A method for the seasonal performance rating of a residential water heating heat pump

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Abstract

To save electricity in the South African residential market, energy efficient air source water heating heat pumps have been widely implemented in combination with conventional hot water storage vessels, also known as geysers. The performance of these heat pump installations are significantly influenced by seasonal changes in the surrounding ambient conditions as well as the municipal water supply temperature. As heat pumps are designed and built by different manufacturers, differences in terms of the sub-components used and their specifications are common. As a result, each heat pump model must be tested to determine its energy saving capabilities. From the literature review it became evident that very little research has been done world-wide on the performance verification of residential heat pump water heaters. It was further found that there are currently no standard for the performance testing of a residential heat pump water heater in South Africa.

The aim of this study was therefore to research and develop a laboratory testing methodology that will accurately represent a residential heat pump's in-field performance taking into account the seasonal influences on these systems. In order to reach this objective, the seasonal performance of air source water heating heat pumps were measured and reviewed for different climate regions in South Africa. The measured data was then used to generate general performance curves at different ambient conditions. The performance curves were verified and validated with laboratory tests as well as a Flownex® SE simulation model. The results were then used to determine which factors must be included in a laboratory test to accurately represent the in-field performance. Based on this a proposed laboratory testing methodology was developed.

KEYWORDS: Heat pump, testing methodology, simulation model, performance rating.

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Abbreviations

COP:	Coefficient of performance
EES:	Engineering Equation Solver
HPWH:	Heat pump water heater
NERSA:	National Energy Regulator of South Africa
RH:	Relative humidity
SABS:	South African Bureau of Standards
SANS:	South African national standard
SCOP:	Seasonal coefficient of performance

Nomenclature

A:	Cross sectional area of the restrictor [m^2]
A_{EQ}:	Evaporator heat transfer area [m^2]
A_Q:	Heat transfer area [m^2]
A_{Qs}:	Secondary heat transfer area [m^2]
COP_{inst}:	Instantaneous coefficient of performance [kW/kW].
D_h:	Hydraulic diameter [m]
D_w:	Wall thickness [m]
E_{tot}:	Total electrical energy consumed during the heating cycle [Joules]
L_p:	Fluid path length [m]
N_i:	Number of increments [-]
N_r:	Total number of tubes [-]
P_d:	Compressor discharge pressure [Pa]
P_{elec}:	Amount of electricity consumed [kWh]
$P_{kJ/minute}$	Electricity used per minute [kJ/ minute]
P_s:	Compressor suction pressure [Pa]
Q_{th}	Thermal energy added to the water [kW]
R_I:	Inside surface roughness [µm]
R_s:	Secondary surface roughness [µm]
$T_{a_{in}}$:	Air dry bulb temperature before the heat pump [°C]
$T_{w_{in}}$:	Water temperature into the heat pump [°C]
T_Δ:	Temperature difference [°C].
T_A:	Ambient dry bulb temperature [°C]
T_{End}:	Temperature of the water at the end of the heating cycle [K]
T_{ID}:	Tube internal diameter [m]
T_L:	Tube pass length [m]
T_{Start}:	Temperature of the water at the start of the heating cycle [K]
T_T:	Geyser tank temperature [°C]
T_{WT}:	Tube wall thickness [m]
T_d:	Compressor discharge temperature [°C]
T_{in}:	Water temperature entering the heat pump [°C]

T_{out}:	Water temperature exiting the heat pump [°C]
T_s:	Compressor suction temperature [°C]
T_t:	Heat pump inlet water temperature [°C]
U_{ff}:	Internal energy of the water at final tank temperature [kJ/kg.K]
U_{fs}:	Internal energy of the water at initial tank temperature [kJ/kg.K]
V_d:	Dead volume [m^3]
V_s:	Swept volume [m^3]
V_t:	Volume of test tank [m^3]
W_{elec}	Electricity used per second [kW].
W_{HP}:	Electricity used in total [kWh]
$\dot{m}$:	Water mass flow rate through the heat pump [kg/s]
ρ_f:	Density of water at final tank temperature [kg/m^3]
ρ_s:	Density of water at initial tank temperature [kg/m^3]
COP_{sys}:	System coefficient of performance [kW/kW].
COP:	Coefficient of performance [kW/kW]
Cp:	Specific heat capacity of water [kJ/kgK]
$Fluid$:	R410A
M:	Water mass [kg]
N:	Rotations per minute [rpm]
R:	Surface roughness [µm]
RH:	Relative Humidity [%]
X:	Number of parallel circuits per row [-]
n:	Number of tube passes [-]
x:	Number of stages [-]
β:	Discharge coefficient [-]
η:	Isentropic efficiency [%]

CHAPTER 1 : INTRODUCTION

1.1 Background

The fast increasing price of electricity in South Africa has led to a national drive towards energy efficient technologies and solutions (Carte Blanche, 2012). This is not only true for the industrial and commercial sectors but for the residential sector as well (Winkler *et al.*, 2006:xi). The major drive towards energy efficient technologies started in 2010 when the National Energy Regulator of South Africa (NERSA) approved annual electrical price increases above 24% for 2010, 2011 and 2012 (Eskom, 2010a). With the demand for energy efficient products rising exponentially, the number of companies supplying and installing these products also increased significantly.

The sudden and dramatic growth in the energy efficient product market has also led to consequent problems in this regard. These are mainly due to inexperience and/or a general lack of knowledge about the products, both in terms of suppliers and consumers. Most of the energy efficient products sold in South Africa entered the country as products of high quality already available to the European market. There is, however, speculation that some companies used the opportunity to import low quality products to ensure larger profits. Problems caused by the variance in the performance due to incorrect installation or poor product quality has unfortunately grown to such an extent that social media and consumers are questioning the functionality of these technologies as a whole. The products that have come under the most scrutiny have been residential type heat pumps and solar water heaters (Carte Blanche, 2013).

The conventional electrically heated hot water storage vessel, also known as a geyser, has been shown by several local and international studies to be responsible for the largest amount of electricity consumed in a residential home. It is therefore the most popular item to be replaced with a product using less electricity (Winkler *et al.,* 2006:123). This is where air source heat pump water heaters (HPWHs) can make a big impact in terms of the South African electricity crisis.

With the first vapour compression cycle already designed, patented and built in 1835 (Perkins, 1835:12), the technology used in an air source HPWH is nothing new and has been proven over the course of many years. In more recent times the technology of vapour compression cycles has been improved to a well-designed and highly reliable system. Computers have significantly aided in the advancement of the vapour compression cycle design, as simulation tools are being developed to accurately predict the behaviour of these systems in the real world (Zhang *et al.*, 2007). Some of the biggest advantages being offered by simulation models are that they give a much higher degree of control as well as a quick and effective way of studying changes made to the system. Unfortunately there are also some disadvantages to simulation models. In many cases the complexity of thermal-fluid systems is such that designing a workable simulation model is already a significant accomplishment, and this model is often based on several assumptions. Therefore, sensitivity regarding the design is usually neglected as a whole, as the simulation models are set up to test certain outcomes with regards to specific changes made to the system design (Niemand, 2003).

Although the technology behind water heating heat pumps is well known, there are many different variations in the components used, and applications for vapour compression cycles. The focus of most research conducted on heat pump technology has been on systems using either the ground or water as a heat source, as the winter air temperatures in Europe and America are considered to be too low for air source units. The major technology behind air, water or ground source heat pumps remains exactly the same, but these systems are very different with regards to components used, performance influencing factors and installation methods (Zhang *et al.*, 2007:1). Therefore, it is important to study the performance of every new design and installation method within a specific climate zone.

The South African Bureau of Standards (SABS) has set up a testing standard for new residential water heating heat pumps entering the market, but unfortunately this standard only tests the heat pump’s functionality and not the performance requirements (SABS, 2012). Most manufacturers and suppliers did, however, also submit their residential type heat pump products to the country’s electricity supplier, Eskom, for further testing. Eskom established a requirement specifying that residential heat pumps must perform 2.8 times better than a conventional geyser in order to qualify for their energy efficient reward or rebate (Eskom, 2010b). The method used during these tests was largely based on variances in ambient conditions. It is still unclear whether these tests gave an accurate representation of the actual in-field heat pump performance as there was, for instance, no water drawn from the geyser in the form of tapping profiles during these tests (Eskom, 2010b), as can be found in the British standard BS EN 16147:2011 (BSI, 2011). These tests were, however, discontinued in 2013 (Eskom, 2014) as the rebate given by Eskom was discontinued. This left the country without any comparative tests for residential type heat pump water heaters.

1.2 Problem statement

The number of residential heat pumps installed in the South African market is constantly growing. Along with this, questions are being raised with regards to the actual in-field efficiencies thereof. A thorough literature study revealed that there is currently no local performance test methodology for evaluating a residential heat pump entering the South African market. Internationally the British standard BS EN 16147:2011 (BSI, 2011) is used as a laboratory performance testing methodology, but no research was found indicating whether this test method can be applied or adopted for the South African climate.

1.3 Objectives

The primary objective of the study is to develop a laboratory test methodology from measured and simulated data that would give an accurate representation of a heat pump's seasonal in-field performance.

The secondary objective will be a literature study of the Eskom rebate test methodology and the British standard BS EN 16147:2011 to determine the advantages of each of these methodologies. These advantages will then be considered for the laboratory test methodology being developed as the primary objective.

1.4 Method of investigation

To successfully complete this study the following method will be applied:

Firstly a critical literature study will be conducted. The literature study will serve as a basis for:

i. understanding what has been done;
ii. identifying limitations within this field of study;
iii. determining if successful studies on this topic were done in other countries;
iv. determining what simulation models have been successfully developed within other studies; and
v. evaluating how the Eskom rebate methodology compares to the British standard BS EN 16147:2011.

Secondly, measurement equipment will be used to gather seasonal in-field data as well as data from within a controlled laboratory test environment. The measurement equipment will track the heat pump's performance indicators and major performance influencing factors, as identified in the literature survey.

The heat pump used during the in-field and laboratory tests will thereafter be modelled within Flownex® SE simulation software (Flownex® SE, 2014) to predict the heat pump's performance. The simulation model's results will be compared to performance data from the manufacturer before being used as verification and validation for the above-mentioned test results. All performance influencing factors will then be studied to determine which factors should be closely monitored in the proposed laboratory test methodology.

Benefits found within the Eskom rebate test and British standard will be considered for the proposed laboratory testing methodology. Finally all results and conclusions will be used to develop a proposed laboratory testing methodology for South Africa.

1.5 Limitations of the study

A black box method will be used to simulate the heat pump within the simulation model. Therefore subcomponent details will be limited to that which are required to predict the outlet conditions of the major components using the specified inlet conditions.

Even though the performance of a heat pump water heater directly affects its economic feasibility, this study will exclude any economic analysis. The focus is towards determining a method for performance testing, rather than determining if heat pump water heaters are economically a good investment for different users.

The proposed testing methodology as listed in Chapter 7 will be limited to the thermal performance test of the HPWHs, excluding general requirements, marking requirements and safety requirements on HPWHs.

CHAPTER 2 : LITERATURE SURVEY

2.1 Introduction

This chapter reviews applicable literature to gain insight into the factors influencing the performance of a residential heat pump water heater and how it can accurately be measured, as identified by other studies. The main topics of the literature survey are as follows:

a. A review of different heat pump technologies.
b. Performance rating of an air source heat pump water heater.
c. Performance influencing factors.
d. The Eskom rebate test methodology.
e. The British standard for testing a residential heat pump.
f. Optimisation of an air source heat pump.
g. Numeric modelling of an air source heat pump.
h. Linear regression of in-field data.

2.2 A review of different heat pump technologies

A heat pump water heater (HPWH) operates on an electrically driven vapour compression cycle and pumps energy from air in its surroundings to water in a geyser, thus raising the temperature of the water. This definition was given to HPWHs in a review study completed by Hepbasli and Kalinci (2009). Their study aimed to investigate why residential HPWH units have been available for more than 20 years, but only had limited success in certain markets. The study focused on reviewing HPWH systems in terms of energy and exergy aspects. HPWH technology along with its historical development was considered by the authors before a comprehensive review was completed on previous studies. HPWHs were then numerically modelled for performance purposes by using the energy and exergy analysis methods found in the reviewed studies.

The study by Hepbasli and Kalinci (2009) states that heat pumps are heat generating devices that can be used to heat water or air for either residential or commercial applications. A HPWH is a promising technology and uses the same mechanical principles as refrigerators and air conditioners - the only difference being that refrigerators and air conditioners are primarily used to extract energy and then discharge it into the surroundings as waste product, while heat pumps extract energy from the surroundings and use it as the primary product (Harris *et al.*, 2005). A heat pump can also be described as a machine that transfers heat from a heat source to a heat sink by employing a refrigeration gas cycle. Figure 1 illustrates the workings of a simplified HPWH system. The major advantage of this technology is its financial attractiveness to consumers as the technology offers an average of 66% reduction in energy consumed to heat water compared to heating water with a traditional electrical resistance element (Zhang *et al.*, 2007).

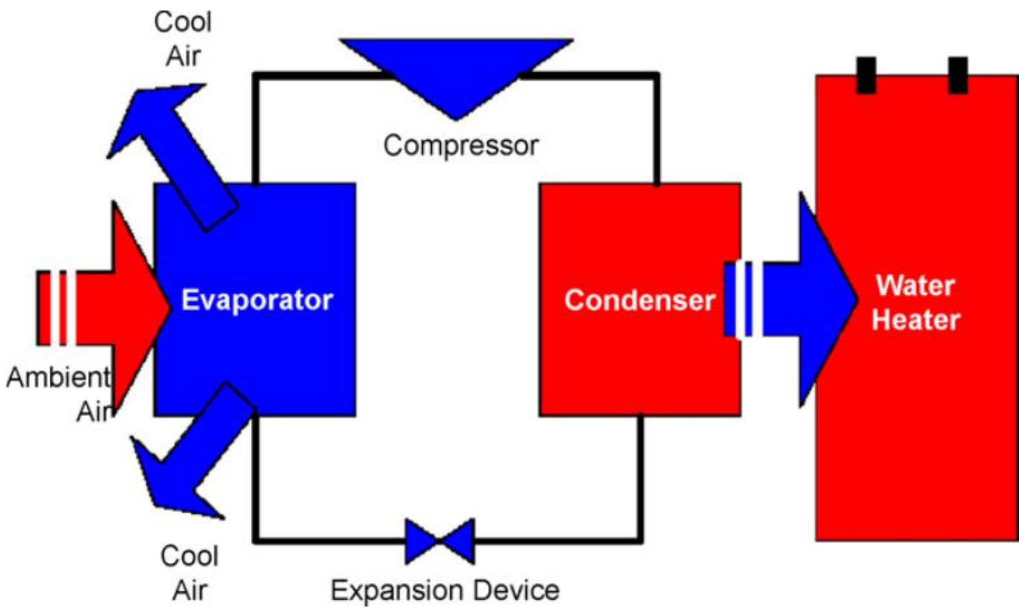

Figure 1: Schematic of a simplified HPHW system by Hepbasli and Kalinci (2009).

The Coefficient of Performance (COP) is a single value summarising the heat pump's ability to reduce electrical consumption. This value can be calculated by dividing the amount of energy required to heat the water by the electrical energy consumed to heat the water. This COP value is dependent on multiple factors, such as i) the temperature of the water, ii) the refrigerant used in the vapour compression cycle, iii) the quality and characteristics of the components used within the gas cycle, and iv) primarily the energy available within the surroundings (Hepbasli & Kalinci, 2009). Since 1950, research has been performed on HPWHs in an attempt to increase the COP. The research involved studying the components within the cycle, thermodynamic properties within the refrigeration gas cycle and the performance influencing factors. The later studies focused on how to minimize the losses and optimize the heat transfer capabilities (US Department of Energy, 2007) .

The air source heat pump simulated in the study of Hepbasli and Kalinci (2009) is a combination unit. The heat pump is located on top of the water tank with a refrigerant coil (running into the geyser) being used as the condenser. The heat pump and hot water storage tank is therefore integrated to function as a single unit. This type of heat pump is highly efficient as there are very little to no pipe losses between the heat pump and the storage vessel. The study also pointed out, however, that the major disadvantage of this type of heat pump is that the entire system will need to be replaced if either the heat pump or water tank should fail.

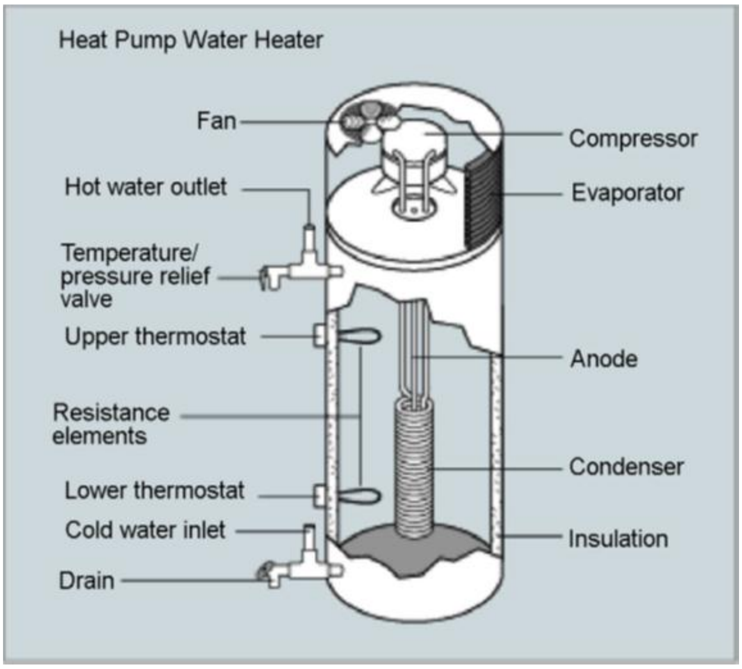

Figure 2: Heat pump system layout by Hepbasli and Kalinci (2009).

The study ultimately showed that the performance of a heat pump can be summarized within a single seasonal coefficient of performance (SCOP) value, but also that there are multiple factors that can influence the SCOP significantly.

2.3 Performance rating of an air source heat pump water heater

Before one can determine the SCOP of a heat pump, it is important to know how performance can be measured accurately. An approach suggested by Ito *et al.* (1999) was successfully implemented in the study by Morrison *et al.* (2004), showing that various primary and secondary influencing factors must be taken into account in order to accurately determine the in-field SCOP of a HPWH. Similar to solar geysers, heat pumps are primarily dependent on the energy available from the sun as shown by almost all studies done with regards to heat pump performance (CSA, 2005). Results within most studies were therefore represented as the COP in relation to the ambient temperature as can be seen in the study of Zhang *et al.* (2007).

As mentioned above, the most important contribution made by the study of Ito *et al.* (1999) is a method considering not only the primary factor of air temperature influence, but also secondary influencing factors. The secondary influencing factors taken into account by this study were relative humidity (RH), water usage patterns and the temperature of municipal water entering the hot water system. The study pointed out that the secondary factors are not only far from negligible, but that these factors should be included in future studies as major influencing factors due to the substantial effect it has on the heat pump performance.

The study by Morrison *et al.* (2004) focused on a method of accurately determining the performance of an air source HPWH during seasonal climate changes. The study aimed to find a method of taking into account the primary and secondary influencing factors, as it was found that in-field data showed an unexpected lower COP than experimental laboratory tests. As in-field tests and testing equipment are very expensive and somewhat unreliable, the study used a “black box” method for testing the heat pump installation. This method entails using only inlet and outlet conditions of the heat pump installation to determine the performance of the system. The advantage of such a method is that less measuring equipment are required and therefore higher quality equipment can be used. The accuracy of the installed equipment will therefore be higher and a smaller equipment measurement error will be applicable to the final COP results. This method also requires no technical knowledge of the internal components of the heat pump, refrigeration gas being used or heat losses throughout the system, as all the internal performance influencing factors are summarized by these inlet and outlet conditions. This method is therefore very efficient in determining an accurate performance factor, with little effort.

The critical factors to be measured include:

i. the power consumed by the unit;
ii. the geyser temperature;
iii. the amount of water drawn from the geyser to the house;
iv. the ambient temperature;
v. the water temperature entering the geyser; and
vi. the RH of the air.

The disadvantages of the black box method are that minor internal component inefficiencies or failures could influence the results without the error being picked up. This can be prevented by verifying the results with a simulation model and/or multiple in-field test sites.

The result obtained by Morrison *et al.* (2004) primarily features a single graph summarizing the results obtained at different environmental conditions. The graph is set up to show the relationship between the COP and the temperature difference between the tank temperature and the ambient temperature. By plotting the graph of COP against the temperature difference rather than just the ambient temperature alone, all the major influencing factors are taken into account. Figure 3 gives the single summarising plot obtained by Morrison *et al.* (2004).

