaces ($\int e \rho \mathbf{V} \cdot \mathbf{n} dA_{cs}$). The final term in Equation 4.2, refers to the energy fluxes crossing the control volume boundaries. The net energy flux can be expressed as the energy ⁴ times the average mass flow crossing the control volume surfaces:

$$\int e \mathbf{V} \cdot \mathbf{n} dA_{cs} = \sum \dot{m}_{out} e_{out} - \sum \dot{m}_{in} e_{in} \quad (4.3)$$

The advection flow ($\int e \mathbf{V} \cdot \mathbf{n} dA_{cs}$), into and out of the system can be divided into the following three fluxes:

- ASHP flux: $\dot{Q}_{ASHP} = \dot{m}_{ASHP} C_p (T_{ASHP-o} - T_{ASHP-i}) > 0$ (diagram positions: **B16** and **F16**, Figure 4.1 and Figure 4.3)
- SWH flux: $\dot{Q}_{SWH} = \dot{m}_{SWH} C_p (T_{SWH-o} - T_{SWH-i}) > 0$ (**C10** and **F11**, Figure 4.1 and Figure 4.3)
- User flux: $\dot{Q}_{user} = \dot{m}_{user} C_p (T_{municipal,in} - T_{draw-off}) < 0$ (**E1** and **H16**, Figure 4.1 and Figure 4.3)

The SWH and ASHP are leading to an increase in the net system energy rate and the user load leading to a decrease in the net system energy rate. The $\frac{D}{Dt} \int e \rho d\forall_{cv}$ term of Equation 4.2 refers to energy additions taking place by method of heat transfer or work done by the system or on the system. The term can be expressed as:

$$\frac{D}{Dt} \int e d\forall_{cv} = (\sum \dot{Q}_{in} - \sum \dot{Q}_{out})_{cv} + (\sum \dot{W}_{in} - \sum \dot{W}_{out})_{cv} \quad (4.4)$$

However, the pump work done on the system is assumed negligible ($\sum \dot{W}_{in} - \sum \dot{W}_{out})_{cv} \approx 0$). Energy is therefore mainly transferred through heat losses to the environment:

$$(\sum \dot{Q}_{in} - \sum \dot{Q}_{out})_{cv} = \sum \dot{Q}_{convective} \quad (4.5)$$

The system control volume demarked in Figure 4.1 is subdivided into smaller control volumes (marked out in Figure 4.3) for the remaining fluxes due to the variation of internal temperatures

³The rate at which energy leaves or enters the coincident control volumes must be the same as the energy leaving or entering the studied control volume through its control surfaces ($\dot{Q}_{net,in} + \dot{W}_{net,in})_{cv} = (\dot{Q}_{net,in} + \dot{W}_{net,in})_{coincident,cv} = \frac{D}{Dt} \int e \rho d\forall_{cv}$.

⁴Elevation energy is ignored in the reduced and detailed simulations and it was consequently assumed that the energy ($e = u + Pv + V^2/2 + gz$) can be approximated by the total enthalpy ($e \approx h + V^2/2 = h_0$).

throughout the system. The diffusion ($\frac{D}{Dt} \int e dV_{cv}$), out of the control volumes can be divided into the following four fluxes:

- Ring-main flux: $\dot{Q}_{ring-main} = UA_{ring-main} (T_{draw-off} - T_{db}) < 0$ (E2, Figure 4.3)
- Storage vessel flux: $\dot{Q}_{tank-i} = UA_{tanks} (T_{tank} - T_{db}) < 0$ (E6, Figure 4.3)
- SWH pipe losses: $\dot{Q}_{SWH-pipes} = UA_{SWH-pipes} (T_{SWH-pipes} - T_{db}) < 0$ (E11, Figure 4.3)
- Storage vessel pipe losses: $\dot{Q}_{pipe-losses} = UA_{pipe-losses} (T_{pipes-tank} - T_{db}) < 0$ (E6 and B16, Figure 4.3)

With a finite, fixed time-step k , then the total energy transferred to the system control volume in the j -th time step is given by:

$$Q_{in-j} = (\dot{Q}_{ASHP} + \dot{Q}_{SWH})_j \times k = \dot{Q}_{in-j} \times k \quad (4.6)$$

The total energy leaving the system control volume in the j -th time step is defined as:

$$Q_{out-j} = (\dot{Q}_{ringmain} + \dot{Q}_{tank} + \dot{Q}_{SWH-pipes} + \dot{Q}_{pipe-losses} + \dot{Q}_{user})_j \times k = \dot{Q}_{out-j} \times k \quad (4.7)$$

The change in the system energy state in the j -th time-step can be calculated by summing all of the energy transferred to and from the system control volume in the j -th time-step:

$$U_{final} = Q_{out-j} - Q_{in-j} + U_{initial} \quad (4.8)$$

Equations 4.6, 4.7 and 4.8 is repeated for each interval until all of the time-steps have been calculated sequentially. Thus the process is repeated until the termination time, M :

$$U_{j+1} = Q_{out-j} - Q_{in-j} + U_j \quad \forall j \in \{0..M\} \quad (4.9)$$

The process can be summarised in the following steps:

- Determine from boundary conditions and the control system of the various components the rate of energy leaving and entering each system component in a given time-step.
- Integrate the energy fluxes over the time-step to obtain the following energy state of the system and components.
- Repeat the process until the termination time-step has been reached.
- For analyses, the average of the rate energy fluxes can be used to obtain a 'steady-state' (1st law energy balance) equivalent of the system for the period that was simulated.

With the calculation of the change in energy-state demonstrated in the previous paragraphs, the calculation of the energy fluxes will be showcased in the following Section 4.1.2.

4.1.2 Simulation components

Figure 4.3 illustrates the system with energy boundaries drawn around the various components corresponding to the energy fluxes presented in the previous paragraph. The section

will showcase the equations used to simulate the energy fluxes and the difference between the reduced and detailed simulation methodology.

4.1.2.1 Air source heat pump flux

Before the energy calculation of the ASHP could be conducted, the control system has to determine if the ASHP has to be switched on or not. It is determined if the internal temperature of the second coldest storage vessel (T_4 , F5 Figure 4.3) is colder than the activation temperature ($T_{activate}$). After the relation have been established, the system/program waits for 15-minutes (simulation time) before activating the ASHP. However, the simulation would have to determine the amount of energy to transfer to the fluid due to the variable control philosophy used by the real-world system. Thus part-time ASHP operation is taken into account in this study.

For the ASHP, the maximum available energy that can be extracted from the environment has to be calculated. The COP regression equation (Equation 3.32) presented in Section 3.5.1 is used in the calculation along with the ASHP electrical consumption approximated by Equation H.8 (presented in Appendix H.8). The maximum useful thermal energy of the compressor can be calculated by:

$$\dot{Q}_{ASHP-available} = \bar{\beta} \times \dot{P}_{electrical} \quad (4.10)$$

Where the regression equations for the COP ($\bar{\beta}$) and electrical consumption ($\dot{P}_{electrical}$) are given by:

$$\bar{\beta} = \alpha_0 + (\alpha_1 \times T_{db}) + (\alpha_2 \times T_{inlet}) + (\alpha_3 \times \phi) \quad (4.11)$$

$$\dot{P}_{electrical} = b_1 + (b_2 \times T_{inlet}^2) + \cos(b_3 + T_{db}) + \cos(b_4 + T_{wb}) \quad (4.12)$$

The environmental conditions influencing the performance of the ASHP is the relative humidity (ϕ), dry - (T_{db}), wet bulb (T_{wb}) and ASHP inlet temperatures (T_{inlet}) as outlined in Section 3.5. For the ASHP control volume, the heat losses to the environment are neglected. Due to the relatively small control volume of the ASHP; it is assumed that the thermal capacitance can also be neglected ($\frac{\partial}{\partial t} \int e \rho dV_{cv} \approx 0$). Due to the high rate of energy transfer to the ASHP control volume; the temperature difference is assumed to be nearly instant. From the maximum available energy ($\dot{Q}_{ASHP-available}$), the mass flow is calculated that would deliver the specified set-point temperature ($T_{ASHP-setpoint}$)⁵.

$$\dot{m}_{ASHP-predict} = \frac{\dot{Q}_{ASHP-available}}{Cp \times (T_{ASHP-setpoint} - T_{inlet})} \quad (4.13)$$

$$\dot{m}_{ASHP} = \dot{m}_{ASHP-predict}$$

Therefore, from Equations 4.10 and 4.13, the required ASHP flux ($\dot{Q}_{ASHP}$) is given by:

$$\dot{Q}_{ASHP} = \dot{m}_{ASHP} Cp (T_{ASHP-setpoint} - T_{inlet}) \quad (4.14)$$

⁵With the set-point temperature at 60 [°C] as introduced in Section 3.2.

However, if the predicted ASHP mass flow ($\dot{m}_{ASHP-predict}$) from Equation 4.13 is greater than the maximum delivery mass flow specified by the OEM ($\dot{m}_{ASHP-max}$)⁶; then only the maximum delivery mass flow is enforced. It is thus assumed that if the maximum available energy ($\dot{Q}_{ASHP-available}$) is larger than the required energy (to heat the water to the set-point temperature ($T_{ASHP-setpoint}$)); then only part of the maximum available energy is transferred to the water.

$$\begin{aligned}\dot{m}_{ASHP-max} &= \dot{m}_{ASHP} \\ \dot{Q}_{ASHP} &= \dot{m}_{ASHP} C_p (T_{ASHP-setpoint} - T_{inlet}) \\ \forall \dot{m}_{ASHP-predict} &> \dot{m}_{ASHP-max}\end{aligned}\quad (4.15)$$

The methodology above corresponds to that which was used in the reduced simulation, with aspects shared by the detailed simulation. For the detailed simulation, thermal capacitance is taken into account ($\frac{\partial}{\partial t} \int \rho p dV_{cv} \neq 0$), along with the heat loss of the ASHP. Consequently, the detailed simulation, the change of internal energy of the ASHP have to be calculated for each time-step. Apart from the thermal capacitance, the detailed simulation uses a pump and resistive curve to enforce the mass flow through the component. The conservation of energy for the ASHP control volume can be expressed as:

$$\frac{m_{ashp-volume} C_p (T_{ashp-new} - T_{ashp})}{\delta t} = \dot{Q}_{advection} - \dot{Q}_{heat-losses} + \dot{Q}_{friction} + \dot{W}_{pump} + Q_{ASHP-available} \quad (4.16)$$

The work-energy added by the pump to the ASHP control volume is given by:

$$\dot{W}_{pump} = (\dot{m}_{ASHP} g (a_1 (\frac{\dot{m}_{ASHP}}{\rho} 3600)^2 + a_2 (\frac{\dot{m}_{ASHP}}{\rho} 3600) + a_3)) \quad (4.17)$$

The mass flow in the detailed simulation is also calculated according to the control procedure outlined by Equations 4.13, 4.14 and 4.15. With the mass flow, a function of these equations, the friction factor (f) is consequently varied to obtain the required mass flow. The friction energy of the ASHP is given by:

$$\dot{Q}_{friction} = f \frac{l}{D} \frac{\rho V^2}{2} \frac{\dot{m}_{ASHP}}{\rho} \quad (4.18)$$

With the heat losses of the ASHP given by:

$$\dot{Q}_{heat-losses} = U A_{ASHP} (T_{ashp} - T_{db}) \quad (4.19)$$

For the detailed simulation, the energy flux addition to the system ($\dot{Q}_{ASHP}$) in the j -th time-step from Equation 4.16, corresponds to the advection energy of the control volume and is given by:

$$\dot{Q}_{advection} = \dot{m}_{ashp} C_p (T_{inlet} - T_{ashp}) = -\dot{Q}_{ASHP} \quad (4.20)$$

⁶With the maximum mass flow presented in Table 3.1.

Table 4.1 lists all of the parameters used in Equations 4.10 to 4.20 with the base unit and source of the applicable parameter.

TABLE 4.1: ASHP pipe heat loss parameters and units. In order of appearance.

<i>Term</i>	<i>Symbol</i>	<i>Unit</i>	<i>Source</i>
Maximum useful ASHP flux	$\dot{Q}_{(ASHP-available)}$	[W]	Simulation regression output
COP regression approximation	$\hat{\beta}$	[dim]	Simulation regression output
ASHP electrical consumption regression	$\dot{P}_{(electrical)}$	[W]	Simulation regression output
COP constant regression coefficient	α_0	[dim]	Regression Appendix H.3
COP dry bulb regression coefficient	α_1	[1/°C]	Regression Appendix H.3
Ambient dry bulb temperature	T_{db}	[°C]	Boundary condition input
COP inlet regression coefficient	α_2	[1/°C ²]	Regression Appendix H.3
ASHP inlet temperature	T_{inlet}	[°C]	Boundary condition input
COP relative humidity regression coefficient	α_3	[dim]	Regression Appendix H.3
Relative humidity	ϕ	[dim]	Boundary condition input
ASHP constant electrical regression coefficient	b_0	[W]	Regression Appendix H.4
ASHP inlet electrical regression coefficient	b_1	[W/°C ²]	Regression Appendix H.4
ASHP dry bulb electrical regression coefficient	b_2	[W/°C]	Regression Appendix H.4
ASHP wet bulb electrical regression coefficient	b_3	[W/°C]	Regression Appendix H.4
Set point temperature mass flow	$\dot{m}_{ASHP-predict}$	[kg s ⁻¹]	Equation 4.13
Specific heat at constant pressure (water)	C_p	[J/kg°C]	Thermodynamic tables
ASHP set-point temperature	$T_{ASHP-setpoint}$	[°C]	Fixed simulation input
Required ASHP flux	$\dot{Q}_{ASHP}$	[W]	Equations 4.14 and 4.15
Maximum delivery mass flow	$\dot{m}_{ASHP-max}$	[kg s ⁻¹]	Fixed simulation input
ASHP control volume mass	$m_{ashp-volume}$	[kg]	Fixed simulation input
Internal temperature of ASHP	$T_{(ashp)}$	[°C]	Simulation output
Internal temperature for next time step	$T_{(ashp-new)}$	[°C]	Simulation output
Time step	δt	[s]	Fixed simulation input
Advection ASHP flux	$\dot{Q}_{advection}$	[W]	Simulation output
ASHP heat loss flux	$\dot{Q}_{heat-losses}$	[W]	Simulation output
ASHP friction flux	$\dot{Q}_{friction}$	[W]	Simulation output
ASHP pump flux	$\dot{W}_{pump}$	[W]	Simulation output
Pump regression coefficient one	a_1	[sm ^{-5/2}]	Appendix J
Pump regression coefficient two	a_2	[s/m ²]	Appendix J
Pump regression coefficient three	a_3	[m]	Appendix J
Fluid density	ρ	[kg/m ³]	Thermodynamic tables
Gravitation acceleration	g	[m/s ²]	Physical constant
Friction coefficient	f	[dim]	Simulation output
ASHP inside pipe diameter	D	[m]	Fixed simulation input
ASHP characteristic pipe length	l	[m]	Fixed simulation input
ASHP axial flow velocity	V	[m s ⁻¹]	Simulation output
ASHP heat loss coefficient	UA_{ASHP}	[W/°C]	Fixed simulation input

4.1.2.2 Solar water heater (SWH) flux

For the SWH component, both the detailed and reduced simulation includes thermal capacitance. Thermal capacitance was included due to the transient behaviour of the SWH. The SWH pump is activated once the internal SWH temperature ($T_{SWH-outlet}$) is 12 [°C] warmer than the third tank temperature (T_5 , C7 Figure 4.3) of the stratified storage vessel arrangement. Finally, the circulating water pump deactivates when the temperature of the outlet on the SWH is 4 [°C] warmer than the middle storage tank temperature (T_3 , E8, Figure 4.3)⁷. The efficiency of

⁷Corresponding to the control procedure outlined in Section 3.2.2

the SWH is calculated according to the regression equation presented in Section 3.5.2, and the useful solar energy can be calculated from:

$$A = \frac{(T_{inlet} - T_{db})}{I_{rr}} \quad (4.21)$$

$$\eta_{solar} = a_1 A^2 + a_2 A + a_3 \quad (4.22)$$

$$\dot{Q}_{usefull-solar} = \eta_{solar}(I_{rr}A_c) \quad \forall \quad I_{rr} \geq 80[W] \quad (4.23)$$

With the heat loss for the SWH manifold defined by:

$$\dot{Q}_{heatloss-solar} = (1 - \eta_{solar})(I_{rr}A_c) \quad \forall \quad I_{rr} > 80[W] \quad (4.24)$$

However, Equation 4.21 will become undefined once the solar irradiance (I_{rr}) falls below zero. Consequently, the useful solar energy during these time-steps ($\dot{Q}_{usefull-solar}$) is expressed as:

$$\dot{Q}_{usefull-solar} = 0 \quad \forall \quad I_{rr} < 80[W] \quad (4.25)$$

With a steady-state $UA_{swh-manifold}$ heat loss coefficient used to calculate the SWH heat loss for these corresponding conditions:

$$\dot{Q}_{heatloss-solar} = UA_{swh-manifold}(T_{SWH} - T_{db}) \quad \forall \quad I_{rr} < 80[W] \quad (4.26)$$

The advection energy for the SWH, and consequently, SWH flux ($\dot{Q}_{swh}$) added to the system can be calculated from:

$$\dot{Q}_{advection} = \dot{m}_{swh} Cp(T_{inlet} - T_{SWH}) = -\dot{Q}_{SWH} \quad (4.27)$$

The reduced model explicitly defines the mass flow through the component; according to the outcome determined by the control system. Thus the internal energy evolution of the SWH for the reduced model is expressed as:

$$\frac{m_{swh-volume} Cp(T_{SWH-new} - T_{SWH})}{\delta t} = \dot{Q}_{advection} - \dot{Q}_{heatloss-solar} + \dot{Q}_{usefull-solar} \quad (4.28)$$

Contrastingly, for the detailed simulations, the SWH components use the same energy equations as presented for the ASHP pump ($\dot{W}_{pump}$, Equation 4.17) and the friction energy ($\dot{Q}_{friction}$, Equation 4.18) to calculate the mass flow. For conditions where the SWH has to be switched off, the friction coefficient (f) is to be chosen as high as possible to ensure the mass flow is as low as possible ($\dot{m}_{SWH} < 0.0001$). The change in internal energy of the SWH can be calculated by:

$$\frac{m_{swh-volume} Cp (T_{SWH-new} - T_{SWH})}{\delta t} = \dot{Q}_{advection} - \dot{Q}_{heatloss-solar} + \dot{Q}_{friction} + \dot{W}_{pump} + \dot{Q}_{usefull-solar} \quad (4.29)$$

Table 4.2 lists all of the parameters used in Equations 4.21 to 4.29 with the base unit and source of the applicable parameter.

TABLE 4.2: SWH flux parameters and units. In order of appearance.

<i>Term</i>	<i>Symbol</i>	<i>Unit</i>	<i>Source</i>
SWH performance parameter	A	$[\text{°C m}^2/\text{W}]$	Simulation output
Ambient dry bulb temperature	T_{db}	$[\text{°C}]$	Boundary condition input
Inlet temperature	T_{inlet}	$[\text{°C}]$	Internal simulation output
Solar irradiance	I_{rr}	$[\text{W}/\text{m}^2]$	Boundary condition input
Regression coefficient one	a_0	$[\text{W}^2/\text{m}^4 \text{ °C}^2]$	Fixed simulation input
Regression coefficient two	a_1	$[\text{W}/\text{m}^2 \text{ °C}]$	Fixed simulation input
Regression coefficient three	a_2	$[\text{dim}]$	Fixed simulation input
SWH collector efficiency	η_{SWH}	$[\text{dim}]$	Simulation regression calculation
Useful absorbed solar energy	$\dot{Q}_{usefull}$	$[\text{W}]$	Simulation output
SWH manifold heat loss	$\dot{Q}_{heatloss-solar}$	$[\text{W}]$	Simulation output
SWH steady-state heat loss coefficient	$UA_{swh-manifold}$	$[\text{W}/\text{°C}]$	Fixed simulation input
SWH manifold temperature	T_{SWH}	$[\text{°C}]$	Internal simulation output
Manifold temperature for next time step	$T_{SWH-new}$	$[\text{°C}]$	Internal simulation output
SWH steady state mass flow	$\dot{m}_{swh}$	$[\text{kg s}^{-1}]$	Internal simulation output
Advection SWH flux	$\dot{Q}_{advection}$	$[\text{W}]$	Simulation output
SWH system flux	$\dot{Q}_{SWH}$	$[\text{W}]$	Simulation output
SWH friction flux	$\dot{Q}_{friction}$	$[\text{W}]$	Simulation output
SWH pump flux	$\dot{W}_{pump}$	$[\text{W}]$	Simulation output
SWH control volume mass	$m_{swh-volume}$	$[\text{kg}]$	Fixed simulation input
Time step	δt	$[\text{s}]$	Fixed simulation input
Specific heat at constant pressure	Cp	$[\text{J}/\text{kg °C}]$	Thermodynamic tables

4.1.2.3 Heat loss storage vessels

The storage vessels have the largest thermal mass relative to all of the components in the system. Accordingly, in both simulations, the thermal capacitance is taken into account. The total heat loss of the five storage vessels is expressed by:

$$\dot{Q}_{tank-losses} = \sum \dot{Q}_{tank-i} \quad \forall i \in \{1, 2, 3, 4, 5\} \quad (4.30)$$

Each tank has a distinct heat loss coefficient for the top, bottom and sides of the storage vessel:

$$\dot{Q}_{tank-i} = (UA_{sides-i} + UA_{bottom-i} + UA_{top-i}) \times (T_{tank_i} - T_{db}) \quad (4.31)$$

The heat loss coefficients were estimated according to the analyses presented in Appendix B.1. The advection flow out and into the storage vessel components are represented by:

$$\dot{Q}_{advection} = \sum_1^n \dot{m}_{in-j} Cp T_{in-j} - \sum_1^m \dot{m}_{out-i} Cp T_{tank-i} \quad (4.32)$$

$$\forall \ n = 3 \ m = 2$$

With T_{in-j} , designating the internal temperature of the adjacent components supplying mass ($\dot{m}_{in-j}$) to the i -th storage vessel component (T_{tank-i}). For the reduced simulation, a PDE was derived for the change in internal energy of the storage vessels. The derivation of the RK-MOL PDE for a storage vessel is demonstrated in detail in Appendix F; with the following PDE that was obtained:

$$\frac{\partial T}{\partial t} = \frac{k_{fluid}}{\rho_{fluid} C p_{fluid}} \frac{\partial^2 T}{\partial x^2} - \frac{\dot{m}_{net} C p_{fluid}}{A_w \rho_{fluid} C p_{fluid}} \frac{\partial T}{\partial x} + \frac{2h_{sidewalls}}{\rho_{fluid} C p_{fluid}} \frac{\partial T}{\partial r} \quad (4.33)$$

The conservation of energy for the storage vessels is expressed in the detailed simulation by:

$$\frac{m_{tank-volume-i} C p (T_{tank-new-i} - T_{tank-i})}{\delta t} = \dot{Q}_{advection} - \dot{Q}_{tank-i} \quad (4.34)$$

Table 4.3 lists all of the parameters used in Equations 4.30 to 4.34 with the base unit and source of the applicable parameter.

TABLE 4.3: Storage vessel flux parameters and units. In order of appearance.

Term	Symbol	Unit	Source
Total storage vessel losses	$\dot{Q}_{tank-losses}$	[W]	Simulation output
i -th storage vessel losses	$\dot{Q}_{tank-i}$	[W]	Simulation output
Side tank heat loss coefficient	$UA_{sides-i}$	[W/°C]	Fixed simulation input
Bottom tank heat loss coefficient	$UA_{bottom-i}$	[W/°C]	Fixed simulation input
Top tank heat loss coefficient	UA_{top-i}	[W/°C]	Fixed simulation input
Internal i -th tank temperature	$T_{(tank_i)}$	[°C]	Simulation output
Ambient dry bulb temperature	T_{db}	[°C]	Boundary condition input
Advection flux	$\dot{Q}_{advection}$	[W]	Simulation output
Supply mass flow inward at j -th port	$\dot{m}_{in-j}$	[kg s ⁻¹]	Internal simulation input
Supply mass flow out k -th port	$\dot{m}_{out-k}$	[kg s ⁻¹]	Internal simulation input
Specific heat at constant pressure	Cp	[J/kg°C]	Thermodynamic tables
Fluid density	ρ	[kg/m ³]	Thermodynamic tables
Thermal conductivity	k_{fluid}	[W/m°C]	Thermodynamic tables
Wetted flow area	A_w	[m ²]	Fixed simulation input
Side heat loss coefficient	$h_{sidewalls}$	[W/m ² °C]	Fixed simulation input
Pipe control volume mass	$m_{tank-volume-i}$	[kg]	Fixed simulation input
Internal i -th tank temperature for next time step	$T_{(tank-new-i)}$	[°C]	Simulation output
Time step	δt	[s]	Fixed simulation input

4.1.2.4 Ring-main heat losses

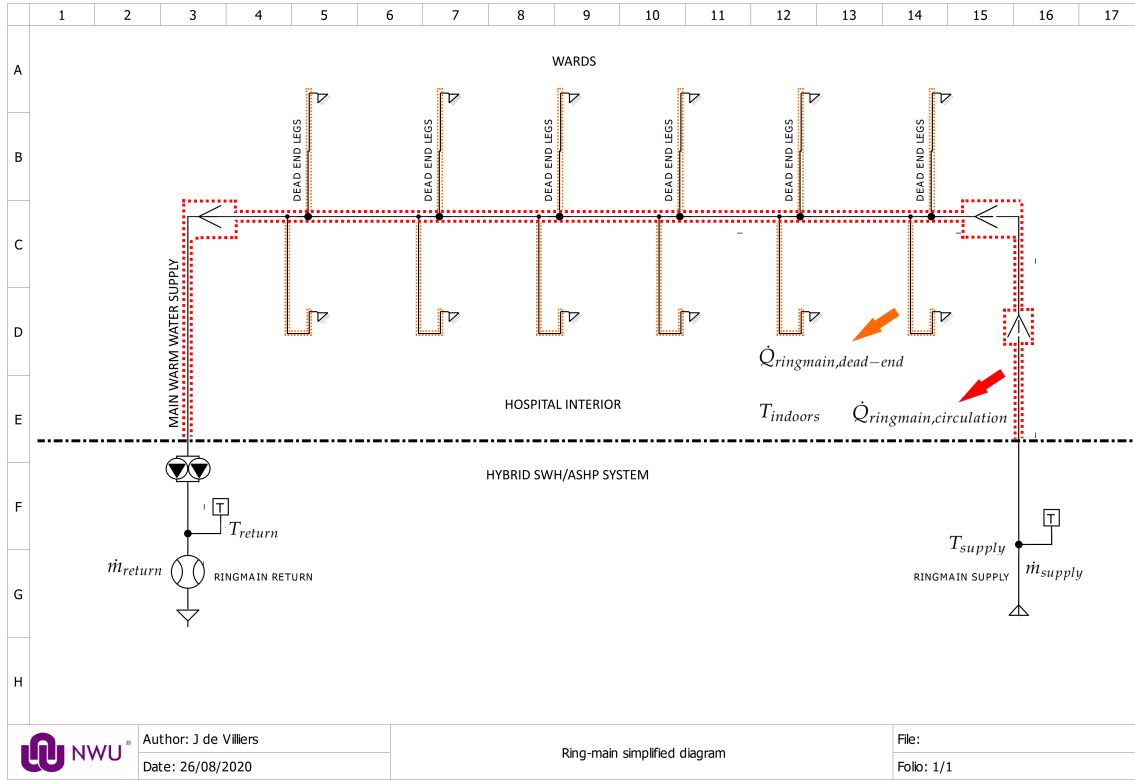


FIGURE 4.4: Ring-main diagram illustrating the control volumes of the dead-end and main ring-main losses.

The steady-state heat loss for the ring-main can be approximated by two components (Guo and Goumba, 2018). The first entails the circulation heat losses incurred by piping the water through the main artery of the ring-main system ($\dot{Q}_{ringmain,circulation}$). The second entails the heat losses incurred in the pipes of the various draw-off branches to the many wards of the hospital ($\dot{Q}_{ringmain,dead-end}$). Figure 4.4 serves to illustrate the former two control volumes of the energy fluxes graphically. For ease of analyses, a single draw-off with a linear decrease in temperature between the supply (T_{supply}) and return temperatures (T_{return}) was assumed. The circulation losses can be calculated from:

$$\dot{Q}_{ringmain,circulation} = UA_{ringmain,circulation} \times (T_{ringmain,average} - T_{indoors}) \quad (4.35)$$

$$\dot{Q}_{ringmain,circulation} = K_d \times L_{ring} \times (T_{ringmain,average} - T_{indoors}) \quad (4.36)$$

Where L_{ring} is the total length of the ring-main in [m]; K_d is the coefficient of pipe heat loss [W/m°C]; $T_{ringmain,average}$ comprises the average temperature in the ring-main and $T_{indoors}$ designates the temperature of the indoor hospital air. The K_d is calculated according to the pipe diameter (d_o) as in EU standard 12828:2012+A1 (EU Standard, 2014):

$$K_d = 2.6 \times d_o + 0.2 \quad (4.37)$$

The losses incurred in the dead-end legs of the ring-main are given by:

$$\dot{Q}_{ringmain,dead-end} = UA_{ringmain,dead-end} \times (T_{ringmain,average} - T_{indoors}) \quad (4.38)$$

The above can be rewritten as:

$$\dot{Q}_{ringmain,dead-end} = \left(\frac{\pi \times Cp \times L_{dead} \times d_i^2}{4} \right) \times \rho \times N_{dead-pipes} \times (T_{ringmain,average} - T_{indoors}) / \delta T \quad (4.39)$$

With L_{dead} the average length of the dead-end pipes [m]; d_i the inside diameter of the dead-end pipes [m]; and $N_{dead-pipes}$ designating the number of dead-end pipe sections. Therefore for the reduced simulation, the ring-main losses could be calculated from:

$$\dot{Q}_{ring-main} = \dot{Q}_{ringmain,dead-end} + \dot{Q}_{ringmain,circulation} \quad (4.40)$$

The ring-main average temperature ($T_{ringmain,average}$) of the previous time step is used for the heat loss calculation in Equations 4.36 and 4.39. For the reduced simulation, the following assumptions are made:

- The internal temperature ($T_{indoors}$) in all of the hospital wards and main building are maintained at the same temperature.
- The heat loss characteristics of the piping remain uniform throughout the network of the hospital building.
- Since warm water might not always be drawn off from the same 'dead-end' leg; it is assumed that the total user load is drawn roughly at the middle of the ring-main circuit at all times.

The draw-off temperature for the following time step is thus approximated by the following relation:

$$T_{draw-off} = T_{supply} - \frac{0.5 \times \dot{Q}_{ring-main}}{\dot{m}_{ring-main} Cp} \quad (4.41)$$

Contrastingly, for the detailed simulation, the heat losses are based on the internal component temperatures that the ring-main comprises of:

$$\dot{Q}_{ringmain,steady} = UA_{10} \times (T_{10} - T_{db}) + UA_{11} \times (T_{11} - T_{db}) + UA_F \times (T_F - T_{db}) \quad (4.42)$$

With T_{10} , T_{11} and T_f , corresponding to the internal temperatures of the ring-main system pipes and the T-piece component (**B4**, **E2** and **H4** in Figure 4.3). The change in pipe component internal temperatures is similar to that of the storage vessel component (Equation 4.34). The main difference is that the pipe component supports a sub-pump component; leading to the additional requirement for pump and friction energy (Equations 4.17 and 4.18).

$$\frac{\dot{m}_{pipe-volume} Cp (T_{pipe-new} - T_{pipe})}{\delta t} = \dot{Q}_{advection} - \dot{Q}_{heat-losses} + \dot{Q}_{friction} + \dot{W}_{pump} \quad (4.43)$$

The T-piece component in the detailed simulation follows the same principle as that of the storage vessel components. The component supports three mass flux ports that can connect to other components. From the incompressible fluid, the summation of the mass fluxes can be expressed by:

$$\sum_1^n \dot{m}_{in-i} = \sum_1^m \dot{m}_{out-j} \quad (4.44)$$

$$n + m < 3 \quad \forall \quad n \neq 3; \quad m \neq 3$$

$$i, j \in \{1, 2\}$$

The advection of the T-piece component is given by:

$$\dot{Q}_{advection} = \sum_1^n \dot{m}_{in-i} Cp T_{in_i} - \sum_1^m \dot{m}_{out-j} Cp T_{connection} \quad (4.45)$$

With T_{in_j} corresponding to the internal temperatures and mass fluxes of adjacent components ($\dot{m}_{in-j}$) that flow into the control volume. $T_{connection}$ corresponds to the internal temperature of the T-piece component. The conservation of energy for the T-piece component is given by:

$$\frac{m_{connection-volume} Cp (T_{connection-new} - T_{connection})}{\delta t} = \dot{Q}_{advection} - \dot{Q}_{heat-losses} \quad (4.46)$$

Table 4.4 lists all of the parameters used in Equations 4.35 to 4.46 with the base unit and source of the applicable parameter.

TABLE 4.4: Ring-main parameters and units. In order of appearance.

Term	Symbol	Unit	Source
Ring-main circulation losses	$\dot{Q}_{ringmain,circulation}$	[W]	Simulation output
Ring-main main heat loss coefficient	$UA_{ringmain,circulation}$	[W/°C]	Fixed simulation input
Average ring-main temperature	$T_{ring-main,average}$	[°C]	Internal simulation output
Ambient indoor temperature	$T_{indoors}$	[°C]	Boundary condition input
Coefficient of pipe heat loss	K_d	[W/m°C]	Fixed simulation input
Ring-main pipe length	L_{ring}	[m]	Fixed simulation input
Ring-main pipe diameter	d_o	[m]	Fixed simulation input
Dead-end rate of energy losses	$\dot{Q}_{ringmain,dead-end}$	[W]	Simulation output
Coefficient of pipe heat loss dead-end legs	$UA_{ringmain,dead-end}$	[W/m°C]	Fixed simulation input
Specific heat at constant pressure	C_p	[J/kg°C]	Thermodynamic tables
Ring-main dead-end pipe length	L_{dead}	[m]	Fixed simulation input
Ring-main pipe inside diameter	d_i	[m]	Fixed simulation input
Fluid density	ρ	[kg/m³]	Thermodynamic tables
Number of dead-end pipes	$N_{dead-pipes}$	[dim]	Fixed simulation input
Time constant	δT	[s]	Fixed simulation input
Ring-main flux	$\dot{Q}_{ring-main}$	[W]	Internal simulation output
Ring-main user supply temperature	$T_{draw-off}$	[°C]	Simulation output
Ring-main supply temperature	T_{supply}	[°C]	Simulation output
Pipe heat loss coefficient	UA_{10}	[W/°C]	Fixed simulation input
Pipe heat loss coefficient	UA_{11}	[W/°C]	Fixed simulation input
T-piece heat loss coefficient	UA_F	[W/°C]	Fixed simulation input
Internal pipe component temperature	T_{10}	[°C]	Simulation output / Initial conditions
Internal pipe component temperature	T_{11}	[°C]	Simulation output / Initial conditions
Internal T-piece component temperature	T_F	[°C]	Simulation output / Initial conditions
Pipe control volume mass	$m_{pipe-volume}$	[kg]	Fixed simulation input
Time step	δt	[s]	Fixed simulation input
Internal pipe temperature	T_{pipe}	[°C]	Simulation output / Initial conditions
Internal pipe temperature next time step	$T_{pipe-new}$	[°C]	Simulation output
Advection flux	$\dot{Q}_{advection}$	[W]	Simulation output
Heat loss flux	$\dot{Q}_{heat-losses}$	[W]	Simulation output
Friction flux	$\dot{Q}_{friction}$	[W]	Simulation output
Pump flux	$\dot{W}_{pump}$	[W]	Simulation output
Supply mass flow inward at i -th port	$\dot{m}_{in-i}$	[kg s ⁻¹]	Internal simulation input
Supply mass flow out j -th port	$\dot{m}_{out-kj}$	[kg s ⁻¹]	Internal simulation input
T-piece control volume mass	$m_{connection-volume}$	[kg]	Fixed simulation input
Internal T-piece connection temperature	$T_{connection}$	[°C]	Simulation output / Initial conditions
Next time step internal temperature	$T_{connection-new}$	[°C]	Simulation output

4.1.2.5 User energy load

For the reduced simulation, the user energy load is calculated from the draw-off temperature ($T_{draw-off}$) approximated by Equation 4.41 (from the ring-main energy flux); the user load ($\dot{m}_{user}$) and the municipality water temperature ($T_{municipal,in}$). With $\dot{m}_{user}$ and $T_{municipal,in}$ as input boundary conditions. The user load flux can be calculated from:

$$\dot{Q}_{user} = \dot{m}_{user} C_p (T_{municipal,in} - T_{draw-off}) \quad (4.47)$$

For the detailed simulation, the internal temperature (E2, Figure 4.3) of the T-piece component is used in contrast to the approximation offered for the ring-main supply temperatures $T_{draw-off}$ by Equation 4.41. The T-piece component uses the same equations as presented in the

previous section (Equations 4.44, 4.45 and 4.46). Consequently, the conservation of energy for the T-piece component is given by:

$$\dot{Q}_{user} = \dot{m}_{user} Cp (T_{municipal,in} - T_F) \quad (4.48)$$

Table 4.5 lists all of the parameters used in Equations 4.47 and 4.48 with the base unit and source of the applicable parameter.

TABLE 4.5: User load parameters and units. In order of appearance.

<i>Term</i>	<i>Symbol</i>	<i>Unit</i>	<i>Source</i>
User load flux	$\dot{Q}_{user}$	[W]	Simulation output
User mass flow	$\dot{m}_{user}$	[kg s ⁻¹]	Boundary condition input
Specific heat at constant pressure	Cp	[J/kg°C]	Thermodynamic tables
Municipal input temperature	$T_{municipal,in}$	[°C]	Boundary condition input
Ring-main draw of temperature	$T_{draw-off}$	[°C]	Simulation component output - Equation 4.41
Internal T-piece temperature	T_F	[°C]	Initial condition / simulation output

4.1.2.6 Storage vessel pipe losses

Thermal capacitance for the storage vessel pipes is neglected for the reduced simulation. To approximate the losses in the pipes between the storage vessels, the average temperature of the storage vessels are used:

$$\dot{Q}_{pipe-losses} = UA_{pipe-losses} \times (T_{(pipes-tank)} - T_{db}) \quad (4.49)$$

However, the term will not actively alter the systems internal energy if the heat loss is not added as a boundary condition to components with thermal capacitance (such as the ring-main system). Consequently, the additional heat loss (Equation 4.49) was subtracted from the storage vessel PDE (Equation 4.33) for the reduced simulation.

For the detailed simulation, the storage vessel pipe losses were taken as the heat loss fluxes of the pipe and T-piece components designated at *H9*, *H11*, *H14*, *G16*, *F16* and *B12* in Figure 4.3. With the advection, friction and pump energy fluxes taking the same form as illustrated in the previous sections (4.1.2.1 and 4.1.2.2) for the detailed simulation (Equations 4.18, 4.17 and 4.45).

$$\frac{m_{pipe-volume} Cp (T_{(pipes-tank-new)} - T_{(pipes-tank)})}{\delta t} = \dot{Q}_{advection} - \dot{Q}_{heat-losses} + \dot{Q}_{friction} + \dot{W}_{pump} \quad (4.50)$$

Table 4.6 lists all of the parameters used in Equations 4.49 to 4.50 with the base unit and source of the applicable parameter.

TABLE 4.6: Storage vessel pipe heat loss parameters and units. In order of appearance.

<i>Term</i>	<i>Symbol</i>	<i>Unit</i>	<i>Source</i>
Pipe heat loss coefficient	$UA_{(pipe-losses)}$	[W/°C]	Fixed simulation input
Internal temperature	$T_{pipes-tank}$	[°C]	Initial condition / simulation output
Ambient dry bulb temperature	T_{db}	[°C]	Boundary condition input
Water mass of pipes	$m_{(pipes-volume)}$	[kg]	Fixed simulation input
Specific heat at constant pressure	Cp	[J/kg°C]	Thermodynamic tables
Internal temperature for next time step	$T_{(pipes-tanks-new)}$	[°C]	Simulation output
Time step	δt	[s]	Fixed simulation input
Control volume advection flux	$\dot{Q}_{advection}$	[W]	Simulation output
Control volume heat loss	$\dot{Q}_{(heat-losses)}$	[W]	Simulation output
Friction flux	$\dot{Q}_{friction}$	[W]	Simulation output
Pump flux	$\dot{W}_{pump}$	[W]	Simulation output

4.1.2.7 Heat loss SWH pipes

The SWH pipes leading from the SWH manifold to the storage vessel acts as an extended thermal battery to the SWH. Due to the high build-up of thermal energy, and the discontinuous flow through these pipes, the thermal capacitance had to be taken into account. Thus, for the reduced and detailed simulation, the conservation of energy for these pipes was taken as:

$$\frac{m_{(SWH-pipes)} Cp (T_{(SWH-pipes-new)} - T_{(SWH-pipes)})}{\delta t} = \dot{Q}_{advection} - \dot{Q}_{(heat-losses)} \quad (4.51)$$

With the advection component of the SWH pipes given by:

$$\dot{Q}_{advection} = \dot{m}_{swh} Cp (T_{swh} - T_{(SWH-pipes)}) \quad (4.52)$$

With T_{SWH} corresponding to the outlet temperature of the SWH in the previous time-step (Equation 4.27). Thus, from the conservation of energy Equations 4.28 and 4.29, presented in Section 4.1.2.2, and Equation 4.51, the heat loss can be calculated from:

$$\dot{Q}_{(SWH-pipes)} = UA_{(SWH-pipes)} \times (T_{(SWH-pipes)} - T_{db}) = \dot{Q}_{(heat-losses)} \quad (4.53)$$

Table 4.7 lists all of the parameters used in Equations 4.49 to 4.53 with the base unit and source of the applicable parameter.

TABLE 4.7: SWH pipe heat loss parameters and units. In order of appearance.

<i>Term</i>	<i>Symbol</i>	<i>Unit</i>	<i>Source</i>
Water mass of SWH pipes	$m_{(SWH-pipes)}$	[kg]	Fixed simulation input
Specific heat at constant pressure	C_p	[J/kg°C]	Thermodynamic tables
SWH pipe heat loss coefficient	$UA_{(SWH-pipes)}$	[W/°C]	Fixed simulation input
Internal temperature	$T_{SWH-pipes}$	[°C]	Initial condition / simulation output
Internal temperature for next time step	$T_{(SWH-pipes-new)}$	[°C]	Simulation output
Time step	δt	[s]	Fixed simulation input
Control volume advection flux	$\dot{Q}_{advection}$	[W]	Simulation output
Control volume heat loss	$\dot{Q}_{(heat-losses)}$	[W]	Simulation output
SWH mass flow	$\dot{m}_{swh}$	[kg s ⁻¹]	Fixed simulation input
SWH outlet temperature	T_{swh}	[°C]	Simulation output
SWH heat loss flux	$\dot{Q}_{(SWH-pipes)}$	[W]	Simulation output
Ambient dry bulb temperature	T_{db}	[°C]	Boundary condition input

The differences in equations, assumptions and boundary conditions between the reduced and detailed system components are summarised in Appendix E:

- T-piece component (Table E.1)
- Storage vessel component (Table E.2)
- Pipe component (Table E.3)
- SWH component (Table E.4)
- ASHP component (Table E.5)

The fundamental derivation on the finite difference method that was used in the simulations is presented in Appendix G.

4.2 Simulation architecture

With the main conservation continuity equations derived, the algorithm will be presented that was used to deploy the derived component equations. This section presents the simulations that were developed and the differences between them. The section was consequently divided into the following subsections:

- Discretization (4.2.1) - The establishment of a spatial and temporal grid to implement in the simulation programs.
- Component boundary conditions (4.2.2) - The decomposition of the energy equation into characteristic fields for each component is presented.
- Simulations (4.2.3) - A discussion on the most notable difference between the RK-MOL and Euler simulation and the numerical method used by each method, respectively.
- Network Theory (4.2.3) - Discussion on the use of Kirchoff's circuit laws to obtain additional equations to solve the system.

4.2.1 Discretization

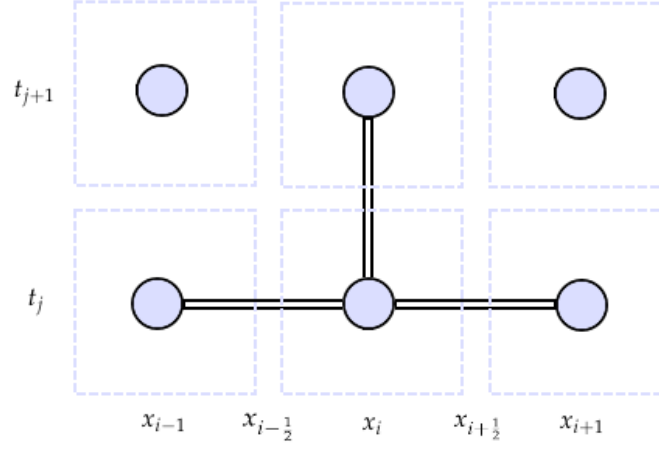


FIGURE 4.5: Numerical stencil employed.

To approximate the solution to the PDEs and DEs, a spatial and temporal numerical grid is established. This study uses a 1D finite difference method. The discretized system for the detailed simulation is shown in Figure 4.3. The components with thermal capacitance are indicated as boxes in Figure 4.3. The length of the tank/pipe system is divided into m discrete layers to the direction of flow with h defining the spatial distance between the centre of the cells/components:

$$x_i = ih \quad \text{with} \quad i \in 0, 1, 2, 3..m$$

The cell interfaces to neighbouring cells is defined as:

$$\begin{aligned} x_{i+\frac{1}{2}} &= (i + 0.5)h \quad \text{with} \quad i \in 0, 1, 2, 3..m \\ x_{i-\frac{1}{2}} &= (i - 0.5)h \quad \text{with} \quad i \in 0, 1, 2, 3..m \end{aligned}$$

The width of the cells is defined as h and can be expressed as:

$$h = \Delta x_i = x_{i+\frac{1}{2}} - x_{i-\frac{1}{2}} \quad \text{with} \quad i \in 0, 1, 2, 3..m$$

The detailed and reduced simulations employ fixed time-steps and define this finite time-step by k . Furthermore, the termination of the integration is defined as the finite lapsed time M , where M is a multiple of the time-step k and n is the number of time steps.

$$\begin{aligned} k = \Delta t_j &= t_j - t_{j-1} \quad \text{with} \quad j \in 1, 2, 3..M \\ T_0 &= t_0 = 0 \end{aligned}$$

The finite difference method is employed to make approximations $E_i^n \in \mathbb{R}^m$ of the energy solution $e(x_i, t_j)$ at discrete points. The true values at the mesh grids are given by:

$$e_i^m = e(x_i, t_j) \tag{4.54}$$

E_i^n is stated as an approximation to the cell average of the true values $e(x, t_n)$, and is written as:

$$\bar{e}_i^n = \frac{1}{h} \int_{x_{(i-\frac{1}{2})}}^{x_{(i+\frac{1}{2})}} e(x, t_n) dx \quad (4.55)$$

Initial data for the numerical method is specified by using pointwise values, $E_i^0 = e_i^0$, at the boundary conditions and cell averaged values, $E_i^0 = \bar{e}_i^0$, at all of the other system components. To obtain the properties in these cell averages a piecewise constant function $E_k(x, t) \quad \forall x, t \in E_i^j$ is defined. The function value is assigned the approximation E_i^j in the (i, j) grid cell.

$$E_k(x, t) = E_j^n \quad \forall (x, t) \in [x_{i-\frac{1}{2}}, x_{i+\frac{1}{2}}) x[t_j, t_{j+1}) \quad (4.56)$$

Figure 4.5 illustrates the numerical stencil that was used in both the RK-MOL and detailed simulations. With the system discretization defined, the boundary conditions have to be enforced on the system in each time step. The boundary conditions can be the ambient conditions, user load, and the energy conditions at the end domain of the spatial discretization. If one defines the finite spatial approximations as $a \leq x \leq b$, then the boundary conditions can be specified at a or b . The boundary conditions are specified in EES by using the parametric table. Initial conditions are specified in the computer code itself. With the numerical method, the temporal and spatial grid established, the energy coupling between the components have to be established.

4.2.2 Component boundary conditions

Due to the assumption of a linear system of equations and the finite difference method; great care has to be exercised with the boundary conditions of each cell. Based on the discretization of the previous paragraph, the unwinding scheme was implemented and will be presented. From the previous sections, the thermal energy capacitance was not assumed negligible ($\frac{\partial}{\partial t} \int e \rho dV_{cv} \neq 0$) for some components. Thus the advection terms in the thermal energy equation must be aligned with the direction of flow. In mathematical terms, one has to, decouple the linear system of equations so that the solution is composed into characteristic fields. Thus for each cell, there will be two types of energy equations to be employed by the upwind method depending on the flow direction:

$$E_i^{n+1} = E_i^n - \frac{k}{h} A^+ (E_i^n - E_{i-1}^n) - \frac{k}{h} A^- (E_{i+1}^n - E_i^n) \quad (4.57)$$

From linear algebra, the system can be decomposed as $A^+ + A^- = A$. Thus depending on the sign of the eigenvector, either the A^+ or A^- term will be zero in the previous equation. Figure 4.6 illustrates Equation 4.57 graphically. If the flow is to the right, then the energy terms on the source terms for the cell boundaries are to the left of the cell centres. When the flow is to the left, then energy sources are to the right of the cell centres. Although the fundamental mass, linear momentum and energy equations were established for the components, additional restrictions are required for a system of components to be solved.

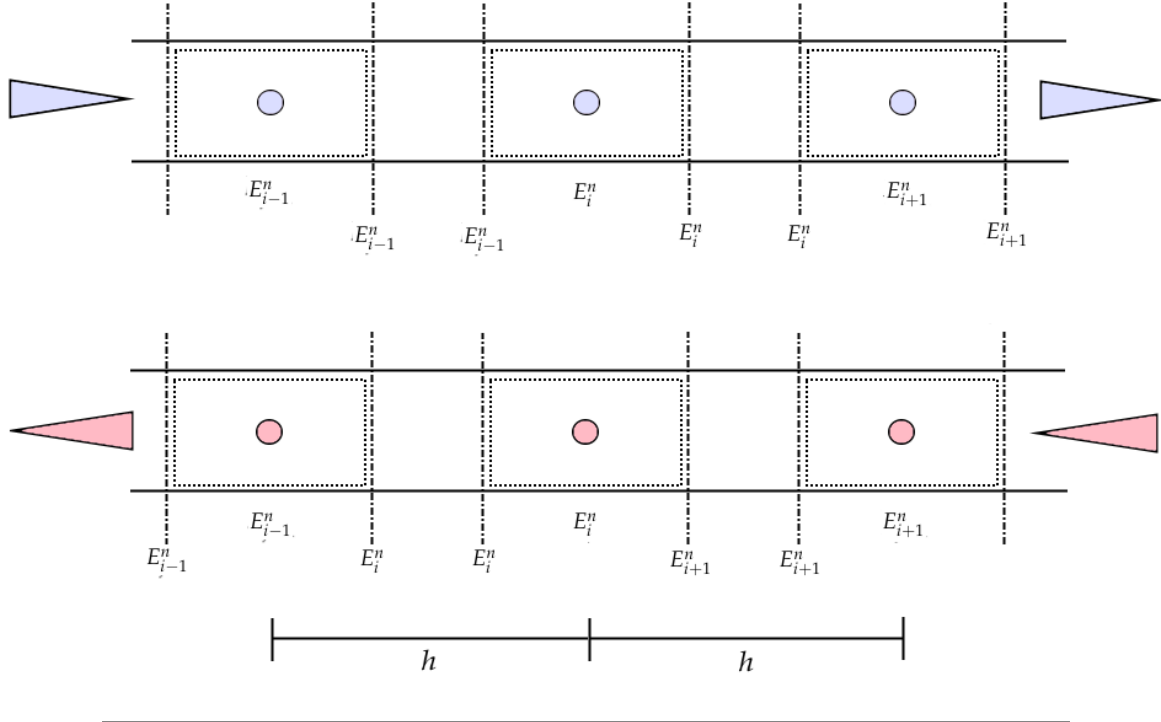


FIGURE 4.6: Numerical unwinding scheme, illustrating the direction of flow and energy.

4.2.3 Solution methodology

The discretisation, component equations and energy coupling were discussed in the previous sections. To solve the change in energy state between components in the network; the mass distribution throughout the network has to be calculated. The detailed and reduced simulations use different strategies to solve the mass distributions based on the assumptions that were made. The detailed simulation is based on an explicit, 1D finite-difference, Euler solver with Mass-Momentum and Energy conservation. The reduced simulation is based on an explicit, 1D finite-difference, Runge-Kutta method of lines solver with only mass and energy conservation. The detailed simulation consequently had 1114 equations in contrast with the 189 equations of the reduced simulation. The solution methodologies and calculation of mass fluxes will be showcased in the following sections.

4.2.3.1 Runge-Kutta method of lines solver

The most notable difference between the RK-MOL solver and the Euler model is the use of the linear momentum equation. The cross-coupling between the mass-momentum and energy equations lead to the necessity of iterative calculations. In large thermo-hydraulic networks, these calculations can be computationally expensive. Furthermore, this coupling due to the inherent characteristics of the PDE problem leads to stiffness and consequently requires a fine spatial mesh. The stiffness also necessitates small time steps due to the explicit numerical method that is applied. Furthermore, the nature of the PDE changes as the control system changes continuously. By eliminating one of the fundamental equations, the solver speed can be drastically enhanced.

However, the linear momentum equation is critical to determine the flow distribution in a 1D flow network. Fortunately, in the case study, the steady-state mass flow for various branches are known beforehand. The mass flows are mainly known since most of the loops in the network have at least one water pump; leading to a known steady-state mass flow. Thus, the linear momentum equation can be neglected with a compromise on accuracy but gain in computational effort. By explicitly defining the flow distribution, the focus can fall on the solving of the energy equation. The methodology followed by the simulation is showcased in the flowchart in Figure 4.7. To summarise, the methodology of the simulation can be divided into:

1. Inlet conditions to the ASHP and SWH are guessed according to the boundary conditions of the current time step and the user load of the previous time-step.
2. The control logic of the ASHP and SWH determines if these components need to be switched on or not and defines the required mass flow through these components.
3. The mass flow through the network is calculated algebraically, according to the previous two steps.
4. The energy equation is solved at the inlet of the ASHP and SWH according to the mass flows calculated in the previous steps.
5. If the guessed inlet conditions of the ASHP and SWH differ from those calculated in the previous step; then steps two to five is repeated until convergence is obtained.
6. Boundary conditions for the next time step are loaded and repeated until all the time-steps have been solved.

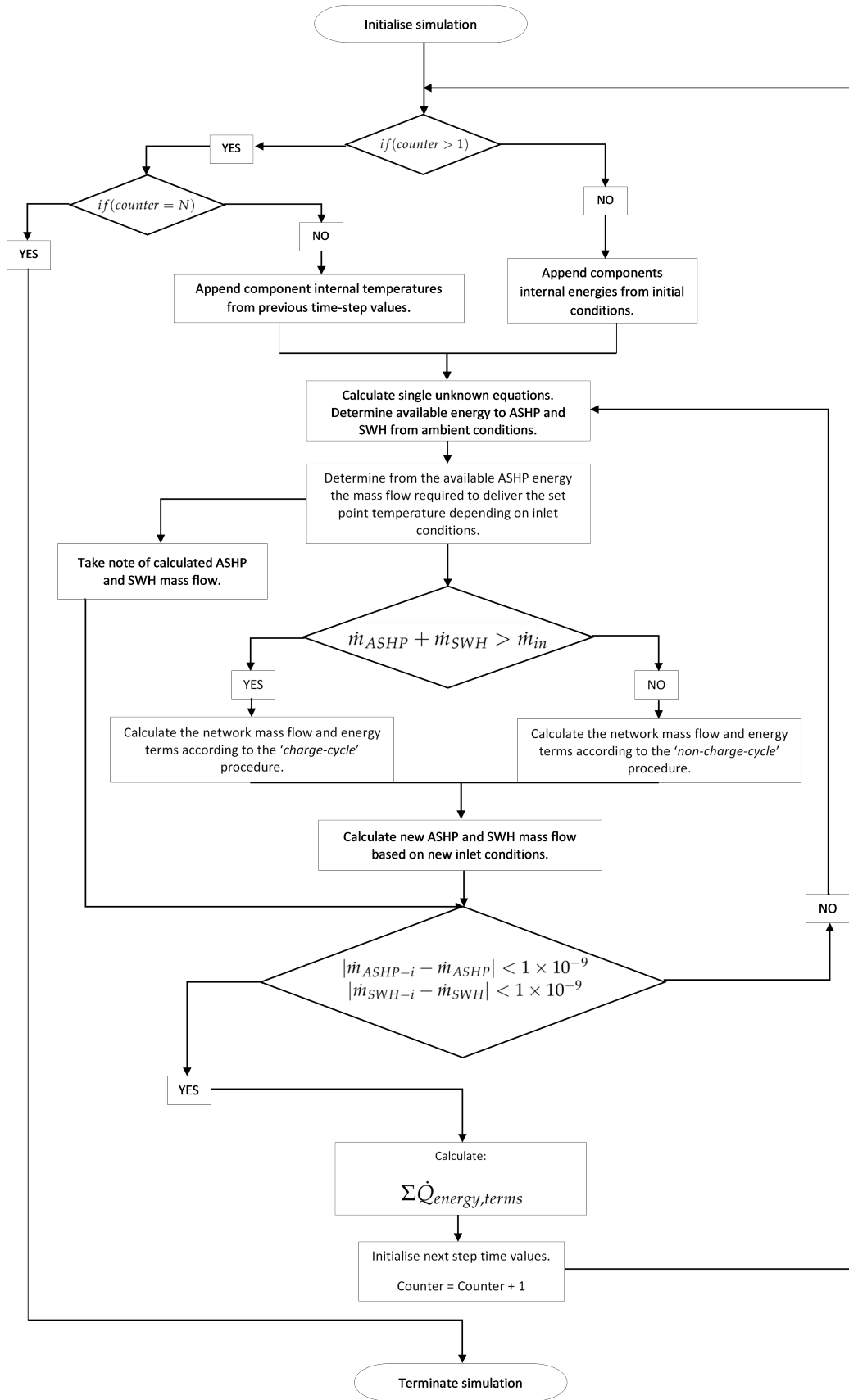


FIGURE 4.7: Flow chart of RK-MOL computer model.

The source code of this model is showcased in Appendix L.

4.2.3.2 Euler solver

The detailed simulation model utilises the explicit Euler numerical method to predict the internal temperature of different components. The principal difference between this model and the RK-MOL simulation is that the linear momentum equation is taken into account (Equation 4.63 and Equation 4.62) and is implemented in the form of the Darcy-Weisbach equation, as discussed in Section G.0.3. Pump curves were used so that the network behaves realistically in terms of mass flow distribution. Consequently, iterative calculations are required to solve the mass flows. The Euler model serves as a reference simulation to compare the RK-MOL model in terms of simulation time and accuracy. The source code for this simulation is showcased in Appendix K.

The Euler simulation requires additional network equations to link the linear momentum equations of the individual components to that of the network. Using an analogical model in the form of Kirchoff's circuit laws, the additional equations can be obtained. Figure 4.8 depicts a simple loop circuit with two pressure boundaries. The pump component (5) forces a counter-clockwise circulating flow through pipe components 4, 3, and 2. A relatively small through flow is established through pipe components 1 and 6 and the rest of the network. The pipe components can be defined as resistances and the pump as a voltage/pressure source. The nodes are defined as the points between components, and in this example, they are A, B, C and D. The sum of the pressure drops and rises around the loop from Kirchoff's voltage law should be zero:

$$0 = \Delta P_{A,D} + \Delta P_{D,C} + \Delta P_{C,B} + \Delta P_{B,A} \quad (4.58)$$

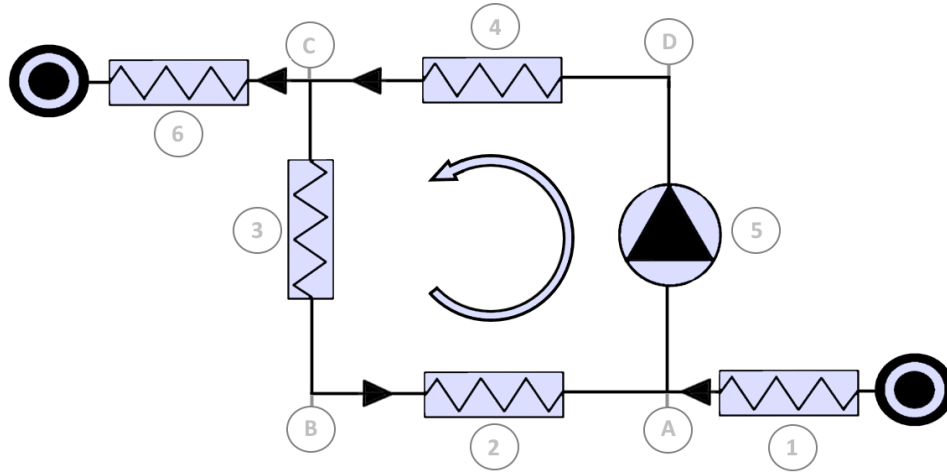


FIGURE 4.8: Simple flow network.

From Kirchoff's current laws, the continuity of mass must be enforced at junctions A and C in Figure 4.8. With a junction defined as a node where more than two components or cells meet. On the systems' network (Figure 4.3) three pressure loops can be identified for the SWH-, ASHP- and ring-main loop, respectively:

$$0 = \Delta P_2 + \Delta P_{12} + \Delta P_{swh} + \Delta P_{swh-pump} + \Delta P_{13} + \Delta P_4 + \Delta P_3 \quad (4.59)$$

$$0 = \Delta P_3 + \Delta P_4 + \Delta P_5 + \Delta P_6 + \Delta P_7 + \Delta P_9 + \Delta P_{ASHP} + \Delta P_{ASHP-pump} + \Delta P_8 + \Delta P_1 + \Delta P_2 \quad (4.60)$$

$$0 = \Delta P_3 + \Delta P_4 + \Delta P_5 + \Delta P_6 + \Delta P_7 + \Delta P_{10} + \Delta P_{11} + \Delta P_{ring-main-pump} \quad (4.61)$$

With the pressure rise given by the pumps as ⁸:

$$\Delta P_{pump} = \rho g (a_1 Q^2 + a_2 Q^1 + a_3) \quad (4.62)$$

The resistances in the various components are approximated by the Darcy-Weisbach equation ⁹:

$$\Delta P = \frac{fL}{D_i} \frac{\rho}{2} \left(\frac{Q}{\left(\frac{D_i}{2}\right)\pi} \right)^2 \quad (4.63)$$

The additional equations ensure that the system of equations can be solved.

The calculation methodology conducted in EES for the Euler simulations can be divided into the following steps:

1. The heat losses are calculated for all components based on the internal energies calculated in the previous time step or initial conditions. The mass flow that enters and leave the system is enforced.
2. The ASHP and SWH controllers determine (based on the current boundary and system conditions) the mass fluxes that needs to be enforced through these components and guesses a friction coefficient (f) accordingly.
3. Based on Kirchoff's circuit laws, the pump and resistive curves in the three loops (Equations 4.59, 4.60 and 4.61) are solved along with the advection terms in all components. This step generally requires iterative calculation, even if the ASHP and SWH are not active ¹⁰.
4. Consequently, based on the mass flow distribution calculated, the new temperatures for all components can be determined.
5. The mass flow enforced through the ASHP is variable and dependant on inlet conditions (T_8). Therefore iterative calculations are required to solve the network, and steps two to four are repeated until convergence for the current time step is obtained.

⁸The pump curves in the detailed simulation are approximated by 2nd order parabolas.

⁹The ASHP and SWH component varies the friction factor f to maintain fixed mass flow conditions as specified by the controller routines of these components.

¹⁰In EES, the maximum number of iterations were set at 2500, with the relative residual limit set at $< 1e - 9$, and finally, the change in variables limit was set at $< 1e - 9$.

6. The 1st law energy flux calculations are conducted. Current temperatures are noted, and the steps above are repeated for the next time-step. The process is repeated until all time steps have been simulated.

The methodology, as described above, is illustrated in Figure 4.9. With the fundamentals equations established the process of verification will be discussed in the following section.

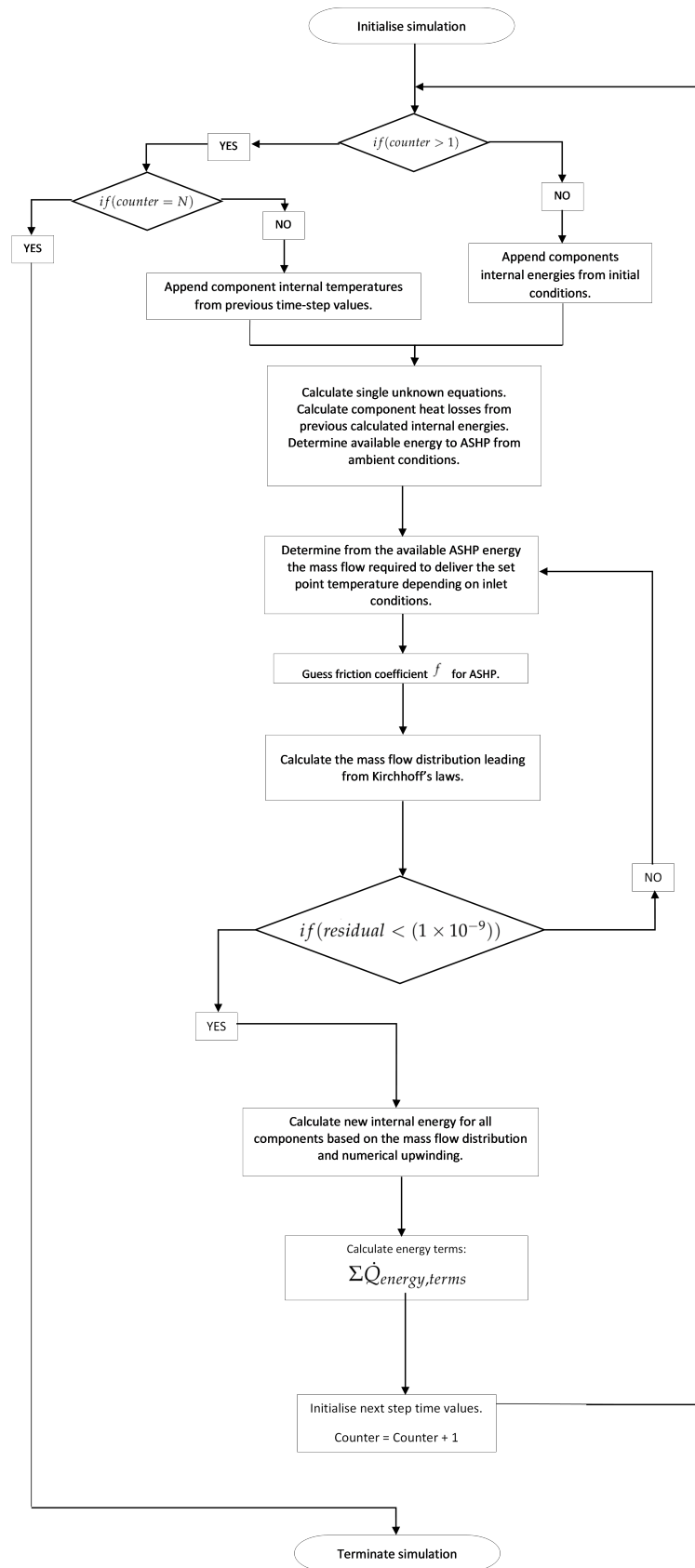


FIGURE 4.9: Flow chart of detailed computer model.

4.3 Verification

Verification is the process of performing several tests and checks before the software is tested and validated in its real-world application. For this study, the verification of the simulation programs involved the analyses of the mathematical topics of convergence, stability and consistency. Hereafter, the developed simulations were compared against a MATLAB® Simulink® model with a few case studies. The comparison to the experimental results will be discussed at the end of this Chapter 4.12.

4.3.1 Convergence, stability and consistency

For any numerical problem, the question of stability, convergence and consistency arises. The study will use thermodynamic and established fluid flow methods to address these problems with the developed simulations. As a quick overview to the reader; the developed simulation should satisfy the following conditions; furthermore let τ_i be the local truncation error at time-step i and M be the number of time-steps:

- Convergent if $\lim_{h \rightarrow 0} |x(t_i) - \dot{x}(t_i)| = 0$, hold as the approximation in the interval $[1, M]$.
- The stability can loosely be defined as the ability of the numerical method to keep small perturbations in the initial data (ϵ) as subsequent small changes ($f(\epsilon)$) in the following approximations as time progresses $t \rightarrow \infty$.
- The method is consistent if the $\lim_{h \rightarrow 0} \max |\tau_i| = 0$ holds for the differential equation with $i \in [1, M]$.

4.3.1.1 Convergence

From the first law of thermodynamics, energy has to be conserved. This study will use the first law to test the convergence of the numerical methods implemented. From Figure 3.19 and Equation 4.2 it can be seen that the net energy flux for the system fluctuates during the course of the day. Leading from the fact that due to advection ($\int e\rho\mathbf{V} \cdot \mathbf{n}dA_{cs}$), external heat addition or removal ($\frac{D}{Dt} \int e\rho dV_{cv}$), energy is being extracted or added to the thermal capacitance ($\frac{\partial}{\partial t} \int e\rho dV_{cv}$) of the system control volume.