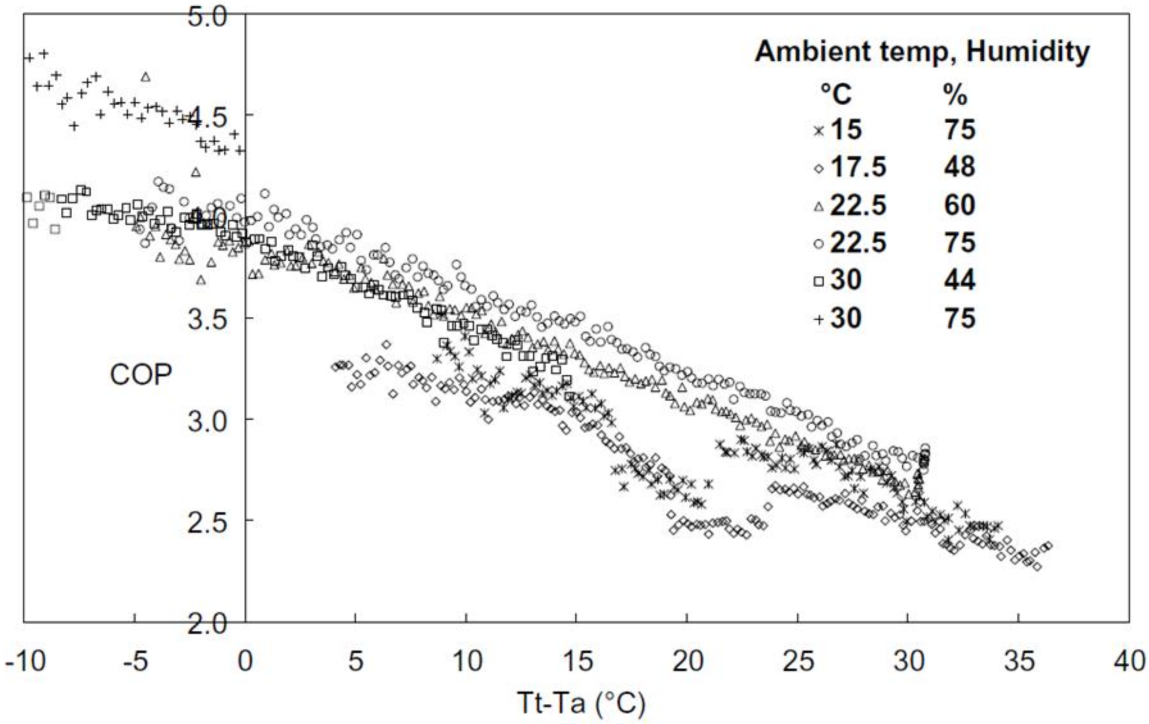

Figure 3: Results plot by Morrison *et al.* (2004).

By taking the secondary influences into consideration, the graph is able to show the information in a more linear perspective with a smaller standard deviation. The major secondary influence normally neglected is the water's ability to absorb energy from the gas cycle. The water's ability to absorb energy decreases substantially at higher water temperatures. Therefore, if small amounts of water are drawn from the tank, the heat pump will reheat the water at higher inlet water temperatures leading to a low performance factor, even at high ambient temperatures. *Morrison et al.* (2004) conclude that the differences in experimental tests and in-field tests can be contributed to the fact that experimental tests do not consider smaller tapping profiles although this is commonly found in a residential environment.

To conclude, the Morrison *et al.* (2004) study described an accurate and efficient method for testing and interpreting in-field heat pump performance data. It further developed a graphical method to summarize the results obtained from tests so that it can be easily interpreted. This method will be used further in this study to represent the results obtained.

2.4 Performance influencing factors

A study conducted by Palmiter *et al*. (2011) studied the effects of improper refrigeration charge and air flow. This was done after a study by Mowris *et al*. (2004) showed a reduction in the performance of air-conditioners and suggested that a similar problem can most likely be found within air source heat pump units.

The study conducted by Palmiter *et al*. (2011) found that the performance of a heat pump is not only dependent on the ambient conditions but is also influenced by factors such as an incorrect refrigeration charge or a reduction in air flow through the heat pump. The object of the study was to measure the effects of improper air flow and refrigeration charge on the seasonal performance of an air source HPWH. The tests were conducted for three different refrigerant charges at 75%, 100% and 125% of nominal value, as well as two air flow rates at 75% and 100% of the rated airflow. In addition, tests were conducted in six climate zones to estimate the SCOP of the heat pumps running under varying conditions.

The results by Palmiter *et al.* (2011) showed that a heat pump with a refrigerant charge varying from 25% below to 25% above nominal value could show a decrease of as much as 20% or an increase in performance of 5%, depending on ambient conditions. Results also indicated that heat pumps with an accumulator at the compressor inlet shows relatively no change in performance. The results of a 25% reduction in air flow through the heat pump unit showed that there is a 3% decrease in performance for dry surface conditions, but a decrease of as much as 8% for wet surface conditions. Results further indicated that the performance is not linearly decreased as air flow is reduced, but rather that there is a sharp decrease in performance at 20% to 25% reduction in air flow.

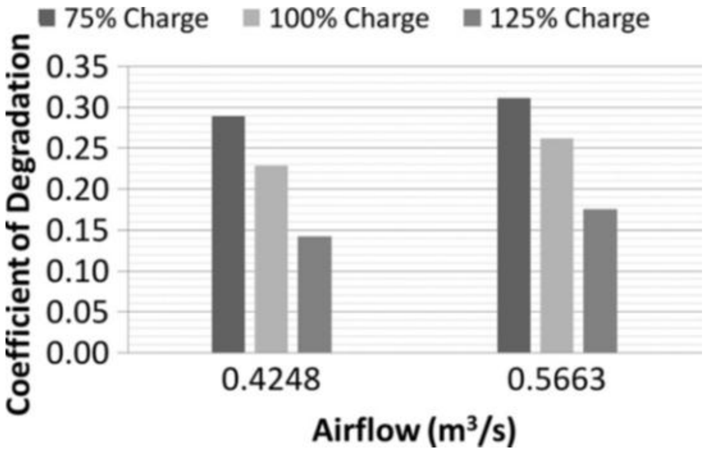

Figure 4: Coefficient of degradation against air flow rate by Palmiter *et al.* (2011).

It was found that all heat pumps considered for this study have an accumulator before the compressor, therefore removing the sensitivity to refrigeration charge as was found in the study of Mowris *et al.* (2004). Further analysis of the events causing a 20% or more air flow reduction suggested that it should be noted within a service schedule but not specifically studied in a method to determine heat pump performance in-field. Therefore it was decided to ignore these factors within the current study as it is unlikely to have a major effect on performance.

2.5 The Eskom rebate test methodology

A procedure for testing residential HPWHs was developed by Eskom in consultation with SABS in 2012 (Eskom, 2012). The following tests and verifications formed part of the procedure:

- A thermal efficiency test.
- Compliance in terms of Eskom's rules as set out in 2010 (Eskom, 2010b).
- Acceptability of the installation in accordance with the local South African National Standard (SANS) regulations.

The Eskom test was not compulsory for heat pump manufacturers and importers. However, as mentioned in the introduction, only heat pumps that have passed this test were eligible for the energy efficiency rebate. This is because the test required heat pumps to have a COP of 2.8, therefore using 66% less electrical energy than a conventional geyser. Not all the heat pump suppliers in South Africa submitted their heat pumps for testing, but most of the popular residential heat pump brands did. The tests also required that a data pack be submitted for evaluation to confirm provisional compliance with all requirements. These requirements are stipulated in the Eskom rebate test methodology (Eskom, 2010b). As part of the evaluation, the heat pump supplier was responsible for the installation of their heat pump at the SABS test premises.

Before the test was started, the installation was inspected to confirm that the heat pump unit and the way in which it was installed complied with:

- All aspects of the product specification supplied to SABS.
- Applicable requirements from the Eskom rebate scheme.
- Selected installation requirements as set out by SANS 10252-2: *Water supply and drainage for buildings.*

The physical testing phase would follow after this inspection and included the thermal performance test applicable to this study.

2.5.1 Test room

A test room complying with the specifications as set out by the Eskom rebate test methodology was used for all tests. The atmospheric dry bulb temperature and RH inside the test room was controlled so as to produce conditions stable enough for accurate testing.

2.5.2 Storage tanks

Two variations of tanks were supplied by the testing authority for the test phase, namely a 300 litre or a 500 litre tank, depending on the heat pump's application in the market. These tanks were fitted with the ports and connections most commonly found on commercially available geysers. These tanks were, however, not be fitted with any internal components such as elements, strainers and anodes.

2.5.3 Tank temperature verification equipment

To verify the water temperature inside the tank, the total volume of water was mixed by circulating it through an external circulation pump. This circulation loop was isolated during testing. The split systems equipment were installed according to a specific diagram as listed within the Eskom rebate test specification.

2.5.4 Testing conditions and equations

The test procedure were repeated for 5 different ambient dry bulb temperature conditions namely 5°C, 15°C, 25°C, 35°C and 45°C. During the test, these temperatures were controlled at an accuracy of ± 3°C while the RH was controlled at 50% ± 7%. The initial water temperature for each test was 30°C ± 1.5°C. When these conditions were confirmed to be stable, the measuring phase of the test would start. The initial water tank temperature was recorded and thereafter the heat pump would be activated to heat the geyser.

When analysing the test condition accuracy specified above it is evident that it allows for too much of a variance especially at lower temperatures. With the allowed variance it is found that for a heat pump tested during the 5°C test, the air temperature is allowed to vary from 2°C to 8°C as the specification states ± 3°C. The specification further allows for a 7% variation on RH resulting in a range from 43% to 57%. If the same heat pump unit is submitted to the test by two different suppliers the test results can vary significantly if one was tested at 2°C and 43% relative humidity while the other was tested at 8°C and 57% relative humidity. The test conditions as listed here will be taken into consideration and updated accordingly in the proposed testing methodology.

During the test the following data were continuously recorded:

1. T_{in}: Water temperature entering the heat pump [°C].
2. T_{out}: Water temperature exiting the heat pump [°C].
3. $\dot{m}$: Water flow rate through the heat pump [kg/s].
4. P_{elec}: Amount of electricity consumed [kWh].

This information was used primarily for calculating the instantaneous COP with the following equation:

$$COP_{inst} = \dot{m} \times Cp \times \frac{(T_{out} - T_{in})}{P_{elec}} \qquad [2.1]$$

COP_{inst}: Instantaneous coefficient of performance [kW/kW].

$\dot{m}$: Water mass flow rate through the heat pump [kg/s].

Cp: Specific heat capacity of water [kJ/kgK].

T_{out}: Temperature of the water at the outlet [K].

T_{in}: Temperature of the water at the inlet [K].

P_{elec}: Electricity used per second [kW].

with Cp taken as a constant of 4.184 [kJ/kg-K].

The heat pump was monitored until it reached the target water temperature of 55°C. Once the heat pump indicated that the target temperature had been reached, the tank was mixed again. The temperature of the water tank was then recorded, for the calculation of the system COP. The following equation is then used to calculate the system COP of the heat pump being tested.

$$COP_{sys} = \frac{V_t \times [(\rho_f \times U_{ff}) - (\rho_s \times U_{fs})]}{E_{tot}} \qquad [2.2]$$

With:

COP_{sys}: System coefficient of performance.

V_t: Volume of test tank [m^3].

ρ_f: Density of water at final tank temperature [kg/m^3].

ρ_s: Density of water at initial tank temperature [kg/m^3].

U_{ff}: Internal energy of the water at final tank temperature [kJ/kg.K].

U_{fs}: Internal energy of the water at initial tank temperature [kJ/kg.K].

E_{tot}: Total electrical energy consumed during the heating cycle [Joules].

The values of these two performance indicating factors were recorded for all 5 ambient test temperatures. After all 10 values had been recorded, the average instantaneous COP and the average system COP was calculated and checked for compliance to the 2.8 COP standard, as defined by Eskom to qualify for the rebate programme.

All expenses for the tests conducted were for the suppliers' account. Since the rebate programme was discontinued, suppliers have no reason for subjecting their heat pumps to this performance verification test any longer. South African consumers are therefore left with claims made by suppliers when comparing heat pumps in the South African market, without any independent proof or verification of the indicated performance. It is therefore possible for a supplier to select a competitive performance value irrespective of the product's true energy saving capabilities, thereby misleading the market and the consumer.

2.6 The British standard for testing a residential heat pump

The British or European standard listed as BS EN 16147:2011 (BSI, 2011) is an example of the standard currently lacking in South Africa. This standard lists the requirements for testing and marking of a residential HPWH's SCOP. This standard states that the markings on the unit shall consist of at least the information as required by the safety standard, and if any performance results are given it shall be given according certain criteria as illustrated in Table 1 below. It further states that only the main results of the tests as set out in Tables 7 to 11 within the British standard BS EN 16147:2011: *Heat pumps with electrically driven compressors — Testing and requirements for marking of domestic hot water units* (BSI, 2011) may be indicated. An example of such a table can be found later on in this study in Chapter 7, Table 5.

Table 1: Presentation of main results according to BS EN 16147:2011 (BSI, 2011).

Result	Unit
Heating up time	h:min
Heating up energy input	kWh
Standby power input	W
Class of the measured tapping cycle and the determined electrical energy consumption W_{EL-TC} for each measured cycle	No. kWh
COP_{DHW} and class of used tapping cycle	-
Reference hot water temperature	°C
Maximum quantity of usable hot water	l
Temperature operating range: Minimal and maximal heat source temperature, minimal start and maximal mean temperature domestic hot water	°C

By using the same testing procedure and labels on all the residential heat pumps in the European market, consumers can quickly and accurately compare the products available in the market. The testing procedure set out within the British standard consists of multiple hot water tapping profiles. These water tapping profiles are set up to simulate the average water consumption of a small to extra-large household over a 24 hour period. These tapping profiles are at different flow rates and water volumes, and attempts to simulate a typical tapping profile found in a residential home ranging from washing hands with hot water to taking a full bath. These tests were specifically set up to determine the energy consumption and performance of a heat pump during hot water heating in a residential environment.

The one factor that is not considered within the British standard tests is the varying seasonal climate that these heat pumps will be running in. These seasonal changes are deliberately set as a constant as the test is used to compare one heat pump with another. It is therefore unnecessary to determine and calculate the exact performance of the heat pump at different seasonal conditions. The only important factor is that the heat pumps being tested are tested at the same seasonal conditions with only the heat pump as variance between tests. The test conditions set out can be found in Table 2 and Table 3 below. This test method represents the heat pump's performance very well, however, it is only for one climate condition. Sensitivity to low and high ambient temperatures are therefore neglected as a whole, even though it can drastically reduce a heat pump's COP.

Table 2: Test conditions applicable to all systems tested according to BS EN 16147:2011 (BSI, 2011).

Measured variable	Set value
Power supply voltage	Rated voltage
Power supply frequency	Rated frequency
Air flow rate on the heat source side	Nominal, as indicated by the manufacturer. When only a range is given, tests are to be carried out at the maximum value.
Temperature of the incoming cold water (°C)	10
Hot water flow rate (l/min)	4 or 10 (see Table 6)

Table 3: Test conditions for particular types of systems according to BS EN 16147:2011 (BSI, 2011).

Type of heat source	Heat source Air[a] temperature in °C	Heat source inlet/outlet or bath[b] temperature in °C	Range of ambient temperature of heat pump in °C	Ambient temperature of storage tank in °C
Outside air heat pump (placed indoor side)	7 (6)		from 15 to 30	20
Outside air heat pump (outdoor side)	7 (6)		heat source temperature	20
Indoor air	15 (12)		heat source temperature	15
Exhaust air	20 (12)		from 15 to 30	20
Water		10 / 7	from 15 to 30	20
Brine		0 / -3	from 15 to 30	20
Direct evaporation		4	from 15 to 30	20

[a] All air temperatures are dry-bulb with wet bulb temperatures in brackets.
[b] Brine mean bath temperature for direct evaporation testing.

The test method used by the British standard BS EN 16147:2011 (BSI, 2011) can be applied at the different environmental conditions found in a particular country to determine a linear regression function to be used for calculating an accurate SCOP. This SCOP function can be used to determine either the performance at a specific climate or the average performance of a region in a country.

Currently the performance figures given by the manufacturers in South Africa are not only measured at different environmental conditions, but also hardly mentioned. It was further found that the test procedures listed by the manufacturers mainly test for differences in ambient dry bulb temperatures without considering any other influential factors. It is therefore suggested that South Africa should adopt the British standard’s method or implement a similar method, assisting consumers in choosing a product of high quality and performance. This is commonly found within the European market for other energy efficient products as well, such as refrigerators and ovens.

2.7 Optimisation of an air source heat pump

Some of the most detailed simulation models on residential HPWHs can be found in optimisation studies. The simulation models found within these studies can be applied to a study on seasonal performance rating either directly or with minor changes. This is because a study on optimisation also focuses on modelling the current performance of a heat pump system with any sensitivity included, before attempting to optimise the system.

In 2011 Guo *et al.* set out to conduct an optimisation study on an air source heat pump installation. As the study aimed to optimise the installation, the heat pump design, installation methodology, operating strategy and the controlling logic would need to be incorporated into the model. The investigation methodology consisted of the construction of an experimental test setup for initial tests on the original heat pump layout, as well as final tests on the system once it had been optimised. In addition to the experimental test setup, a simulation model was developed with the help of the initial test results to study changes made to the system without spending large amounts of money to physically implement every suggested change.

The experimental setup consisted of the heat pump unit fitted within an environmental test chamber connected to a water tank located on the outside. The heat pump used in this test had an expansion valve, evaporator and compressor within the unit while the condenser formed part of the tank. This setup can be seen in Figure 5 below.

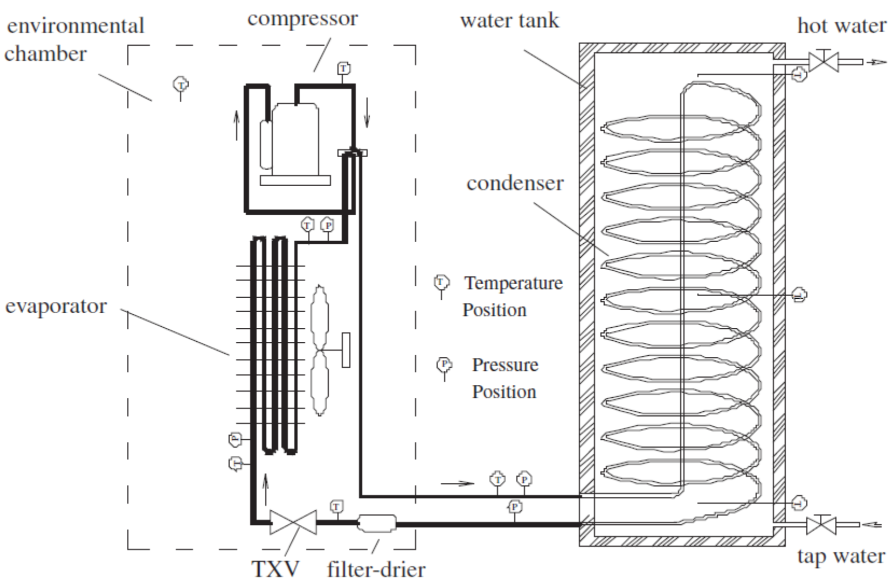

Figure 5: Experimental heat pump setup (Guo *et al.*, 2011).

The figure also indicates the position of the temperature and pressure measurements used to monitor the performance of the heat pump. The test chamber had the ability to simulate ambient conditions between 5°C and 40°C, and these conditions were used to set up a performance function indicating COP at different ambient conditions. The measurements to determine the COP involved measuring the total power consumed during the heating cycle as well as the total energy added to the water in the tank. The single COP value can then be determined by using the following equation:

$$COP = \frac{C_p M(T_{End} - T_{Start})}{W_{HP}} \qquad [2.3]$$

COP: Coefficient of performance [kW/kW].

M: Water mass [kg].

Cp: Specific heat capacity of water [kJ/kg-K].

T_{End}: Temperature of the water at the end of the heating cycle [K].

T_{Start}: Temperature of the water at the start of the heating cycle [K].

W_{HP}: Electricity used in total [kWh].

with Cp taken as a constant of 4.184 [kJ/kg-K].

This method uses the heat pump installation as a black box, thus only recording the major input and output values. The performance indicators recorded within this method are influenced by all the components and losses in the HPWH installation, but the detail regarding each sub-component of the installation is not required. This method therefore gives an accurate representation of the COP using relatively basic equations and requiring little effort. The disadvantage of this method is that the detail with regards to varying performance due to tank temperature or even unintentional errors or influences is not captured, displayed or evaluated leaving the final performance COP value vulnerable to installation or recording errors. As the study by Guo *et al.* (2011) aimed to use the experimental data for verification only, this method proves to be an effective and accurate way of studying the performance of an entire installation compared to other methods.

A basic numerical simulation model was then set up by Guo *et al.* (2011) to model the experimental results obtained. The numerical model uses mass, momentum and energy balance relationships to model the gas cycle, with additional heat transfer equations to account for the losses to the atmosphere. Compared to other studies, this numerical model is very basic but was able to generate results within a maximum variation of 9.8% from experimental results. The study by Guo *et al.* (2011) therefore showed that some degree of accuracy is fairly easy to obtain with the advanced thermodynamic relationships found in literature today, but also that every percentage of accuracy beyond this point is only gained by exponentially increasing the details within the simulation model, the depth of research and the total effort.

Experimental results indicated that the average COP ranged from 2.82 to 5.51 under typical climate conditions (Guo *et al.,* 2011). The study further concluded that the optimal starting time for the heat pump is between 12:00 and 14:00 with the allowable running time extending to 22:00, if there is no electrical price difference to take into account. The assumption is therefore made that the client has sufficient hot water capacity installed to supply hot water to a house from 22:00 to 12:00 the next day without reheating the water. The study further looked at set temperatures for different climate zones concluding that the set water temperature should be 46°C in the summer and 50°C in colder months due to the decrease in municipal water temperatures. However, there is talk in industry of a new regulation that is expected to be implemented in the near future. The expected regulation will require hot water to be stored at 60°C; due to possible bacterial growth at lower temperatures. The uncertainty regarding this implementation of this regulation excludes these temperatures from this study.

2.8 Numeric modelling of an air source heat pump

In all air source heat pump applications, performance of the heat pump is significantly influenced by instantaneous ambient air temperatures. A study done by Yokoyama *et al.* (2005) focused specifically on the influence of ambient conditions on a heat pump's performance. The study also indicated that doing in-depth tests in a physical test environment can be very expensive and time consuming. They therefore used a method of measuring only key variables in a system with a test setup and then using these measured values in an in-depth numerical study to calculate the rest of the unknown parameters. The measured parameters in the study consisted of the water temperature in the geyser, water temperatures at the heat pump inlet and outlet, air temperatures at the heat pump inlet and outlet, the temperature of the gas leaving the compressor, the temperature of the gas entering the evaporator and the duration of the heating cycle. Some of the key measured values such as heating time were not used within the design of the numeric model components but were later used for verification of the numeric model results.

The heat pump installation used in their study is very similar to the typical installations used in South Africa and can be seen in Figure 6.

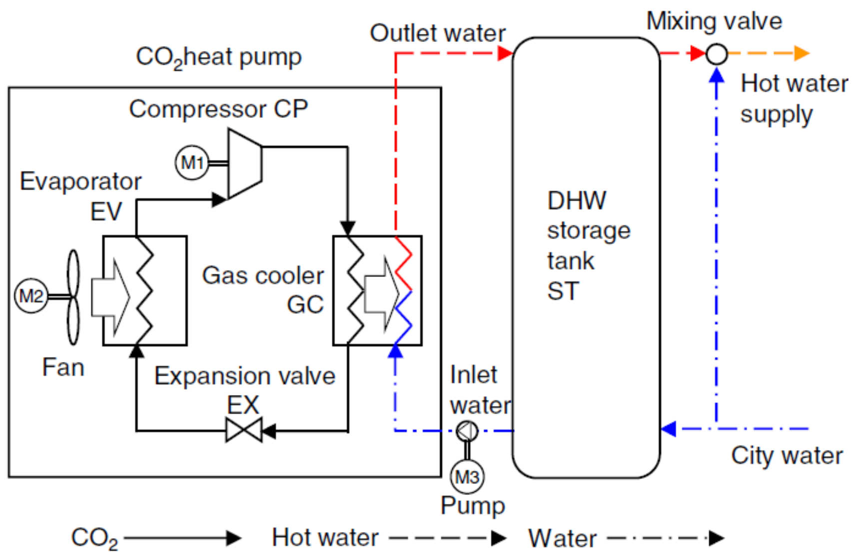

Figure 6: CO_2 heat pump test setup (Yokoyama *et al.*, 2005).

The figure above shows a split type or retrofitted residential heat pump water heater with the water pipes connecting the heat pump unit to the geyser. The heat pump in the figure indicates four gas cycle components to be simulated, namely the evaporator, compressor, gas cooler (also referred to as a condenser) and the electronic expansion valve. These gas cycle components where modelled along with the water storage tank to numerically represent the entire installation.

Numerical models can vary significantly with regards to the detail simulated within each component. If insufficient details are included, the accuracy of the model will decrease, thus skewing the results. However, it is also possible to include unnecessary detail that takes up a lot of time and effort to research and incorporate into the model without a noticeable increase in accuracy. The study conducted by Yokoyama *et al.* (2005) is one of the most detailed numerical studies that were found in the available literature. The study set out to model each of the heat pump's components numerically by considering mass, momentum and energy balance relationships. The outlet conditions from each component were assumed to be the inlet conditions for the next component in the gas cycle, with the losses between components considered negligible. The HPWH refrigeration cycle was then set up to loop while functions controlling the ambient temperature and water temperature boundary conditions act as external influencing factors.

The numerical model set up by Yokoyama *et al.* (2005) was then used to keep all parameters near constant while varying the ambient temperature for each simulated heating cycle. This ensures that the changes in performance can be contributed to the change in ambient temperatures. The results obtained in the study showed that the calculated values are within 8% of experimental results for all the simulated ambient conditions. Due to the 3°C air temperature variance during the laboratory test conditions, the accuracy of 8% was deemed adequate to validate the simulation model.

The only disadvantage of the approach taken was that the results were generated with the tank temperature changing due to a numerically fitted equation rather than with the energy generated by the heat pump. This is because only the influences of ambient air and water temperatures on each component in the system were simulated.

Most components in a heat pump can be modelled to various degrees as was seen in the study of Yokoyama *et al.* (2005). The detail required within a simulation model is therefore determined by the accuracy requirements from the simulation model. The model that will be developed in the current study is required to verify and validate the effects of influencing factors seen on the measurements obtained in laboratory tests and from in-field data. The secondary purpose of the simulation model is to determine the error between the simulation model and laboratory tests, when only major influencing factors are taken into account.

McKinley and Alleyne (2008) studied various internal influencing factors to ultimately model a refrigeration gas cycle more accurately. The study focused mainly on developing a dynamic evaporator model. The study accurately states that all vapour compression cycles are essentially heat management devices and that heat exchanger models have the largest effect on the simulation's accuracy. The study furthermore states that models including a loop or cycle such as a residential heat pump must be able to run in real-time. The study proposed a lumped parameter or moving-boundary heat exchanger model for accurately simulating primarily the evaporator of a heat pump water heater. The accuracy of the model was increased further than previous studies by including the finned surfaces, non-linear air temperature distribution and non-circular passages. The model can therefore be used for single pass and cross-flow heat exchangers as found within heat pump evaporators.

The study by McKinley and Alleyne (2008) ultimately used a mathematical model to simulate the entire refrigeration cycle as found in a residential HPWH with special emphasis on the evaporator as it dictates the amount of energy entering the refrigeration cycle. Figure 7 portrays the evaporator from a mathematical viewpoint. The time varying inputs or boundary conditions are the air mass flow rate, the air inlet temperatures, the air humidity, the refrigerant inlet and outlet mass flow rates and the inlet enthalpy. As the evaporator model forms part of a vapour compression simulation, these inputs will therefore be provided by the boundary conditions from the models of the components before and after the evaporator, namely the expansion valve and the compressor respectively.

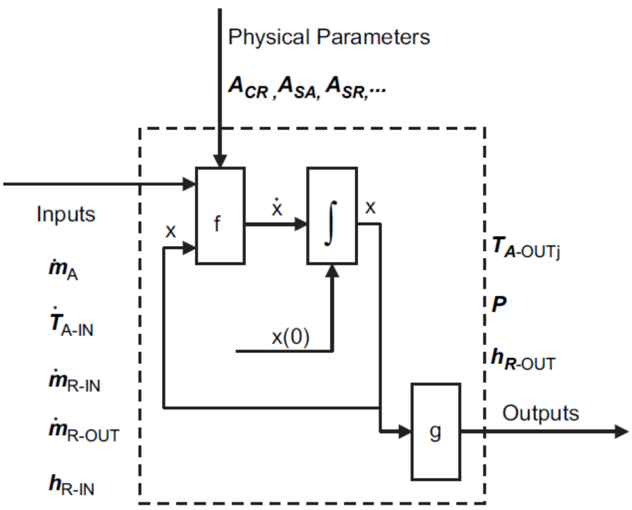

Figure 7: Mathematical evaporator model (McKinley & Alleyne, 2008).

To derive the functions needed to solve the model, the heat exchanger is divided into control volumes or zones that can vary with time and are tracked by the model, hence the term moving boundary method. Figure 8 shows that the evaporator is divided into three zones, namely the superheated, two-phase, and sub-cooled regions, therefore distinguishing on the basis of refrigerant gas phases.

As the refrigerant changes from a liquid to a gas phase, its ability to absorb energy from the air changes as well. This study therefore accurately simulated an evaporator capable of taking into account the phase change of the refrigerant and the effects thereof in a dynamic real time model.

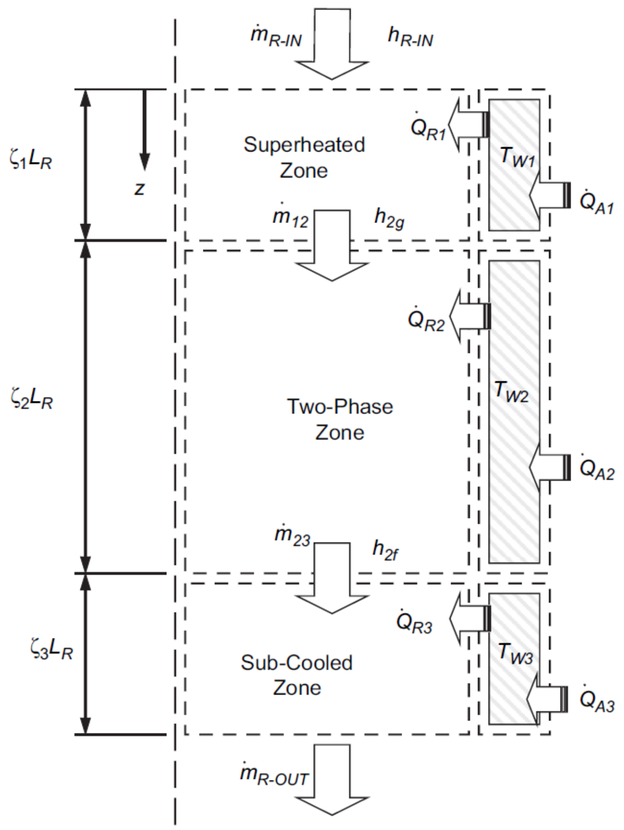

Figure 8: Evaporator schematic (McKinley & Alleyne, 2008).

This study highlighted the extensive details that can be applied to components within a basic numerical simulation model as well as the high degree of accuracy that can be obtained from such a model. The results from Yokoyama *et al.* (2005) and McKinley and Alleyne (2008) were evaluated and it was determined that the accuracy of Yokoyama *et al.* (2005) would be sufficient for this study, as the increase in accuracy from the study by

Yokoyama *et al.* (2005) to that of McKinley and Alleyne (2008) is marginal and that this increase in accuracy was obtained only with a significant amount of additional research, effort and simulation model complexity that would be unnecessary for the current study.

2.9 Linear regression of in-field data

A study conducted by Huchtemann and Müller (2012) analysed in-field test data in order to compare several types of German heat pumps. The tests contained 77 heat pump systems, but after recording data for two years only 43 test sites yielded usable data. Of the 43 units that were tested, 21 were air to water heat pumps, 17 were brine to water heat pumps with horizontal ground source exchangers and five were brine to water heat pumps with vertical ground source heat exchangers. These systems were evaluated during 2008 and 2009 in various locations within Germany's various climate regions. The study found a mean SCOP of 2.3 for air source heat pumps and 2.9 for ground source devices with no real difference in performance with regards to vertical or horizontal ground source heat exchangers. It is important to take into account the differences between the South African and German climates. Berlin, Germany has an average high ambient temperature of 13°C compared to Pretoria, South Africa with an average high ambient temperature of 23°C (Climate data, 2015).

The study compared and evaluated the heat pumps by identifying heating curves generated from linear ambient air temperature dependent regression functions for each of the 43 heat pumps used by Huchtemann and Müller (2012). These linear regression functions were then overlaid on the same plot to quickly and effectively study the performance. After the performance of the heat pumps were studied on a high level, the detailed data was analysed to determine what caused the differences in performance. The use of regression functions to accurately track the performance of an in-field installed heat pump system showed adequate accuracy and efficiency. The accuracy of this regression function is, however, influenced by the accuracy of the data recorded and the factors taken into account within the equations used to generate the performance factor or COP.

The performance factor within the study by Huchtemann and Müller (2012) was defined as the quotient of the heating energy (Q) supplied by the heat pump per unit of electrical energy (W) used by the heat pump. The study showed, in conclusion, that one air source heat pump site had a performance factor of 3 compared to a normal geyser element. This illustrated the potential of this technology even in cold climate zones such as Germany. In addition, the difference between the mean and maximum performance factor found between sites points out the necessity for optimisation with regards to timers being used to control the runtime of a heat pump. From the site data it was found that the performance factor for a site is much higher when timers are set to dictate the allowable running time in terms of the changing climate within a day.

This is not only applicable to Germany but also to the South African climate. The figure below (World Weather & Climate Information, 2015) shows an average difference of 20°C between the minimum and maximum daily temperatures, while the average difference between winter and summer is only 13°C. Although the figure only shows data for Bloemfontein, South Africa, the differences remain fairly the same throughout the South African climate regions.

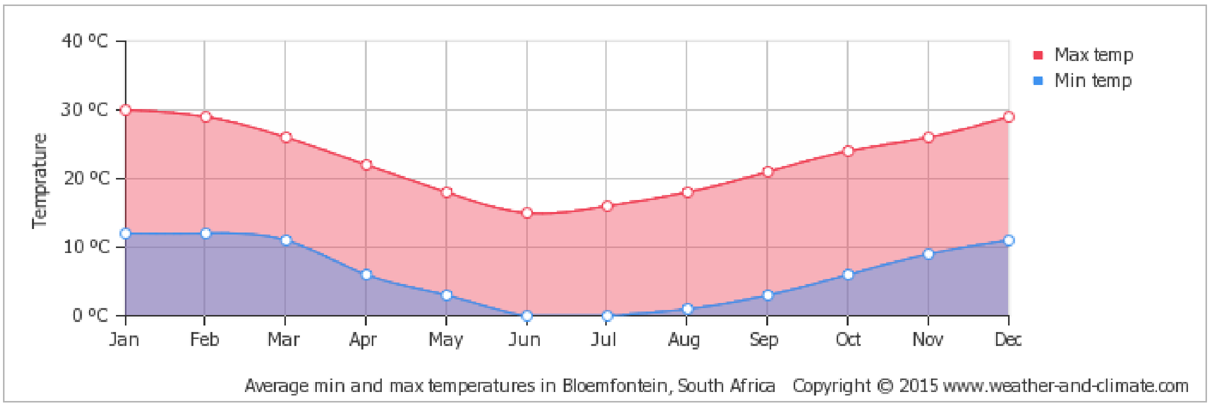

Figure 9: Bloemfontein average, minimum and maximum temperatures (World Weather & Climate Information, 2015).

The study by Huchtemann and Müller (2012) further suggests that the use of a buffer tank will reduce the number of operating intervals and increase performance. This is because short operational intervals at high storage tank temperatures are ineffective, and can be avoided by implementing the buffer tank. The practical study showed that the heating performance of an air source system is not only dependent on the source temperature but also on the heat sink or water temperature. A buffer tank is therefore used to ensure that a larger volume of water has to be drawn before both tanks are reheated again. This is accomplished by the use of two temperature probes to dictate when the heating cycle starts and stops. The two temperature sensors are installed at the inlet of each tank as can be seen in the figure below.

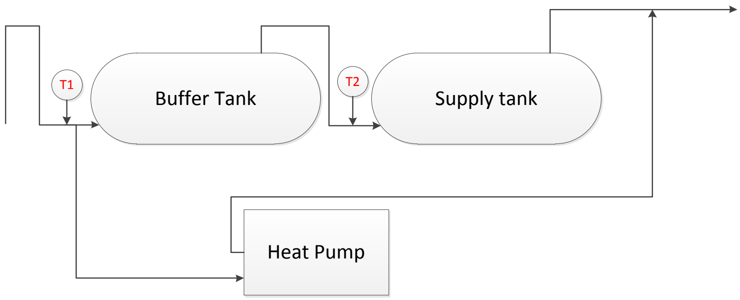

Figure 10: The use of a buffer tank.

Temperature sensor T2 is used to start the reheating cycle when cold water starts entering the supply tank, while T1 is used to stop the reheat cycle indicating that both the tanks have been heated to the desired storage temperature. This configuration extends the available hot water, and therefore the time before the heat pump needs to reheat the water during the coldest part of the day. This is favourable as the HPWH could be set to only reheat the water during the highest heat source and lowest heat sink temperatures.

2.10 Conclusion

From the literature study the following conclusions can be made which will be applied in the current study:

1. Two laboratory testing methodologies were considered within the literature study. It was found that the Eskom rebate test methodology focuses mainly on the performance of the heat pump by varying the ambient conditions as heat source. Within the British standard it was found that the focus is more on the heat sink, by varying the inlet water temperatures within the testing methodology. The literature study also revealed that the heat source and heat sink are equally important within a laboratory testing methodology. The current study will therefore use the benefits of both methodologies described above when developing the proposed testing methodology.

2. A simulation model based upon fundamental theory should be developed for the heat pump to validate the readings from the in-field measurements and laboratory tests. A simulation package equipped for this should preferably be used to simulate the heat pump for this purpose as it is very time consuming to set up a new numerical simulation model in a mathematical simulation environment.

3. The method described in Section 2.3 from the study conducted by Morrison *et al.* (2004) will be used to represent the data in the results chapter of the current study.

This method of data representation has the ability to summarise the results more clearly compared to other techniques.

4. The in-field data will be reduced similarly to the method proposed by Huchtemann and Müller (2012).

CHAPTER 3 : IN-FIELD MEASUREMENT METHODOLOGY

In-field data was primarily gathered to determine the actual in-field performance of a HPWH within the South African climate conditions when used at an occupied residential home. The in-field performance data will be used to determine if the laboratory tests could accurately predict the in-field performance. This chapter explains the methodology used to gather, reduce and present the data obtained from the in-field residential HPWHs. The results of the in-field measurements will be discussed in Chapter 6.

3.1 Data gathering

The infield data used within this study was measured by Eskom for five residential HPWHs installed in five different climate regions throughout South Africa. The climate regions selected were Bloemfontein in the Free State, Potchefstroom in the North West Province, Pretoria in Gauteng, Tzaneen in Limpopo and Durban in KwaZulu-Natal. Due to the price of a quality data acquisition system, only one heat pump was installed in each of the five climate regions. This of course increases the risk of losing an entire climate region's data, should there be a malfunction on the acquisition system or the heat pump. It was, however, deemed a calculated risk as the ambient conditions throughout these climate zones overlap, allowing verification of the data acquired.

The data acquisition system was set up to record nine performance parameters every minute for a period of one year. These parameters were then used to determine the performance changes with temperature within a 24 hour day, and also during the climate changes of the four seasons.

The nine measuring parameters included in the data acquisition systems can be divided into two groups. The first set of parameters shall be used to determine the system COP as was done by the study of Guo *et al.* (2011), described in Section 2.7. The second set of parameters will be used to determine the instantaneous COP as was done by Yokoyama *et al.* (2005) in Section 2.8. Installing equipment to measure both types of performance indicators reduces the risk of losing data due to instrumentation failure. Should both performance indicating values be available, it will be compared with one another for verification purposes. Although these values should correlate well, it is expected that the system COP will be as much as 10% lower than the instantaneous COP due to pipe and tank losses throughout the system.

The installation of the heat pump units for this study was done according to the schematic shown in Figure 11. Instrumentation was installed to enable the measurement of the following influencing factors:

- Ambient dry bulb temperature.
- Ambient RH.
- Municipal water temperature.
- Hot water consumption using a flow meter in the cold water supply.
- Geyser water outlet temperature.
- Heat pump water inlet temperature.
- Heat pump water outlet temperature.
- Water flow rate through the heat pump.
- Electrical energy consumption of the heat pump.

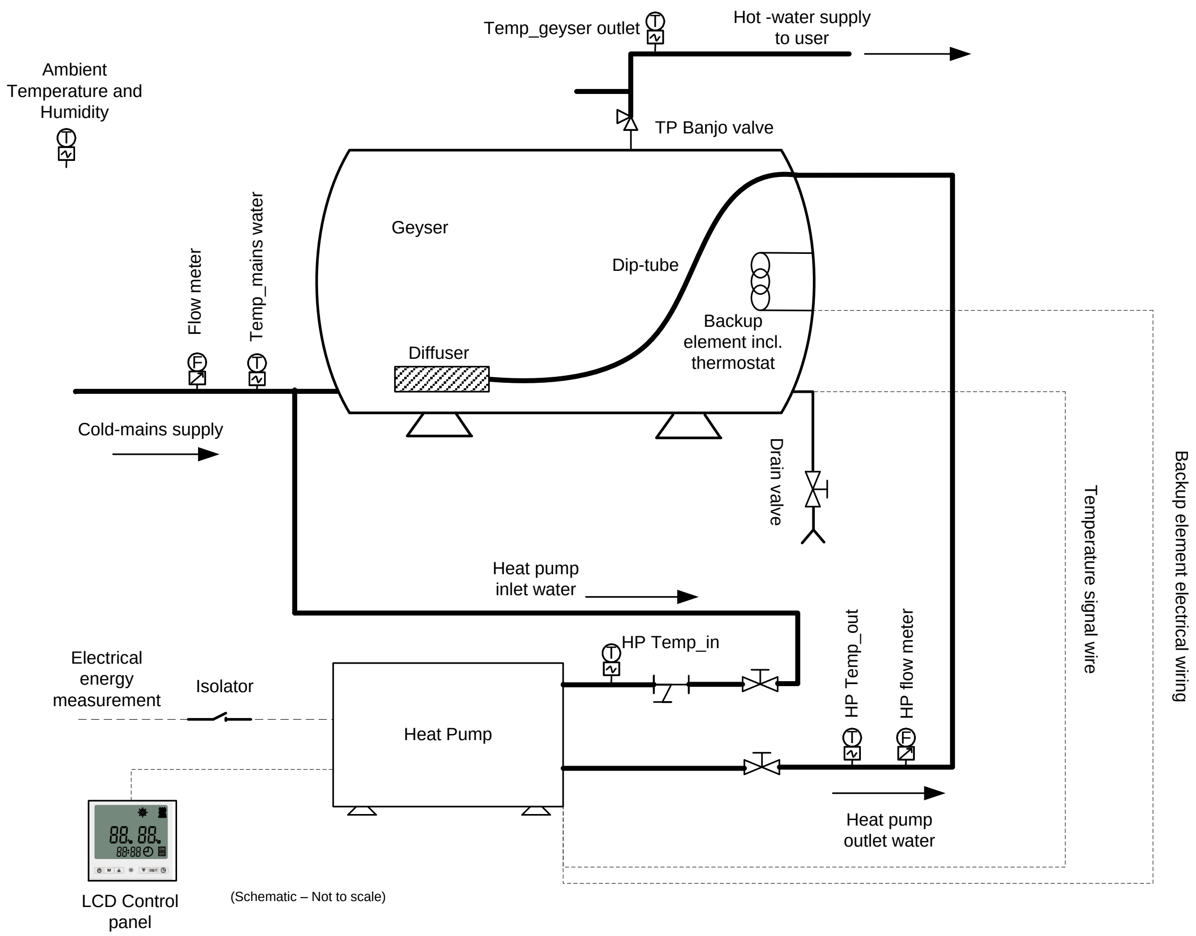

Figure 11: In-field heat pump installation schematic.