If one takes the average of the net energy flux added to and removed from the system ($\Sigma\dot{Q}_{net} = \frac{D}{Dt} \int e\rho dV_{cv} - \int e\rho\mathbf{V} \cdot \mathbf{n}dA_{cs}$), over an extended period ($t \rightarrow M$), then $\frac{\Sigma\dot{Q}_{net}}{n}$ should come close to the steady-state of the system. Thus as the time-step approaches a finite time M , in k time-steps then the following should hold from thermodynamics:

$$\frac{\partial}{\partial t} \int e\rho dV_{cv} = 0 \quad (4.64)$$

With the use of different quotients and the approximation of partial derivatives; errors are inevitable, and Equation 4.64 can be rewritten as:

$$\frac{1}{k} \sum_1^k \frac{e_i}{h} \rho dV_{cv} \approx 0 \quad i \in \{1, 2..k\} \quad (4.65)$$

Consider a case where Equation 4.65 holds to be true. Furthermore, consider that for some reason at a particular time-step, local convergence is violated. By the mathematical principle of *reductio ad absurdum*, global convergence would consequently be impossible, and this would violate Equation 4.65 and the first law of thermodynamics. Thus, if it can be shown that the averaged net energy rate approach zero as time advances, then the numerical method and problem should be convergent and local convergence is implied.

Figure 4.10 illustrates the averaged net energy flux evaluated over the experimental period. For the RK-MOL simulation, the averaged energy flux was calculated with time-steps of 10, 30, 60 and 300 [s], respectively. The Euler simulation was evaluated with a time-step of 1 [s]. From Figure 4.10, it can be seen that for the RK-MOL simulation, the averaged flux parameter converged close to $-80[\text{W}]$ over the experimental period. The figure helped to indicate that for this particular simulation, with the chosen initial conditions, time step and numerical method, global convergence was obtained.

For the Euler method, only a single time step was evaluated (1 [s]). The average net energy converged to under $200[\text{W}]$ at the end of the period with a steadily decline. The slow convergence of the Euler method to that of the RK-MOL simulation is that it has 20 components where thermal capacitance was taken into account compared to six components. Finally, the possibility of introducing an error by choosing the incorrect initial temperature is exaggerated by more thermal capacity components. From the analyses presented above, the simulations appear to converge to the proper solution.

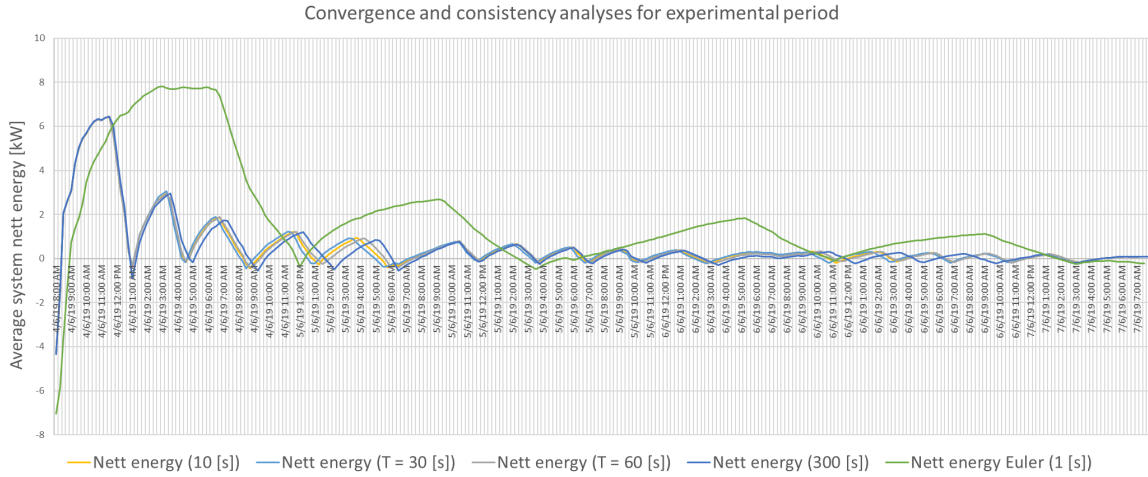


FIGURE 4.10: Convergence and consistency analyses for RK-MOL model.

4.3.1.2 Stability

The advection terms in the energy equation should be bounded to ensure stability and convergence. The famous CFL (*Courant–Friedrichs–Lewy*) condition is used to ensure that the advection terms ($\int \rho \mathbf{v} \cdot \mathbf{n} dA_{cs}$) do not cause the higher-order Taylor terms of the thermal capacitance to become unbounded. The simulations in this study use fixed time steps and control volumes in both the Euler and RK-MOL simulations.

Consider a pipe component of volume $\forall$ and density ρ with simulation time step δt . Furthermore, consider a steady mass flow of $\dot{m}$ flowing through the pipe control volume. If the case

is observed where external energy flux is negligible ($\frac{D}{Dt} \int e \rho dV_{cv} \approx 0$), then the ratio of the energy equation for this control volume reduces to:

$$\begin{aligned} \frac{D}{Dt} \int e \rho dV_{cv} &= \frac{\partial}{\partial t} \int e \rho dV_{cv} + \int e \rho \mathbf{V} \cdot \mathbf{n} dA \\ 0 &= \frac{\partial}{\partial t} \int e \rho dV_{cv} + \int e \rho \mathbf{V} \cdot \mathbf{n} dA \\ \frac{\partial}{\partial t} \int e \rho dV_{cv} &= - \int e \rho \mathbf{V} \cdot \mathbf{n} dA \\ Cp \frac{T_{i+1} - T_i}{\delta t} \rho V_{cv} &= - \dot{m} Cp (T_{in} - T_{out}) \end{aligned}$$

$$(\rho V_{cv}) Cp (T_{i+1} - T_i) = (-\dot{m} \delta t) Cp (T_{in} - T_{out}) \quad (4.66)$$

With the CFL condition given by:

$$CFL \equiv \frac{|u| \delta t}{\delta x} \leq 1 \quad (4.67)$$

It can be shown that by combining Equations 4.66 and 4.67, the following can be obtained:

$$\frac{Cp |T_{i+1} - T_i|}{Cp (T_{out} - T_{in})} = \frac{(\dot{m} \delta t)}{(\rho V_{cv})} \ll 1 \quad (4.68)$$

The above states that the amount of mass that flows through a control volume in a given time step should be less than the total mass of the element. Table 4.8 illustrates the CFL conditions at the components with the smallest thermal capacitance and highest expected mass flows. In the RK-MOL simulation, this is the storage vessel (1500 [kg]). In the detailed simulation, this is the T-section components (5 [kg]). For the worst-case mass flow, the maximum delivery of the ASHP is taken (0.522 [L s⁻¹]) to calculate the maximum CFL conditions.

Apart from the CFL condition, the time steps have to be chosen to ensure that the control system can be simulated practically. Taking too large time steps will lead to the erratic operation that can traverse to the unstable simulation of the system. Therefore, to satisfy both requirements of CFL and control system simulation, for the RK-MOL simulation, a time step of 60 [s] was chosen and for the Euler simulation a time step of 1 [s] was used.

TABLE 4.8: The maximum CFL conditions for the simulations evaluated with various time-steps.

CFL		
δt [s]	RK - MOL simulation	Detailed/Euler simulation
1	0.0003	0.1044
10	0.0035	1.044
30	0.0104	3.132
60	0.0209	6.264
120	0.0418	12.528

4.3.1.3 Consistent

Consistency for one-step methods implies that an error between the approximation of the numerical method and the true differential equation will approach zero as the time-step is reduced ($\lim_{h \rightarrow 0} \max |\tau_i| = 0$). As the exact solution to the true differential equation is difficult to obtain;

an alternative had to be devised. For multistep methods such as the RK-MOL model, several approximations are made at each time step. These numerous approximations exacerbate perturbation errors if mishandled. Therefore one would want the multistep error to reduce to zero as the time-step is reduced ($h \rightarrow 0$). Accordingly, the time step was varied along the figures introduced in Table 4.8 for the reduced simulation ¹¹. From here, the average of the system net energy flux ($\frac{1}{k} \sum_1^k \frac{e_i}{h} \rho dV_{cv}$) was calculated at each time step. If it could be shown that this term approaches zero as the time step is reduced, then consistency could be proven.

From Figure 4.10, the results of the proposed analyses are showcased. It can be seen that for the RK-MOL simulation, all of the time-steps led to a net average energy flux of $\lesssim 80[W]$ over the three days. Smaller time steps did not only lead to quicker convergence but also a net flux value closer to zero for the RK-MOL simulation. With convergence, stability and consistency proved for the simulations; the comparison to a commercial code could be conducted.

4.3.2 MATLAB verification

Test models were modelled in MATLAB® Simulink® to verify the mathematical approach followed in the previous sections. These models are presented in Appendix I. Three test cases were used for comparison with the RK-MOL simulation model and the Euler simulation model. Steady-state boundary conditions were enforced onto these case studies. The three case studies where as follows:

- Case 1 - Warm water is pumped through the system with the ASHP and SWH not active and no heat loss on the system pipes.
- Case 2 - Warm water is pumped through the system with the ASHP and SWH not active and heat loss on all system components.
- Case 3 - Water is extracted from the system with the ASHP and SWH operational as well as heat loss on all system components.

The boundary conditions for the various cases are reported in Table 4.9.

TABLE 4.9: Test case conditions for the validation simulations

<i>Description</i>	<i>Parameter</i>	<i>Case 1</i>	<i>Case 2</i>	<i>Case 3</i>	<i>Unit</i>
System mass in	$\dot{m}_{system,in}$	0.3	1	1	$[kg\ s^{-1}]$
System temperature in	$T_{system,in}$	20	20	20	$[^{\circ}C]$
Ambient dry bulb temperature	T_{db}	11.91	11.91	11.91	$[^{\circ}C]$
Ambient wet bulb temperature	T_{wb}	0.33	0.33	0.33	$[^{\circ}C]$
Relative humidity	RH	49.41	49.41	49.41	$[\%]$
Initial system temperature	$T_{i,system}$	11.91	11.91	11.91	$[^{\circ}C]$
Initial tank temperatures	$T_{i,tank} i1..5$	11.91	11.91	11.91	$[^{\circ}C]$
Irradiance	G	0	0	850	$[W/m^2]$
Heat loss active	UA_{system}	0	1	1	$[dim]$
ASHP active	$ASHP_{active}$	0	1	1	$[dim]$
SWH active	SWH_{active}	0	1	1	$[dim]$

¹¹For the detailed simulations as from Table 4.8 the CFL conditions are violated at most of the time steps, and 1 [s] was the uppermost time step used for the detailed simulation.

From the results of the case studies (Figure 4.11, Table 4.13), one can note that the steady-state temperatures seem to correspond between the Simulink, RK-MOL and the Detailed models. Table 4.13 showcases a temperature comparison for the storage vessels between the different models at 10 000 [s] intervals. For the sake of verification, it is assumed that the MATLAB Simulink models resemble the true system solutions with the Detailed and RK-MOL simulations compared consequently. Table 4.11 presents the errors of the RK-MOL and Euler simulations to that of the MATLAB Model.

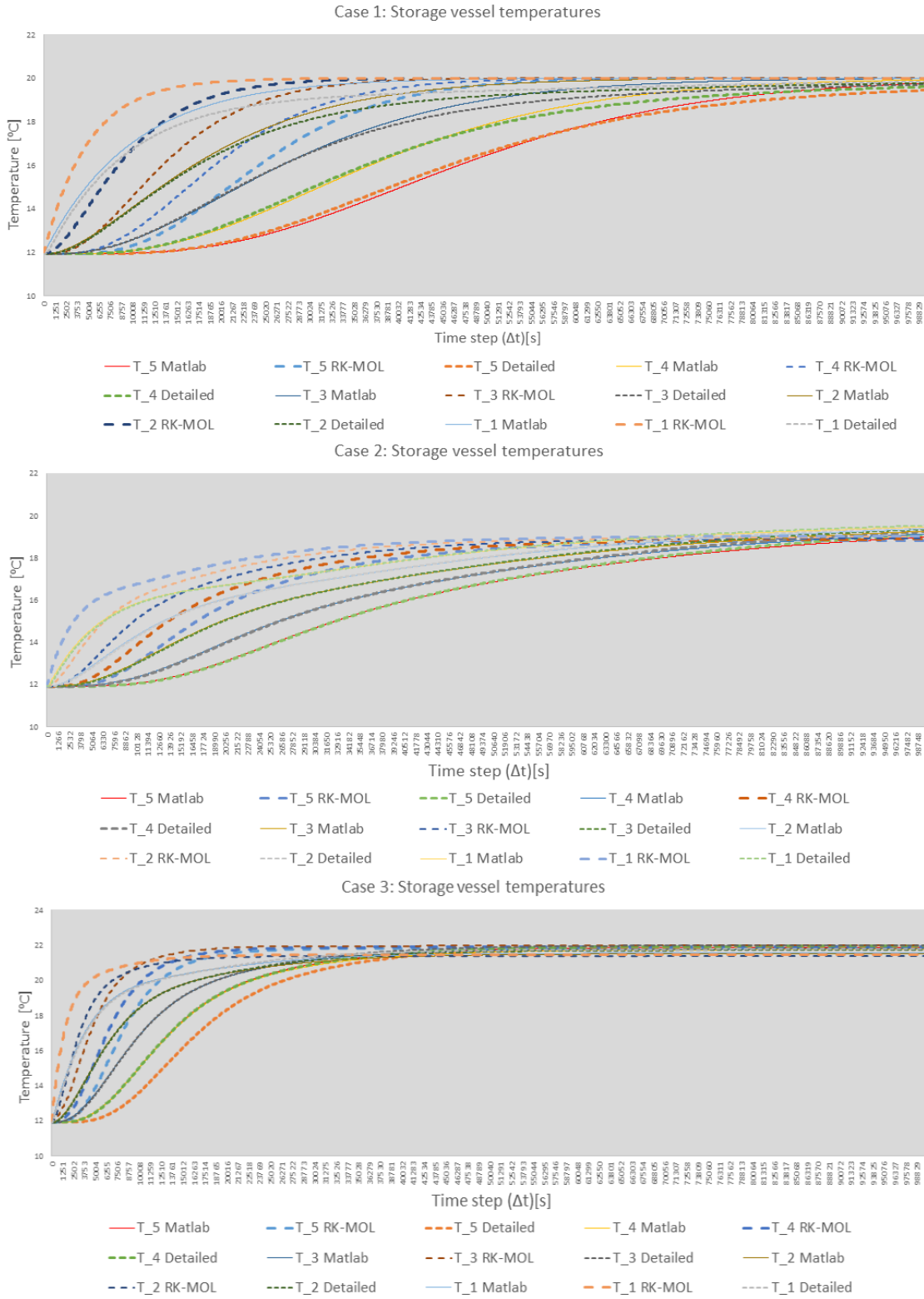


FIGURE 4.11: Storage vessel temperature comparison on case studies.

From Figure 4.11, it can be noted that the models converge approximately to the same temperatures. The RK-MOL simulation showcased a steady-state error of between 2.13 and -4.13 [%] ($\Rightarrow$ 0.49[°C] to -1.53[°C]), whilst the detailed model had errors of between 1.99 and -1.02 [%] ($\Rightarrow$ 0.43[°C] to -0.22[°C]). However, it can be noted that the RK-MOL seems to reach steady-state temperatures quicker than the detailed and Simulink models. The RK-MOL simulation had a maximum temperature error of -30.24 [%] ($\Rightarrow$ -3.88[°C]), and the Euler model had a maximum of 9.27 [%] ($\Rightarrow$ 1.39[°C]). These errors were present in the transient advection-dominated regions of the case studies.

The RK-MOL's quick convergence to steady-state conditions can be circumscribed to the fact that thermal capacitance is neglected on most components (except that of the storage vessel and the SWH). Contrastingly, the detailed and Simulink models have to heat the thermal mass of the pipes along with that of the storage vessels. The additional components consequently led to the delay in convergence. The case studies illustrate conditions where some system temperatures (T_5) is increased by a significant amount ($\Delta T \approx 8$ [°C]). Thus, the additional heat-up of thermal capacitance is of significant energy expense. During the first half of the simulations, the advection terms dominate the transfer of energy with heat transfer playing a minor role. One can note that the detail simulation appears to follow the MATLAB simulation consistently during the advection-dominated region.

Table 4.10 lists the simulation time and time step of the simulations on each case study¹². It can be noted that the detailed simulation is by far the slowest of all the simulation in all scenarios. Apart from taking the thermal capacitance of all the components into account, the simulation used pump curves and the linear momentum equation to determine the flow throughout the network. However, the above is not uncommon in commercial software packages. The MATLAB Simulink model was able to simulate the system quicker than the detailed model in all case studies taking the same detail into account. The difference lies in the numerical method employed. A fixed time step and an explicit Euler method were used in this study for the detailed simulation. Contrastingly, Matlab Simulink allows for dynamic time stepping and employs numerical methods with high orders of convergence. The advanced numerical methods allow for the economic simulation of the system. However, on all simulations, the RK-MOL simulates the quickest of the simulations. This aids in showing that even under the increased complexity of the third case study; the RK-MOL simulation was able to simulate the quickest of all the compared simulations.

TABLE 4.10: Simulation time and time step comparison for the case study data set.

		RK-MOL	Detailed	MATLAB model
Case study 1	Simulation time [s]	121	37 467	328.09
	Time step size [s]	60	1	<i>Dynamic</i>
Case study 2	Simulation time [s]	128.5	38 387	352.72
	Time step size [s]	60	1	<i>Dynamic</i>
Case study 3	Simulation time [s]	123.3	38 461	3775.32
	Time step size [s]	60	1	<i>Dynamic</i>

¹²All simulations in this chapter were conducted on a laptop with the following specifications: Intel Core i7-3632QM @ 2.2 GHz CPU, 12 GB DDR 3 1600 [Hz] RAM, Windows 10 Pro 64-bit operating system, EES Professional 32 Bit and MATLAB 64 Bit.

TABLE 4.11: Storage vessel temperature prediction errors relative to MATLAB.

	Timestep [s]	T1 [°C]		T2 [°C]		T3 [°C]		T4 [°C]		T5 [°C]	
		Euler	RK-MOL	Euler	RK-MOL	Euler	RK-MOL	Euler	RK-MOL	Euler	RK-MOL
Case 1	10000	1.76	-10.99	0.32	-19.07	-0.10	-15.79	-0.09	-7.34	-0.03	-2.94
	20000	2.41	-4.91	0.80	-15.16	-0.39	-24.48	-0.72	-24.10	-0.51	-19.19
	30000	2.84	-1.89	1.66	-7.91	0.17	-17.56	-0.93	-25.72	-1.26	-30.24
	40000	2.91	-0.70	2.28	-3.66	1.06	-9.96	-0.35	-18.48	-1.39	-27.63
	50000	2.69	-0.26	2.53	-1.61	1.77	-5.13	0.54	-11.25	-0.80	-19.74
	60000	2.31	-0.10	2.45	-0.69	2.14	-2.52	1.30	-6.30	0.07	-12.47
	70000	1.86	-0.04	2.16	-0.29	2.17	-1.19	1.75	-3.35	0.86	-7.34
	80000	1.44	-0.01	1.79	-0.12	1.99	-0.55	1.89	-1.71	1.38	-4.13
	90000	1.08	-0.01	1.41	-0.05	1.69	-0.25	1.80	-0.85	1.60	-2.24
	100000	0.80	0.00	1.08	-0.02	1.37	-0.11	1.58	-0.41	1.60	-1.18
Case 2	10000	0.17	-5.84	0.33	-12.48	0.31	-15.67	0.18	-12.08	0.08	-7.80
	20000	0.01	-6.30	0.05	-8.65	0.14	-13.25	0.19	-17.45	0.17	-20.31
	30000	-0.03	-6.50	-0.05	-7.85	-0.03	-10.12	0.01	-13.37	0.06	-17.89
	40000	-0.09	-4.83	-0.11	-6.26	-0.13	-7.95	-0.12	-10.06	-0.09	-13.24
	50000	-0.16	-2.98	-0.19	-4.20	-0.21	-5.65	-0.22	-7.30	-0.21	-9.53
	60000	-0.23	-1.45	-0.26	-2.36	-0.29	-3.48	-0.31	-4.81	-0.31	-6.48
	70000	-0.28	-0.23	-0.32	-0.88	-0.36	-1.69	-0.39	-2.69	-0.41	-3.95
	80000	-0.33	0.75	-0.38	0.29	-0.42	-0.29	-0.46	-1.01	-0.49	-1.94
	90000	-0.37	1.53	-0.42	1.22	-0.47	0.81	-0.52	0.30	-0.56	-0.36
	100000	-0.40	2.13	-0.46	1.94	-0.52	1.67	-0.57	1.31	-0.62	0.85
Case 3	10000	0.23	-6.37	0.05	-12.39	-0.06	-23.83	-0.19	-32.19	9.27	-24.20
	20000	-0.23	-3.30	-0.15	-4.53	-0.13	-8.26	-0.27	-12.04	5.62	-10.83
	30000	-0.64	-1.08	-0.57	-1.61	-0.45	-3.84	-0.42	-4.96	1.99	-4.14
	40000	-0.83	-0.14	-0.80	-0.28	-0.70	-1.99	-0.66	-2.37	0.78	-1.51
	50000	-0.93	0.22	-0.91	0.24	-0.82	-1.26	-0.81	-1.33	0.27	-0.42
	60000	-0.97	0.36	-0.97	0.43	-0.89	-0.97	-0.88	-0.93	0.06	0.01
	70000	-1.00	0.42	-1.00	0.51	-0.92	-0.86	-0.92	-0.77	-0.04	0.18
	80000	-1.01	0.44	-1.01	0.54	-0.93	-0.82	-0.93	-0.71	-0.08	0.25
	90000	-1.01	0.45	-1.01	0.56	-0.94	-0.80	-0.94	-0.69	-0.09	0.28
	100000	-1.01	0.45	-1.02	0.56	-0.94	-0.80	-0.95	-0.68	-0.10	0.29

4.4 Validation

Validation was achieved by comparing energy fluxes as predicted by the simulations to that obtained from the experimental period. The results and relative errors are presented in Table 4.12.

TABLE 4.12: Average energy comparison for test period between simulations and experimental data

	Experimental data		Reduced simulation		Detailed simulation	
	Energy [W]		Energy [W] Error [%]		Energy [W] Error [%]	
Tank losses	3436.25		3739.18	8.82	3415.52	0.60
Pipe losses	773.21		800.08	3.48	769.92	0.42
User load	6640.94		6626.45	0.22	6844.59	3.06
Ringmain load	7365.40		7481.74	1.58	7225.0	1.90
Solar flux	643.64		647.30	0.57	639.176	0.69
Solar pipe losses	645.91		662.37	2.55	613.66	4.99
HP flux	18710.60		18598.64	0.60	17991.29	3.84
Energy deficit	492.54		-63.89	N/A	-238.22	N/A
Average error [%]				2.54		2.22

The RK-MOL model compares relatively well with a global averaged error of 2.54 [%]. The Euler model compared slightly better than the RK-MOL simulation, with a global averaged

error of 2.22 [%]. The ASHP flux showcased an error of 0.6 [%] and 3.84 [%] for the RK-MOL and detailed simulations, respectively. The same ASHP regression equation was used for both the experimental analyses and the simulations. These commonalities help explain the relatively small error.

The user energy consumption is a function of the mass flow extracted, the hot water supply temperature and municipal inlet temperature. Two of these variables are fixed as simulation inputs from the experimental data. The supply temperature is a function of the simulation model dynamics. With limited simulation derived variables, a comparatively small error of 0.22 [%] and 3.06 [%] for the RK-MOL and detailed simulations were obtained, respectively.

The solar flux was measured directly on the experiment with the simulation using a regression graph. The error was indicating that the regression graph does predict proper SWH performance. An averaged error of 0.57 [%] and 0.69 [%] was obtained for the RK-MOL and detailed simulations, respectively.

The ring-main energy flux is a function of the mass flow, supply temperature and heat loss characteristics. With the mass flow as input to the simulation, the heat loss coefficient and the input temperature would determinate the ring main energy flux. As previously mentioned. The supply temperature is predicted accurately as the user energy load is accurately predicted. The ring-main average errors were 1.58 [%], and 1.9 [%] was obtained for the RK-MOL and detailed simulations, respectively.

Storage vessel and system pipe losses worked on the same averaged temperature stratification assumptions from the experimental data (Chapter 3.4.1.2). The heat loss coefficient is calculated in Appendix B, and the dry bulb temperature is measured from ambient conditions. The averaged errors were 8.85 [%] and 0.6 [%] for the storage vessel losses on the RK-MOL and detailed model, respectively. For the storage pipes, the averaged errors were 3.48 [%] and 0.42 [%], respectively. One could note from these measurements that the detailed model was able to model stratification more effectively than the RK-MOL simulation. The discrepancy in temperature prediction can only be explained by the temperature evolution of the boundary conditions of the coldest and warmest storage vessel. With linear momentum and thermal capacitance neglected in the RK-MOL simulation; a difference would have been expected.

The variable that will lead to a difference in the solar energy losses will be the temperature in the SWH manifold. Average errors of was 2.55 [%], and 4.99 [%] were obtained for the RK-MOL and detailed simulations respectively. The temperature difference can be explained by the method in which the SWH control system was switched on due to the difference in predicted tank temperatures between the simulation programs.

4.5 Concluding remarks

It is evident from the results of this chapter that model reducing measures can lead to significant reductions in simulation times. The RK-MOL simulation was able to simulate the case studies 2.71–, 2.74 and 30.61 times quicker than the MATLAB Simulink simulations, respectively. It is, however, indisputable from the case studies that this reduction in simulation time occurred with a handicap. Even though the simulations attain roughly the same steady-state temperatures (suggesting steady-state convergence as discussed in Section 4.3.1.1), the transient start-up led to discrepancies in temperature prediction.

The discrepancies can be explained by the assumptions made in the RK-MOL simulations. The deviation is observable once components that are assumed quasi-steady have to undergo a large temperature differential ¹³. The RK-MOL simulation will under-predict the effect of the thermal inertia of quasi-steady components on the components with thermal capacitance (non-'quasi-steady'). Furthermore, this illustrates that for start-up conditions, the reduced model will not be able to simulate the system effectively. It is clear, however, that the simulation can simulate transient conditions once the system has reached its operational temperature band. Supporting this claim is the validation results obtained from the experimental data and the net energy flux that was attained. However, the significance in the choice of the initial conditions is of paramount importance. Not only from a numerical standpoint; but also to simulate the dynamics and control system effectively for short periods of simulation time.

The Euler simulation was able to follow the Simulink model during transient start-up and steady-state conditions. However, the simulation was noticeably slower than MATLAB. This fact can easily be explained by the numerical method used and the fixed time step that offers a slower simulation time. What this comparison helps in illustrating is that even with advanced numerical methods, detailed simulations have long simulation times. These simulations do, however, offer detailed analyses during start-up conditions and conditions that will lead to large temperature differentials in the components. Still, once these operational temperatures have been reached; then an efficient simulation like the RK-MOL simulation can be utilised to simulate the system. To conclude, the RK-MOL simulation allows designers to evaluate control systems and component specifications quickly once the feasibility of start-up conditions have been established by means of a detailed simulation.

This chapter illustrated the theory, constraints and assumptions under which the simulations were developed. The verification of the various simulations was also illustrated with mathematical principles and compared with case studies to commercial code. Finally, the simulations were validated against experimental data, and the discrepancies were discussed. With the feasibility of the simulations and the limits of application established, the RK-MOL will be implemented in parametric studies in the following Chapter 5.

¹³In other words, a condition where advection dominated flow would lead to high-temperature differentials in components where thermal capacitance is neglected.

TABLE 4.13: Case study storage vessel temperatures as presented in Figure 4.11. Temperatures values are presented in 10 000 [s] intervals from 0 to 100 000 [s]

	Timestep [s]	T1 [°C]			T2 [°C]			T3 [°C]			T4 [°C]			T5 [°C]		
		MATLAB	Euler	RK-MOL	MATLAB	Euler	RK-MOL	MATLAB	Euler	RK-MOL	MATLAB	Euler	RK-MOL	MATLAB	Euler	RK-MOL
Case 1	10000	17.02	16.73	18.90	14.05	14.00	16.73	12.56	12.57	14.54	12.06	12.07	12.95	11.94	11.94	12.29
	20000	18.91	18.45	19.83	16.72	16.58	19.25	14.53	14.58	18.08	13.07	13.16	16.22	12.34	12.40	14.70
	30000	19.60	19.04	19.97	18.39	18.09	19.85	16.58	16.55	19.49	14.77	14.91	18.57	13.41	13.58	17.46
	40000	19.85	19.28	19.99	19.26	18.82	19.97	18.08	17.89	19.88	16.50	16.56	19.55	14.92	15.13	19.04
	50000	19.95	19.41	20.00	19.68	19.18	19.99	19.00	18.66	19.97	17.86	17.77	19.87	16.44	16.58	19.69
	60000	19.98	19.52	20.00	19.86	19.37	20.00	19.50	19.09	19.99	18.78	18.54	19.97	17.70	17.69	19.91
	70000	19.99	19.62	20.00	19.94	19.51	20.00	19.76	19.33	20.00	19.34	19.01	19.99	18.61	18.45	19.98
	80000	20.00	19.71	20.00	19.98	19.62	20.00	19.89	19.49	20.00	19.66	19.29	20.00	19.20	18.94	19.99
	90000	20.00	19.78	20.00	19.99	19.71	20.00	19.95	19.61	20.00	19.83	19.47	20.00	19.56	19.25	20.00
	100000	20.00	19.84	20.00	20.00	19.78	20.00	19.98	19.71	20.00	19.92	19.60	20.00	19.77	19.45	20.00
Case 2	10000	15.81	15.79	16.74	14.27	14.22	16.05	13.01	12.96	15.04	12.32	12.29	13.80	12.04	12.03	12.97
	20000	16.70	16.69	17.75	16.03	16.02	17.41	15.02	15.00	17.01	13.95	13.92	16.38	13.06	13.04	15.71
	30000	17.26	17.26	18.38	16.85	16.85	18.17	16.28	16.28	17.92	15.50	15.50	17.58	14.61	14.60	17.22
	40000	17.84	17.85	18.70	17.47	17.49	18.56	17.06	17.08	18.41	16.54	16.56	18.20	15.89	15.90	17.99
	50000	18.32	18.35	18.87	18.01	18.05	18.77	17.67	17.70	18.67	17.27	17.31	18.53	16.80	16.83	18.40
	60000	18.68	18.72	18.95	18.44	18.49	18.88	18.17	18.22	18.80	17.85	17.90	18.70	17.48	17.53	18.61
	70000	18.95	19.01	19.00	18.77	18.83	18.93	18.55	18.62	18.87	18.30	18.37	18.79	18.01	18.08	18.72
	80000	19.17	19.23	19.02	19.02	19.09	18.96	18.85	18.93	18.90	18.65	18.74	18.84	18.42	18.51	18.78
	90000	19.33	19.40	19.04	19.21	19.29	18.98	19.08	19.17	18.92	18.92	19.02	18.87	18.74	18.84	18.81
	100000	19.46	19.53	19.04	19.36	19.45	18.99	19.25	19.35	18.93	19.13	19.24	18.88	18.99	19.10	18.82
Case 3	10000	19.75	19.70	21.01	18.42	18.41	20.70	16.85	16.86	20.87	15.01	15.04	19.84	15.01	13.62	18.65
	20000	20.73	20.78	21.42	20.40	20.43	21.32	20.20	20.22	21.86	19.37	19.43	21.71	19.42	18.33	21.52
	30000	21.23	21.37	21.46	21.05	21.17	21.39	21.15	21.25	21.97	20.85	20.94	21.89	20.94	20.52	21.80
	40000	21.44	21.62	21.47	21.34	21.51	21.40	21.55	21.70	21.98	21.40	21.54	21.91	21.51	21.34	21.83
	50000	21.51	21.71	21.47	21.45	21.64	21.40	21.71	21.89	21.98	21.62	21.79	21.91	21.75	21.69	21.84
	60000	21.54	21.75	21.47	21.49	21.70	21.40	21.77	21.96	21.98	21.71	21.90	21.91	21.84	21.83	21.84
	70000	21.56	21.77	21.47	21.51	21.72	21.40	21.79	21.99	21.98	21.74	21.94	21.91	21.88	21.88	21.84
	80000	21.56	21.78	21.47	21.52	21.73	21.40	21.80	22.00	21.98	21.75	21.96	21.91	21.89	21.91	21.84
	90000	21.56	21.78	21.47	21.52	21.74	21.40	21.80	22.01	21.98	21.76	21.96	21.91	21.90	21.92	21.84
	100000	21.56	21.78	21.47	21.52	21.74	21.40	21.81	22.01	21.98	21.76	21.97	21.91	21.90	21.92	21.84

Chapter 5

Simulation implementation

The goal of this chapter is to assess the influence of a few design variables on annual hybrid SWH/ASHP system electricity consumption and ring-main supply temperatures. The simulations presented in Chapter 4 was verified and validated against a commercial code and experimental data presented in Chapter 4.3.2 and 4.4, respectively. Consequently, the RK-MOL simulation, due to its swift simulation times, will be applied to evaluate design scenarios with various year-long simulations. The simulation times were noted to obtain insight into the computational expense of conducting these parametric studies. The most prominent design variables that will influence system performance was identified from the experimental results presented in Chapter 3.6.3 and will be discussed in the following paragraphs.

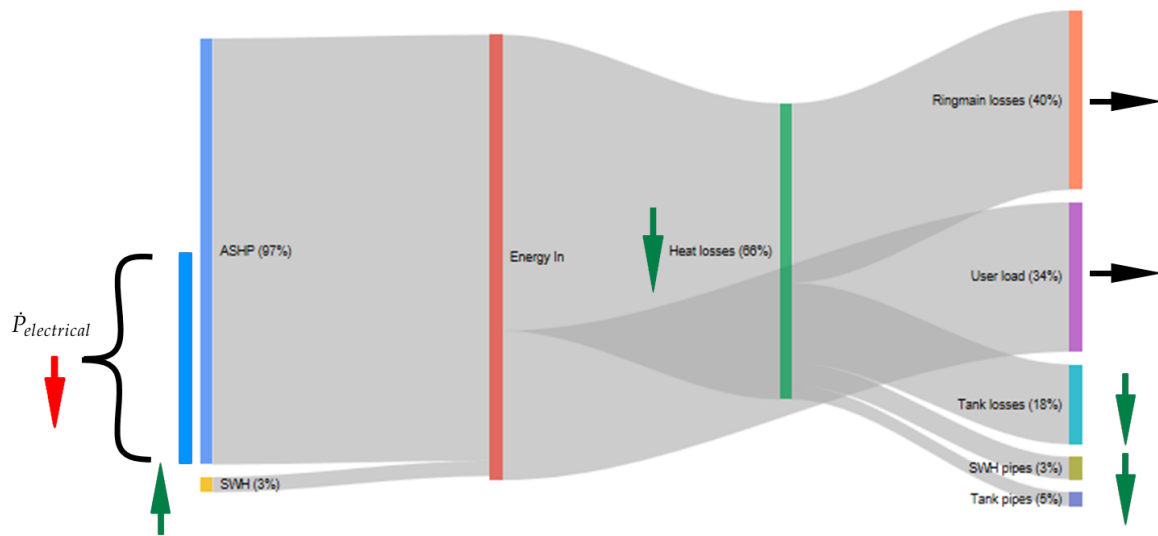


FIGURE 5.1: Sankey diagram illustrating energy flux distribution and possible areas of improvement.

Referring to the Sankey diagram, Figure 5.1 (altered from Figure 3.17 of Chapter 3); one can argue that for the same boundary conditions, advances in system performance can be attained by the reduction of a few energy expenditures. For this study, an improvement in system performance entails the supply of hot water in the target temperature band (55 - 60 [°C]) and at lower electrical consumption.

Focusing on the energy expenditures of the Sankey diagram (Figure 5.1), the relatively constant energy fluxes are indicated with the black arrows. The ring-main, user, and system pipe losses cannot be improved significantly due to the design principle of 'hot water on tap'. Furthermore, the thermal insulation on these systems was of client specification and insulated to a reasonable degree, reducing the possibility of further improvements. Finally, the energy expenditure on user consumption is a function of the hospital hot water demand and cannot be altered.

In the Sankey-diagram (Figure 5.1), the most plausible areas of improvement are indicated by the green arrows. The third-largest energy expenditure on the system is the storage vessel losses. Altering the volume and aspect ratio of the storage vessels can lead to a reduction in energy load on the system. Apart from the reduction in thermal capacitance, the heat losses of the storage vessels can be reduced.

Additionally, a more efficient control strategy for the SWH could be implemented. Leading to a reduction in SWH losses and an increase in solar thermal contribution. Finally, the size or number of SWH's can be increased to reduce the systems' reliance on the ASHP, and possibly in turn, a reduction in electrical consumption.

The electrical consumption of the ASHP is a fraction of the total ASHP energy addition and is indicated by the bracket and red arrow in Figure 5.1. The electrical consumption is governed by the COP of the ASHP, as discussed in the previous Chapters (3.5 and 4.1.2.1). A smaller rated ASHP might have a lower initial capital cost and lead to a reduction in electrical consumption. In the following Section 5.1, the suggested improvements will be investigated using a few parametric studies.

5.1 Parametric study

Apart from the alteration of system components, the influence of weather conditions on the base-line system was studied. The parameters that were altered to study the effect thereof included:

- Ambient conditions (5.1.1) - Analyses on the influence of temperature, humidity, irradiance and atmospheric pressure.
- Storage vessel (5.1.2) - Variation on the capacity of the storage vessels and their aspect ratios.
- ASHP size (5.1.3) - Variation on the rating of the ASHP.
- Solar field size and control system (5.1.4) - Increase in the number of SWH banks connected to the system and alter its control system and flow rate.

For all the parametric studies presented in this section, the following conditions are assumed constant:

- User consumption - The average user consumption profile from the experimental period is unchanged throughout the year.
- Indoor hospital temperature - Indoor temperatures are assumed constant throughout the year, and seasonal variations are neglected.
- Component configuration - The components interconnection, as presented in Figure 3.2, will remain the same.

- Weather - Apart from the ambient condition parametric study (5.1.1) presented in the following section, all weather conditions in this chapter is based on ambient conditions in Pretoria, South-Africa.

5.1.1 Effect of different climate-zones

Design year data of six Southern-African cities was obtained from the Council of Science and Industrial Research (CSIR) and used as simulation input. A fixed step size of 60 [s] was used for the year simulations (leading to 525 600 time steps). A notebook was able to simulate, simultaneously, the six cities' in 14 hours¹. All other system components were kept as per the original system based in Potchefstroom. The cities that were used for the parametric study includes:

- Pretoria (PTA)
- Windhoek (WHK)
- Cape Town (CPT)
- Bloemfontein (BFN)
- Durban (DBN)
- Port Elizabeth (PE)

This list includes three cities near the coast (Cape Town, Durban and Port Elizabeth), and cities with semi-arid (Windhoek and Bloemfontein) and subtropical climates (Pretoria). The averaged ambient conditions for the six cities are tabulated in Table 5.1.

A study by Li et al. showcased that ASHP performance is dependant on the atmospheric pressure and consequent altitude of the installation (Li et al., 2017). The study found that the high altitudes and cold temperatures of Tibet led to a 9.5 - 12 [%] reduction in COP in comparison to coastal regions (based on non-frosting conditions). The reduction in COP can be circumscribed due to the lower density of the air at higher altitudes, reducing the heat transfer potential. Consequently, to compensate for altitude variations, this study scaled the ASHP COP (Equation 3.32) depending on the atmospheric pressure. The atmospheric pressure for the cities that were studied is listed in Table 5.1. The average annual energy fluxes for the simulations are showcased in Table 5.2.

TABLE 5.1: Average design year data obtained from the CSIR on various Southern-African cities.

Annual ambient conditions	PTA	WHK	CPT	BFN	DBN	PE
Dry bulb temperature T_{db} [°C]	16.98	19.34	15.93	15.22	20.2	17.05
Wet bulb temperature T_{wb} [°C]	11.84	9.489	13.15	9.896	17.35	14.56
Relative humidity RH [%]	60.68	30.82	76.5	59.67	77.03	78.46
Solar irradiance G [W/m ²]	430.72	490.15	409.28	457.12	355.2	377.47
Lowest elevation z [m]	1 339	1 655	0	1 395	0	0
Atmospheric pressure P_{atm} [kPa]	87.33	83.06	101.46	85.59	101.06	101.05

¹All simulations in this chapter were executed on a notebook with the following specifications: AMD @ 2.2 GHz CPU, 8 GB DDR 4 1600 [Hz] RAM, Windows 10 Pro 64-bit operating system, EES Professional 32 Bit

TABLE 5.2: Energy flux and ASHP performance comparison on simulation results of various Southern-African cities.

System variable	PTA	WHK	CPT	BFN	DBN	PE
<i>Energy fluxes [W]</i>						
ASHP flux	16 905.44	16 549.4	17 027.14	16 984.94	16 593.17	16 948.0
ASHP electrical	6 809.78	5 833.4	6 281.04	7 911.86	5 279.0	5 960.94
SWH flux	797.15	1 062.55	765.58	859.47	580.31	661.26
Storage vessel losses	3 129.63	2 927.98	3 220.20	3 277.67	2 854.92	3 124.27
Storage pipe losses	669.64	626.54	688.99	701.80	610.91	668.49
SWH pipe losses	709.50	872.82	681.33	730.92	527.01	619.41
User load flux	6 136.5	6 128.92	6 135.45	6 153.12	6 129.05	6 133.71
Ring-main losses	7 283.78	7 270.72	7 285.54	7 287.10	7 275.86	7 285.30
<i>ASHP performance</i>						
COP_{ashp} [dim]	2.48	2.84	2.71	2.15	3.14	2.84

It can be seen from Tables (5.1 and 5.2) that the fluxes that showed the most extensive variation between the different climates were the ASHP electrical consumption, SWH and storage vessel energy fluxes. Cities with the highest dry bulb temperatures had the lowest heat losses in the pipes and storage vessels. The reduction in heat losses can be explained by the smaller temperature differential between the bulk fluid temperature throughout the system and ambient conditions. Higher solar irradiance led to high solar flux values and, consequently, high SWH losses. The ring-main losses remain relatively constant as the indoor building temperature was assumed the same for the different scenarios. The user load flux remained relatively the same since the user consumption was assumed constant between all the studied cities.

The ASHP flux remained relatively constant throughout the various cities, with notable exceptions for Windhoek and Durban. These cities had the lowest heat input requirement due to lower heat losses. The electrical energy consumed by the ASHP gives insight into the COP obtained for the different regions. The ASHP tended to consume less energy in regions with higher dry bulb temperatures (Windhoek, Durban and Port-Elizabeth). A drastic reduction in COP was observed for Bloemfontein. Bloemfontein has the lowest dry bulb temperatures, along with the second-highest elevation of the inspected dataset. These conditions consequently led to unfavourable operating conditions for the AHSP compared to the other cities that were studied.

Pretoria and Port Elizabeth have roughly the same dry bulb temperatures (16.98 [°C] and 17.05 [°C] respectively), but the relative humidity differs by 17.78 [%] and the atmospheric pressure by 13.72 [kPa]. Taking these differences into account, one notes that even though the load between these cities are roughly the same, the ASHP consumed 848 [W] more electricity in Pretoria than in Port Elizabeth. The increased air density of the coastal regions will inevitably lead to improved ASHP performance. Durban had the highest annual dry bulb temperatures of all cities studied, and with the increased air density, the lowest electricity consumption. The higher relative humidity at the coastal regions did not amount to the same impact on COP as the air density difference and dry bulb temperatures. For the dataset, Windhoek has the second highest annual dry bulb temperatures (19.34[°C]), the lowest relative humidity (30.82 [%]) and the highest elevation (1 655 [m]) of all the studied cities. Windhoek, due to its desert climate, had high dry bulb temperatures and consequently favourable COP conditions even with

the relatively high elevation of the city.

The results presented in the paragraphs above help to illustrate that climatic zones can have a significant impact on ASHP-SWH hybrid systems. The largest deviation in electrical consumption observed in the city dataset was nearly 3 [kW] between Durban and Bloemfontein. The design of these systems is, therefore, directly dependant on the ambient conditions and will directly impact the choice of sub-components with the lowest energy consumption. The sections to follow will focus on the variation of component specifications based on ambient conditions for Pretoria, South-Africa.

5.1.2 Storage vessel capacity and aspect ratio

From Chapters 3.6.2 and 4.4, it was noted that the storage vessels was the second-largest system heat loss and accounted for roughly 18 [%] of total energy expenditure. By decreasing the storage vessel energy load (without compromising warm water supply during peak periods) would lead to energy savings.

The storage vessel capacity was varied between 2 and 4 [kL] in increments of 500 [L] to test the hypothesis. The stratified storage vessel arrangement was kept at five separate tanks with only the tank capacities and aspect ratios varied. Stratification ensures that the heat loss is minimised and the SWH and ASHP work with the appropriate inlet temperatures². Table 5.3 lists the different storage vessel capacities used as the primary input to this parametric study. For the 2, 2.5, 3.5 and 4 [kL] storage vessels, the geometry was specified to minimize the area to volume ratio for the specified volume³. However, the 3 [kL] storage vessel dimensions were kept the same for this parametric study as obtained from the original system as presented in Chapter 3.

TABLE 5.3: Storage vessel capacities and aspect ratios.

Tank parameters	2000 [L]	2500 [L]	3000 [L]	3500 [L]	4000 [L]
Radius [m]	0.683	0.736	0.61	0.823	0.860
Height [m]	1.366	1.471	2.566	1.646	1.721
Area [m ²]	8.757	10.197	12.174	12.761	13.949
Volume [m ³]	2.00	2.50	3.00	3.50	4.00
Area to volume ratio [m ⁻¹]	4.39	4.07	4.06	3.647	3.49
Heat loss coefficient [W/K × m ²]	1.39	1.393	1.394	1.398	1.399

From Table 5.4, the results of the parametric study are tabulated. A decrease in the storage vessel volume led to a decrease in the load of the ASHP. The heat loss of the smaller storage vessels was less than that of the larger storage vessels. The maximum and minimum delivery temperatures remained relatively the same. These temperatures indicated that even during peak user load, the system was able to supply warm water as per the performance of the original system. By reducing the storage vessels to a capacity of 2 [kL] from 3 [kL], the electrical consumption was reduced by 389 [W] ($\Rightarrow$ 5.72[%]), the storage vessel losses and ASHP energy

²The SWH efficiency is increased by working with colder inlet temperatures and the COP of the ASHP is increased by a reduction in the inlet temperatures.

³Define the area to volume ratio (AVR) as $AVR = A_{area} / \forall$ where $\forall = h\pi R^2$ and $A_{area} = 2\pi(R^2 + Rh)$. The minimum AVR can be obtained from $\frac{dAVR}{dr}(\frac{2\pi R^2}{\forall} + \frac{2}{R}) = 0$. With the minimum radius corresponding to the minimum AVR given by $R = (0.5\frac{\forall}{\pi})^{1/3}$.

fluxes were reduced by 873 [W] ($\Rightarrow$ 27.92[%]) of storage vessel thermal energy. The COP of the ASHP remains relatively the same. The constant COP indicated that colder inlet conditions are not present and that the ASHP can charge all of the storage vessels to the original design intent. The ASHP annual running hours decrease by 3 [%] if the storage vessel capacity is reduced from 2 [kL] to 3 [kL]. This helps to illustrate that the buffer capacity might have been overspecified for the original system.

TABLE 5.4: Energy flux values, supply temperatures and ASHP performance obtained from storage vessel parametric study.

System variable	2000 [L]	2500 [L]	3000 [L]	3500 [L]	4000 [L]
<i>Energy fluxes [W]</i>					
ASHP flux	16 025.94	16 406.36	16 905.44	17 106.34	17 399.81
ASHP electrical consumption	6 420.0	6 584.59	6 809.78	6 869.43	6 909.31
SWH flux	750.89	766.43	797.15	784.49	735.14
Storage vessel losses	2 255.86	2 621.51	3 129.63	3 293.45	3 605.9
Storage pipe losses	668.68	668.67	669.64	669.76	670.57
SWH pipe losses	647.94	674.31	709.5	656.13	642.65
Ring-main losses	7 258.88	7 253.45	7 283.78	7 280.65	7 288.43
<i>Temperatures [°C]</i>					
Maximum supply	58.79	58.79	58.74	58.77	58.69
Minimum supply	50.28	50.28	50.97	50.7	51.52
Average supply	56.12	56.09	56.23	56.22	56.25
<i>ASHP performance</i>					
Annual utilization [%]	50.36	51.13	53.31	53.87	54.63
COP_{ashp} [dim]	2.5	2.5	2.48	2.49	2.518

5.1.3 ASHP capacity

The rating of the ASHP determines the maximum mass flow that can be supplied to the system at any moment. However, the maximum delivery flow should not only be larger than the worst-case scenario user mass flow. Due to the variability of ambient conditions, the ASHP will not be able to supply its' maximum heat rating at all times. Therefore, the ASHP should be able to meet user demand even under the worst environmental conditions and most demanding user consumption. If the ASHP cannot attain the requirement, then the buffer storage should aid the ASHP in supplying water in the specified temperature range. In this parametric study, the ASHP rating will be varied to study the effects of delivery temperatures and electricity consumption on the original system. The ASHP was scaled by adjusting the electrical consumption regression equations derived for the 90 [kW] ASHP of the original case study. It was assumed that the COP of the smaller machines was the same as the larger machines for the same ambient conditions.

TABLE 5.5: Energy flux, delivery temperatures and ASHP performance from the ASHP parametric study.

System variable	35 [kW]	45 [kW]	55 [kW]	70 [kW]	90 [kW]
<i>Energy fluxes [kW]</i>					
ASHP flux	15 308.46	17 112.4	17 604.82	17 186.48	16 905.44
ASHP electrical consumption	5 738.45	6 597.52	6 974.44	6 938.28	6 809.75
SWH flux	504.24	485.785	464.74	702.44	797.15
Storage vessel losses	2 072.52	2 612.06	3 135.36	3 138.38	3 129.63
Storage pipe losses	440.82	560.51	671.44	671.77	669.64
SWH pipe losses	451.83	449.11	444.47	640.197	709.5
User load flux	5 856.33	6 369.87	6 335.62	6 224.2	6136.5
Ring-main losses	7 104.8	7 618.19	7 532.18	7 375.84	7 283.78
<i>Temperatures [°C]</i>					
Minimum ring-main	41.8	53.02	50.37	50.59	50.97
Average ring-main	55.4	57.77	57.38	56.66	56.23
Maximum ring-main	57.82	58.47	58.79	58.78	58.74
Standard deviation ring-main	2.298	0.2414	1.386	1.865	1.906
Average top tank (T_5)	34.6	54.29	58.51	58.69	58.66
<i>ASHP performance</i>					
Annual utilization [%]	99.99	99.99	89.59	68.02	53.31
COP_{ashp} [dim]	2.67	2.59	2.52	2.48	2.48

Table 5.5 lists the results obtained for this study by varying the rating of the ASHP. The ring-main minimum supply temperatures fall below 50 [°C] for ASHP ratings below 45 [kW]. Maximum temperatures remain relatively the same for all of the ASHP ratings except the 35 [kW] unit. Notably, the standard deviation on the ring-main supply temperatures for an ASHP unit under 45 [kW] is significant. These figures indicate that the buffer storage vessels are not able to aid the smaller ASHP units in delivering the rated supply temperatures.

Furthermore, the average temperature of the warmest vessel indicates that smaller ASHP units cannot adequately charge the 3 [kL] storage vessels. The storage vessel and system pipe losses are lower due to the temperatures in the storage vessels being maintained at lower temperatures. Furthermore, the COP of the smallest unit is the largest as it draws water from the bottom tank. The tank is maintained at lower temperatures and consequently improves the COP of the ASHP. The annual ASHP utilisation indicates that the 35 and 45 [kW] units appear to be operational for all hours of the year. From a maintenance standpoint, continuous operation will not be ideal. The 45 [kW] unit, whilst able to keep ring-main temperatures in the target supply temperature band, cannot load the buffer tanks. This indicates that the unit is ideally sized if the continuous operation would not have been a problem. The larger units have large standard deviations since the control system switches these units of and on according to tank temperatures. The largest ASHP unit has the lowest energy consumption for ASHP units that can deliver target temperatures (55, 70 and 90 [kW]). As these units have roughly the same COP; the maximum delivery heat determines which system will be the most economical for part-load conditions. Thus, the 90 [kW] ASHP unit appears to be sized correctly for the user profile and components selected for the hospital system.

5.1.4 Solar field size and control system

By increasing the number of SWH banks in series, an increase in outlet temperature can be obtained. By increasing the number of SWH banks in parallel, the flow-rate can be increased, but higher pump losses are to be expected. From Chapter 3.6.3, it can be recalled that high temperatures (up to 120 [°C]) were measured in the SWH manifold. It can be concluded that by adding more SWH banks in series, the results would lead to a decrease in SWH performance, including the following:

- Higher heat losses
- Lower SWH efficiency due to higher internal temperatures
- Increase in exergy generated by the SWH

Therefore, the study will assess the impact of adding additional SWH banks in parallel. Leading to possible higher solar contributions. However, the results from Chapter 3.6.3 and 4.4 indicate that the SWH contribution and SWH pipe losses are relatively the same over the 24 [h] period. Thus, additional banks in parallel will not increase system performance significantly, and the same observation will be made. Therefore, the energy contribution of a single SWH bank has to be improved.

Consequently, the SWH controller was adjusted. The maximum setpoint temperature was lowered along with the activation (8 [°C]) and deactivation temperature (2 [°C]) differentials⁴. By adjusting these temperatures, the high temperature built up in the SWH can be decreased. Lowering the heat losses of the SWH and increasing the number of times the SWH activates by contributing useful energy to the storage vessels. The flow-rate was reduced to 0.05 [[L s⁻¹]], to decrease the on-off control of the SWH and allow the SWH to operate continuously. Lower temperatures and slower flow-rates should improve ASHP utilisation.

All parameters on the original system will be used as input apart from the lowering of the maximum SWH activation and deactivation temperatures and flow rate. Additionally, the number of banks was increased from one to six to evaluate the scalability of additional banks. The results of the parametric study are tabulated in Table 5.6.

⁴With the SWH control temperatures and differentials as presented in Section 3.2.2.

TABLE 5.6: Energy flux values and ASHP performance obtained from SWH bank parametric study.

System variable [W]	No of SWH banks in parralel					
	1	2	3	4	5	6
ASHP flux	16 704.9	16 569.2	16 222.63	15 810.5	15 731.88	15 913.84
ASHP electrical	6 752.95	6 676.23	6 597.2	6 483.13	6 481.03	6 536.62
Storage vessel losses	3 122.44	3 108.52	3 094.86	3 072.2	3 068.07	3 084.8
Storage pipe losses	668.13	665.09	662.09	657.16	656.22	659.85
User load flux	6 143.84	6 137.13	6 115.25	6 055.36	6 047.13	6 080.47
Ring-main losses	7 263.67	7 213.53	7 166.25	7 090.32	7 071.38	7 111.63
Solar energy	1 158.38	2 245.77	3 335.25	4 442.37	5 555.33	6 638.68
SWH manifold losses	362.8	724.24	1 086.57	1 384.09	1 852.15	2 445.19
SWH flux	796.3	1 521.53	2 248.68	3 058.28	3 706.18	4 193.5
SWH pipe losses	494.1	1 022.82	1 581.11	2 131.50	2 730.35	3 315.58
<i>ASHP performance</i>						
COP_{ashp} [dim]	2.47	2.48	2.46	2.44	2.42	2.43

From this study, it can be noted that by increasing the number of SWH banks from one to six, the ASHP flux is reduced by 791 [W] ($\Rightarrow$ 4.73[%]). Furthermore, electrical consumption was reduced by 216 [W] ($\Rightarrow$ 3.2[%]). The reduction of the SWH control system set-point temperatures led to gains in solar contribution with the solar pipe losses decreasing by 215 [W] ($\Rightarrow$ 30.35[%]) for a single solar bank.

5.2 Concluding remarks

This chapter explored the use of the RK-MOL simulation through various parametric studies. A total of 22 year-long simulations were conducted with a 60 [s] fixed time step. All of the simulations were conducted in a 38 [h] period. These figures help signify the quick design analyses that can be conducted on major components of the system by using the reduced simulation. From the conclusions of Chapter 3.6.3, a few prominent observations were made regarding the design of the case study system. The following hypotheses were presented regarding possible improvements on the system and will be repeated here for the reader's convenience:

- The storage vessel volume is too large for the particular installation, and a smaller storage buffer would have been sufficient.
- The ASHP have been oversized for the particular installation, and a smaller machine would have been sufficient.
- The SWH system is underutilised due to extensive heat losses.

From the analyses in Section 5.1.2, it was noted that by decreasing the tank capacity from 3000 [L] to 2000 [L] the heat loss was reduced by nearly 27 [%] (873 [W]). The maximum and minimum supply temperatures remained relatively the same, indicating that a reduction in the storage buffer volume will decrease electrical consumption and will not compromise the water supply temperature. By decreasing the storage vessel volumes to 2 [kL], the ring-main would circulate the total storage vessel volume three times per day. However, the ring-main will drain stored thermal energy too quickly if the storage vessel capacity is reduced by a too large degree.

Therefore, a compromise between worst-case scenario user load and storage losses should be reached. To conclude, under the current user load, 90 [kW] ASHP unit and solar bank; smaller storage vessels could lead to energy savings of up to 5.72 [%] (389.78 [W]) and a reduction in running hours of 3 [%].

The ASHP appears to be sized correctly for the current application. The results of Section 5.1.3 illustrates that smaller heat pumps with a rating can meet the systems' current average supply temperature ($\approx 57^{\circ}\text{C}$). However, the 35 [kW] ASHP units showed a significant reduction in minimum supply temperatures (41.8°C). These temperatures correspond with unfavourable COP conditions or high user demand. Smaller ASHP units would require inline electrical resistive heaters to keep minimum supply temperatures above 50°C . From the standard deviation of the supply temperature and the heat losses on the storage vessels, it can be concluded that the smaller ASHP unit struggles to keep the temperatures constant throughout the system. However, it could be argued that a larger ASHP would lead to lower electrical consumption and consequent payback period than a smaller unit with electrical resistive heaters. Furthermore, the smaller units have an annual utilisation of nearly 100 [%]—the high percentage signals continuous operation. From a maintenance standpoint, this would not be ideal. The 90 [kW] unit displays the best energy consumption of the largest units (55, 70 and 90 [kW]) due to its part-time operation and larger compressor.

It has been suggested that the minimum activation temperature, deactivation and maximum temperatures have to be decreased to reduce the SWH pipe losses and the losses on the SWH header manifold. From Section 5.1.4, it could be seen that by implementing these strategies, a reduction in SWH pipe losses of up to 215 [W] for a single SWH bank was attainable. Compared to the original design year simulated for conditions in Pretoria (Section 5.1.1). Apart from the reduction in SWH losses, useful SWH energy was increased by the same amount. However, several SWH have to be added in parallel to offset the thermal energy addition of the ASHP over the 24 [h] period. The ASHP energy addition on the system is roughly 16.9 [kW] compared to 1.158 [kW] for a single solar bank. Even if the full thermal energy contribution of the ASHP is to be offset by solar, due to the ring-main system and variable irradiance, the heat pump cannot be entirely replaced. Apart from this, a notable finding is that even though ASHP thermal energy was reduced by a maximum of 0.719 [kW] (by increasing the number of solar banks to six), only a 0.216 [kW] reduction in electrical consumption was observed.

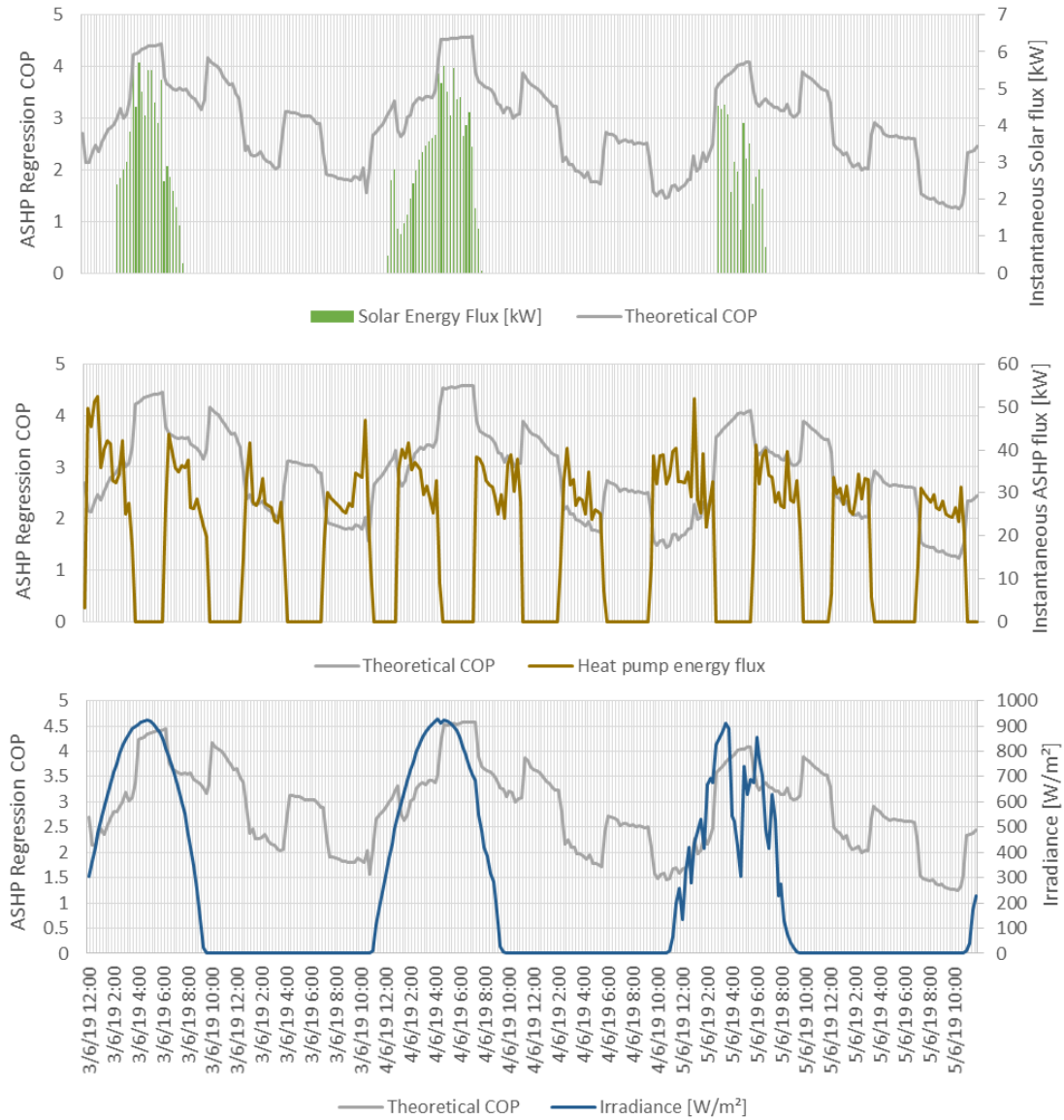


FIGURE 5.2: Simulation data on a three day period corresponding to a reduction in SWH control temperatures and one SWH bank.

Figure 5.2 illustrates data obtained from the simulation for a three day period. From Figure 5.2, the theoretical COP⁵ that could be obtained was plotted against the instantaneous solar energy, -ASHP energy, and irradiance. From Figure 5.2, it can be noted that high solar irradiance occurs at the same time that favourable COP conditions are predicted. Due to higher dry bulb temperatures during daylight hours; the ASHP uses less electricity during these hours to add thermal energy to the system. However, the SWH can only add useful energy during daylight hours. The total thermal load remains constant, and thus, a reduction in ASHP derived thermal energy have to be supplemented by solar derived thermal energy. As more SWH banks

⁵The serrations in predicted COP is due to the variation in inlet conditions of the ASHP. When the ASHP is operational, warmer water is obtained from the storage vessels. Consequently, an instant reduction in COP is observed.

are added, the ratio of solar to heat pump derived thermal energy is consequently altered. The ASHP is forced to add the remainder of the required thermal energy under unfavourable COP conditions. However, it could be argued that the ASHP under these conditions is still more efficient than the use of resistive heaters.

The addition of SWH technology poses a slight reduction in electrical consumption, the COP of the ASHP, and thermal energy derived from the ASHP. Even though additional SWH banks reduce electrical consumption, it does not reduce it as significantly in contrast to the thermal energy the SWH banks contribute. The impact of SWH banks would have been more profound on a system with pure resistive heaters. However, SWH systems contributions are more variable than ASHP, and cannot replace resistive heaters alone.

To conclude, the following observations were made with reference to the simulation results:

- Ambient conditions can influence ASHP electrical consumption up to 3 [kW] between different cities⁶.
- For relatively the same dry-bulb conditions, relative humidity and air pressure can influence ASHP electrical consumption by nearly 848 [W], on average, annually.
- A reduction in storage vessel volume can maintain set-point temperatures and decrease electrical consumption by 389 [W] ($\Rightarrow$ 5.72[%]).
- A 45 [kW] ASHP unit can lead to 300 [W] decrease in electrical consumption, but a decrease in minimum temperatures are to be expected. The unit cannot charge the storage vessels to set-point temperatures and will operate continuously.
- By decreasing set-point temperatures on the SWH controller, a 220 [W] ($\Rightarrow$ 30.35[%]) reduction in thermal energy losses can be attained for a single SWH bank.
- By increasing the number of SWH banks to six, a 216 [W] ($\Rightarrow$ 3.2[%]) electrical saving was observed.
- An increase in the number of SWH banks in parallel increases solar energy contribution and decreases thermal ASHP input. However, the ASHP is forced to add the rest of the thermal demand under less favourable COP conditions.

The following chapter will discuss the main findings of the study and recommendation for further study.

⁶Under the six cities that were studied in Section 5.1.1.

Chapter 6

Conclusion and Recommendations

This chapter will recollect the problem statement, goals and scope of the study. A discussion will follow on the findings of the various chapters.

6.1 Overview

Due to the rotation of the earth and climatic conditions, the available solar radiation, dry- and wet bulb temperatures along with humidity differs hourly. The latter poses a challenge to the designer of combined heat pump and solar water heating systems. The difficulty can be circumscribed to the fact that several factors influence the performance of such systems: The size and performance of the solar collectors, the capacity of the storage vessels, the hot water demand profile, the size and performance of the heat pump, ambient conditions and the control systems. The design and evaluation of the syntheses between the sub-components of such systems are therefore not straightforward.

Simulation programs are often used for the design and performance evaluation of hot water systems. Researchers and designers often use these simulations to design systems for maximum efficiency, user comfort, or even a combination thereof. The problem, however, with most of these component-based simulations is that they require specialist knowledge to calibrate realistically and are often computationally expensive. Furthermore, it can be challenging to obtain internal component variables that are required for the successful calibration of the simulation. A simulation that uses readily available manufacturer performance curves and matrices of sub-components diminishes the need for complicated calibration and reduce computational effort. These semi-empirical component models aid the designer to focus on the layout and sizing of components and control systems of these components.

Naturally, researchers and designers would aim to conduct parametric studies to evaluate the effect of design parameters on system performance. However, parametric studies in a limited number of independent parameters become near impossible when a single simulation can take days to complete. This problem becomes even more apparent when extended periods are simulated. The above consequently set the scene for the study.

From Chapter 1.2, the **problem statement** and **need for the study** were defined as:

"To better understand the synthesis of the various system components of a hybrid HP/SWH installation. There exists a need for a simulation model that uses readily available manufacturer data of solar, heat pump, and storage vessel sub-components for integrated, long-term, transient thermal system simulation. Furthermore, such a validated simulation model needs to be computationally inexpensive. Nevertheless, also simple to calibrate to enable extensive parametric and optimisation studies to be conducted."

Consequently, a comprehensive literature study was undertaken to develop a simulation adhering to the above requirements. The literature study was divided into six sections. Each of these sections focused on an aspect of the problem statement. These sections included heat pumps, solar water heaters, the combination of the former and latter, and water storage vessel simulations. Finally, methods to reduce the computational effort and typical simulation structures were also investigated.

Validated commercial-scale SWH/ASHP hot water models were scarce from literature. A notable finding was that discarding the linear momentum equations in specific thermal networks can lead to a drastic reduction in computational effort. A list of concluding remarks on each section was given. The lists assisted in highlighting the primary findings and some deficiencies from literature.

To successfully validate the developed simulation, a real-world SWH/ASHP hybrid system was analysed. The hybrid solar-heat pump heated water system of a hospital in Potchefstroom, South-Africa, was consequently studied. The system was monitored during the cold winter months of June-July (2019) using the systems' SCADA and some additional sensors. The period was chosen to strain the system to its design limits. An energy balance based on the various energy fluxes was presented and calculated accordingly.

Due to sensor shortages, a statistical model based on the manufacturer and experimental data was used to model the ASHP performance. From the statistical analyses, it was determined that relative humidity had an impact on the performance of the ASHP, and in turn, on the energy balance. Consequently, experimental data was used to compensate for the deficiency of relative humidity data proceeding from the OEM data. The statistical model that was used is a linear multivariate model with a strong positive correlation between COP and dry bulb temperature, a strong negative correlation for ASHP inlet temperature and a comparatively small negative correlation for relative humidity.

With the experimental data captured, a few conclusions on the systems' dynamics and design characteristics were made. ASHP energy flux contribution accounted for 97 [%] of energy entering the system, with the remainder supplied by the SWH. Notably, SWH contributions and losses were nearly the same in magnitude. This could be explained by the parameters of the SWH controller that led to significant heat losses. Ring-main losses and the user load accounted for 40 [%] and 34 [%] of energy expenditure, respectively. The remaining energy losses were made up by the storage vessel (18 [%]) and pipe losses (8 [%]). The system provided the case study and input for the developed simulations.

Two simulation models based on the hospital case study was developed in EES®. Furthermore, one simulation used reducing measures, whereas the other employed additional equations and was referred to as the detailed simulation. The assumptions, fundamental equations and control logic for the sub-components was presented. For the reduced simulation, the most notable assumption was the use of only the mass and energy conservation equations and the use of thermal capacitance only on some components. Contrastingly, the detailed simulation did take the linear momentum conservation into account along with thermal capacitance on all of the components. Consequently, the reduced simulation had 189 equations compared to the 1114 equations of the detailed simulation. The difference in the number of equations led to a significant difference in computational expense.

Both of the simulation models used a 1D finite difference method, with explicit numerical methods and fixed time steps. The reduced simulation used the method of lines along with a Runge-Kutte integration method. The detailed simulation used the explicit Euler method for

integration. Most of the loops in the system have a water pump. Consequently, many of the mass flows were known beforehand. The magnitude of these mass flows was based on the system conditions at the evaluated time-step and controller output. Thus, the reduced simulation was able to enforce mass flows explicitly. For the detailed simulation, additional equations had to be obtained in the form of Kirchoff's circuit laws, to solve the linear momentum equation for the system. The linear momentum equation consequently led to iterative calculations. The difference in the control logic and solving sequence between these simulations were illustrated.

The mathematical principles of convergence, stability and consistency were used to verify the reduced and detailed simulations. Convergence was tested by evaluating the systems' average net energy flux over an extended period. Consistency was assessed by varying the time-step of the simulations and evaluating the consequential impact on the net energy flux. Finally, for stability, the CFL condition was used to ensure that the advection flow in the various components would be bounded. Apart from the CFL conditions, the time-step was chosen to ensure that the control systems did not cause erratic behaviour. From the tested numerical methods, the simulations average net energy flux converged to under 80 [W] for the three-day experimental period.

To further verify the simulations, a third simulation in Matlab® Simulink® was developed. The developed simulation applied the same ASHP and SWH performance curves used in detail and reduced simulations. Consequently, three case studies were set up in Matlab Simulink. From these studies, the RK-MOL simulation produced steady-state errors between 2.13 and -4.13 [%] ($\Rightarrow$ 0.49[°C] to -1.53[°C]). The simulation reported a maximum error of -30.24 [%] ($\Rightarrow$ -3.88[°C]) in the most transient region after system start-up. Contrastingly, the detailed simulation produced steady-state errors of between 1.99 and -1.02 [%] ($\Rightarrow$ 0.43[°C] to -0.22[°C]). The detailed simulation reported a maximum error of 9.27 [%] ($\Rightarrow$ 1.39[°C]) during the transient regions. The RK-MOL simulation was, however, able to simulate the most detailed case study nearly 31 times quicker than the Simulink equivalent.

However, from the results, it was concluded that the RK-MOL simulation would not be able to simulate the systems during start-up conditions accurately. Due to the assumptions that were used, the thermal inertia of components with no thermal capacitance will have no impact on components where thermal capacitance are taken into account. Therefore, the use of the simulation is limited to systems where the mass flow is non-zero, known beforehand in the branches, and components do not undergo a large temperature differential ($\approx \Delta 8$ [°C]). Therefore, it was proposed to use the simulation to test system performance once operational temperatures had been reached. Within these constraints, the RK-MOL simulation has the potential to conduct long term simulations at drastically quickened time-frames. The detailed simulation was able to follow the Simulink model consistently. The detailed simulation did, however, have significant longer simulation times compared to the Simulink model. The longer simulation times could be circumscribed from the numerical method that was used.

With the simulations verified; the simulations were consequently validated against the experimental data. The reduced and detailed simulations showcased an average energy flux error of 2.54 [%] and 2.22 [%], respectively. With the reduced simulation validated, the simulation was consequently used in parametric studies to investigate the design of the case study and syntheses of SWH/ASHP hybrid systems.

The reduced simulation was able to simulate 22 design year scenarios, with a time-step of 60 [s] in a 38 [h] period on a standard notebook. For the parametric study, the variation in climatic

conditions, storage vessel-, ASHP capacity and alterations on the SWH were investigated. During these studies, a few findings were made. SWH/ASHP systems at the coastal cities consume a maximum of 3 [kW] less electricity than inland cities. The denser air and higher dry bulb temperatures led to a maximum increase of 24 [%] in ASHP COP. Higher dry bulb temperatures tended to raise COP conditions, and higher inlet temperatures decreased COP. Consistent with the statistical model derived for the ASHP.

Reducing the storage vessel size from 3 [kL] to 2 [kL] led to a 389 [W] ($\Rightarrow$ 5.72[%]) and 873 [W] ($\Rightarrow$ 27.92[%]) reduction in electrical and thermal energy consumption, respectively. With the reduction in storage vessel capacity, the delivery temperatures were not compromised. The reduced storage vessel capacity reduced the thermal load on the ASHP. Consequently, lowering electrical consumption. Lower rated ASHP units down to 45 [kW] can maintain ring-main temperatures in the design parameters. However, the smaller ASHP units' annual utilisation approached 100 [%]. These figures indicated that these machines work increasingly longer and nearly continuously to maintain ring-main temperatures. From a maintenance standpoint, the smaller ASHP units will require field services more often. Apart from maintenance, the smaller units (35 and 45 [kW]) are not able to load the storage vessels to the set-point temperatures.

From the experimental results, SWH contribution and losses were nearly the same. The problem was investigated utilising a parametric study where the SWH controller and flow-rate were altered. Consequently, a 220 [W] ($\Rightarrow$ 30.35[%]) reduction in thermal losses on the SWH pipes was observed. Following the improvement, the number of SWH banks in parallel was increased from one to six. A 791 [W] ($\Rightarrow$ 4.73[%]) reduction in thermal energy and a 216 [W] ($\Rightarrow$ 3.2[%]) reduction in electrical energy for the ASHP was observed. Finally, it was remarked from the predicted COP, that favourable COP conditions and peak solar irradiance overlapped. Thus, additional SWH banks reduce the thermal contribution of the ASHP over the 24 [h] period. The SWH bank will consequently charge the storage vessels during peak solar irradiance. One can recall that the ASHP activation temperature probe is situated near the two coldest storage vessels in the stratified tank arrangement. Thus, as the SWH charges the stratified storage vessels, the ASHP temperature probe will not cause the ASHP to activate. Consequently, the SWH pump and ASHP will not be active simultaneously. Ultimately, the ASHP is forced to add the remaining thermal energy demand during less favourable COP conditions.

6.2 Conclusions

From the problem statement, a few requirements can be identified for the simulation program:

- Make use of manufacturer performance maps or curves for the main system components.
- Computationally inexpensive.
- Make use of the simulation for a preliminary analysis on the synthesis of the various components the SWH/ASHP compromises of.
- Easy to calibrate and use for extensive parametric studies or optimisation studies.

The following paragraphs will aim to analyse in what manner the study was able to achieve the objectives or not.

One of the initial aims of the study was to use readily available manufacturer data to simulate components of the SWH/ASHP system. The SWH could be simulated using the SWH efficiency curve. The use of the curves ensured one could circumvent the need to use evacuated

heat pipe models that require extensive knowledge and data on the design of these components. However, the SWH efficiency curve has a limited range of applicability. By altering the number of tubes, the SWH efficiency curve is altered. Therefore, the use of these curves is restricted to the configuration of SWH banks in parallel. A detailed simulation can be used to calibrate the curves for steady-state conditions. The curve consequently is used in the reduced simulations as introduced in this study.

For the ASHP, the manufacturer data could be implemented in the computer simulation by using regression equations. However, the implementation could not be implemented seamlessly as the relative humidity was not varied over the full parameter space by the manufacturer. By using the experimental data, the regression equation could be adjusted to take the variation of relative humidity into account. Furthermore, using the regression equations, the modelling of the ASHP components such as the compressor, evaporator, condenser and capillary tube was circumvented. Most of the data on these components would be proprietary and not easily attainable for system designers. However, due to the limited amount of data most manufacturers publish on their ASHP, the implementation of regression equations is somewhat limited. Finally, the quality of the data the suppliers publish will ultimately define the regression equations, and consequently, the quality of the output derived from the reduced simulation.

From the results of the study, it can be concluded that the use of these regression curves and performance matrices reduces computational complexity. Furthermore, with the study neglecting the momentum equation, a noticeable reduction in simulation times was obtained. With a notebook capable of running six design year simulations simultaneously with a 60 [s] time-step. With these figures in mind, the feasibility of running optimisation studies have consequently been established. However, the decrease in simulation times came with a drawback. By neglecting thermal capacitance in the ring-main network and other system pipes, the simulation shows temperature discrepancies to the MATLAB simulation during start-up conditions. Consequently, the reduced simulation will only be able to simulate conditions where operational temperatures have been reached (and is reflected in the initial conditions). Finally, the accuracy of the reduced simulation can be improved by taking the thermal capacitance of the ring-main into account.

Information on the insulation properties of the storage vessels and system piping could easily be obtained. From the calibration standpoint, the use of regression curves limits the need for extensive calibration on sub-components. The simulation requires an ASHP performance map, SWH efficiency curve, insulation properties of pipes storage vessels, control system set-points, and finally, the design mass flows through the hydraulic loops.

The reduced simulation could quickly evaluate design syntheses between the components of the SWH/ASHP system. The simulation offers a wealth of data such as the impact on the change of parameters on the supply temperature, electricity usage, ASHP operational runtime, heat-losses and performance of the ASHP and SWH. Furthermore, swift simulation enables designers to evaluate control systems and their set-points with dynamic simulations. However, the effective use of the simulation is dependant on the quality of OEM data on components and the limit of its' use to non-start-up conditions.

6.3 Recommendation and future work

The reduced simulations within the limitations reported in the verifications section can significantly reduce computational effort on year-long simulations. The optimisation of the system

was not in the scope of the study. However, the possibility of using the developed simulation for this objective is noticeable. A simple example will be given in the case study to support this statement:

During parametric studies and optimisation, an objective function is generally defined. This can be to evaluate the systems annual electrical consumption ($P_{electrical}$), initial capital cost ($E_{initial}$) or user discomfort ($T_{discomfort}$). Sometimes a multi-objective objective function ($f_{objective-system}$) can be defined using a combination of these requirements such as:

$$f_{objective-system} = \gamma_1 P_{electrical} + \gamma_2 E_{initial} + \gamma_3 T_{discomfort} \quad (6.1)$$

Where γ_1, γ_2 and γ_3 are constants that assign weights to the various objectives. It can be said that by minimising $f_{objective-system}$, an 'optimum' with regards to the objectives can be attained. However, when optimising; the effect of varying a combination of independent variables on the objective function has to be evaluated. For this study, it can be the effect of increasing the number of ASHP in parallel, the capacity of the ASHP, and the number of SWH banks at the same time to evaluate the effect on system performance. By varying the number of SWH and ASHPs from one to five and the capacity of the ASHP in five ratings, a total of 125 simulations will have to be evaluated if a rectangular grid is employed. Thus each additional independent parameter significantly increases the number of simulations. Furthermore, increasing the number of independent variables may lead to the 'curse of dimensionality'. Per illustration, Figure 6.1 showcases a typical Pareto frontier obtained from the simulation with numerous system configurations that minimize Equation 6.1.

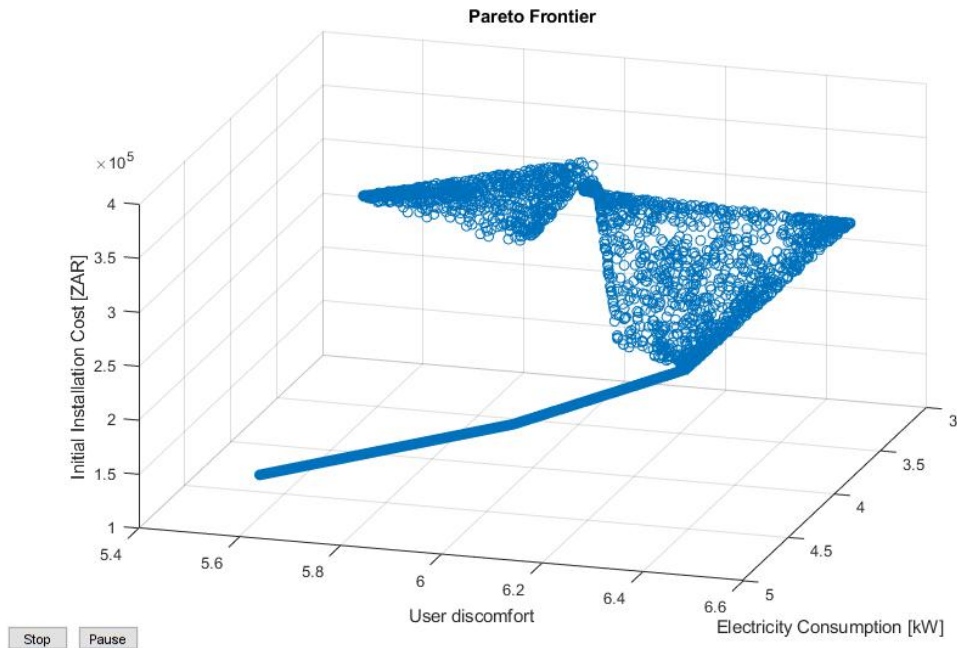


FIGURE 6.1: Pareto frontier example for multi-objective optimisation.

Per the example above, it becomes clear that for optimisation, a quick but accurate solver is required to evaluate a significant amount of simulations. Consequently, the simulation poses a

foundation for future work in the field of ASHP/SWH systems that could include the following:

- The predictions made by this simulation relies to a large degree on the ASHP performance map that was used as this study was limited by the data from the manufacturer to fit a statistical model. An experimental setup can be used to evaluate the performance of ASHP at various ambient conditions and relative humidities. The experiment will give insight into the impact of relative humidity at high dry-bulb temperatures on ASHP performance. From these conditions, a thorough statistical ASHP model can be derived that works in frosting, condensing and dry-air conditions with OEM system specifications. The electrical demand of the ASHP can be measured to assess the COP at these conditions.
- The reduced computer model can be used to test the viability of a smart ASHP controller. The controller can anticipate peak user consumption and consequently charge the system storage vessels accordingly. Possible electrical savings can be achieved if the system is charged under more favourable COP and irradiance conditions. The impact of the controller on warm water supply and electrical consumption can be scrutinised. Finally, the set-point temperatures and parameters can be optimised based on the output of the reduced simulation.
- A techno-economic study can be conducted on the payback period of ASHP/SWH hybrid systems by varying the ASHP, storage vessel, SWH bank capacities and controllers by making use of the reduced simulation.
- A full multi-objective, stochastic optimisation can be conducted in a limited amount of variables by using the reduced model. The reduced simulation can be used to derive a response surface that characterises the performance of the system. The regression equations can be used, in turn, in an optimisation routine. Consequently, a system design can be obtained adhering to the proposed cost-functions set out during optimisation.
- The storage vessels can be simulated by using CFD with various mass flows. The results obtained from CFD can be used to derive a data-driven model (such as statistical models). The derived regression equation can then be applied in the system simulation to offer CFD level detail at a reduced computational effort. This can help to evaluate stratification effects on heat losses and system performance with year-long simulations.
- Aspects of the reduced simulation can be converted into the object-orientated format as was practised in the development of the detail simulation.

Appendix A

Instrumentation

As part of the agreement with the hospital group, the system may not have been altered in any way. Therefore, the verification of the sensors on the SCADA network of the hospital could not be conducted. This appendix will present the verification of the additional sensors used by the study to capture the required data for simulation. Finally, at the end of the appendix, the study will discuss the propagation of errors on the energy fluxes. The appendix is divided into the following sections:

- Solar irradiance (A.1)
- Temperature measurement (A.2)
- Electrical current measurement (A.3)
- Water flow measurements (A.4)
- Propagation of errors in energy flux calculations (A.5)

A.1 Solar irradiance measurement

A CM11 pyranometer is used by the faculty solely for the purpose of calibration and verification of experimental pyranometer. Data on the pyranometer can be obtained from Table A.1. The study used an SP-215 pyranometer as simulation input for the experimental period. The CM11 and SP-215 have calibration uncertainties of 3 [%] and 5 [%], respectively. To ensure that the calibration constant used by the SP-215 is correct, it was compared to the CM11. For the SP-215, the following conversion is used and was obtained from the calibration sheet of the manufacturer to calculate irradiance:

$$I_{SP-215} = (V \times 1000[\frac{mV}{V}]) \times 0.5[\frac{W}{(mV \times m^2)}] \quad (A.1)$$

For the CM11 (reference pyranometer) the equation to obtain the irradiance is given by:

$$I_{CM11} = \frac{V}{4.97 \times 10^{-6}[\frac{V \times m^2}{W}]} \quad (A.2)$$

TABLE A.1: Technical data of single version CM11 with some indicators according to ISO 9060 classification.

<i>Performance indicator</i>	<i>Value</i>
Spectral range	310 to 2800nm(50[%] points) 340 to 2200nm(95[%] points)
Irradiance range	0 to 1400 [W/m ²] (max. 4000 [W/m ²])
Stability	< 0.5[%] per year
Resolution	< ± 1 [W/m ²]
Non-linearity	<0.5%
Spectral sensitivity	< 2[%]
Tilt response	< 0.2[%]
Temperature response	< 1[%] at (−10 [°C].. + 40 [°C])
Response time (95% reponse)	12[s]



FIGURE A.1: Pyranometer verification against reference pyranometer.

As showcased in Figure A.1, the SP-215 was connected to the U12-008 Hobo® logger, which in turn was connected to a notebook. The live data stream was written to a data file for comparison. The output of the CM11 was measured by means of a Fluke® 117 multimeter. The output measured by the multimeter was logged manually along with the time in a spreadsheet program. The measurements are noted in Table A.2 with the relative error of the SP-215 measured against the CM11. With the reported relative error below 4.5 [%] it was decided not to alter the calibration constant issued by the manufacturer as this error is between the maximum calibration uncertainties of both pyranometers.

TABLE A.2: Verification of SP-215 pyranometer against CM11 pyranometer as measured on the 07/05/2019.

<i>Time</i>	<i>Instrument</i>	<i>Voltage</i> [mV]	<i>Irradiance</i> [W/m ²]	<i>Relative error</i> [%]
10:44	SP-215	2200	550.0	2.38
	CM11	2.8	563.38	
10:50	SP-215	2241	560.25	0.56
	CM11	2.8	563.38	
10:51	SP-215	2263	565.75	3.04
	CM11	2.9	583.5	
10:51	SP-215	2314	578.5	4.16
	CM11	3.0	603.621	

A.2 Temperature measurement

The Hobo® loggers used TMC temperature probes manufactured by OnSet® to measure the additionally required temperatures for the study. Table A.3 tabulates the technical data of the TMP temperature probes. To verify the temperatures probes used during the study; three different tests were conducted. The tests where as follows:

- Steady-state air
- Ice water solution
- Warm water

The TMP probes were compared to that of the Fluke® 712 temperature calibrator (Table A.3). The results of the three tests are summarized in Table A.5. The first test that was undertaken was the steady-state air temperature test. The various probes were coiled together by a metal wire. Hereafter, they were left to normalise to steady-state room temperatures. From Tables A.3 and A.5 it can be noted that the temperature difference was 0.162 to 0.21 [°C] higher than the maximum measurement uncertainty of the temperature probe.

The second test was conducted in an ice water solution (Figure A.2). Water was added between the ice cubes to create a solution that would encompass the whole sensor. Theoretically, the temperature in this solution should be close to 0 [°C]. It can be seen that the TMP probes have an 0.15 – 0.3 [°C] error. With the first temperature probe error falling within its' measurement error range. The second temperature probe error falls between 0.102 - 0.142 [°C] from the maximum accuracy deviation for the TMP probe.

The last test was conducted in warm water to test the accuracy of the probes at high temperatures. It should be noted that these probes have a measurement range with a maximum of 50 [°C] in water. Consequently, the warm water test is at the upper limit of the temperature probe range. The probes were installed between the metal pipes and the insulation of the circulating pipes. Thus, the probes were never in direct contact with the fluid and rarely saw temperatures as high as that was measured during the test. It can be recalled that the probes could not be installed with direct contact to the water as no modifications were allowed on the hospital system.

From the temperature measurements, the error to that of the reference measurement varied between 0.854 – 1.027 [°C]. By taking the absolute error, it can be seen that the maximum relative error for Temp 1 was 1.5 [%], and Temp 2 was 1.39 [%]. To conclude for the TMP probes, a

measurement error of 1.39 [%] would have to be used for the mixing point probe as the probe is exposed to temperatures above 50 [°C], on occasion. For the municipal inlet, temperatures never exceed 50 [°C], and thus the standard measurement uncertainty as reported by the manufacturer can be used ¹.

TABLE A.3: Fluke®712 temperature calibrator data.

<i>Performance indicator</i>	<i>Value</i>
Range	−200°C to 800°C
Resolution	0.1 [°C]
Accuracy	0.33 [°C]

TABLE A.4: OnSet®TMP temperature probe data.

<i>Performance indicator</i>	<i>Value</i>
Measurement range in water	−40 to 50[°C]
Measurement range in air	−40 to 100 [°C]
Drift	< 0.1 [°C] per year
Resolution	< 0.002 at 25 [°C]
Response time in air	2 min to 90 [%] in 1 [m/s] air
Response time in stirred water	30 [s] to 90 [%] < 0.002 [°C]
Accuracy	±0.15 [°C] from 0 to 70 [°C]

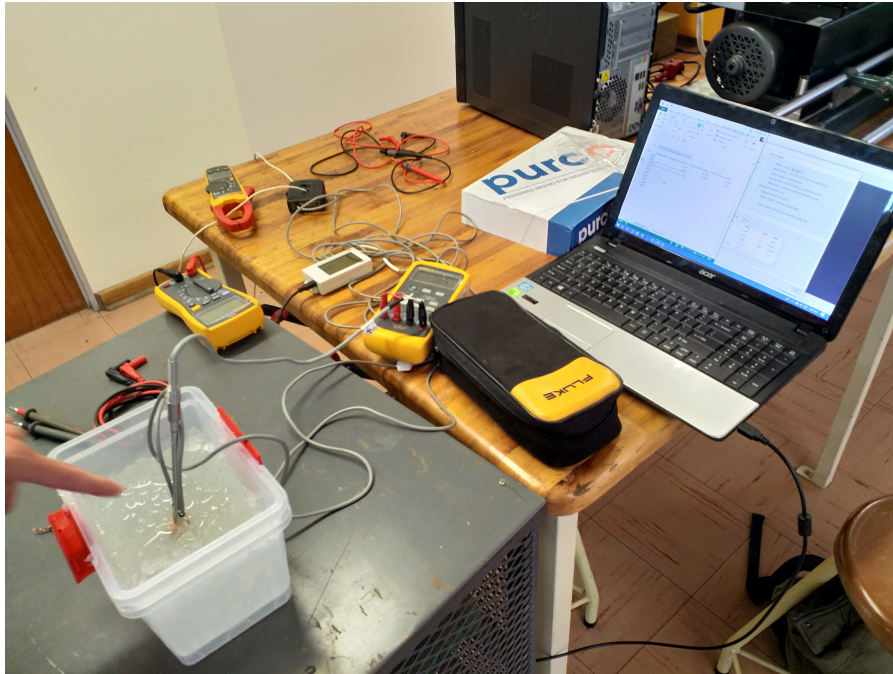


FIGURE A.2: Thermocouple verification against reference temperature probes.

¹With mixing point as defined in Chapter 3.3. Figure 3.2 illustrated the positions of the various sensors, with the mixing point at diagram position *H13* and the municipal inlet at diagram position *H15*.

TABLE A.5: Temperature probe relative error analyses as measured on the 07/05/2019.

<i>Time</i>	<i>Test</i>	<i>Instrument</i>	<i>Temperature</i> [°C]	<i>Temperature difference</i> [°C]
08:40	Steady state	Fluke	21.07	
		Temp 1	21.382	0.312
		Temp 2	21.438	0.36
08:41	ICE-Bath	Fluke	-0.02	
		Temp 1	0.124	0.144
		Temp 1	0.277	0.277
08:42	ICE-Bath	Fluke	-0.04	
		Temp 1	0.108	0.148
		Temp 2	0.252	0.292
08:51	Warm water test	Fluke	68.36	
		Temp 1	69.327	0.967
		Temp 2	69.243	0.883
08:52	Warm water test	Fluke	68.05	
		Temp 1	69.077	1.027
		Temp 2	68.999	0.949
08:52	Warm water test	Fluke	67.63	
		Temp 1	68.607	0.977
		Temp 2	68.52	0.89
08:53	Warm water test	Fluke	67.2	
		Temp 1	68.192	0.992
		Temp 2	68.054	0.854
08:53	Warm water test	Fluke	66.96	
		Temp 1	67.945	0.985
		Temp 2	67.841	0.881

Apart from the measurements required on the water circuit; the ambient conditions also had to be measured. The relative humidity and dry bulb temperatures were measured by using a HOBO® U14-001 sensor. The main parameters of the dry bulb sensors are showcased in Table A.6 and the humidity sensor is shown in A.7.

TABLE A.6: HOBO® U14-001 Dry bulb sensor characteristics

<i>Performance indicator</i>	<i>Value</i>
Measurement range	−20 to 50[°C]
Accuracy	±0.21°C from 0 to 50[°C]
Resolution	< 0.02°C at 25[°C]
Response time	15 minutes (to 90% in airflow of 1 [m s ^{−1}])
Drift	Negligible

TABLE A.7: HOBO® U14-001 Relative humidity sensor characteristics

<i>Performance indicator</i>	<i>Value</i>
Measurement range	0 – 100[%] RH, –20 to 50[°C]
Accuracy (10 - 90 [%] RH)	±2.5[%]
Maximum accuracy deviation (10 - 90 [%] RH)	±3.5 [%] including hysteresis at 25 [°C]
Accuracy (below 10 and above 90 [%] RH)	±5 [%] typical
Resolution	0.05 [%] RH
Response time	2 minutes (to 90 [%] in airflow of 1 [m/sec])
Drift	< 1 [%] per year

A.3 Electrical current measurement

A CT was used to measure the electrical current on the SWH circulating pumps. The measurement was undertaken to determine the mass flow through the pump using the methodology as presented in Chapter 3.5.2.2. The pump curve and additional data on the circulation pump is presented in Appendix J. To test the validity of the CT; a resistance bank was used to test the output signal of the CT and logger against a Fluke® 376 clamp-on multimeter (Figure A.3). The resistance bank consists of six 750 [W] resistors (6.4 [Ω]). The measurement of the Fluke 376 was compared to the measurement that the Hobo® logger relayed to the notebook. For the first measurement; only one resistor was connected to the circuit. For the second test, two resistors were connected in parallel. The results of this comparison are presented in Table A.8.



FIGURE A.3: Electrical ammeter calibration against reference meters.

From Table A.8, it can be seen that the CT and the Fluke® compares within a 50 [mA]. The margin lies within the resolution and measurement accuracy of the multimeter and CT, as reported in Table A.9 and A.10.

TABLE A.8: Electrical current transformer verification.

<i>Number of resistors</i>	<i>Fluke [A]</i>	<i>Hobo Measurement [A]</i>	<i>Difference [A]</i>	<i>Relative error [%]</i>
1	3.2	3.15	0.0499	1.56
2	6.6	6.61	-0.0166	-0.25

TABLE A.9: Fluke® 376 specifications

<i>Performance indicator</i>	<i>Value</i>
Measurement range	0 – 999.9[A]
Accuracy	2[%] $\pm$ 5 digits (10 - 100 [Hz]) 2.5[%] $\pm$ 5 digits (100 - 500 [Hz])
Resolution	0.1[A]

TABLE A.10: Onset CTV-B 20A Current transformer

<i>Performance indicator</i>	<i>Value</i>
Measurement range	2 – 20[A]
Accuracy with UX120-006M	± 2.1 [%] of full scale, including logger accuracy
Response time	200 milliseconds (from 10% to 90% of amplitude)
Voltage rating	600 [VAC]
Output	0-2.5 [VDC]

However, it can be seen from Appendix J and Table J.1 that the electrical current that is consumed by the circulating pump is lower than the accuracy as reported in Table A.10 and the smallest current that the transformer can measure. Therefore, a steady-state measurement was taken by the handheld ammeter to calculate the volumetric flow from the pump curves. The CT was thus only used to indicate when the SWH pump was active or not. Figure A.4 illustrates the typical output of the CT on the logger, with the blue line representing SWH operation.

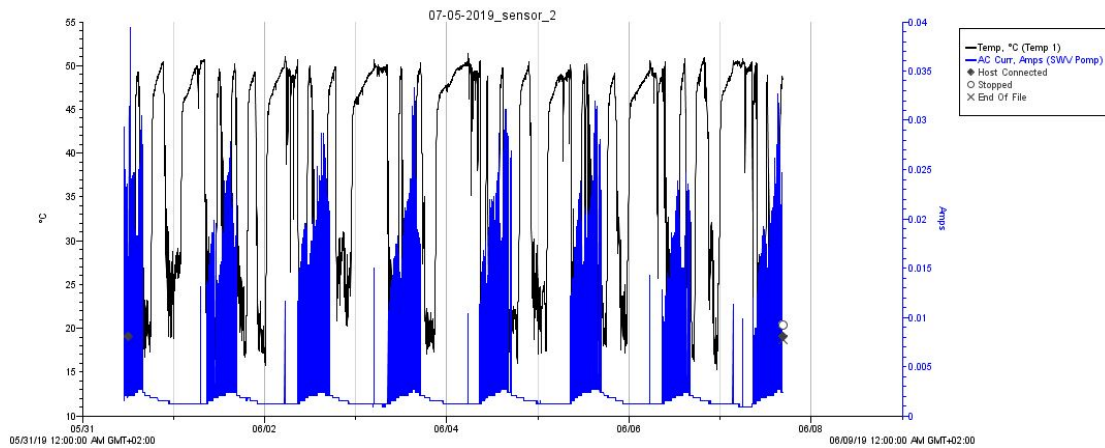


FIGURE A.4: SWH pump activation graph.

A.4 SCADA water flow meters

The flow meters that were installed by the hospital technical team is the SENSUS[®] HRI-Mei B4 DN:40-125 flow meter on both the municipal supply and the ring-main return (Diagram positions, *H15* and *H4* in Figure 3.2). The data on these flow meters are presented in Table A.11.

TABLE A.11: SENSUS HRI-Mei B4 DN:40-125 flow meter

<i>Performance indicator</i>	<i>Value</i>
Maximum peak flow	60 [m ³ /h]
Continuous flow	40 [m ³ /h]
Minimum flow rate horizontal	0.2[m ³ /h]
Pulse value	0.01 [m ³ /h]
Maximum accuracy horizontal	1.8 [%]

A.5 Propagation of uncertainty - energy fluxes

During the course of the study, energy fluxes are calculated and compared between the experimental and simulations results. As these energy fluxes are based on the measurements of various sensors; the errors have to be taken into account. Therefore, from the errors presented on the various instrumentation information from the tables in Chapter 3 and the tables presented earlier in this appendix, the consequent compounded error could be calculated for the outward energy fluxes.

For these calculations, the specific heat at constant pressure (C_p) for water was assumed constant and independent to the other variables throughout the study. Figure A.5 illustrates the variance of C_p ([kJ/kg × K]) as the temperatures (°C) was varied from 10 - 90 [°C] at 80 [kPa]:

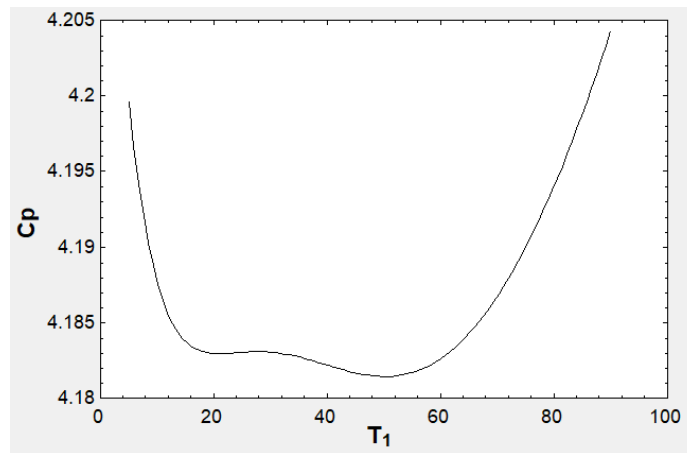


FIGURE A.5: Specific heat at constant pressure variance example.

One can note that the variation is small and the relative error is roughly 0.5 [%] between the smallest and largest value on the 10 - 90 [°C] at 80 [kPa] parameter space. The calculated heat losses presented in Appendix B are assumed to be exact whilst calculating the propagation of uncertainty with regards to sensor data. All of the energy fluxes are based on multiplication.

Therefore, assume that there are N sensors that the energy flux calculation comprises of. Furthermore, assume the relative error of the i -th sensor to be represented by ϵ_i . The propagation of errors (ϵ_c) for the energy fluxes as reported in Chapter 3.4, can accordingly be calculated by:

$$\epsilon_c = \sqrt{\sum_1^n (\epsilon_i)^2} \quad (\text{A.3})$$

Appendix B

Heat loss coefficient calculations

This appendix will discuss the approximation of the various heat loss coefficients used in the simulation programs and experimental data. The appendix was divided as follows:

- Tank heat losses (B.1)
- Pipe heat losses (B.2)
- Ring-main losses (B.3)

The calculations of the heat losses were scripted in EES® and is presented in Appendix M.

B.1 Tank heat losses

The heat loss on the storage vessels is expressed as:

$$\dot{Q}_{\text{tank-losses}} = UA_{\text{system}} \times \Delta T \quad (\text{B.1})$$

$$\dot{Q}_{\text{tank-losses}} = UA_{\text{system}} \times (T_{\text{tank}} - T_{\text{ambient}}) \quad (\text{B.2})$$

With the UA_{system} of the system as the only non-direct measurable parameter, a calculation was conducted to approximate the heat transfer coefficient. For enclosed pipes, there is predominantly two types of convective heat transfer: namely forced and natural/convective. In certain circumstances, both of these regimes can prevail, and thus, both need to be taken into consideration. However, apart from establishing the regime the thermal resistance circuit of the storage vessel had to be determined so that the overall heat transfer coefficient (UA_{system}) could be calculated.

$$UA_{\text{system}} = \frac{1}{R_{\text{system}}} \quad (\text{B.3})$$

where as

$$R_{\text{system}} = \sum R_{\text{components}} \quad (\text{B.4})$$

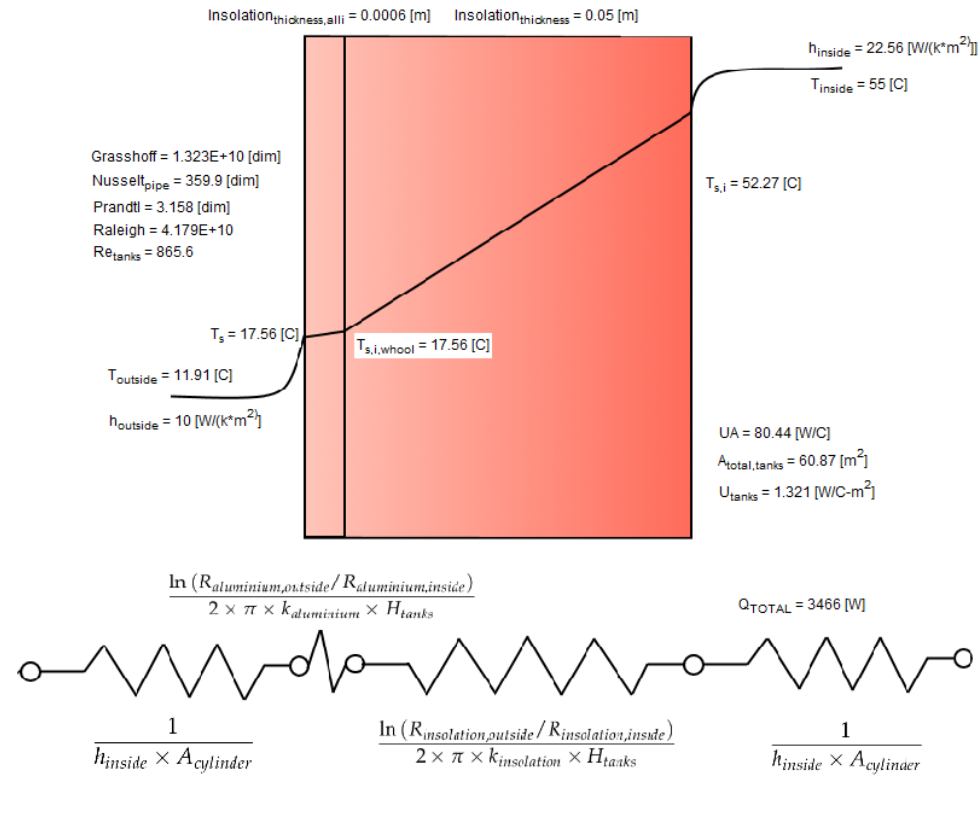


FIGURE B.1: Storage vessel thermal circuit and calculated heat loss results.

During the calculation, the heat loss due to convection and conduction will be taken into account but not the heat loss due to radiation¹. Each storage vessel is divided into three areas. The heat loss through the sidewall of the tank and the heat loss on the top and bottom of the tank. Figure B.1 illustrates the thermal circuit on the storage vessels and the temperature profile that was calculated. Consequently, the thermal circuit consisted of the following thermal resistances:

- Convective heat transfer on the inside of the storage vessel.
- Conductive heat transfer through the inner insulation.
- Conductive heat transfer through the aluminium sheeting around the storage vessel.
- Convective heat transfer on the outside of the storage vessel.

The convective heat loss for each storage tank on the system reduces to:

$$R_{conv,circumference,inside} = \frac{1}{h_{inside} \times A_{cylinder}} \quad (B.5)$$

¹Figure 3.5 illustrates the epoxy and aluminum sheet isolation utilised on the storage vessels. Furthermore, the temperature difference between the ambient conditions and the sheet temperature is relatively low. Per illustration, assume the outside sheet metal temperature to be 20 [°C]. Commercial aluminium sheets have a emissivity of roughly 0.09. The heat transfer can be calculated from: $q_{rad}'' = \epsilon \sigma (T_{amb}^4)$. Consequently, the heat loss amounts to roughly $q_{rad} = 0.0025 [W m^{-2}]$

$$R_{conv,circumference,outside} = \frac{1}{h_{outside} \times A_{cylinder}} \quad (B.6)$$

For the top and bottom of the storage vessel, the resistances can be defined by:

$$R_{conv,top,inside} = \frac{1}{h_{inside} \times A_{top}} \quad (B.7)$$

$$R_{conv,top,outside} = \frac{1}{h_{outside} \times A_{top}} \quad (B.8)$$

$$R_{conv,bot,inside} = \frac{1}{h_{inside} \times A_{bottom}} \quad (B.9)$$

$$R_{conv,bot,outside} = \frac{1}{h_{outside} \times A_{bottom}} \quad (B.10)$$

Furthermore, the storage vessels are assumed to be perfect cylinders with the heat losses of the corners assumed negligible. It is also assumed that the top and bottom is flat with no curvature. The top and bottom areas of the cylinders are represented by A_{top} [m²] and A_{bottom} [m²]. Finally the circumferential area is designated by $A_{cylinder}$ [m²]. The parameters h_{inside} [W/K × m²] and $h_{outside}$ is defined as the convective heat transfer coefficient for forced and natural convection inside and outside the tanks, respectively. With $h_{outside}$ assumed as 10 [W/K × m²]. These coefficients are assumed the same for all the inner regions of the storage vessels. The storage tanks are covered with an aluminium cladding apart from the foam-based insulation. The conductive resistances on the top and bottom is therefore given by:

$$R_{cond,top,insolation} = \frac{L_{insolation}}{k_{insolation} \times A_{top}} \quad (B.11)$$

$$R_{cond,top,aluminium} = \frac{L_{metal,sheet}}{k_{aluminium} \times A_{top}} \quad (B.12)$$

$$R_{cond,bottom,insolation} = \frac{L_{insolation}}{k_{insolation} \times A_{top}} \quad (B.13)$$

$$R_{cond,bottom,aluminium} = \frac{L_{insolation}}{k_{aluminium} \times A_{top}} \quad (B.14)$$

Hereafter, the conductive losses on the cylindrical sides of the tank were calculated from:

$$R_{cond,insolation} = \frac{\ln(R_{insolation,outside}/R_{insolation,inside})}{2 \times \pi \times k_{insolation} \times H_{tanks}} \quad (B.15)$$

$$R_{cond,aluminium} = \frac{\ln(R_{aluminium,outside}/R_{aluminium,inside})}{2 \times \pi \times k_{aluminium} \times H_{tanks}} \quad (B.16)$$

The thermal conductivity and thickness of the cladding and insulation foam was obtained from the storage vessel manufacturers' catalogues. With $R_{aluminium,outside}$ and $R_{insolation,inside}$ designating the outer and inner most diameters of the storage vessel. With the components of the thermal circuit defined, the calculation of the heat loss coefficient could be determined. The

Appendix B. Heat loss coefficient calculations

temperature inside the tanks, T_{inside} , was taken as the average of storage vessel temperatures from the experimental period. The following flux equations were obtained, by dividing the heat loss between the bottom, top and sides of the storage vessel:

$$q_{heat,loss,sides} = \frac{T_{inside} - T_{outside}}{R_{cond,insolation} + R_{cond,aluminium} + R_{conv,circumference,inside} + R_{conv,circumference,outside}} \quad (B.17)$$

$$q_{heat,loss,top} = \frac{T_{inside} - T_{outside}}{R_{cond,top,insolation} + R_{cond,top,aluminium} + R_{cond,bottom,insolation} + R_{cond,bottom,aluminium}} \quad (B.18)$$

$$q_{heat,loss,bottom} = \frac{T_{inside} - T_{outside}}{R_{cond,top,insolation} + R_{cond,top,aluminium} + R_{cond,bottom,insolation} + R_{cond,bottom,aluminium}} \quad (B.19)$$

With the total energy flux for all of the storage vessels given by:

$$\dot{Q}_{TOTAL} = (q_{heat,loss,sides} + q_{heat,loss,top} + q_{heat,loss,bottom}) \times 5 \quad (B.20)$$

The overall heat transfer coefficient U [W/K] could now be calculated from the equivalent thermal circuit:

$$UA_{system} = \frac{\dot{Q}_{total}}{T_{inside} - T_{outside}} \quad (B.21)$$

$$U_{tanks} = UA_{system} / A_{total,tanks} \quad (B.22)$$

To determine if the heat transfer regime inside the tank is within forced or natural convective transfer regimes; the Grashoff and Reynolds number were calculated. The non-dimensional Grashoff number is defined as the ratio between the buoyancy and viscous effects. Thus, serving as the indicator of buoyancy-driven flow. For the critical length, the height of the tank was used (H_{tanks}). The rationale was that stratification and buoyancy-driven flow is in the vertical direction of the tanks:

$$Gr = \frac{g \times \beta \times (T_{inside} - T_{s,i}) \times (H_{tanks}^3)}{(\mu / \rho)^2} \quad (B.23)$$

$$Re = \frac{\rho \times V \times D_{tanks}}{\mu} \quad (B.24)$$

These preliminary calculations indicated that the forced convection was laminar. This can be attributed to the low flow-rate through the relatively large diameter of the tanks. From the previous, the Prandtl number could be calculated along with the Raleigh number.

$$Ra = Gr \times Pr \quad (B.25)$$

$$Pr = \frac{Cp \times \mu}{k} \quad (B.26)$$

From here, the Nussult number could be calculated. The Nussult number was calculated using the relation for flow over vertical plates (Bergmain et al., 2013):

$$Nu_{tank} = \left(\frac{4}{3}\right) \times \left(\left(\frac{Gr}{4}\right)^{0.25}\right) \times g \times Pr \quad (B.27)$$

$$Nu_{tank2} = \left(0.825 + \left(\frac{0.387 \times Ra \times h^{1/6}}{\left(1 + (0.492/Pr)^{9/16} \right)^{8/27}} \right) \right)^2 \quad (B.28)$$

The Reynolds number is smaller than 2100, and therefore the flow is laminar.

$$Re = \frac{\rho \times V \times Diameter_{tanks}}{\mu} < 2100 \quad (B.29)$$

The Grashoff number is larger than the critical value for laminar flow and, therefore, turbulent flow is observed for natural convection.

$$Gr = \frac{g \times \beta \times (T_{inside} \times T_s) \times Diameter_{tanks}^3}{\frac{\mu}{\rho}} > 10^4 \quad (B.30)$$

The heat transfer coefficient inside the tanks could be calculated. The heat loss coefficient could consequently be used to calculate the thermal resistances.

$$h_{tank,inside} = \left(\frac{k}{H_{tanks}} \right) \times Nu_{tank} \quad (B.31)$$

$$FlowRegime = \frac{Gr}{Re^2} \quad (B.32)$$

With the ratio as $\frac{Gr}{Re^2} \gg 1$ for this case; the flow regime inside the tank is dominated by turbulent free convection flow.

B.2 Pipe heat losses

The pipe heat loss fluxes are calculated from:

$$\dot{Q}_{pipe-losses-system} = UA_{pipes} \times \Delta T \quad (B.33)$$

$$\dot{Q}_{pipe-losses-system} = UA_{pipes} \times (T_{tank} - T_{ambient}) \quad (B.34)$$

As the ambient temperatures are measured directly; an assumption has to be made for the internal temperature of the various pipes. For the solar water heater pipes, the measured outlet temperature of the SCADA system is used. For the various pipes between the tanks; the local storage tank temperature is used as the internal temperature term of these connecting pipes. The system heat transfer coefficient UA_{system} is calculated in the paragraphs below. The Reynolds number was calculated, and from the flow conditions, it was determined that the flow regime was turbulent:

$$\dot{Q}_{flow,pipe} = \frac{\dot{m}_{ringmain}}{\rho} \quad (B.35)$$

$$V_{pipe} = \dot{Q}_{flow,pipe} / A_{flow,area} \quad (B.36)$$

$$Re_{pipe} = \frac{\rho \times V_{pipe} \times D_{pipe,i}}{\mu} \quad (B.37)$$

$$Re_{pipe} > 2100$$

Due to the turbulent flow through the system pipes, free convective flow can be neglected. Furthermore, due to the relatively large length to diameter ratio, the flow was assumed to be fully developed. Following the above line of thought, the Dieter-Boelter equation for cooling was employed to calculate the convective heat transfer coefficient in the pipes (Bergmain et al., 2013):

$$Pr = \frac{cp \times \mu}{k} \quad (B.38)$$

$$Nusselt_{pipe} = 0.023 \times Re_{pipe}^{4/5} \times P_r^{0.3} \quad (B.39)$$

$$h_{pipe,i} = \frac{k}{D_{swh,pipe,i}} \times Nusselt_{pipe} \quad (B.40)$$

As with the tank heat losses, the UA_{pipes} of the system have to be determined. The pipe heat losses were divided into the SWH pipe heat losses and the pipe losses between the storage tanks. For the heat transfer within the pipes, three thermal resistances are at play. The conductivity through the insulation. The convective heat transfer on the outside and inside of the pipes, respectively. The aluminium metal coating due to its high thermal conductivity and thickness was neglected:

$$R_{conv,i} = \frac{1}{h_{pipe,i} A_{area,i}} \quad (B.41)$$

$$R_{conv,o} = \frac{1}{h_{pipe,o} A_{area,o}} \quad (B.42)$$

$$R_{cond,pipe,swh} = \frac{\ln \frac{A_{pipe,o}}{A_{pipe,i}}}{2 \times \pi \times k_{insolation} \times L_{swh,pipe}} \quad (B.43)$$

$$R_{cond,pipe,tank} = \frac{\ln \frac{A_{pipe,o}}{A_{pipe,i}}}{2 \times \pi \times k_{insolation} \times L_{tank,pipe}} \quad (B.44)$$

From test data the theoretical heat loss could be calculated and is given by:

$$Q_{heat,loss,pipes,tank} = \frac{T_{inside} - T_{outside}}{R_{conv,i} + R_{conv,o} + R_{cond,pipe,tank}} \quad (B.45)$$

$$Q_{heat,loss,pipes,swh} = \frac{T_{inside} - T_{outside}}{R_{conv,i} + R_{conv,o} + R_{cond,pipe,swh}} \quad (B.46)$$

The heat loss coefficient of the system pipes could be calculated from:

$$UA_{pipes,tank} = \frac{Q_{heat,loss,pipes,tank}}{(T_{inside} - T_{outside})} \quad (B.47)$$

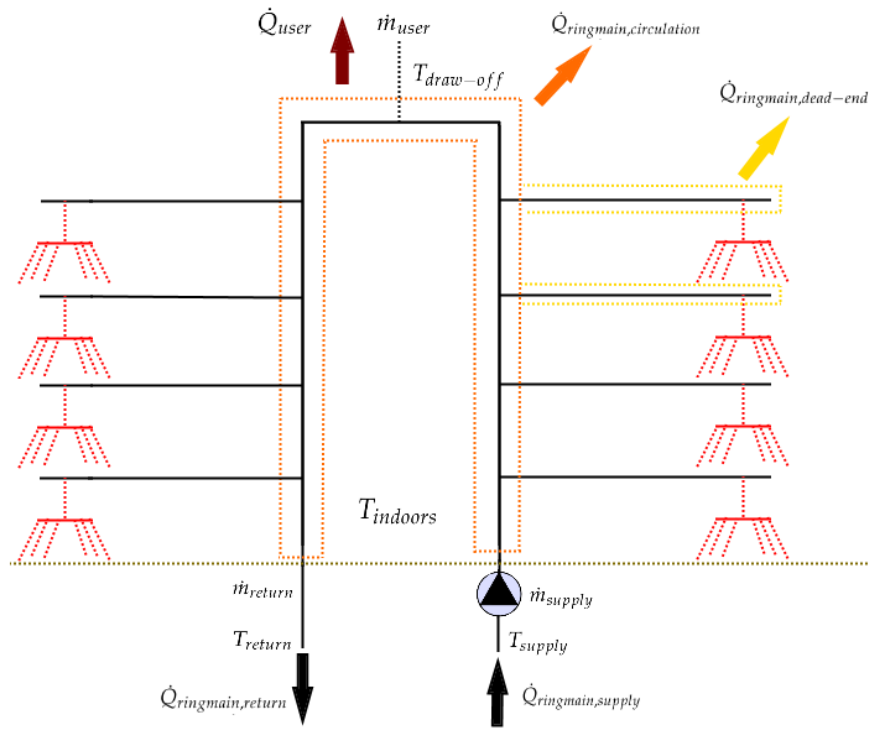


FIGURE B.2: Simplified diagram on the ring-main system..

$$UA_{pipes,swh} = \frac{Q_{heat,loss,pipes,swh}}{(T_{inside} - T_{outside})} \quad (B.48)$$

B.3 Ringmain losses

A special case was required to determine the heat loss characteristics in the ring-main. The latter refers to the condition, whereas no flow was extracted from the ring-main by the hospital. As soon as the heat loss characteristics were determined, the user load could be added to determine the total energy loss in the ring-main. Thus, one can conclude that the heat loss in the ring-main is the sum of the user load and the heat loss characteristics of the system:

$$\dot{Q}_{ringmain,losses} = \dot{Q}_{ringmain,steady-state} + \dot{Q}_{user} \quad (B.49)$$

$$\dot{Q}_{ringmain,steady-state} = f(UA_{ring-main}, T_{ringmain,average}, T_{indoors}) \quad (B.50)$$

$$\dot{Q}_{user} = f(\dot{m}_{user}, T_{municipal,in}, T_{draw-off}) \quad (B.51)$$

Consequently, the experimental data were analysed for time-steps, where the following conditions were upheld:

$$\dot{m}_{supply} = \dot{m}_{return} \quad (B.52)$$

$$\dot{m}_{user} = 0[\text{kg s}^{-1}] \quad (B.53)$$

At these time steps the boundary conditions where noted and the steady-state heat loss was calculated using the following equation:

$$\dot{Q}_{ringmain,losses} = \dot{m}_{supply} \times C p_{water} (T_{supply} - T_{return}) \quad (B.54)$$

$$\forall \dot{m}_{user} = 0 [\text{kg s}^{-1}] \quad (B.55)$$

The steady-state heat loss can theoretically be expressed as two components (Guo and Goumba, 2018). The first entails the circulation losses incurred by piping the water through the main arteries of the ring-main system ($\dot{Q}_{ringmain,circulation}$). The second compromise the losses incurred in the pipes of the various draw-off locations to the main artery of the ring-main ($\dot{Q}_{ringmain,dead-end}$). Figure B.2 serves to illustrate the former two energy fluxes graphically. The assumption of a single draw-off with linear decrease in temperature between the supply (T_{supply}) and return temperatures (T_{return}) are illustrated. The former can be calculated by:

$$\dot{Q}_{ringmain,circulation} = U A_{ringmain,circulation} \times (T_{ringmain,average} - T_{indoors}) \quad (B.56)$$

$$\dot{Q}_{ringmain,circulation} = K_d \times L_{ring} \times (T_{ringmain,average} - T_{indoors}) \quad (B.57)$$

$$\forall \dot{m}_{user} = 0 [\text{kg s}^{-1}] \quad (B.58)$$

Where L_{ring} is the total length of the ring-main in [m]; K_d is the coefficient of pipe heat loss [W/m/K]; $T_{ringmain,average}$ compromises the average temperature in the ring-main and $T_{indoors}$ designates the temperature of the indoor air. The K_d is calculated according to the pipe diameter as in EU standard 12828:2012+A1 (EU Standard, 2014):

$$K_d = 2.6 \times d_o + 0.2 \quad (B.59)$$

The losses incurred in the dead-end sections of the ring-main is given by:

$$\dot{Q}_{ringmain,dead-end} = U A_{ringmain,dead-end} \times (T_{ringmain,average} - T_{indoors}) \quad (B.60)$$

$$\dot{Q}_{ringmain,dead-end} = \left(\frac{\pi \times C p_{water} \times L_{dead} \times d_i^2}{4} \right) \times \rho \times N_{dead-pipes} \times (T_{ringmain,average} - T_{indoors}) / \delta_T \quad (B.61)$$

With L_{dead} the average length of the dead-end pipes [m]; d_i the inside diameter of the dead-end pipes [m]; $N_{dead-pipes}$ compromises the number of dead-end pipe sections. Using the time-steps where no user load was extracted from the system, the above two terms where calculated and compared to that of the measured ring-main losses. The ring-main characteristics were then adjusted until the error between the calculated and measured ring-main losses were close to zero.

$$\begin{aligned} & \text{Guess } K_d \text{ and } \delta_T \\ & \models (\dot{Q}_{ringmain,dead-end} + \dot{Q}_{ringmain,circulation}) - \dot{Q}_{ringmain,losses} \rightarrow 0 [\text{kW}] \\ & \forall \dot{m}_{user} = 0 [\text{kg s}^{-1}] \end{aligned}$$

As stated above this step was necessary to characterise the ring-main system for analyses and use in the simulation environment. The main characteristics are given in Table 3.4.

Appendix C

Daily energy balance example

For the test period (06/06/2019 - 08/06/2019) a simplified calculation process will be shown. This is based on the average energy flux values as reported in Tables C.1, C.2 and C.3. As with any closed, non-deforming control volume, one has to list all energy fluxes protruding the system boundary for a 1st law energy balance:

- Air Source Heat pump (In)
- Solar Water Heater (In)
- System pipe losses system (Out)
- Solar pipe losses (Out)
- Storage vessel losses (Out)
- User load (Out)
- Steady-state ring-main losses (Out)

From here, the inputs for each of these energy fluxes were identified as:

Heat pump (Q_{ASHP})

- Heat pump inlet temperature ($T_{In-ASHP}$)
- Ambient dry bulb temperature (T_{db})
- Ambient relative humidity (RH)
- Three-phase power consumption ($\dot{P}_{electrical}$)

Solar (Q_{Solar})

- Solar supply temperature ($T_{solar-supply-n}$)
- Solar inlet temperature (T_{mix-n})
- Steady-state SWH mass flow ($\dot{m}_{steady-state}$)
- Number of running minutes of solar (δt)

Pipe losses system ($Q_{Pipe-losses-system}$)

- Ambient dry bulb temperature (T_{db})
- Average pipe temperature ($T_{pipes-tank}$)

Pipe losses solar ($Q_{Pipe-losses-solar}$)

- Ambient dry bulb temperature (T_{db})
- Average solar outlet temperature ($T_{pipes-solar}$)

Storage vessel losses (Q_{Tank})

- Ambient dry bulb temperature (T_{db})
- Average storage vessel temperatures ($T_{tank-average}$)

User load (Q_{User})

- Mass water consumed by users ($\dot{m}_{user}$)
- System inlet temperature (T_{inlet})
- System supply temperature ($T_{draw-off}$)

Steady-state ring-main losses ($Q_{Ringmain}$)

- Ring-main return mass flow ($\dot{m}_{return}$)
- Ring-main user mass flow ($\dot{m}_{user}$)
- Ring-main supply temperature (T_{supply})
- Ring-main return temperature (T_{return})

Using a combination of the SCADA and HOBO logger data average and summation of hourly values could be determined for the inputs listed above. From the daily average and summation values, the energy fluxes can be calculated:

Heat pump energy flux

By using the daily average ambient conditions along with the average electrical consumption, an over-prediction on the contribution of the ASHP would be obtained. The reason being that the COP is sensitive to environmental conditions and very hourly. The other being that the machine was not continuously under operations and therefore, the average power consumption would not be representative of the actual operation of the machine. As the COP and power consumption are multiplied with each other, the hourly energy contribution had to be summed to calculate the average values.

Let COP_{ASHP-i} and $\dot{P}_{electrical-i}$ designate the i -th averaged hourly COP and electrical consumption of the ASHP. Furthermore let T_{db-i} , RH_i , $T_{In-ASHP-i}$ represent the ambient conditions of the i - th hour. By using the regression equation derived in Appendix H one can calculate the COP for the heat pump. As an example, the conditions as reported on 08:00 on the 06/06/2019 will be used per illustration:

$$COP_{ASHP-1} = f(T_{db-1}, RH_1, T_{In-ASHP-1})$$

$$COP_{ASHP-1} = 3.45693852134674 + (0.075939057244308 \times T_{db-1}) - (0.0302670035556761 \times T_{In-ASHP-1}) - (0.00967840826037918 \times RH_1)$$

$$COP_{ASHP-1} = 1.68$$

The COP given by the manufacturer is based on the compressor. The SCADA system measures the total power consumed by the system. Therefore, to obtain the actual heat addition, the electrical consumption apart from the ASHP compressor had to be deducted. The auxiliary components include the fans, pump and ring-main pumps. The power factor of these devices

was also taken into account. It was assumed that some of the components used a fixed amount of electrical power. This was due to the shortage of instrumentation. The above is given in equation format as follows:

$$\dot{P}_{electrical-1} = ((\dot{P}_{3-phase_1} - (\dot{P}_{fans} + \dot{P}_{pump}) \times PF_{fans})) \times PF_{ashp}) - \dot{P}_{aux}$$

The heat addition could be calculated as:

$$\begin{aligned}\dot{Q}_{ASHP-1} &= \dot{P}_{electrical-1} \times COP_{ASHP-1} \\ \dot{Q}_{ASHP-1} &= 21\ 611.31[\text{W}]\end{aligned}$$

The former was repeated for all of the hours between 08:00 AM 06/06/2020 and 08:00 AM 08/06/2020. The average heat addition of the ASHP over the full period was calculated as:

$$\dot{Q}_{ASHP} = 18\ 799.23[\text{W}]$$

Solar energy flux

The ammeter connected to the solar pump due to its measurement hysteresis is only used for indication purposes to showcase when the pump was active or not. However, steady-state measurements were taken with an accurate clamp-on ammeter to determine the steady-state flow rate ($\dot{m}_{steady-state}$) delivered by the pump¹. Due to the bang-bang control philosophy, and the transient SWH operation, the following calculation method was employed to calculate useful solar energy contribution. The methodology showcased was done for each minute interval. The instantaneous energy flux for the $j - th$ minute can be calculated from:

$$\dot{Q}_{Solar-j} = \dot{m}_{steady-state} \times C_{pWater} \times (T_{solar-supply-j} - T_{mix-j})$$

Whereas the energy transferred for the $j - th$ minute interval can be calculated from:

$$Q_{Solar-j} = \dot{m}_{steady-state} \times C_{pWater} \times (T_{solar-supply-j} - T_{mix-j}) \times \delta t$$

With $\dot{m}_{steady-state}$, $T_{solar-supply-n}$ and T_{mix-n} took as the average values of these respective variables over the $j - th$ minute. The specific heat at constant pressure is assumed constant. By summing all of the minute energy values in the time interval of between 08:00-09:00 [AM] 06/06/2020 the following total energy was obtained:

$$Q_{Solar-1} = 2\ 058\ 838.47[\text{J}]$$

By averaging the value over said hour as if the SWH was in continuous operation; the following was obtained:

$$\dot{Q}_{Solar-1} = 571.9[\text{W}]$$

The above mentioned, steps were repeated for every hour of the dataset (08:00 [AM] 06/06/2020 - 08:00 [AM] 08/06/2020). Hereafter the averaged rate of solar energy contribution was determined as:

$$\dot{Q}_{Solar} = 607.8[\text{W}]$$

Pipe losses system

The pipe losses through the system are divided into the solar pipe and system pipes. The division was made as there is a significant difference in temperatures between that of the former and latter. The method that was used to approximate the heat loss coefficient of these pipes is illustrated in Appendix B.2. The heat loss coefficient of the pipes is given in Table 3.14. The

¹With the sensors employed as discussed in Chapter 3.4.2.2

heat losses could be calculated by taking the average temperature measurements from 08:00 [AM] 06/06/2020 - 08:00 [AM] 08/06/2020. The heat loss in the system pipes is given by:

$$\begin{aligned}\dot{Q}_{pipe-losses-system} &= UA_{pipes-tanks} \times \Delta T \\ \dot{Q}_{pipe-losses-system} &= UA_{pipes-tanks} \times (T_{pipes-tank} - T_{db})\end{aligned}$$

For the pipes connecting the various storage tanks, the internal temperature ($T_{pipes-tank}$) was assumed as the average tank temperature (T_{tank}) as explained in Chapter 3.4.1.2.

$$T_{pipes-tank} = 54.26[^\circ\text{C}]$$

The average ambient temperatures for the period is reported as:

$$T_{db} = 11.91[^\circ\text{C}]$$

The overall heat loss coefficient was taken as 18.26 [W/K]. Where the average energy flux for the system pipe losses is given by:

$$\dot{Q}_{pipe-losses-system} = 773.31[\text{W}]$$

Solar pipe losses

The solar pipe heat losses follow the same methodology as that of the system pipe losses and is given by:

$$\begin{aligned}\dot{Q}_{pipe-losses-solar} &= UA_{pipes-solar} \times \Delta T \\ \dot{Q}_{pipe-losses-solar} &= UA_{pipes-solar} \times (T_{pipes-solar} - T_{db})\end{aligned}$$

The overall heat loss coefficient was taken as 20 [W/K]. The solar pipes temperature was taken as the outlet temperature measured by the SCADA system.

$$\begin{aligned}T_{pipes-solar} &= T_{supply-solar} \\ T_{pipes-solar} &= 44.21[^\circ\text{C}]\end{aligned}$$

Where the average energy flux for the solar pipe losses is given by:

$$\dot{Q}_{pipe-losses-solar} = 884.2[\text{W}]$$

Storage vessel losses system

The heat loss for the storage vessels can be expressed as:

$$\begin{aligned}\dot{Q}_{tank-losses} &= UA_{system} \times \Delta T \\ \dot{Q}_{tank-losses} &= UA_{system} \times (T_{tank-average} - T_{db})\end{aligned}$$

The tank temperature was taken as the daily average tank temperature. This temperature is calculated according to the methodology described in Chapter 3.4.1.2. The derivation of the tank heat loss coefficients is demonstrated in Appendix B. With the tank characteristics given in Table 3.3. The average tank and ambient temperatures for the period were:

$$\begin{aligned}T_{tank-average} &= 54.25[^\circ\text{C}] \\ T_{db} &= 11.74[^\circ\text{C}]\end{aligned}$$

The hourly average energy flux on the storage tank amounts to:

$$\dot{Q}_{tank-losses} = 3\ 436.70[\text{W}]$$

User load

The user load energy flux can be calculated from:

$$\dot{Q}_{user} = \dot{m}_{user} \times C p_{water} \times (T_{draw-off} - T_{inlet})$$

The draw-off ($T_{draw-off}$) is taken as the average of the ring-main supply and return temperatures². The inlet temperature (T_{inlet}) is measured by a HOBO logger that has been placed near the system inlet. The user mass flow ($\dot{m}_{user}$) is calculated by the inlet flow meters on the ring-main system. . For the average daily example, the total consumed water is added, and the average temperatures are used to calculate the total energy. This later is then equally divided between all of the hours of the experimental period:

$$\begin{aligned} m_{total} &= 9\,667.32[\text{L}] \\ T_{inlet} &= 15.35[^\circ\text{C}] \\ T_{draw-off} &= 57.55[^\circ\text{C}] \\ Q_{user} &= 1\,713\,435\,797[\text{J}] \\ \dot{Q}_{user} &= \frac{Q_{user}}{3600 \times 24 \times 3} \\ \dot{Q}_{user} &= 6610.48[\text{W}] \end{aligned}$$

Ring-main energy losses

The ring main energy losses can be calculated from:

$$\dot{Q}_{ring-main} = \dot{m}_{supply} \times C p_{water} \times T_{supply} - \dot{m}_{supply} \times C p_{water} \times T_{return}$$

The ring-main compromises of both heat losses ($\dot{Q}_{ringmain,steady-state}$) and the energy flux that is extracted by the user ($\dot{Q}_{user,ringmain}$):

$$\begin{aligned} \dot{Q}_{ringmain,steady-state} &= \dot{Q}_{ring-main} - \dot{Q}_{user,ringmain} \\ \Rightarrow \dot{Q}_{ringmain,steady-state} &= \dot{m}_{return} C p_{water} (T_{supply} - T_{return}) - \dot{m}_{user} C p_{water} (T_{supply} - T_{draw-off}) \end{aligned}$$

The ring-main supply temp (T_{supply}) and return temperatures (T_{return}) are measured by the SCADA. The draw-off temperature ($T_{draw-off}$) is assumed as the average of the supply and ring-main return temperatures. Additionally, the return mass flow ($\dot{m}_{return}$) and system inlet mass flow ($\dot{m}_{out}$) is measured by the SCADA system. The ring-main supply mass flow is determined by:

$$\dot{m}_{supply} = \dot{m}_{return} + \dot{m}_{out}$$

For the average daily example, the total circulated water is added, and the average temperatures are used to calculate the total energy. This later is then equally divided between all of the hours of the day. The total mass flow circulated on the return of the ring-main and the user consumption sums up to:

$$\begin{aligned} m_{return} &= 73\,453.2[\text{L}] \\ m_{user} &= 9\,667.3[\text{L}] \end{aligned}$$

The average temperature parameters on the ring-main during the period was:

$$\begin{aligned} Q_{user,ringmain} &= 60.9[^\circ\text{C}] \\ T_{return} &= 54.2[^\circ\text{C}] \\ T_{draw-off} &= 57.6[^\circ\text{C}] \end{aligned}$$

The total energy expenditure for the period on the ring-main was calculated as:

$$Q_{user,ringmain} = 134\,395\,082.6[\text{J}]$$

²With the draw off temperature as defined in Chapter 3.4.1.1

$$Q_{ring-main} = 2\,042\,292\,773[\text{J}]$$
$$Q_{ringmain,steady-state} = 1\,907\,897\,690[\text{J}]$$

Dividing the energy between all of the hours of the period the following was obtained for The heat losses in the ring-main was calculated by dividing the energy between all of the hours of the period:

$$\dot{Q}_{ringmain,steady-state} = 7\,360.71[\text{W}]$$

Net energy and energy ratio

The net energy contribution for the period into the system is calculated as:

$$\dot{Q}_{in} = \dot{Q}_{ASHP} + \dot{Q}_{Solar}$$
$$\dot{Q}_{in} = 19\,407.03[\text{W}]$$

The total energy expenditure for the period was calculated as:

$$\dot{Q}_{out} = \dot{Q}_{Ringmain-steady-state} + \dot{Q}_{User} + \dot{Q}_{Storage-tank-losses} + \dot{Q}_{System-pipe-losses} + \dot{Q}_{SWH-losses}$$
$$\dot{Q}_{out} = 19\,064.951$$

The nett energy for the period can be calculated as:

$$\dot{Q}_{nett} = \dot{Q}_{in} - \dot{Q}_{out}$$
$$\dot{Q}_{nett} = 342.08[\text{W}]$$

The nett energy ratio can be calculated as:

$$\dot{Q}_{nett-ratio} = \frac{\dot{Q}_{in}}{\dot{Q}_{out}}$$
$$\dot{Q}_{nett-ratio} = 1.017$$

TABLE C.1: Daily averages for input values.