3.2 Data reduction

3.2.1 Initial data reduction

Data was recorded every minute for a period of one year. The data recorded was outlined in table format with the rows indicating the recorded data at different times. The columns of the table indicated the different parameters recorded, such as ambient temperature, relative humidity, power consumption, inlet and outlet water temperatures, etc. as can be seen in Annexure A, Section A.1.

Initially the data was reduced by software specifically designed for data processing. It was, however, found that the results obtained were far below the expected values. Further investigation into what caused the lower values proved to be extremely challenging, due to the excessive volume of data recorded for each site.

It was then decided to write a Microsoft Excel data reduction engine that makes use of programmed macros to perform investigative tasks. The tasks performed by these macros were the following:

- Segregate the datasets into daily 24 hour periods.
- Calculate the following for every 24 hour period:
    - Total hot water usage.
    - Average system COP.
    - Average instantaneous COP.
    - Average ambient dry bulb temperature during operations.
    - RH during operations.
    - The integrity of individual data points with respect to the rest of the data.
    - Total electrical energy used for the 24 hour period.
- Transfer the results from the calculations above to tables summarising the results.
- Populate a table that summarises the results in representable plots and charts.
- Colour code the table to indicate outlying data points.

These tables were then used to identify recording errors within the data as well as instrumentation failures that may have occurred. The tables indicated that the water flow meters used to determine the system COP failed on three of the five installations due to hard water and scaling. The flow meters showed a decrease in the flow recorded as time progressed, and the recorded values degraded until it showed only readings when water was drawn from the geyser at a high flow rate. The recorded data from the two remaining flow meters was deemed untrustworthy as there was no method of verifying the data recorded. The system COP was therefore excluded from this study. The tables further indicated that although the loggers did have recording errors in the form of major outliers, over 90% of the data could be used to accurately calculate the instantaneous COP. The instantaneous COP results could also be verified against results from other sites during similar climate conditions. A further investigation revealed that all the columns in a single recorded row of data will either be correct or outlying. Outliers could therefore be isolated and deleted by implementing rules to delete rows where the ambient temperature increased or decreased unnaturally for a row in the data. More details about the initial macro results for the in-field tests can be found in Annexure A, Section A.2.

3.2.2 Data reduction methodology

A second Microsoft Excel data reduction engine was then developed to calculate the instantaneous COP for every running cycle of a heat pump during the year. The tasks performed by the macros in this Excel data reduction engine were the following:

- Remove columns containing recoded values for system COP.
- Delete the outliers according to ambient temperature.
- Sort the data according to power consumption.
- Delete all recorded values where the heat pump was not operational.
- Sort the values according to ambient relative humidity and separate the recordings into 10% intervals. These intervals were defined as follows:
  - 5% ≤ $RH_{10\%}$ < 15% summarised under 10%.
  - 15% ≤ $RH_{20\%}$ < 25% summarised under 20%.
  - 25% ≤ $RH_{30\%}$ < 35% summarised under 30%.
  - 35% ≤ $RH_{40\%}$ < 45% summarised under 40%.
  - 45% ≤ $RH_{50\%}$ < 55% summarised under 50%.
  - 55% ≤ $RH_{60\%}$ < 65% summarised under 60%.
  - 65% ≤ $RH_{70\%}$ < 75% summarised under 70%.
  - 75% ≤ $RH_{80\%}$ < 85% summarised under 80%.
  - 85% ≤ $RH_{90\%}$ < 95% summarised under 90%.
  - The data below 5% and above 95% occurred so rarely that it was deemed inaccurate and therefore negligible.
- Arrange the summarised ambient RH values according to ambient dry bulb temperatures.
- Calculate for each of the summarised RH data sets the following:

  - Temperature difference between the water in the geyser and the ambient air dry bulb temperature.
  - Electricity consumption of the heat pump.
  - Energy added to the water by the heat pump.
  - Instantaneous COP.
- Plot the resulting COP values against the temperature difference as was suggested by Morrison *et al.* (2004) in Section 2.3. Draw linear regression lines through the resulting plots as suggested by Huchtemann and Müller (2012).
- Reduce the resulting linear regression lines to a single performance indication line for comparison with the other data sets.

More details about the macro programming code for the in-field tests and data reduction can be found in Annexure A, Section A.3.

3.2.3 Relevant equations

The primary objective of this study, as mentioned in Chapter 1, is to determine the efficiency of residential HPWHs in various climate zones in order to develop a laboratory testing methodology for South Africa. The efficiency of a residential HPWH is described by the coefficient of performance (COP) which is the quotient of the thermal energy added to the water and the electrical power consumed during this period.

The formula used for the calculation of the temperature difference between the water in the geyser and the ambient air is given by:

$$T_{\Delta} = T_T - T_A \qquad [3.1]$$

Where:

T_{Δ}: Temperature difference [°C].

T_T: Geyser tank temperature [°C].

T_A: Ambient dry bulb temperature [°C].

The measurement equipment used to track the electricity used by the heat pump gives data at one minute time intervals representing the performance of the heat pump for that minute. The electrical usage is therefore also recorded as a kilojoule per minute value and not a kilojoule per second value as is required for the calculation of the COP. The recorded value is therefore converted to a kilojoule per second value so that it can be used to calculate the COP. The formula for the calculation of the electricity used per second is as follows:

$$W_{elec} = \frac{P_{kJ/minute}}{60} \qquad [3.2]$$

W_{elec} Electricity used per second [kW].

$P_{kJ/minute}$ Electricity used per minute [kJ/ minute].

The formula for the calculation of the energy added by the heat pump as water is flowing through it and heated up is given by:

$$Q_{th} = \dot{m} \times Cp \times (T_{out} - T_{in}) \qquad [3.3]$$

Where:

Q_{th} Thermal energy added to the water [kW].

$\dot{m}$ Water mass flow rate through the heat pump [kg/s].

Cp Specific heat capacity of water [kJ/kgK].

T_{out} Temperature of the water at the outlet [K].

T_{in} Temperature of the water at the inlet [K].

The instantaneous COP of a HPWH is then given by the quotient of the energy added to the water and the electrical work required to do so:

$$COP = \frac{Q_{th}}{P_{elec}} \qquad [3.4]$$

Where:

COP Coefficient of performance [kW/kW]

Q_{th} Thermal energy added to the water [kW].

P_{elec} Electricity used per second [kW].

3.3 Conclusion

The in-field data was recorded, but errors were found within the data due to instrumentation failure and logging errors. As a direct effect of these errors the system COP could no longer be calculated. It was, however, found that the data could be filtered with a Microsoft Excel macro to the point where the instantaneous COP could be calculated.

The macro filtered the data allowing only a 1% variance or error. After the data was filtered it was found that more than 90% of the data in each of the data set was reliable, and could therefore be used to determine the actual in-field performance of the residential HPWH.

The results of the in-field measurements will be compared to the laboratory test results and the simulation model results in Chapter 6.

More details about the data processing of the in-field tests and results thereof can be found in Annexure A, Section A.4.

CHAPTER 4 : LABORATORY TEST METHODOLOGY

The primary goal of the laboratory tests was to determine if the in-field performance of a residential heat pump water heater can be predicted with a proper testing methodology. The methodology used for the laboratory tests within this study will be used as the baseline for determining which influencing factors are accurately captured and what needs to be added to the methodology before an accurate representation of the in-field performance is obtained. The laboratory test results used in this study were generated by a reputable testing facility within South Africa. As there is currently no methodology for performance testing of residential HPWHs in South Africa, the method used for the Eskom rebate test was also applied as a baseline here to test the residential heat pump unit in a controlled environment. This section explains the methodology used to gather, process and present the data obtained in the laboratory tests. The results of the laboratory tests will be discussed in Chapter 6.

4.1 Data gathering

The tests were conducted within a climate controlled environment on a residential heat pump installation that included the geyser and relevant pipe work. The test recorded the water inlet temperature, the water outlet temperature, the water flow rate, the electricity consumption, and the heat pump inlet air dry bulb temperature and relative humidity. These parameters are required to compare the performance of the heat pump against the available in-field data.

Five separate tests were conducted, each at a 50% relative humidity and with ambient dry bulb temperatures ranging from 5°C to 42°C. For each of the tests the water in the geyser is mixed to a homogeneous temperature and recorded as the starting temperature. Hereafter the heat pump is activated to heat the water until the heat pump's logic indicates that it has reached its set temperature (55°C). For the duration of the heating cycle the performance indicating factors and the performance influencing factors are measured. Finally the water in the geyser is again mixed to a homogeneous temperature before the final water temperature is recorded. The major performance indicating factors that were measured are as follows:

$\dot{m}$ Water mass flow rate through the heat pump [kg/s].

T_{out} Temperature of the water at the heat pump outlet [K].

T_{in} Temperature of the water at the heat pump inlet [K].

P_{elec} Electricity being used [kW].

More details about the recorded data of the laboratory tests and results thereof can be found in Annexure B, Section B.1.

These recorded values are used upon completion of the test to calculate the instantaneous COP.

The major performance influencing factors recorded were as follows:

T_T: Geyser tank temperature [°C].

T_A: Ambient dry bulb temperature [°C].

RH: Relative Humidity [%].

These values are recorded to measure the effect thereof on the recorded performance.

The results generated by the laboratory tests are expected to result in a performance indication line that correlates to within 10% of the in-field performance data. It was, however, noted in the literature that this test will result in a performance indication line similar to a pump curve. It is therefore important to note that even though it gives the performance within 10%, the values of tank and ambient temperature will determine where on this performance line the heat pump is performing. The average performance of the heat pump on this performance line can be obtained by theoretically adding tapping profiles after the initial heating cycle, as was found in the British standard (described in Section 2.6), but it will not be included for these laboratory tests.

The environmental test chamber could only reduce the air temperature to a minimum of 5°C, as the facility struggled to provide a stable test environment at lower temperatures. The relative humidity could only be kept at 50% due to the limitations of the environmental test chamber. Finally, the system COP could not be determined due to a limited number of calibrated temperature sensors available within the laboratory test facility.

The installation was set up similar to the in-field installation and therefore has the exact same schematic as Figure 11 in Section 3.1.

4.2 Data reduction

4.2.1 Data reduction methodology

The data reduction methodology used is as follows:

- Use the data gathered during the laboratory tests and determine an instantaneous COP for each second of the recorded data.
- Plot the instantaneous COP against the water and air temperature difference for each of the five tests conducted at the different ambient conditions.
- Generate a linear regression line at 50% relative humidity for each one of the five tests and plot them against one another.
- Reduce the resulting linear regression lines at 50% relative humidity to a single performance indication line for comparison with the other data sets.

4.2.2 Relevant equations

The relevant equations for the in-field data, laboratory test results and simulation model are the exact the same. The same data was recorded or generated within each data set. This reduced the complexity of data processing within the study before the data sets could be verified and validated against one another. The same equations as explained in Chapter 3 can therefore be used for the data processing of the laboratory tests.

4.3 Conclusion

The laboratory tests were conducted on certain sample conditions namely 5°C, 15°C, 25°C, 35°C and 42°C, all at 50% humidity, as it was not deemed necessary to test all temperatures and relative humilities found within the in-field data. The methodology for the laboratory tests closely resembled the test methodology used for the laboratory tests conducted by Eskom for the residential heat pump rebate programme. The results of the laboratory tests will be compared to the in-field data and the simulation model results in Chapter 6. More details about the data processing of the laboratory tests and results thereof can be found in Annexure B.

CHAPTER 5 : FLOWNEX SE SIMULATION MODEL METHODOLOGY

The primary goal of the simulation model is to verify and validate the in-field data and laboratory tests against a theoretical simulation model. From the literature study it was found that the best approach for a performance study is laboratory tests and/or in-field data compared to a fundamental numerical simulation model based on mass, energy and momentum conservation laws. These studies also pointed out that such models take a long time to develop in a mathematically based program. It was therefore decided to build the model in a simulation environment created for fast and effective construction. As this simulation model requires specific design specifications from the manufacturer to increase the accuracy thereof, it cannot be applied to heat pumps in general, but will serve as confirmation that the data gathered infield and within laboratory tests can be theoretically confirmed. This chapter will list the methodology used in developing and implementing the simulation model, but the results will only be shown and discussed in chapter 6.

5.1 Background

The simulation program selected for this study is Flownex® SE (Flownex® SE, 2014). Flownex was developed to combine a very extensive range of simulation capabilities within a built-in component library, enabling the user to create complete systems quickly and effectively. Although Flownex® offers a dynamic simulation option, it was decided to limit the study to only a steady state solution. The advantage of a steady state solution for this simulation model is that less components are required leading to a smaller error margin within the obtained results.

The Flownex® simulation environment also caters for refrigerant cycle design with pre-constructed components available to build into the simulation. These building blocks are pre-programmed with the fundamental equations leaving only the characteristics and boundary conditions of the component as variables to be supplied by the user before the component block is fully functional. The simulation model is developed to generate the same results as obtained in the in-field measurements and laboratory test results, allowing for quick and effective comparison between data sets.

Each component is sized and specified according to the design specifications and drawings obtained from the component manufacturers. After the model is developed, it will be improved with test results obtained from the heat pump manufacturer. These test results are generated by the heat pump manufacturer independent of this study. The simulation model results are then compared to the results obtained by the manufacturer at 25°C ambient dry bulb temperature. If it is found that the performance of a certain component is slightly over- or under predicted compared to the manufacturer's test results, the simulation model component can be improved by changing one of the independent variables. The independent variable to be changed would be any variable influencing only the component being improved, without affecting the rest of the components in the refrigeration cycle.

Once the performance results at 25°C ambient dry bulb temperature are deemed accurate in comparison with the manufacturer's data, the Flownex® model can be used to generate simulation results for all the ambient temperature conditions of the laboratory tests listed in Chapter 4 namely 5°C, 15°C, 25°C, 35°C and 45°C at 50% relative humidity. The simulation results of each temperature are compared to the laboratory tests, after which the results are reduced to a single performance line to be compared to both the in-field results and the laboratory test results.

5.2 Simulation model layout

The components that are simulated within this simulation model are the compressor, condenser, expansion valve and evaporator. These components can be found directly within the Flownex® SE simulation library. The expansion valve is simulated using a pipe reduction component with a variable cross sectional area. Figure 12 shows the graphical user interface for the Flownex® SE simulation model developed for this study. The outlet water temperature is the major output value of the simulation model and can be seen highlighted in yellow within Figure 12.

The simulation inputs required to solve the simulation model are initially as follows:

- $T_{a_{in}}$: Air dry bulb temperature before the heat pump [°C].
- $T_{w_{in}}$: Water temperature into the heat pump [°C].
- T_s: Compressor suction temperature [°C].
- P_s: Compressor suction pressure [Pa].
- T_d: Compressor discharge temperature [°C].
- P_d: Compressor discharge pressure [Pa].

Not all of these input values were recorded within the in-field measurements or the laboratory tests. It was therefore decided to investigate methods of reducing the input values to values that were recorded within the in-field measurements and laboratory tests, allowing for more accurate comparison between the simulation model and the other data sets. The investigation indicated that the values not recorded could be calculated by integrating an Engineering Equation Solver (EES) model into the Flownex® SE environment.

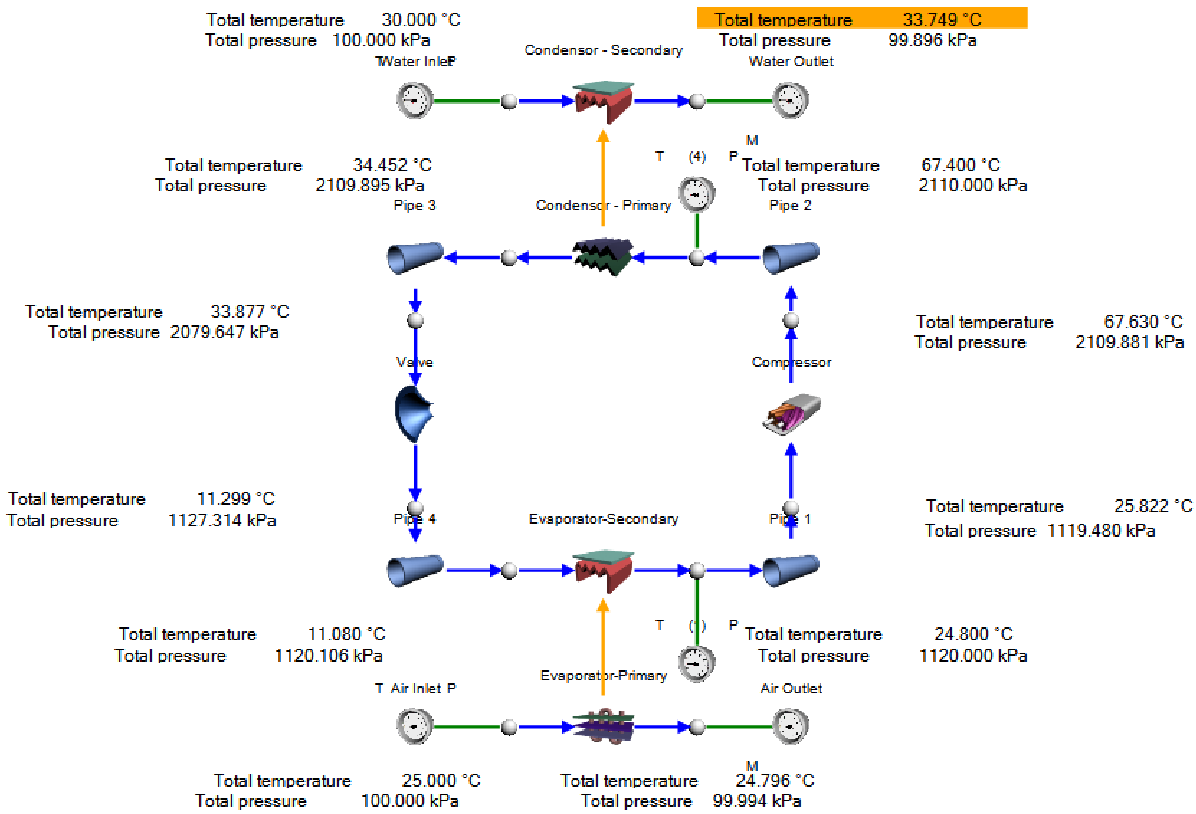

Figure 12: Flownex® SE simulation model.

5.3 Integrated EES functions

After the manufacturer's data was studied it was found that the condenser inlet refrigerant temperature and pressure have a strong correlation to the inlet water temperature. It was further found that the evaporator outlet refrigerant temperature and pressure have a strong correlation to the inlet air temperature. After these correlations were studied, functions were developed to determine the refrigeration temperatures and pressures required by the Flownex® SE simulation model as input values. The inlet water and air temperatures are therefore exported from Flownex® SE to the EES model to be used as input values for the developed functions. The EES model then calculates the refrigeration temperatures and pressures before exporting them back to Flownex® SE as input values. By integrating the EES model into the Flownex® SE model the inputs are reduced to:

- $T_{a_{in}}$: Air dry bulb temperature before the heat pump [°C].
- $T_{w_{in}}$: Water temperature into the heat pump [°C].

These values were recorded in the in-field measurements and laboratory tests, and could be used as input values to compare the results from the simulation model to the in-field results and laboratory results.

5.4 Component specifications

The following lists the input variables used within each of the component blocks of the Flownex® SE model.

Compressor

- η: Isentropic efficiency [%].
- N: Rotations per minute [rpm].
- x: Number of stages [-].
- V_s: Swept volume [m^3].
- V_d: Dead volume [m^3].
- $Fluid$: R410A.

Condenser refrigerant side

- A_Q: Heat transfer area [m^2].
- L_p: Fluid path length [m].
- N_i: Number of increments [-].
- D_w: Wall thickness [m].
- R: Surface roughness [µm].
- $Material$: Copper.

Condenser water side

- D_h: Hydraulic diameter [m].
- A_{Qs}: Secondary heat transfer area [m^2].
- R_s: Secondary surface roughness [µm].

Expansion valve

- A: Cross sectional area of the restrictor [m^2].
- β: Discharge coefficient [-].

Evaporator refrigerant side

- X: Number of parallel circuits per row [-].
- n: Number of tube passes [-].
- N_r: Total number of tubes[-].
- T_{ID}: Tube internal diameter [m].
- T_L: Tube pass length [m].
- R_I: Inside surface roughness [µm].

Evaporator air side

- A_{EQ}: Evaporator heat transfer area [m^2]
- T_{WT}: Tube wall thickness [m].
- Evaporator characteristics.

More details about the Flownex® input tab can be found in Annexure C, Section C.1.

5.5 Conclusion

A simulation model was developed to verify and validate the data generated within in-field measurements and laboratory test results. This simulation model was developed within the Flownex® SE software environment, allowing for fast and effective simulation of the entire gas cycle.

The disadvantage of using a simulation environment is that it requires manufacturer details specific to the heat pump. This means that the simulation model cannot be applied to heat pumps in general within the South African market.

The simulation model was developed using performance values and specifications from the manufacturers of the heat pump's sub-components. The simulation model was developed to generate the same results as obtained in the in-field measurements and laboratory test results, allowing for quick and effective comparison between data sets.

The results of the simulation model will be compared to the laboratory test results as well as the in-field measurements in Chapter 6.

CHAPTER 6 : RESULTS

The results to be discussed in this chapter were generated by following the methodologies listed in Chapters 3, 4 and 5. The results of the in-field data, laboratory tests and the simulation model will be discussed separately before being compared to one another.

6.1 In-field measurements results

The data from the different sites were first separated in terms of the relative humidity as mentioned in Chapter 3. The COP was then calculated per row of data for each one of the data sheets generated in-field (see Annexure A for an example of a data sheet). Hereafter the data sheets were sorted according to a temperature difference $[T_t - T_a]$ between the tank temperature (T_t) and the ambient temperature (T_a).

6.1.1 Recorded data for each relative humidity

The table below indicates how many recorded relative humidity values are present within each predefined range. One recorded value represents one minute of running data.

Table 4: Number of in-field recorded values for each defined RH

RH data range	Number of data rows
5% ≤ $RH_{10\%}$ < 15%	4969
15% ≤ $RH_{20\%}$ < 25%	16958
25% ≤ $RH_{30\%}$ < 35%	30264
35% ≤ $RH_{40\%}$ < 45%	37601
45% ≤ $RH_{50\%}$ < 55%	37406
55% ≤ $RH_{60\%}$ < 65%	38215
65% ≤ $RH_{70\%}$ < 75%	32154
75% ≤ $RH_{80\%}$ < 85%	50385
85% ≤ $RH_{90\%}$ < 95%	6783

From Table 4 it can be seen that there are far less data for the two extreme predefined ranges, namely 10% and 90%. The data in these two ranges mainly comes from a single in-field test site eliminating the ability of the data to be verified against other sites. As these data sets cannot be sufficiently verified, it is deemed unreliable. The low number of recordings within these conditions does, however, indicate that they are not frequently found within South Africa, thus lowering the need for testing within these conditions. The

10% and 90% lines are therefore removed within the results comparison for the in-field data as well as for the remainder of the study.

6.1.2 Results summary for in-field measurements

In Figure 13 the in-field data is shown for the different predefined relative humidity ranges. The figure shows the coefficient of performance (COP) for the difference between the current tank temperature (T_t) and the ambient dry bulb temperature (T_a). The temperature difference decreases to a negative value, when the water in the geyser drops below the ambient dry bulb temperature. The temperature difference increases to above zero as the water temperature in the geyser rises above the ambient conditions. The conditions for operation are therefore favourable at a low temperature difference, with the conditions becoming less favourable as the temperature difference increases.

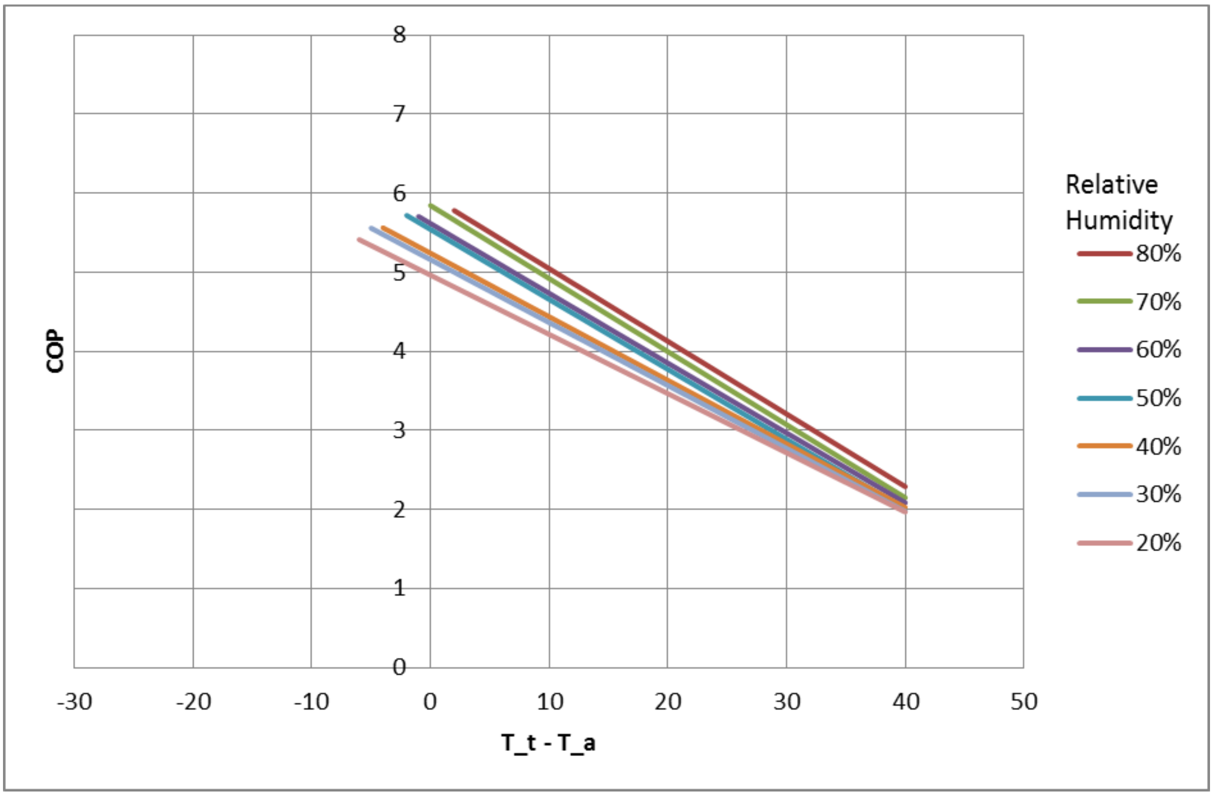

Figure 13: In-field measurement results.

Further observations that can be made from the in-field results:

- The 10% and the 90% relative humidity performance lines will not be taken into account during the data reduction process to follow, due to the reasons mentioned before.

- The remaining performance lines show an increase in performance as the relative humidity increases. The influence of relative humidity is not as significant as the influence of the water tank temperature and the ambient dry bulb temperature, but if it is not considered, the standard deviation of the performance line is increased to the point where detection of outliers and errors in the data becomes impossible.
- The 50% relative humidity line is close to an average performance line, with a 0.4 COP deviation to each side. It is therefore suggested that it is not required to do multiple relative humidity tests within a laboratory test environment. However, it is required to specify the humidity for the test at the average value of 50% knowing the performance can increase or decrease slightly from this point with a change in relative humidity.
- The maximum COP difference recorded between the 80% and 20% relative humidity performance lines is 0.9. The minimum COP difference recorded between the 80% and 20% relative humidity performance lines was 0.36.

In Figure 14, a linear regression line is fitted through the data. As mentioned in Chapter 3 the performance lines need to be reduced to a single performance line summarising the in-field performance of the HPWH that was tested, for comparison with the laboratory and simulation results. This line can now be used to quickly and effectively compare this in-field data set to the linear regression line drawn up for the laboratory tests as well as the simulation model results, to be shown later in this chapter.

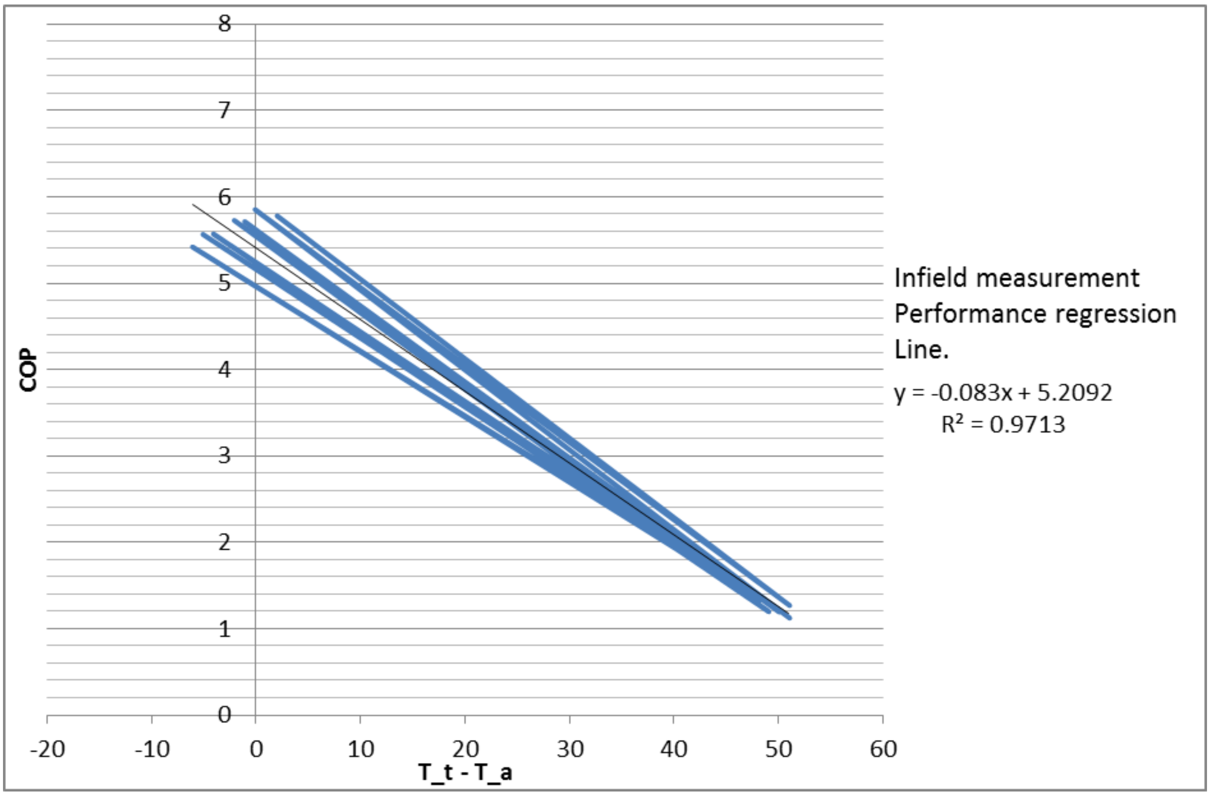

Figure 14: In-field measurements performance regression line.

6.2 Laboratory test results

As mentioned in Chapter 4, the laboratory test results were generated with a methodology closely resembling the Eskom rebate methodology as given in Section 2.5.

The laboratory test recorded data for every second for the duration of the heating cycle. This data was then summarised in a table with the time steps represented as rows and the different recorded values as columns. The instantaneous COP was calculated per row of data for each one of the five test conditions. The resulting tables were then sorted according to the temperature difference $[T_t - T_a]$ between the tank temperature (T_t) and the ambient dry bulb temperature (T_a).

6.2.1 Results summary for laboratory measurements

In Figure 15 below the laboratory results obtained can be seen before it was reduced to a single performance line. The figure indicates the results obtained for the tests conducted at 5°C, 15°C, 25°C, 35°C and 42°C, all at 50% relative humidity. The figure shows remarkable similarities for all test conditions when compared to Figure 17 generated by the Flownex® SE simulation model, to be discussed in Section 6.3. From the in-field data it was observed that the highest dry bulb air temperature recorded was 36°C. This means that the 42°C performance line in the laboratory tests and simulation model will skew the final linear regression performance line in each of the data sets respectively. For both the laboratory tests and simulation model, the 42°C performance lines will therefore be removed before the final performance linear regression lines are drawn.

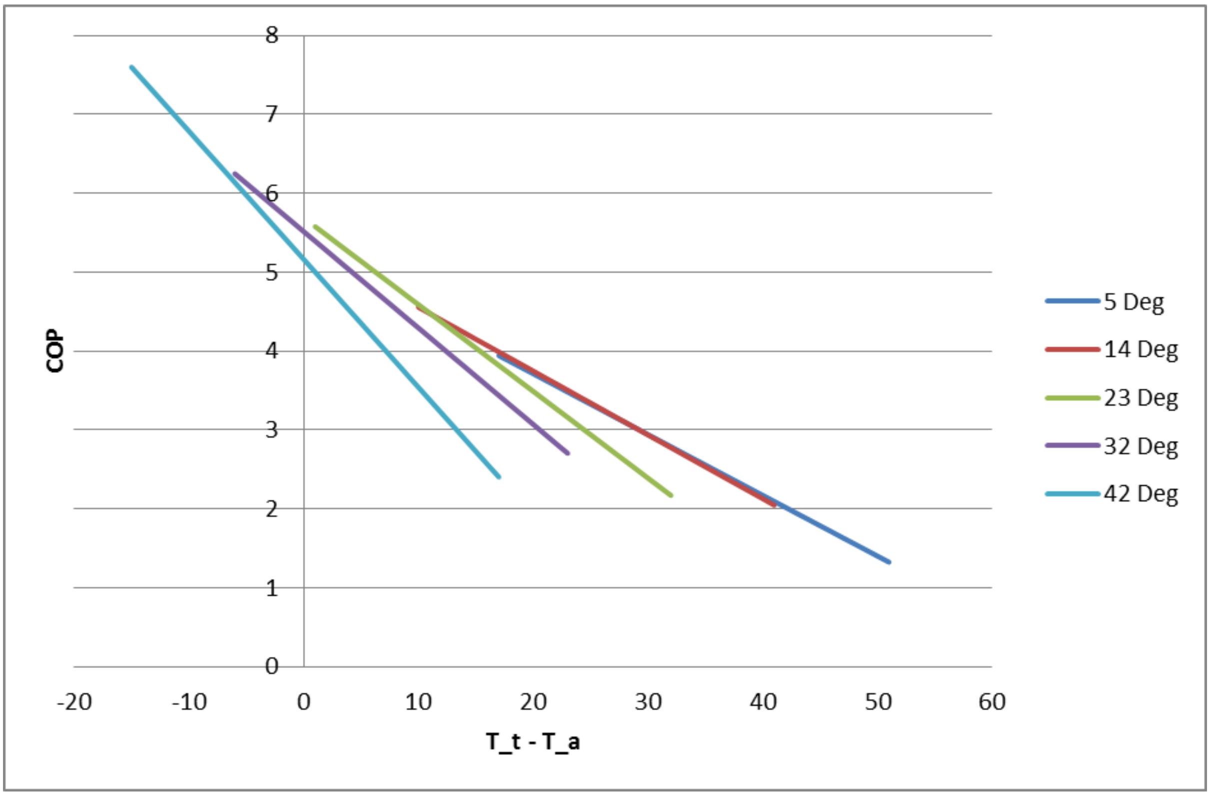

Figure 15: Laboratory test results.

In Figure 16 below, a linear regression line is fitted through the laboratory test results. As mentioned in Chapter 4, the resulting performance lines shall be used for summarizing the performance of the HPWH tested in the laboratory. This line can now be used to quickly and effectively compare this data set to the linear regression line drawn up for the in-field measurements as well as the simulation model results, to be discussed later in this chapter.

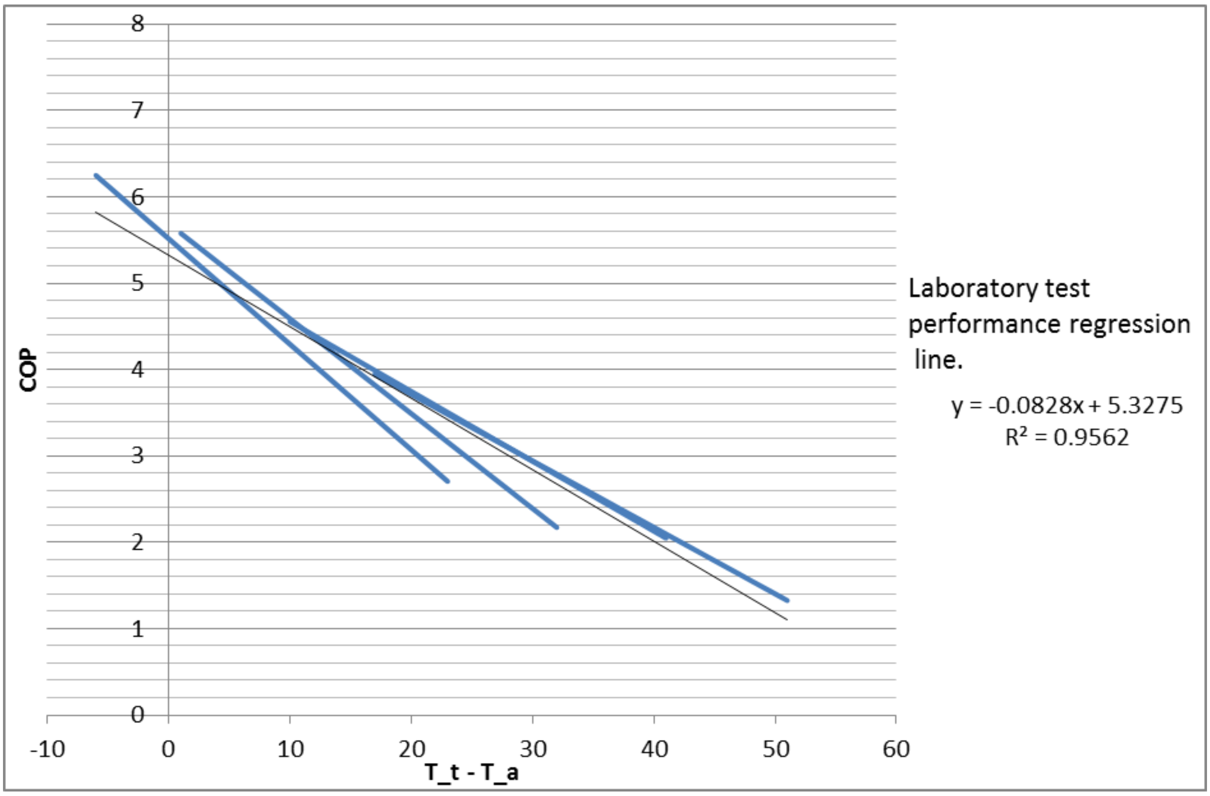

Figure 16: Laboratory test data regression line.

6.3 Simulation model results

The results discussed here were generated with the simulation test methodology as described in Chapter 5. The simulation model was used to generate results for each of the laboratory test conditions, namely 5°C, 15°C, 25°C, 35°C and 42°C, all at 50% relative humidity. The simulation results were then tabulated with the rows progressing as the water temperature is increased. The COP was then calculated per row of generated data before being sorted according to the temperature difference [T_t – T_a] between the tank temperature (T_t) and the ambient dry bulb temperature (T_a).

6.3.1 Summary of simulation results

The results shown in Figure 17 closely resembles that obtained within the laboratory tests shown in Figure 15. The 5°C performance line within the Flownex® SE results is, however, much higher than for the laboratory test results. This can be contributed to the increase in inefficiencies found within some of the sub-components at low temperatures that was not incorporated into the Flownex® SE model. As mentioned in Section 6.2, the 42°C performance line will be removed before the final performance linear regression line is drawn, as it was found that the highest ambient conditions recorded in-field was 36°C dry bulb. The 42°C performance line will therefore skew the final resulting performance line being used for comparison to the in-field data and laboratory test results.