	Tdb [°C] <i>Average</i>	RH [%] <i>Average</i>	Twb [°C] <i>Average</i>	P_average HP [W] <i>Average</i>	COP heat pump <i>Average</i>	Q_bruikbaar_vergelyk [W] <i>Average</i>	Q_solar [J] <i>Sum</i>	Q_son [W] <i>Average</i>	Average Tank Temperature [°C] <i>Average</i>	T_solar [°C] <i>Average</i>	User load reading [L] <i>Sum</i>	Inlet temp system [°C] <i>Average</i>	T_draw_off [°C] <i>Average</i>	Ringmain reading [L] <i>Sum</i>	Ringmain Supply temp [°C] <i>Average</i>	Ringmain_return [°C] <i>Average</i>
03/03/2019	11.24	49.81	-0.18	11969.07	2.12	19817.62	69876132.45	808.75	54.21	56.55	3014.02	15.40	57.43	24584.40	60.74	54.13
04/03/2019	13.35	48.84	1.72	9688.63	2.33	18140.72	61789739.61	715.16	54.24	48.20	3687.37	15.46	57.64	24063.60	60.95	54.32
05/03/2019	12.85	48.80	0.65	10490.85	2.24	18060.29	59557525.12	689.32	53.97	41.45	3398.79	15.50	57.35	24662.40	60.59	54.11
06/03/2019	9.54	50.59	-1.38	12131.27	1.98	19538.17	45483303.72	526.43	54.56	42.97	2581.158516	15.09	57.65	24727.2	61.03	54.27
	11.74	49.51	0.20	11069.95	2.17	18889.2	236706700.90	684.92	54.25	47.29	12681.34	15.36	57.52	98037.60	60.83	54.21

TABLE C.2: 4th and 5th of July SCADA and HOBO logger data.

Time	Tdb [°C]	RH [%]	Twb [°C]	P_avg HP [W]	COP	Q_ASHP [W]	Q_solar [J]	Q_son [W]	AVG Tank Temp [°C]	T_solar [°C]	User load reading [L]	Inlet temp system[°C]	T_draw_off [°C]	Ringmain reading [L]	Ringmain Supply temp [°C]	Ringmain_return [°C]
8:00	5.43	64.74	-0.70	11414.72	1.66	18910.24	2599546.68	722.10	52.03	56.28	272.54	13.72	57.24	1020.00	60.20	54.29
9:00	9.31	53.46	0.33	0.00	2.39	0.00	8473793.86	2353.83	54.64	93.58	208.42	14.03	56.56	1015.20	59.64	53.48
10:00	13.01	43.80	0.96	0.00	2.73	0.00	5195301.17	1443.14	54.46	90.80	192.38	14.36	56.64	1004.40	59.46	53.82
11:00	16.84	31.00	-0.43	20167.32	2.58	52111.04	11085427.30	3079.29	49.73	92.67	128.26	15.20	58.44	1035.60	61.96	54.92
12:00	18.84	26.10	-1.02	0.00	2.87	0.00	10647577.21	2957.66	56.30	90.83	160.32	15.30	58.46	1003.20	61.30	55.62
13:00	20.06	25.30	-0.39	0.00	3.19	0.00	6643850.93	1845.51	56.05	83.95	208.42	15.52	58.15	1005.60	61.05	55.26
14:00	20.73	24.21	-0.44	0.00	3.26	0.00	12589042.10	3496.96	55.42	110.82	160.32	15.53	57.76	1069.20	60.42	55.10
15:00	20.87	23.56	-0.69	7768.37	3.28	25472.15	7489525.96	2080.42	48.30	86.70	320.64	15.47	57.19	1006.80	60.10	54.28
16:00	20.79	23.69	-0.68	20986.75	2.62	55073.52	5152067.24	1431.13	55.29	87.51	144.29	15.51	59.09	1002.00	62.50	55.67
17:00	19.18	27.22	-0.15	3263.22	2.70	8824.85	0.00	0.00	56.86	82.14	48.10	15.68	59.10	1038.00	62.28	55.91
18:00	16.55	34.23	0.71	0.00	2.89	0.00	0.00	0.00	56.86	54.96	128.26	15.49	58.55	1016.40	61.86	55.23
19:00	14.71	38.34	0.67	0.00	2.82	0.00	0.00	0.00	56.55	47.18	48.10	15.69	58.16	1008.00	61.55	54.77
20:00	12.54	43.94	0.60	3632.62	2.64	9597.11	0.00	0.00	51.98	45.35	48.10	15.60	57.04	1008.00	60.43	53.65
21:00	10.51	49.26	0.32	19496.82	1.98	38617.73	0.00	0.00	50.82	39.04	144.29	15.00	57.04	1012.80	60.39	53.68
22:00	9.15	54.25	0.39	20027.23	1.73	34600.10	0.00	0.00	53.45	38.00	48.10	15.33	56.64	1041.60	59.98	53.30
23:00	7.41	60.53	0.27	20237.13	1.56	31636.65	0.00	0.00	53.73	35.44	64.13	15.91	56.49	1033.20	59.97	53.01
0:00	6.20	64.80	0.07	19952.54	1.45	29004.05	0.00	0.00	53.87	33.93	80.16	15.38	56.49	1022.40	60.03	52.94
1:00	5.46	65.85	-0.41	20087.59	1.38	27758.52	0.00	0.00	54.44	32.80	16.03	16.42	56.74	1042.80	60.34	53.15
2:00	5.01	68.13	-0.38	20307.86	1.32	26751.42	0.00	0.00	55.09	30.91	0.00	17.75	57.00	1042.80	60.77	53.23
3:00	4.24	71.90	-0.38	20337.80	1.24	25136.95	0.00	0.00	55.33	30.17	0.00	18.47	57.14	1045.20	60.96	53.32
4:00	3.75	72.79	-0.68	20285.19	1.19	24195.15	0.00	0.00	55.48	29.91	16.03	16.19	57.17	1051.20	60.94	53.39
5:00	3.06	76.40	-0.69	20414.32	1.12	22849.11	0.00	0.00	55.59	31.34	80.16	14.78	57.16	1036.80	60.96	53.37
6:00	2.85	76.47	-0.87	19412.74	1.15	22395.04	0.00	0.00	54.32	17.08	208.42	13.67	57.14	1008.00	60.41	53.86
7:00	3.19	75.38	-0.75	19465.33	1.17	22689.31	0.00	0.00	54.56	15.71	288.58	13.67	57.00	1015.20	60.20	53.80
8:00	6.66	65.43	0.62	18051.79	1.48	26686.67	2711253.28	753.13	54.06	52.25	448.90	13.73	57.69	1004.40	60.51	54.87
9:00	11.05	52.01	1.55	0.00	2.54	0.00	6901564.37	1917.10	54.72	89.81	368.74	14.09	56.82	1046.80	59.72	53.93
10:00	15.63	39.26	1.78	11752.91	2.50	29332.72	6343826.76	1762.17	49.83	79.87	256.51	14.45	57.59	1038.00	60.68	54.50
11:00	18.66	32.75	1.97	0.00	3.06	0.00	6638510.61	1844.03	54.88	79.73	176.35	14.99	57.04	1003.20	59.88	54.21
12:00	20.56	29.15	1.98	0.00	3.18	0.00	8984036.64	2495.57	54.82	79.34	208.42	15.21	56.85	1005.60	59.82	53.87
13:00	21.89	26.32	1.70	8439.00	3.28	27673.84	9765911.79	2712.75	49.46	83.51	288.58	15.62	57.36	1012.80	59.40	54.32
14:00	22.46	25.77	1.90	15658.71	2.76	43142.42	11495808.78	3193.28	55.75	90.72	208.42	15.57	59.20	1052.40	62.34	56.06
15:00	22.77	24.67	1.55	0.00	3.28	0.00	5989561.33	1663.77	56.50	72.40	224.45	15.38	58.58	1002.00	61.50	55.66
16:00	22.71	25.68	2.06	0.00	3.39	0.00	2483394.55	689.83	56.29	63.40	176.35	15.39	58.29	1000.80	61.29	55.29
17:00	20.84	28.75	2.02	0.00	3.23	0.00	475871.52	132.19	55.40	52.21	64.13	15.51	57.47	1005.60	60.40	54.54
18:00	18.47	32.95	1.88	5435.71	3.08	16719.24	0.00	0.00	51.28	44.62	176.35	15.30	56.60	1051.20	59.86	53.34
19:00	16.07	38.85	2.06	20634.02	2.33	48060.07	0.00	0.00	52.92	43.43	32.06	17.37	58.42	1087.20	62.01	54.84
20:00	12.91	48.10	2.17	21306.99	1.96	41742.69	0.00	0.00	55.75	42.55	48.10	16.73	59.12	1098.00	62.93	55.31
21:00	11.06	53.64	2.02	16075.82	1.79	28814.66	0.00	0.00	56.84	38.54	96.19	15.97	59.41	1036.80	63.18	55.64
22:00	10.68	55.28	2.09	0.00	2.18	0.00	0.00	0.00	57.54	32.16	96.19	15.85	58.88	1011.60	62.54	55.23
23:00	9.51	59.64	2.05	0.00	2.23	0.00	0.00	0.00	57.31	28.03	48.10	15.52	58.54	1006.80	62.31	54.76
0:00	9.21	60.93	2.07	0.00	2.29	0.00	0.00	0.00	56.95	26.07	128.26	15.15	58.28	1014.00	61.95	54.62
1:00	8.79	61.88	1.89	0.00	2.20	0.00	0.00	0.00	55.85	24.25	16.03	15.21	57.35	1010.40	60.85	53.85
2:00	8.22	63.51	1.71	15688.99	1.95	30642.13	0.00	0.00	46.28	22.76	64.13	15.75	56.90	1033.20	60.46	53.34
3:00	7.51	65.76	1.52	20191.34	1.56	31596.82	0.00	0.00	52.93	22.97	0.00	18.17	56.35	1039.20	59.86	52.85
4:00	7.21	67.02	1.50	20200.10	1.51	30415.72	0.00	0.00	53.71	23.21	0.00	16.87	56.42	1048.80	60.09	52.76
5:00	6.69	68.44	1.30	19892.15	1.44	28590.97	0.00	0.00	54.29	25.82	96.19	14.92	56.55	1057.20	60.13	52.98
6:00	5.57	72.39	1.00	19592.96	1.35	26433.05	0.00	0.00	54.13	22.73	304.61	14.06	56.76	1030.80	60.14	53.38
7:00	5.22	74.00	0.97	19606.63	1.30	25526.17	0.00	0.00	54.41	16.39	160.32	14.12	56.81	1066.80	60.10	53.51

TABLE C.3: 6th and 7th of July SCADA and HOBO logger data.

Time	Tdb [°C]	RH [%]	Twb [°C]	P_avg HP [W]	COP	Q_ASHP [W]	Q_solar [J]	Q_son [W]	AVG Tank Temp [°C]	T_solar [°C]	User load reading [L]	Inlet temp system [°C]	T_draw_off [°C]	Ringmain reading [L]	Ringmain Supply temp [°C]	Ringmain_return [°C]
8:00	8.34	64.88	2.11	12887.30	1.68	21611.31	2058838.47	571.90	54.58	44.94	208.42	14.05	57.04	1015.20	60.31	53.78
9:00	12.80	53.28	3.51	0.00	2.55	0.00	5751249.20	1597.57	54.67	78.33	192.38	14.37	56.56	1006.80	59.67	53.44
10:00	18.46	34.55	2.44	0.00	3.08	0.00	8061438.16	2239.29	54.55	75.13	256.51	14.90	56.64	1004.40	59.55	53.73
11:00	21.53	26.32	1.38	14195.92	3.03	42950.37	8334746.49	2315.21	50.61	80.69	176.35	15.27	57.54	1047.60	60.86	54.22
12:00	22.77	22.80	0.43	12812.78	2.86	36705.32	8714378.48	2420.66	52.85	81.52	192.38	15.63	59.57	1015.20	62.52	56.61
13:00	23.15	20.91	-0.42	883.08	3.04	2685.88	9748853.98	2708.01	54.60	78.93	208.42	15.76	58.74	1000.80	61.66	55.83
14:00	23.56	20.76	-0.18	0.00	3.29	0.00	8386899.91	2329.69	56.39	75.58	192.38	15.79	58.36	1002.00	61.39	55.33
15:00	23.72	20.96	0.09	0.00	3.37	0.00	6033462.81	1675.96	56.12	72.21	208.42	15.69	58.10	1057.20	61.12	55.08
16:00	23.33	21.72	0.25	0.00	3.43	0.00	2467657.63	685.46	54.41	64.92	144.29	15.88	56.83	1018.80	59.41	54.26
17:00	20.89	27.74	1.52	0.00	3.26	0.00	0.00	0.00	53.70	50.63	48.10	15.95	55.87	1006.80	58.70	53.04
18:00	17.97	33.58	1.71	18079.79	2.65	47976.51	0.00	0.00	48.66	40.83	320.64	15.61	57.45	1005.60	60.74	54.17
19:00	15.89	38.11	1.61	21012.58	2.23	46822.55	0.00	0.00	55.11	38.32	112.22	15.42	58.88	1032.00	62.34	55.41
20:00	13.51	44.29	1.60	5193.72	2.16	11218.49	0.00	0.00	56.73	32.67	48.10	15.62	58.86	1087.20	62.28	55.44
21:00	11.03	51.02	1.28	0.00	2.32	0.00	0.00	0.00	56.68	25.01	176.35	15.86	58.17	1051.20	61.68	54.66
22:00	9.21	55.78	0.84	0.00	2.32	0.00	0.00	0.00	56.40	20.49	64.13	15.45	57.86	1010.40	61.40	54.32
23:00	8.00	59.17	0.52	8414.19	2.15	18120.50	0.00	0.00	50.33	17.61	64.13	15.65	57.31	1014.00	60.80	53.83
0:00	6.78	62.64	0.16	19993.17	1.57	31356.38	0.00	0.00	52.36	15.91	80.16	15.95	56.54	1009.20	59.69	53.39
1:00	5.91	65.83	0.01	19590.85	1.46	28541.02	0.00	0.00	53.13	16.24	48.10	15.70	56.40	1035.60	59.75	53.06
2:00	5.00	68.37	-0.33	19811.98	1.37	27071.05	0.00	0.00	53.52	16.35	16.03	16.19	56.37	1042.80	59.81	52.94
3:00	4.28	71.74	-0.36	19850.30	1.28	25431.35	0.00	0.00	53.99	15.36	16.03	17.68	56.47	1052.40	60.02	52.91
4:00	3.70	74.04	-0.49	19951.27	1.21	24177.24	0.00	0.00	54.43	14.21	32.06	16.46	56.60	1062.00	60.17	53.02
5:00	3.05	76.38	-0.70	20186.89	1.14	22959.13	0.00	0.00	54.84	12.31	80.16	14.92	56.71	1054.80	60.24	53.18
6:00	2.66	78.48	-0.70	19760.04	1.11	21924.22	0.00	0.00	54.47	12.47	192.38	14.05	56.62	1006.80	59.97	53.26
7:00	2.91	77.76	-0.59	19156.43	1.25	23895.73	0.00	0.00	52.04	14.23	320.64	14.08	56.93	1023.60	60.07	53.80
8:00	5.32	71.04	0.50	19501.97	1.33	26034.84	946516.33	262.92	54.30	32.84	192.38	14.19	56.75	1048.80	60.07	53.42
9:00	8.50	60.98	1.39	12186.86	1.75	21341.81	7029334.56	1952.59	54.26	97.04	320.64	14.26	57.35	1005.60	60.51	54.19
10:00	11.80	49.72	1.61	0.00	2.51	0.00	6507644.16	1807.68	54.69	82.45	128.26	14.42	56.80	1006.80	59.69	53.90
11:00	14.77	39.88	1.24	0.00	2.76	0.00	7248785.11	2013.55	54.37	76.21	112.22	14.76	56.47	1052.40	59.37	53.56
12:00	17.31	35.06	1.74	0.00	3.00	0.00	4092764.89	1136.88	53.87	78.61	224.45	15.13	56.29	1006.80	58.87	53.71
13:00	18.34	31.21	0.99	18566.92	2.69	49950.02	5276244.06	1465.62	49.10	78.13	64.13	15.45	57.80	996.00	61.46	54.14
14:00	19.00	24.58	-1.72	16899.15	2.51	42338.15	9532386.06	2647.89	55.50	82.87	224.45	15.40	59.54	994.80	62.70	56.39
15:00	19.17	22.57	-2.70	0.00	3.03	0.00	4184502.06	1162.36	56.95	75.94	144.29	15.27	58.83	1003.20	61.95	55.72
16:00	18.31	23.89	-2.67	0.00	3.10	0.00	665126.49	184.76	56.75	70.14	160.32	15.20	58.42	1034.40	61.75	55.08
17:00	16.46	26.63	-2.79	14291.95	2.73	38948.82	0.00	0.00	50.99	46.62	144.29	15.33	58.25	1082.40	61.59	54.90
18:00	14.27	32.21	-2.11	21046.96	2.16	45462.23	0.00	0.00	55.48	39.88	208.42	15.17	59.08	1048.80	62.38	55.78
19:00	12.32	36.62	-2.08	16901.51	1.98	33475.76	0.00	0.00	57.49	33.81	64.13	15.56	60.23	1008.00	63.91	56.55
20:00	10.89	40.39	-2.05	620.07	1.99	1231.82	0.00	0.00	57.92	27.47	32.06	15.80	59.21	1008.00	62.92	55.49
21:00	9.07	45.58	-2.08	0.00	2.23	0.00	0.00	0.00	57.62	22.56	32.06	15.15	58.69	1006.80	62.62	54.77
22:00	7.21	52.34	-1.90	0.00	2.26	0.00	0.00	0.00	57.37	19.61	192.38	14.96	58.67	1010.40	62.37	54.97
23:00	6.21	56.01	-1.91	12142.61	2.10	25517.87	0.00	0.00	46.84	17.52	80.16	14.67	58.43	1035.60	61.44	55.42
0:00	4.87	61.43	-1.93	19646.55	1.46	28637.57	0.00	0.00	52.49	14.82	0.00	17.28	56.47	1026.00	59.71	53.22
1:00	4.23	66.26	-1.50	19671.72	1.33	26189.86	0.00	0.00	53.63	15.28	16.03	16.35	56.24	1045.20	59.79	52.70
2:00	3.54	68.33	-1.75	19875.18	1.25	24805.84	0.00	0.00	54.25	15.64	64.13	15.14	56.41	1051.20	59.92	52.89
3:00	2.65	70.59	-2.16	20082.27	1.15	23070.12	0.00	0.00	55.18	19.56	0.00	15.61	56.92	1060.80	60.55	53.29
4:00	1.87	72.39	-2.57	20096.17	1.08	21656.49	0.00	0.00	55.50	21.61	16.03	16.16	56.99	1063.20	60.70	53.29
5:00	1.26	74.41	-2.79	20122.56	1.03	20702.38	0.00	0.00	55.25	19.83	16.03	14.17	56.66	1053.60	60.34	52.99
6:00	0.72	75.57	-3.10	19748.56	1.00	19780.22	0.00	0.00	54.92	23.04	96.19	13.39	56.72	1040.40	60.25	53.18
7:00	0.86	76.55	-2.80	19749.49	1.00	19772.31	0.00	0.00	54.75	19.90	48.10	13.27	56.43	1038.00	59.86	53.00

Appendix D

HP Manufacturer test data

TABLE D.1: 90 [kW] ASHP manufacturer test data

<i>Inlet Temp</i>	<i>DB</i>	<i>WB</i>	<i>Q-heat</i>	<i>COP</i>
15	-5	-6	43.18	2.46
15	0	-1	50.56	5.78
15	5	4	58.12	3.25
15	10	9	66.42	3.67
15	15	13	74.73	4.22
15	20	15	83.03	4.64
25	-5	-6	41.25	2.26
25	0	-1	48.62	2.56
25	5	4	55.8	2.99
25	10	9	64.43	3.37
25	15	13	74.49	3.88
25	20	15	80.54	4.27
35	-5	-6	41.88	2.09
35	0	-1	49.13	2.37
35	5	4	56.38	2.76
35	10	9	65.1	3.12
35	15	13	70.24	3.17
35	20	15	78.05	3.48
45	-5	-6	40.15	1.84
45	0	-1	47.1	2.09
45	5	4	54.05	2.44
45	10	9	62.44	2.75
45	15	13	70.24	3.17
45	20	15	78.05	3.48
50	-5	-6	41.02	1.72
50	0	-1	48.12	1.95
50	5	4	55.21	2.27
50	10	9	65.1	2.57
50	15	13	73.23	2.96
50	20	15	81.37	3.25

Appendix E

Component library - Simulations

This appendix contains comparison Tables between the Reduced and Detailed simulations as introduced in Chapter 4.1.1. The general outline of these tables are as follows:

- Assumptions - The assumptions that were made are presented.
- Fundamental equations - The fundamental mass, energy and momentum equations are presented for each component.
- Numerical upwinding - The boundary conditions for the components are specified according to the mass flow enforced through the component.

The Appendix have comparison Tables for the following components:

- T-piece component (E.1)
- Storage vessel component (E.2)
- Pipe component (E.3)
- SWH Component (E.4)
- ASHP component (E.5)

TABLE E.1: Reduced and detailed 1D model fundamental equations for T-Piece components.

Reduced 1D model	Detailed 1D model
<i>T-piece component</i>	
<i>Assumptions</i>	
Negligible thermal capacitance. $\Rightarrow \frac{m(u_{i+1}-u_i)}{\delta t} \approx 0$ Negligible density change. $\Rightarrow \frac{\partial \rho}{\partial t} \approx 0$ Negligible friction. $\Rightarrow \dot{Q}_{friction} \approx 0$ Through mass flow explicitly defined. $\Rightarrow \dot{m} = f(\text{Program})$	Non-negligible thermal capacitance. $\Rightarrow \frac{m(u_{i+1}-u_i)}{\delta t} \neq 0$ Negligible density change. $\Rightarrow \frac{\partial \rho}{\partial t} \approx 0$ Negligible friction. $\Rightarrow \dot{Q}_{friction} \neq 0$ Mass flow is a function of pump and friction energy. $\Rightarrow \dot{m} = f(\dot{W}_{pump}, \dot{Q}_{friction})$
<i>Fundamental Equations</i>	
<i>Continuity of mass</i>	
$\Sigma \dot{m} = 0$ $\dot{m}_1 + \dot{m}_2 + \dot{m}_3 = 0$	$\Sigma \dot{m} = 0$ $\dot{m}_1 + \dot{m}_2 + \dot{m}_3 = 0$
<i>Continuity of energy</i>	
$\Sigma \dot{m}h = 0$ $\dot{m}_1 h_1 + \dot{m}_2 h_2 + \dot{m}_3 h_3 = \Sigma \dot{m}h$ $\dot{Q}_{convection} = 0$	$\frac{\dot{m}(u_{i+1}-u_i)}{\delta t} = \Sigma \dot{m}h - \dot{Q}_{convection}$ $\dot{m}_1 h_1 + \dot{m}_2 h_2 + \dot{m}_3 h_3 = \Sigma \dot{m}h$ $\dot{Q}_{convection} = UA \times (T_{internal} - T_{amb})$
<i>Continuity of linear momentum</i>	
n/a	n/a
<i>Numerical Upwinding / Boundary Conditions</i>	
$T_{port-1} = T_1 \quad \forall \quad \dot{m}_1 > 0$ $T_{port-2} = T_2 \quad \forall \quad \dot{m}_2 > 0$ $T_{port-3} = T_3 \quad \forall \quad \dot{m}_3 > 0$ $T_{port-1} = T_{internal} \quad \forall \quad \dot{m}_1 < 0$ $T_{port-2} = T_{internal} \quad \forall \quad \dot{m}_2 < 0$ $T_{port-3} = T_{internal} \quad \forall \quad \dot{m}_3 < 0$	$T_{port-1} = T_1 \quad \forall \quad \dot{m}_1 > 0$ $T_{port-2} = T_2 \quad \forall \quad \dot{m}_2 > 0$ $T_{port-3} = T_3 \quad \forall \quad \dot{m}_3 > 0$ $T_{port-1} = T_{internal} \quad \forall \quad \dot{m}_1 < 0$ $T_{port-2} = T_{internal} \quad \forall \quad \dot{m}_2 < 0$ $T_{port-3} = T_{internal} \quad \forall \quad \dot{m}_3 < 0$
<i>Inputs</i>	
UA	$m, UA, a_1, a_2, a_3, f, l, D$

TABLE E.2: Reduced and detailed 1D model fundamental equations for storage vessel components.

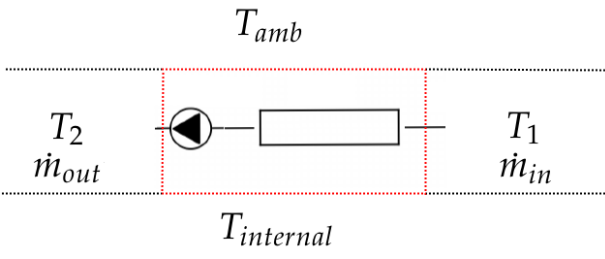
Reduced 1D model	Detailed 1D model
Storage vessel components	
	
Assumptions	
Non-Negligible thermal capacitance. $\Rightarrow \frac{m(u_{i+1}-u_i)}{\delta t} \neq 0$	Non-negligible thermal capacitance. $\Rightarrow \frac{m(u_{i+1}-u_i)}{\delta t} \neq 0$
Negligible density change. $\Rightarrow \frac{\partial \rho}{\partial t} \approx 0$	Negligible density change. $\Rightarrow \frac{\partial \rho}{\partial t} \approx 0$
Non-Negligible pipe friction. $\Rightarrow \dot{Q}_{friction} \neq 0$	Non-negligible pipe friction. $\Rightarrow \dot{Q}_{friction} \neq 0$
Non-Negligible work input from pumps. $\Rightarrow \dot{W}_{pump} \neq 0$	Non-negligible pump work. $\Rightarrow \dot{W}_{pump} \neq 0$
Through mass flow explicitly defined. $\Rightarrow \dot{m} = f(Program)$	Mass flow is a function of pump and friction energy. $\Rightarrow \dot{m} = f(\dot{W}_{pump}, \dot{Q}_{friction})$
	Pump curve approximated by 2nd order parabola. $\Rightarrow H_{pump} = a_1(\frac{\dot{m}}{\rho} 3600)^2 + a_2(\frac{\dot{m}}{\rho} 3600) + a_3$
Fundamental Equations	
Continuity of mass	
$\Sigma \dot{m} = 0$ $\dot{m}_{in} = \dot{m}_{out}$	$\Sigma \dot{m} = 0$ $\dot{m}_{in} = \dot{m}_{out}$
Continuity of energy	
$\frac{\dot{m}(u_{i+1}-u_i)}{\delta t} = \Sigma \dot{m}h - \dot{Q}_{convection}$ $\Sigma \dot{m}h = \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out}$ $\dot{Q}_{convection} = UA \times (T_{internal} - T_{amb})$	$\frac{\dot{m}(u_{i+1}-u_i)}{\delta t} = \Sigma \dot{m}h - \dot{Q}_{convection} - \dot{Q}_{friction} + \dot{W}_{pump}$ $\Sigma \dot{m}h = \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out}$ $\dot{Q}_{convection} = UA \times (T_{internal} - T_{amb})$
Continuity of linear momentum	
$\dot{m}_{in} = \dot{m}_{pump-steady-state} \delta(a-1)$	$\dot{Q}_{friction} = f \frac{1}{D} \frac{\rho V^2}{2} \times \frac{\dot{m}}{\rho}$ $\dot{W}_{pump} = \dot{m}g(a_1(\frac{\dot{m}}{\rho} 3600)^2 + a_2(\frac{\dot{m}}{\rho} 3600) + a_3)$
Numerical Upwinding / Boundary Conditions	
$T_{internal} = T_2 \quad \forall \quad \dot{m}_{in} > 0$ $T_{port} = T_1 \quad \forall \quad \dot{m}_{in} > 0$ $T_{internal} = T_1 \quad \forall \quad \dot{m}_{in} < 0$ $T_{port} = T_2 \quad \forall \quad \dot{m}_{in} < 0$	$T_{internal} = T_2 \quad \forall \quad \dot{m}_{in} > 0$ $T_{port} = T_1 \quad \forall \quad \dot{m}_{in} > 0$ $T_{internal} = T_1 \quad \forall \quad \dot{m}_{in} < 0$ $T_{port} = T_2 \quad \forall \quad \dot{m}_{in} < 0$
Inputs	
$UA_{tank}, T_{amb}, \dot{m}_{in,i}, m$	$UA_{tank}, T_{amb}, T_{internal,i}, \dot{m}_{in,i}, a_1, a_2, a_3, f, l, D, m$

TABLE E.3: Reduced and detailed 1D model fundamental equations for pipe components.

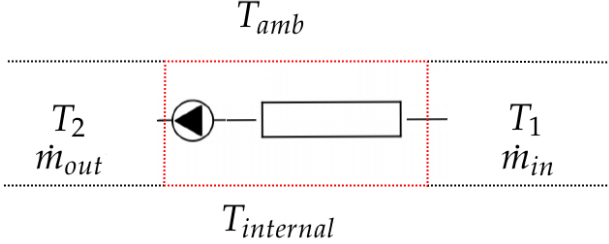
Reduced 1D model	Detailed 1D model
Pipe component	
	
Assumptions	
Negligible thermal capacitance. $\Rightarrow \frac{m(u_{i+1}-u_i)}{\delta t} \approx 0$ Negligible density change. $\Rightarrow \frac{\partial \rho}{\partial t} \approx 0$ Negligible pipe friction. $\Rightarrow \dot{Q}_{friction} \approx 0$ Negligible work input from pumps. $\Rightarrow \dot{W}_{pump} \approx 0$ Through mass flow explicitly defined. $\Rightarrow \dot{m} = f(\text{Program})$ Advection flow restriction. $\Rightarrow \frac{\partial \dot{m}_{in}}{\partial t} \approx 0$	Non-negligible thermal capacitance. $\Rightarrow \frac{m(u_{i+1}-u_i)}{\delta t} \neq 0$ Negligible density change. $\Rightarrow \frac{\partial \rho}{\partial t} \approx 0$ Non-negligible pipe friction. $\Rightarrow \dot{Q}_{friction} \neq 0$ Non-negligible pump work. $\Rightarrow \dot{W}_{pump} \neq 0$ Mass flow is a function of pump and friction energy. $\Rightarrow \dot{m} = f(\dot{W}_{pump}, \dot{Q}_{friction})$ Pump curve approximated by 2nd order parabola. $\Rightarrow H_{pump} = a_1(\frac{\dot{m}}{\rho} 3600)^2 + a_2(\frac{\dot{m}}{\rho} 3600) + a_3$
Fundamental Equations	
Continuity of mass	
$\Sigma \dot{m} = 0$ $\dot{m}_{in} = \dot{m}_{out}$	$\Sigma \dot{m} = 0$ $\dot{m}_{in} = \dot{m}_{out}$
Continuity of energy	
$\Sigma \dot{m}h = \dot{Q}_{convection}$ $\Sigma \dot{m}h = \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out}$ $\dot{Q}_{convection} = UA \times (T_{port} - T_{amb})$	$\frac{\dot{m}(u_{i+1}-u_i)}{\delta t} = \Sigma \dot{m}h - \dot{Q}_{convection} - \dot{Q}_{friction} + \dot{W}_{pump}$ $\Sigma \dot{m}h = \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out}$ $\dot{Q}_{convection} = UA \times (T_{internal} - T_{amb})$
Continuity of linear momentum	
$\dot{m}_{in} = \dot{m}_{pump-steady-state} \delta(a-1)$	$\dot{Q}_{friction} = f \frac{l}{D} \frac{\rho V^2}{2} \times \frac{\dot{m}}{\rho}$ $\dot{W}_{pump} = \dot{m}g(a_1(\frac{\dot{m}}{\rho} 3600)^2 + a_2(\frac{\dot{m}}{\rho} 3600) + a_3)$
Numerical Upwinding / Boundary Conditions	
$T_{port} = T_1 \quad \forall \quad \dot{m}_{in} > 0$ $T_{port} = T_2 \quad \forall \quad \dot{m}_{in} < 0$	$T_{internal} = T_2 \quad \forall \quad \dot{m}_{in} > 0$ $T_{port} = T_1 \quad \forall \quad \dot{m}_{in} > 0$ $T_{internal} = T_1 \quad \forall \quad \dot{m}_{in} < 0$ $T_{port} = T_2 \quad \forall \quad \dot{m}_{in} < 0$
Inputs	
$UA_{pipes}, T_{amb}, \dot{m}_{in,i}$	$UA_{pipes}, T_{amb}, T_{internal,i}, \dot{m}_{in,i}, a_1, a_2, a_3, f, l, D, m$

TABLE E.4: Reduced and detailed 1D model fundamental equations for SWH components.

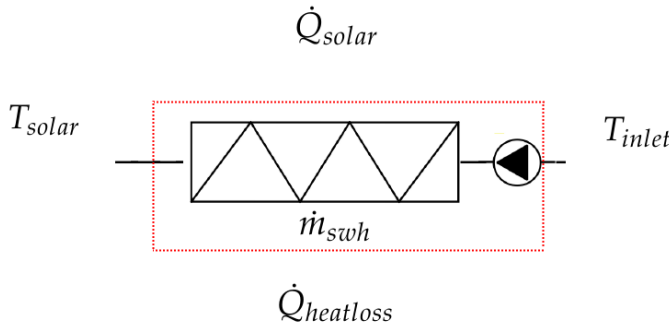
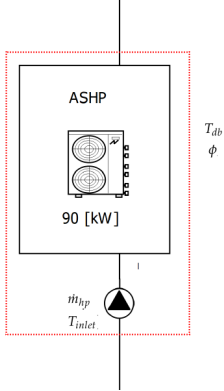
Reduced 1D model	Detailed 1D model
SWH component	
	
Assumptions	
Non-negligible thermal capacitance. $\Rightarrow \frac{m(u_{i+1}-u_i)}{\delta t} \neq 0$ Negligible density change. $\Rightarrow \frac{\partial \rho}{\partial t} \approx 0$ Non-negligible pipe friction. $\Rightarrow \dot{Q}_{friction} \neq 0$ Through mass flow explicitly defined. $\Rightarrow \dot{m} = f(\text{Program})$	Non-negligible thermal capacitance. $\Rightarrow \frac{m(u_{i+1}-u_i)}{\delta t} \neq 0$ Negligible density change. $\Rightarrow \frac{\partial \rho}{\partial t} \approx 0$ Non-negligible pipe friction. $\Rightarrow \dot{Q}_{friction} \neq 0$ Non-negligible pump work. $\Rightarrow \dot{W}_{pump} \neq 0$ $\Rightarrow \dot{m} = f(\dot{W}_{pump}, \dot{Q}_{friction})$ $\Rightarrow H_{pump} = a_1(\frac{\dot{m}}{\rho} 3600)^2 + a_2(\frac{\dot{m}}{\rho} 3600) + a_3$
Fundamental Equations	
Continuity of mass	
$\Sigma \dot{m} = 0$ $\dot{m}_{in} = \dot{m}_{out}$	$\Sigma \dot{m} = 0$ $\dot{m}_{in} = \dot{m}_{out}$
Continuity of energy	
$\frac{\dot{m}(u_{i+1}-u_i)}{\delta t} = \Sigma \dot{m}h - \dot{Q}_{convection} - \dot{Q}_{friction} + \dot{W}_{pump}$ $\Sigma \dot{m}h = \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out}$ $\dot{Q}_{convection} = (1 - \eta_{solar}) \times I_{rr} \times A_c$ $\dot{Q}_{solar} = \eta_{solar} \times I_{rr} \times A_c$ $A = (T_{inlet} - T_{db}) / I_{rr}$ $\eta_{solar} = a_1 A^2 + a_2 A + a_3$ $\Sigma \dot{m}u = \dot{m}_{swh} Cp(T_{solar} - T_{inlet})$	$\frac{\dot{m}(u_{i+1}-u_i)}{\delta t} = \Sigma \dot{m}h - \dot{Q}_{convection} - \dot{Q}_{friction} + \dot{W}_{pump}$ $\Sigma \dot{m}h = \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out}$ $\dot{Q}_{convection} = (1 - \eta_{solar}) \times I_{rr} \times A_c$ $\dot{Q}_{solar} = \eta_{solar} \times I_{rr} \times A_c$ $A = (T_{inlet} - T_{db}) / I_{rr}$ $\eta_{solar} = a_1 A^2 + a_2 A + a_3$ $\Sigma \dot{m}u = \dot{m}_{swh} Cp(T_{solar} - T_{inlet})$
Continuity of linear momentum	
$\dot{m}_{in} = \dot{m}_{pump-steady-state} \delta(a - 1)$ $\dot{m}_{pump-steady-state} = f(\text{program})$	$\dot{Q}_{friction} = f \frac{1}{D} \frac{\rho V^2}{2} \times \frac{\dot{m}}{\rho}$ $\dot{W}_{pump} = \dot{m} g (a_1 (\frac{\dot{m}}{\rho} 3600)^2 + a_2 (\frac{\dot{m}}{\rho} 3600) + a_3)$
Numerical Upwinding / Boundary Conditions	
$T_{port-1} = T_1 \quad \forall \quad \dot{m}_{in} > 0$ $T_{port-2} = T_{solar} \quad \forall \quad \dot{m}_{in} > 0$	$T_{port-1} = T_1 \quad \forall \quad \dot{m}_{in} > 0$ $T_{port-2} = T_{solar} \quad \forall \quad \dot{m}_{in} > 0$
Inputs	
$m, u_{initial}, I_{rr}, A_c, T_{db}, T_{inlet}, \dot{m}_{pump-steady-state}$	$m, u_{initial}, I_{rr}, A_c, T_{db}, T_{inlet}, a_1, a_2, a_3, f, l, D$

TABLE E.5: Reduced and detailed 1D model fundamental equations for ASHP component.

Reduced 1D model	Detailed 1D model
<i>ASHP component</i>	
	
<i>Assumptions</i>	
Negligible thermal capacitance. $\Rightarrow \frac{m(u_{i+1}-u_i)}{\delta t} \approx 0$ Negligible density change. $\Rightarrow \frac{\partial \rho}{\partial t} \approx 0$ Negligible pipe friction. $\Rightarrow \dot{Q}_{friction} \approx 0$ Through mass flow explicitly defined. $\Rightarrow \dot{m} = f(\text{Program})$	Non-negligible thermal capacitance. $\Rightarrow \frac{m(u_{i+1}-u_i)}{\delta t} \neq 0$ Negligible density change. $\Rightarrow \frac{\partial \rho}{\partial t} \approx 0$ Non-negligible pipe friction. $\Rightarrow \dot{Q}_{friction} \neq 0$ Mass flow is a function of pump and friction energy. $\Rightarrow \dot{m} = f(\dot{W}_{pump}, \dot{Q}_{friction})$
<i>Fundamental Equations</i>	
<i>Continuity of mass</i>	
$\Sigma \dot{m} = 0$ $\dot{m}_{in} = \dot{m}_{out} = \dot{m}_{hp}$	$\Sigma \dot{m} = 0$ $\dot{m}_{in} = \dot{m}_{out} = \dot{m}_{hp}$
<i>Continuity of energy</i>	
$\dot{Q}_{ashp-available} = f(T_{db}, \phi, T_{inlet})$ $\Sigma \dot{m}h = \dot{Q}_{ashp-available}$ $-\dot{m}_1 h_{inlet} + \dot{m}_2 h_{ashp} = \Sigma \dot{m}h$	$\dot{Q}_{ashp-available} = f(T_{db}, \phi, T_{inlet})$ $\frac{\dot{m}(u_{i+1}-u_i)}{\delta t} + \Sigma \dot{m}h = -\dot{Q}_{convection} - \dot{Q}_{friction} + \dot{W}_{pump}$ $-\dot{m}_1 h_{inlet} + \dot{m}_2 h_{ashp} = \Sigma \dot{m}h$
<i>Continuity of linear momentum</i>	
$\dot{Q}_{convection} \approx 0$ $\dot{m}_{in} = \dot{m}_{pump-steady-state} \delta(a-1)$	$\dot{Q}_{convection} = UA \times (T_{internal} - T_{amb})$ $\dot{Q}_{friction} = f \frac{l}{D} \frac{\rho V^2}{2} \times \frac{\dot{m}}{\rho}$ $\dot{W}_{pump} = \dot{m} g (a_1 (\frac{\dot{m}}{\rho} 3600)^2 + a_2 (\frac{\dot{m}}{\rho} 3600) + a_3)$
<i>Numerical Upwinding / Boundary Conditions</i>	
$T_{port-1} = T_1 \quad \forall \quad \dot{m}_1 > 0$ $T_{port-2} = T_2 \quad \forall \quad \dot{m}_2 > 0$	$T_{port-1} = T_1 \quad \forall \quad \dot{m}_1 > 0$ $T_{port-2} = T_2 \quad \forall \quad \dot{m}_2 > 0$
<i>Inputs</i>	
$T_{db}, \phi, T_{inlet}, T_{fixed}$	$m, T_{db}, \phi, T_{inlet}, UA, f, l, D, u_{initial}, T_{fixed}$

Appendix F

Runga-Kutte method of lines derivation

F.1 Energy flux and PDE

The method discusses the numerical method that was used in the reduced model. Consider a stationary, non-deforming control volume that encompasses an incompressible fluid. This layer represents one of the layers in the storage tanks of the hot water system. Furthermore, assume a coordinate system where the x-axis is in the length of the storage tank. It is assumed that conduction can only take place by method between the layers of the fluid. The convective heat transfer losses can only move radially outwards in the layer towards the sidewall of the storage tank. The conduction between the layers by means of the sidewall is neglected. The mass flux between these layers are the flow that circulates through the storage tanks system. The energy terms is thus as follows:

- Conduction losses along the x-axis
- Convection losses in the radial direction
- Mass flux along the x-axis

By combining these. the following partial differential equation is obtained:

$$\frac{\partial T}{\partial t} = \frac{k_{fluid}}{\rho_{fluid} C p_{fluid}} \frac{\partial^2 T}{\partial x^2} - \frac{\dot{m}_{nett} C p_{fluid}}{A_w \rho_{fluid} C p_{fluid}} \frac{\partial T}{\partial x} + \frac{h_{sidewalls} P_{perimeter}}{A_w \rho_{fluid} C p_{fluid}} \delta T \quad (F.1)$$

$$\frac{\partial T}{\partial t} = \frac{k_{fluid}}{\rho_{fluid} C p_{fluid}} \frac{\partial^2 T}{\partial x^2} - \frac{\dot{m}_{nett} C p_{fluid}}{A_w \rho_{fluid} C p_{fluid}} \frac{\partial T}{\partial x} + \frac{2h_{sidewalls}}{\rho_{fluid} C p_{fluid}} \frac{\partial T}{\partial r} \quad (F.2)$$

Where $A_w = \pi R^2$ and $P_{perimeter} = 2\pi R$. The above-mentioned PDE is a mixed problem as the dependant variable T is partially dependant in both time and spatial parameters. Which further complicates the matter is that the spatial dependence is both of first and second-degree significance. The origin of this PDE will be illustrated by first observing the diffusion component and then the advection component of the PDE. Furthermore, from heat transfer, if one observes a control volume of dimensions dx , dy and dz respectively. With the differential areas of this control volume to be of the same size. If we assume that heat transfer can only take place via conduction to the control volume then from Reynolds Theorem, we can write the energy equation for this control volume as:

$$\rho C p \frac{\partial T}{\partial t} dx dy dz = \sum \dot{Q}_{in} - \sum \dot{Q}_{out} \quad (F.3)$$

$$\sum \dot{Q}_{in} = q_x + q_y + q_z \quad (F.4)$$

By using the Taylor expansion for the velocity term, the following equation was obtained.

$$\begin{aligned} \sum \dot{Q}_{out} &= (q_x + \frac{\partial q_x}{\partial x} dx) + (q_y + \frac{\partial q_y}{\partial y} dy) + (q_z + \frac{\partial q_z}{\partial z} dz) \\ \rho C_p \frac{\partial T}{\partial t} dx dy dz &= -\frac{\partial q_x}{\partial x} dx - \frac{\partial q_y}{\partial y} dy - \frac{\partial q_z}{\partial z} dz \end{aligned}$$

Then Fouriers laws state that the energy form of conduction heat rate for a isotropic material can be expressed as:

$$q_x = -k dy dz \frac{\partial T}{\partial x} \quad (F.5)$$

$$q_y = -k dx dz \frac{\partial T}{\partial y} \quad (F.6)$$

$$q_z = -k dx dy \frac{\partial T}{\partial z} \quad (F.7)$$

The second order conductive terms is thus obtained:

$$\rho C_p \frac{\partial T}{\partial t} = -\frac{\partial}{\partial x} (k \frac{\partial T}{\partial x}) - \frac{\partial}{\partial y} (k \frac{\partial T}{\partial y}) - \frac{\partial}{\partial z} (k \frac{\partial T}{\partial z}) \quad (F.8)$$

From the former it can clearly be seen that the rate of heat conduction through the medium is second degree dependence on spatial parameters. This concludes the derivation of the diffusion term of the PDE. For the mass flux the same rationale can be followed. If one assumes that energy can only be transferred via mass flux in the x-direction then the following can be obtained:

$$\begin{aligned} u_{in}(\rho_{fluid} C_{pfluid})(T_{in}) dy dz - u_{out}(\rho_{fluid} C_{pfluid})(T_{out}) dy dz &= \rho_{fluid} C_{pfluid} \frac{\partial T}{\partial t} \forall \\ (u_{in} T_{in} - u_{out} T_{out})(\rho_{fluid} C_{pfluid}) dy dz &= (\rho_{fluid} C_{pfluid}) \frac{\partial T}{\partial t} dx dy dz \\ (u_x T_x - u_{x+\delta x} T_{x+\delta x})(\rho_{fluid} C_{pfluid}) dy dz &= (\rho_{fluid} C_{pfluid}) \frac{\partial T}{\partial t} dx dy dz \\ (u_x T_x - (u_x + \frac{\partial u}{\partial x} dx)(T_x + \frac{\partial T}{\partial x} dx))(\rho_{fluid} C_{pfluid}) dy dz &= (\rho_{fluid} C_{pfluid}) \frac{\partial T}{\partial t} dx dy dz \\ \frac{\partial \rho}{\partial x} \approx 0; \frac{\partial y}{\partial x} \approx 0; \frac{\partial z}{\partial x} \approx 0; \frac{\partial u}{\partial x} \approx 0 & \\ (T_x - (T_x + \frac{\partial T}{\partial x} dx))u_x(\rho_{fluid} C_{pfluid}) dy dz &= (\rho_{fluid} C_{pfluid}) \frac{\partial T}{\partial t} dx dy dz \\ -(\frac{\partial T}{\partial x} dx) \dot{m}_{nett}(C_{pfluid}) &= (\rho_{fluid} C_{pfluid}) \frac{\partial T}{\partial t} dx dy dz \\ -\frac{\dot{m}_{nett}(C_{pfluid})}{A_{aw}(\rho_{fluid} C_{pfluid})} (\frac{\partial T}{\partial x}) &= \frac{\partial T}{\partial t} \end{aligned} \quad (F.9)$$

It was assumed that the control volume was of small enough infinitesimal small size that the change in speed and mass flow is of the first-order Taylor expression. Furthermore, the fluid is assumed to be incompressible with the mass flux flowing is equal to the mass flux that is flowing out. This concludes the derivation of the advection term of the PDE.

F.2 Method of lines

The PDE describing the process has been formulated. The focus will now fall on how to solve this equation. The method of lines methodology was employed to help solve these equations.

The former technique entails using finite difference approximations for partial derivative terms in all variables except one. The former is left continuous and a weak formulation of the original problem is obtained in the form of an ODE. This can then be solved using the techniques derived for the numerical solving of ODEs from literature.

It is assumed that the problem has continuous partial derivatives in the variable space to the power of three. It is further assumed that the problem can be expressed by the famous Taylor power series from Calculus. A grid has to be established for this method. In the direction of flow, the system is subdivided into m discrete layers along the pipes and tanks. The step size is further defined as $h = l/m$ where l defines the length of the tank/pipe system. Furthermore, a time step of k is selected. The grid for this system is thus defined as (x_i, t_j, r_k) , where the variable x_i is defined as $x_i = ih$, further with $i = 0, 1, \dots, m$, and the time-step defined as $t_j = jk$ with $j = 0, 1, 2, \dots$. The pipe is furthermore divided into various concentric cylinders. Thus a number $n > 0$ exists whereas the radius of the h^{th} layer is defined as R . The h^{th} layer is also divided into n concentric circles where the radius of these circles are defined as: $r_k = mk$ with $m = R/n$ and $k = 0, 1, \dots, n$. The Taylor series is used and is defined as:

$$f(x) = \sum_{n=0}^{\infty} \frac{f^n(a)}{n!} (x-a)^n \quad (F.10)$$

$$f(x) = f(a) + \frac{f'(a)}{1!} (x-a) + \frac{f''(a)}{2!} (x-a)^2 + \frac{f'''(a)}{3!} (x-a)^3 + \dots \quad (F.11)$$

A function $T(x_i, t_j)$ is to be approximated that describes the change in temperature in the fluid system due to fluid location (x_i) and time (t_j). The difference method is used using the above series in the variable t to obtain the following difference quotient:

$$\frac{\partial T}{\partial t}(x_i, t_j) = \frac{T(x_i, t_j + k) - T(x_i, t_j)}{k} - \frac{k}{2} \frac{\partial^2 T(x_i, \mu_j)}{\partial t^2} \quad (F.12)$$

Whereas $\mu_j \in (t_j, t_{j+1})$ and thus the Taylor series in x is used to form the difference quotient in the spatial parameters x_i as:

$$\frac{\partial^2 T}{\partial x^2}(x_i, t_j) = \frac{T(x_i + h, t_j) - 2T(x_i, t_j) + T(x_i - h, t_j)}{h^2} - \frac{h^2}{12} \frac{\partial^4 T(\phi_i, t_j)}{\partial x^4} \quad (F.13)$$

Whereas $\phi_i \in (x_{i-1}, x_{i+1})$. The same can be done for the first-order difference coefficient in the spatial parameter r and x :

$$\frac{\partial T}{\partial r}(r_k, t_j) = \frac{T(r_k + m, t_j) - T(r_k, t_j)}{m} - \frac{m}{2} \frac{\partial^2 T(\eta, t_j)}{\partial r^2} \quad (F.14)$$

$$\frac{\partial T}{\partial x}(x_i, t_j) = \frac{T(x_i + h, t_j) - T(x_i, t_j)}{h} - \frac{h}{2} \frac{\partial^2 T(\zeta, t_j)}{\partial x^2} \quad (F.15)$$

The same is done for the time parameter and, thus, the following difference coefficient is obtained:

$$\frac{\partial T}{\partial t}(x_i, t_j) = \frac{T(x_i, t_j + k) - T(x_i, t_j)}{k} - \frac{k}{2} \frac{\partial^2 T(\zeta, t_j)}{\partial t^2} \quad (\text{F.16})$$

Whereas $\eta_i \in (r_i, r_{i+1})$ and $\zeta_i \in (x_i, x_{i+1})$. Neglecting the higher-order Taylor terms an approximation to the partial derivatives can be obtained. Thus the original PDE describing the temperature in the fluid element can be rewritten as:

$$\frac{T(x_i, t_j + k) - T(x_i, t_j)}{k} = \frac{k_{fluid}}{\rho_{fluid} C p_{fluid}} \left(\frac{T(x_i + h, t_j) - 2T(x_i, t_j) + T(x_i - h, t_j)}{h^2} \right) - \frac{\dot{m}_{nett} C p_{fluid}}{A_w \rho_{fluid} C p_{fluid}} \left(\frac{T(x_i + h, t_j) - T(x_i, t_j)}{h} \right) + \frac{2h_{sidewalls}}{\rho_{fluid} C p_{fluid}} \left(\frac{T(r_k + m, t_j) - T(r_k, t_j)}{m} \right) \quad (\text{F.17})$$

It is assumed that the ambient temperature can be expressed by the function T_a that represents the ambient temperature conditions at time-step j and is expressed as:

$$T_{amb} = T_a(t_j) \quad \forall j \geq 0$$

Moreover, the size of the grid will be chosen as such that the former difference quotient in parameter r reduces to the difference between the bulk fluid temperature and the dry-bulb ambient temperature around the pipe or tank system.

$$\frac{T(r_k + m, t_j) - T(r_k, t_j)}{m} \approx \frac{T_a(t_j) - T_{amb}}{R} \quad (\text{F.18})$$

Thus the difference quotient form of the original PDE for the temperature in a fluid layer can be expressed as:

$$\begin{aligned} \frac{T(x_i, t_j + k) - T(x_i, t_j)}{k} &= \frac{k_{fluid}}{\rho_{fluid} C p_{fluid}} \left(\frac{T(x_i + h, t_j) - 2T(x_i, t_j) + T(x_i - h, t_j)}{h^2} \right) - \frac{\dot{m}_{nett} C p_{fluid}}{A_w \rho_{fluid} C p_{fluid}} \left(\frac{T(x_i + h, t_j) - T(x_i, t_j)}{h} \right) + \\ &\quad \frac{h_{sidewalls} P_{perimeter}}{A_w \rho_{fluid} C p_{fluid}} (T_a(t_j) - T_{amb}) \\ A_w h \rho_{fluid} C p_{fluid} \frac{T(x_i, t_j + k) - T(x_i, t_j)}{k} &= k_{fluid} A_w \frac{T(x_i + h, t_j) - 2T(x_i, t_j) + T(x_i - h, t_j)}{h} - \dot{m}_{out} C p_{fluid} (T(x_i + h, t_j) - T(x_i, t_j)) + \\ &\quad \dot{m}_{in} C p_{fluid} (T(x_i - h, t_j) - T(x_i, t_j)) + h_{sidewalls} P_{perimeter} (T(x_i, t_j) - T_a(t_j)) \end{aligned}$$

Because the control volumes are of the same size we can divide by the step-size h . The former PDE has now been reduced into a ODE expression. This equation can also be expressed in the enthalpy form where as $h_{i,j} = T(x_i, t_j)$:

$$\begin{aligned} m_{element} \frac{h(x_i, t_j + k) - h(x_i, t_j)}{k} &= k_{fluid} A_w \frac{T(x_i + h, t_j) - 2T(x_i, t_j) + T(x_i - h, t_j)}{h} - \dot{m}_{out} (h(x_i + h, t_j) - h(x_i, t_j)) + \\ &\quad \dot{m}_{in} (h(x_i - h, t_j) - h(x_i, t_j)) + h_{sidewalls} P_{perimeter} (T(x_i, t_j) - T_a(t_j)) \\ m_{element} \frac{h(x_i, t_j + k) - h(x_i, t_j)}{k} &= k_{fluid} A_w \frac{T(x_i + h, t_j) - T(x_i, t_j)}{h} + k_{fluid} A_w \frac{T(x_i - h, t_j) - T(x_i, t_j)}{h} - \dot{m}_{out} (h(x_i + h, t_j) - h(x_i, t_j)) + \\ &\quad \dot{m}_{in} (h(x_i - h, t_j) - h(x_i, t_j)) + h_{sidewalls} P_{perimeter} (T(x_i, t_j) - T_a(t_j)) \\ h(x_i, t_j + k) &= h(x_i, t_j) + \frac{k}{m_{element}} [k_{fluid} A_w \frac{T(x_i + h, t_j) - T(x_i, t_j)}{h} + k_{fluid} A_w \frac{T(x_i - h, t_j) - T(x_i, t_j)}{h} - \\ \dot{m}_{out} (h(x_i + h, t_j) - h(x_i, t_j)) + \dot{m}_{in} (h(x_i - h, t_j) - h(x_i, t_j)) + h_{sidewalls} P_{perimeter} (T(x_i, t_j) - T_a(t_j))] \\ \frac{\partial \rho}{\partial t} &\approx 0 \\ \dot{m}_{in} &= \dot{m}_{out} \\ h(x_i, t_j + k) &= h(x_i, t_j) + \frac{k}{m_{element}} [k_{fluid} A_w \frac{T(x_i + h, t_j) - T(x_i, t_j)}{h} + k_{fluid} A_w \frac{T(x_i - h, t_j) - T(x_i, t_j)}{h} - \\ &\quad \dot{m}_{out} (h(x_i + h, t_j) - h(x_i, t_j)) + \dot{m}_{in} (h(x_i - h, t_j) - h(x_i, t_j)) + h_{sidewalls} P_{perimeter} (T(x_i, t_j) - T_a(t_j))] \end{aligned}$$

This equation can be integrated to obtain the temperature in the next time-step using general numerical methods that have been developed for ODE methods.

F.3 Runge-Kutte

In the reduced model the fourth order Runge-Kutte method is employed. The ODE that was derived in the former section is applicable to a single component. To solve the fluid properties throughout the whole system the Runge-Kutte method for systems have to be used. It is noteworthy that the derived ODE of the former section is an autonomous ODE ($y' = y(t)$). A non-autonomous ODE is explicitly defined by a time parameter t or is of the form $y' = y(t) + t$. The general expression for this method can be written as:

$$\begin{aligned} y(x_o) &= y_o \\ k_1 &= hf(x_n, y_n) \\ k_2 &= hf(x_n + \frac{1}{2}h, y_n + \frac{1}{2}k_1) \\ k_3 &= hf(x_n + \frac{1}{2}h, y_n + \frac{1}{2}k_2) \\ k_4 &= hf(x_n + h, y_n + k_3) \\ \forall n &= 0, 1, 2, \dots, n-1 \end{aligned}$$

The derived ODE can be rewritten in the RK format as follows:

$$\begin{aligned} k_{1i,j} &= \frac{k}{m_{element}} [k_{fluid} A_w \frac{T(x_i+h, t_j) - T(x_i, t_j)}{h} + k_{fluid} A_w \frac{T(x_i-h, t_j) - T(x_i, t_j)}{h} - \dot{m}_{out}(h(x_i + h, t_j)) + \\ &\quad \dot{m}_{in}(h(x_i - h, t_j)) + h_{sidewalls} P_{perimeter}(T(x_i, t_j) - T_a(t_j))] \\ k_{2i,j} &= \frac{k}{m_{element}} [k_{fluid} A_w \frac{(T(x_i+h, t_j) + 0.5k_{1i+h,j}/temp) - (T(x_i, t_j) + 0.5k_{1i,j}/temp)}{h} + \\ &\quad k_{fluid} A_w \frac{(T(x_i-h, t_j) + 0.5k_{1i-h,j}/temp) - (T(x_i, t_j) + 0.5k_{1i,j}/temp)}{h} - \dot{m}_{out}(h(x_i + h, t_j) + 0.5k_{1i+h,j}) + \dot{m}_{in}(h(x_i - \\ &\quad h, t_j) + 0.5k_{1i-h,j}) + h_{sidewalls} P_{perimeter}((T(x_i, t_j) + 0.5k_{1i,j}/temp) - T_a(t_j))] \\ k_{3i,j} &= \frac{k}{m_{element}} [k_{fluid} A_w \frac{(T(x_i+h, t_j) + 0.5k_{1i+h,j}/temp) - (T(x_i, t_j) + 0.5k_{2i,j}/temp)}{h} + \\ &\quad k_{fluid} A_w \frac{(T(x_i-h, t_j) + 0.5k_{2i-h,j}/temp) - (T(x_i, t_j) + 0.5k_{2i,j}/temp)}{h} - \dot{m}_{out}(h(x_i + h, t_j) + 0.5k_{2i+h,j}) + \dot{m}_{in}(h(x_i - \\ &\quad h, t_j) + 0.5k_{2i-h,j}) + h_{sidewalls} P_{perimeter}((T(x_i, t_j) + 0.5k_{2i,j}/temp) - T_a(t_j))] \\ k_{4i,j} &= \frac{k}{m_{element}} [k_{fluid} A_w \frac{(T(x_i+h, t_j) + k_{3i+h,j}/temp) - (T(x_i, t_j) + k_{3i,j}/temp)}{h} + \\ &\quad k_{fluid} A_w \frac{(T(x_i-h, t_j) + k_{3i-h,j}/temp) - (T(x_i, t_j) + k_{3i,j}/temp)}{h} - \dot{m}_{out}(h(x_i + h, t_j) + k_{3i+h,j}) + \dot{m}_{in}(h(x_i - h, t_j) + \\ &\quad k_{3i-h,j}) + h_{sidewalls} P_{perimeter}((T(x_i, t_j) + k_{3i,j}/temp) - T_a(t_j))] \end{aligned}$$

As the reduced model used in EES explicitly describes the time evolution of the enthalpy term; a expression for the temperature terms had to be defined. This was also necessary as to be used in the conductive and convective terms of the energy equation in the RK formulation.

$$k_{1-temperature} = f(k_1) \quad (F.19)$$

$$k_{1i,j/temp} = f(h(x_i, t_{j-1}) + (0.5 \times k_{1i,j})) \quad (F.20)$$

$$k_{2i,j/temp} = f(h(x_i, t_{j-1}) + (0.5 \times k_{2i,j})) \quad (F.21)$$

$$k_{3i,j/temp} = f(h(x_i, t_{j-1}) + k_{3i,j}) \quad (F.22)$$

The next time-step enthalpy can be calculated from:

$$k_{1i,j+1} = k_{1i,j} + \frac{1}{6}(k_1 + 2k_2 + 2k_3 + k_4) \quad (F.23)$$

Appendix G

Fundamental equation derivation - Simulations

The thermo-hydraulic simulation consists of various components, each having a set of equations that govern fluid flow. Some fundamental equations are often shared between the various components. Accordingly, a particular case was observed to derive these basic equations. In this section, Newtons' second law of motion (conservation of linear momentum), the first law of thermodynamics (conservation of energy) along with the conservation of mass will be used to derive the basic equations for the reduced and detailed simulations. Conservation of species will be neglected for the purposes of this study.

Consider a stationary, non-deforming control volume that partially encompasses an incompressible fluid at first (Figure G.1). Consider this same infinitesimally small control volume to be filling up due to mass fluxes. Finally assume the dimensions of this control volume to be dx , dy and dz respectively with the former dimensions all of the same magnitude. The velocity components of mass flowing through this control volume is defined as u, v and w in the x, y and z directions respectively. With the control volume defined; the conservation of mass, energy and momentum will be discussed in the following sections.

G.0.1 Conservation of mass

The mass has to be conserved in the control volume. Thus, an expression is derived for the mass accumulation in the control volume. The control volume (dV) is fixed, so the change in density (ρ_{cv}) of the control volume should express the change in mass. An incompressible fluid is observed; thus, the fluid properties such as the density of the fluid (ρ_{fluid}) remains constant:

$$\frac{\partial \rho_{fluid}}{\partial t} = 0 \quad (G.1)$$

However, as mass fluxes flow into the control volume (due to it only partially encompassing the fluid), the density of the control volume (ρ_{cv}) changes. Thus $\frac{\partial \rho_{cv}}{\partial t}$ is not necessarily equal to zero. One can approximate the change in density of the control volume ($\frac{\partial}{\partial t} \int_{cv} \rho dV_{CV}$) as follows:

$$\frac{\partial}{\partial t} \int_{cv} \rho dV_{CV} \approx \frac{\partial \rho_{cv}}{\partial t} \delta x \delta y \delta z = \Sigma \dot{m}_{nett} \quad (G.2)$$

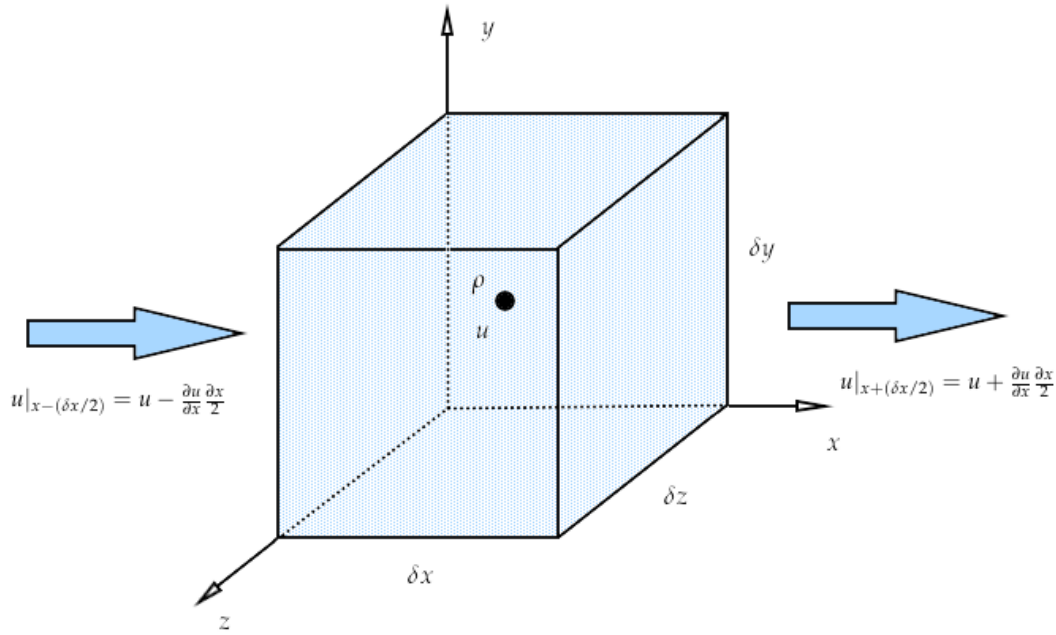


FIGURE G.1: An stationary, non-deforming control volume that encompasses partially an incompressible fluid at first and is of infinitesimally small size.

The mass flux ($\Sigma \dot{m}_{nett}$) through the control volume can be obtained by assuming a constant linear acceleration of the fluid through the control volume (Figure G.2). In other words the Taylor expression of the velocity in the x direction is used and the higher order terms are neglected to obtain the following:

$$u|_{x+(\delta x/2)} = u + \frac{\partial u}{\partial x} \frac{\delta x}{2} \quad (G.3)$$

$$u|_{x-(\delta x/2)} = u - \frac{\partial u}{\partial x} \frac{\delta x}{2} \quad (G.4)$$

By multiplying Equations G.3 and G.4 by the fluid density (ρ_{fluid}) and differential area (normal to which the fluid is flowing through), one obtains the following velocity equations:

$$u|_{x+(\delta x/2)}(\rho \delta y \delta z) = (u(\rho \delta y \delta z)) + \left(\frac{\partial u}{\partial x} \frac{\delta x}{2} (\rho \delta y \delta z) \right) \quad (G.5)$$

$$u|_{x-(\delta x/2)}(\rho \delta y \delta z) = (u(\rho \delta y \delta z)) - \left(\frac{\partial u}{\partial x} \frac{\delta x}{2} (\rho \delta y \delta z) \right) \quad (G.6)$$

Subtracting Equations G.5 and G.6 above one can obtain a expression for the net mass flux in the x direction:

$$u|_{x+(\delta x/2)}(\rho \delta y \delta z) - u|_{x-(\delta x/2)}(\rho \delta y \delta z) = (u(\rho \delta y \delta z)) + \left(\frac{\partial u}{\partial x} \frac{\delta x}{2} (\rho \delta y \delta z) \right) - (u(\rho \delta y \delta z)) - \left(\frac{\partial u}{\partial x} \frac{\delta x}{2} (\rho \delta y \delta z) \right) \quad (G.7)$$

$$\frac{\partial \rho_{fluid} u}{\partial x} \delta x \delta y \delta z = (u(\rho \delta y \delta z)) + \left(\frac{\partial u}{\partial x} \frac{\delta x}{2} (\rho \delta y \delta z) \right) - (u(\rho \delta y \delta z)) - \left(\frac{\partial u}{\partial x} \frac{\delta x}{2} (\rho \delta y \delta z) \right) \quad (G.8)$$

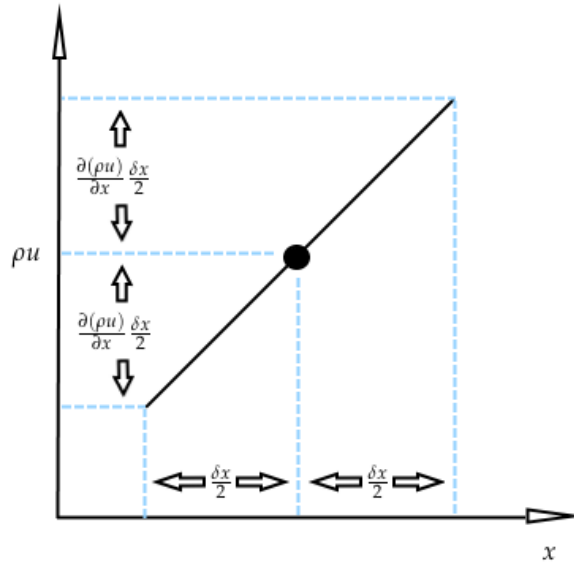


FIGURE G.2: Graphical representation of Taylor expression for fluid flow velocity through the control volume.

By applying the same methodology in the y and z directions, respectively, the following can be obtained:

$$\frac{\partial \rho_{fluid} v}{\partial y} \delta x \delta y \delta z = (v(\rho \delta x \delta z)) + \left(\frac{\partial v}{\partial y} \frac{\delta y}{2} (\rho \delta x \delta z) \right) - (v(\rho \delta x \delta z)) - \left(\frac{\partial v}{\partial y} \frac{\delta y}{2} (\rho \delta x \delta z) \right) \quad (G.9)$$

$$\frac{\partial \rho_{fluid} w}{\partial z} \delta x \delta y \delta z = (w(\rho \delta y \delta x)) + \left(\frac{\partial w}{\partial z} \frac{\delta z}{2} (\rho \delta y \delta x) \right) - (w(\rho \delta y \delta x)) - \left(\frac{\partial w}{\partial z} \frac{\delta z}{2} (\rho \delta y \delta x) \right) \quad (G.10)$$

As stated before the change in control volume density (ρ_{cv}) can only be attributed to the mass flux ($\Sigma \dot{m}_{nett}$) moving through the partially filled control volume. The change in fluid density¹ (ρ_{fluid}) is zero. By adding the mass flux equations in all three directions (Equations G.8, G.9 and G.10) and the temporal change in mass term (Equation G.2); the following equation was obtained:

$$\frac{\partial \rho_{cv}}{\partial t} \delta x \delta y \delta z + \frac{\partial \rho_{fluid} u}{\partial x} \delta x \delta y \delta z + \frac{\partial \rho_{fluid} v}{\partial y} \delta x \delta y \delta z + \frac{\partial \rho_{fluid} w}{\partial z} \delta x \delta y \delta z = 0 \quad (G.11)$$

By assuming all surfaces of the control volume to have the same differential area (dA), then one can rewrite Equation G.11 to:

$$\frac{\partial \rho_{cv}}{\partial t} \forall_{cv} + \frac{\partial \rho_{fluid} u}{\partial x} \delta x dA + \frac{\partial \rho_{fluid} v}{\partial y} \delta y dA + \frac{\partial \rho_{fluid} w}{\partial z} \delta z dA = 0 \quad (G.12)$$

¹The system fluid density (ρ_{fluid}) includes the density of the fluid mass fluxes crossing the control volume boundaries and the control volume content.

$$\frac{\partial \rho_{cv}}{\partial t} \forall_{cv} + \left(\frac{\partial u}{\partial x} \delta x dA + \frac{\partial v}{\partial y} \delta y dA + \frac{\partial w}{\partial z} \delta z dA \right) \times \rho_{fluid} = 0 \quad (G.13)$$

It can be recognised from Figure G.2 and Equation G.13, that the average linear acceleration ($\frac{\partial u}{\partial x}$) times the axial distance in which the average velocity travels (δx) in the differential volume, results into the average speed of that component through the control volume ($u_{average}$). In symbolic terms:

$$\frac{\partial u}{\partial x} \delta x = u_{average} \quad (G.14)$$

$$\frac{\partial v}{\partial y} \delta y = v_{average} \quad (G.15)$$

$$\frac{\partial w}{\partial z} \delta z = w_{average} \quad (G.16)$$

Thus the accumulation of mass ($\frac{\partial \rho_{cv}}{\partial t} \forall_{cv}$) in the control volume is dependant on the average flow rate of the various velocity components **normal** to the control surfaces of the control volume:

$$\frac{\partial \rho_{cv}}{\partial t} \forall_{cv} + (u_{average} + v_{average} + w_{average}) \times \rho_{fluid} dA = 0 \quad (G.17)$$

Define the net average volumetric flow rate in the x , y and z directions normal to the respective control surfaces as $\dot{Q}_x$, $\dot{Q}_y$ and $\dot{Q}_z$.

$$\frac{\partial \rho_{cv}}{\partial t} \forall_{cv} + (\dot{Q}_x + \dot{Q}_y + \dot{Q}_z) \times \rho_{fluid} = 0 \quad (G.18)$$

It is assumed that the fluid is incompressible ($\frac{\partial \rho_{fluid}}{\partial t} \approx 0$), and all of the system components are encompassing the fluid fully with no component partially filled ($\frac{\partial \rho_{cv}}{\partial t} \forall_{cv} \approx 0$). For this system and simulation, it can be concluded that the system and control volume densities are the same ($\rho_{cv} \approx \rho_{fluid}$). Under these assumptions, Equation G.21 can be rewritten to give the mass fluxes as:

$$\dot{m}_x + \dot{m}_y + \dot{m}_z = 0 \quad (G.19)$$

$$\sum \dot{m}_{in} - \sum \dot{m}_{out} = 0 \quad (G.20)$$

As a final validation, by dividing Equation G.12 with the control volume ($\forall_{cv}$) the differential form of the continuity equation or the continuity of mass of the Navier-Stokes equations can be obtained:

$$\frac{\partial \rho_{cv}}{\partial t} + \frac{\partial \rho_{fluid} u}{\partial x} + \frac{\partial \rho_{fluid} v}{\partial y} + \frac{\partial \rho_{fluid} w}{\partial z} = 0 \quad (G.21)$$

G.0.2 Conservation of energy

Reynolds theorem for control volumes supports the finite difference methodology showcased above and, thus, the same can be done for the rate of energy ($\frac{\partial e}{\partial t}$) change in a fixed, non-deforming control volume:

$$\frac{D}{Dt} \int e \rho d\forall_{sys} = \frac{\partial}{\partial t} \int e \rho d\forall_{cv} + \int e \rho \mathbf{V} \cdot \mathbf{n} dA \quad (G.22)$$

From the first law of Thermodynamics energy can only change form and have to be conserved. The above equation states that the rate of change in energy of a system ($\frac{D}{Dt} \int e \rho d\forall_{sys}$), must be equal to the sum of the rate of change of energy in the control volume ($\frac{\partial}{\partial t} \int e \rho d\forall_{cv}$), and the net rate at which energy leaves and enters the control volume through its surfaces ($\int e \rho \mathbf{V} \cdot \mathbf{n} dA$).

The $\frac{D}{Dt} \int e \rho d\forall_{sys}$ term of Equation G.22 refers to energy additions taking place by method of heat transfer or work done by the system or on the system. Thus this term can be rewritten as:

$$\frac{D}{Dt} \int e d\forall_{sys} = (\sum \dot{Q}_{in} - \sum \dot{Q}_{out})_{sys} + (\sum \dot{W}_{in} - \sum \dot{W}_{out})_{sys} \quad (G.23)$$

However, in this case study, no work is done on the system or by the system. Energy is mainly transferred through heat transfer. In the detailed simulation, however, the work performed on the fluid by the pumps ($\dot{W}_{pumps}$) are taken into account. Although in both detailed and reduced simulations, the energy is transferred mainly by conduction and convective heat transfer. Finally, energy addition is mainly taking place utilizing the ASHP and to a lesser extent, the SWH. Thus for the reduced simulation:

$$(\sum \dot{Q}_{in} - \sum \dot{Q}_{out})_{sys} = \sum \dot{Q}_{convective} + \sum \dot{Q}_{conductive} \quad (G.24)$$

Consequently, for the detailed simulation:

$$(\sum \dot{Q}_{in} - \sum \dot{Q}_{out})_{sys} + \sum \dot{W}_{pumps} = \sum \dot{Q}_{convective} + \sum \dot{Q}_{conductive} + \sum \dot{Q}_{friction} + \sum \dot{W}_{pumps} \quad (G.25)$$

In Equation G.22, $\frac{\partial}{\partial t} \int e d\forall_{cv}$ refers to the rate of energy added to the control volume. Thermal capacitance was neglected ($\frac{\partial}{\partial t} \int e d\forall_{cv} \approx 0$) for components with high mass flow rates and small volumes in the reduced simulation. Mostly on the various pipe control volumes. This was mainly assumed as the smallest volume in a simulation generally restricts the maximum numerical time step. Consequently, slowing down simulation times. Furthermore, the capacitance of these components is relatively low when compared to the sensible storage vessels. These components will be referred to as quasi-steady.

The reduced simulation does, however, take into account the thermal capacitance of the large storage vessels. In the detailed simulation that was developed, all components have thermal capacitance, and the thermal capacitance was never assumed as zero.

The final term in Equation G.22, refers to the energy fluxes crossing the control volume. Thus for all of the components in both simulations; the nett energy flux can be expressed as the energy times the average mass flow crossing the system control surfaces:

$$\int e \mathbf{V} \cdot \mathbf{n} dA = \sum \dot{m}_{out} e_{out} - \sum \dot{m}_{in} e_{in} \quad (G.26)$$

Elevation energy is ignored in the reduced and detailed simulations, and it was consequently assumed that the energy could be approximated by the total enthalpy:

$$e = u + Pv + V^2/2 + gz \quad (G.27)$$

$$e \approx h + V^2/2 = h_0 \quad (G.28)$$

From classical Thermodynamics, enthalpy can be expressed as the internal energy plus the pressure energy. Whereas the pressure energy refers to total pressure energy:

$$h = u + Pv \quad (G.29)$$

Thus to conclude, for components with thermal capacitance in the reduced simulation; the energy equation in Euler integrand form reduces to:

$$\sum \dot{Q}_{convective} + \sum \dot{Q}_{conductive} = \frac{h_{i+1} - h_i}{\delta t} \forall_{cv} + (\sum \dot{m}_{out} e_{out} - \sum \dot{m}_{in} e_{in}) \quad (G.30)$$

In the detail simulation, the work exercised by pumps are taken into account, and Equation G.30 is rewritten as:

$$\sum \dot{Q}_{convective} + \sum \dot{Q}_{conductive} + \sum \dot{Q}_{friction} + \sum \dot{W}_{pumps} = \frac{h_{i+1} - h_i}{\delta t} \forall_{cv} + (\sum \dot{m}_{out} e_{out} - \sum \dot{m}_{in} e_{in}) \quad (G.31)$$

For quasi-steady components in the reduced simulation, the energy equation reduces to:

$$\sum \dot{Q}_{convective} + \sum \dot{Q}_{conductive} = (\sum \dot{m}_{out} e_{out} - \sum \dot{m}_{in} e_{in}) \quad (G.32)$$

G.0.3 Conservation of linear momentum

Mass and energy equations have been linked in the previous paragraphs; an expression is to be obtained for the drop in pressure energy of the enthalpy term. This expression will only be required for the detailed simulations, as the reduced model explicitly enforces mass flows in the various branches.

The linear momentum equation will be critical to determine the flow distribution in the pipe network by establishing a relationship between continuity of linear momentum, mass and energy. If a small mass particle $\rho d\forall$ is observed, then the linear momentum for this particle can be expressed as $\mathbf{V} \rho d\forall$. If the same control volume approach is followed for the linear momentum, then the following equation from Reynolds theorem is obtained:

$$\frac{D}{Dt} \int \mathbf{V} \rho d\forall_{sys} = \frac{\partial}{\partial t} \int \mathbf{V} \rho d\forall_{cv} + \int \mathbf{V} \rho \mathbf{V} \cdot \mathbf{n} dA \quad (G.33)$$

Combining Equation G.33 with Newton's law of motion, the following is obtained:

$$\frac{D}{Dt} \int \mathbf{V} \rho dV_{sys} = \sum F_{sys} \quad (G.34)$$

Equation G.34 is divided into surface and body forces. In this study, body forces such as gravity are neglected. Surface forces on the control volume are inertial, non-deforming and can be approximated as steady flow. The assumption is that the change in linear momentum is not of significant difference between the various time steps. Thus phenomenon such as water hammer is neglected. Also, the control volumes of most of the pipes do not change in diameter significantly as to allow for momentum to accumulate in these control volumes significantly when the flowrate changes. Furthermore, the study focuses on incompressible flow. Thus it is assumed that the change in linear momentum of the control volume can be expressed as:

$$\frac{\partial}{\partial t} \int \mathbf{V} \rho dV_{cv} \approx 0 \quad (G.35)$$

Consequently, Equation G.33 reduces to the change in momentum fluxes over the various control surfaces:

$$\int \mathbf{V} \rho \mathbf{V} \cdot \mathbf{n} dA = \sum \dot{m} \mathbf{V} \quad (G.36)$$

Under the assumptions of the previous paragraphs and Equations G.34, G.35 and G.36; Equation G.33 reduces to:

$$\sum F_{sys} = \sum \dot{m} \mathbf{V} \quad (G.37)$$

External forces ($\sum F_{sys}$) includes wall shear stresses and the forces of the component wall on the fluid element. As the developed simulation will mostly deal with hydrodynamic problems, the focus will fall on flow within pipes. The detailed simulation takes the friction forces into account that offers resistance in the various pipes. The reduced model does not take these into account as the mass flow in the network is defined in an explicit manner. Thus from Equation G.37, the following is obtained:

$$R_x + p_{in} dA - p_{out} dA = \dot{m}_{in} V_{in} - \dot{m}_{out} V_{out} \quad (G.38)$$

$$\text{under the assumptions of } \frac{\partial \rho_{fluid}}{\partial t} \approx 0$$

$$\models V_{in} = V_{out}$$

Equation G.38 reduces to

$$R_x + p_{in} dA - p_{out} dA = 0 \quad (G.39)$$

The total pressure is quantified as $p = p_o + 1/2 \rho V^2$. R_x in Equation G.39 refers to the shear stresses incurred on the fluid. Reynolds theorem (as a control volume approach) only establishes that such a force exist but does not aid in quantifying this force. The assumptions in the following paragraph will lead to an expression that can be used to calculate the resistance force.

Consider the viscous flow through a horizontal circular tube with radius (R) Figure G.3. Furthermore, it is assumed that the flow is parallel to the pipe walls with $v_r = 0$ and $v_\theta = 0$.

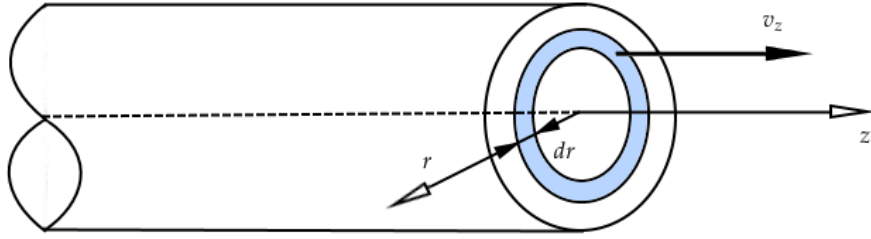


FIGURE G.3: A differential annular ring in a pipe.

Also, it is assumed that the flow is asymmetric steady and that the flow velocity (v_z) in the z -direction is independent of t and θ so that the velocity of the flow is only a function of the radial coordinate on the flow $v_z = f(r)$. Thus from the famous Navier-Stokes equations for cylindrical polar coordinates in the r direction, that can be written as:

$$\rho \left(\frac{\partial v_r}{\partial t} + v_r \frac{\partial v_r}{\partial r} + \frac{v_\theta}{r} \frac{\partial v_r}{\partial \theta} - \frac{v_\theta^2}{r} + v_z \frac{\partial v_r}{\partial z} \right) = -\frac{\partial p}{\partial r} + \rho g_r + \mu \left(\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial v_r}{\partial r} \right) - \frac{v_r}{r^2} + \frac{1}{r^2} \frac{\partial^2 v_r}{\partial \theta^2} - \frac{2}{r^2} \frac{\partial v_\theta}{\partial \theta} + \frac{\partial^2 v_r}{\partial z^2} \right) \quad (\text{G.40})$$

From the assumptions expressed in the former paragraph, Equation G.40 reduces to:

$$0 = -\frac{\partial p}{\partial z} + \mu \left(\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial v_z}{\partial r} \right) \right) \quad (\text{G.41})$$

One can assume that for a given length of pipe (L) that the pressure drop ($\frac{\partial p}{\partial z}$) is constant. Thus Equation G.41 is integrated twice to obtain the following two equations:

$$r \frac{\partial v_z}{\partial r} = \frac{1}{2\mu} \left(\frac{\partial p}{\partial z} r^2 \right) + c_1 \quad (\text{G.42})$$

$$v_z = \frac{1}{4\mu} \left(\frac{\partial p}{\partial z} \right) r^2 + c_1 \ln(r) + c_2 \quad (\text{G.43})$$

The shape of the velocity profile is dependant on the constants of integration c_1 and c_2 . With the boundary conditions chosen the flow (Q) through a pipe can be determined by multiplying the averaged flow velocity ($V_{average}$) and the flow area (A). It can be shown from the former two equations and dimensional analyses that:

$$Q = AV_{average} = \frac{\frac{\pi}{4C} \Delta p D^4}{\mu l} \quad (\text{G.44})$$

Where C has been determined theoretically and experimentally for selected geometry. Rewriting Equation G.44 into the famous Darcy-Weisbach form Equation G.33 under the assumptions made reduces to:

$$\Delta h = f \frac{l}{D} \frac{V^2}{2g} \quad (\text{G.45})$$

$$\Delta p = f \frac{l}{D} \frac{\rho V^2}{2} \quad (\text{G.46})$$

By coupling, Equation G.46 with the conservation of energy (Equation G.31) and mass equation (Equation G.20) the flow distribution in the pipe network can be determined. The energy expended in the friction flow ($\dot{Q}_{friction}$) of a pipe forms part of the rate of energy added to the control volume term ($\frac{D}{Dt} \int e \rho dV_{sys}$), and can be expressed as:

$$\dot{Q}_{friction} = \frac{\Delta p \dot{m}}{\rho} \quad (\text{G.47})$$

$$\dot{Q}_{friction} = f \frac{l}{D} \frac{\rho V^2}{2} \times \frac{\dot{m}}{\rho} \quad (\text{G.48})$$

Equation G.48, in effect, implies that pressure differences move at the speed of sound. Thus friction losses incurred reduces the pressure energy. Mechanical work exerted on the fluid by the method of centrifugal pumps rises this energy to combat the friction losses and consequently pressure term in enthalpy. Considering it can be assumed that the fluid is incompressible, the specific volume does not change ($\frac{\partial v_{fluid}}{\partial t} \approx 0$). Thus mechanical work leads to a relatively large change in pressure energy (Pv) rather than internal energy in the fluid enthalpy. Leading to the mathematical coupling between mass, momentum and energy equations. The main equations derived from this section are listed in Table G.1. The difference between the reduced and detailed simulation also is easily illustrated from this table (G.1). With the fundamental equations and assumptions discussed, the focus will shift to the implementation of these equations in the following sections.

TABLE G.1: Reduced and detailed 1D model fundamental continuity equations.

Reduced 1D model	Detailed 1D model
<i>Continuity of mass</i>	
$0 = \sum \dot{m}_{in} - \sum \dot{m}_{outj}$	$0 = \sum \dot{m}_{in} - \sum \dot{m}_{outj}$
<i>Continuity of energy</i>	
Quasi-steady components: $\sum \dot{m}_{in} e_{in} - \sum \dot{m}_{out} e_{out}$ $= (\sum \dot{Q}_{in} - \sum \dot{Q}_{out})_{sys}$	All components $\sum \dot{m}_{in} e_{in} - \sum \dot{m}_{out} e_{out} + \frac{\partial}{\partial t} \int e \rho dV_{cv}$ $= (\sum \dot{Q}_{in} - \sum \dot{Q}_{out})_{sys} + \dot{W}_{pump} - \dot{Q}_{friction}$
For tank components: $\sum \dot{m}_{in} e_{in} - \sum \dot{m}_{out} e_{out} + \frac{\partial}{\partial t} \int e \rho dV_{cv}$ $= (\sum \dot{Q}_{in} - \sum \dot{Q}_{out})_{sys}$	
<i>Continuity of linear momentum</i>	
$\Delta p \approx 0$	$\Delta p = f \frac{l}{D} \frac{\rho V^2}{2}$
<i>Energy expression</i>	
$e \approx u$	$e \approx h = u + Pv$

TABLE G.2: Reduced and detailed 1D model fundamental equations of components.

Reduced 1D model	Detailed 1D model
Pipe component	
$\frac{m(u_{i+1}-u_i)}{\delta t} \approx 0$ $\Rightarrow \Sigma \dot{m}h = \dot{Q}_{convection}$	$\frac{m(u_{i+1}-u_i)}{\delta t} \neq 0$ $\Rightarrow \frac{m(u_{i+1}-u_i)}{\delta t} = \Sigma \dot{m}h - \dot{Q}_{heatloss}$ $\frac{\dot{m}(u_{i+1}-u_i)}{\delta t} = \Sigma \dot{m}h - \dot{Q}_{convection} - \dot{Q}_{friction} + \dot{W}_{pump}$
Storage tank component	
$\frac{m(h_{i+1}-h_i)}{\delta t} = \Sigma \dot{m}h + \Sigma \dot{Q}_{heatloss}$ $\Sigma \dot{Q}_{heatloss} = \Sigma \dot{Q}_{convection}$	$\frac{m(u_{i+1}-u_i)}{\delta t} = \Sigma \dot{m}h + \Sigma \dot{Q}_{heatloss}$ $\Sigma \dot{Q}_{heatloss} = \Sigma \dot{Q}_{convection} - \Sigma \dot{Q}_{friction}$
Pump component	
$\dot{m}_{pump-steady-state} = b$ $\dot{m}_{pump} = \dot{m}_{pump-steady-state} \delta(a-1)$	$\dot{W}_{pump} = \dot{m}g(a_1(\frac{\dot{m}}{\rho}3600)^2 + a_2(\frac{\dot{m}}{\rho}3600) + a_3)$
Heat Pump component	
$\dot{m}_{hp} = \dot{m}_{pump-steady-state}$ $\dot{Q}_{ashp-available} = f(T_{db}, \phi, T_{inlet})$ $\dot{Q}_{ashp-available} = \dot{m}_{hp}Cp(T_{fixed} - T_{inlet})$ $\Sigma \dot{Q}_{ashp-available} = \dot{Q}_{ashp-available}$ $\frac{m(u_{i+1}-u_i)}{\delta t} \approx 0$ $\Sigma \dot{m}h = \dot{Q}_{ashp-available}$	$\dot{W}_{hp-pump} = \dot{m}_{hp}g(a_1(\frac{\dot{m}_{hp}}{\rho}3600)^2 + a_2(\frac{\dot{m}_{hp}}{\rho}3600) + a_3)$ $\dot{Q}_{ashp-available} = f(T_{db}, \phi, T_{inlet})$ $\Sigma \dot{m}h = \dot{m}_{hp}Cp(T_{fixed} - T_{inlet})$ $\Sigma \dot{Q}_{HP} = \dot{Q}_{ashp-available} + \dot{W}_{hp-pump} - \dot{Q}_{friction}$ $\dot{Q}_{friction} = f \frac{l}{D} \frac{\rho V^2}{2} \times \frac{\dot{m}}{\rho}$ $\frac{m(u_{i+1}-u_i)}{\delta t} = \Sigma \dot{m}h + \dot{Q}_{HP} - \dot{Q}_{heatloss}$
T-piece component	
$\frac{m(u_{i+1}-u_i)}{\delta t} \approx 0$ $\dot{Q}_{heat-loss} \approx 0$ $\Sigma \dot{m}h = 0$	$\dot{Q}_{friction} \approx 0$ $\frac{m(u_{i+1}-u_i)}{\delta t} + \Sigma \dot{m}h = \Sigma \dot{Q}_{heatloss}$
Solar water heater component	
$A = f(I_{rr}, T_{db}, T_{inlet})$ $\eta_{solar} = a_1 A^2 + a_2 A + a_3$ $\dot{Q}_{solar} = \eta_{solar} \times I_{rr} \times A_c$ $\dot{Q}_{heatloss} = (1 - \eta_{solar}) \times I_{rr} \times A_c$ $\Sigma \dot{m}h = \dot{m}_{swh}Cp(T_{solar} - T_{inlet})$ $\dot{m}_{pump} = \dot{m}_{pump-steady-state} \delta(a-1)$ $\frac{m(u_{i+1}-u_i)}{\delta t} = \Sigma \dot{m}h - \dot{Q}_{heatloss} + \dot{Q}_{solar}$	$A = f(I_{rr}, T_{db}, T_{inlet})$ $\eta_{solar} = a_1 A^2 + a_2 A + a_3$ $\dot{Q}_{solar} = \eta_{solar} \times I_{rr} \times A_c$ $\dot{Q}_{heatloss} = (1 - \eta_{solar}) \times I_{rr} \times A_c$ $\Sigma \dot{m}h = \dot{m}_{swh}Cp(T_{solar} - T_{inlet})$ $\dot{W}_{swh-pump} = \dot{m}_{swh}g(a_1(\frac{\dot{m}_{swh}}{\rho}3600)^2 + a_2(\frac{\dot{m}_{swh}}{\rho}3600) + a_3)$ $\frac{m(u_{i+1}-u_i)}{\delta t} = \Sigma \dot{m}h - \dot{Q}_{heatloss} + \dot{Q}_{solar} + \dot{W}_{swh-pump}$
Extra restrictions	
$\Sigma \dot{m} = 0$ $\Sigma \dot{m}h = 0$ n/a	$\Sigma \dot{m} = 0$ $\Sigma \dot{m}h = 0$ $\Sigma \Delta p = 0$
Runge-Kutte MOL solver	Euler solver

Appendix H

ASHP Regression Model

This appendix will illustrate the derivation of the ASHP regression equation employed in the 1st law energy balance and the simulation programs developed in the study. The appendix is divided into the following sections:

- The system-level analyses on heat pumps are presented, and the determination of independent variables influencing system performance (H.1).
- The psychrometric analyses on relative humidity and its impact on ASHP performance are discussed (H.2).
- The statistical analyses on the proposed regression equation for the coefficient of performance is presented (H.3).
- The statistical analyses on the proposed regression equation for electrical consumption is presented (H.4).

H.1 Basic principles of the ASHP

The fundamental workings of the ASHP will be presented to showcase the main factors influencing ASHP performance. From the third law of thermodynamics, the entropy of a system is minimum at absolute zero. From this, it can be noted that the possibility to extract energy from air is theoretically possible for all temperatures above absolute zero. However, this is hindered by the practical limitations imposed by heat transfer and boundary conditions near the source and sink of the heat pump. Heatpumps, using the second law of thermodynamics, uses work input to transport energy by the method of heat transfer in the direction of opposing thermal gradient.

The ASHP works on the principles of utilising the latent heat and phase change properties of a refrigerant to transport heat between two spatial locations. The main components of an ASHP are the compressor, evaporator, condenser and capillary tube. Figure H.1 illustrates the typical vapour compression cycle employed in ASHP's with an accompanying typical T-s diagram. The condenser is connected to the cycle sink (points 3 and 4) and the evaporator to the cycle heat source (points 7 and 8). The refrigerant is at low pressure and temperature in the evaporator. (points 6 and 7). This consequently ensures that the relative warmer ambient air heats the refrigerant in the evaporator to superheated vapour (points 8 and 1) due to the favourable heat gradient. A compressor compresses the vapour to a higher operating pressure (point 2). The higher operating pressure in the condenser allows the higher specific volume of the refrigerant to create a favourable temperature gradient to the heat sink of the cycle. It was

allowing heat to be extracted from the sink to the source (points 3 and 4). The pressure at the outlet of the condenser has to be reduced to match the evaporator pressure (points 5 and 6) to create a closed cycle. A capillary tube is thus employed that creates a pressure drop, and due to irreversibility, the refrigerant enters at a higher entropy (points 6 and 7). For the ideal cycle, the HP is characterised mainly by the discharge pressure of the compressor and the heat transfer characteristics between the evaporator and the cold space.

The ASHP used in this study was designed to deliver a fixed outlet temperature. By utilising a system-level approach, it can be noted that the heat pump performance will be highly determinate on the heat source and sink conditions (Carnot efficiency). This will depend on the mass flow and thermal properties of the refrigerant and ambient air. Psychrometrics help illustrates the impact of relative humidity on heat pump performance and its eventual inclusion in the regression model. Thus from this section, one can conclude that for a specific heat pump, the following factors will impact the performance of the machine:

- Psychrometrics (T_{db}, RH) thus (Points 7 and 8)
- Inlet water temp (T_{inlet}) thus (Point 3 and 4)

The dry and wet bulb conditions determine the available enthalpy at the source (evaporator). Furthermore, the inlet water conditions determine the system conditions at the sink (condenser). The following section will aid in explaining the impact of relative humidity on the available energy to an ASHP cycle.

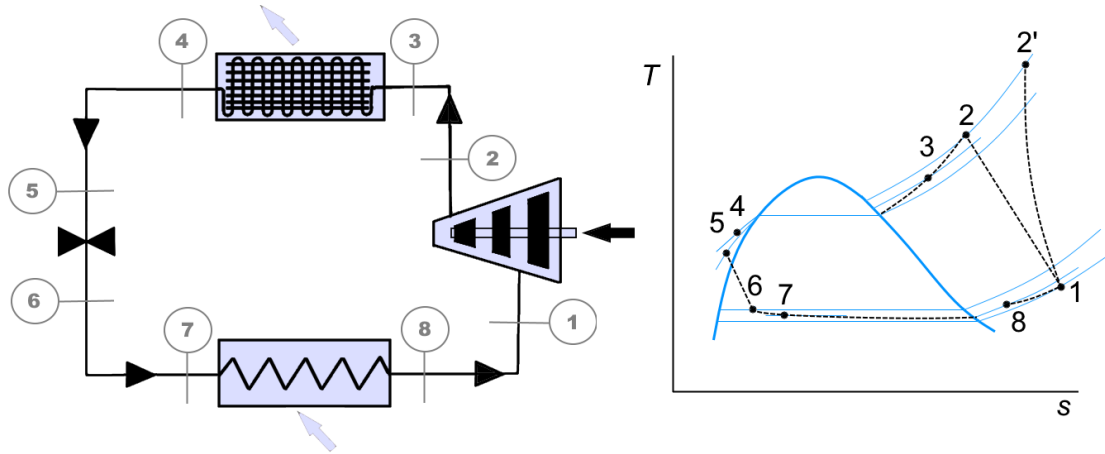


FIGURE H.1: Basic ASHP system schematic with typical T-s diagram.

H.2 Psychrometry and the evaporator of the ASHP

The properties of ambient air are dependant on the dry and wet bulb temperatures along with the atmospheric pressures. For a single installation, the atmospheric pressure can be assumed constant. Therefore the dry and wet bulb temperatures will influence the thermal characteristics of the air and the ease at which the energy is to be extracted. Using Dalton's law of ideal gas mixtures and the concept of partial pressures one, can rewrite the energy equation for a fixed volume of ambient air as:

$$m \times h_{mix} = (m_a \times h_a) + (m_v \times h_v) \quad (H.1)$$

$$\begin{aligned} \models h_a &= f(T = T_{db}, P = P_a) \\ \models h_v &= f(x = 1, P = P_v) \end{aligned}$$

The relative humidity (ϕ) is given as the ratio of the mole fraction of vapour in the mixture to the mole fraction of vapour in a saturated mixture at the same temperature:

$$P_v = P_g \left(\frac{\phi}{100} \right) \quad (\text{H.2})$$

Whereas the various partial pressures are given by:

$$\begin{aligned} P_v &= f(T = T_{db}, x = 1) \\ P_a &= P_{atm} - P_v \end{aligned} \quad (\text{H.3})$$

Which leads to the humidity ratio (ω) defined as the ratio of water mass to vapour mass in the gas-vapour mixture and the consequent equivalent enthalpy of the gas-vapour mixture:

$$\omega = \frac{MW_{water} \times P_v}{MW_{air} \times P_a} \quad (\text{H.4})$$

$$h_{mix} = \frac{(\omega \times h_v) + h_a}{\omega + 1} \quad (\text{H.5})$$

From the above analyses, it can quickly be recognised that the enthalpy of the gas-vapour mixture increases as the dry bulb temperature (T_{db}) and consequently enthalpy of the dry air increases (h_a). With the increase in relative humidity (ϕ), the enthalpy of the gas-vapour mixture at the evaporator of the heat pump increases. Nevertheless, higher enthalpy at the evaporator does not guarantee increased performance as the energy has to be transferred to the evaporator by heat transfer.

Three types of heat transfer conditions can manifest at the evaporator of an ASHP unit based on the ambient dry bulb temperature, relative humidity, atmospheric pressure and evaporator temperature:

- Frosting
- Condensation
- Dry air heat transfer

The influence of these parameters on ASHP performance is of intensive research, and some observations have been made:

It is well established from literature and industry that frosting is not beneficial to ASHP performance, and consequently, lower the COP of ASHP systems (Wu et al., 2020; Li et al., 2017). The frosting creates a significant thermal resistance that inhibits heat transfer to take place to the evaporator—lowering the heat input to the ASHP. These conditions generally correspond with low dry bulb temperatures ($T_{db} < 7[^\circ\text{C}]$) and high relative humidities. Recent studies aim to use methods to reduce frosting at these conditions by using desiccant coated heat exchangers or active dehumidification (Wang et al., 2018; Zhang, Hihara, and Saikawa, 2012). To conclude, from literature, it could be seen that for dry bulb temperatures lower than 7 [°C], higher relative humidities led to reductions in ASHP performance and COP.

Condensate can form on the evaporator coil of a heat-pump once the evaporator temperature is colder than the dew point temperature. From literature, droplet condensation has been established as a heat transfer enhancer with the heat transfer coefficient orders of magnitude larger than film condensation (Bergmain et al., 2013; Eimann et al., 2020). Furthermore, studies have shown that increased relative humidity leads to increased condensate forming on the sub-cooled surface and increase in total heat transfer Danilo, Dominique, and Frédéric, 2016. However, studies have also shown that if the condensate that forms on the evaporator, is not removed with a high enough rate, then the droplets might grow and even form film condensation (El Fil, Kini, and Garimella, 2020). Consequently, creating an increase in thermal resistance and reduction in ASHP performance. To reduce the build-up of condensate is therefore beneficial to HVAC equipment, and a recent study have proposed alterations in the surfaces to enhance condensate removal (Chung, Kadala, and Taylor, 2020). Thus it can be concluded that for some conditions an increase of relative humidity will increase COP, but once condensate build-up is severe, a reduction in heat transfer will occur.

For non-frosting, frosting and condensing regions the following conditions do always hold: Higher dry-bulb (T_{db}) temperatures generally lead to increases in ASHP performance (Underwood, Royapoor, and Sturm, 2017; Wu et al., 2020; Li et al., 2017). Furthermore, decreases in inlet water temperatures decrease the performance of ASHP units (Guo and Goumba, 2018; Vega and Cuevas, 2018).

To conclude, the following findings with regards to the impact of ambient conditions on ASHP performance can be made:

- For frosting conditions, an increase in relative humidity leads to a reduction in the COP of an ASHP (Wu et al., 2020; Li et al., 2017; Wang et al., 2018; Zhang, Hihara, and Saikawa, 2012; Underwood, Royapoor, and Sturm, 2017).
- It has been shown that for non-frosting conditions; an increase in relative humidity increases the rate of dropwise condensation. Dropwise condensation, in turn, increases heat transfer under conditions outlined from literature (Danilo, Dominique, and Frédéric, 2016). It can thus be argued that higher relative humidity will increase ASHP performance under these conditions as relatively higher energy heat (opposed to dry-air) is available at lower thermal resistance (Eimann et al., 2020). However, condensation build-up can degrade heat transfer, and consequently, performance (Chung, Kadala, and Taylor, 2020). Thus, from heat transfer literature, and thermodynamics, one can argue that increased relative humidity will generally increase ASHP COP for ambient conditions where condensation occurs.
- Higher water inlet temperatures degrade COP performance (Guo and Goumba, 2018; Vega and Cuevas, 2018).
- Higher dry bulb temperatures, based on ASHP and heat transfer literature, increase COP performance for frosting, condensing and non-condensing regions (Tabatabaei, Treur, and Waumans, 2016; Underwood, Royapoor, and Sturm, 2017; Guo and Goumba, 2018; Kinab et al., 2010).

Based on the findings, the study resumed deriving a performance map based solely on OEM and experimental data. Deriving an expression that will accurately describe ASHP performance in all of the conditions outlined above falls out of the scope of the study with the data and experimental setup that was available. The following section will describe the statistical analyses on the proposed regression formulas for COP (H.3) and electrical consumption (H.4).

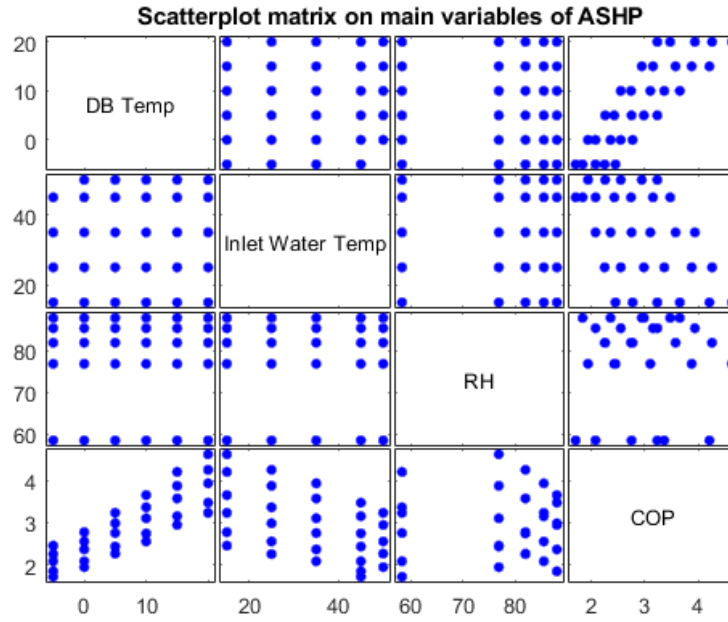


FIGURE H.2: Scatter-plot matrix on ASHP variables from MATLAB®.

H.3 Statistical Analyses on the Coefficient of Performance

From conventional mathematical modelling practises, the primary factors have been identified that influence the behaviour of the process which one is attempting to predict. Secondly, the tentative relationships between the variables have to be established. From the previous sections, the following primary independent variables were identified:

- Inlet water temperature (T_{inlet})
- DB temperature (T_{db})
- Relative humidity (ϕ)

The regression aims to predict the ASHP usable heat energy transferred, electrical consumption and COP (β). The regression analyses of the COP will now be illustrated; the other regressions followed the same method. For simplicity, it is assumed that a linear multi-variate relationship exists between the variables mentioned above and COP:

$$\beta = \alpha_0 + (\alpha_1 \times T_{db}) + (\alpha_2 \times T_{inlet}) + (\alpha_3 \times \phi) + \epsilon \quad (\text{H.6})$$

With ϵ representing the error between the actual and predicted COP (β) value. Using the manufacturer data in Appendix D and the MATLAB function *fitnlm* a regression was conducted on the data with the software estimating the coefficients of $\alpha_0, \alpha_1, \alpha_2$ and α_3 respectively. From Figure H.2, the scatterplot of the various variables against COP was plotted. It can be seen that the dry bulb temperature has a somewhat positive relation to COP. The inlet water temperature has a negative correlation with COP. The relative humidity, however, was not varied enough by the manufacturer to assume its effects on the COP. However, it seems to have a 2nd order effects on the COP. To make an accurate fit, the RH has to be varied over the full extent of the

performance space to avoid extrapolation. This observation was also noted by the studies of Simon and Underwood (Simon et al., 2016; Underwood, Royapoor, and Sturm, 2017).

Consequently, the RH coefficient was tuned using in-field test data. The estimated coefficients are tabulated in Table H.1 with regards to the OEM data and that obtained from MATLAB. The main statistical parameters for the fitness of the regression is tabulated in Table H.2, also regarding the OEM data.

TABLE H.1: Statistical coefficient output of *fitnlm* function on OEM data.

<i>Coefficient</i>	<i>Estimate</i>	<i>SE</i>	<i>tStat</i>	<i>pValue</i>
α_0	3.456e+00	1.435e-01	2.408e+01	2.636e-19
α_1	7.5934e-02	2.088e-03	3.636e+01	7.973e-24
α_2	-3.0275e-02	1.4120e-03	-2.142e+01	4.796e-18
α_3	-7.520e-04	1.689e-03	-4.452e-01	6.598e-01

TABLE H.2: Statistical goodness-of fit output of *fitnlm* function on OEM data.

<i>Term</i>	<i>Symbol</i>	<i>Value</i>
Number of observations	N	30
Root mean squared error	RMS	0.0977
Coefficient of multiple determination	R^2	0.985
Adjusted coefficient of multiple determination	adjusted R^2	0.984
P-value	p	5.81e-24

The P-value (ρ) is small for all of the coefficients except for α_3 , which represents the RH coefficient. In this instance, the null hypothesis (H_0) states that there is no linear relationship between the coefficient and variable pair to that of the COP. It can thus be concluded that the null hypothesis is valid for the last coefficient. Thus extra data or further analyses are required to make use of the linear COP to RH relationship. One can note, however, that there is a significant probability that a test statistic will not be obtained to support the null hypothesis (H_0) on the first three coefficients ($\alpha_0, \alpha_1, \alpha_2$) with regards to the test data (OEM) used.

Figure H.3 plots the residual error to that of the fitted values. These type of plots can aid in revealing patterns in the data and support or contradict the assumptions made on the relations between the independent and dependant variables. From the previously mentioned Figure, it can be noticed that it appears to showcase a U shape with the residuals. The COP was under-predicted near a COP of three. By contrast, COP was over-predicted on the lower and upper extremities of the variable space. The former and latter usually demonstrates non-linear relationships in the data. By increasing the linear relationship to a higher degree relationship, the same plot was obtained. Leading towards the rationale that the RH term might have been responsible. Therefore the model was over-fitted with regards to RH using the manufacturer data. The regression was mitigated with in-field test data and the 1st law energy balance.

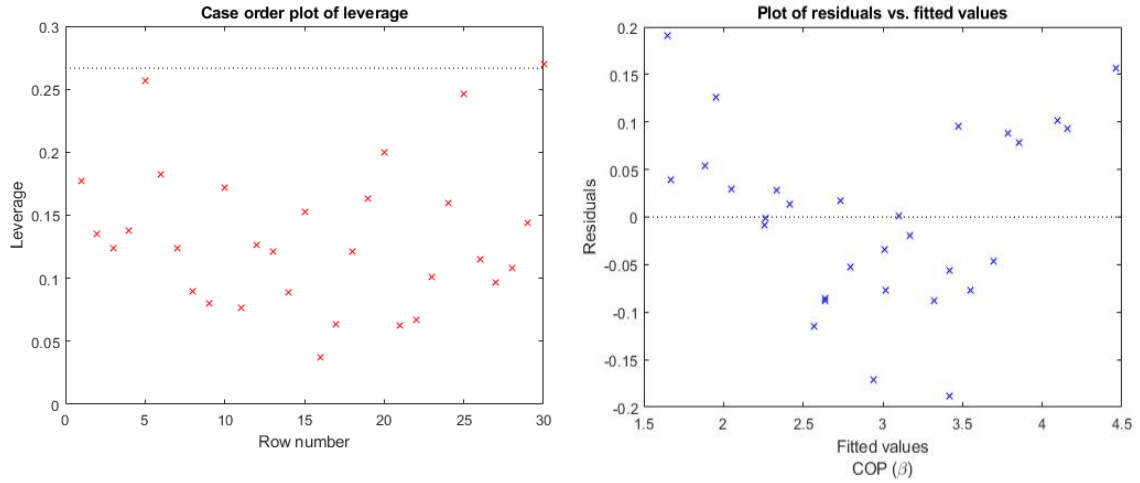


FIGURE H.3: *Left:* The leverage plot for the fitted values of the manufacturer. *Right:* The residual plot for the fitted values.

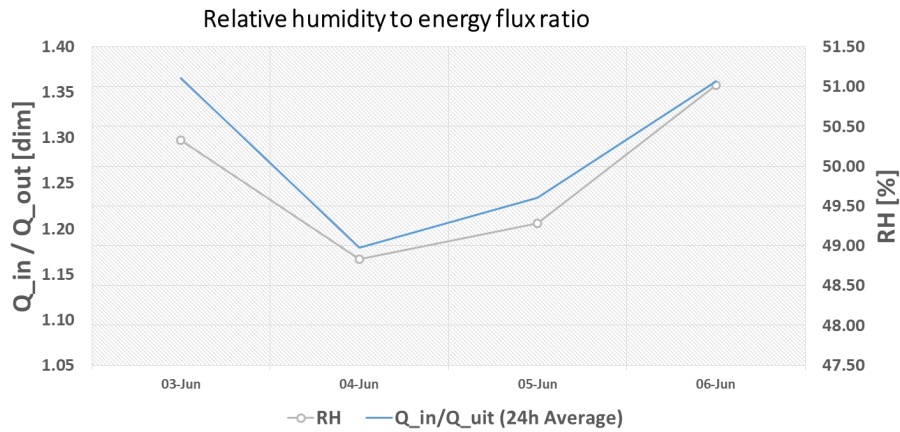


FIGURE H.4: The relative humidity to energy flux relationship prior to taking humidity into account on ASHP regression model.

Figure H.4 illustrates the energy balance when the humidity coefficient to the regression equation has not been added. It was noted that there appears to be a relationship between the RH and the energy flux. The ASHP is by far the broadest energy flux term in the system and apart from this carries nearly 97 [%] of total energy contribution. As the regression equation expresses ASHP performance, a relatively small error can pose a devastating effect on the energy balance. By using in-field test data, the regression equation was used, and the α_3 coefficient was adjusted to minimise the Q_{in}/Q_{out} relationship. Thus the dry and inlet coefficients were obtained from the regression analyses in MATLAB. With the RH coefficient derived from the experimental results. After this final analyses, Figure H.5 was obtained.

Thus, in a nutshell, take the OEM data (Appendix D) and assume a linear multivariate relationship according to Figure H.2:

$$\beta \approx \alpha_0 + (\alpha_1 \times T_{db}) + (\alpha_2 \times T_{inlet}) + (\alpha_3 \times \phi) \quad (H.7)$$

Analyse the data from Tables H.1 and H.2 as well as Figure H.4. Concluding the null hypotheses for (α_3) , the parameter is tuned using in-field data to obtain the correct fit.

$$\begin{aligned} &\text{Guess } \alpha_3 \\ &\models (\beta - \beta_{\text{experiment}}) \rightarrow 0 \\ &\forall i \in E \end{aligned}$$

Where E is the calibration period of the experimental data. After fitting (α_3) against the in-field data, Figure H.5 was obtained opposed to Figure H.4. Showcasing a good fit in all of the variables and therefore concluding the statistical analyses. This equation was consequently used in energy balance and simulations.

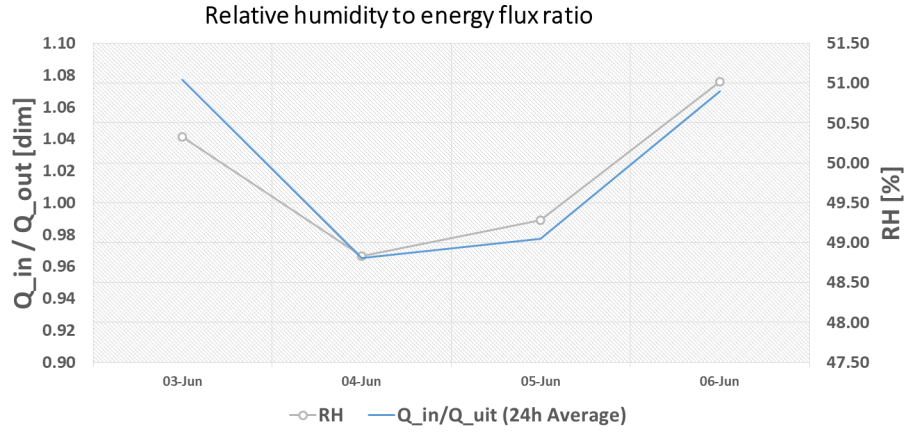


FIGURE H.5: The relative humidity to energy flux relationship after taking humidity into account on regression model.

H.4 Statistical Analyses Electrical Consumption

From the previous section, a regression equation was derived for the prediction of ASHP performance based on the experimental data. However, the simulation programs have no source for electrical consumption. The COP has to be multiplied by the electrical consumption to calculate the available energy. Therefore, a regression equation was devised for the power consumed by the compressor. This enabled the simulation to predict electrical consumption on the ASHP and the COP depending on the environmental conditions.

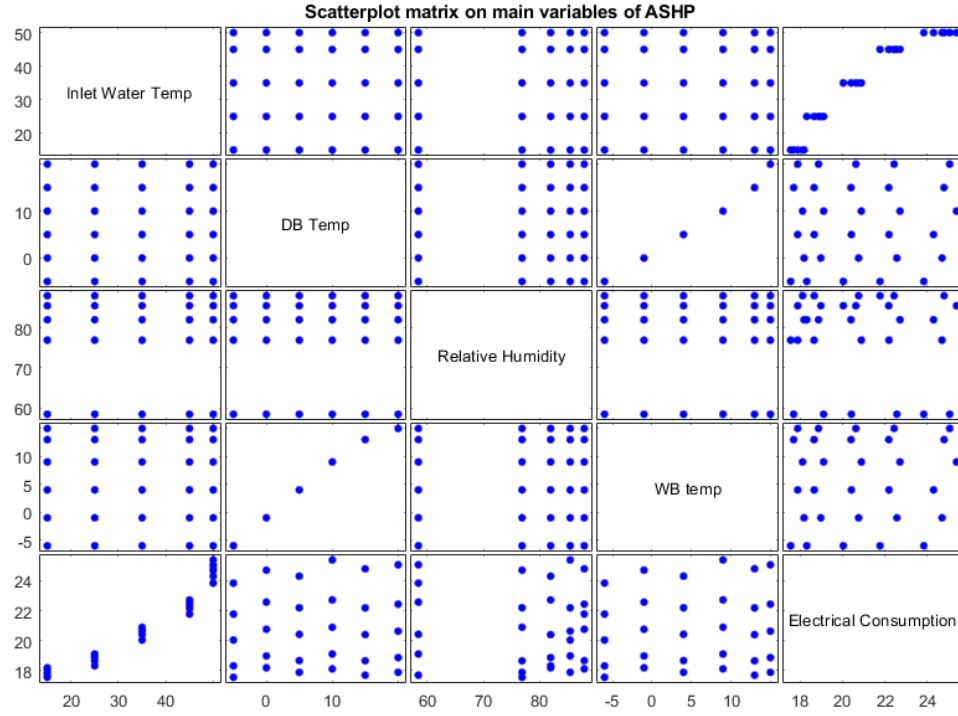


FIGURE H.6: Scatter-plot matrix on ASHP variables from MATLAB.

From Figure H.6, one can note a second-order relation between the electrical consumption and the inlet water temperature of the machine. This relation can be circumscribed since the machine has favourable drivers for heat transfer potential (to the sink) when the inlet water temperature is cold. As the water becomes increasingly warmer, the temperature difference between the set-point and the sink temperature decreases. Decreasing the amount of energy that needs to be transferred. The flow rate on the water circuit is increased to transfer the same amount of energy (within the set-point temperature range). Additionally, the refrigerant mass flow in the ASHP also needs to increase to enable the cycle to operate within the design and set-point parameters. This consequently leads to higher consumption of electricity. From the same Figure (H.6), the relative humidity seems to follow no apparent profile to that of the electrical consumption. The wet bulb and dry bulb temperatures vary the electrical consumption sinusoidally for fixed inlet conditions. From the above, a non-linear multivariate regression equation was proposed in the following format:

$$\dot{P}_{electrical} = b_1 + b_2 \times T_{inlet}^2 + \cos(b_3 + T_{db}) + \cos(b_4 + T_{wb}) \quad (H.8)$$

TABLE H.3: Statistical coefficient output of *fitnlm* function on OEM data.

<i>Coefficient</i>	<i>Estimate</i>	<i>SE</i>	<i>tStat</i>	<i>pValue</i>
b_0	1.713e+01	1.629e-01	1.051e+02	1.020e-35
b_1	2.862e-03	1.032e-04	2.772e+01	7.726e-21
b_2	1.004e+00	1.350e-01	7.440e+00	6.699e-08
b_3	-1.383e+00	1.190e-01	-1.162e+01	8.460e-12

TABLE H.4: Statistical goodness-of fit output of *fitnlm* function on OEM data.

<i>Term</i>	<i>Symbol</i>	<i>Value</i>
Number of observations	N	30
Root mean squared error	RMS	0.479
Coefficient of multiple determination	R^2	0.967
Adjusted coefficient of multiple determination	adjusted R^2	0.964
P-value	p	1.95e-19

Using Matlab® *fitnlm*, the coefficients in Table H.3 was obtained, and the goodness-of-fit is showcased in Table H.4. In this instance, the null hypothesis states that the relationship showcased in the above equations do not hold between the dependant and independent parameters. From Table H.3, it can be noted that the p value remains small for all of the parameters and that the null hypothesis is invalid for all the fitted coefficients. Table H.4 illustrates that a reasonable good coefficient of multiple determination was obtained given the number of observations.

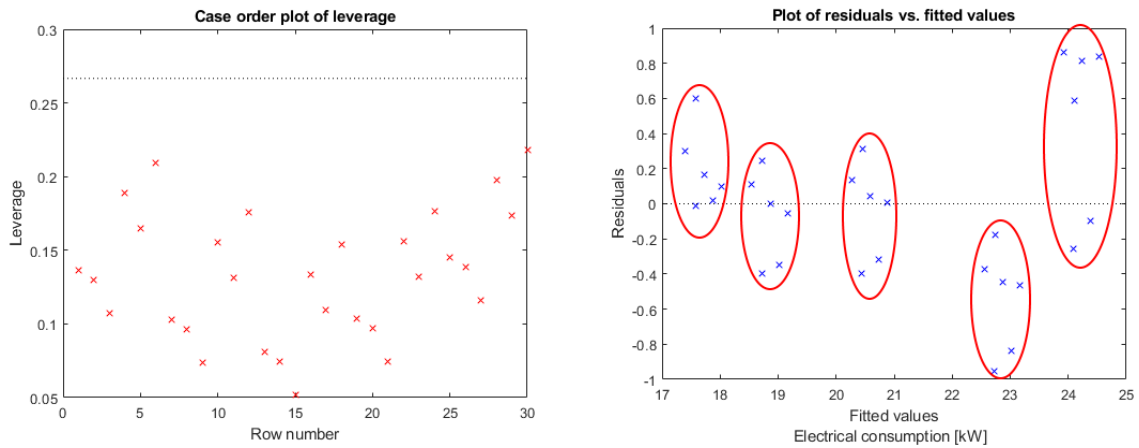


FIGURE H.7: *Left:* The leverage plot for the fitted values of the manufacturer.
Right: The residual plot for the fitted values.

From the leverage plot in Figure H.7; it can be noted that none of the data points has a high influence potential. The data set is thus adequately influential. With no outliers in the data set the model could be fitted robustly. From the residual plot in the same Figure, a slight pattern can be recognised. In the regression model showcases a higher residual near full electrical load on the ASHP. The pattern is a six-point data set groupings (17.5, 19, 20.5, 23, 24.5[kW]) that corresponds with the five inlet temperature conditions (15, 25, 35, 45, 50[°C]). It can be seen that

the residual spread increases as electrical consumption increases. With the model having a absolute residual spread of 0.6 [kW] up to 24 [kW] consumption and 1 [kW] above 24 [kW]. This could indicate that the machine is more sensitive then what the model predicts to dry and wet bulb temperature variations at the warmest inlet temperatures. Furthermore, from the OEM data and the top right corner of Figure H.6, it can be noted that at the higher inlet temperatures, the ASHP electrical consumption is more sensitive to variations in dry and wet bulb temperatures. The variation could be explained by the fact that the machine has to apply increased flow rates to ensure that the target outlet water temperature is attained. Consequently, higher electrical consumption is required by the compressor. As the purpose of the study is to evaluate system performance and not the analyses of sub-components within the heat pump, the regression equation was deemed adequate for use in the developed simulations.

Appendix I

MATLAB Simulink simulations

This appendix will showcase the MATLAB simulation models that were used for verification against the developed EES® simulations. During the study, various simulations were conducted in MATLAB ®Simulink ®to conduct the verification studies for the comparisons to the developed EES models. These models used the following libraries from Simscape ® and Simulink ® to simulate the ASHP/SWH environment:

- Simscape® Thermal Liquid Library
- Simscape® Thermal Element Library
- Simulink® Block Library

The boundary conditions that were used in the three case studies are reported in Table I.1. The following sections will discuss, by method of various screenshots; the Simulink models that were developed. The MATLAB Simulink model for the first and second case study uses the same architecture with the only difference that the heat losses have been activated for the second and third case study as per the boundary specifications in Table I.1.

TABLE I.1: Test case conditions for the validation simulations

<i>Description</i>	<i>Parameter</i>	<i>Case 1</i>	<i>Case 2</i>	<i>Case 3</i>	<i>Unit</i>
System mass in	$\dot{m}_{system;n}$	0.3	1	1	$[\text{kg s}^{-1}]$
System temperature in	$T_{system;n}$	20	20	20	$[\text{°C}]$
Ambient dry bulb temperature	T_{db}	11.91	11.91	11.91	$[\text{°C}]$
Ambient wet bulb temperature	T_{wb}	0.33	0.33	0.33	$[\text{°C}]$
Relative humidity	RH	49.41	49.41	49.41	$[\%]$
Initial system temperature	$T_{i,system}$	11.91	11.91	11.91	$[\text{°C}]$
Initial tank temperatures	$T_{i,tank}1..5$	11.91	11.91	11.91	$[\text{°C}]$
Irradiance	G	0	0	850	$[\text{W/m}^2]$
Heat loss active	UA_{system}	0	1	1	$[\text{dim}]$
ASHP active	$ASHP_{active}$	0	1	1	$[\text{dim}]$
SWH active	SWH_{active}	0	1	1	$[\text{dim}]$

I.1 Case study one and two

From Figure I.1 the different subsystem of the MATLAB® Simulink® program has been marked and will be discussed in the following paragraphs. The paragraphs were divided as follows:

- A - Municipal inlet sub-component (I.1.1)
- B - Stratified storage vessels (I.1.2)
- C - Ring-main circuit (I.1.3)
- D - System outlet (I.1.4)

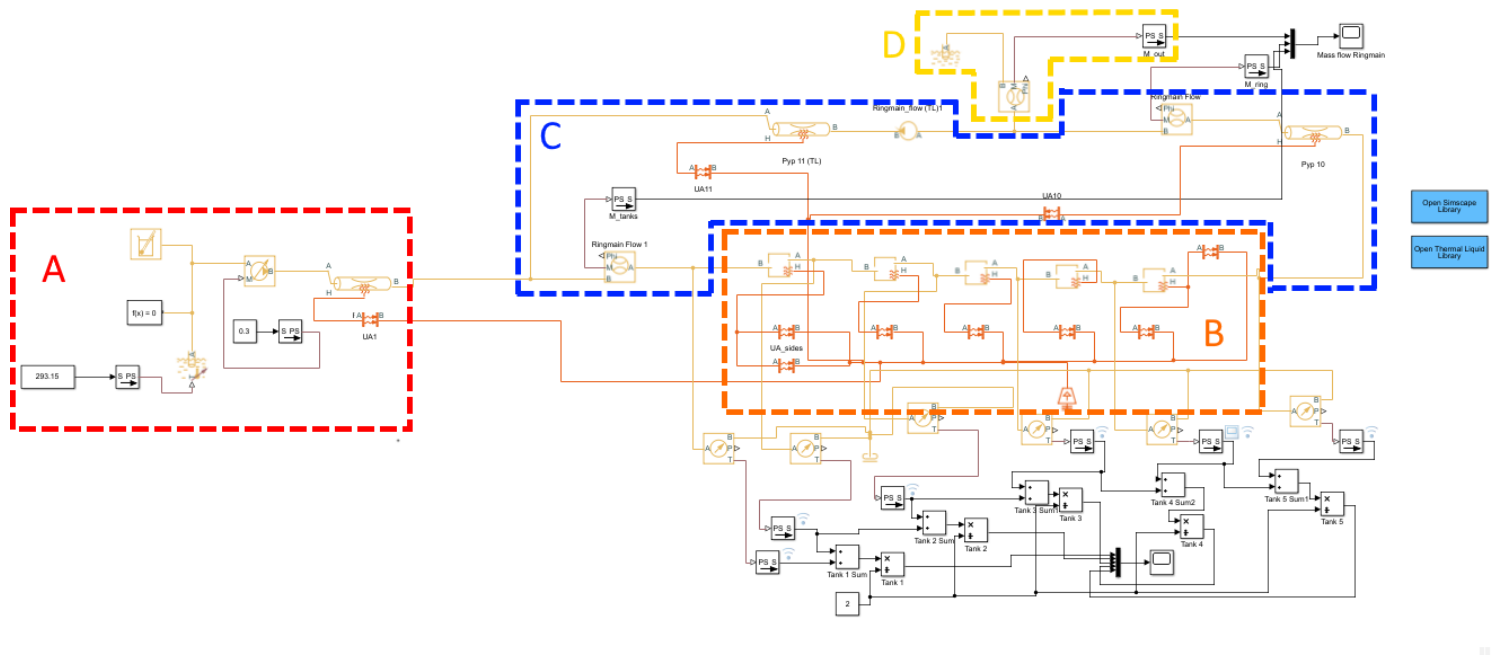


FIGURE I.1: MATLAB Simulink verification model.

I.1.1 Municipal inlet sub-component

The municipal supply subsystem (Figure I.2) consisted of the *Controlled reservoir* thermal liquid block with the inlet temperature specified as reported in Table I.1. The mass flow was enforced by the *Controlled Mass Flow Rate Source* (TL) component and resembled the extracted mass flow through the system. The mass flow component enforces no pressure differential between the inlet and outlet of the component and the fluid is not compressed. The *Thermal Liquid Settings* (TL) block used 2-D pressure and temperature data to simulate the properties of liquid water throughout the network.

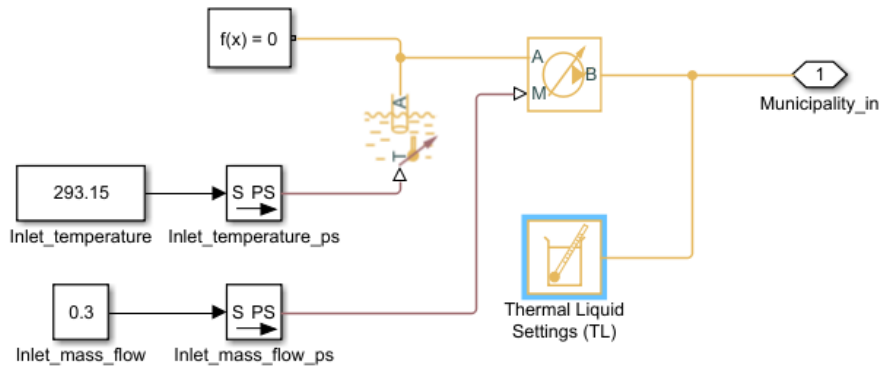


FIGURE I.2: MATLAB® Simulink® municipal inlet subsystem model.

I.1.2 Stratified storage vessel

Each tank in the stratified storage vessel arrangement (Figure I.3) was simulated by using the *2-Port Constant Volume Chamber* (TL) that assumes negligible flow resistance and the accumulation of energy and mass. The heat loss on these components was simulated using the *Convective Heat Transfer* block. For the first case study, the heat transfer coefficient was taken as $h < 0.0001$ to establish adiabatic conditions. For the second case study, the true heat transfer values were inputted. For the heat transfer network, a *Temperature Source* component was used that resembles the thermal sink to the system, and ambient temperatures were inputted for the component.

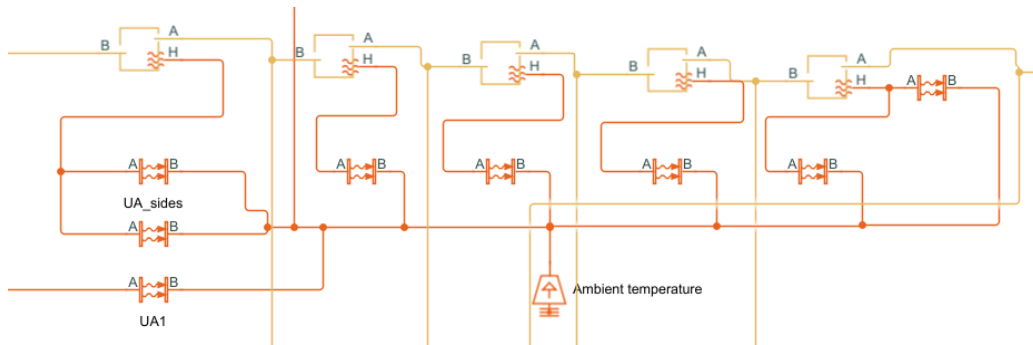


FIGURE I.3: MATLAB® Simulink® municipal inlet subsystem model.

I.1.3 Ring-main circuit

For the ring-main circuit (Figure I.4), the mass flow on the return was enforced by the *Mass Flow Rate Source* (TL) component. For this component, no compression takes place and there is no pressured difference between the inlet and outlet of the component. The piping trough the system is simulated by the *Pipe* (TL) component. The *Pipe* (TL) component, takes the resistance of the tubes into account; fluid compressibility and the heat losses to the environment via the *Convective Heat Transfer* introduced in the previous paragraph. Mass flow measurements at the supply of the ring-main and inlet of the coldest storage vessel were measured by the *Controlled Mass Flow Rate Source* (TL) component.

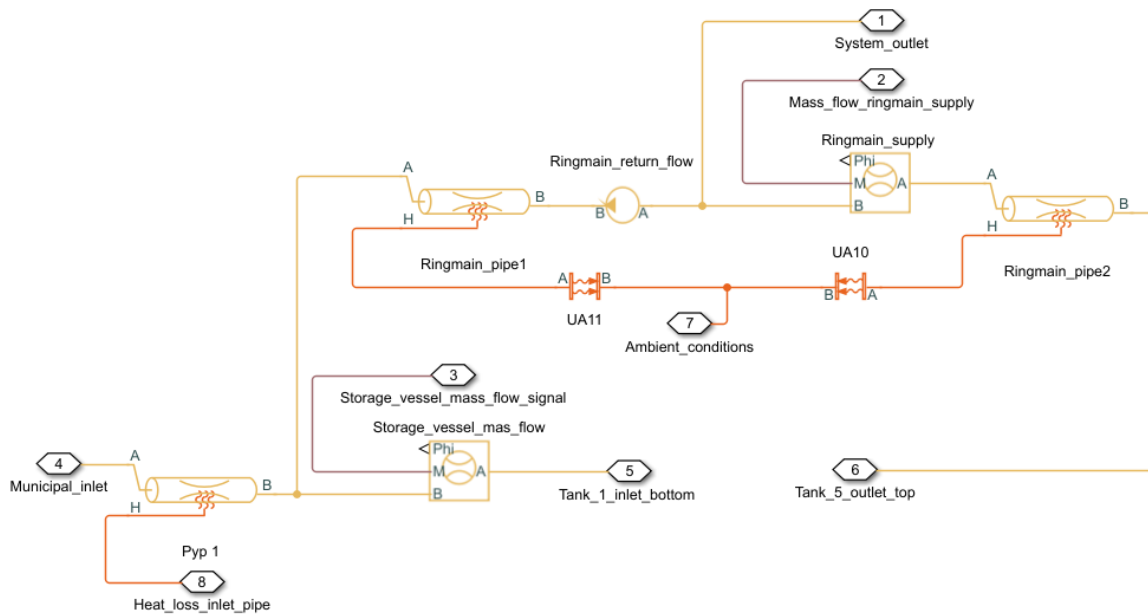


FIGURE I.4: MATLAB® Simulink® ring-main subsystem model.

I.1.4 System outlet

The outlet the various wards (Figure I.5) was simulated using the *Reservoir* (TL) block that maintains the sink at a fixed temperature and pressure. As the mass flow was always into the thermal reservoir, the temperature that was used as input had no effect on the system. The mass flow leaving the system was measured by the *Mass & Energy Flow Rate Sensor* (TL) and used for analyses and debugging.

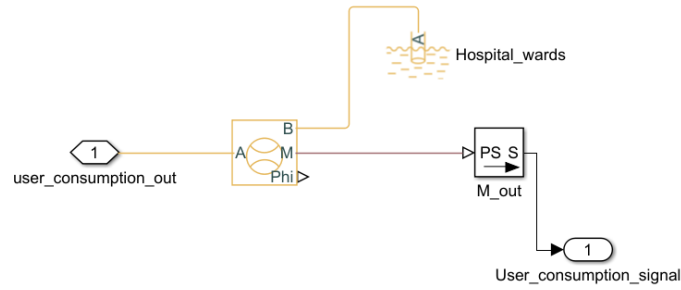


FIGURE I.5: MATLAB® Simulink® outlet subsystem model.

I.2 Case study three

From Figure I.6, the different sections of the MATLAB® Simulink® program have been marked and will be discussed sequentially in the following paragraphs. For the third case study, the SWH and ASHP have been added. All of the other subsystems outlined in Section I.1 remain the same for this simulation apart from the inputs to some of the components. Consequently, these components will not be reintroduced in these sections. The paragraphs were divided as follows:

- A - Solar water heater (I.2.1)
- B - Air source heat pump (I.2.2)

I.2.1 Solar water heater

The SWH subsystem model I.7 used the *2-Port Constant Volume Chamber (TL)* block for the SWH manifold along with an inlet and outlet pipe for the SWH simulated by making use of the *Pipe (TL)* component. Heat losses on these components were taken into account by using the *Convective Heat Transfer* component to a constant ambient temperature source *Temperature Source*. With the physical ports and components defined; the simulation requires two control logic network blocks to control the following variables:

- SWH energy - The amount of heat energy that needs to be added to the *2-Port Constant Volume Chamber (TL)* component have to be determined from the measured ambient conditions.
- SWH mass flow - The mass flow through the SWH have to be determined based on the set-point temperatures and the internal temperature of the SHW.