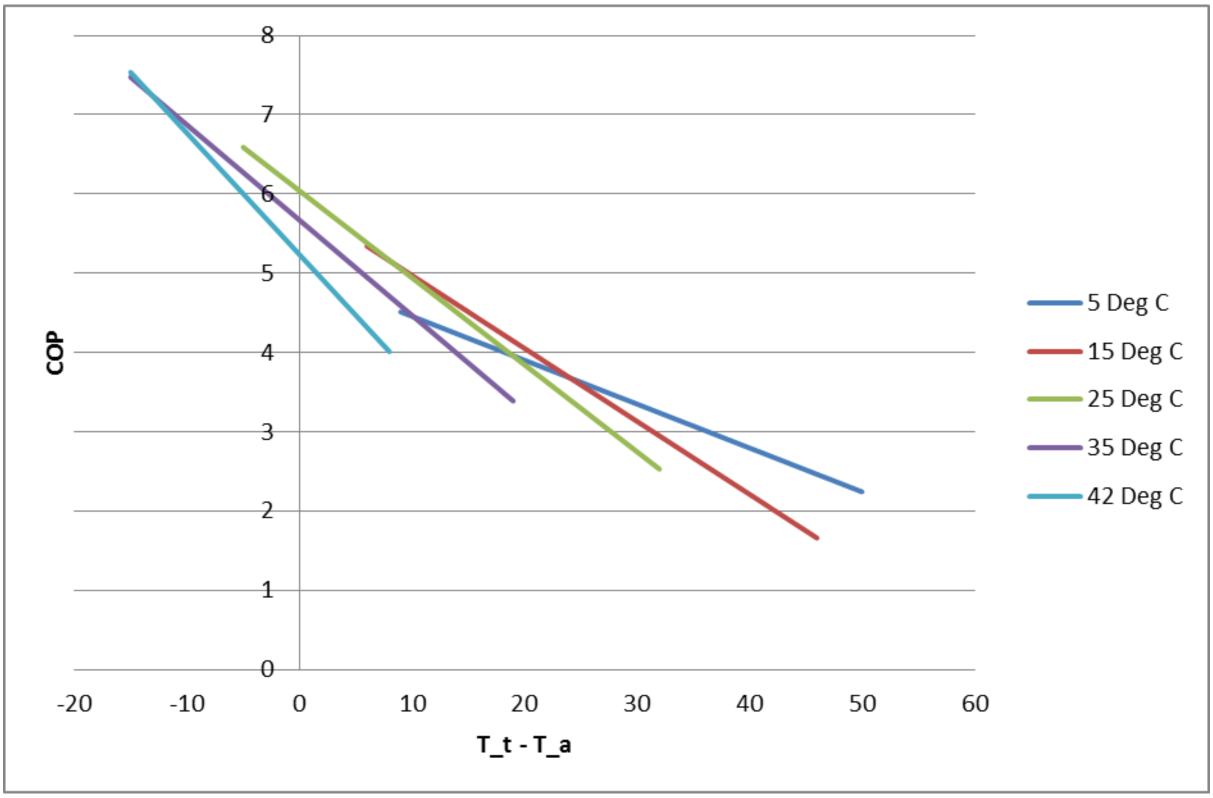

Figure 17: Flownex® simulation model results.

In Figure 18 below, a linear regression line is fitted through the generated simulation model results. As mentioned in Chapter 5, the resulting performance lines shall be used for summarizing the performance of the HPWH simulated in Flownex® SE. As mentioned before, this line can now be used to quickly and effectively compare this data set to the linear regression line drawn up for the in-field measurements as well as the laboratory test results.

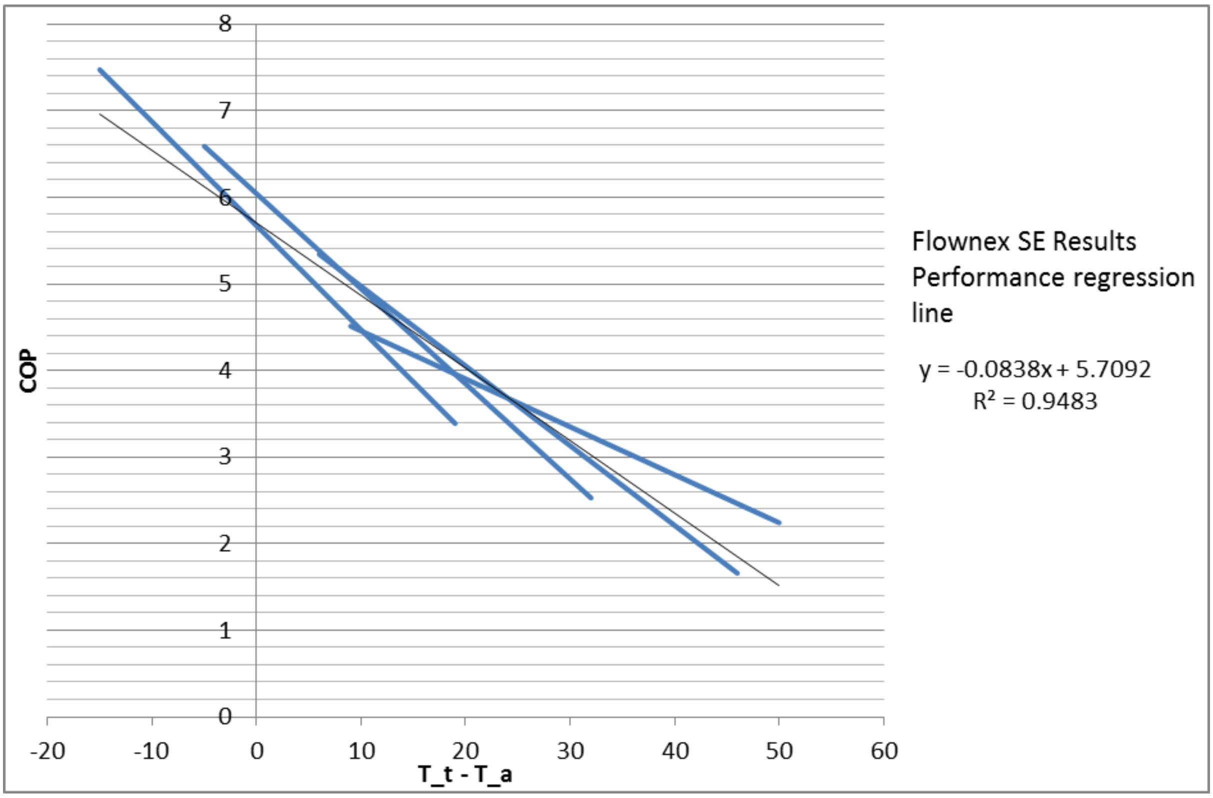

Figure 18: Simulation model data regression line.

6.4 Results comparison

Figure 19 below compares the final performance linear regression lines for the in-field measurements, laboratory tests and generated simulation results. This figure also includes a line indicating the performance specifications as supplied by the manufacturer of the HPWH. These results supplied by the manufacturer were reduced to this performance line based on the methodology listed in Chapter 4. The manufacturer's results are listed at different ambient conditions similar to this study, however the performance line was generated at a constant inlet water temperature of 30°C. This performance line based on the manufacturer's tests indicates a higher efficiency compared to the results generated within this study. The higher efficiency indicated by the manufacturer can be contributed mostly to the constant water temperature used within the manufacturer's specification. It is further suspected that the manufacturer's specifications were generated at a relative humidity higher than 70% due to the location of the factory being in a very humid area.

The constant inlet water temperature as currently specified is unrealistically low compared to inlet water temperatures found on average in-field. When other manufacturer specifications were studied in more detail, it was found that the test conditions were either not given, or that the specification indicated conditions far more favourable to the heat pump performance than the conditions found within a South African residential environment. The difference between this line and the performance lines generated within the study highlights the necessity for a performance standard in South Africa.

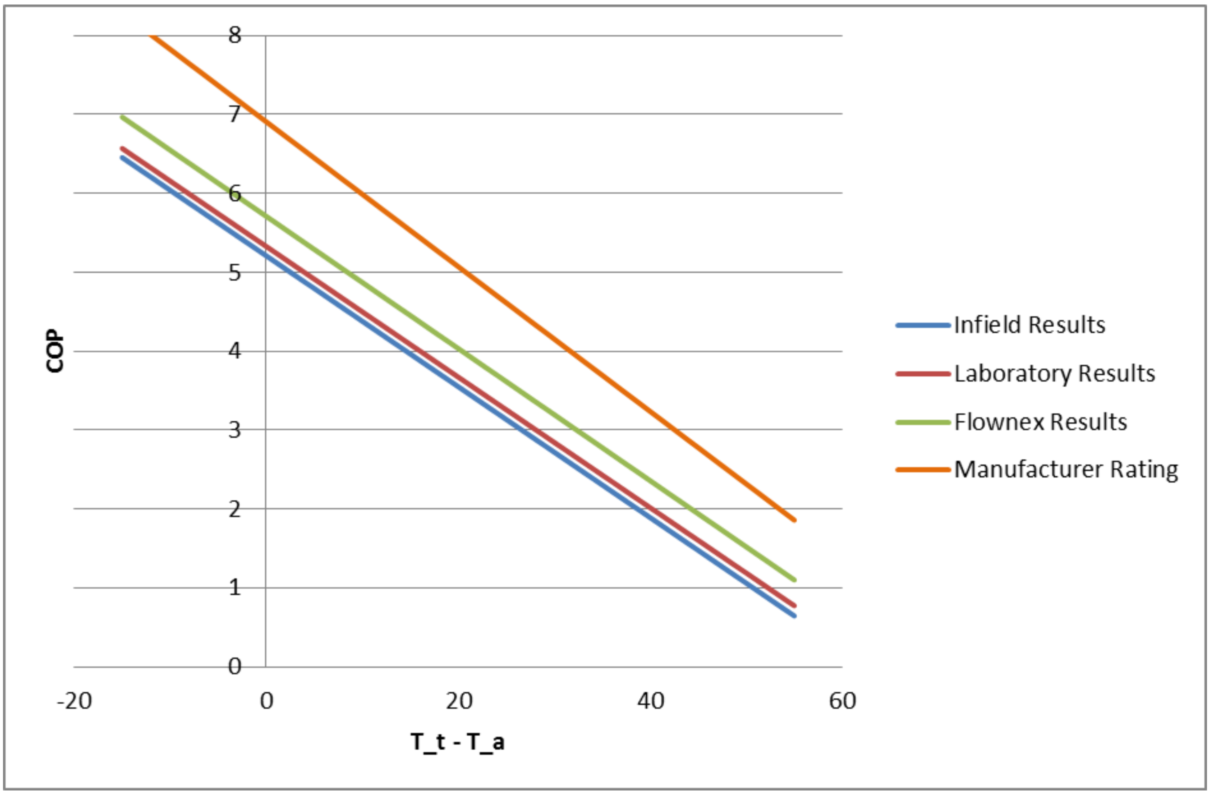

Figure 19: Performance line comparison.

Figure 19 above gives the performance lines for the tested heat pump considering the three major performance influencing factors, namely water temperature, dry bulb air temperature and relative humidity. For the three performance lines generated in this study the coefficient of variance between the lines is 8.4% on average while the average standard deviation for the data is 0.6. The performance line generated for the laboratory results has a coefficient of variance of 7.8% on average when compared to the Flownex® simulation model results and an average standard deviation of 0.5. The laboratory test results generated to represent the in-field data have a 3% average coefficient of variance when compared to the in-field data, with an average standard deviation of 0.2. This variance is very small and can be contributed to measurement accuracy and potential fouling on the heat pumps installed in-field. The heat pumps in-field were running for more than 1 year before data was recorded while the heat pump used in the laboratory tests was still brand new and only functionally tested before being used. It was further found that the lagging used on the heat pumps installed in-field had a lower R rating and thickness compared to the laboratory test installation. With all factors taken into account a

3% difference is considered to be small enough therefore proving that the laboratory test can give an accurate representation of the in-field performance of a heat pump water heater.

It is, however, important to note that even though this performance line gives an accurate representation of the performance at a certain condition, the British standard indicates that it is not only important to consider the performance line but also the position on this performance line that is most frequently found in-field. The next section will expand on this requirement.

6.5 The influence on performance considering the climate conditions

The effects included in the British standard can be theoretically applied to this study by studying the water temperatures and climate conditions during the in-field operations.

The charts to be discussed below indicate firstly the dry bulb air temperatures and secondly the relative humidity during the heat pumps' operation in-field. The data used to formulate the charts excludes all the data when the heat pump was not operational. These charts were formulated to verify that all relative humidity and temperature ranges are in fact represented within the in-field data. The charts will also be used as basis to formulate the final test methodology, by indicating the relative humidity and dry bulb temperatures most commonly found within the different regions of South Africa.

6.5.1 Climate in Bloemfontein during heat pump operation

From Figure 20 it can be seen that the relative humidity levels for the Bloemfontein site in the Free State falls in the average to low category with the largest concentration of data found around 30% relative humidity. The average relative humidity recorded for Bloemfontein was 46.4% and it can be seen that the relative humidity is rarely within the two extremes namely 10% and 90%. The ambient dry bulb temperatures recorded for Bloemfontein are lower than the national average dry bulb temperature of 16°C even though the minimum recorded dry bulb temperature was only 0.3°C. The temperatures in Bloemfontein are known to drop below 0°C but the heat pump did not run during these conditions. The maximum recorded dry bulb temperature for Bloemfontein was 36.2°C with the average recorded dry bulb temperature for Bloemfontein at 13.4°C. The figure below summarises the recorded climate in Bloemfontein.

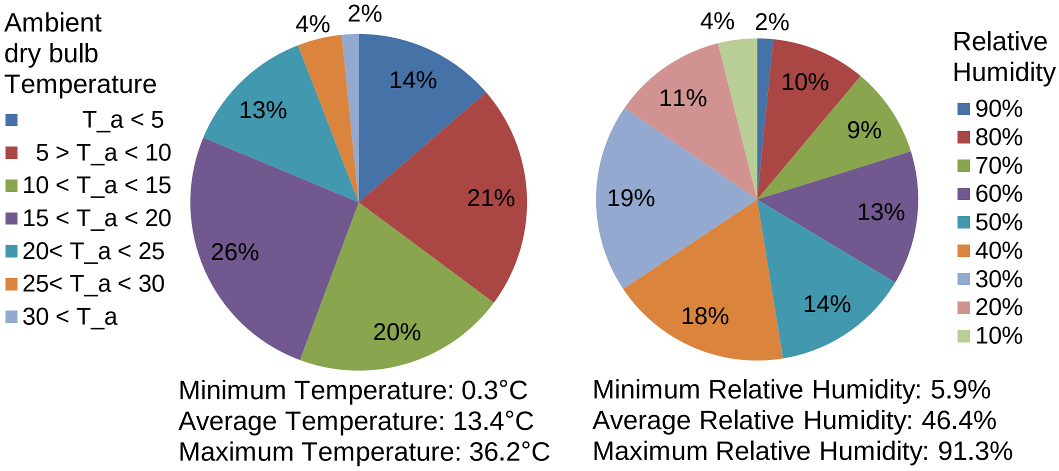

Figure 20: Climate during heat pump operation in Bloemfontein.

6.5.2 Climate in Potchefstroom during heat pump operation

The relative humidity levels at the Potchefstroom test site in the North-West province are very similar to the conditions found in Bloemfontein. The relative humidity is, however, consistently slightly higher than the relative humidity in Bloemfontein. The largest concentration of data was recorded around 40% relative humidity while the average relative humidity recorded for Potchefstroom was 47.2%. The Potchefstroom site also recorded the highest and lowest relative humidity, and dry bulb temperatures in the study. The minimum recorded dry bulb temperature for Potchefstroom was -2.1°C, while the maximum recorded dry bulb temperature for Potchefstroom was 36.7°C. The average dry bulb temperature recorded for Potchefstroom during operation was 15.6°C. The figure below summarises the recorded climate in Potchefstroom.

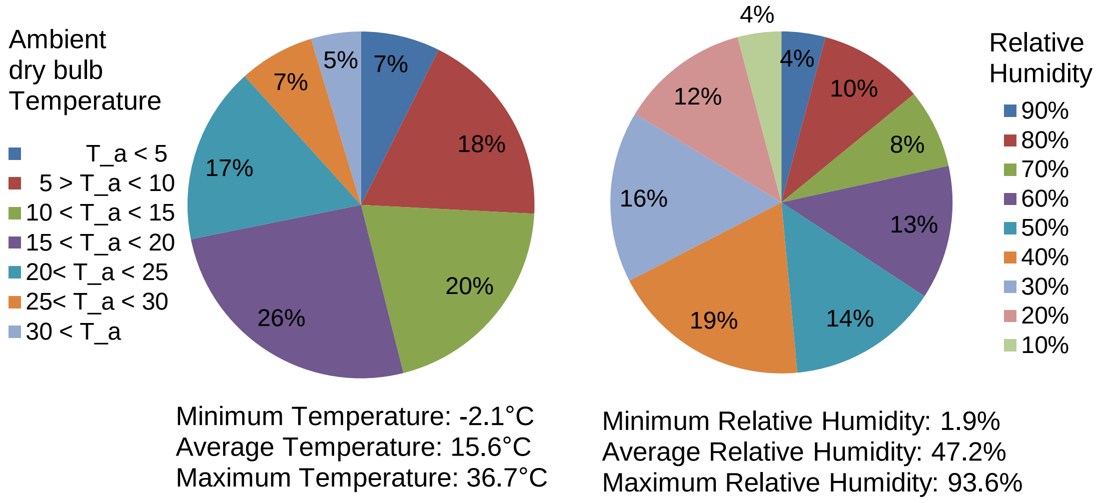

Figure 21: Climate during heat pump operation in Potchefstroom.

6.5.3 Climate in Centurion during heat pump operation

The relative humidity levels at the Centurion site in Pretoria, Gauteng, showed a large concentration of data around the 50% relative humidity average, with the number of recorded values decreasing towards the extremes of 10% and 90%. The calculated relative humidity is slightly higher than 50% at 52.7%. The recorded dry bulb temperature for Centurion showed very little data below 5°C during heat pump operation. This can be due to the test site’s water consumption patterns, reducing the running time in the colder early morning hours. The minimum recorded dry bulb temperature for Centurion was 3.6°C, while the maximum recorded dry bulb temperature was 35.9°C. The average dry bulb temperature recorded during operation was 18.2°C. Figure 22 summarises the recorded climate in Centurion.

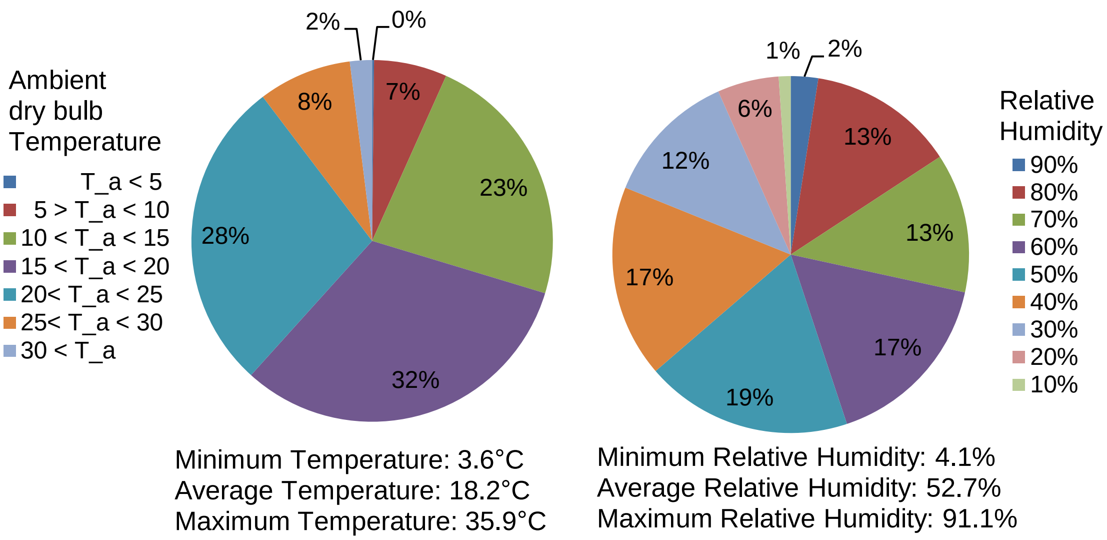

Figure 22: Climate during heat pump operation in Centurion.

6.5.4 Climate in Tzaneen during heat pump operation

Tzaneen in the Limpopo province was selected to represent a slightly higher relative humidity than Centurion, but it was found that the relative humidity in the specific area where the heat pump was installed was much higher than expected. 41% of all the data recorded measures within the 75% relative humidity to 85% relative humidity range. The average relative humidity for Tzaneen was 68.3%. The dry bulb temperatures for Tzaneen at these high relative humidity values are average to low. The minimum recorded dry bulb temperature for Tzaneen was 2.3°C, while the maximum recorded dry bulb for Tzaneen was 33.3°C. The average dry bulb temperature during operation was 13.0°C. The figure below summarises the recorded climate for the Tzaneen site.

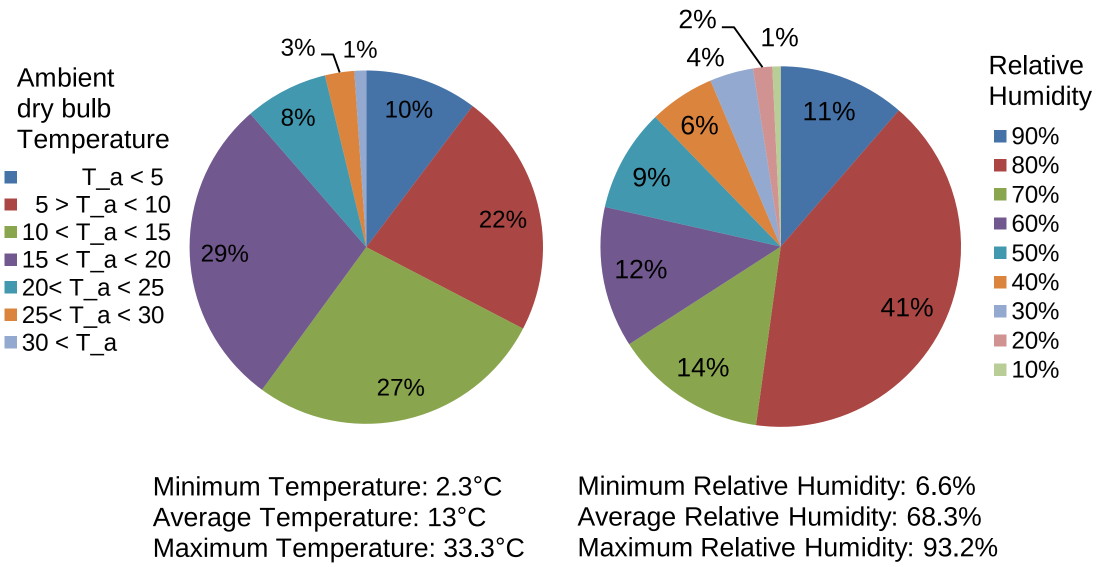

Figure 23: Climate during heat pump operation in Tzaneen.