For the heat addition of solar energy to the SWH manifold, the *Controlled Heat Flow Rate Source* controllable heat flux component was used to add energy. The heat energy, in turn, was driven by various Simulink® blocks that make use of the SWH efficiency curve, as presented in Chapter 3.5.2. The *fcn* block was used to contain the SWH efficiency curve. The curve, as one might recall, requires solar irradiance, SWH inlet temperature and the ambient temperature to predict performance. Therefore, the *Pressure & Temperature Sensor (TL)* components were used to obtain the inlet temperature of the SWH subsystem.

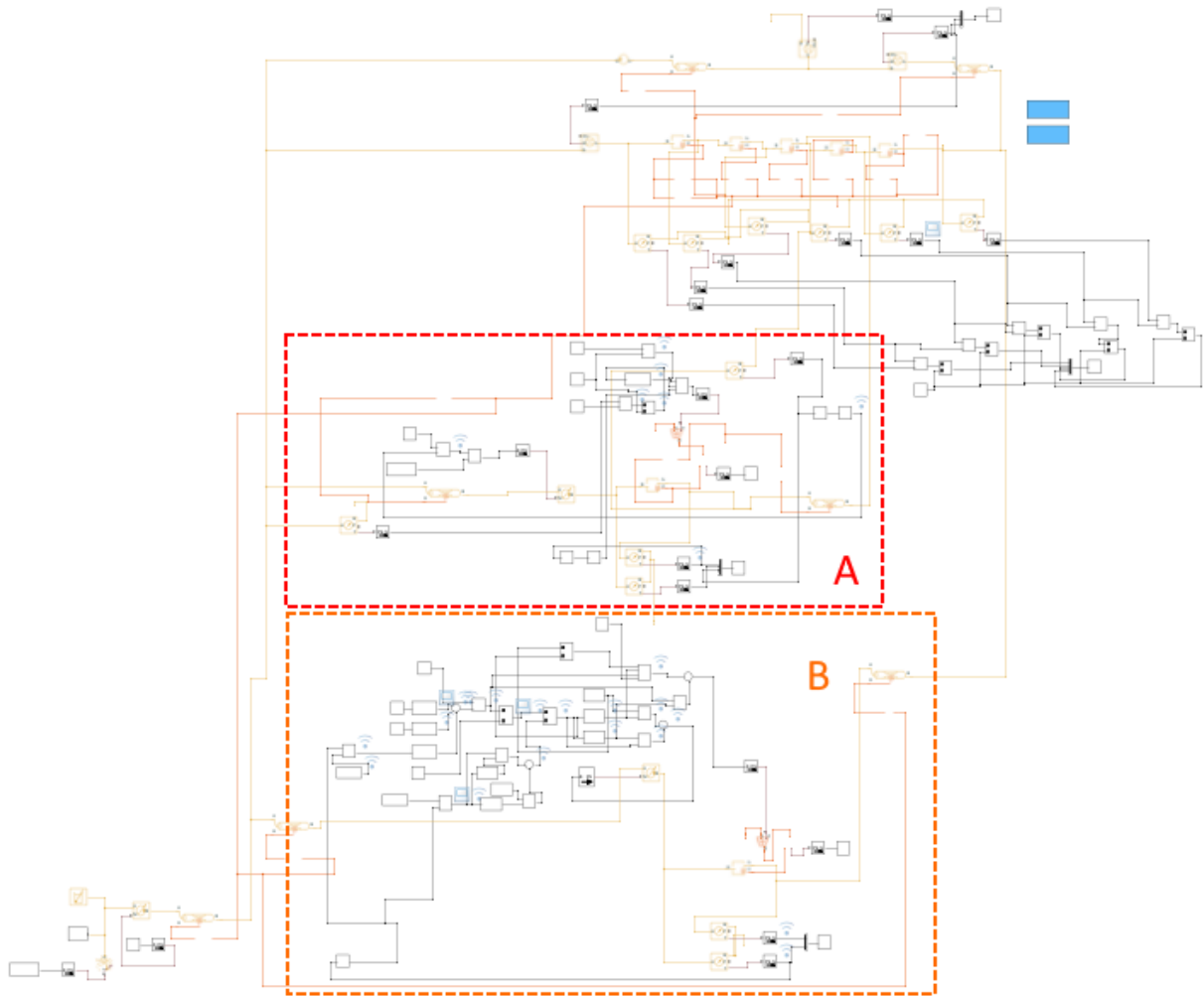


FIGURE I.6: MATLAB Simulink verification model - third case study.

For the control logic to determine the SWH mass flow; temperature measurements were required along with the *Memory*, previous time step block. The simulation compares the set-point temperature to the previous time-step temperature or initial condition to determine if the fixed mass flow has to be enforced through the SWH or not. Consequently, the storage vessel temperature of the middle storage tank was fed into the SWH control logic. If this temperature was higher than the set-point; then the boolean gate would multiply this value with the constant SWH mass flow and feed the output to the *Controlled Mass Flow Rate Source (TL)* mass flow controller. Consequently, the methodology outlined above determined the mass flow that is forced through the SWH manifold.

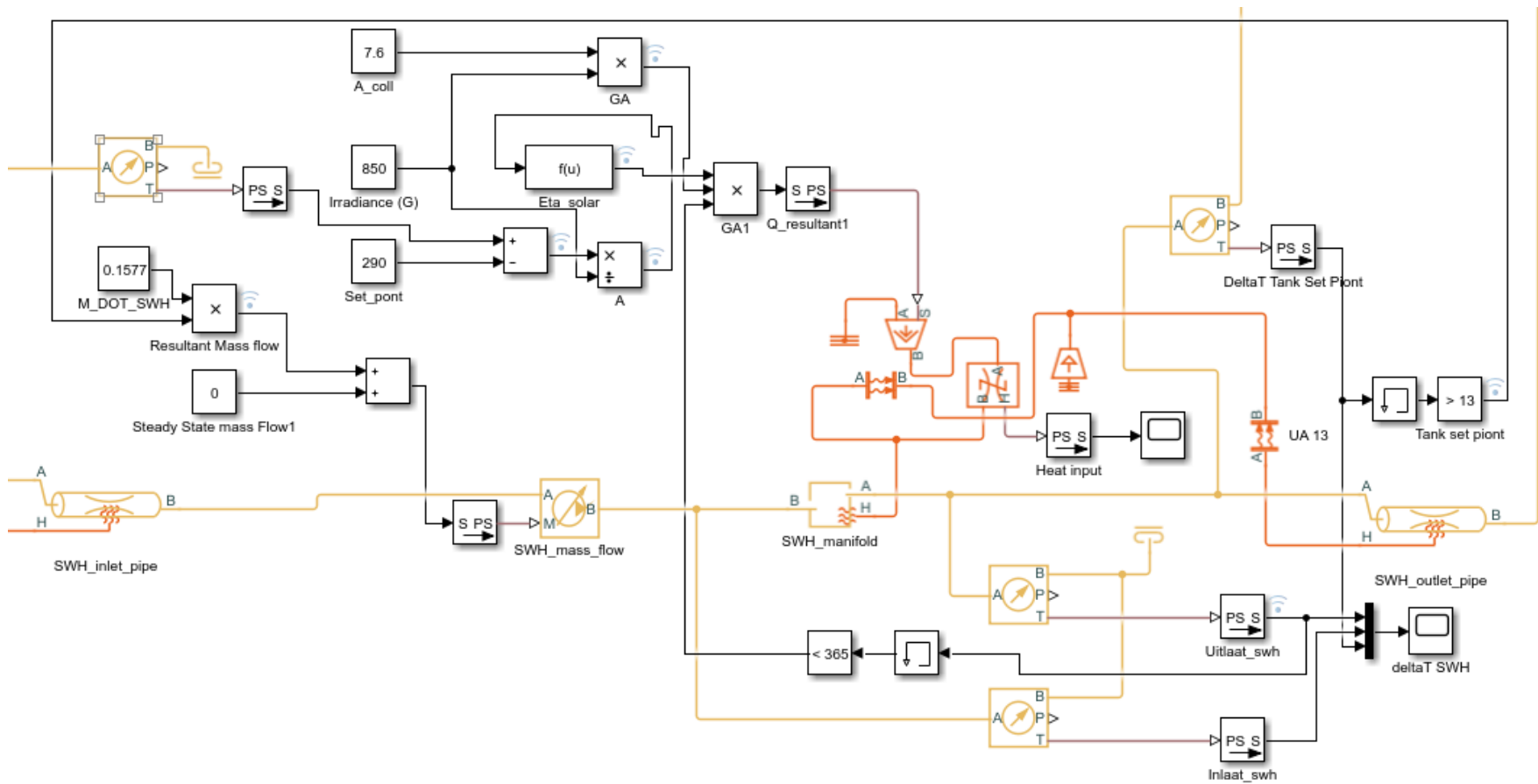


FIGURE I.7: MATLAB® Simulink® solar water heater subsystem model.

I.2.2 Air source heat pump

The waterside of the ASHP evaporator (Figure I.8) was modelled by a *2-Port Constant Volume Chamber (TL)* component with the heat transfer components the same as that of the SWH, as outlined in the previous section. For the ASHP component, there are two types of variables that need to be controlled by Simulink® logic gates:

- Heat energy - The amount of heat energy transferred to the waterside of the ASHP evaporator (*2-Port Constant Volume Chamber (TL)*) have to be calculated according to the environmental conditions and the control logic.
- Mass flow - The mass flow through the ASHP have to be determined and enforced according to the controller of the AHSP.

The maximum available heat energy that can be extracted from the environment is calculated according to the regression equation presented in Appendix H and Chapter 3.5.1. The equation is housed in the user-defined function block (*Fcn*). Consequently, the temperature of the previous time-step at the inlet of the ASHP is obtained from a *Memory* block. Control gates determine from the inlet temperature, and maximum available energy; the mass flow that is required to obtain the set-point temperature (as discussed in Chapter 4.1.2.1).

The calculated mass flow is compared to the maximum OEM specified mass flow. If the mass flow is smaller than the maximum mass flow; then the maximum available heat energy is transferred to the control volume. If the calculated mass flow is higher than the OEM specified mass flow; then only a fraction of the maximum heat energy is transferred.

The calculated mass flow is enforced by a *Controlled Heat Flow Rate Source* component from a fixed reference temperature component *Temperature Source*. Accordingly, the mass flow through the ASHP is enforced by the *Controlled Mass Flow Rate Source (TL)* component.

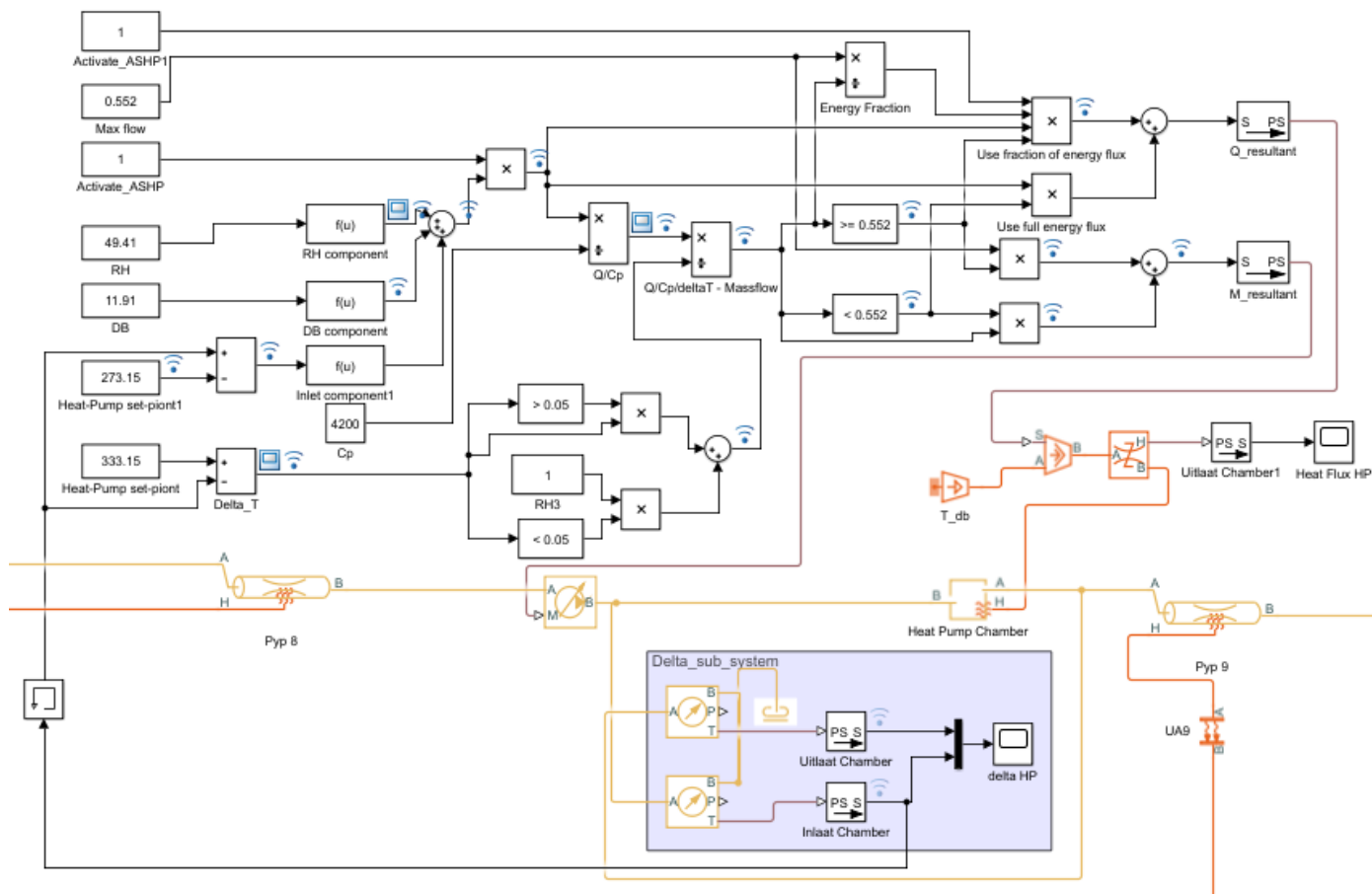


FIGURE I.8: MATLAB® Simulink® air source heat pump subsystem model.

Appendix J

Pump information and curves

The Appendix lists the applicable information regarding the water circulation pump (CCG 08-25c) used in the SWH circuit. As recalled from Chapter 3.5.2.2, the pump curves for only the high-speed setting was supplied by the manufacturer. By using the pump similarity laws, the low and medium speed settings were obtained along with the power curves.

The SCADA monitored electrical voltage on the second phase of the ASHP. The SWH pump's electrical connection was on the same electrical phase of the ASHP. An electrical current measurement was taken while the pump was operational to determine the typical electrical current that the pump uses. Consequently, the CT and HOBO logger was only used to determine if the pump was active or not (Chapter A.3). The measured voltage, steady-state current, and an assumed power factor were used to calculate the electrical power used by the circulation pump. Finally, the flow through the pump could be determined by using the power curve derived from the similarity laws. The SWH pump information was as follows:

- OEM supplied table with a singular operating point on the three-speed settings (Table J.1).
- Picture pertaining to the installed circulation pump (Figure J.1)
- Pump curve with head to volumetric flow (Figure J.2)
- Electrical consumption curve with electrical consumption to volumetric flow (Figure J.3)

TABLE J.1: The SWH (CCG 08-25c) circulating pump information.

<i>Speed</i>	<i>Power</i> [W]	<i>Amps</i> [A]	<i>Max head</i> [m]	<i>Max flow</i> [m^3/h]
High	165	0.61	8	4.8
Medium	115	0.42	7	2.9
Low	75	0.3	5	1.5



FIGURE J.1: Solar water heater circulating pump

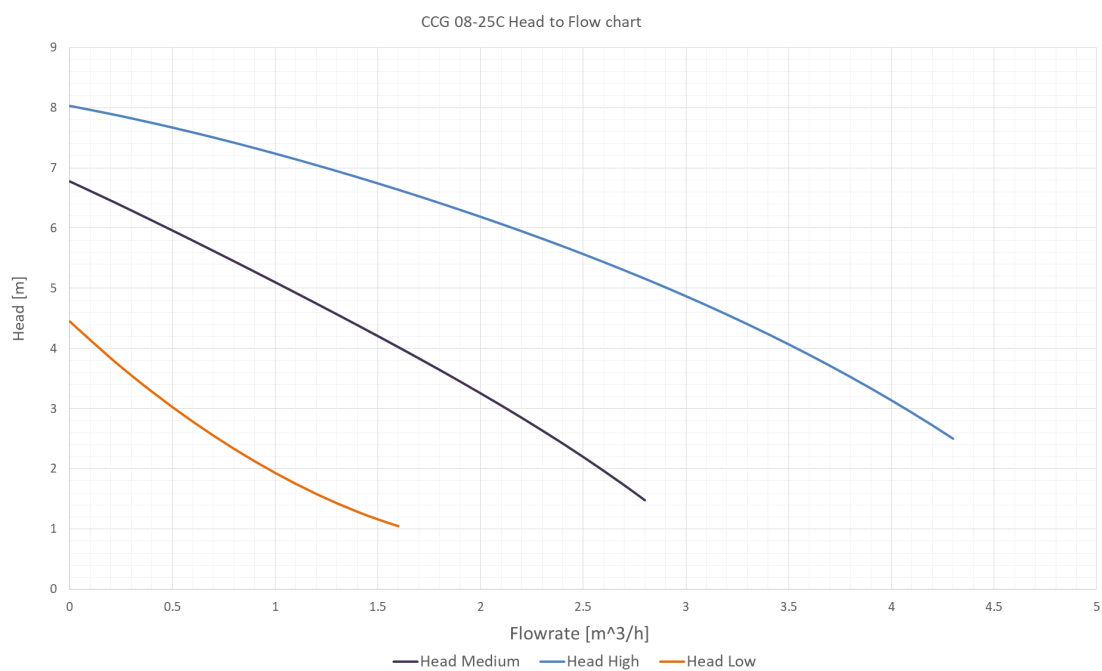


FIGURE J.2: OEM solar pump head to flow curves.

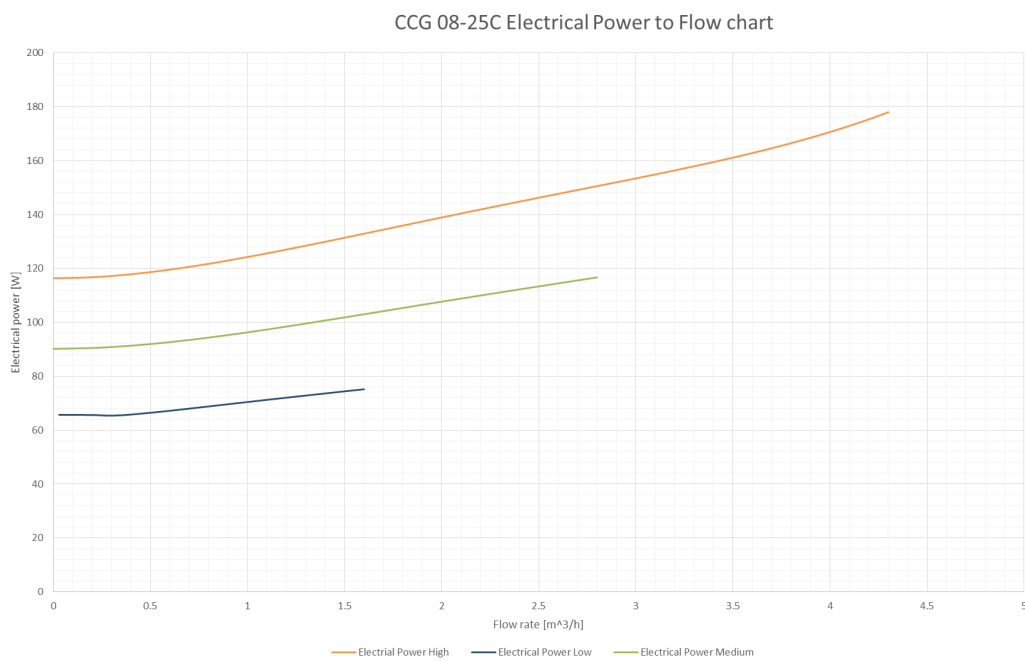


FIGURE J.3: Derived electrical power to flow curves.

Appendix K

Computer code (EES): Detailed system simulation

```
{THSE – Thermal Hydraulic Simulation Enviroment  
02/2020 by Jean-Pierre de Villiers  
24875252  
-----}  
  
Module Pyp_komponent(teller ,m_poort1 ,T_poort1 ,T_poort2 ,m_element ,T_internal ,deltaT ,f ,D_i ,D_o ,L ,UA ,T_db ,Q_pomp ,Head_pomp ,m_out ,  
T_out , T_internal_new ,deltaP ,T_poort_gebruik ,Q_losses ,Q_in_pyp ,Q_toets_in , Q_toets_uit)  
m_out = m_poort1  
  
vloei_area = (pi# / 4 )* D_i  
buite_area = (pi#)* D_o*L  
  
rho = Density(Water,T=T_internal ,P=80 [kPa])  
Cp = Cp(Water,T=T_internal ,P=80 [kPa])  
  
Q = m_out/rho  
  
delta_P = f*(-L/D_i)*(rho/2)*(((Q/(((D_i/2)^2)*pi#)))^2){*maalFaktor}  
deltaP = delta_P + (Head_pomp*rho*g#)  
  
T_internal = T_out  
T_poort_gebruik = toetsVloei(m_poort1 ,T_poort1 ,T_poort2)  
maalFaktor = If(m_poort1,0,-1,1,1)  
  
UA_gebruik = if(T_internal ,T_db,0,0,UA)  
Q_heat_loss = UA_gebruik*(T_db-T_internal)  
Q_hydraulic_loss = ((delta_P/rho)*m_out)  
Q_vloei = ((m_poort1*cp*T_poort_gebruik) - (m_out*cp*T_internal))*maalFaktor  
safe = 1 {steadyStateTester(T_internal ,T_internal_new ,0.001)}  
Q_losses = Q_heat_loss + Q_hydraulic_loss  
Q_in_pyp = Q_pomp  
  
Q_thermal_mass = (m_element*Cp*(T_internal_new - T_internal )/deltaT)  
  
Q_toets_in = (m_poort1*cp*T_poort_gebruik) + Q_pomp  
Q_toets_uit = (m_out*cp*T_internal) + Q_heat_loss + Q_hydraulic_loss  
  
Q_thermal_mass = Q_vloei + Q_heat_loss + Q_hydraulic_loss + Q_pomp  
End  
  
Module SWH_komponent(teller ,m_poort1 ,T_poort1 ,m_element ,T_internal ,deltaT ,f ,D_i ,D_o ,L ,UA ,Area_collector ,T_db ,Q_pomp ,Head_pomp ,  
m_out ,T_out , T_internal_new ,deltaP ,Q_solar ,G ,Q_heat_loss ,Q_vloei ,activateLoop_SWH ,Q_in_solar ,Q_uit_solar ,Q_in_swh ,  
Q_uit_swh)  
m_out = m_poort1  
  
m_poort1 = if(activateLoop_SWH,0,0.0000001,0.0000001,0.1577)  
  
vloei_area = (pi# / 4 )* D_i  
buite_area = (pi#)* D_o*L  
  
rho = Density(Water,T=T_internal ,P=80 [kPa])  
Cp = Cp(Water,T=T_internal ,P=80 [kPa])  
h_inlaat = Enthalpy(Water,T=T_poort1 ,P=80 [kPa])  
  
Q = m_out/rho  
  
delta_P = f*(-L/D_i)*(rho/2)*(((Q/(((D_i/2)^2)*pi#)))^2)  
deltaP = delta_P + (Head_pomp*rho*g#)  
  
T_internal = T_out  
UA_gebruik = if(T_internal ,T_db,0,0,UA)  
eta_solar = eta_swh(T_db,G,h_inlaat,80)  
eta_solar_gebruik = if(eta_solar ,0,0,0,eta_solar)
```

Appendix K. Computer code (EES): Detailed system simulation

```
Q_heat_loss = UA_gebruik*(T_db-T_internal)
Q_hydraulic_loss = ((delta_P/rho)*m_out)
Q_vloei = (m_poort1*cp*(T_poort1-T_internal))
Q_thermal_mass = (m_element*cp*(T_internal_new - T_internal )/deltaT)
Q_solar = eta_solar_gebruik*G*Area_collector

(m_element*cp*(T_internal_new - T_internal )/deltaT) = Q_vloei + Q_heat_loss + Q_hydraulic_loss + Q_pomp + Q_solar

Q_in_solar = Q_pomp + Q_solar + (m_poort1*cp*(T_poort1))
Q_uit_solar = Q_heat_loss + Q_hydraulic_loss + (m_poort1*cp*(T_internal))

Q_in_swh = Q_pomp + Q_solar
Q_uit_swh = Q_heat_loss + Q_hydraulic_loss

End

Module ASHP_komponent( teller , m_poort1 , T_poort1 , m_element , T_internal , deltaT , D_i , D_o , L , UA , T_db , RH , Q_pomp , Head_pomp , m_out , T_out ,
    T_internal_new , deltaP_total , T_set_point , f , activateLoop_ASHP , Q_hp , Q_hp2 , Q_hp_losses , T_wb , Q_hp3 , Q_in_ashp , Q_uit_ashp ,
    Q_ashp_pomp )
m_poort1 = m_out

vloei_area = (pi# / 4 ) * D_i
buite_area = (pi#) * D_o * L

polisie = nullTester(m_poort1)
polisie2 = nullTester(Q_besikbaar)

rho = Density(Water , T=T_internal , P=80 [kPa])
Cp = Cp(Water , T=T_internal , P=80 [kPa])

Q = m_out / rho

{ delta_P = f*(-L/D_i)*(rho/2)*(((Q/(((D_i/2)^2)*pi#)))^2)*polisie*activateLoop_ASHP
  delta_P_total = delta_P + (Head_pomp*rho*g#)*polisie*activateLoop_ASHP}
{ delta_P = f*(-L/D_i)*(rho/2)*(((Q/(((D_i/2)^2)*pi#)))^2)*activateLoop_ASHP
  delta_P_total = (delta_P + (((Head_pomp*rho*g#))))*activateLoop_ASHP}

delta_P_resistance = f*(-L/D_i)*(rho/2)*(((Q/(((D_i/2)^2)*pi#)))^2)
delta_head_pomp = (((Head_pomp*rho*g#)))
{RK = activateLoop_ASHP*(delta_P_resistance + delta_head_pomp)}
deltaP_total = delta_head_pomp + (delta_P_resistance)

T_internal = T_out
{Q_besikbaar_toets = Q_ECDH90MT_nuwe(T_db , RH , T_internal)*1000[w/kw]"Kyk wat is die beskikbare energie wat vannuit die lug
  getrek kan word"}
Q_besikbaar_toets = Q_ECDH90MT_nuwe(T_db , RH , (T_internal+T_poort1)/2 , T_wb)*1000[w/kw]"Kyk wat is die beskikbare energie wat
  vannuit die lug getrek kan word"

hp_mass_flow_limit = 0.522 [kg/s]
m_out = ashpThrottle(hp_mass_flow_limit , Q_besikbaar_toets , T_internal , T_set_point , Cp , activateLoop_ASHP)
Q_besikbaar = ashpHeatThrottle(T_internal , T_set_point , T_set_point , Q_besikbaar_toets , hp_mass_flow_limit , activateLoop_ASHP) "
  Kyk of die delta T baie klein begin raak dan sit die hitte af maar hou die vloei aan"

UA_gebruik = if(T_internal , T_db , 0 , 0 , UA)
Q_heat_loss = UA_gebruik*(T_db-T_internal)
Q_hydraulic_loss = ((delta_P_resistance/rho)*m_out)
Q_vloei = ((m_poort1*cp*T_poort1) - (m_out*cp*T_internal))
Q_thermal_mass = (m_element*cp*(T_internal_new - T_internal )/deltaT)
Q_hp = -Q_vloei
Q_hp2 = Q_besikbaar
Q_hp3 = Q_besikbaar_toets*activateLoop_ASHP

Q_ashp_pomp = Q_pomp

{ (m_element*cp*(T_internal_new - T_internal )/deltaT) = Q_vloei + Q_heat_loss + Q_hydraulic_loss + (Q_pomp*activateLoop_ASHP)
  + Q_besikbaar}
Q_hp_losses = Q_heat_loss + Q_hydraulic_loss + Q_pomp
(m_element*cp*(T_internal_new - T_internal )/deltaT) = Q_vloei + Q_heat_loss + Q_hydraulic_loss + (Q_pomp) + Q_besikbaar
{Q_new = -Q_vloei - Q_heat_loss - Q_hydraulic_loss - Q_pomp}

Q_in_ashp = Q_pomp + Q_besikbaar + (m_poort1*cp*T_poort1)
Q_uit_ashp = Q_hydraulic_loss + Q_heat_loss - (m_out*cp*T_internal)

End

Module T_stuk_komponent( teller , m_poort1 , m_poort2 , m_poort3 , T_poort1 , T_poort2 , T_poort3 , m_element , T_internal , deltaT , UA , T_db ,
    T_internal_new , Q_losses )

0 = m_poort1 + m_poort2 + m_poort3

rho = Density(Water , T=T_internal , P=80 [kPa])
Cp = Cp(Water , T=T_internal , P=80 [kPa])

UA_gebruik = if(T_internal , T_db , 0 , 0 , UA)

Q_heat_loss = UA_gebruik*(T_db-T_internal)
{Q_hydraulic_loss = ((delta_P_resistance/rho)*m_out)
T_gebruik_3 = if(m_poort3 , 0 , T_internal , 0 , T_poort3)
T_gebruik_2 = if(m_poort2 , 0 , T_internal , 0 , T_poort2)
T_gebruik_1 = if(m_poort1 , 0 , T_internal , 0 , T_poort1)
Q_vloei = ((m_poort3*cp*T_gebruik_3) + (m_poort2*cp*T_gebruik_2) + (m_poort1*cp*T_gebruik_1))
```

Appendix K. Computer code (EES): Detailed system simulation

```
throttle = if(abs(Q_vloei),0.0001, 0,0,1)
Q_thermal_mass = (m_element*Cp*(T_internal_new - T_internal )/deltaT)

Q_losses = Q_heat_loss

(m_element*Cp*(T_internal_new - T_internal )/deltaT) = (Q_vloei*throttle) + Q_heat_loss
End

module Pomp_komponent(m_poort1 ,T_poort1 ,a1 ,a2 ,a3 ,Q_hydraulic_gain ,Head)
m_out = m_poort1
Head = (a1*Q^2) + (a2*Q^1) + a3 [m]
Q = m_poort1/rho*3600
rho = Density(Water,T=T_poort1 ,P=80 [kPa])
Q_hydraulic_gain = ((rho *Head*g#)/rho)*m_poort1
End

Function Q_ECDH90MT_nuwe(T_db,RH,T_inlaat ,T_wb)
Q_ECDH90MT_nuwe = (3.45693852134674 + ((0.075939057244308 )*T_db) + ((-0.0302670035556761 )*T_inlaat)+((-0.0171650168580766*RH)
)*(1.73939260105266e+01) + ((2.52005071301143e-03)*T_inlaat^2)+ (cos(T_wb+(-1.30006073814026e+00)))+ (cos(T_wb
+(9.34619431336624e-01))))
End

{Function Q_ECDH90MT_nuwe(T_db,T_wb,T_inlaat)
Q_ECDH90MT = (((5.21360306967587e+01) + ((-7.25366503289549e-02)*T_inlaat) + (( -3.77401257644041e-03)*T_inlaat^2) +
((7.15089269079599e-05)*T_inlaat^3) + ((1.80183230654392e+00)*T_db) + ((-2.49616311303084e-01)*T_wb)))*1.3
End}
{ ashpThrottle(hp_mass_flow_limit ,Q_besikbaar_toets ,T_internal ,T_set_point ,Cp,activateLoop_ASHP)}
Function ashpThrottle(hp_mass_flow_limit ,Q_besikbaar ,T_internal ,T_set_point ,Cp,activate_ASHP)
IF(activate_ASHP = 1) then {mag die HP aan wees}
if(Q_besikbaar = 0) THEN {limit floei as daar wel die masjien aan mag wees}
ashpThrottle = hp_mass_flow_limit
ELSE
m_toets = Q_besikbaar/(Cp*abs(T_set_point-T_internal)) {Bereken wat die massafloei moet wees}
if((m_toets > hp_mass_flow_limit)) then
ashpThrottle = hp_mass_flow_limit
else
ashpThrottle = m_toets
endif
ENDIF
else
ashpThrottle = 0.0001
endif
End

{Function ashpThrottle(hp_mass_flow_limit ,Q_besikbaar ,T_internal ,T_set_point ,Cp,activate_ASHP)
IF(activate_ASHP = 1) then
if(T_internal >= T_set_point) THEN {limit floei as daar wel die masjien aan mag wees}
ashpThrottle = hp_mass_flow_limit
ELSE
m_toets = Q_besikbaar/(Cp*(T_set_point-T_internal)) {Bereken wat die massafloei moet wees}
if((m_toets > hp_mass_flow_limit) OR (Q_besikbaar = 0)) then
ashpThrottle = hp_mass_flow_limit
else
ashpThrottle = m_toets
endif
ENDIF
else
ashpThrottle = 0.0001
endif
End}

{ ashpHeatThrottle(T_internal ,T_set_point+3,T_set_point ,Q_besikbaar_toets ,hp_mass_flow_limit ,activateLoop_ASHP) }
Function ashpHeatThrottle(T1,T_maks,T_set_point ,Q_in ,hp_mass_flow_limit ,activate_ashp)

Cp = Cp(Water,T=T1,P=80 [kPa])

if((T1 >= T_set_point) OR (activate_ashp = 0)) THEN
ashpHeatThrottle = 0
ELSE
m_toets = Q_in / (4200* (T_maks- T1))
if(m_toets > hp_mass_flow_limit) then
ashpHeatThrottle = Q_in { hp_mass_flow_limit*Cp* (T_maks- T1)} { Q_in*(hp_mass_flow_limit/m_toets)}
else
ashpHeatThrottle = Q_in
endif
ENDIF
End

{Function ashpHeatThrottle(T1,T_maks,T_set_point ,Q_in)
if((T1 >= T_maks) OR ((T_set_point - T1) < 2)) THEN
ashpHeatThrottle = 0
ELSE
ashpHeatThrottle = Q_in
ENDIF
End}

Function steadyStateTester(T1,T2,tolerance)
toetsWaarde = abs(T1 - T2)
if(toetsWaarde < 0.002) THEN
steadyStateTester = 0
ELSE
steadyStateTester = 1
ENDIF
End
```

Appendix K. Computer code (EES): Detailed system simulation

```
Function nullTester(M1)
if (M1 = 0) THEN
nullTester = 0
ELSE
nullTester = 1
ENDIF
End

"Bepaal die massavloei wat deur die HP moet vloei"
Function hpMassaVloei(h_inlet,Q_hp,T_set_point,P_druk,activate_heat_pump)
if(activate_heat_pump = 1) then "Indien die HP nie op tydskakelaar is nie bereken die volgende"
h_outlet = Enthalpy(Water,T=T_set_point,P=P_druk) "Bereken die verlangde uitlaat entalpie"
if(h_outlet = h_inlet) then
hpMassaVloei = 0.001 [kg/s]
else
hpMassaVloeiToets = Q_hp/(h_outlet - h_inlet) "Bereken die gestadigde massavloei wat deur die stelsel benodig word"
if (hpMassaVloeiToets>0.522) then
hpMassaVloei = 0.522
else
hpMassaVloei = hpMassaVloeiToets
endif
endif
else
hpMassaVloei = 0.001
endif
End

Function toetsVloei(M1,T1,T2)
if (M1 >= 0 ) THEN
toetsVloei = T1
ELSE
toetsVloei = T2
ENDIF
End

Module T_piece_komponent (M1,T1_f,T1_b,M2,T2_f,T2_b,M3,T3_f,T3_b)
{Neem aan dat 2 en 3 in vloei}
{Massabehoud tel altyd}
M3 = M2 + M1
{Vir die energiebalaans hang dit af van waar energie verkry word}
(IF (M3,0,T3_b,T3_f,T3_f)*M3) = (IF (M2,0,T2_b,T2_f,T2_f)*M2)+(IF (M1,0,T1_b,T1_f,T1_f)*M1)
end

Function Lyn_VorigeToetser(teller,vorigeWaarde,tabel$)
if(teller = 1) then
Lyn_VorigeToetser = vorigeWaarde
else
Lyn_VorigeToetser = tablevalue(TABLENAME$,teller-1,tabel$)
endif
End

function NodeTemp (M1,T1,M2,T2,M3,T3) {M3 = M2 + M1}
if ((M1 >= 0) and (M2 >= 0)) then
NodeTemp = T3
endif
if ((M1 >= 0) and (M2 > 0)) then
NodeTemp = T3
endif
end

Function eta_svv(T_amb,G,h_inlaat,P_atm)
{I_inlaat_water - Inlaat temperatuur van die SWV
T_amb - Die omgewings temperatuur
G - Sonbestraling
A_col - Die area van die kollektor}
{'Die energie wat by die tank gevoeg moet word as die tank temperatuur daal'}
if (G> 80) THEN
T_inlaat_water = Temperature(Water,P=P_atm,h=h_inlaat)
A = (T_inlaat_water-T_amb)/G
{eta_son = (m_dot(h_uitlaat-h_inlaat))/(G*A_col)}
{eta_svv =(-(4072*a^2) + (359.07*a) + 52.334)*1.92/100}
eta_svv = -(6250*a^2) + (12.5*a) + 53.125)*1.63/100
ELSE
eta_svv = 0
ENDIF
End

Function svv_controller(T_intern,T_2,T_3,T_max,P_atm,T_delta_hoog,T_delta_laag)
if ((T_intern - T_2)>T_delta_hoog) then
if (T_3 > T_max) then
svv_controller = 0
else
svv_controller = 1
endif
else
if ((T_intern-T_2) > T_delta_laag) Then
svv_controller = 1
else
svv_controller = 0
endif
endif
End
```

Appendix K. Computer code (EES): Detailed system simulation

```
Function hpTimerControlFunc(T_probe,T_probe_set_piont,teller,timer_hp_oud,activate_timer_hp_oud,t_set_point_2)
"Indien die program 'n 1 terugstuur dan moet die timer loop en die HP aangeskakel word"
if(T_probe < T_probe_set_piont) then "Indien die toetsnode se temperatuur kouer is as die set point begin aftel"
hpTimerControlFunc = 1
else
{if(teller > 1 ) then "Indien ons verby die eerste tydstap is."}
if(activate_timer_hp_oud = 1) then
if((timer_hp_oud > (900)) AND (T_probe >= t_set_point_2 )) then
hpTimerControlFunc = 0
else
hpTimerControlFunc= 1
endif
else
hpTimerControlFunc = 0
endif
{endif}
endif
end

Function hpTimerCounterFunc(hpTimerControlPar, tydstap_grootte,timer_hp_oud)
if (hpTimerControlPar =1) then
hpTimerCounterFunc = tydstap_grootte + timer_hp_oud
else
hpTimerCounterFunc = 0
endif
end

{Globale simulasie waardes}
delta_tyd = 1[s]
AFLOOP_TYD = delta_tyd * (teller -1)

T_delta_hoog = 12[C]
T_delta_laag = 8 [C]
T_max = 63 [C]
T_set = 52 [C]
T_set2 = 55 [C]
timer_hp_oud_init = 0 [s]
activate_timer_hp_oud_init = 0

activateLoop_ASHP =hpTimerControlFunc(Lyn_VorigeToetser(teller,T_intern_aanvanklik_4,'T_internal_4_new'),T_set,teller,
Lyn_VorigeToetser(teller,timer_hp_oud_init,'timer_hp_oud'),Lyn_VorigeToetser(teller,activate_timer_hp_oud_init,'
activate_timer_hp_oud'),T_set2)
activate_timer_hp_oud = activateLoop_ASHP
timer_hp_oud = hpTimerCounterFunc(activateLoop_ASHP,delta_tyd,Lyn_VorigeToetser(teller,timer_hp_oud_init,'timer_hp_oud'))

activateLoop_SWH = swv_controller(Lyn_VorigeToetser(teller,T_intern_aanvanklik_SWH,'T_internal_SWH_new'),Lyn_VorigeToetser(
teller,T_intern_aanvanklik_5,'T_internal_5_new'),Lyn_VorigeToetser(teller,T_intern_aanvanklik_7,'T_internal_7_new'),
T_max,85 [kPa],T_delta_hoog,T_delta_laag)

{***** (Pyp komponente)
*****}

{Pyp 1 konneksie}
m_poort_aanvanklik_1 = -0.222 [kg/s]
T_poort_aanvanklik_1 = 46.61 [C]
T_intern_aanvanklik_1 = 46.61 [C]
m_1 = 50 [kg]
f_1 = 0.1 [dim]
D_i_1 = 0.08 [m]
D_o_1 = 0.1 [m]
L_1 = 10 [m]
UA_1= 4.78{ 3.9 [dim]}

{Pyp 2 konneksie}
m_poort_aanvanklik_2 = 0.3247 [kg/s]
T_poort_aanvanklik_2 = 46.76 [C]
T_intern_aanvanklik_2 = 46.76 [C]
m_2 = 50 [kg]
f_2 = 0.1 [dim]
D_i_2 = 0.08 [m]
D_o_2 = 0.1 [m]
L_2 = 10 [m]
UA_2= 4.78 [dim]

{Pyp 3 konneksie}
m_poort_aanvanklik_3 = -0.1973 [kg/s]
T_poort_aanvanklik_3 = 50 [C]
T_intern_aanvanklik_3 = 50 [C]
m_3 = 3000 [kg]
f_3 = 0.4 [dim]
D_i_3 = 0.5 [m]
D_o_3 = 0.51 [m]
L_3 = 1 [m]
UA_3= 15.11 [dim] { 18.23 }

{Pyp 4 konneksie}
m_poort_aanvanklik_4 = -0.1973 [kg/s]
T_poort_aanvanklik_4 = 52 [C]
T_intern_aanvanklik_4 = 52 [C]
m_4 = 3000 [kg]
f_4 = 0.4 [dim]
D_i_4 = 0.5 [m]
D_o_4 = 0.51 [m]
L_4 = 1 [m]
```



```

UA_4= 15.11 [dim]

{Pyp 5 konneksie}
m_poort_aanvanklik_5 = -0.1973 [kg/s]
T_poort_aanvanklik_5 = 54 [C]
T_intern_aanvanklik_5 = 54 [C]
m_5 = 3000 [kg]
f_5 = 0.4 [dim]
D_i_5 = 0.5 [m]
D_o_5 = 0.51 [m]
L_5 = 1 [m]
UA_5= 15.11 [dim]

{Pyp 6 konneksie}
m_poort_aanvanklik_6 = -0.1973 [kg/s]
T_poort_aanvanklik_6 = 56 [C]
T_intern_aanvanklik_6 = 56 [C]
m_6 = 3000 [kg]
f_6 = 0.4 [dim]
D_i_6 = 0.5 [m]
D_o_6 = 0.51 [m]
L_6 = 1 [m]
UA_6= 15.11 { 18.23 [dim]}

{Pyp 7 konneksie}
m_poort_aanvanklik_7 = 0.1973 [kg/s]
T_poort_aanvanklik_7 = 58 [C]
T_intern_aanvanklik_7 = 58 [C]
m_7 = 3000 [kg]
f_7 = 0.4 [dim]
D_i_7 = 0.5 [m]
D_o_7 = 0.51 [m]
L_7 = 1 [m]
UA_7= 15.11 {18.23 [dim]}

{Pyp 8 konneksie}
m_poort_aanvanklik_8 = 0.522 [kg/s]
T_poort_aanvanklik_8 = 46.81 [C]
T_intern_aanvanklik_8 = 46.81 [C]
m_8 = 50 [kg]
f_8 = 3 [dim]
D_i_8 = 0.08 [m]
D_o_8 = 0.1 [m]
L_8 = 10 [m]
UA_8= 4.78 [dim]

{Pyp 9 konneksie}
m_poort_aanvanklik_9 = 0.522 [kg/s]
T_poort_aanvanklik_9 = 46.16 [C]
T_intern_aanvanklik_9 = 46.16 [C]
m_9 = 50 [kg]
f_9 = 3 [dim]
D_i_9 = 0.08 [m]
D_o_9 = 0.1 [m]
L_9 = 10 [m]
UA_9= 4.78 [dim]

{Pyp 10 konneksie}
m_poort_aanvanklik_10 = 0.3247 [kg/s]
T_poort_aanvanklik_10 = 54.82 [C]
T_intern_aanvanklik_10 = 54.82 [C]
m_10 = 500 [kg]
f_10 = 20 [dim]
D_i_10 = 0.08 [m]
D_o_10 = 0.1 [m]
L_10 = 100 [m]
UA_10= 90 [dim] {77}

{Pyp 11 konneksie}
m_poort_aanvanklik_11 = 0.02487 [kg/s]
T_poort_aanvanklik_11 = 50 [C]
T_intern_aanvanklik_11 = 50 [C]
m_11 = 500 [kg]
f_11 = 20 [dim]
D_i_11 = 0.08 [m]
D_o_11 = 0.1 [m]
L_11 = 100 [m]
UA_11= 90 [dim] {77}

{Pyp 12 konneksie}
m_poort_aanvanklik_12 = 0 [kg/s]
T_poort_aanvanklik_12 = 15.45 [C]
T_intern_aanvanklik_12 = 15.45 [C]
m_12 = 30 [kg]
f_12 = 0.32 [dim]
D_i_12 = 0.08 [m]
D_o_12 = 0.1 [m]
L_12 = 10 [m]
UA_12=0.001{ 5.63 [dim]}

{Pyp 13 konneksie}
m_poort_aanvanklik_13 = 0 [kg/s]
T_poort_aanvanklik_13 = 15.44 [C]
T_intern_aanvanklik_13 = 15.44 [C]

```

Appendix K. Computer code (EES): Detailed system simulation

```
m_13 = 30 [kg]
f_13 = 0.32 [dim]
D_i_13 = 0.08 [m]
D_o_13 = 0.1 [m]
L_13 = 10 [m]
UA_13= 0.001{5.63 [dim]}

{*****(T-stukke)
*****}
{T_stuk A konneksies}
m_poort_aanvanklik_a = 0 [kg/s]
T_poort_aanvanklik_a = 27.37 [C]
T_intern_aanvanklik_a = 27.37 [C]
m_a = 5 [kg]
UA_a = 2 [dim]

{T_stuk B konneksies}
m_poort_aanvanklik_b = 0 [kg/s]
T_poort_aanvanklik_b = 46.68 [C]
T_intern_aanvanklik_b = 46.68 [C]
m_b = 5 [kg]
UA_b = 2 [dim]

{T_stuk C konneksies}
m_poort_aanvanklik_c = 0 [kg/s]
T_poort_aanvanklik_c = 46.81 [C]
T_intern_aanvanklik_c = 46.81 [C]
m_c = 5 [kg]
UA_c = 2 [dim]

{T_stuk d konneksies}
m_poort_aanvanklik_d = 0 [kg/s]
T_poort_aanvanklik_d = 47.27 [C]
T_intern_aanvanklik_d = 47.27 [C]
m_d = 5 [kg]
UA_d = 2 [dim]

{T_stuk e konneksies}
m_poort_aanvanklik_e = 0 [kg/s]
T_poort_aanvanklik_e = 59 [C]
T_intern_aanvanklik_e = 59 [C]
m_e = 5 [kg]
UA_e = 2 [dim]

{T_stuk f konneksies}
m_poort_aanvanklik_f = 0 [kg/s]
T_poort_aanvanklik_f = 53 [C]
T_intern_aanvanklik_f = 53 [C]
m_f = 5 [kg]
UA_f = 2 [dim]

{*****(Ander komponente)
*****}
{SWH 4 konneksies}
m_poort_aanvanklik_SWH = 0 [kg/s]
T_poort_aanvanklik_SWH = 4.823 [C]
T_intern_aanvanklik_SWH = 4.823 [C]
m_SWH = 30 [kg]
area_SWH_collector = 7.4[m^2]
D_i_swh = 0.05
D_o_swh = 0.06
L_swh = 15
UA_swh = 30
{f_swh = 1.5}

{f_swh = 0.5}

{ASHP konneksies}
m_poort_aanvanklik_ASHP = 0.522 [kg/s]
T_poort_aanvanklik_ASHP = 60 [C]
T_intern_aanvanklik_ASHP = 60 [C]
m_ASHP = 5 [kg]
area_ASHP_collector = 7.44 [m^2]
D_i_ashp = 0.05 [m]
D_o_ashp = 0.06 [m]
L_ashp = 10 [m]
UA_ashp = 0 {2 [W/K]}
Q_oud_losses_aanvanklik = 0 [W]

T_set_point_ASHP = 60 [DEG]

{*****(Konneksies tussen komponente)
*****}

{Drukval vir die verskeie loops}
{ASHP loop}
Drukval_ASHP_A = deltaP_8 + deltaP_ASHP + deltaP_9
Drukval_ASHP_B = deltaP_1 + deltaP_2 + deltaP_3+ deltaP_4+ deltaP_5+ deltaP_6+ deltaP_7
0 = Drukval_ASHP_A + Drukval_ASHP_B
0 = if(activateLoop_ASHP,0,m_out_9,Drukval_ASHP_A + Drukval_ASHP_B)
0 = if(activateLoop_ASHP,0,F_ASHP,F_ASHP,dummy)}
```

Appendix K. Computer code (EES): Detailed system simulation

```
{T-stuk konneksies}
CALL T_stuk_komponent(teller ,m_system_in,-m_dot_1,-m_dot_8,T_system_in,Lyn_VorigeToetser(teller ,T_intern_aanvanklik_1 ,
T_intern_1_new'),Lyn_VorigeToetser(teller ,T_intern_aanvanklik_8 ,T_intern_8_new'),m_a,Lyn_VorigeToetser(teller ,
T_intern_aanvanklik_a ,T_intern_a_new'),delta_tyd,UA_a,T_db, T_intern_a_new,Q_loss_A)

CALL T_stuk_komponent(teller ,m_out_1,-m_dot_2,-m_dot_12,Lyn_VorigeToetser(teller ,T_intern_aanvanklik_1 ,T_intern_1_new'),
Lyn_VorigeToetser(teller ,T_intern_aanvanklik_2 ,T_intern_2_new'),Lyn_VorigeToetser(teller ,T_intern_aanvanklik_12 ,
T_intern_12_new'),m_b,Lyn_VorigeToetser(teller ,T_intern_aanvanklik_b ,T_intern_b_new'),delta_tyd,UA_b,T_db,
T_intern_b_new,Q_loss_B)

CALL T_stuk_komponent(teller ,m_out_2,m_out_11,-m_dot_3,Lyn_VorigeToetser(teller ,T_intern_aanvanklik_2 ,T_intern_2_new'),
Lyn_VorigeToetser(teller ,T_intern_aanvanklik_11 ,T_intern_11_new'),Lyn_VorigeToetser(teller ,T_intern_aanvanklik_3 ,
T_intern_3_new'),m_c,Lyn_VorigeToetser(teller ,T_intern_aanvanklik_c ,T_intern_c_new'),delta_tyd,UA_c,T_db,
T_intern_c_new,Q_loss_C)

CALL T_stuk_komponent(teller ,m_out_4,-m_dot_5,m_out_13,Lyn_VorigeToetser(teller ,T_intern_aanvanklik_4 ,T_intern_4_new'),
Lyn_VorigeToetser(teller ,T_intern_aanvanklik_5 ,T_intern_5_new'),Lyn_VorigeToetser(teller ,T_intern_aanvanklik_13 ,
T_intern_13_new'),m_d,Lyn_VorigeToetser(teller ,T_intern_aanvanklik_d ,T_intern_d_new'),delta_tyd,UA_d,T_db,
T_intern_d_new,Q_loss_D)

CALL T_stuk_komponent(teller ,m_out_9,-m_dot_10,m_out_7,Lyn_VorigeToetser(teller ,T_intern_aanvanklik_9 ,T_intern_9_new'),
Lyn_VorigeToetser(teller ,T_intern_aanvanklik_10 ,T_intern_10_new'),Lyn_VorigeToetser(teller ,T_intern_aanvanklik_3 ,
T_intern_7_new'),m_e,Lyn_VorigeToetser(teller ,T_intern_aanvanklik_e ,T_intern_e_new'),delta_tyd,UA_e,T_db,
T_intern_e_new,Q_loss_E)

CALL T_stuk_komponent(teller ,m_dot_10,-m_dot_11,-m_dot_system_out,Lyn_VorigeToetser(teller ,T_intern_aanvanklik_10 ,
T_intern_10_new'),Lyn_VorigeToetser(teller ,T_intern_aanvanklik_11 ,T_intern_11_new'),Lyn_VorigeToetser(teller ,
T_intern_aanvanklik_f ,T_intern_f_new'),m_f,Lyn_VorigeToetser(teller ,T_intern_aanvanklik_f ,T_intern_f_new'),
delta_tyd,UA_f,T_db, T_intern_f_new,Q_loss_F)

{Pyp_konneksies}
CALL Pyp_komponent(teller ,m_dot_1,Lyn_VorigeToetser(teller ,T_intern_aanvanklik_a ,T_intern_a_new'), Lyn_VorigeToetser(teller
,T_intern_aanvanklik_b ,T_intern_b_new'),m_1, Lyn_VorigeToetser(teller ,T_intern_aanvanklik_1 ,T_intern_1_new'),
delta_tyd,f_1,D_i_1,D_o_1,L_1,UA_1,T_db,0,0,m_out_1,T_out_1,T_intern_1_new,deltaP_1,T_gebruik_in_1,Q_losses_1,
Q_in_pyp_1,Q_toets_in_1,Q_toets_uit_1)

CALL Pyp_komponent(teller ,m_dot_2 , Lyn_VorigeToetser(teller ,T_intern_aanvanklik_b ,T_intern_b_new'), Lyn_VorigeToetser(
teller ,T_intern_aanvanklik_c ,T_intern_c_new'),m_2, Lyn_VorigeToetser(teller ,T_intern_aanvanklik_2 ,T_intern_2_new
'),delta_tyd,f_2,D_i_2,D_o_2,L_2,UA_2,T_db,0,0,m_out_2,T_out_2,T_intern_2_new,deltaP_2,T_gebruik_in_2,Q_losses_2,
Q_in_pyp_2,Q_toets_in_2,Q_toets_uit_2)

CALL Pyp_komponent(teller ,m_dot_3, Lyn_VorigeToetser(teller ,T_intern_aanvanklik_c ,T_intern_c_new'), Lyn_VorigeToetser(
teller ,T_intern_aanvanklik_4 ,T_intern_4_new'),m_3, Lyn_VorigeToetser(teller ,T_intern_aanvanklik_3 ,T_intern_3_new')
),delta_tyd,f_3,D_i_3,D_o_3,L_3,UA_3,T_db,0,0,m_dot_4,T_out_3,T_intern_3_new,deltaP_3,T_gebruik_in_3,Q_losses_3,
Q_in_pyp_3,Q_toets_in_3,Q_toets_uit_3)

CALL Pyp_komponent(teller ,m_dot_4, Lyn_VorigeToetser(teller ,T_intern_aanvanklik_3 ,T_intern_3_new'), Lyn_VorigeToetser(
teller ,T_intern_aanvanklik_d ,T_intern_d_new'),m_4, Lyn_VorigeToetser(teller ,T_intern_aanvanklik_4 ,T_intern_4_new
'),delta_tyd,f_4,D_i_4,D_o_4,L_4,UA_4,T_db,0,0,m_out_4,T_out_4,T_intern_4_new,deltaP_4,T_gebruik_in_4,Q_losses_4,
Q_in_pyp_4,Q_toets_in_4,Q_toets_uit_4)

CALL Pyp_komponent(teller ,m_dot_5 , Lyn_VorigeToetser(teller ,T_intern_aanvanklik_d ,T_intern_d_new'), Lyn_VorigeToetser(
teller ,T_intern_aanvanklik_6 ,T_intern_6_new'),m_5, Lyn_VorigeToetser(teller ,T_intern_aanvanklik_5 ,T_intern_5_new
'),delta_tyd,f_5,D_i_5,D_o_5,L_5,UA_5,T_db,0,0,m_dot_6,T_out_5,T_intern_5_new,deltaP_5,T_gebruik_in_5,Q_losses_5,
Q_in_pyp_5,Q_toets_in_5,Q_toets_uit_5)

CALL Pyp_komponent(teller ,m_dot_6, Lyn_VorigeToetser(teller ,T_intern_aanvanklik_5 ,T_intern_5_new'), Lyn_VorigeToetser(
teller ,T_intern_aanvanklik_7 ,T_intern_7_new'),m_6, Lyn_VorigeToetser(teller ,T_intern_aanvanklik_6 ,T_intern_6_new
'),delta_tyd,f_6,D_i_6,D_o_6,L_6,UA_6,T_db,0,0,m_dot_7,T_out_6,T_intern_6_new,deltaP_6,T_gebruik_in_6,Q_losses_6,
Q_in_pyp_6,Q_toets_in_6,Q_toets_uit_6)

CALL Pyp_komponent(teller ,m_dot_7 , Lyn_VorigeToetser(teller ,T_intern_aanvanklik_6 ,T_intern_6_new'), Lyn_VorigeToetser(
teller ,T_intern_aanvanklik_e ,T_intern_e_new'),m_7, Lyn_VorigeToetser(teller ,T_intern_aanvanklik_7 ,T_intern_7_new')
),delta_tyd,f_7,D_i_7,D_o_7,L_7,UA_7,T_db,0,0,m_out_7,T_out_7,T_intern_7_new,deltaP_7,T_gebruik_in_7,Q_losses_7,
Q_in_pyp_7,Q_toets_in_7,Q_toets_uit_7)

CALL Pyp_komponent(teller ,m_dot_8 , Lyn_VorigeToetser(teller ,T_intern_aanvanklik_a ,T_intern_a_new'), Lyn_VorigeToetser(
teller ,T_intern_aanvanklik_ASHP ,T_intern_ASHP_new'),m_8, Lyn_VorigeToetser(teller ,T_intern_aanvanklik_8 ,
T_intern_8_new'),delta_tyd,f_8,D_i_8,D_o_8,L_8,UA_8,T_db,0,0,m_dot_ASHP,T_out_8,T_intern_8_new,deltaP_8,
T_gebruik_in_8,Q_losses_8,Q_in_pyp_8,Q_toets_in_8,Q_toets_uit_8)

CALL Pyp_komponent(teller ,m_out_ASHP , Lyn_VorigeToetser(teller ,T_intern_aanvanklik_ASHP ,T_intern_ASHP_new'),
Lyn_VorigeToetser(teller ,T_intern_aanvanklik_e ,T_intern_e_new'),m_9, Lyn_VorigeToetser(teller ,T_intern_aanvanklik_9
,T_intern_9_new'),delta_tyd,f_9,D_i_9,D_o_9,L_9,UA_9,T_db,0,0,m_out_9,T_out_9,T_intern_9_new,deltaP_9,
T_gebruik_in_9,Q_losses_9,Q_in_pyp_9,Q_toets_in_9,Q_toets_uit_9)
m_out_9 = m_dot_9

CALL Pyp_komponent(teller ,m_dot_10, Lyn_VorigeToetser(teller ,T_intern_aanvanklik_e ,T_intern_e_new'),Lyn_VorigeToetser(
teller ,T_intern_aanvanklik_f ,T_intern_f_new'),m_10, Lyn_VorigeToetser(teller ,T_intern_aanvanklik_10 ,
T_intern_10_new'),delta_tyd,f_10,D_i_10,D_o_10,L_10,UA_10,T_db,0,0,m_out_10,T_out_10,T_intern_10_new,deltaP_10,
T_gebruik_in_10,Q_losses_10,Q_in_pyp_10,Q_toets_in_10,Q_toets_uit_10)

CALL Pyp_komponent(teller ,m_dot_11 , Lyn_VorigeToetser(teller ,T_intern_aanvanklik_f ,T_intern_f_new'), Lyn_VorigeToetser(
teller ,T_intern_aanvanklik_c ,T_intern_c_new'),m_11, Lyn_VorigeToetser(teller ,T_intern_aanvanklik_11 ,
T_intern_11_new'),delta_tyd,f_11,D_i_11,D_o_11,L_11,UA_11,T_db,Q_pomp_Ring,H_pomp_Ring,m_out_11,T_out_11,
T_intern_11_new,deltaP_11,T_gebruik_in_11,Q_losses_11,Q_in_pyp_11,Q_toets_in_11,Q_toets_uit_11)

CALL Pyp_komponent(teller ,m_dot_12 , Lyn_VorigeToetser(teller ,T_intern_aanvanklik_b ,T_intern_b_new'), Lyn_VorigeToetser(
teller ,T_intern_aanvanklik_SWH ,T_intern_SWH_new'),m_12, Lyn_VorigeToetser(teller ,T_intern_aanvanklik_12 ,
T_intern_12_new'),delta_tyd,f_12,D_i_12,D_o_12,L_12,UA_12,T_db,0,0,m_dot_SWV,T_out_12,T_intern_12_new,deltaP_12,
T_gebruik_in_12,Q_losses_12,Q_in_pyp_12,Q_toets_in_12,Q_toets_uit_12)
```

Appendix K. Computer code (EES): Detailed system simulation

```
CALL Pyp_komponent(teller ,m_out_SWH , Lyn_VorigeToetser(teller ,T_intern_aanvanklik_SWH,'T_internal_SWH_new'),Lyn_VorigeToetser
(teller ,T_intern_aanvanklik_D,'T_internal_D_new'),m_13, Lyn_VorigeToetser(teller ,T_intern_aanvanklik_13,'
T_internal_13_new'),delta_tyd,f_13,D_i_13,D_o_13,L_13,UA_13,T_db,0,0,m_out_13,T_out_13,T_internal_13_new,deltaP_13,
T_gebruik_in_13,Q_losses_13,Q_in_pyp_13,Q_toets_in_13,Q_toets_uit_13)

{Pomp konneksies}
{CALL Pomp_komponent(m_dot_SWV,Lyn_VorigeToetser(teller ,T_intern_aanvanklik_12,'T_internal_12_new'),0.6492,-0.31638,4.4474,
Q_pomp_SWH,H_pomp_SWH) {SWH pomp}}
CALL Pomp_komponent(m_dot_SWV,Lyn_VorigeToetser(teller ,T_intern_aanvanklik_12,'T_internal_12_new'),-0.1579,-0.5788,7.9989,
Q_pomp_SWH,H_pomp_SWH) {SWH pomp}
CALL Pomp_komponent(m_dot_8,Lyn_VorigeToetser(teller ,T_intern_aanvanklik_ASHP,'T_internal_ASHP_new'),-0.1579,-0.5788,7.9989,
Q_pomp_ASHP,H_pomp_ASHP) {ASHP pomp}
CALL Pomp_komponent(m_dot_11 ,Lyn_VorigeToetser(teller ,T_intern_aanvanklik_11,'T_internal_11_new'),-0.0414,-1.4047,5.5923,
Q_pomp_Ring,H_pomp_Ring) {Ring-main pomp}

{SWH}
CALL SWH_komponent(teller ,m_dot_SWV,Lyn_VorigeToetser(teller ,T_intern_aanvanklik_12,'T_internal_12_new'),m_SWH,
Lyn_VorigeToetser(teller ,T_intern_aanvanklik_SWH,'T_internal_SWH_new'),delta_tyd,f_swh,D_i_swh,D_o_swh,L_swh,UA_swh,
area_SWH_collector ,T_db,Q_pomp_SWH,H_pomp_SWH,m_out_SWH,T_out_SWH,T_internal_SWH_new,deltaP_swh,Q_solar,G,
Q_heat_loss_solar,-Q_solar_contribution,activateLoop_SWH,Q_in_solar,Q_uit_solar,Q_in_swh,Q_uit_swh)

{ASHP component}
Q_potential = Q_ECDH90MT_nuwe(T_db,RH,Lyn_VorigeToetser(teller ,T_intern_aanvanklik_8,'T_internal_8_new'),T_wb)*1000[w/kw]
CALL ASHP_komponent(teller ,m_dot_ASHP,Lyn_VorigeToetser(teller ,T_intern_aanvanklik_8,'T_internal_8_new'),m_ASHP,
Lyn_VorigeToetser(teller ,T_intern_aanvanklik_ASHP,'T_internal_ASHP_new'),delta_tyd,D_i_ashp,D_o_ashp,L_ashp,UA_ashp,T_db
,RH,Q_pomp_ASHP,H_pomp_ASHP,m_out_ASHP,T_out_ASHP,T_internal_ASHP_new,deltaP_ASHP,T_set_point_ASHP,F_ASHP,
activateLoop_ASHP,Q_hp,Q_hp2,Q_hp_losses,T_wb,Q_hp3,Q_in_ashp,Q_uit_ashp,Q_ashp_pomp)

{Q_losses_new = 0}

{SWH loop}
Drukval_SWH_A = deltaP_12 + deltaP_SWH + deltaP_13
Drukval_SWH_B = deltaP_2 + deltaP_3 + deltaP_4
0 = Drukval_SWH_A + Drukval_SWH_B
{0 = if(activateLoop_SWH,0,m_out_SWH,m_out_SWH,Drukval_SWH_A + Drukval_SWH_B)}

{Ring-main loop}
Drukval_RL_A = deltaP_3 + deltaP_4 + deltaP_5 + deltaP_6 + deltaP_7 + deltaP_10
Drukval_RL_B = deltaP_11
0 = Drukval_RL_A + Drukval_RL_B

{Enegietermes}
Q_in = m_dot_system_out*Cp(Water,T=T_system_in,P=80 [kPa])*T_system_in
Q_uit = m_dot_system_out*Cp(Water,T=Lyn_VorigeToetser(teller ,T_intern_aanvanklik_f,'T_internal_f_new'),P=80 [kPa])*
Lyn_VorigeToetser(teller ,T_intern_aanvanklik_f,'T_internal_f_new')
Q_user = Q_uit - Q_in
Q_losses_tank = Q_losses_3 + Q_losses_4 + Q_losses_5 + Q_losses_6 + Q_losses_7 + Q_loss_C + Q_loss_D + Q_loss_E
Q_losses_pipe = Q_losses_1 + Q_losses_2 + Q_losses_8 + Q_losses_9 + Q_losses_12 + Q_losses_13 + Q_heat_loss_solar
Q_losses_pipe_2 = Q_losses_1 + Q_losses_2 + Q_losses_8 + Q_losses_9 + Q_loss_B + Q_loss_A
Q_losses_solar = Q_losses_13 + Q_losses_12 + Q_uit_swh
Q_pipe_losses_solar = Q_losses_13 + Q_losses_12
Q_pompe = Q_in_pyp_1 + Q_in_pyp_2 + Q_in_pyp_3 + Q_in_pyp_4 + Q_in_pyp_5 + Q_in_pyp_6 + Q_in_pyp_7 + Q_in_pyp_8 + Q_in_pyp_9 +
Q_in_pyp_10 + Q_in_pyp_11 + Q_in_pyp_12 + Q_in_pyp_13
Q_solar_energy = Q_solar_contribution
Q_ringmain_losses = m_dot_11 * Cp(Water,T=Lyn_VorigeToetser(teller ,T_intern_aanvanklik_11,'T_internal_11_new'),P=80 [kPa]) * (
Lyn_VorigeToetser(teller ,T_intern_aanvanklik_e,'T_internal_e_new')-Lyn_VorigeToetser(teller ,T_intern_aanvanklik_11,'
T_internal_11_new'))
Q_ringmain_losses2 = Q_losses_10 + Q_losses_11 + Q_loss_F
Q_netto = Q_in_ashp + Q_uit_ashp
Q_netto2 = Q_in_solar + Q_uit_swh
Q_netto3 = Q_hp + Q_ashp_pomp + Q_losses_pipe_2 + Q_ringmain_losses2 - Q_user + Q_in_swh + Q_losses_solar + Q_pompe + Q_losses_tank

Q_in_total = Q_hp + Q_ashp_pomp + Q_in_swh + Q_pompe + Q_in
Q_uit_total = Q_uit - Q_losses_pipe_2 - Q_ringmain_losses2 - Q_losses_solar - Q_losses_tank
Q_verskil = Q_in_total - Q_uit_total
```

Appendix L

Computer code (EES): Reduced system simulation

{WaterTank_stratifikasie_model_rev31 (test data) rev 6

09/2019 by Jean-Pierre de Villiers
24875252

```
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Procedure stratifikasieTank3 (activate_heat_pump, electrical_element, T_set_aux, n, P_atm, q_element, T_hp, T_munisipaliteit_koud,
    T_ringfeed_return, T_wb_i, T_db_i, k_water, h_kante, h_top, h_onder, rho_water, cp_water, h_node, area, area2, T_aanvangs[1..5],
    m_dot_return_i, m_dot_svv_aanvanklik, m_dot_hp_aanvanklik, m_dot_in, m_dot_poort_i, m_dot_return, n_step, tydstep_grootte,
    teller, G, A_col, T_delta_hoog, T_delta_laag, L_d, d_o, K_d, N_drawoffs, l_dl, d_d, T_indoors, T_probe_hp, RH, UA_solar,
    UA_solar_manifold, UA_tank_pipes, N_hp, N_solar, T_svv_i, m_solar_bank, T_svv_pype_i, m_solar_pype, density_ratio, m_dot_ashp_max,
    ashp_ratio:tyd, stratifikasieTank[1..5], stratTankEntalp[1..5], T_return, m_dot[1..5], m_dot_hp_probe, h_poort_1, m_svv_nou,
    M_dot_1, M_dot_2, M_dot_3, M_dot_4, M_dot_5, h_svv_nou, eta_svv, konduksie_term[1..5], konveksie_term[1..5], geforseerdeVloed
    [1..5], Q_dot_munisipaliteit, T_aftap, Q_losses_d, q_probe_hp, Q_losses_dead_end, Q_son_probe, Q_return, Q_top, T_meng, T_top_meng,
    m_dot_hp, m_dot_poort, m_dot_svv, timer_hp, activate_timer_hp, Q_netto, m_dot_ringfeed, TS, cycle$, Q_hp_probe,
    hp_timer_control_internal, Q_losses_tank, Q_user, Q_user2, Q_ringmain_normal, Q_losses_tank_pype, COP_hp, Q_elektries_hp,
    COP_hp_potential, Stelsel_entalpie, T_svv, T_room, Q_hp_probe_test, Q_netto2, Q_netto3, Q_solar_losses, Q_hello_ashp, die_probe,
    m_dot_hp_swh, m_hello_ashp, diagnose_condition, Q_supplies_energy, Q_in, Q_out, Q_net, Q_ringmain_test, Q_heat_pump_loop_test,
    Sigma_ringmain_massflux, Sigma_ashp_loop, Q_tester, Q_tester_2, Q_tester_3, Q_tester_4, mass_balance_tanks_top,
    mass_balance_tanks_bottom, Q_tank_top, Q_tank_bottom, Q_net, laaste, kode_blok, h_svv_new, Q_son_probe2, Q_son_losses_manifold,
    h_svv_pype, T_svv_pype)
```

"------(Massavloei en energie vergelykings)

T\$=concat('The time is now ', TIME\$)
{ Time_function\$(1)}
IF (teller >=2) then

```
h_oud_1 = tablevalue(TABLENAMES, teller -1, 'stratTankEntalp[1]')
h_oud_2 = tablevalue(TABLENAMES, teller -1, 'stratTankEntalp[2]')
h_oud_3 = tablevalue(TABLENAMES, teller -1, 'stratTankEntalp[3]')
h_oud_4 = tablevalue(TABLENAMES, teller -1, 'stratTankEntalp[4]')
h_oud_5 = tablevalue(TABLENAMES, teller -1, 'stratTankEntalp[5]')
```

```
T_oud_1 = Temperature(Water, h=h_oud_1, P=P_atm)
T_oud_2 = Temperature(Water, h=h_oud_2, P=P_atm)
T_oud_3 = Temperature(Water, h=h_oud_3, P=P_atm)
T_oud_4 = Temperature(Water, h=h_oud_4, P=P_atm)
T_oud_5 = Temperature(Water, h=h_oud_5, P=P_atm)
```

T_aftap_oud = tablevalue(TABLENAMES, teller -1, 'T_aftap')

```
{m_dot_poort = tablevalue(TABLENAMES, teller -1, 'm_dot_poort')}
m_dot_return = tablevalue(TABLENAMES, teller, 'm_dot_return')
T_omgewing_db = tablevalue(TABLENAMES, teller, 'T_omgewing_db')
T_omgewing_wb = tablevalue(TABLENAMES, teller, 'T_omgewing_wb')
m_dot_svv_oud = tablevalue(TABLENAMES, teller -1, 'm_svv_nou')
{h_meng_oud = tablevalue(TABLENAMES, teller -1, 'm_svv_nou')}
```

```
h_svv_oud = tablevalue(TABLENAMES, teller -1, 'h_svv_new')
h_svv_pype_oud = tablevalue(TABLENAMES, teller -1, 'h_svv_pype')
T_svv_pype = Temperature(Water, h = h_svv_pype_oud, P=P_atm)
```

```
timer_hp_oud = tablevalue(TABLENAMES, teller -1, 'timer_hp')
activate_timer_hp_oud = tablevalue(TABLENAMES, teller -1, 'activate_timer_hp')
```

```
else
T_oud_1 = T_aanvangs[1]
T_oud_2 = T_aanvangs[2]
T_oud_3 = T_aanvangs[3]
T_oud_4 = T_aanvangs[4]
T_oud_5 = T_aanvangs[5]
h_oud_1 = Enthalpy(Water, T=T_oud_1, P=P_atm)
h_oud_2 = Enthalpy(Water, T=T_oud_2, P=P_atm)
h_oud_3 = Enthalpy(Water, T=T_oud_3, P=P_atm)
h_oud_4 = Enthalpy(Water, T=T_oud_4, P=P_atm)
```

Appendix L. Computer code (EES): Reduced system simulation

```

h_oud_5 =Enthalpy (Water,T=T_oud_5,P=P_atm)

T_omgewing_db = T_db_i
T_omgewing_wb = T_wb_i
T_aftap_oud = T_oud_5-2
m_dot_svw_oud = m_dot_svw_aanvanklik
m_dot_poort = m_dot_poort_i

timer_hp_oud = 0
activate_timer_hp_oud = 0
m_dot_ringfeed = m_dot_return
{m_svw_nou = m_dot_svw_aanvanklik}

h_svw_oud =Enthalpy (Water,T = T_svw_i,P=P_atm)
h_svw_pype_oud =Enthalpy (Water,T = T_svw_pype_i,P=P_atm)
T_svw_pype = T_svw_pype_i
EndIf
"Initialiseer die stelsel entalpie vir die verskeie"
h_hp = Enthalpy (Water,T=T_hp,P=P_atm) "Vaste entalpie vir alle waardes"
h_munispaliteit = Enthalpy (Water,T=T_munispaliteit_koud,P=P_atm) "(Berekening)"
"-----HP aktiewe logika berekeninge"
"-----"
"Die HP se vergelykings"
"Bepaal vannuit die vorige berekeninge of dit charged on non charged vloei is"
"Gebruik die vorige entalpie en massavloei en bepaal die inlaat kondisies van die HP"
"Indien die strome meng bepaal die gemengde entalpie op die inlaat kant van die HP"
"Bepaal die HP se massavloei vannuit die kondisies"
"Bereken die stelsel se nuwe massavloei"
"Bereken die stelsel se nuwe energie vergelykings"
"Bepaal of die HP aan geskakel moet word"

T_swh_max = T_set_aux + 8
t_probe_hp_2 = T_probe_hp

hpTimerControl = hpTimerControl(T_oud_2,T_probe_hp-3,teller ,timer_hp_oud ,activate_timer_hp_oud ,t_probe_hp_2+1)
{pre_COP = Q_ECDH90M_COP (T_omgewing_db,RH, Temperature (Water,P=P_atm,h=h_oud_1))
  hpTimerControl = hpTimerControl_nuwe(T_oud_2,T_probe_hp,teller ,timer_hp_oud ,activate_timer_hp_oud ,pre_COP)}

if (hpTimerControl =1) then
timer_hp = tydstep_grootte + timer_hp_oud
activate_timer_hp = 1
hp_timer_control_internal = 0
else
timer_hp = 0
activate_timer_hp = 0
hp_timer_control_internal = 0
endif
if(timer_hp > 900) then
hp_timer_control_internal = 1
endif

h_svw = h_svw_oud {h_uitlaat_svw(T_omgewing_db,G,A_col,h_poort_1,P_atm,m_dot_svw_aanvanklik)}
T_svw = Temperature (Water,P=P_atm,h=h_svw)
T_svw_oud = Temperature (Water,P=P_atm,h=h_svw_oud)
m_svw_activate = m_svw_activate(h_svw,T_oud_3,T_oud_5,T_swh_max,P_atm,T_delta_hoog,T_delta_laag,m_dot_svw_oud)

{(T_db,RH,T_wb,h_munispaliteit,T_set_point,P_druk,activate_heat_pump,N_hp,m_dot_user,m_dot_ring_return,m_dot_svw,N_solar,
  h_poort,m_dot_in)}
m_dot_poort_a = m_poort_calc (T_omgewing_db,RH,T_omgewing_wb,h_munispaliteit,T_hp,P_atm,hp_timer_control_internal,N_hp,
  m_dot_in,m_dot_return,m_svw_activate*m_dot_svw_aanvanklik,N_solar,h_oud_1,m_dot_in,density_ratio,m_dot_ashp_max,
  ashp_ratio)

if(m_dot_poort_a <= 0) THEN "Laai - Siklus / Charge Cycle vir die ASHP en die SWH"

cycle$ = 'Charge'
h_poort_1 = h_oud_1
if(m_dot_poort_a = 0) THEN "Indien die water vir die SWH en ASHP nie getrek word van die stelsel kry hy dit direk van die
  munispaliteit af"
{if(m_dot_svw_oud > 0) then
  h_meng = h_oud_1
  else
  h_meng = h_munispaliteit
  endif}
h_meng = h_munispaliteit
h_meng_son = h_munispaliteit
die_probe = 22

ELSE
if(m_dot_in >= 0) then "Daar kom water in"
if((-m_dot_poort_a > ((m_svw_activate*m_dot_svw_aanvanklik)*N_solar))) THEN "Die water word verby die SWH getrek"

if ((m_dot_in + (-m_dot_poort_a -((m_svw_activate*m_dot_svw_aanvanklik)*N_solar)))>0.1) then "M_ashp > 0"
h_meng = ((m_dot_in* h_munispaliteit) + ((-m_dot_poort_a -((m_svw_activate*m_dot_svw_aanvanklik)*N_solar))*h_poort_1))/(
  m_dot_in + (-m_dot_poort_a -((m_svw_activate*m_dot_svw_aanvanklik)*N_solar)))

else
h_meng = h_oud_1
endif
h_meng_son = h_oud_1
{h_meng_calc (T_omgewing_db,RH,T_omgewing_wb,h_munispaliteit,T_hp,P_atm,activate_heat_pump,N_hp,m_dot_in,m_dot_return,
  m_dot_svw_oud,N_solar,h_oud_1,m_dot_in)} { ((m_dot_in* h_munispaliteit) - ((m_dot_poort+(m_dot_svw_oud*N_solar))*
  h_poort_1))/(m_dot_in - (m_dot_poort+(m_dot_svw_oud*N_solar)))} "h_meng is die inlaat entalpie van slegs die hittepomp"

die_probe =44

```

Appendix L. Computer code (EES): Reduced system simulation

```
ELSE
h_meng_son = (((-m_dot_poort_a*h_oud_1) + ((m_dot_poort_a+((m_svv_activate*m_dot_svv_aanvanklik)*N_solar))* h_munispaliteit))/
m_svv_activate+m_dot_svv_aanvanklik*N_solar)

h_meng = h_munispaliteit

die_probe = 33
ENDIF
endif
ENDIF

T_meng = Temperature(Water,P=P_atm,h=h_meng)
q_probe_hp = Q_ECDH90MT_nuwe(T_omgewing_db,RH, Temperature(Water,P=P_atm,h=h_meng),T_omgewing_wb,density_ratio,ashp_ratio)
if (hp_timer_control_internal >= 1) then
m_dot_hp = hpMassaVloei(h_meng,q_probe_hp,T_hp,P_atm,activate_heat_pump,m_dot_ashp_max)
else
m_dot_hp = 0
endif
Q_hp_probe_test = m_dot_hp*(h_hp-h_meng)

if (T_svv_pype > T_omgewing_db) then
Q_solar_losses = UA_solar * (T_svv_pype- T_omgewing_db)*N_solar {SWH supply pipe losses}
else
Q_solar_losses = 0
endif
eta_svv = eta_svv(T_omgewing_db,G,h_poort_1,P_atm)

Q_son_probe = m_dot_svv_aanvanklik*(h_svv-h_meng_son)*N_solar*m_svv_activate {SWH vloei energie}

if (eta_svv > 0) then
Q_son_probe2 = (eta_svv*G*A_col*N_solar)/1000 {SWV - inkomende energie}
Q_son_losses_manifold= ((1-eta_svv-((100-(53.125*1.63))/100))*G*A_col*N_solar)/1000 {SWV - inkomende energie - optical losses
have been added}
else
Q_son_probe2 = 0
if (T_svv_oud > T_omgewing_db ) then
Q_son_losses_manifold = UA_solar_manifold * (T_svv_oud- T_omgewing_db)*N_solar
else
Q_son_losses_manifold = 0
endif
endif

h_svv_new = h_uitlaat_svv_new(T_omgewing_db,G,A_col,h_meng_son,P_atm,m_dot_svv_aanvanklik,h_svv_oud,UA_solar_manifold,
m_svv_activate,tydstap_grootte,m_solar_bank)
m_dot_svv = m_dot_svv_aanvanklik*m_svv_activate
h_svv_pype = h_uitlaat_svv_pype(T_omgewing_db,h_svv_oud,P_atm,m_dot_svv,h_svv_pype_oud,UA_solar,tydstap_grootte,m_solar_pype)
)

m_dot_hp_probe = m_dot_hp
COP_hp_potential = Q_ECDH90M_COP(T_omgewing_db,RH, Temperature(Water,P=P_atm,h=h_meng),density_ratio)

ELSE "Nie laai Siklus / Non -charge cycle - validated geen probleem met energie balans"
die_probe = 0
cycle$ = 'Non-Charge'
h_poort_1 = h_munispaliteit
h_meng = h_munispaliteit "Eintlik uitkomstenaar"
h_meng_son = h_munispaliteit
q_probe_hp = Q_ECDH90MT_nuwe(T_omgewing_db,RH, Temperature(Water,P=P_atm,h=h_munispaliteit),T_omgewing_wb,density_ratio,
ashp_ratio)

T_meng = Temperature(Water,P=P_atm,h=h_meng)
if (hp_timer_control_internal >= 1) then
m_dot_hp = hpMassaVloei(h_munispaliteit,q_probe_hp,T_hp,P_atm,activate_heat_pump,m_dot_ashp_max)
else
m_dot_hp = 0
endif
m_dot_hp_probe = m_dot_hp
Q_hp_probe_test = m_dot_hp*(h_hp-h_meng)

h_svv = h_svv_oud {h_uitlaat_svv(T_omgewing_db,G,A_col,h_poort_1,P_atm,m_dot_svv_aanvanklik)}
T_svv = Temperature(Water,P=P_atm,h=h_svv)
T_svv_oud = Temperature(Water,P=P_atm,h=h_svv_oud)
if (T_svv_pype > T_omgewing_db) then
Q_solar_losses = UA_solar * (T_svv_pype- T_omgewing_db)*N_solar {SWH supply pipe losses}
else
Q_solar_losses = 0
endif
eta_svv = eta_svv(T_omgewing_db,G,h_munispaliteit,P_atm)
m_svv_activate = m_svv_activate(h_svv,T_oud_3,T_oud_5,T_swh_max,P_atm,T_delta_hoog,T_delta_laag,m_dot_svv_oud)
Q_son_probe = m_dot_svv_aanvanklik*(h_svv-h_meng_son)*N_solar*m_svv_activate
if (eta_svv > 0) then
Q_son_probe2 = (eta_svv*G*A_col*N_solar)/1000
Q_son_losses_manifold= ((1-eta_svv-((100-(53.125*1.63))/100))*G*A_col*N_solar)/1000 {SWV - inkomende energie}
{SWV - inkomende energie}
{SWV - inkomende energie}
else
Q_son_probe2 = 0
if (T_svv_oud > T_omgewing_db ) then
Q_son_losses_manifold = UA_solar_manifold * (T_svv_oud- T_omgewing_db)*N_solar
```

Appendix L. Computer code (EES): Reduced system simulation

```
else
Q_son_losses_manifold= 0
endif
endif
h_svv_new = h_uitlaat_svv_new(T_omgeving_db,G,A_col,h_meng_son,P_atm,m_dot_svv_aanvanklik,h_svv_oud,UA_solar_manifold,
m_svv_activate,tydstap_grootte,m_solar_bank)
m_dot_svv = m_dot_svv_aanvanklik*m_svv_activate
h_svv_pype = h_uitlaat_svv_pype(T_omgeving_db,h_svv_oud,P_atm,m_dot_svv,h_svv_pype_oud,UA_solar,tydstap_grootte,m_solar_pype)
m_dot_hp_probe = m_dot_hp

COP_hp_potential = Q_ECDH90M_COP(T_omgeving_db,RH,Temperature(Water,P=P_atm,h=h_meng),density_ratio)

ENDIF

"Alle massavloei berekeninge"
"Koue water toevoer wat by die onderkant van die tank invloei"
m_dot_hp_swh = m_dot_in - (m_dot_hp*N_hp)#####[Massavloei vergelyking [10]##### "
m_dot_uit = m_dot_in
m_dot_poort = m_dot_hp_swh - (m_dot_svv*N_solar) #####[Massavloei vergelyking [6]##### " "Die koue water
temperatuur wat onder by die water kan invloei"

m_dot[1] = m_dot_poort + m_dot_return #####[Massavloei vergelyking [1]##### "
M_dot_1 = m_dot[1]
m_dot[2] = m_dot[1] #####[Massavloei vergelyking [2]##### "
M_dot_2 = m_dot[2]

m_dot[3] = m_dot[2] + (m_dot_svv*N_solar) #####[Massavloei vergelyking [3]##### "
M_dot_3 = m_dot[3]

m_dot[4] = m_dot[3] #####[Massavloei vergelyking [4]##### "
M_dot_4 = m_dot[4]

m_dot[5] = m_dot[4] #####[Massavloei vergelyking [5]##### "
M_dot_5 = m_dot[5]

h_svv_nou = h_svv
m_svv_nou = m_dot_svv

if (m_dot_in > 0) then
Q_dot_munisipaliteit = m_dot_in*h_munisipaliteit
else
Q_dot_munisipaliteit = 0
endif
m_dot_ringfeed = (m_dot_hp*N_hp) + m_dot[5] #####[Massavloei vergelyking [7]##### "

"=====
"Energie vergelyking 6"

if(m_dot_poort >= 0) THEN "As dit nie 'n laai siklus is nie sal die verskafde water 'n mengsel wees"

h_ringfeed_top = (m_dot[5]*h_oud_5 + (m_dot_hp*h_hp*N_hp))/m_dot_ringfeed"Geen verliese in die ringmain word in ag geneem"
Q_hello_ashp = (m_dot_hp*(h_hp)*N_hp) + (m_dot[5]*h_oud_5) - (h_ringfeed_top * m_dot_ringfeed )
m_hello_ashp = (m_dot_hp*N_hp) + (m_dot[5]) - (m_dot_ringfeed )
Q_tank_top = -(m_dot[5]*h_oud_5)
h_poort_bo = h_oud_5
diagnose_condition = 11
else
if ((m_dot_poort < 0) AND (m_dot_hp > 0) ) then "As dit nie 'n laai siklus is nie sal die verskafde water hoofsaaklik HP water
wees"
"Maar daar moet wel bepaal word of die vloei op die node 5 inwaarts of uitwaarts is."
if(m_dot[5] < 0) then
h_ringfeed_top = h_hp " ((m_dot[5]*h_hp) + (m_dot_hp*h_hp*2))/m_dot_ringfeed"
h_poort_bo = h_hp {h_oud_5}
diagnose_condition = 221
else
h_ringfeed_top = (m_dot[5]*h_oud_5 + (m_dot_hp*h_hp*N_hp))/m_dot_ringfeed
h_poort_bo = h_oud_5
diagnose_condition = 500
endif

Q_hello_ashp = (m_dot_hp*(h_hp)*N_hp) + (m_dot[5]*h_hp) - (h_ringfeed_top * m_dot_ringfeed )
m_hello_ashp = (m_dot_hp*N_hp) + (m_dot[5]) - (m_dot_ringfeed )
Q_tank_top = -(m_dot[5]*h_hp)

else "Anderste is dit 'n laai siklus waar die HP niks bydra nie want hy is nie aan nie maar die SWH werk dalk"
m_dot_ringfeed = m_dot[5] {+ (m_dot_svv*N_solar)}
h_ringfeed_top = h_oud_5 " ((m_dot[5]*h_hp) + (m_dot_hp*h_hp*2))/m_dot_ringfeed"
h_poort_bo = h_oud_5
Q_hello_ashp = (m_dot_hp*(h_hp)*N_hp) - (m_dot[5]*h_oud_5) + (h_ringfeed_top * m_dot_ringfeed )
m_hello_ashp = (m_dot_hp*N_hp) -(m_dot[5]) + (m_dot_ringfeed)
diagnose_condition = 33
Q_tank_top = -(m_dot[5]*h_oud_5)
endif
endif

T_top_meng = Temperature(Water,P=P_atm,h=h_ringfeed_top)
Q_top = (m_dot_ringfeed)*h_ringfeed_top
Q_losses_d = k_d * l_d * (T_aftap_oud - (T_indoors) ) / 1000[W/kW]"verliese na die omgewing op die las kant"{k_d * l_d * (
T_top_meng - (T_omgeving_db +T_indoors) ) / 1000[W/kW]}
```


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```
Q_losses_dead_end = ((( pi# * Cp(Water,T=T_aftap_oud,P=P_atm) * (d_d^2) * l_dl )/4)* density(Water,T=T_aftap_oud,P=P_atm) *
N_drawoffs * (T_aftap_oud - (T_indoors)))/ tydstep_grootte
((( pi# * Cp(Water,T=T_aftap_oud,P=P_atm) * (d_d^2) * l_dl )/4)* density(Water,T=T_aftap_oud,P=P_atm) * N_drawoffs * (
T_top_meng - (T_omgewing_db+T_indoors)))/ tydstep_grootte}
T_room = T_indoors {(T_omgewing_db+T_indoors)}

h_ringfeed_return = (h_ringfeed_top*(m_dot_return + (m_dot_in/2)) - Q_losses_dead_end - Q_losses_d)/(m_dot_return + (m_dot_in
/2)) {h_ringfeed_aftap - ((Q_losses_d/m_dot_ringfeed)/2)}
h_ringfeed_aftap = (h_ringfeed_top+h_ringfeed_return)/2
T_aftap = Temperature(Water,P=P_atm,h=h_ringfeed_aftap)
T_return = Temperature(Water,P=P_atm,h=h_ringfeed_return)

{T_svv_nou = Temperature(Water,P=P_atm,h=h_svv)}
{T_ringfeed_return = T_return }

"=====

if(m_dot_poort <= 0) THEN "Charge cycle - Laai siklus"

if((m_dot_hp > 0) and (m_dot[5] < 0)) then
kode_blok = 1
"Node 1 se energie"
tyd = teller* tydstep_grootte
digtheid1 = Density(Water,T=T_oud_1,P=P_atm)
digtheid_oud1 = digtheid1
m_1 = h_node*area*digtheid1
Q_return = (m_dot_return*h_ringfeed_return)

{
stratTankEntalp[1]= h_oud_1 +((h_kante*area2*(T_omgewing_db-T_oud_1)+((k_water*area)/h_node)*(T_oud_2 -
T_oud_1)+(m_dot_poort*h_poort_1)+(m_dot_return*h_ringfeed_return)-(m_dot_1*h_oud_1)+(h_oud_2*area*(T_omgewing_db-T_oud_1
)))/(rho_water*area*h_node))* tydstep_grootte}
{stratTankEntalp[1]= h_oud_1 +(((m_dot_poort*h_poort_1)+(m_dot_return*h_ringfeed_return)-(m_dot[1]*h_oud_1))/(m_1))*
tydstep_grootte}

energy_corrector_node1 = upwindFunction(m_dot[1],0,-h_oud_1,-h_oud_2)
konduksie_term[1] = (((k_water*area)/h_node)*(T_oud_2 - T_oud_1))
konveksie_term[1] = (h_kante*area2*(T_omgewing_db-T_oud_1))+(h_oud_2*area*(T_omgewing_db-T_oud_1))+(h_top*area*(T_omgewing_db-
T_oud_1))
tank_pyp_verlies[1] = ((UA_tank_pipes/5)*(T_omgewing_db-T_oud_1))
geforseerdeVloed[1] = (m_dot_poort*h_poort_1)+(m_dot_return*h_ringfeed_return)+(m_dot[1]*energy_corrector_node1)
k_node1_1= ((geforseerdeVloed[1] +konveksie_term[1]+konduksie_term[1] +tank_pyp_verlies[1])/(m_1))* tydstep_grootte
k_node1_1_Tn = Temperature(Water,h=((k_node1_1*0.5)+h_oud_1),P=P_atm)
k_node1_1_T = k_node1_1_Tn

"Node 2 se energie"
digtheid2 = Density(Water,T=T_oud_2,P=P_atm)
digtheid_oud2 = digtheid2
m_2 = h_node*area*digtheid2
{
stratTankEntalp[2] = h_oud_2 +((h_kante*area2*(T_omgewing_db-T_oud_2)+((k_water*area)/h_node)*(T_oud_3 -
T_oud_2)+(k_water*area)/h_node)*(T_oud_1 - T_oud_2)+(m_dot_1*h_oud_1)-(m_dot_2*h_oud_2))/(rho_water*area*h_node))*
tydstep_grootte}
{stratTankEntalp[2] = h_oud_2 +(((m_dot[1]*h_oud_1)-( m_dot[2]*h_oud_2))/(m_2))* tydstep_grootte}

energy_corrector_node2_1 = upwindFunction(m_dot[1],0,h_oud_1,h_oud_2)
energy_corrector_node2_2 = upwindFunction(m_dot[2],0,-h_oud_2,-h_oud_3)

konduksie_term[2] = (((k_water*area)/h_node)*(T_oud_1 - T_oud_2))+((k_water*area)/h_node)*(T_oud_3 - T_oud_2))
konveksie_term[2] = (h_kante*area2*(T_omgewing_db-T_oud_2))+(h_oud_2*area*(T_omgewing_db-T_oud_2))+(h_top*area*(T_omgewing_db-
T_oud_2))
geforseerdeVloed[2] = (m_dot[1]*energy_corrector_node2_1)+( m_dot[2]*energy_corrector_node2_2)
tank_pyp_verlies[2] = ((UA_tank_pipes/5)*(T_omgewing_db-T_oud_2))
k_node2_1= ((geforseerdeVloed[2]+konveksie_term[2]+konduksie_term[2]+tank_pyp_verlies[2])/(m_2))* tydstep_grootte
k_node2_1_Tn = Temperature(Water,h=((k_node2_1*0.5)+h_oud_2),P=P_atm)
{k_node2_1_T = T_oud_2 + k_node2_1_Tn}
k_node2_1_T =k_node2_1_Tn

"Node 3 se energie"
digtheid3 = Density(Water,T=T_oud_3,P=P_atm)
digtheid_oud3 = digtheid3
m_3 = h_node*area*digtheid3
{
stratTankEntalp[3] = h_oud_3 +((h_kante*area2*(T_omgewing_db-T_oud_3)+((k_water*area)/h_node)*(T_oud_4 -
T_oud_3)+(k_water*area)/h_node)*(T_oud_2 - T_oud_3)+(m_dot_2*h_oud_2)-(m_dot_3*h_oud_3)+(m_dot_svv*h_svv))/(rho_water*
area*h_node))* tydstep_grootte}
{stratTankEntalp[3] = h_oud_3 +((( m_dot[2]*h_oud_2)-(m_dot[3]*h_oud_3)+(m_dot_svv*h_svv))/(m_3))* tydstep_grootte}
energy_corrector_node3_1 = upwindFunction(m_dot[2],0,h_oud_2,h_oud_3)
energy_corrector_node3_2 = upwindFunction(m_dot[3],0,-h_oud_3,-h_oud_4)
konduksie_term[3] = (((k_water*area)/h_node)*(T_oud_4 - T_oud_3))+((k_water*area)/h_node)*(T_oud_2 - T_oud_3))
konveksie_term[3] = (h_kante*area2*(T_omgewing_db-T_oud_3))+(h_oud_2*area*(T_omgewing_db-T_oud_2))+(h_top*area*(T_omgewing_db-
T_oud_2))
geforseerdeVloed[3] = ( m_dot[2]*energy_corrector_node3_1)+(m_dot[3]*energy_corrector_node3_2)+(m_dot_svv*h_svv_pype_oud*
N_solar)
tank_pyp_verlies[3] = ((UA_tank_pipes/5)*(T_omgewing_db-T_oud_3))
k_node3_1 = ((geforseerdeVloed[3]+konveksie_term[3]+konduksie_term[3]+tank_pyp_verlies[3])/(m_3))* tydstep_grootte
k_node3_1_Tn = Temperature(Water,h=((k_node3_1*0.5)+h_oud_3),P=P_atm)
{k_node3_1_T = T_oud_3 + k_node3_1_Tn}
k_node3_1_T = k_node3_1_Tn

"Node 4 se energie"
digtheid4 = Density(Water,T=T_oud_4,P=P_atm)
cp_water_4 =Cp(Water,T=T_oud_4,P=P_atm)
digtheid_oud4 = digtheid4
m_4 = h_node*area*digtheid4
```

Appendix L. Computer code (EES): Reduced system simulation

```
q_aux_4 = Q_aux(T_set_aux, T_oud_4, q_element, cp_water_4, m_4, tydstop_grootte, electrical_element, m_dot[3], h_oud_4, -m_dot[4],
h_oud_5, h_kante, area2, T_omgewing_db, 0, area, T_oud_3, T_oud_5, h_node, k_water)
{
    stratTankEntalp[4] = h_oud_4 + ((h_kante*area2*(T_omgewing_db-T_oud_4) + ((k_water*area)/h_node)*(T_oud_5 -
T_oud_4) + ((k_water*area)/h_node)*(T_oud_3 - T_oud_4) + (m_dot[3]*h_oud_3) - (m_dot[4]*h_oud_4) + q_aux_4) / (rho_water*area*h_node
))* tydstop_grootte
}
stratTankEntalp[4] = h_oud_4 + (((m_dot[3]*h_oud_3) - (m_dot[4]*h_oud_4) + q_aux_4) / (m_4)) * tydstop_grootte
konduksie_term[4] = (((k_water*area)/h_node)*(T_oud_5 - T_oud_4)) + (((k_water*area)/h_node)*(T_oud_3 - T_oud_4))
konveksie_term[4] = (h_kante*area2*(T_omgewing_db-T_oud_4)) + (h_oud_4*area*(T_omgewing_db-T_oud_4)) + (h_top*area*(T_omgewing_db-
T_oud_4))
geforseerdeVloed[4] = (m_dot[3]*h_oud_4) - (m_dot[4]*h_oud_5)
tank_pyp_verlies[4] = ((UA_tank_pipes/5)*(T_omgewing_db-T_oud_4))
k_node4_1 = ((geforseerdeVloed[4] + q_aux_4 + konveksie_term[4] + konduksie_term[4] + tank_pyp_verlies[4]) / (m_4)) * tydstop_grootte
k_node4_1_Tn = Temperature(Water, h=((k_node4_1*0.5)+h_oud_4), P=P_atm)
{k_node4_1_T = T_oud_4 + k_node4_1_Tn}
k_node4_1_T = k_node4_1_Tn

"Node 5 se energie"
{m_dot_ringfeed = m_dot[5] + (m_dot_hp*2)}
digtheid5 = Density(Water, T=T_oud_5, P=P_atm)
cp_water_5 = Cp(Water, T=T_oud_5, P=P_atm)
m_5 = h_node*area*digtheid5
q_aux_5 = Q_aux(T_set_aux, T_oud_5, q_element, cp_water_5, m_5, tydstop_grootte, electrical_element, m_dot[4], h_oud_5, -m_dot[5],
h_poort_bo, h_kante, area2, T_omgewing_db, h_top, area, T_oud_4, T_oud_5, h_node, k_water)
{q_aux_1 = Q_aux(T_set_aux, T_oud_5, q_element, cp_water_5, m_5, tydstop_grootte)}
{T_meng = ((T_hp*m_dot_hp) + (T_oud_5 * m_dot[5])) / m_dot_ringfeed}
{
    stratTankEntalp[5] = h_oud_5 + ((h_kante*area2*(T_omgewing_db-T_oud_5) + ((k_water*area)/h_node)*(T_oud_4 -
T_oud_5) + (m_dot[4]*h_oud_4) - (m_dot[5]*h_oud_5) + (h_top*area*(T_omgewing_db-T_oud_5)) + q_aux_5) / (rho_water*area*h_node)) *
tydstop_grootte
}
stratTankEntalp[5] = h_oud_5 + (((m_dot[4]*h_oud_4) - (m_dot[5]*h_oud_5) + q_aux_5) / (m_5)) * tydstop_grootte
konduksie_term[5] = (((k_water*area)/h_node)*(T_oud_4 - T_oud_5))
geforseerdeVloed[5] = (m_dot[4]*h_oud_5) - (m_dot[5]*h_poort_bo)
konveksie_term[5] = (h_kante*area2*(T_omgewing_db-T_oud_5)) + (h_oud_5*area*(T_omgewing_db-T_oud_5)) + (h_top*area*(T_omgewing_db-
T_oud_5))
tank_pyp_verlies[5] = ((UA_tank_pipes/5)*(T_omgewing_db-T_oud_5))
k_node5_1 = (konduksie_term[5] + q_aux_5 + konveksie_term[5] + geforseerdeVloed[5] + tank_pyp_verlies[5]) / (m_5) *
tydstop_grootte
k_node5_1_Tn = Temperature(Water, h=((k_node5_1*0.5)+h_oud_5), P=P_atm)
{k_node5_1_T = T_oud_5 + k_node5_1_Tn}
k_node5_1_T = k_node5_1_Tn

k_node1_2 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node1_1_T)) + (m_dot_poort*h_poort_1) + (m_dot_return*h_ringfeed_return) + (m_dot
[1]*(energy_corrector_node1 + (0.5*k_node2_1))) + (h_kante*area2*(T_omgewing_db-((k_node1_1_T)))) + (h_oud_5*area*(
T_omgewing_db-((k_node1_1_T)))) + (h_top*area*(T_omgewing_db-((k_node1_1_T)))) + (((k_water*area)/h_node/2))*(((k_node2_1_T)
- ((k_node1_1_T)))) / (m_1/2)) * tydstop_grootte
k_node2_2 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node2_1_T)) + (m_dot[1]*(energy_corrector_node2_1 + (0.5*k_node2_1))) + (m_dot
[2]*(energy_corrector_node2_2 + (0.5*k_node2_1))) + (h_kante*area2*(T_omgewing_db-((k_node2_1_T)))) + (h_top*area*(
T_omgewing_db-((k_node2_1_T)))) + (h_oud_5*area*(T_omgewing_db-((k_node2_1_T)))) + (((k_water*area)/h_node/2))*(((k_node3_1_T)
- ((k_node2_1_T)))) + (((k_water*area)/h_node/2))*(((k_node1_1_T) - ((k_node2_1_T)))) / (m_2/2)) *
tydstop_grootte
k_node3_2 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node3_1_T)) + (m_dot[2]*(energy_corrector_node3_1 + (0.5*k_node3_1))) + (m_dot
[3]*(energy_corrector_node3_2 + (0.5*k_node4_1))) + (m_dot_svw*h_svw_pype_oud*N_solar) + (h_kante*area2*(T_omgewing_db-((
k_node3_1_T)))) + (h_top*area*(T_omgewing_db-((k_node3_1_T)))) + (h_oud_5*area*(T_omgewing_db-((k_node3_1_T)))) + (((k_water*
area)/h_node/2))*(((k_node4_1_T) - ((k_node3_1_T)))) + (((k_water*area)/h_node/2))*(((k_node2_1_T) - ((k_node3_1_T))))
)/ (m_3/2)) * tydstop_grootte
k_node4_2 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node4_1_T)) + (m_dot[3]*(h_oud_4 + (0.5*k_node4_1))) - (m_dot[4]*(h_oud_5 + (0.5*
k_node5_1))) + q_aux_4 + (h_kante*area2*(T_omgewing_db-((k_node4_1_T)))) + (h_top*area*(T_omgewing_db-((k_node4_1_T)))) + (
h_oud_5*area*(T_omgewing_db-((k_node4_1_T)))) + (((k_water*area)/h_node/2))*(((k_node5_1_T) - ((k_node4_1_T)))) + (((k_wa
ter*area)/h_node/2))*(((k_node3_1_T) - ((k_node4_1_T)))) / (m_4/2)) * tydstop_grootte
k_node5_2 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node5_1_T)) + (m_dot[4]*(h_oud_5 + (0.5*k_node5_1))) - (m_dot[5]*(h_poort_bo)) +
q_aux_5 + (h_kante*area2*(T_omgewing_db-((k_node5_1_T)))) + (h_top*area*(T_omgewing_db-((k_node5_1_T)))) + (h_oud_5*area*(
T_omgewing_db-((k_node5_1_T)))) + (((k_water*area)/h_node/2))*(((k_node4_1_T) - ((k_node5_1_T)))) / (m_5/2)) *
tydstop_grootte

k_node1_2_Tn = Temperature(Water, h=((h_oud_1 + (0.5*k_node1_2))), P=P_atm) "Verandering in entalpie"
{k_node1_2_T = T_oud_1 + k_node1_2_Tn}
k_node1_2_T = k_node1_2_Tn
k_node2_2_Tn = Temperature(Water, h=((h_oud_2 + (0.5*k_node2_2))), P=P_atm)
{k_node2_2_T = T_oud_2 + k_node2_2_Tn}
k_node2_2_T = k_node2_2_Tn
k_node3_2_Tn = Temperature(Water, h=((h_oud_3 + (0.5*k_node3_2))), P=P_atm)
{k_node3_2_T = T_oud_3 + k_node3_2_Tn}
k_node3_2_T = k_node3_2_Tn
k_node4_2_Tn = Temperature(Water, h=((h_oud_4 + (0.5*k_node4_2))), P=P_atm)
{k_node4_2_T = T_oud_4 + k_node4_2_Tn}
k_node4_2_T = k_node4_2_Tn
k_node5_2_Tn = Temperature(Water, h=((h_oud_5 + (0.5*k_node5_2))), P=P_atm)
{k_node5_2_T = T_oud_5 + k_node5_2_Tn}
k_node5_2_T = k_node5_2_Tn

k_node1_3 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node1_2_T)) + (m_dot_poort*h_poort_1) + (m_dot_return*h_ringfeed_return) + (m_dot
[1]*(energy_corrector_node1 + (0.5*k_node2_2))) + (h_kante*area2*(T_omgewing_db-((k_node1_2_T)))) + (h_oud_5*area*(
T_omgewing_db-((k_node1_2_T)))) + (h_top*area*(T_omgewing_db-((k_node1_2_T)))) + (((k_water*area)/h_node/2))*(((k_node2_2_T)
- ((k_node1_2_T)))) / (m_1/2)) * tydstop_grootte
k_node2_3 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node2_2_T)) + (m_dot[1]*(energy_corrector_node2_1 + (0.5*k_node2_2))) + (m_dot
[2]*(energy_corrector_node2_2 + (0.5*k_node3_2))) + (h_kante*area2*(T_omgewing_db-((k_node2_2_T)))) + (((k_water*area)/h_node
/2))*(((k_node3_2_T) - ((k_node2_2_T)))) + (h_oud_5*area*(T_omgewing_db-((k_node2_2_T)))) + (h_top*area*(T_omgewing_db-((
k_node2_2_T)))) + (((k_water*area)/h_node/2))*(((k_node1_2_T) - ((k_node2_2_T)))) / (m_2/2)) * tydstop_grootte
k_node3_3 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node3_2_T)) + (m_dot[2]*(energy_corrector_node3_1 + (0.5*k_node3_2))) + (m_dot
[3]*(energy_corrector_node3_2 + (0.5*k_node4_2))) + (m_dot_svw*h_svw_pype_oud*N_solar) + (h_kante*area2*(T_omgewing_db-((
k_node3_2_T)))) + (h_oud_5*area*(T_omgewing_db-((k_node3_2_T)))) + (h_top*area*(T_omgewing_db-((k_node3_2_T)))) + (((k_water*
area)/h_node/2))*(((k_node4_2_T) - ((k_node3_2_T)))) + (((k_water*area)/h_node/2))*(((k_node2_2_T) - ((k_node3_2_T))))
)/ (m_3/2)) * tydstop_grootte
```

Appendix L. Computer code (EES): Reduced system simulation

```
k_node4_3 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node4_2_T)))+(m_dot[3]*(h_oud_4+(0.5*k_node4_2_T)))-(m_dot[4]*(h_oud_5+(0.5*
k_node5_2_T)))+(q_aux_4+(h_kante*area2*(T_omgewing_db-((k_node4_2_T))))+(((k_water*area)/(h_node/2))*((k_node5_2_T)) - ((
k_node4_2_T))))+(h_onder*area*(T_omgewing_db-((k_node4_2_T))))+(h_top*area*(T_omgewing_db-((k_node4_2_T))))+(((k_water*
area)/(h_node/2))*((k_node3_2_T)) - ((k_node4_2_T))))/(m_4/2)* tydstap_grootte
k_node5_3 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node5_2_T)))+(m_dot[4]*(h_oud_5+(0.5*k_node5_2_T)))-(m_dot[5]*(h_poort_bo)) +
q_aux_5+(h_kante*area2*(T_omgewing_db-((k_node5_2_T))))+(((h_top*area*(T_omgewing_db-((k_node5_2_T))))+(h_onder*area*(
T_omgewing_db-((k_node5_2_T))))+(((k_water*area)/(h_node/2))*((k_node4_2_T)) - ((k_node5_2_T))))/(m_5/2)*
tydstap_grootte

k_node1_3_Tn = Temperature(Water,h=((h_oud_1+(k_node1_3))),P=P_atm) "Verandering in entalpie"
{k_node1_3_T = T_oud_1 + k_node1_3_Tn}
k_node1_3_T = k_node1_3_Tn
k_node2_3_Tn = Temperature(Water,h=((h_oud_2+(k_node2_3))),P=P_atm)
{k_node2_3_T = T_oud_2 + k_node2_3_Tn}
k_node2_3_T = k_node2_3_Tn
k_node3_3_Tn = Temperature(Water,h=((h_oud_3+(k_node3_3))),P=P_atm)
{k_node3_3_T = T_oud_3 + k_node3_3_Tn}
k_node3_3_T = k_node3_3_Tn
k_node4_3_Tn = Temperature(Water,h=((h_oud_4+(k_node4_3))),P=P_atm)
{k_node4_3_T = T_oud_4 + k_node4_3_Tn}
k_node4_3_T = k_node4_3_Tn
k_node5_3_Tn = Temperature(Water,h=((h_oud_5+(k_node5_3))),P=P_atm)
{k_node5_3_T = T_oud_5 + k_node5_3_Tn}
k_node5_3_T = k_node5_3_Tn

k_node1_4= (((UA_tank_pipes/5)*(T_omgewing_db-k_node1_3_T)))+(m_dot_poort*h_poort_1)+(m_dot_return*h_ringfeed_return)+(m_dot
[1]*(energy_corrector_node1+(k_node2_3)))+(h_kante*area2*(T_omgewing_db-((k_node1_3_T))))+(h_onder*area*(T_omgewing_db
-((k_node1_3_T))))+(h_top*area*(T_omgewing_db-((k_node1_3_T))))+(((k_water*area)/(2*h_node))*((k_node2_3_T)) - ((
k_node1_3_T))))/(2*m_1)* tydstap_grootte
k_node2_4= (((UA_tank_pipes/5)*(T_omgewing_db-k_node2_3_T)))+(m_dot[1]*(energy_corrector_node2_1+(k_node2_3)))+(m_dot[2]*(
energy_corrector_node2_2+(k_node3_3)))+(h_kante*area2*(T_omgewing_db-((k_node2_3_T))))+(((k_water*area)/(h_node))*((
k_node3_3_T)) - ((k_node2_3_T))))+(h_onder*area*(T_omgewing_db-((k_node2_3_T))))+(h_top*area*(T_omgewing_db-((
k_node2_3_T))))+(((k_water*area)/(h_node*2))*((k_node1_3_T)) - ((k_node2_3_T))))/(2*m_2)* tydstap_grootte
k_node3_4 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node3_3_T)))+(m_dot[2]*(energy_corrector_node3_1+(k_node3_3)))+(m_dot[3]*(
energy_corrector_node3_2+(k_node4_3)))+(m_dot_sww*h_sww_pype_oud*N_solar)+(h_kante*area2*(T_omgewing_db-((k_node3_3_T)))
)+(h_onder*area*(T_omgewing_db-((k_node3_3_T))))+(h_top*area*(T_omgewing_db-((k_node3_3_T))))+(((k_water*area)/(h_node
*2))*((k_node4_3_T)) - ((k_node1_3_T))))+(((k_water*area)/(h_node*2))*((k_node2_3_T)) - ((k_node3_3_T))))/(m_3*2)*
tydstap_grootte
k_node4_4 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node4_3_T)))+(m_dot[3]*(h_oud_4+(k_node4_3_T)))-(m_dot[4]*(h_oud_5+(k_node5_3_T))
+q_aux_4+(h_kante*area2*(T_omgewing_db-((k_node4_3_T))))+(((k_water*area)/(h_node*2))*((k_node5_3_T)) - ((k_node4_3_T))
)))+(h_onder*area*(T_omgewing_db-((k_node4_3_T))))+(h_top*area*(T_omgewing_db-((k_node4_3_T))))+(((k_water*area)/(h_node
*2))*((k_node3_3_T)) - ((k_node4_3_T))))/(m_4*2)* tydstap_grootte
k_node5_4 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node5_3_T)))+(m_dot[4]*(h_oud_5+(k_node5_3_T)))-(m_dot[5]*(h_poort_bo))+q_aux_5
+(h_kante*area2*(T_omgewing_db-((k_node5_3_T))))+(((h_top*area*(T_omgewing_db-((k_node5_3_T))))+(h_onder*area*(
T_omgewing_db-((k_node5_3_T))))+(((k_water*area)/(h_node*2))*((k_node4_3_T)) - ((k_node5_3_T))))/(m_5*2)*
tydstap_grootte

stratTankEntalp[1]= h_oud_1 + (1/6*(k_node1_1+(2*k_node1_2)+(2*k_node1_3))+k_node1_4)
stratTankEntalp[2]= h_oud_2 + (1/6*(k_node2_1+(2*k_node2_2)+(2*k_node2_3))+k_node2_4)
stratTankEntalp[3]= h_oud_3 + (1/6*(k_node3_1+(2*k_node3_2)+(2*k_node3_3))+k_node3_4)
stratTankEntalp[4]= h_oud_4 + (1/6*(k_node4_1+(2*k_node4_2)+(2*k_node4_3))+k_node4_4)
stratTankEntalp[5]= h_oud_5 + (1/6*(k_node5_1+(2*k_node5_2)+(2*k_node5_3))+k_node5_4)

"=====
"Bereken die temperatuur van die nuut berekende entalpie"
stratifikasieTank[1] = Temperature(Water,h=stratTankEntalp[1],P=P_atm)
stratifikasieTank[2] = Temperature(Water,h=stratTankEntalp[2],P=P_atm)
stratifikasieTank[3] = Temperature(Water,h=stratTankEntalp[3],P=P_atm)
stratifikasieTank[4] = Temperature(Water,h=stratTankEntalp[4],P=P_atm)
stratifikasieTank[5] = Temperature(Water,h=stratTankEntalp[5],P=P_atm)

"=====

else
"Node 1 se energie"
kode_blok = 2
tyd = teller* tydstap_grootte
digtheid1 = Density(Water,T=T_oud_1,P=P_atm)
digtheid_oud1 = digtheid1
m_1 = h_node*area*digtheid1
Q_return = (m_dot_return*h_ringfeed_return)

corrector = maalFaktor(m_dot[2],0)

{
    stratTankEntalp[1]= h_oud_1 + ((h_kante*area2*(T_omgewing_db-T_oud_1)+(k_water*area)/(h_node)*(T_oud_2 -
T_oud_1)+(m_dot_poort*h_poort_1)+(m_dot_return*h_ringfeed_return)-(m_dot_1*h_oud_1)+(h_onder*area*(T_omgewing_db-T_oud_1
)))/(rho_water*area*h_node))* tydstap_grootte}
{stratTankEntalp[1]= h_oud_1 +(((m_dot_poort*h_poort_1)+(m_dot_return*h_ringfeed_return)-(m_dot[1]*h_oud_1))/(m_1))*
tydstap_grootte}

energy_corrector_node1 = upwindFunction(m_dot[1],0,-h_oud_1,-h_oud_2)
konduksie_term[1] = (((k_water*area)/(h_node)*(T_oud_2 - T_oud_1))
konveksie_term[1] = (h_kante*area2*(T_omgewing_db-T_oud_1))+(h_onder*area*(T_omgewing_db-T_oud_1))+(h_top*area*(T_omgewing_db-
T_oud_1))
tank_pyp_verlies[1] = ((UA_tank_pipes/5)*(T_omgewing_db-T_oud_1))
geforseerdeVloed[1] = (m_dot_poort*h_oud_1)+(m_dot_return*h_ringfeed_return)+(m_dot[1]*energy_corrector_node1)
k_node1_1= ((geforseerdeVloed[1] +konveksie_term[1]+konduksie_term[1])/(m_1))* tydstap_grootte
k_node1_1_Tn = Temperature(Water,h=((k_node1_1*0.5)+h_oud_1),P=P_atm)
k_node1_1_T = k_node1_1_Tn

"Node 2 se energie"
```

Appendix L. Computer code (EES): Reduced system simulation

```
digtheid2 = Density (Water ,T=T_oud_2 ,P=P_atm)
digtheid_oud2 = digtheid2
m_2 = h_node*area*digtheid2
{
    stratTankEntalp [2] = h_oud_2 +((h_kante*area2*(T_omgewing_db-T_oud_2)+((k_water*area)/h_node)*(T_oud_3 -
    T_oud_2)+((k_water*area)/h_node)*(T_oud_1 - T_oud_2)+(m_dot_1*h_oud_1)-(m_dot_2*h_oud_2))/(rho_water*area*h_node))*
    tydstep_grootte
}
{stratTankEntalp [2] = h_oud_2 +(((m_dot[1]*h_oud_1)-( m_dot[2]*h_oud_2))/(m_2))* tydstep_grootte}
energy_corrector_node2_1 = upwindFunction(m_dot[1],0,h_oud_1,h_oud_2)
energy_corrector_node2_2 = upwindFunction(m_dot[2],0,-h_oud_2,-h_oud_3)
konduksie_term [2] = (((k_water*area)/h_node)*(T_oud_1 - T_oud_2))+(((k_water*area)/h_node)*(T_oud_3 - T_oud_2))
konveksie_term [2] = (h_kante*area2*(T_omgewing_db-T_oud_2))+(h_ouder*area*(T_omgewing_db-T_oud_2))+(h_top*area*(T_omgewing_db-
T_oud_2))
geforseerdeVloed [2] = (m_dot[1]*energy_corrector_node2_1 )+( m_dot[2]*energy_corrector_node2_2 )
tank_pyp_verlies [2] = ((UA_tank_pipes/5)*(T_omgewing_db-T_oud_2))
k_node2_1= ((geforseerdeVloed [2]+konveksie_term [2]+konduksie_term [2]+tank_pyp_verlies [2])/(m_2))* tydstep_grootte
k_node2_1_Tn = Temperature (Water ,h=((k_node2_1*0.5)+h_oud_2 ),P=P_atm)
{k_node2_1_T = T_oud_2 + k_node2_1_Tn}
k_node2_1_T =k_node2_1_Tn

"Node 3 se energie"
digtheid3 = Density (Water ,T=T_oud_3 ,P=P_atm)
digtheid_oud3 = digtheid3
m_3 = h_node*area*digtheid3
{
    stratTankEntalp [3] = h_oud_3 +((h_kante*area2*(T_omgewing_db-T_oud_3)+((k_water*area)/h_node)*(T_oud_4 -
    T_oud_3)+((k_water*area)/h_node)*(T_oud_2 - T_oud_3)+(m_dot_2*h_oud_2)-(m_dot_3*h_oud_3)+(m_dot_svw*h_svw))/(rho_water*
    area*h_node))* tydstep_grootte
}
{stratTankEntalp [3] = h_oud_3 +((( m_dot[2]*h_oud_2)-(m_dot[3]*h_oud_3)+(m_dot_svw*h_svw))/(m_3))* tydstep_grootte}
energy_corrector_node3_1 = upwindFunction(m_dot[2],0,h_oud_2,h_oud_3)
energy_corrector_node3_2 = upwindFunction(m_dot[3],0,-h_oud_3,-h_oud_4)
konduksie_term [3] = (((k_water*area)/h_node)*(T_oud_4 - T_oud_3))+(((k_water*area)/h_node)*(T_oud_2 - T_oud_3))
konveksie_term [3] = (h_kante*area2*(T_omgewing_db-T_oud_3))+(h_ouder*area*(T_omgewing_db-T_oud_2))+(h_top*area*(T_omgewing_db-
T_oud_2))
geforseerdeVloed [3] = ( m_dot[2]*energy_corrector_node3_1)+(m_dot[3]*energy_corrector_node3_2)+(m_dot_svw*h_svw_pype_oud*
N_solar)
tank_pyp_verlies [3] = ((UA_tank_pipes/5)*(T_omgewing_db-T_oud_3))
k_node3_1 = ((geforseerdeVloed [3]+konveksie_term [3]+konduksie_term [3]+tank_pyp_verlies [3])/(m_3))* tydstep_grootte
k_node3_1_Tn = Temperature (Water ,h=((k_node3_1*0.5)+h_oud_3 ),P=P_atm)
{k_node3_1_T = T_oud_3 + k_node3_1_Tn}
k_node3_1_T = k_node3_1_Tn

"Node 4 se energie"
digtheid4 = Density (Water ,T=T_oud_4 ,P=P_atm)
cp_water_4 =Cp(Water ,T=T_oud_4 ,P=P_atm)
digtheid_oud4 = digtheid4
m_4 = h_node*area*digtheid4
q_aux_4 = Q_aux(T_set_aux , T_oud_4 , q_element , cp_water_4 , m_4 , tydstep_grootte ,electrical_element ,m_dot[3],h_oud_4,-m_dot[4] ,
h_oud_5,h_kante,area2,T_omgewing_db,0,area,T_oud_3,T_oud_5,h_node,k_water)
{
    stratTankEntalp [4] = h_oud_4 +((h_kante*area2*(T_omgewing_db-T_oud_4)+((k_water*area)/h_node)*(T_oud_5 -
    T_oud_4)+((k_water*area)/h_node)*(T_oud_3 - T_oud_4)+(m_dot_3*h_oud_3)-(m_dot_4*h_oud_4)+q_aux_4)/(rho_water*area*h_node
    ))* tydstep_grootte
}
{stratTankEntalp [4] = h_oud_4 +(((m_dot[3]*h_oud_3)-(m_dot[4]*h_oud_4)+q_aux_4)/(m_4))* tydstep_grootte}

energy_corrector_node4_1 = upwindFunction(m_dot[3],0,h_oud_3,h_oud_4)
energy_corrector_node4_2 = upwindFunction(m_dot[4],0,-h_oud_4,-h_oud_5)
konduksie_term [4] = (((k_water*area)/h_node)*(T_oud_5 - T_oud_4))+(((k_water*area)/h_node)*(T_oud_3 - T_oud_4))
konveksie_term [4] = (h_kante*area2*(T_omgewing_db-T_oud_4))+(h_ouder*area*(T_omgewing_db-T_oud_4))+(h_top*area*(T_omgewing_db-
T_oud_4))
geforseerdeVloed [4] = (m_dot[3]*energy_corrector_node4_1)+(m_dot[4]*energy_corrector_node4_2)
tank_pyp_verlies [4] = ((UA_tank_pipes/5)*(T_omgewing_db-T_oud_4))
k_node4_1 =((geforseerdeVloed [4] +q_aux_4+konveksie_term [4]+konduksie_term [4]+tank_pyp_verlies [4])/(m_4))* tydstep_grootte
k_node4_1_Tn = Temperature (Water ,h=((k_node4_1*0.5)+h_oud_4 ),P=P_atm)
{k_node4_1_T = T_oud_4 + k_node4_1_Tn}
k_node4_1_T = k_node4_1_Tn

"Node 5 se energie"
{m_dot_ringfeed = m_dot[5] + (m_dot_hp*2)}
digtheid5 = Density (Water ,T=T_oud_5 ,P=P_atm)
cp_water_5 =Cp(Water ,T=T_oud_5 ,P=P_atm)
m_5 = h_node*area*digtheid5
q_aux_5 = Q_aux(T_set_aux , T_oud_5 , q_element , cp_water_5 , m_5 , tydstep_grootte ,electrical_element ,m_dot[4],h_oud_5,-m_dot[5] ,
h_poort_bo,h_kante,area2,T_omgewing_db,h_top,area,T_oud_4,T_oud_5,h_node,k_water)
{q_aux_1 = Q_aux(T_set_aux , T_oud_5 , q_element , cp_water_5 , m_5 , tydstep_grootte)}
{T_meng = ((T_hp*m_dot_hp)+(T_oud_5 * m_dot[5]))/m_dot_ringfeed}
{
    stratTankEntalp [5] = h_oud_5 +((h_kante*area2*(T_omgewing_db-T_oud_5)+((k_water*area)/h_node)*(T_oud_4 -
    T_oud_5)+(m_dot_4*h_oud_4)-(m_dot_5*h_oud_5)+(h_top*area*(T_omgewing_db-T_oud_5))+q_aux_5)/(rho_water*area*h_node))*
    tydstep_grootte
}
{stratTankEntalp [5] = h_oud_5 +(((m_dot[4]*h_oud_4)-(m_dot[5]*h_oud_5)+q_aux_5)/(m_5))* tydstep_grootte}
konduksie_term [5] = (((k_water*area)/h_node)*(T_oud_4 - T_oud_5))
geforseerdeVloed [5] = (m_dot[4]*h_oud_4)-(m_dot[5]*h_oud_5)
konveksie_term [5] = (h_kante*area2*(T_omgewing_db-T_oud_5))+(h_ouder*area*(T_omgewing_db-T_oud_5))+(h_top*area*(T_omgewing_db-
T_oud_5))
tank_pyp_verlies [5] = ((UA_tank_pipes/5)*(T_omgewing_db-T_oud_5))
k_node5_1 = (konduksie_term [5] + q_aux_5 + konveksie_term [5] + geforseerdeVloed [5]+tank_pyp_verlies [5] )/(m_5)*
tydstep_grootte
k_node5_1_Tn = Temperature (Water ,h=((k_node5_1*0.5)+h_oud_5 ),P=P_atm)
{k_node5_1_T = T_oud_5 + k_node5_1_Tn}
k_node5_1_T = k_node5_1_Tn

k_node1_2= (((UA_tank_pipes/5)*(T_omgewing_db-k_node1_1_T)))+(m_dot_poort*(h_oud_1+(0.5*k_node1_1)))+(m_dot_return*
h_ringfeed_return)+(m_dot[1]*(energy_corrector_node1+(0.5*k_node1_1)))+(h_kante*area2*(T_omgewing_db-((k_node1_1_T))))+(
h_ouder*area*(T_omgewing_db-((k_node1_1_T))))+(h_top*area*(T_omgewing_db-((k_node1_1_T))))+(((k_water*area)/(h_node/2))
*(((k_node2_1_T) - ((k_node1_1_T)))))/(m_1/2))* tydstep_grootte
```

Appendix L. Computer code (EES): Reduced system simulation

```

k_node2_1= (((UA_tank_pipes/5)*(T_omgewing_db-k_node2_1_T)))+(m_dot[1]*(energy_corrector_node2_1+(0.5*k_node2_1)))+(m_dot
[2]*(energy_corrector_node2_2+(0.5*k_node2_1)))+(h_kante*area2*(T_omgewing_db-((k_node2_1_T)))+(h_top*area*(
T_omgewing_db-((k_node2_1_T)))+(h_onder*area*(T_omgewing_db-((k_node2_1_T)))+( ((k_water*area)/(h_node/2))*(((
k_node3_1_T) - ((k_node2_1_T))))+(((k_water*area)/(h_node/2))*(((k_node1_1_T) - ((k_node2_1_T)))))/(m2/2))*
tydstap_grootte
k_node3_2 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node3_1_T)))+( m_dot[2]*(energy_corrector_node3_1+(0.5*k_node2_1)))+(m_dot
[3]*(energy_corrector_node3_2+(0.5*k_node3_1)))+(m_dot_svv*h_svv_pype_oud*N_solar)+(h_kante*area2*(T_omgewing_db-((
k_node3_1_T)))+(h_top*area*(T_omgewing_db-((k_node3_1_T)))+(h_onder*area*(T_omgewing_db-((k_node3_1_T)))+( ((k_water*
area)/(h_node/2))*(((k_node4_1_T) - ((k_node3_1_T))))+(((k_water*area)/(h_node/2))*(((k_node2_1_T) - ((k_node3_1_T))))
)/(m3/2))* tydstap_grootte
k_node4_2 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node4_1_T)))+(m_dot[3]*(energy_corrector_node4_1+(0.5*k_node3_1)))+(m_dot[4]*(
energy_corrector_node4_2+(0.5*k_node4_1)))+(q_aux_4+(h_kante*area2*(T_omgewing_db-((k_node4_1_T)))+(h_top*area*(
T_omgewing_db-((k_node4_1_T)))+(h_onder*area*(T_omgewing_db-((k_node4_1_T)))+( ((k_water*area)/(h_node/2))*(((
k_node5_1_T) - ((k_node4_1_T))))+(((k_water*area)/(h_node/2))*(((k_node3_1_T) - ((k_node4_1_T)))))/(m4/2))*
tydstap_grootte
k_node5_2 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node5_1_T)))+(m_dot[4]*(h_oud_4+(0.5*k_node4_1)))-(m_dot[5]*(h_oud_5+(0.5*
k_node5_1_T)))+(q_aux_5+(h_kante*area2*(T_omgewing_db-((k_node5_1_T)))+(h_top*area*(T_omgewing_db-((k_node5_1_T)))+(
h_onder*area*(T_omgewing_db-((k_node5_1_T)))+( ((k_water*area)/(h_node/2))*(((k_node4_1_T) - ((k_node5_1_T)))))/(m5/2)
))* tydstap_grootte

k_node1_2_Tn = Temperature(Water,h=((h_oud_1+(0.5*k_node1_2))),P=P_atm) "Verandering in entalpie"
{k_node1_2_T = T_oud_1 + k_node1_2_Tn}
k_node1_2_T = k_node1_2_Tn
k_node2_2_Tn = Temperature(Water,h=((h_oud_2+(0.5*k_node2_2))),P=P_atm)
{k_node2_2_T = T_oud_2 + k_node2_2_Tn}
k_node2_2_T = k_node2_2_Tn
k_node3_2_Tn = Temperature(Water,h=((h_oud_3+(0.5*k_node3_2))),P=P_atm)
{k_node3_2_T = T_oud_3 + k_node3_2_Tn}
k_node3_2_T = k_node3_2_Tn
k_node4_2_Tn = Temperature(Water,h=((h_oud_4+(0.5*k_node4_2))),P=P_atm)
{k_node4_2_T = T_oud_4 + k_node4_2_Tn}
k_node4_2_T = k_node4_2_Tn
k_node5_2_Tn = Temperature(Water,h=((h_oud_5+(0.5*k_node5_2))),P=P_atm)
{k_node5_2_T = T_oud_5 + k_node5_2_Tn}
k_node5_2_T = k_node5_2_Tn

k_node1_3= (((UA_tank_pipes/5)*(T_omgewing_db-k_node1_2_T)))+(m_dot_poort*(h_oud_1+(0.5*k_node1_2)))+(m_dot_return*
h_ringfeed_return)+(m_dot[1]*(energy_corrector_node1+(0.5*k_node1_2)))+(h_kante*area2*(T_omgewing_db-((k_node1_2_T)))+(
h_onder*area*(T_omgewing_db-((k_node1_2_T)))+(h_top*area*(T_omgewing_db-((k_node1_2_T)))+( ((k_water*area)/(h_node/2)
))*(((k_node2_2_T) - ((k_node1_2_T))))/(m1/2))* tydstap_grootte
k_node2_3= (((UA_tank_pipes/5)*(T_omgewing_db-k_node2_2_T)))+(m_dot[1]*(energy_corrector_node2_1+(0.5*k_node1_2)))+(m_dot
[2]*(energy_corrector_node2_2+(0.5*k_node2_2)))+(h_kante*area2*(T_omgewing_db-((k_node2_2_T)))+( ((k_water*area)/(h_node
/2))*(((k_node2_2_T) - ((k_node2_2_T))))+(h_onder*area*(T_omgewing_db-((k_node2_2_T)))+(h_top*area*(T_omgewing_db-((
k_node2_2_T)))+( ((k_water*area)/(h_node/2))*(((k_node1_2_T) - ((k_node2_2_T)))))/(m2/2))* tydstap_grootte
k_node3_3 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node3_2_T)))+( m_dot[2]*(energy_corrector_node3_1+(0.5*k_node2_2)))+(m_dot[3]*(
energy_corrector_node3_2+(0.5*k_node3_2)))+(m_dot_svv*h_svv_pype_oud*N_solar)+(h_kante*area2*(T_omgewing_db-((
k_node3_2_T)))+(h_onder*area*(T_omgewing_db-((k_node3_2_T)))+(h_top*area*(T_omgewing_db-((k_node3_2_T)))+( ((k_water*
area)/(h_node/2))*(((k_node4_2_T) - ((k_node3_2_T))))+(((k_water*area)/(h_node/2))*(((k_node2_2_T) - ((k_node3_2_T))))
)/(m3/2))* tydstap_grootte
k_node4_3 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node4_2_T)))+(m_dot[3]*(energy_corrector_node4_1+(0.5*k_node3_2)))+(m_dot[4]*(
energy_corrector_node4_2+(0.5*k_node4_2)))+(q_aux_4+(h_kante*area2*(T_omgewing_db-((k_node4_2_T)))+( ((k_water*area)/(
h_node/2))*(((k_node5_2_T) - ((k_node4_2_T))))+(h_onder*area*(T_omgewing_db-((k_node4_2_T)))+(h_top*area*(
T_omgewing_db-((k_node4_2_T)))+( ((k_water*area)/(h_node/2))*(((k_node3_2_T) - ((k_node4_2_T)))))/(m4/2))*
tydstap_grootte
k_node5_3 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node5_2_T)))+(m_dot[4]*(h_oud_4+(0.5*k_node4_2)))-(m_dot[5]*(h_oud_5+(0.5*
k_node5_2)))+(q_aux_5+(h_kante*area2*(T_omgewing_db-((k_node5_2_T)))+(h_top*area*(T_omgewing_db-((k_node5_2_T)))+(
h_onder*area*(T_omgewing_db-((k_node5_2_T)))+( ((k_water*area)/(h_node/2))*(((k_node4_2_T) - ((k_node5_2_T)))))/(m5/2)
))* tydstap_grootte

k_node1_3_Tn = Temperature(Water,h=((h_oud_1+(k_node1_3))),P=P_atm) "Verandering in entalpie"
{k_node1_3_T = T_oud_1 + k_node1_3_Tn}
k_node1_3_T = k_node1_3_Tn
k_node2_3_Tn = Temperature(Water,h=((h_oud_2+(k_node2_3))),P=P_atm)
{k_node2_3_T = T_oud_2 + k_node2_3_Tn}
k_node2_3_T = k_node2_3_Tn
k_node3_3_Tn = Temperature(Water,h=((h_oud_3+(k_node3_3))),P=P_atm)
{k_node3_3_T = T_oud_3 + k_node3_3_Tn}
k_node3_3_T = k_node3_3_Tn
k_node4_3_Tn = Temperature(Water,h=((h_oud_4+(k_node4_3))),P=P_atm)
{k_node4_3_T = T_oud_4 + k_node4_3_Tn}
k_node4_3_T = k_node4_3_Tn
k_node5_3_Tn = Temperature(Water,h=((h_oud_5+(k_node5_3))),P=P_atm)
{k_node5_3_T = T_oud_5 + k_node5_3_Tn}
k_node5_3_T = k_node5_3_Tn

k_node1_4= (((UA_tank_pipes/5)*(T_omgewing_db-k_node1_3_T)))+(m_dot_poort*(h_oud_1+(k_node1_3)))+(m_dot_return*
h_ringfeed_return)+(m_dot[1]*(energy_corrector_node1+(k_node1_3)))+(h_kante*area2*(T_omgewing_db-((k_node1_3_T)))+(
h_onder*area*(T_omgewing_db-((k_node1_3_T)))+(h_top*area*(T_omgewing_db-((k_node1_3_T)))+( ((k_water*area)/(2*h_node)
))*(((k_node2_3_T) - ((k_node1_3_T))))/(2*m1))* tydstap_grootte
k_node2_4= (((UA_tank_pipes/5)*(T_omgewing_db-k_node2_3_T)))+(m_dot[1]*(energy_corrector_node2_1+(k_node1_3)))+(m_dot[2]*(
energy_corrector_node2_2+(k_node2_3)))+(h_kante*area2*(T_omgewing_db-((k_node2_3_T)))+( ((k_water*area)/(h_node)
))*(((k_node3_3_T) - ((k_node2_3_T)))+(h_onder*area*(T_omgewing_db-((k_node2_3_T)))+(h_top*area*(T_omgewing_db-((
k_node2_3_T)))+( ((k_water*area)/(h_node*2))*(((k_node1_3_T) - ((k_node2_3_T))))/(2*m2))* tydstap_grootte
k_node3_4 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node3_3_T)))+( m_dot[2]*(energy_corrector_node3_1+(k_node2_3)))+(m_dot[3]*(
energy_corrector_node3_2+(k_node3_3)))+(m_dot_svv*h_svv_pype_oud*N_solar)+(h_kante*area2*(T_omgewing_db-((k_node3_3_T))
)+(h_onder*area*(T_omgewing_db-((k_node3_3_T)))+(h_top*area*(T_omgewing_db-((k_node3_3_T)))+( ((k_water*area)/(h_node
*2))*(((k_node4_3_T) - ((k_node3_3_T)))+( ((k_water*area)/(h_node*2))*(((k_node2_3_T) - ((k_node3_3_T)))))/(m3*2))*
tydstap_grootte
k_node4_4 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node4_3_T)))+(m_dot[3]*(energy_corrector_node4_1+(k_node3_3)))+(m_dot[4]*(
energy_corrector_node4_2+(k_node4_3)))+(q_aux_4+(h_kante*area2*(T_omgewing_db-((k_node4_3_T)))+( ((k_water*area)/(h_node
*2))*(((k_node5_3_T) - ((k_node4_3_T)))+(h_onder*area*(T_omgewing_db-((k_node4_3_T)))+(h_top*area*(T_omgewing_db-((
k_node4_3_T)))+( ((k_water*area)/(h_node*2))*(((k_node3_3_T) - ((k_node4_3_T)))))/(m4*2))* tydstap_grootte