6.5.5 Climate in Durban during heat pump operation

Durban in KwaZulu-Natal was included in the study due to its average to high temperatures and relative humidity. No relative humidity values were recorded below 10%, with very little recordings lower than 30%. 57% of all the data was recorded between 75% and 85% relative humidity, while the average relative humidity of the site is 70.5%. The dry bulb temperatures for Durban are also very high with no recorded values below 7°C. The dry bulb temperatures for the Durban site are mostly around the high average of 19.3°C, with the minimum recorded dry bulb temperature at 7.9°C. The maximum recorded dry bulb temperature for Durban was 36.2°C, but these high temperatures were not frequently found in the recorded data. The figure below summarises the recorded climate for Durban.

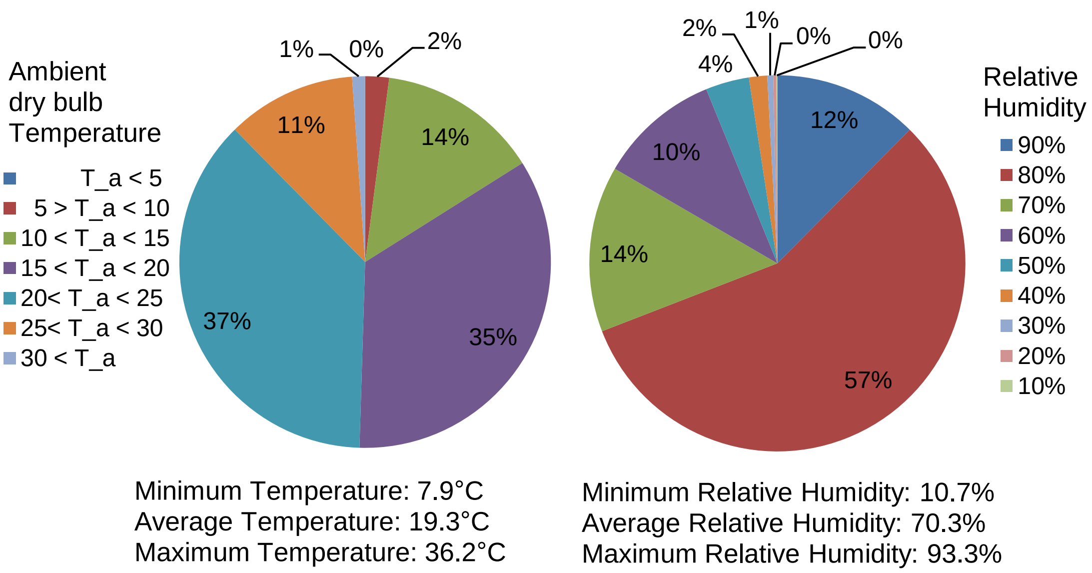

Figure 24: Climate during heat pump operation in Durban.

6.6 Results comparison for different regions considering climate data

The data from all the sites were combined to determine if the climate conditions typically found in South Africa were adequately covered in the study. The relative humidity data in Figure 25 shows that the data from Tzaneen and Durban had a substantial effect on the average relative humidity. The higher relative humidity on the Tzaneen site increased the average relative humidity from 50% to 55.8%. With this increase considered it can still be seen that adequate data was captured for the relative humidity ranges frequently found within the South African climate.

The summarised dry bulb temperatures in Figure 25 show that adequate data was captured to represent the climate in South Africa. From the data it became evident that a 42°C test condition within the simulation model and laboratory tests are not required as this will be rarely found in South Africa.

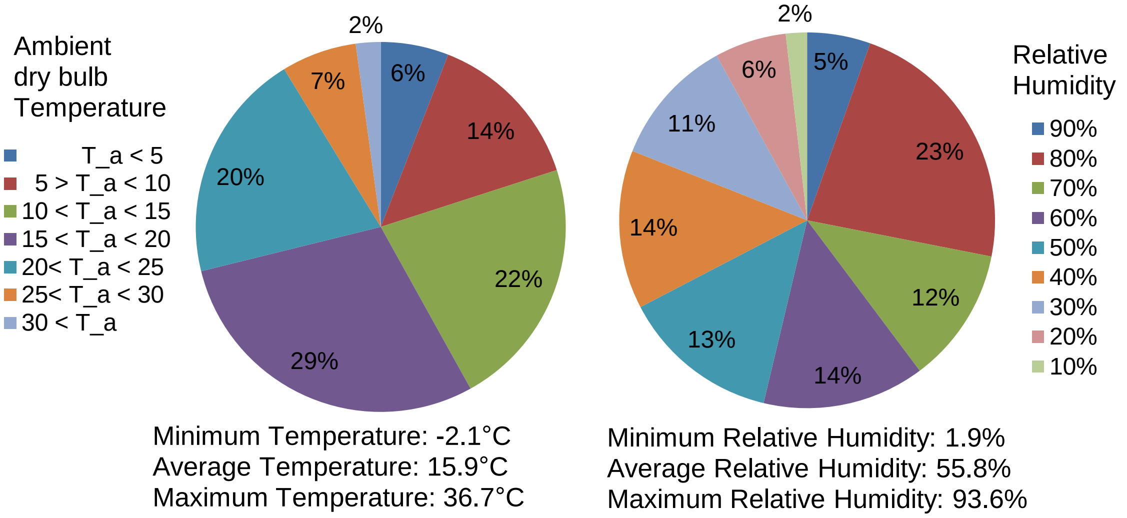

Figure 25: Average climate during heat pump operation on all sites.

If the climate data summarised above is theoretically applied to the performance lines according to the British standard, we find that the middle part of the performance line is not a good indication of the average performance of the HPWH. Within the in-field measurements it was also found that the water temperatures in the tank is rarely drawn to a temperature lower than 45°C, before the water is reheated back to 55°C. This is because water drawn from a tap within a residential home is on average relatively small compared to the total water volume stored within the geyser at 55°C. The water temperature (T_t) used within the x-axis of Figure 19 therefore rarely drops below 45°C; limiting the maximum COP realistically obtainable in-field. Furthermore, if an average dry bulb temperature is considered as portrayed in Figure 25, we find that the average performance of the heat pump installed in-field is limited by the temperature difference ($T_t - T_a$) to a very specific part of the heat pump performance line.

For an average in-field heating cycle the heat pump would therefore be operating between 29°C and 39° temperature difference (($T_t - T_a$) as indicated in Figure 26 below. The average in-field COP for the heat pump used in this study is therefore 2.4 if the British standard is considered and 3.1 if only the Eskom rebate methodology is followed. The importance of incorporating the British standard's test methodology can therefore be seen as the difference between a 3.1 COP value and a 2.4 COP value is far from negligible. The heat pump used within this study passed the Eskom rebate test specifications stating that the heat pump must have a COP of at least 2.8. If the British standard test methodology was incorporated with the Eskom rebate test methodology, the heat pump would have given a COP value lower than 2.8. However, if the British standard was considered, the required COP value would have most likely been lower. The required COP value will therefore be recalculated for the recommended testing methodology described in Chapter 7.

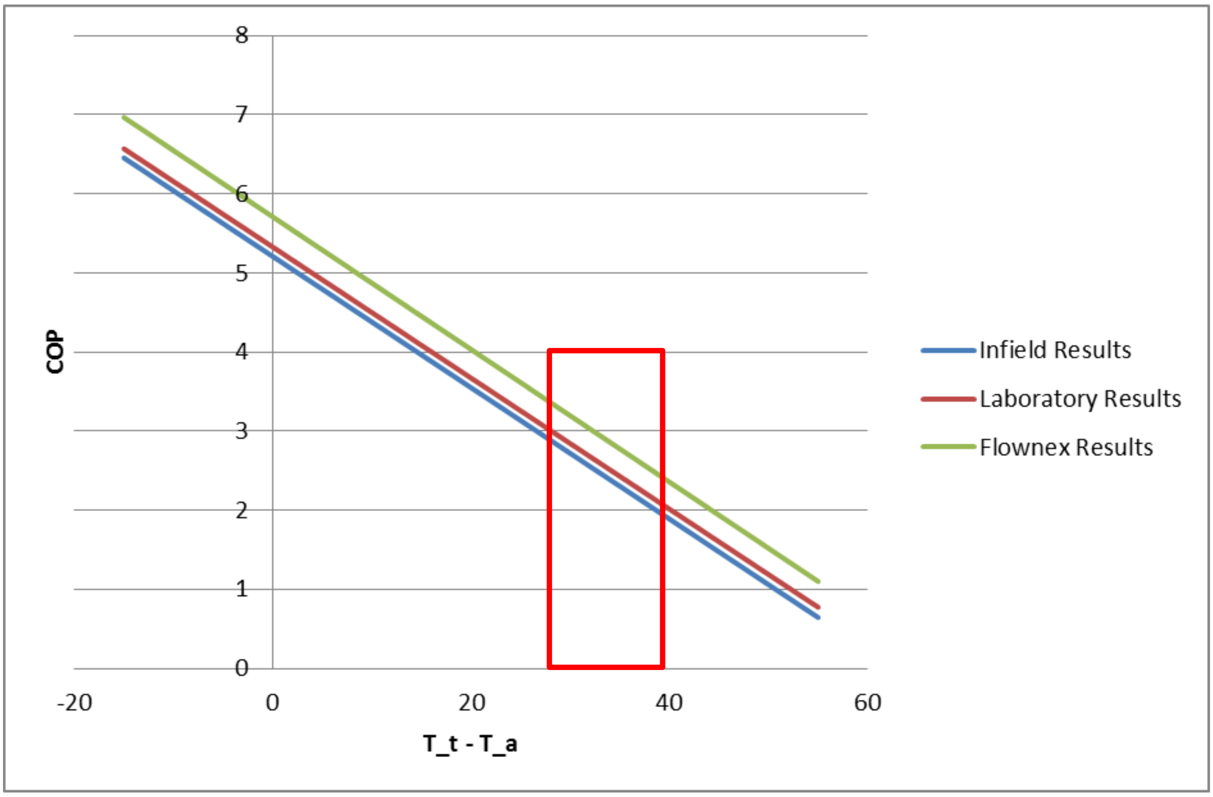

Figure 26: Results comparison with climate consideration.

It is therefore critical that the methodology used within the British standard must be incorporated into the laboratory test methodology to ensure the average performance of a heat pump is accurately determined in-field.

CHAPTER 7 : PROPOSED TEST METHODOLOGY

This test methodology will apply to residential heat pump water heaters, used with storage vessels that falls within the scope of the local standard SANS 151: *Fixed electric storage water heaters*. As mentioned in Section 1.5, this chapter will be limited to the thermal performance test of the heat pump water heater, therefore excluding general requirements, marking requirements and safety requirements on HPWHs.

With all results of this study considered, the proposed test methodology for a thermal performance test on a residential HPWH is a combination of the Eskom rebate test methodology and the British standard. It was further established that the test facility must be able to maintain environmental conditions within a lower variance than the variances suggested in the Eskom rebate test methodology. As mentioned before, the Eskom rebate methodology could result in a COP difference of 0.5 or 33% at low ambient conditions, without taking into account the accuracy of the testing equipment.

7.1 Test procedure and requirements

The proposed methodology consists of three tests. All three tests will be completed consecutively, without the installation being altered in any way between tests. The test process as listed in Phase 1, Phase 2 and Phase 3 below will be completed at three specified ambient test conditions. The test room temperature may not vary with more than ± 1°C for the duration of the test. The actual ambient conditions within the test room must be recorded for the duration of the tests. The specified ambient conditions during the three tests are as follows:

- Low temperature test at 3°C dry bulb temperature and 50% relative humidity.
- Average temperature test at 18°C dry bulb temperature and 50% relative humidity.
- High temperature test at 37°C dry bulb temperature and 50% relative humidity.

For each of these tests the following shall be completed:

Phase 1: System COP

During the first phase, water in the storage vessel will be heated from an initial temperature of 30°C up to the heat pump set water temperature of at least 55°C.

Phase 2: Tapping profile

During this phase the performance of the heat pump is monitored while a residential tapping profile is being simulated. This is achieved by drawing water from the top of the storage vessel at different flow rates and volumes, at different times during the test. The water drawn from the storage vessel is checked for compliancy against the local SANS 151: *Fixed electric storage water heaters* regulation stating that the water temperature must remain above 50°C.

Phase 3: Instantaneous COP

During this phase the average COP based on the results of phase 2 is calculated.

The heat pump shall adhere to all requirements as listed in Phases 1, 2 and 3 at all times in order to successfully pass the performance test. Only heat pump installations conforming to all requirements are allowed to provide their performance results to customers. Only the performance results as listed in Section 7.3 may be listed.

7.1.1 Phase 1 (System COP)

For phase 1, the following methodology is proposed to determine the system COP of a heat pump installation. The heat pump must be installed as per the supplier's installation requirements. After the water and electrical connections are completed, the test storage vessel temperature will be increased or decreased to 30°C ± 0.5°C. The water in the tank will be mixed while increasing or decreasing the tank temperature to ensure the water in the tank is at a constant temperature. Care shall be taken to ensure that the water temperature is as close to 30°C as possible, as a variance in temperature has a significant effect on a heat pump's performance. At the same time the test room's dry bulb temperature and relative humidity will be adjusted to the set point of the appropriate ambient test condition.

Once all the conditions have been met, the measurement equipment will be activated to record all parameters as set out in Section 7.2.3. The heat pump will then be switched on and allowed to heat the water until the set water temperature of 55°C or higher has been reached. The point in time when the heat pump switches off will be indicated in the recorded data by a drop in the electrical current drawn by the unit for an extended period of time. The duration of the heating cycle will then be recorded, where after the power supply to the heat pump will be switched off to avoid any reheating taking place. The water in the testing storage vessel will then be mixed until a constant temperature is reached, before the tank temperature is recorded.

The final mixed water temperature of the storage tank shall be within ±2°C of the heat pump set water temperature for all test conditions. If this requirement is not met, the unit has failed the test requirements. If the water temperature complies with the requirement, the system COP will be calculated with equation 7.1 as applied in the Eskom rebate methodology.

$$COP_{sys} = \frac{V_t \times [(\rho_f \times U_{ff}) - (\rho_s \times U_{fs})]}{E_{tot}} \quad [7.1]$$

With:

V_t: Volume of test tank [m^3].

ρ_f: Density of water at final tank temperature [kg/m^3].

ρ_s: Density of water at initial tank temperature [kg/m^3].

U_{ff}: Internal energy of the water at final tank temperature [kJ/kg.K].

U_{fs}: Internal energy of the water at initial tank temperature [kJ/kg.K].

E_{tot}: Total electrical energy consumed during the heating cycle [Joules].

The calculated system COP will then be compared with the applicable requirements as listed below:

- The system COP may not be less than 0.8 at 3°C dry bulb temperature and 50% relative humidity.
- The system COP may not be less than 2.0 at 18°C dry bulb temperature and 50% relative humidity.
- The heat pump shall remain operational without the activation of a safety alarm for all testing conditions.

Phase 2 shall directly follow after phase 1 with no interruptions.

7.1.2 Phase 2 (Tapping Profile)

With the power to the heat pump switched off, the measurement equipment recording the parameters will be reactivated. The system will be allowed to remain undisturbed for 20 minutes, before the electrical supply to the heat pump will be reactivated, followed by allowing the system to remain undisturbed for a further 10 minutes. Should the heat pump enter a heating cycle within this time; the heat pump shall be allowed to reheat the water to the set temperature before the tapping profile is started. The heat pump performance from the start of phase 2 including all reheating cycles will be taken into account for the average water temperature difference and average instantaneous COP. The tapping profile as set out in Table 5 shall then follow. Water will be drawn from the test vessel at the time intervals as set out in Column 1 of Table 5. The volume of water drawn from the test vessel for each time interval should comply with the value listed in Column 3, while being drawn at the flow rate listed in Column 4. Column 2 of Table 5 indicates the description of the tapping profile being simulated. The time intervals are calculated by adding the minutes to the start time as listed in Column 1 of Table 5.

Table 5: Example of a tapping profile.

Start time	Tapping profile description	Volume (litre)	Flow rate (l/min)
Start	Basin	2	6
+ 20 min	Basin	2	6
+ 40 min	Shower	45	10
+ 65 min	Shower	35	10
+ 120 min	Scullery	6	10
+ 650 min	Shower	35	10
+ 680 min	Scullery	6	10
+ 720 min	Bath	50	20
+ 735 min	Bath top up	15	20
+ 780 min	Basin	2	6
+ 1440 min	Full draw down	Total vessel volume	20

The tapping profile is set up to simulate the typical water usage of a residential home. This tapping profile includes periods of time that the heat pump will be required to heat standing losses, as well as a full draw down of the tank volume after 24 hours.

Phase 2 shall conform to the following requirements at all times:

- The temperature of the water drawn from the storage vessel may not drop below 50°C except for the final full capacity draw down test.
- At least 70% of the storage vessel's volume must be drawn at a temperature above 45°C during the full draw down test.
- The heat pump shall remain operational without the activation of a safety alarm for the entire test phase.
- The heat pump shall be allowed to reheat the water at any time during the tapping profile test.

7.1.3 Phase 3 (Instantaneous COP)

Due to differences found within installations in the South African market, the tank temperature as described in Chapter 6 of this this study is not necessarily the temperature of the water entering the heat pump. The water inlet temperature will therefore be used as (T_t) within the proposed phase 3 testing methodology.

To accurately determine the instantaneous COP the volume of recorded data for phase 2 needs to be reduced. All the data where the heat pump was not operational in the recorded data for phase 2 must be deleted, as well as data recoded during the heat pump start-up sequence. For each recorded time interval the temperature difference of the heat pump inlet water temperature (T_t) and the ambient dry bulb temperature (T_a) must be calculated using equation 7.2.

$$T_\Delta = T_t - T_a \qquad [7.2]$$

Where:

T_Δ: Temperature difference [°C].

T_t: Heat pump inlet water temperature [°C].

T_A: Ambient dry bulb temperature [°C].

After the temperature difference has been calculated, the instantaneous COP for each of the recorded time intervals is calculated using equation 7.3 and 7.4. Equation 7.3 is used to determine the total heat energy supplied to the water.

$$Q_{th} = \dot{m} \times Cp \times (T_{out} - T_{in}) \qquad [7.3]$$

Where:

Q_{th} Thermal energy added to the water [kW].

$\dot{m}$ Water mass flow rate through the heat pump [kg/s].

Cp Specific heat capacity of the water [kJ/kgK].

T_{out} Temperature of the water at the outlet [K].

T_{in} Temperature of the water at the inlet [K].

The value for the specific heat capacity Cp of the water is calculated per time interval from the inlet water temperature T_{in}.

Equation 7.4 can then be used to determine the instantaneous COP of the HPWH.

$$COP_{inst} = \frac{Q_{th}}{P_{elec}} \qquad [7.4]$$

COP_{inst} Instantaneous coefficient of performance [kW/kW].

Q_{th} Thermal energy added to the water [kW].

P_{elec} Electricity used [kW].

The average instantaneous COP is calculated across all recorded time steps. The calculated instantaneous COP will then be compared with the applicable requirements listed below:

- The average instantaneous COP may not be less than 1 at 3°C dry bulb temperature and 50% relative humidity.
- The average instantaneous COP may not be less than 2.2 at 18°C dry bulb temperature and 50% relative humidity.

7.2 Testing equipment

7.2.1 Test room specifications

The test room must be able to accommodate an entire heat pump and installation in height and width without affecting the heat pump's air flow. The test room must be able to maintain a mean dry bulb temperature to an accuracy of ±0.5°C, while maintaining a mean relative humidity within ±5%. The test room must be capable of supplying water from an external vessel to the heat pump installation during the tests. The supply water temperature will be maintained at 16°C ± 1°C as this is the average municipal water temperature. The test room shall be equipped with a waterproof floor and drain point for condensation due to the nature of the tests being conducted.

7.2.2 Test storage vessel

The test storage vessel shall conform to the local requirements as set out in SANS 151: *Fixed electric storage water heaters*. The storage vessel shall be 150 litres, 250 litres or 450 litres depending on the heating capacity and application of the HPWH being tested. The volume of the tank must be verified to an accuracy of ±0.5 litres as it will be used for the calculation of the system COP.

7.2.3 Measuring equipment

The measurement equipment will be used to record the following:

Temperatures:

- Ambient dry bulb air temperature inside the test chamber, measured in front of the heat pump evaporator inlet.
- Heat pump inlet water temperature.
- Heat pump outlet water temperature.
- Geyser inlet water temperature.
- Geyser outlet water temperature.
- Geyser tank temperature at the thermostat pocket of the geyser.

Relative humidity:

- Ambient relative humidity inside the test chamber.

Water flow rate:

- Supply water volume flow rate for use during phase 2, measured in the main cold water supply line to the installation.
- Water volume flow rate through the heat pump, measured on the inlet to the heat pump.