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Appendix L. Computer code (EES): Reduced system simulation

```
k_node5_4 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node5_3_T)))+(m_dot[4]*(h_oud_4+(k_node4_3)))-(m_dot[5]*(h_oud_5+(k_node5_3))
)+q_aux_5+(h_kante*area2*(T_omgewing_db-((k_node5_3_T)))+(h_top*area*(T_omgewing_db-((k_node5_3_T)))+(h_onder*area
*(T_omgewing_db-((k_node5_3_T)))+(k_water*area)/(h_node*2))*((k_node4_3_T) - ((k_node5_3_T)))/(m_5*2))*
tydstap_grootte

stratTankEntalp[1]= h_oud_1 + (1/6*(k_node1_1+(2*k_node1_2)+(2*k_node1_3))+k_node1_4)
stratTankEntalp[2]= h_oud_2 + (1/6*(k_node2_1+(2*k_node2_2)+(2*k_node2_3))+k_node2_4)
stratTankEntalp[3]= h_oud_3 + (1/6*(k_node3_1+(2*k_node3_2)+(2*k_node3_3))+k_node3_4)
stratTankEntalp[4]= h_oud_4 + (1/6*(k_node4_1+(2*k_node4_2)+(2*k_node4_3))+k_node4_4)
stratTankEntalp[5]= h_oud_5 + (1/6*(k_node5_1+(2*k_node5_2)+(2*k_node5_3))+k_node5_4)

"=====
"Bereken die temperatuur van die nuut berekende entalpie"
stratifikasieTank[1] = Temperature(Water,h=stratTankEntalp[1],P=P_atm)
stratifikasieTank[2] = Temperature(Water,h=stratTankEntalp[2],P=P_atm)
stratifikasieTank[3] = Temperature(Water,h=stratTankEntalp[3],P=P_atm)
stratifikasieTank[4] = Temperature(Water,h=stratTankEntalp[4],P=P_atm)
stratifikasieTank[5] = Temperature(Water,h=stratTankEntalp[5],P=P_atm)

"=====
endif
ELSE "Nie laai suklus"
kode_blok = 3
"Node 1 se energie"
tyd = teller* tydstap_grootte
digtheid1 = Density(Water,T=T_oud_1,P=P_atm)
digtheid_oud1 = digtheid1
m_1 = h_node*area*digtheid1
{
    stratTankEntalp[1]= h_oud_1 + ((h_kante*area2*(T_omgewing_db-T_oud_1)+(k_water*area)/h_node)*(T_oud_2 -
    T_oud_1)+(m_dot_poort*h_poort_1)+(m_dot_return*h_ringfeed_return)-(m_dot_1*h_oud_1)+(h_onder*area*(T_omgewing_db-T_oud_1
    ))/(rho_water*area*h_node))* tydstap_grootte}
{stratTankEntalp[1]= h_oud_1 + (((m_dot_poort*h_poort_1)+(m_dot_return*h_ringfeed_return)-(m_dot_1*h_oud_1))/(m_1))*
    tydstap_grootte}
Q_return = (m_dot_return*h_ringfeed_return)
konduksie_term[1] = (((k_water*area)/h_node)*(T_oud_2 - T_oud_1))
konveksie_term[1] = (h_kante*area2*(T_omgewing_db-T_oud_1)+(h_onder*area*(T_omgewing_db-T_oud_1)) + (h_top*area*(T_omgewing_db
-T_oud_1)))
geforseerdeVloed[1] = (m_dot_poort*h_poort_1)+(m_dot_return*h_ringfeed_return)-(m_dot_1*h_oud_1)
tank_pyp_verlies[1] = ((UA_tank_pipes/5)*(T_omgewing_db-T_oud_1))

k_node1_1= ((geforseerdeVloed[1]+konveksie_term[1]+konduksie_term[1]+tank_pyp_verlies[1])/(m_1))* tydstap_grootte
k_node1_1_Tn = Temperature(Water,h=((0.5*k_node1_1)+h_oud_1),P=P_atm)
{k_node1_1_T = T_oud_1 + k_node1_1_Tn}
k_node1_1_T = k_node1_1_Tn {T_oud_1} {k_node1_1_Tn}

"Node 2 se energie"
digtheid2 = Density(Water,T=T_oud_2,P=P_atm)
digtheid_oud2 = digtheid2
m_2 = h_node*area*digtheid2
{
    stratTankEntalp[2] = h_oud_2 + ((h_kante*area2*(T_omgewing_db-T_oud_2)+(k_water*area)/h_node)*(T_oud_3 -
    T_oud_2)+((k_water*area)/h_node)*(T_oud_1 - T_oud_2)+(m_dot_1*h_oud_1)-(m_dot_2*h_oud_2))/(rho_water*area*h_node))*
    tydstap_grootte}
{stratTankEntalp[2] = h_oud_2 + (((m_dot_1*h_oud_1)-(m_dot_2*h_oud_2))/(m_2))* tydstap_grootte}
konduksie_term[2] = (((k_water*area)/h_node)*(T_oud_3 - T_oud_2))+(((k_water*area)/h_node)*(T_oud_1 - T_oud_2))
konveksie_term[2] = (h_kante*area2*(T_omgewing_db-T_oud_2)) + (h_onder*area*(T_omgewing_db-T_oud_2)) + (h_top*area*(
T_omgewing_db-T_oud_2))
geforseerdeVloed[2] = (m_dot_1*h_oud_1)-(m_dot_2*h_oud_2)
tank_pyp_verlies[2] = ((UA_tank_pipes/5)*(T_omgewing_db-T_oud_2))

k_node2_1= ((geforseerdeVloed[2]+konveksie_term[2] +konduksie_term[2]+tank_pyp_verlies[2])/(m_2))* tydstap_grootte
k_node2_1_Tn = Temperature(Water,h=((0.5*k_node2_1)+h_oud_2),P=P_atm)
{k_node2_1_T = T_oud_2 + k_node2_1_Tn}
k_node2_1_T = k_node2_1_Tn {T_oud_2} {k_node2_1_Tn}

"Node 3 se energie"
digtheid3 = Density(Water,T=T_oud_3,P=P_atm)
digtheid_oud3 = digtheid3
m_3 = h_node*area*digtheid3
{
    stratTankEntalp[3] = h_oud_3 + ((h_kante*area2*(T_omgewing_db-T_oud_3)+(k_water*area)/h_node)*(T_oud_4 -
    T_oud_3)+((k_water*area)/h_node)*(T_oud_2 - T_oud_3)+(m_dot_2*h_oud_2)-(m_dot_3*h_oud_3)+(m_dot_svw*h_svw))/(rho_water*
    area*h_node))* tydstap_grootte}
{stratTankEntalp[3] = h_oud_3 + (((m_dot_2*h_oud_2)-(m_dot_3*h_oud_3)+(m_dot_svw*h_svw))/(m_3))* tydstap_grootte}
konduksie_term[3] = (((k_water*area)/h_node)*(T_oud_4 - T_oud_3))+(((k_water*area)/h_node)*(T_oud_2 - T_oud_3))
konveksie_term[3] = (h_kante*area2*(T_omgewing_db-T_oud_3)) + (h_onder*area*(T_omgewing_db-T_oud_3)) + (h_top*area*(T_omgewing_db
-T_oud_3))
geforseerdeVloed[3] = (m_dot_2*h_oud_2)-(m_dot_3*h_oud_3)+(m_dot_svw*h_svw_pype_oud*N_solar)
tank_pyp_verlies[3] = ((UA_tank_pipes/5)*(T_omgewing_db-T_oud_3))

k_node3_1 = ((geforseerdeVloed[3]+konveksie_term[3]+konduksie_term[3]+tank_pyp_verlies[3])/(m_3))* tydstap_grootte
k_node3_1_Tn = Temperature(Water,h=((0.5*k_node3_1)+h_oud_3),P=P_atm)
{k_node3_1_T = T_oud_3 + k_node3_1_Tn}
k_node3_1_T = k_node3_1_Tn {T_oud_3} {k_node3_1_Tn}

"Node 4 se energie"
digtheid4 = Density(Water,T=T_oud_4,P=P_atm)
cp_water_4 =Cp(Water,T=T_oud_4,P=P_atm)
digtheid_oud4 = digtheid4
m_4 = h_node*area*digtheid4
q_aux_4 = Q_aux(T_set_aux, T_oud_4, q_element, cp_water_4, m_4, tydstap_grootte, electrical_element, m_dot[3], h_oud_3, -m_dot[4],
    h_oud_4, h_kante, area2, T_omgewing_db, 0, area, T_oud_3, T_oud_5, h_node, k_water)
konduksie_term[4] = (((k_water*area)/h_node)*(T_oud_5 - T_oud_4))+(((k_water*area)/h_node)*(T_oud_3 - T_oud_4))
```

Appendix L. Computer code (EES): Reduced system simulation

```
konveksie_term[4] = (h_kante*area2*(T_omgewing_db-T_oud_4))+(h_onder*area*(T_omgewing_db-T_oud_4)) + (h_top*area*(T_omgewing_db-T_oud_4))
geforseerdeVloed[4] = (m_dot[3]*h_oud_3)-(m_dot[4]*h_oud_4)
tank_pyp_verlies[4] = ((UA_tank_pipes/5)*(T_omgewing_db-T_oud_4))

{
    stratTankEntalp[4] = h_oud_4 + ((h_kante*area2*(T_omgewing_db-T_oud_4)+(k_water*area)/h_node)*(T_oud_5 - T_oud_4)+(k_water*area)/h_node)*(T_oud_3 - T_oud_4)+(m_dot_3*h_oud_3)-(m_dot_4*h_oud_4)+q_aux_4)/(rho_water*area*h_node)
)}* tydstap_grootte
{stratTankEntalp[4] = h_oud_4 + (((m_dot[3]*h_oud_3)-(m_dot[4]*h_oud_4)+q_aux_4)/(m_4))* tydstap_grootte}
k_node4_1 = ((geforseerdeVloed[4]+q_aux_4+konveksie_term[4]+konduksie_term[4]+tank_pyp_verlies[4])/(m_4))* tydstap_grootte
k_node4_1_Tn = Temperature(Water,h=((0.5*k_node4_1)+h_oud_4),P=P_atm)
{k_node4_1_T = T_oud_4 + k_node4_1_Tn}
k_node4_1_T = k_node4_1_Tn {T_oud_4} {k_node4_1_Tn}

"Node 5 se energie"
{m_dot_ringfeed = m_dot[5] + (m_dot_hp*2)}
digtheid5 = Density(Water,T=T_oud_5,P=P_atm)
cp_water_5 = Cp(Water,T=T_oud_5,P=P_atm)
m_5 = h_node*area*digtheid5
q_aux_5 = Q_aux(T_set_aux, T_oud_5, q_element, cp_water_5, m_5, tydstap_grootte, electrical_element, m_dot[4], h_oud_4, -m_dot[5], h_oud_5, h_kante, area2, T_omgewing_db, h_top, area, T_oud_4, T_oud_5, h_node, k_water)
konduksie_term[5] = (((k_water*area)/h_node)*(T_oud_4 - T_oud_5))
konveksie_term[5] = (h_kante*area2*(T_omgewing_db-T_oud_5))+((h_top*area*(T_omgewing_db-T_oud_5)))+(h_onder*area*(T_omgewing_db-T_oud_5))
geforseerdeVloed[5] = (m_dot[4]*h_oud_4)-(m_dot[5]*h_poort_bo)
tank_pyp_verlies[5] = ((UA_tank_pipes/5)*(T_omgewing_db-T_oud_5))

{q_aux_1 = Q_aux(T_set_aux, T_oud_5, q_element, cp_water_5, m_5, tydstap_grootte)}
{T_meng = ((T_hp*m_dot_hp)+(T_oud_5 * m_dot[5]))/m_dot_ringfeed}
{
    stratTankEntalp[5] = h_oud_5 + ((h_kante*area2*(T_omgewing_db-T_oud_5)+(k_water*area)/h_node)*(T_oud_4 - T_oud_5)+(m_dot_4*h_oud_4)-(m_dot_5*h_oud_5)+(h_top*area*(T_omgewing_db-T_oud_5))+q_aux_5)/(rho_water*area*h_node))* tydstap_grootte
}
{stratTankEntalp[5] = h_oud_5 + (((m_dot[4]*h_oud_4)-(m_dot[5]*h_oud_5)+q_aux_5)/(m_5))* tydstap_grootte}
k_node5_1 = ((geforseerdeVloed[5]+q_aux_5+konveksie_term[5]+konduksie_term[5]+tank_pyp_verlies[5])/(m_5))* tydstap_grootte
k_node5_1_Tn = Temperature(Water,h=((0.5*k_node5_1)+h_oud_5),P=P_atm)
{k_node5_1_T = T_oud_5 + k_node5_1_Tn}
k_node5_1_T = k_node5_1_Tn {T_oud_5} {k_node5_1_Tn}

k_node1_2= (((UA_tank_pipes/5)*(T_omgewing_db-k_node1_1_Tn))+(m_dot_poort*h_poort_1)+(m_dot_return*h_ringfeed_return)-(m_dot[1]*(h_oud_1+(0.5*k_node1_1)))+(h_kante*area2*(T_omgewing_db-((k_node1_1_Tn)))+(h_onder*area*(T_omgewing_db-((k_node1_1_Tn)))+(h_top*area*(T_omgewing_db-((k_node1_1_Tn)))+(k_water*area)/h_node/2))*((k_node2_1_Tn) - ((k_node1_1_Tn)))/(m_1/2))* tydstap_grootte
k_node2_2= (((UA_tank_pipes/5)*(T_omgewing_db-k_node2_1_Tn))+(m_dot[1]*(h_oud_1+(0.5*k_node1_1)))-(m_dot[2]*(h_oud_2+(0.5*k_node2_1)))+(h_kante*area2*(T_omgewing_db-((k_node2_1_Tn)))+(k_water*area)/h_node/2))*((k_node3_1_Tn) - ((k_node2_1_Tn)))+(h_onder*area*(T_omgewing_db-((k_node2_1_Tn)))+(h_top*area*(T_omgewing_db-((k_node2_1_Tn)))+(k_water*area)/h_node/2))*((k_node1_1_Tn) - ((k_node2_1_Tn)))/(m_2/2))* tydstap_grootte
k_node3_2 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node3_1_Tn))+(m_dot[2]*(h_oud_2+(0.5*k_node2_1)))-(m_dot[3]*(h_oud_3+(0.5*k_node3_1)))+(m_dot_svv*h_svv_pype_oud*N_solar)+((k_water*area)/h_node/2))*((k_node4_1_Tn) - ((k_node3_1_Tn)))+(h_kante*area2*(T_omgewing_db-((k_node3_1_Tn)))+(h_onder*area*(T_omgewing_db-((k_node3_1_Tn)))+(h_top*area*(T_omgewing_db-((k_node3_1_Tn)))+(k_water*area)/h_node/2))*((k_node2_1_Tn) - ((k_node3_1_Tn)))/(m_3/2))* tydstap_grootte
k_node4_2 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node4_1_Tn))+(m_dot[3]*(h_oud_3+(0.5*k_node3_1)))-(m_dot[4]*(h_oud_4+(0.5*k_node4_1)))+(q_aux_4+(h_kante*area2*(T_omgewing_db-((k_node4_1_Tn)))+(k_water*area)/h_node/2))*((k_node5_1_Tn) - ((k_node4_1_Tn)))+(h_onder*area*(T_omgewing_db-((k_node4_1_Tn)))+(h_top*area*(T_omgewing_db-((k_node4_1_Tn)))+(k_water*area)/h_node/2))*((k_node3_1_Tn) - ((k_node4_1_Tn)))/(m_4/2))* tydstap_grootte
k_node5_2 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node5_1_Tn))+(m_dot[4]*(h_oud_4+(0.5*k_node4_1)))-(m_dot[5]*(h_poort_bo+(0.5*k_node5_1)))+(q_aux_5+(h_kante*area2*(T_omgewing_db-((k_node5_1_Tn)))+(h_top*area*(T_omgewing_db-((k_node5_1_Tn)))+(h_onder*area*(T_omgewing_db-((k_node5_1_Tn)))+(k_water*area)/h_node/2))*((k_node4_1_Tn) - ((k_node5_1_Tn)))/(m_5/2))* tydstap_grootte

k_node1_2_Tn = Temperature(Water,h=((h_oud_1+(0.5*k_node1_2))),P=P_atm) "Verandering in entalpie"
{k_node1_2_T = T_oud_1 + k_node1_2_Tn}
k_node1_2_T = k_node1_2_Tn {T_oud_1} {k_node1_2_Tn}
k_node2_2_Tn = Temperature(Water,h=((h_oud_2+(0.5*k_node2_2))),P=P_atm)
{k_node2_2_T = T_oud_2 + k_node2_2_Tn}
k_node2_2_T = k_node2_2_Tn {T_oud_2} {k_node2_2_Tn}
k_node3_2_Tn = Temperature(Water,h=((h_oud_3+(0.5*k_node3_2))),P=P_atm)
{k_node3_2_T = T_oud_3 + k_node3_2_Tn}
k_node3_2_T = k_node3_2_Tn {T_oud_3} {k_node3_2_Tn}
k_node4_2_Tn = Temperature(Water,h=((h_oud_4+(0.5*k_node4_2))),P=P_atm)
{k_node4_2_T = T_oud_4 + k_node4_2_Tn}
k_node4_2_T = k_node4_2_Tn {T_oud_4} {k_node4_2_Tn}
k_node5_2_Tn = Temperature(Water,h=((h_oud_5+(0.5*k_node5_2))),P=P_atm)
{k_node5_2_T = T_oud_5 + k_node5_2_Tn}
k_node5_2_T = k_node5_2_Tn {T_oud_5} {k_node5_2_Tn}

k_node1_3= (((UA_tank_pipes/5)*(T_omgewing_db-k_node1_2_Tn))+(m_dot_poort*h_poort_1)+(m_dot_return*h_ringfeed_return)-(m_dot[1]*(h_oud_1+(0.5*k_node1_2)))+(h_kante*area2*(T_omgewing_db-((k_node1_2_Tn)))+(h_onder*area*(T_omgewing_db-((k_node1_2_Tn)))+(h_top*area*(T_omgewing_db-((k_node1_2_Tn)))+(k_water*area)/h_node/2))*((k_node2_2_Tn) - ((k_node1_2_Tn)))/(m_1/2))* tydstap_grootte
k_node2_3= (((UA_tank_pipes/5)*(T_omgewing_db-k_node2_2_Tn))+(m_dot[1]*(h_oud_1+(0.5*k_node1_2)))-(m_dot[2]*(h_oud_2+(0.5*k_node2_2)))+(h_kante*area2*(T_omgewing_db-((k_node2_2_Tn)))+(k_water*area)/h_node/2))*((k_node3_2_Tn) - ((k_node2_2_Tn)))+(h_onder*area*(T_omgewing_db-((k_node2_2_Tn)))+(h_top*area*(T_omgewing_db-((k_node2_2_Tn)))+(k_water*area)/h_node/2))*((k_node1_2_Tn) - ((k_node2_2_Tn)))/(m_2/2))* tydstap_grootte
k_node3_3 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node3_2_Tn))+(m_dot[2]*(h_oud_2+(0.5*k_node2_2)))-(m_dot[3]*(h_oud_3+(0.5*k_node3_2)))+(m_dot_svv*h_svv_pype_oud*N_solar)+(h_kante*area2*(T_omgewing_db-((k_node3_2_Tn)))+(h_onder*area*(T_omgewing_db-((k_node3_2_Tn)))+(h_top*area*(T_omgewing_db-((k_node3_2_Tn)))+(k_water*area)/h_node/2))*((k_node4_2_Tn) - ((k_node3_2_Tn)))+(k_water*area)/h_node/2))*((k_node2_2_Tn) - ((k_node3_2_Tn)))/(m_3/2))* tydstap_grootte
k_node4_3 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node4_2_Tn))+(m_dot[3]*(h_oud_3+(0.5*k_node3_2)))-(m_dot[4]*(h_oud_4+(0.5*k_node4_2)))+(q_aux_4+(h_kante*area2*(T_omgewing_db-((k_node4_2_Tn)))+(k_water*area)/h_node/2))*((k_node5_2_Tn) - ((k_node4_2_Tn)))+(h_onder*area*(T_omgewing_db-((k_node4_2_Tn)))+(h_top*area*(T_omgewing_db-((k_node4_2_Tn)))+(k_water*area)/h_node/2))*((k_node3_2_Tn) - ((k_node4_2_Tn)))/(m_4))* tydstap_grootte
```

Appendix L. Computer code (EES): Reduced system simulation

```

k_node5_3 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node5_2_T)))+(m_dot[4]*(h_oud_4+(0.5*k_node4_2_T)))-(m_dot[5]*(h_poort_bo+(0.5*
k_node5_2_T))+q_aux_5+(h_kante*area2*(T_omgewing_db-((k_node5_2_T))))+((h_top*area*(T_omgewing_db-((k_node5_2_T)))))+(
h_onder*area*(T_omgewing_db-((k_node5_2_T))))+(((k_water*area)/(h_node/2))*(((k_node4_2_T) - ((k_node5_2_T)))))/(m5/2)
)* tydstap_grootte

k_node1_3_Tn = Temperature(Water,h=((h_oud_1+(k_node1_3_T))),P=P_atm) "Verandering in entalpie"
{k_node1_3_T = T_oud_1 + k_node1_3_Tn}
k_node1_3_T = k_node1_3_Tn {T_oud_1} {k_node1_3_Tn}
k_node2_3_Tn = Temperature(Water,h=((h_oud_2+(k_node2_3_T))),P=P_atm)
{k_node2_3_T = T_oud_2 + k_node2_3_Tn}
k_node2_3_T = k_node2_3_Tn {T_oud_2} {k_node2_3_Tn}
k_node3_3_Tn = Temperature(Water,h=((h_oud_3+(k_node3_3_T))),P=P_atm)
{k_node3_3_T = T_oud_3 + k_node3_3_Tn}
k_node3_3_T = k_node3_3_Tn {T_oud_3} {k_node3_3_Tn}
k_node4_3_Tn = Temperature(Water,h=((h_oud_4+(k_node4_3_T))),P=P_atm)
{k_node4_3_T = T_oud_4 + k_node4_3_Tn}
k_node4_3_T = k_node4_3_Tn {T_oud_4} {k_node4_3_Tn}
k_node5_3_Tn = Temperature(Water,h=((h_oud_5+(k_node5_3_T))),P=P_atm)
{k_node5_3_T = T_oud_5 + k_node5_3_Tn}
k_node5_3_T = k_node5_3_Tn {T_oud_5} {k_node5_3_Tn}

k_node1_4= (((UA_tank_pipes/5)*(T_omgewing_db-k_node1_3_T)))+(m_dot_poort*h_poort_1)+(m_dot_return*h_ringfeed_return)-(m_dot
[1]*(h_oud_1+(k_node1_3_T)))+(h_kante*area2*(T_omgewing_db-((k_node1_3_T))))+(h_onder*area*(T_omgewing_db-((k_node1_3_T))))
)+(h_top*area*(T_omgewing_db-((k_node1_3_T))))+(((k_water*area)/(h_node*2))*(((k_node2_3_T) - ((k_node1_3_T)))))/(m1
*2))* tydstap_grootte
k_node2_4= (((UA_tank_pipes/5)*(T_omgewing_db-k_node2_3_T)))+(m_dot[1]*(h_oud_1+(k_node1_3_T)))-(m_dot[2]*(h_oud_2+(k_node2_3_T))
)+(h_kante*area2*(T_omgewing_db-((k_node2_3_T))))+(h_onder*area*(T_omgewing_db-((k_node2_3_T))))+(h_top*area*(
T_omgewing_db-((k_node2_3_T))))+(((k_water*area)/(h_node*2))*(((k_node3_3_T) - ((k_node2_3_T))))+(((k_water*area)/(
h_node*2))*(((k_node1_3_T) - ((k_node2_3_T)))))/(m2*2))* tydstap_grootte
k_node3_4 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node3_3_T)))+( m_dot[2]*(h_oud_2+(k_node2_3_T)))-(m_dot[3]*(h_oud_3+(k_node3_3_T)
)))+(m_dot_svv*h_svv_pype_oud*N_solar)+(h_kante*area2*(T_omgewing_db-((k_node3_3_T))))+(h_onder*area*(T_omgewing_db-((
k_node3_3_T))))+(h_top*area*(T_omgewing_db-((k_node3_3_T))))+(((k_water*area)/(h_node*2))*(((k_node4_3_T) - (T_oud_3+(
k_node1_3_T))))+(((k_water*area)/(h_node*2))*(((k_node2_3_T) - ((k_node3_3_T)))))/(m3*2))* tydstap_grootte
k_node4_4 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node4_3_T)))+(m_dot[3]*(h_oud_3+(k_node3_3_T)))-(m_dot[4]*(h_oud_4+(k_node4_3_T))
)+q_aux_4+(h_kante*area2*(T_omgewing_db-((k_node4_3_T))))+(((k_water*area)/(h_node*2))*(((k_node5_3_T) - ((k_node4_3_T))
)))+(h_onder*area*(T_omgewing_db-((k_node4_3_T))))+(h_top*area*(T_omgewing_db-((k_node4_3_T))))+(((k_water*area)/(h_node
*2))*(((k_node3_3_T) - ((k_node4_3_T)))))/(m4*2))* tydstap_grootte
k_node5_4 = (((UA_tank_pipes/5)*(T_omgewing_db-k_node5_3_T)))+(m_dot[4]*(h_oud_4+(k_node4_3_T)))-(m_dot[5]*(h_poort_bo+(
k_node5_3_T))+q_aux_5+(h_kante*area2*(T_omgewing_db-((k_node5_3_T))))+((h_top*area*(T_omgewing_db-((k_node5_3_T)))))+(
h_onder*area*(T_omgewing_db-((k_node5_3_T))))+(((k_water*area)/(h_node*2))*(((k_node4_3_T) - ((k_node5_3_T)))))/(m5*2)
)* tydstap_grootte

stratTankEntalp[1]= h_oud_1 + (1/6*(k_node1_1+(2*k_node1_2)+(2*k_node1_3))+k_node1_4)
stratTankEntalp[2]= h_oud_2 + (1/6*(k_node2_1+(2*k_node2_2)+(2*k_node2_3))+k_node2_4)
stratTankEntalp[3]= h_oud_3 + (1/6*(k_node3_1+(2*k_node3_2)+(2*k_node3_3))+k_node3_4)
stratTankEntalp[4]= h_oud_4 + (1/6*(k_node4_1+(2*k_node4_2)+(2*k_node4_3))+k_node4_4)
stratTankEntalp[5]= h_oud_5 + (1/6*(k_node5_1+(2*k_node5_2)+(2*k_node5_3))+k_node5_4)

"=====
"Bereken die temperatuur van die nuut berekende entalpie"
stratifikasieTank[1] = Temperature(Water,h=stratTankEntalp[1],P=P_atm)
stratifikasieTank[2] = Temperature(Water,h=stratTankEntalp[2],P=P_atm)
stratifikasieTank[3] = Temperature(Water,h=stratTankEntalp[3],P=P_atm)
stratifikasieTank[4] = Temperature(Water,h=stratTankEntalp[4],P=P_atm)
stratifikasieTank[5] = Temperature(Water,h=stratTankEntalp[5],P=P_atm)

"=====
ENDIF

if(m_dot_hp > 0) then
Q_hp_probe = q_probe_hp*N_hp
{Q_hp_probe_test = m_dot_hp*(h_hp-h_meng)}
{COP_hp = Q_ECDH90M_COP(T_omgewing_db,RH, Temperature(Water,P=P_atm,h=h_munisipaliteit))}
{if (q_probe_hp > q_hp_probe_test) then
Q_elektries_hp = Q_hp_probe_test/COP_hp_potential*( q_hp_probe_test/q_probe_hp)
else}
Q_elektries_hp = Q_hp_probe_test/COP_hp_potential
{endif}
else
Q_hp_probe = 0 [W]
{Q_hp_probe_test = 0 [W]}
COP_hp = 0
Q_elektries_hp = 0 [W]
endif

Q_losses_tank = konveksie_term[1] + konveksie_term[2] + konveksie_term[3] + konveksie_term[4] + konveksie_term[5]
Q_losses_tank_pype = tank_pyp_verlies[1] + tank_pyp_verlies[2] + tank_pyp_verlies[3] + tank_pyp_verlies[4] + tank_pyp_verlies
[5]

Q_supplies_energy = ((h_hp*m_dot_hp*N_hp) + (h_svv*m_dot_svv*N_solar) + (h_poort_1*m_dot_poort) - Q_dot_munisipaliteit -
Q_son_probe)*(hp_timer_control_internal)

Q_tank_bottom = (m_dot_poort*h_poort_1)+(m_dot_return*h_ringfeed_return)

Q_user2 = Q_top - Q_return - Q_losses_d - Q_losses_dead_end - Q_dot_munisipaliteit

```


Appendix L. Computer code (EES): Reduced system simulation

```
Q_user = m_dot_in*(h_ringfeed_aftap-h_munispaliteit)

{Q_user = Q_top - Q_return - Q_losses_d - Q_losses_dead_end}
{Q_user2 = m_dot_in}
Q_ringmain_normal = Q_losses_d + Q_losses_dead_end
Stelsel_entalpie = stratTankEntalp[1] + stratTankEntalp[2] + stratTankEntalp[3] + stratTankEntalp[4] + stratTankEntalp[5]
Cop_hp = 0

Q_netto = Q_hp_probe_test + Q_son_probe + Q_losses_tank + Q_losses_tank_pype - Q_user -Q_ringmain_normal

Q_netto2 = Q_supplies_energy + Q_son_probe + Q_losses_tank + Q_losses_tank_pype - Q_user - Q_ringmain_normal -
Q_solar_losses [-Q_top + Q_return + Q_hp_probe_test +Q_dot_munispaliteit + Q_losses_tank + Q_son_probe]

Q_in = Q_hp_probe_test + Q_son_probe
Q_out = Q_losses_tank + Q_losses_tank_pype - Q_solar_losses - Q_user - Q_ringmain_normal

Q_netto = Q_hp_probe_test*N_hp + Q_son_probe*N_solar + Q_losses_tank + Q_losses_tank_pype - Q_user -Q_ringmain_normal
Q_netto3 = Q_hp_probe_test*N_hp + Q_son_probe + Q_losses_tank + Q_losses_tank_pype - Q_user -Q_ringmain_normal -
Q_solar_losses

Q_ringmain_test = Q_top - Q_return - Q_ringmain_normal - (m_dot_in*h_ringfeed_aftap)
Q_heat_pump_loop_test = ((h_hp*m_dot_hp*N_hp) + (h_svv*m_dot_svv*N_solar) + (h_poort_1*m_dot_poort) -Q_dot_munispaliteit -
Q_son_probe) - (Q_hp_probe_test*N_hp)
Sigma_ringmain_massflux = m_dot_ringfeed -m_dot_in - m_dot_return
Sigma_ashp_loop = (m_dot_hp*N_hp) + m_dot_svv + m_dot_poort - m_dot_in
Q_tester = -Q_dot_munispaliteit + (h_poort_1*m_dot_poort) + (m_dot_hp*h_meng)
Q_tester_2 = (h_poort_1*m_dot_poort) + (m_dot_hp*h_meng*N_hp)
Q_tester_3 = (m_dot_hp*h_meng)
Q_tester_4 = (h_poort_1*m_dot_poort)

mass_balance_tanks_top = (m_dot_hp*N_hp) - m_dot_ringfeed + m_dot_5
mass_balance_tanks_bottom = m_dot_poort + m_dot_return-m_dot_1

Q_netto_laaste = Q_tank_top + Q_tank_bottom+Q_losses_tank+Q_losses_tank_pype

end

"=====
"Funksie wat die elektriese element simuleer"
Function Q_aux(T_set_aux, T_tank, Q_heat, Cp_water, massa_water,tydstap,elektriese_elementActive,m_oud1,h_oud1,m_oud2,h_oud2,
h_kante,area2,T_omgewing_db,h_top,area,T_tank2,T_tank3,h_node,k_water)
{'Die energie wat by die tank gevoeg moet word as die tank temperatuur daal'}

if(elektriese_elementActive = 1) then
If T_tank < T_set_aux Then
antwQ = (((massa_water * Cp_water * (T_set_aux - T_tank)))/tydstap -((m_oud1*h_oud1)+(m_oud2*h_oud2)))+(h_kante*area2*(
T_omgewing_db-T_tank)))+(h_top*area*(T_omgewing_db-T_tank)))*tydstap +(((k_water*area)/h_node)*(T_tank2 - T_tank)) +(((
k_water*area)/h_node)*(T_tank3 - T_tank))){[kg]*[k]/(kg*k)}*[K] = [kJ] maar [kW] = [kJ/s]}
If antwQ > (Q_heat*tydstap) Then
Q_aux = Q_heat
Else
Q_aux = antwQ/(tydstap)
EndIf
Else
Q_aux = 0
EndIf
else
Q_aux = 0
endif
End

Function Time_function$( )
{'Die energie wat by die tank gevoeg moet word as die tank temperatuur daal'}
Time_function$ = concat('The time is now ',TIME$)

End

"=====
"Funksie wat die hittepomp simuleer"
Function Q_ECDH90MT_nuwe(T_db,RH,T_inlaat,T_wb,density_ratio,ashp_ratio)
"Die ou vergelyking"
{Q_ECDH90MT_nuwe = (51.3346731853813) + ((1.58865575624804)*T_db) + ((-0.0737514686869437)*T_inlaat)+((-0.59115919832997900*
RH) + (0.00170073734429347*RH^2))}

"Die nuwe vergelyking"
Q_ECDH90MT_nuwe =(3.45693852134674 + ((0.075939057244308)*T_db) + ((-0.0302670035556761)*T_inlaat)+((-0.00967840826037918*RH
)))*((1.71332505730209e+01) + ((2.86265884650667e-03)*T_inlaat^2) + (cos(T_db+(1.00463893485746e+00)))+ (cos(T_wb
+(-1.38353732325251e+00)))))*density_ratio*ashp_ratio

End

"=====
Function Q_ECDH90M_COP (T_db,RH,T_inlaat,density_ratio)
Q_ECDH90M_COP = (3.45693852134674 + ((0.075939057244308)*T_db) + ((-0.0302670035556761)*T_inlaat)+((-0.00967840826037918*RH))
*density_ratio
```

Appendix L. Computer code (EES): Reduced system simulation

End

```
Function h_meng_calc (T_db,RH,T_wb,h_munispaliteit,T_set_point,P_druk,activate_heat_pump,N_hp,m_dot_user,m_dot_ring_return,
    m_dot_svv,N_solar,h_poort,m_dot_in,density_ratio)
teller = 1
repeat_control = 0
h_meng_oud = h_munispaliteit
Repeat
if (teller = 1) then
m_dot_ashp = hpMassaVloei(h_munispaliteit,Q_ECDH90MT_nuwe(T_db,RH,h_munispaliteit,T_wb,density_ratio),T_set_point,P_druk,
    activate_heat_pump,)
m_dot_5 = m_dot_ring_return + m_dot_user - (m_dot_ashp * N_hp)
m_dot_2 = m_dot_5 - (m_dot_svv*N_solar)
m_dot_poort = m_dot_2 - m_dot_ring_return
if ((m_dot_ashp > 0) AND (m_dot_poort < 0)) then
h_meng_predict = ((m_dot_in*h_munispaliteit) +((-m_dot_poort-(m_dot_svv*N_solar))*h_poort))/(m_dot_ashp * N_hp)
else
h_meng_predict = h_poort
endif
endif
else
m_dot_ashp = hpMassaVloei(h_meng_oud,Q_ECDH90MT_nuwe(T_db,RH,h_meng_oud,T_wb,density_ratio,ashp_ratio),T_set_point,P_druk,
    activate_heat_pump,m_dot_ashp_max)
m_dot_5 = m_dot_ring_return + m_dot_user - (m_dot_ashp * N_hp)
m_dot_2 = m_dot_5 - m_dot_svv*N_solar
m_dot_poort = m_dot_2 - m_dot_ring_return
if ((m_dot_ashp > 0) AND (m_dot_poort < 0)) then
h_meng_predict = ((m_dot_in*h_munispaliteit) +((-m_dot_poort-(m_dot_svv*N_solar))*h_poort))/(m_dot_ashp * N_hp)
else
h_meng_predict = h_poort
endif
endif
{
    if (abs((h_meng_predict - h_meng_oud)) < 0.01) then
        repeat_control = 1
        teller = 10000
    else}
teller = teller + 1
h_meng_oud = h_meng_predict
verskil = h_meng_predict - h_meng_oud
verskil2 = (m_dot_ashp*h_meng_predict) - (m_dot_in*h_munispaliteit) + (m_dot_poort*h_poort)
{repeat_control = 0}
{endif}
Until (teller = 20)
h_meng_calc = h_meng_predict
End
```

```
Function m_poort_calc (T_db,RH,T_wb,h_munispaliteit,T_set_point,P_druk,activate_heat_pump,N_hp,m_dot_user,m_dot_ring_return,
    m_dot_svv,N_solar,h_poort,m_dot_in,density_ratio,m_dot_ashp_max,ashp_ratio)
teller = 1
repeat_control = 0
h_meng_oud = h_munispaliteit
Repeat
if (teller = 1) then
Q_ashp = Q_ECDH90MT_nuwe(T_db,RH,Temperature(Water,P=P_druk,h=h_munispaliteit),T_wb,density_ratio,ashp_ratio)
m_dot_ashp = hpMassaVloei(h_munispaliteit,Q_ashp,T_set_point,P_druk,activate_heat_pump,m_dot_ashp_max)
m_dot_5 = m_dot_ring_return + m_dot_user - (m_dot_ashp * N_hp)
m_dot_2 = m_dot_5 - (m_dot_svv*N_solar)
m_dot_poort = m_dot_2 - m_dot_ring_return
if (m_dot_poort < 0) then "Charge"
if (m_dot_ashp > 0) then
h_meng_predict = ((m_dot_in*h_munispaliteit) +((-m_dot_poort-(m_dot_svv*N_solar))*h_poort))/(m_dot_ashp * N_hp)
else
h_meng_predict = h_poort
endif
else "m_dot_poort > 0 and m_ashp > 0 Charge"
h_meng_predict = h_munispaliteit
endif
else
Q_ashp = Q_ECDH90MT_nuwe(T_db,RH,Temperature(Water,P=P_druk,h=h_meng_oud),T_wb,density_ratio)
m_dot_ashp = hpMassaVloei(h_meng_oud,Q_ashp,T_set_point,P_druk,activate_heat_pump,m_dot_ashp_max)
m_dot_5 = m_dot_ring_return + m_dot_user - (m_dot_ashp * N_hp)
m_dot_2 = m_dot_5 - m_dot_svv*N_solar
m_dot_poort = m_dot_2 - m_dot_ring_return

if (m_dot_poort < 0) then "Charge"
if (m_dot_ashp > 0) then
h_meng_predict = ((m_dot_in*h_munispaliteit) +((-m_dot_poort-(m_dot_svv*N_solar))*h_poort))/(m_dot_ashp * N_hp)
else
h_meng_predict = h_poort
endif
else "m_dot_poort > 0 and m_ashp > 0 Charge"
h_meng_predict = h_munispaliteit
endif
endif
{
    if (abs((h_meng_predict - h_meng_oud)) < 0.01) then
        repeat_control = 1
        teller = 10000
    else}
teller = teller + 1

verskil = abs(h_meng_predict - h_meng_oud)
h_meng_oud_vergelyk = h_meng_oud
```

Appendix L. Computer code (EES): Reduced system simulation

```
verskil2 = abs((m_dot_ashp*h_meng_predict* N_hp) - (m_dot_in*h_munispaliteit) + (m_dot_poort*h_poort))
h_meng_oud = h_meng_predict
{repeat_control = 0}
{endif}
Until (verskil < 0.0001)
m_poort_calc = m_dot_poort
T_meng_predict = Temperature(Water,P=P_druk,h=h_meng_predict )
End

Function h_uitlaat_svv_new(T_amb,G,A_col,h_inlaat,P_atm,m_dot,h_son_oud,UA_solar,activate_solar,tydstap,m_bank)
{I_inlaat_water = Inlaat temperatuur van die SWV
T_amb = Die omgewings temperatuur
G = Sonbestraling
A_col = Die area van die kollektor}
{'Die energie wat by die tank gevoeg moet word as die tank temperatuur daal'}
conversion = 0.001 [kW/w]
T_inlaat_water = Temperature(Water,P=P_atm,h=h_inlaat)
T_son_oud = Temperature(Water,P=P_atm,h=h_son_oud)

eta_son = eta_svv(T_amb,G,h_inlaat,P_atm)
if ((G>80) AND (eta_son>0)) then

{eta_son = (m_dot(h_uitlaat-h_inlaat))/(G*A_col)}

q_son = (eta_son*G*A_col*conversion)
else
q_son = 0
endif
if (T_son_oud <= T_amb) then
hitteverlies = 0
else
if ((T_amb < 0) AND (T_son_oud < 3)) then
hitteverlies = 0
else
if (eta_son>0) then
hitteverlies = (1-eta_son)*G*A_col*conversion
MODE = 1
else
hitteverlies = UA_solar*(T_son_oud - T_amb )
MODE = 2
endif
endif
endif
{h_uitlaat_svv_new =h_son_oud + ((q_son - (activate_solar*m_dot*(h_son_oud-h_inlaat)) -(hitteverlies*UA_solar*(T_son_oud -
T_amb )))*tydstap)/(m_bank)}
h_uitlaat_svv_new =h_son_oud + ((q_son - (activate_solar*m_dot*(h_son_oud-h_inlaat)) -hitteverlies)*tydstap)/(m_bank)
End

Function h_uitlaat_svv_pype(T_amb,h_inlaat,P_atm,m_dot,h_son_oud,UA_solar,tydstap,m_bank)
{I_inlaat_water = Inlaat temperatuur van die SWV
T_amb = Die omgewings temperatuur}
{'Die energie wat by die tank gevoeg moet word as die tank temperatuur daal'}
conversion = 0.001 [kW/w]
T_inlaat_water = Temperature(Water,P=P_atm,h=h_inlaat)
T_son_oud = Temperature(Water,P=P_atm,h=h_son_oud)

if (T_son_oud<T_amb) then
hitteverlies = 0
else
if ((T_amb < 0) AND (T_son_oud < 3)) then
hitteverlies = 0
else
hitteverlies = UA_solar*(T_son_oud - T_amb )
endif
endif
{h_uitlaat_svv_new =h_son_oud + ((q_son - (activate_solar*m_dot*(h_son_oud-h_inlaat)) -(hitteverlies*UA_solar*(T_son_oud -
T_amb )))*tydstap)/(m_bank)}
h_uitlaat_svv_pype =h_son_oud + (( -m_dot*(h_son_oud-h_inlaat)-hitteverlies)*tydstap)/(m_bank)
End

Function m_svv_activate(h_uitlaat,T_2,T_3,T_max,P_atm,T_delta_hoog,T_delta_laag,m_dot_svv_oud)
if (h_uitlaat >0) then
T_svv =Temperature(Water,P=P_atm,h=h_uitlaat)
if ((T_svv - T_2)>T_delta_hoog) then
if (T_3 > T_max) then
m_svv_activate = 0
else
m_svv_activate = 1
endif
else
if (m_dot_svv_oud > 0) then
if ((T_svv-T_2) > T_delta_laag) Then
m_svv_activate = 1
else
m_svv_activate = 0
endif
else
m_svv_activate = 0
endif
endif
else
endif
endif
```

Appendix L. Computer code (EES): Reduced system simulation

```
m_svv_activate = 0
endif
End

Function eta_svv(T_amb,G,h_inlaat,P_atm)
{I_inlaat_water - Inlaat temperatuur van die SWV
  T_amb - Die omgewings temperatuur
  G - Sonbestraling
  A_col - Die area van die kollektor}
{'Die energie wat by die tank gevoeg moet word as die tank temperatuur daal'}
if (G> 80) THEN
  T_inlaat_water = Temperature(Water,P=P_atm,h=h_inlaat)
  A = (T_inlaat_water-T_amb)/G
  {eta_son = (m_dot(h_uitlaat-h_inlaat))/(G*A_col)}
  {eta_svv = (-4072*a^2) + (359.07*a) + 48)*1.66/100}
  {eta_svv = (-4072*a^2) + (359.07*a) + 40)*1.8/100}
  eta_svv = (-6250*a^2) + (12.5*a) + 53.125)*1.63/100
ELSE
  eta_svv = 0
ENDIF
IF(eta_svv < 0) THEN
  eta_svv = 0
ENDIF
End

Function m_svv(h_uitlaat,T_2,T_3,T_max,P_atm,T_delta_hoog,T_delta_laag,m_dot_svv)
if(m_dot_svv >0) AND (h_uitlaat >0) then
  T_svv =Temperature(Water,P=P_atm,h=h_uitlaat)
  if((T_svv - T_2)>T_delta_hoog) then
    if (T_3 > T_max) then
      m_svv = 0[kg/s]
    else
      m_svv = m_dot_svv
    endif
  else
    if ((T_svv-T_2) > T_delta_laag) Then
      m_svv = m_dot_svv
    else
      m_svv = 0 [kg/s]
    endif
  endif
else
  m_svv = 0 [kg/s]
endif
End

Function hpTimerControl(T_probe,T_probe_set_piont ,teller ,timer_hp_oud,activate_timer_hp_oud,t_set_point_2)
"Indien die program 'n 1 terugstuur dan moet die timer loop en die HP aangeskakel word"
if(T_probe <= T_probe_set_piont ) then "Indien die toetsnode se temperatuur kouer is as die set point begin aftel"
hpTimerControl = 1
else
  {if(teller >1 ) then "Indien ons verby die eerste tydstap is."}
  if(activate_timer_hp_oud = 1) then
    if(T_probe >= t_set_point_2 ) then
      hpTimerControl = 0
    else
      hpTimerControl = 1
    endif
  else
    hpTimerControl = 0
  endif
endif
end

Function hpTimerControl_nuwe(T_probe,T_probe_set_piont ,teller ,timer_hp_oud,activate_timer_hp_oud,COP)
"Indien die program 'n 1 terugstuur dan moet die timer loop en die HP aangeskakel word"
if(T_probe < 53 ) then
  if((T_probe < T_probe_set_piont) OR (COP >3)) then "Indien die toetsnode se temperatuur kouer is as die set point begin aftel"
  hpTimerControl_nuwe = 1
  else
    {if(teller >1 ) then "Indien ons verby die eerste tydstap is."}
    if(activate_timer_hp_oud = 1) then
      if((timer_hp_oud > (900)) ) then
        hpTimerControl_nuwe = 0
      else
        hpTimerControl_nuwe = 1
      endif
    else
      hpTimerControl_nuwe = 0
    endif
  endif
endif
end

"=====
"=====
"Bepaal die massavloei wat deur die HP moet vloei"
Function hpMassaVloei(h_inlet,Q_hp,T_set_point,P_druk,activate_heat_pump,m_dot_ashp_max)
if(activate_heat_pump = 1) then "Indien die HP nie op tydsakelaar is nie bereken die volgende"
h_outlet = Enthalpy(Water,T=T_set_point,P=P_druk) "Bereken die verlangde uitlaat entalpie"
if(h_outlet = h_inlet) then
```

Appendix L. Computer code (EES): Reduced system simulation

```
hpMassaVloei = 0 [kg/s]
else
if ((h_inlet < h_outlet) and (Q_hp > 0)) then
hpMassaVloeiToets = Q_hp/(h_outlet - h_inlet) "Bereken die gestadigde massavloei wat deur die stelsel benodig word"
if (hpMassaVloeiToets>m_dot_ashp_max) then
hpMassaVloei = m_dot_ashp_max
else
hpMassaVloei = hpMassaVloeiToets
endif
else
hpMassaVloei = m_dot_ashp_max
endif
endif
else
hpMassaVloei = 0
endif
End

Function maalFaktor(m_1,m_2)
if (m_1 > m_2) then
maalFaktor = 1
else
maalFaktor = -1
endif
end

Function upwindFunction(m_1,m_2,u_1,u_2)
if (m_1 > m_2) then
upwindFunction = u_1
else
upwindFunction = u_2
endif
end

"=====
"------(Die stelsel inset )
"-----"

rho_water = Density(Water,T=T_aanvangs[1],P=P_atm)
cp_water = Cp(Water,T=25[C],P=P_atm)
k_water = Conductivity(Water,T=T_aanvangs[1],P=P_atm)/1000[W/kW] "Konduksie koëffisient van die water"

"Vaste veranderlikes"
{h_node = 2 [m] "Hoogte van die nodus"}
{radius = 0.6084 [m] "Radius van die tank"
{Vol_tank = 3000 [l] "Volume van 'n enkele node"}
Vol_node = 3 [m^3] "Volume van 'n enkele node"}
area = pi*(radius^2) "Deursnit area"
area2 = 2*pi*radius*h_node "Omtrek area"
Vol_node = area*h_node "Bereken die hoogte van die nodus"
col = (((m_dot_poort_i+m_dot_return)/(area*rho_water))*tydstap_grootte)/h_node "Colesky factorization"
aantalNodusse = 5
"=====
"Die loop se berekninge"
n_step = 120 {1950} "Aantal n_step waardeur die kode moet hardloop"
tydstap_grootte = 60[s] "Die lengte van 'n enkele tydstap"
n_sek = tydstap_grootte * n_step
n_ure = n_sek / 3600 [s/h]
n_dae = n_ure / 24 [h/d]
"=====
"Aanvangs temperatuur parameters vir die verskeie ure"
T_aanvangs[1] = 47 [C]
T_aanvangs[2] = 49 [C]
T_aanvangs[3] = 51 [C]
T_aanvangs[4] = 55 [C]
T_aanvangs[5] = 60 [C]

rhp_aanvangs[1] = Density(Water,T=T_aanvangs[1],P=P_atm)
rhp_aanvangs[2] = Density(Water,T=T_aanvangs[2],P=P_atm)
rhp_aanvangs[3] = Density(Water,T=T_aanvangs[3],P=P_atm)
rhp_aanvangs[4] = Density(Water,T=T_aanvangs[4],P=P_atm)
rhp_aanvangs[5] = Density(Water,T=T_aanvangs[5],P=P_atm)

h_aanvangs[1] = Enthalpy(Water,T=T_aanvangs[1],P=P_atm)
h_aanvangs[2] = Enthalpy(Water,T=T_aanvangs[2],P=P_atm)
h_aanvangs[3] = Enthalpy(Water,T=T_aanvangs[3],P=P_atm)
h_aanvangs[4] = Enthalpy(Water,T=T_aanvangs[4],P=P_atm)
h_aanvangs[5] = Enthalpy(Water,T=T_aanvangs[5],P=P_atm)

T_wb_i = 0.03633
T_db_i = 4.99
"=====
"------(Die stelsel inset massavloei)
"-----"

"Die massavloei in en uit die stelsel"
m_dot_in_i = 0.024 [kg/s] "#####Inset [1]##### - Die massavloei in die stelsel in"
m_dot_uit = m_dot_in "#####[Massavloei vergelyking [8]##### - Behoud van massa"

"Massavloei deur die SWV"
```

Appendix L. Computer code (EES): Reduced system simulation

```
"Word op die stadium deur die Diagram window beheer"
{m_dot_svv = 0.1 [kg/s]} "#####Inset [2]##### - Die massavloei in die stelsel in "{0.069 [kg/s] }"Die
massavloei deur die vakuumbuise"

"Dinamiese funksie wat die massavloei deur die HP bepaal"
h_munispaliteit1 = Enthalpy(Water,T=T_munispaliteit_koud,P=P_atm)
m_dot_hp_aanvanklik = hpMassaVloei(h_munispaliteit1,Q_ECDH90MT_nuwe(T_omgewing_db,RH,T_munispaliteit_koud,T_omgewing_wb,
density_ratio,ashp_ratio),T_hp,P_atm,activate_heat_pump,m_dot_ashp_max)*N_hp "#####Inset [3]##### -
Die massavloei in die stelsel in"

"Koue water toevoer wat by die onderkant van die tank invloei"
m_dot_hp_swh_i = m_dot_uit - m_dot_hp_aanvanklik "#####[Massavloei vergelyking [10]##### "m_dot_hp_swh =
m_dot_in - (m_dot_hp*2) "
m_dot_swh = 0 [kg/s]
m_dot_poort_i = m_dot_hp_swh_i - m_dot_swh"#####[Massavloei vergelyking [6]##### "Die koue water
temperatuur wat onder by die water kan invloei"

"Die ringmain terugvoer massavloei"
m_dot_las = m_dot_uit "#####[Massavloei vergelyking [8]##### "Die massavloei wat deur die hospitaal
gebruik word"
m_dot_return_i = 0.2 [kg/s] "Die ringfeed se warm water return"
{ m_dot_ringfeed = m_dot_return + m_dot_las "Die massavloei deur die ringfeed"}

m_dot_aanvangs_1 = m_dot_poort_i + m_dot_return "#####[Massavloei vergelyking [1]##### "
m_dot_aanvangs_2 = m_dot_aanvangs_1 "#####[Massavloei vergelyking [2]##### "
m_dot_aanvangs_3 = m_dot_aanvangs_2 + (m_dot_svv_aanvanklik*N_solar) "#####[Massavloei vergelyking
[3]##### "
m_dot_aanvangs_4 = m_dot_aanvangs_3"#####[Massavloei vergelyking [4]##### "
m_dot_aanvangs_5 = m_dot_aanvangs_4 "#####[Massavloei vergelyking [5]##### "

"=====
"------(Die stelsel inset temperature)
"-----"

T_delta_hoog = 38.5
T_delta_laag = 29.5
T_ringfeed_return = T_hp - T_ringfeed_delta
T_ringfeed_delta = 10 [C]
{T_swh = 45 [C]}
T_hp = 62[C]
T_probe_hp = 52 [deg]
{G= 1000 [W/m^2]}
A_col =7.6 [m^2]
T_indoors = 23 [C] "Gemiddelde temperatuur van die hospitaal aan die binnekant"
{T_set_aux = 60 [C]} "Word gestel in Diagram window"
"=====
h_hp = Enthalpy(Water,T=T_hp,P=P_atm) "Vaste entalpie vir alle waardes"
{h_svv = Enthalpy(Water,T=T_swh,P=P_atm) "Vaste entalpie afhangend van die beheerprogram"}
h_ringfeed_return = Enthalpy(Water,T=T_ringfeed_return,P=P_atm) "(Berekening)"
h_munispaliteit = Enthalpy(Water,T=T_munispaliteit_koud,P=P_atm) "(Berekening)"

"------(Die stelsel inset areas en hitteverlieskoeffisiente)
"-----"

h_kante = 0.001394 [kW/(m^2 * K)]
h_top = 0.001437 [kW/(m^2*k)]
h_onder = 0.001437 [kW/(m^2*k)]
q_element = 45 [kW]
Q_losses = 0.2[kW]
"=====
"------(OmgewingsOmstandighede)-----"

P_atm = 86.92 [kPa]
P_target = 86.92 [kPa]
{P_target = 101.058 [kPa] {Durban}
P_target = 101.05 [kPa] {pe}
P_target = 101.46 [kPa] {cpt}
P_target = 85.59 [kPa] {bloem}
P_target = 83.06 [kPa] {whk}
P_target = 87.33 [kPa] {pta}}
density_target =Density(Air_ha,T=25,P=P_target)
density_ref = Density(Air_ha,T=25,P=P_atm)
density_ratio = density_target /density_ref
n = 5
"------(ASHP Specs)-----"
ref_ashp_specs = 90 [kW]
target_ashp_specs = 90 [kW]
ashp_ratio = target_ashp_specs/ref_ashp_specs
m_dot_ashp_max = 0.522 [kg/s]

{TIMESTAMP$='00:00'
maand$ = '1'
dag$ = '1'}

"DWH Supply system losses"
L_d = 250 [m]"Pipe delivery length"
d_o = 0.028 [m] "Outside diameter of pipe"
k_d = (2.6*d_o )+0.2 "Heat loss coefficient of pipe"
N_drawoffs = 7 "Number of draw of pypes"
l_dl = 4 [m] "dead leg length"
d_d = 0.01 [m] "inside diameter of dead leg"
```

Appendix L. Computer code (EES): Reduced system simulation

```
T_swh_i = 20 [deg]
T_swh_pype_i = 20 [deg]
m_solar_bank = 25 [kg]
m_solar_pype = 20

"Die verliese tussen al die verskeie pype"
UA_solar_manifold = 0.015 [kW/k]
UA_solar = 0.08{ 0.058[kW/k]}
{UA_solar_pipes = 0.02 [kW/k]}
UA_tank_pipes = 0.01826 [kW/k]

{N_hp = 1
 N_solar = 1}

"Parametriese tabel berekeninge"
tydverloop = teller *tydstap_grootte
{teller = 1}

"=====
CALL stratifikasieTank3 (activate_heat_pump, electrical_element , T_set_aux , n , P_atm , q_element , T_hp , T_munisipaliteit_koud ,
T_ringfeed_return , T_wb_i , T_db_i , k_water , h_kante , h_top , h_onder , rho_water , cp_water , h_node , area , area2 , T_aanvangs [1..5] ,
m_dot_return , m_dot_swh_aanvanklik , m_dot_hp_aanvanklik , m_dot_in , m_dot_poort_i , m_dot_return_i , n_step , tydstap_grootte ,
teller , G , A_col , T_delta_hoog , T_delta_laag , L_d , d_o , k_d , N_drawoffs , l_d1 , d_d , T_indoors , T_probe_hp , RH , UA_solar ,
UA_solar_manifold , UA_tank_pipes , N_hp , N_solar , T_swh_i , m_solar_bank , T_swh_pype_i , m_solar_pype , density_ratio , m_dot_ashp_max
, ashp_ratio : tyd , stratifikasieTank [1..5] , stratTankEntalp [1..5] , T_return , m_dot [1..5] , m_dot_hp_probe , h_poort_1 , m_swh_nou ,
M_dot_1 , M_dot_2 , M_dot_3 , M_dot_4 , M_dot_5 , h_swh_nou , eta_swh , konduksie_term [1..5] , konveksie_term [1..5] , geforseerdeVloed
[1..5] , Q_dot_munisipaliteit , T_aftap , Q_losses_d , q_probe_hp , Q_losses_dead_end , Q_son_probe , Q_return , Q_top , T_meng , T_top_meng ,
m_dot_hp , m_dot_poort , m_dot_swh , timer_hp , activate_timer_hp , Q_netto , m_dot_ringfeed , T$, cycle$ , Q_hp_probe ,
hp_timer_control_internal , Q_losses_tank , Q_user , Q_user2 , Q_ringmain_normal , Q_losses_tank_pype , COP_hp , Q_elektries_hp ,
COP_hp_potential , Stelsel_entalpie , T_swh , T_room , Q_hp_probe_test , Q_netto2 , Q_netto3 , Q_solar_losses , Q_hello_ashp , die_probe ,
m_dot_hp_swh , m_hello_ashp , diagnose_condition , Q_supplies_energy , Q_in , Q_out , Q_net , Q_ringmain_test , Q_heat_pump_loop_test ,
Sigma_ringmain_massflux , Sigma_ashp_loop , Q_tester , Q_tester_2 , Q_tester_3 , Q_tester_4 , mass_balance_tanks_top ,
mass_balance_tanks_bottom , Q_tank_top , Q_tank_bottom , Q_net , laaste , kode_blok , h_swh_new , Q_son_probe2 , Q_son_losses_manifold ,
h_swh_pype , T_swh_pype)

partone$ = CONCAT$( ' Pretoria_HP ' , String$( N_hp ))
parttwo$ = CONCAT$( partone$ , ' SWH ' )
partthree$ = CONCAT$( parttwo$ , String$( N_solar ))
filepath$ = 'C:\Users\user1\Documents\Meesters\1.) 1D EES model\1) Real data simulations\4) Jan 2020\3) Parametric study\1)
Pretoria '
partfour$ = CONCAT$( filepath$ , partthree$ )
endof$ = ' .csv '
file$ = CONCAT$( partfour$ , endof$ )
{$Export file$ tyd , stratifikasieTank [1..5] , stratTankEntalp [1..5] , T_return , m_dot [1..5] , m_dot_hp_probe , h_poort_1 , m_swh_nou ,
M_dot_1 , M_dot_2 , M_dot_3 , M_dot_4 , M_dot_5 , h_swh_nou , eta_swh , konduksie_term [1..5] , konveksie_term [1..5] , geforseerdeVloed
[1..5] , Q_dot_munisipaliteit , T_aftap , Q_losses_d , q_probe_hp , Q_losses_dead_end , Q_son_probe , Q_return , Q_top , T_meng , T_top_meng ,
m_dot_hp , m_dot_poort , m_dot_swh , timer_hp , activate_timer_hp , Q_netto , m_dot_ringfeed , T$, cycle$ , Q_hp_probe ,
hp_timer_control_internal , Q_losses_tank , Q_user , Q_ringmain_normal , Q_losses_tank_pype , COP_hp , Q_elektries_hp ,
COP_hp_potential , Stelsel_entalpie , T_swh , T_room , Q_hp_probe_test , Q_netto_2}
```

Appendix M

Computer code (EES): Tank heat losses

```
"Geometry of tank"
Diameter_tanks = 1.22 [m]
Volume_tanks = 3 [m^3]
Volume_tanks = ((Diameter_tanks^2)/4)*pi#*Height_tanks
A_outside_cylinder = Diameter_tanks*pi#*Height_tanks
A_inside_cylinder = (Diameter_tanks-((Insolation_thickness+Insolation_thickness_alli)*2))*pi#*Height_tanks
A_top = ((Diameter_tanks^2)/4)*pi#

Geometry_ratio = Diameter_tanks/Height_tanks
Geomtery_ratio2 = 35/Grasshoff^0.25

A_total_tank = A_outside_cylinder + A_top + A_top
A_total_tanks = A_total_tank * 5

Radius_tanks_outside = Diameter_tanks/2
Radius_tanks_inside = (Diameter_tanks/2)-Insolation_thickness_alli
Radius_tanks_inside_in = (Diameter_tanks/2)-Insolation_thickness-Insolation_thickness_alli

"Thermal and geometric properties of tanks"
Insolation_thickness = 0.05[m]
Insolation_thickness_alli = 0.0006[m]
Insolation_conductivity = 0.085 [W/(m*k)]
Insolation_conductivity_alli = 205 [W/(m*k)]

"Convective properties inside and outside"
h_outside = 10 [W/(k*m^2)]
h_inside = 22.56 [W/(k*m^2)]
h_inside_pipes = h_pipe_inside{ 7411 [W/(k*m^2)]}

"Temperature properties"
T_inside = 55 [C]
T_outside = 11.91 [C]

"Thermal resisitances sides"
R_conv_in = 1/(h_inside*A_inside_cylinder)
R_conv_out = 1/(h_outside*A_outside_cylinder)

R_conv_top_out = 1/(h_outside*A_top)
R_conv_top_in = 1/(h_inside*A_top)

R_cond = ln(Radius_tanks_inside/Radius_tanks_inside_in)/(2*pi#*Insolation_conductivity*Height_tanks)
R_cond_al = ln(Radius_tanks_outside/Radius_tanks_inside)/(2*pi#*Insolation_conductivity_alli*Height_tanks)

R_cond_top = Insolation_thickness/(Insolation_conductivity*A_top)
R_cond_top_al = Insolation_thickness_alli/(Insolation_conductivity_alli*A_top)

R_cond_bottom = Insolation_thickness/(Insolation_conductivity*A_top)
R_cond_bottom_al = Insolation_thickness_alli/(Insolation_conductivity_alli*A_top)

Regime_raio = Grasshoff / (Re_tanks^2)

"Heat loss"
q_heat_loss_sides = (T_inside-T_outside)/(R_conv_in+R_conv_out+R_cond+R_cond_al)
q_heat_loss_sides = (T_inside-T_s_i)/(R_conv_in)
q_heat_loss_sides = (T_inside-T_s_i_whoool)/(R_conv_in+R_cond)
q_heat_loss_sides = (T_inside-T_s)/(R_conv_in+R_cond+R_cond_al)
q_heat_loss_top = (T_inside-T_outside)/(R_conv_top_in+R_cond_top+R_cond_top_al+R_conv_top_out)
q_heat_loss_bottom = (T_inside-T_outside)/(R_conv_top_in+R_cond_bottom+R_cond_bottom_al+R_conv_top_out)
Q_TOTAL = (q_heat_loss_sides+q_heat_loss_top+q_heat_loss_bottom) * 5

UA_sides = q_heat_loss_sides/(T_inside-T_outside)
U_sides =UA_sides / A_outside_cylinder

UA_top = q_heat_loss_top/(T_inside-T_outside)
U_top =UA_top / A_top

UA_bottom = q_heat_loss_bottom/(T_inside-T_outside)
U_bottom =UA_bottom / A_top
```


Appendix M. Computer code (EES): Tank heat losses

```
UA = Q_TOTAL/((T_inside-T_outside))
U_tanks = UA/A_total_tanks

"Reynolds and Grasshoff calculaton"
m_dot = 0.387 [kg/s]
rho =Density (Water,T=60,P=80000)
Q_flow = m_dot/rho
V = Q_flow/A_top
mu = Viscosity (Water,T=60,P=80000)
Re_tanks = (rho*V*Diameter_tanks)/mu
k=Conductivity (Water,T=40,P=80000)
h = (3.66)*(k/Diameter_tanks)
beta=VolExpCoef (Water,T=55,P=80000)
beta2=VolExpCoef (Air_ha,T=55,P=80000)

anw = (mu/rho)^1

Grasshoff = (g#*beta*(T_inside -T_s_i)*(((Diameter_tanks/2)^3)))/(mu/rho)^2
Raleigh = Grasshoff*Prandtl
{Nusselt_tank = (4/3)*((Grasshoff/4)^(0.25))*g#*Prandtl}
Nusselt_tank = (0.023)*((Re_tanks)^(4/5))*Prandtl^0.4
h_tank_inside= (k/Height_tanks)*Nusselt_tank

Grasshoff2 = (g#*beta*(T_inside -T_s_i)*(((Diameter_tanks/2)^3)))/(mu/rho)^2
Raleigh2 = Grasshoff2*Pr2
Nusselt_tank2 = (0.825 + ((0.387*Raleigh2^(1/6)))/((1 + (0.492/Pr2)^(9/16))^(8/27))))^2
h_tank_inside2= (k/Height_tanks)*Nusselt_tank2

Pr2=Prandtl (Water,T=50,P=120000)

"Pump calculations"
Q_hour = (m_dot * 3600 [s/h])/rho

"Pipe length calculations"
swh_length_pipe = 34.28[m]
swh_pipe_diameter_i = 0.030 [m]
swh_pipe_diameter_o = 0.055 [m]
swh_vloei_area = pi#*(swh_pipe_diameter_i^2)/4
swv_pyp_volume = swh_vloei_area * swh_length_pipe
swv_pyp_buite_area = swh_pipe_diameter_o*pi#*swh_length_pipe
swv_pyp_binne_area = swh_pipe_diameter_i*pi#*swh_length_pipe

"Pipe length calculations"
tank_length_pipe = 30[m]
tank_pipe_diameter_i = 0.030 [m]
tank_pipe_diameter_o = 0.055 [m]
tank_vloei_area = pi#*(tank_pipe_diameter_i^2)/4
tank_pyp_volume = tank_vloei_area * tank_length_pipe
tank_pyp_buite_area = tank_pipe_diameter_o*pi#*tank_length_pipe
tank_pyp_binne_area = tank_pipe_diameter_i*pi#*tank_length_pipe
m_dot_ringmain = 1.25[kg/s]

Cp = Cp(Water,T=50,P=80000)
Prandtl = (cp* mu)/k

Q_flow_pipe = m_dot_ringmain/rho
V_pipe = Q_flow_pipe/swh_vloei_area
Re_pipe = (rho*V_pipe*swh_pipe_diameter_i)/mu
Nusselt_pipe = 0.023 * (Re_pipe^(4/5)) *(Prandtl)^0.3
h_pipe_inside = (k/swh_pipe_diameter_i)*Nusselt_pipe

"Pipe losses"
R_conv_in_pipe = 1/(h_inside_pipes *swv_pyp_binne_area)
R_conv_out_pipe = 1/(h_outside*swv_pyp_buite_area)
R_cond_pipe = ln(swh_pipe_diameter_o/swh_pipe_diameter_i)/(2*pi#*Insolation_conductivity*swh_length_pipe)
R_cond_pipe_tank = ln(tank_pipe_diameter_o/tank_pipe_diameter_i)/(2*pi#*Insolation_conductivity*tank_length_pipe)
Q_heat_loss_pipes = (T_inside-T_outside)/(R_conv_in_pipe+R_conv_out_pipe +R_cond_pipe)
Q_heat_loss_pipes_tank = (T_inside-T_outside)/(R_conv_in_pipe+R_conv_out_pipe +R_cond_pipe_tank)
UA_pipes = Q_heat_loss_pipes/((T_inside-T_outside))
UA_pipes_tank = Q_heat_loss_pipes_tank/((T_inside-T_outside))
```

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