Electrical consumption:

- Instantaneous electrical supply voltage to the heat pump.
- Instantaneous electrical current drawn by the heat pump.

The measurement equipment shall be installed on the test installation as depicted in Figure 27 below:

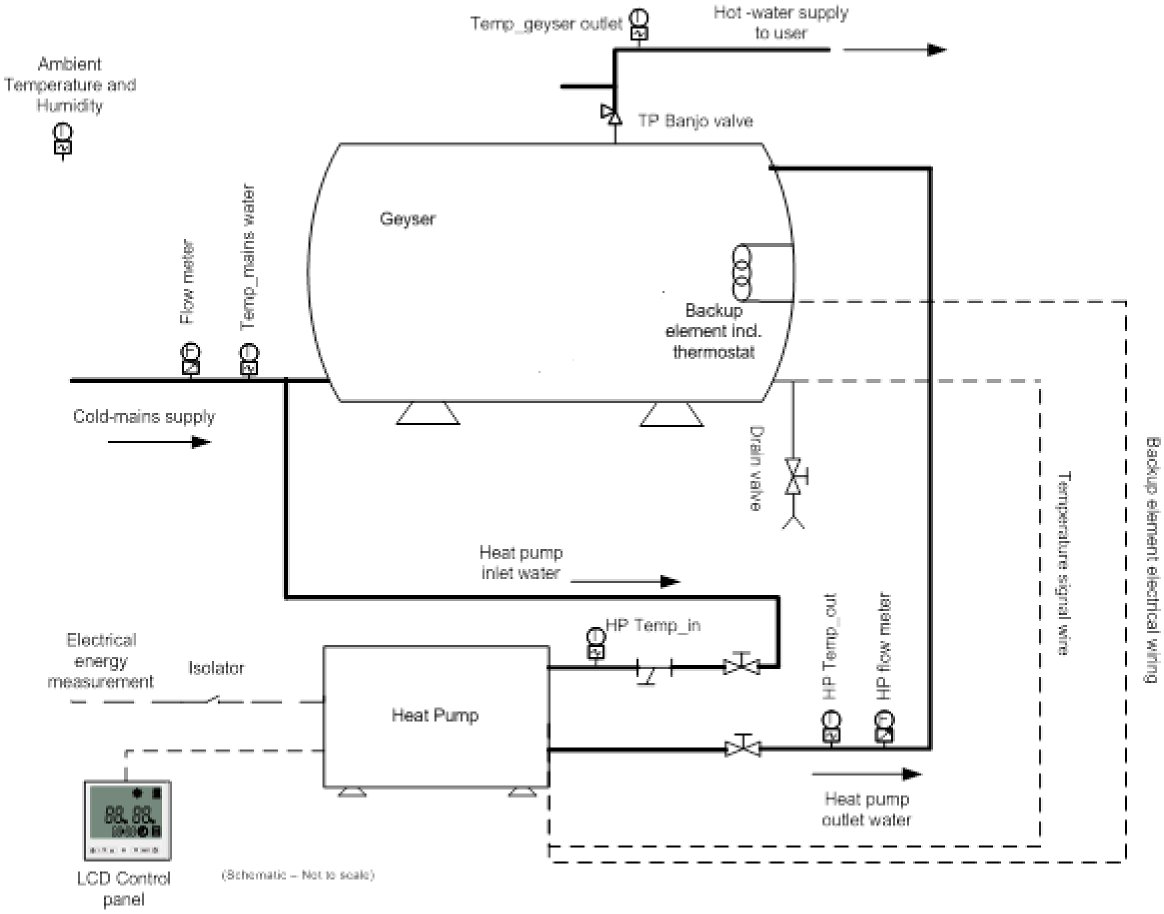

Figure 27: Measurement equipment positions on installation.

All measuring equipment must be capable of recording the respective values within the following accuracies:

- Temperature measuring equipment shall be capable of measuring to an accuracy of ±0.5°C.
- Relative humidity measuring equipment shall be capable of measuring to an accuracy of ±3%.
- Flow measurement equipment shall be capable of measuring to an accuracy of ±0.3 l/min.
- Electrical energy measuring equipment shall be capable of measuring to an accuracy of 5%.

7.3 Performance rating of the heat pump

The final part of the methodology is to plot the instantaneous COP as calculated per recorded time step in equation 7.4 (on the y-axis), against the temperature difference as calculated per recorded time step in equation 7.2 (on the x-axis) for all tests, on the same graph. A linear regression line, similar to what was done in Chapter 6, is then drawn through the performance indication lines of the three tests. This line shall be used as the heat pump's performance line to predict the unit performance under different ambient conditions and water temperatures.

The average of the temperature difference $(T_t - T_a)$ for the results obtained during the 18°C dry bulb temperature test in phase 2 is calculated and the COP value at the average of the temperature difference $(T_t - T_a)$ from the performance line is determined. This COP value shall then be used as the heat pump's rated COP at 18°C and 50% relative humidity.

CHAPTER 8 : CONCLUSIONS AND RECOMMENDATIONS

8.1 Conclusions

The number of heat pumps installed in the South African market is constantly growing. Subsequently, questions are being raised with regards to the actual seasonal in-field efficiencies thereof. The aim of this study was to show that a laboratory test methodology can be developed and used to accurately predict a heat pump's seasonal in-field performance.

The literature study indicated that the three critical performance influencing factors to take into account when testing a heat pump's efficiency is the dry bulb air temperature, the temperature of the water entering the heat pump and the relative humidity. The influence of these factors on the heat pump's performance was then studied within in-field measurements, a laboratory test and a Flownex® SE simulation model. The results obtained from the recorded measurements and generated results were then compared for verification and validation.

The results indicated that a laboratory test methodology developed by combining the Eskom rebate test and British standard could accurately represent the seasonal in-field performance within 3%. With all the results from this study and the literature considered, a proposed laboratory testing methodology was then developed to accurately represent a heat pump's in-field performance.

8.2 Recommendations for future studies

The following recommendations should be considered for future studies within this field.

The system COP could not be calculated within this study due to instrumentation failure. Literature suggests that the benefit of calculating the system COP is that it gives a more accurate representation of a HPWHs performance. However, literature also indicates that this method has a disadvantage as it is only a summarising value. The benefits and disadvantages of tracing performance using instantaneous COP and system COP must be carefully studied to determine which calculation method should be used in laboratory tests.

Future in-field studies should consider installing a new type of water flow meter that only became available on the market close to the end of this study. These flow meters have been designed for hot water billing purposes and are therefore built specifically for the environment it will be used in. These flow meters are available with two temperature probes and electrical consumption meters built into it, to be able to measure COP directly. The recorded values can be downloaded individually or as a calculated COP value. These meters will therefore combine most of the water side measuring equipment used in this study reducing the price of the recording system dramatically. Future studies tracing in-field performance should also include residential homes of different sizes to ensure tapping profiles are generated for small, medium and large residential homes within South Africa.

It is also recommended that the laboratory test methodology as proposed within this study is included and further studied in future studies on this subject. The proposed method can benefit from system COPs being recorded as well as actual tapping profiles found within South Africa.

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Annexure A: In-field data

Annexure A is used to provide supporting tables, figures and detail with regards to the in-field data measurements and results.

A.1 Example of raw recorded data

Table A.1 below shows a sample of the in-field recorded data.

Table A1: Example of in-field recorded data

Date / Time	Energy Used (kJ/Minute)	Ambient Temp [°C]	2553 RH Ambient (%)	Tank Temp [°C]	HP OUT Temp [°C]	HP Water Flow [l/min]
2013/06/03 17:45	0.02	8.15	38.38	38.5	41.26	13.6
2013/06/03 17:46	0.021	8.15	38.13	38.65	41.42	13.6
2013/06/03 17:47	0.021	8.23	38.56	38.97	41.66	13.6
2013/06/03 17:48	0.022	8.07	38.38	39.21	41.98	13.6
2013/06/03 17:49	0.021	7.99	38.38	39.44	42.21	13.6
2013/06/03 17:50	0.021	7.99	38.38	39.68	42.45	13.6
2013/06/03 17:51	0.021	7.99	38.13	40	42.69	13.6
2013/06/03 17:52	0.022	7.91	37.94	40.24	42.92	13.6
2013/06/03 17:53	0.021	7.76	37.94	40.39	43.08	13.6
2013/06/03 17:54	0.022	7.76	38.13	40.71	43.4	13.6
2013/06/03 17:55	0.021	7.68	37.97	41.02	43.71	13.6
2013/06/03 17:56	0.022	7.76	37.94	41.26	43.87	13.6
2013/06/03 17:57	0.022	7.68	38.13	41.58	44.19	13.6
2013/06/03 17:58	0.022	7.68	38.38	41.74	44.35	13.6
2013/06/03 17:59	0.022	7.68	38.56	41.9	44.5	13.6
2013/06/03 18:00	0.022	7.68	38.38	42.13	44.74	13.6
2013/06/03 18:01	0.022	7.6	38.38	42.37	44.9	13.6
2013/06/03 18:02	0.022	7.6	38.5	42.61	45.06	13.6
2013/06/03 18:03	0.021	7.6	38.38	36.61	38.74	13.6
2013/06/03 18:04	0.017	7.52	38.38	30.61	32.42	13.6
2013/06/03 18:05	0.02	7.28	38.13	35.26	37.63	13.6
2013/06/03 18:06	0.021	7.13	38.38	37.79	40.31	13.6
2013/06/03 18:07	0.021	7.13	38.38	39.44	41.9	13.6
2013/06/03 18:08	0.021	7.05	38.56	40.55	43	13.6
2013/06/03 18:09	0.022	7.13	38.53	41.18	43.63	13.6
2013/06/03 18:10	0.021	7.13	38.75	41.66	44.11	13.6
2013/06/03 18:11	0.022	7.13	39.16	42.13	44.58	13.6
2013/06/03 18:12	0.022	7.05	38.75	42.37	44.82	13.6
2013/06/03 18:13	0.023	7.13	38.53	42.61	45.13	13.6
2013/06/03 18:14	0.022	7.05	38.38	42.84	45.29	13.6
2013/06/03 18:15	0.022	7.05	38.53	43.08	45.53	13.6
2013/06/03 18:16	0.023	7.05	38.53	43.32	45.77	13.6
2013/06/03 18:17	0.022	7.05	38.38	43.56	46.01	13.6
2013/06/03 18:18	0.022	7.05	38.38	43.79	46.17	13.6
2013/06/03 18:19	0.023	7.05	38.13	44.11	46.48	13.6
2013/06/03 18:20	0.023	6.97	38.13	44.27	46.64	13.6
2013/06/03 18:21	0.022	6.97	38.13	44.5	46.88	13.6
2013/06/03 18:22	0.023	6.97	38.38	44.74	47.04	13.6
2013/06/03 18:23	0.022	6.89	38.56	44.98	47.35	13.6
2013/06/03 18:24	0.023	6.97	38.38	45.13	47.51	13.6
2013/06/03 18:25	0.023	6.97	38.13	45.29	47.67	13.6
2013/06/03 18:26	0.023	6.97	38.38	45.45	47.75	13.6
2013/06/03 18:27	0.023	6.97	38.13	45.77	48.14	13.6
2013/06/03 18:28	0.024	6.97	37.75	46.01	48.3	13.6
2013/06/03 18:29	0.023	6.97	37.94	46.25	48.54	13.6

A.2 Initial Excel macro results

The following tables and figures give an example to the tables and figures generated for initial data reduction and data error identification.

Table A2: Summary of single month's results.

Performance Parameters for Reporting Period			
Total Hot Water Used [l]:	42485.5	**Average Ambient Temperature**	15.72
Total Energy Used by Unit [kWh]:	1850.32	**Average Error in data:**	0.42%
Average COP	1.71		

Table A3: Summary of all the monthly results.

Day:	Hot Water Usage	COP	Instant COP	Ambient Temp	Humidity	% Untrusted data	Energy used	7AM - 10AM	6PM - 8PM	Off peak
June	3907.5	8.87775618	1.185226316	6.91709905	47.11931	0.05%	259.62	54.03	19.59	186.00
July	3680.5	1.45859952	1.400905996	8.833694189	47.93414	0.43%	255.41	47.93	23.44	184.04
August	4705	1.50024462	1.41855368	9.526622212	44.955	0.29%	267.45	56.56	18.56	192.32
September	5413.5	1.82488586	1.729214741	13.02415037	30.41307	0.31%	200.80	54.43	11.08	135.29
October	4598.5	1.93712185	1.747429294	16.58521549	44.13013	0.92%	172.69	36.84	13.69	122.16
November	4697.5	2.17630367	1.993209029	19.01907192	44.87237	0.25%	150.08	33.02	12.47	104.60
Desember	2367	2.11146626	1.924722329	21.35303589	50.94803	0.14%	111.74	15.99	10.62	85.13
January	2376	2.2372765	1.908791778	23.19843583	47.95502	0.97%	103.13	16.77	7.42	78.94
February	3454.5	2.3590269	2.073796462	20.73321031	68.36491	0.22%	105.57	22.82	7.34	75.42
March	5040.5	2.29697879	1.89482844	18.7049995	64.03961	0.99%	146.69	35.00	15.60	96.09
April	2245	2.1573221	1.531975303	15.05606501	54.47164	0.00%	77.14	19.34	7.78	50.02
Sum :	42485.5						1850.32	392.72	147.58	1310.02
Average:		2.63063475	1.709877579	15.72287271	49.56393	0.004157774				

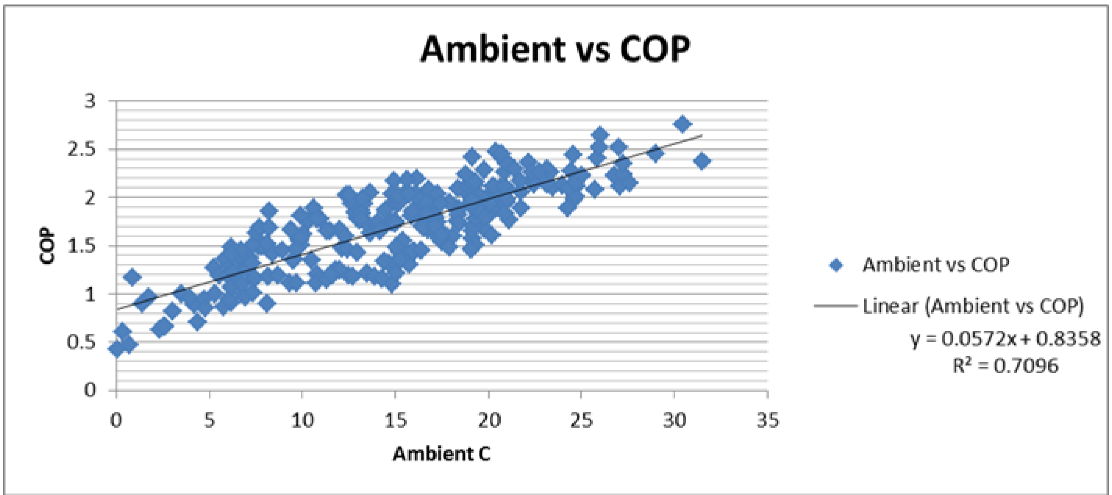

Figure A1: Initial performance regression line without the consideration of RH or tank temperature.

A.3 Excel VBA for final data reduction

This section gives the programming code used to reduce the in-field recorded measurements to the performance lines used within the study.

Button 1:

```
Sub Macro3()

Application.ScreenUpdating = False

'Macro3 Macro


'Sort Data by Energy Consumption

 Dim LastRow As Long, sht As Worksheet, rng As Range, v As Variant, i As Long

  Set sht = ThisWorkbook.Worksheets("Data")

 LastRow = sht.Cells(sht.Rows.Count, "A").End(xlUp).Row


 Range("A4:K4").Select

    Range(Selection, Selection.End(xlDown)).Select

    ActiveWorkbook.Worksheets("Data").Sort.SortFields.Clear

    ActiveWorkbook.Worksheets("Data").Sort.SortFields.Add Key:=Range("H5:H" & LastRow _

        ), SortOn:=xlSortOnValues, Order:=xlDescending, DataOption:=xlSortNormal

    With ActiveWorkbook.Worksheets("Data").Sort

        .SetRange Range("A4:K" & LastRow)

        .Header = xlYes

        .MatchCase = False

        .Orientation = xlTopToBottom
```

```
        .SortMethod = xlPinYin
        .Apply
    End With
'Delete all data with lower than start-up current
Dim ELowRow As Excel.Range
v = 0.012
Set ELowRow = sht.Range("H:H").Find(what:=v, lookat:=xlWhole)
If Not ELowRow Is Nothing Then
i = ELowRow.Row
With Sheets("Data")
    .Rows(i & ":" & .Rows.Count).Delete
End With
Else
MsgBox (v & "Not Found")
End If

'Locate pre start-up conditions by checking water temp in and out
'Calculate temp differance
Dim NewlastRow As Long
NewlastRow = sht.Cells(sht.Rows.Count, "A").End(xlUp).Row
Range("L4").Select
    ActiveCell.FormulaR1C1 = "=RC[-6]-RC[-7]"
    Selection.AutoFill Destination:=Range("L4:L" & NewlastRow)
    Range("L4:L" & NewlastRow).Select
```

```
'Sort table according to temp difference
Range("A4").Select
    Range(Selection, Selection.End(xlToRight)).Select
    Range(Selection, Selection.End(xlDown)).Select
    ActiveWorkbook.Worksheets("Data").Sort.SortFields.Clear
    ActiveWorkbook.Worksheets("Data").Sort.SortFields.Add Key:=Range("L5:L" & NewlastRow) _
        , SortOn:=xlSortOnValues, Order:=xlDescending, DataOption:=xlSortNormal
    With ActiveWorkbook.Worksheets("Data").Sort
        .SetRange Range("A4:L" & NewlastRow)
        .Header = xlYes
        .MatchCase = False
        .Orientation = xlTopToBottom
        .SortMethod = xlPinYin
        .Apply
    End With
'Change formula to readable value
Range("L4").Select
    Range(Selection, Selection.End(xlDown)).Select
    Selection.Copy
    Selection.PasteSpecial Paste:=xlPasteValues, Operation:=xlNone, SkipBlanks _
        :=False, Transpose:=False

'Delete all value lower than 0.5degC
```

```
Range("L4:L" & NewlastRow).Select

Dim n As Long

Dim TLowRow As Excel.Range

Dim DTemp As Variant

DTemp = 1.8

Set TLowRow = sht.Range("L:L").Find(what:=DTemp, lookat:=xlWhole)

If Not TLowRow Is Nothing Then

n = TLowRow.Row

With Sheets("Data")

   .Rows(n & ":" & .Rows.Count).Delete

End With

Else

MsgBox (DTemp & "Not Found")

End If

Selection.Delete

Application.ScreenUpdating = True

End Sub
```

Button 2:

```
Sub Macro1()

' Macro1 Macro

Range("A1").Select

Application.ScreenUpdating = False

'Sort values according to water flow
```

```
Dim LastRow As Long, sht As Worksheet, F As Variant
Set sht = ThisWorkbook.Worksheets("Data")
 LastRow = sht.Cells(sht.Rows.Count, "A").End(xlUp).Row


 Range("A4:K4").Select
    Range(Selection, Selection.End(xlDown)).Select
    ActiveWorkbook.Worksheets("Data").Sort.SortFields.Clear
    ActiveWorkbook.Worksheets("Data").Sort.SortFields.Add Key:=Range("J5:J" &
LastRow _
        ), SortOn:=xlSortOnValues, Order:=xlDescending, DataOption:=xlSortNormal
    With ActiveWorkbook.Worksheets("Data").Sort
        .SetRange Range("A4:K" & LastRow)
        .Header = xlYes
        .MatchCase = False
        .Orientation = xlTopToBottom
        .SortMethod = xlPinYin
        .Apply
    End With


'Check if the flow was recorded correctly
Range("M12").Select
ActiveCell.FormulaR1C1 = "=AVERAGE(R[-8]C[-3]:R[" & LastRow & "]C[-3])"
F = Range("M12").Value
If F < 13.3 Then
F = 13.3075
```

```
End If
Range("J4").Value = F
Range("M12").Clear

'Set all the flow to the average flow as there are double recordings
'that recorded double and some that recorded only half
Range("J4").Select
    Selection.AutoFill Destination:=Range("J4:J" & LastRow)

'change the formating back to normal
Range("J4:J" & LastRow).Select
    Selection.Borders(xlDiagonalDown).LineStyle = xlNone
    Selection.Borders(xlDiagonalUp).LineStyle = xlNone
    With Selection.Borders(xlEdgeLeft)
        .LineStyle = xlContinuous
        .ColorIndex = 0
        .TintAndShade = 0
        .Weight = xlThin
    End With
    With Selection.Borders(xlEdgeTop)
        .LineStyle = xlContinuous
        .ColorIndex = 0
        .TintAndShade = 0
        .Weight = xlThin
```

```
End With
With Selection.Borders(xlEdgeBottom)
    .LineStyle = xlContinuous
    .ColorIndex = 0
    .TintAndShade = 0
    .Weight = xlThin
End With
With Selection.Borders(xlEdgeRight)
    .LineStyle = xlContinuous
    .ColorIndex = 0
    .TintAndShade = 0
    .Weight = xlThin
End With
With Selection.Borders(xlInsideVertical)
    .LineStyle = xlContinuous
    .ColorIndex = 0
    .TintAndShade = 0
    .Weight = xlThin
End With
With Selection.Borders(xlInsideHorizontal)
    .LineStyle = xlContinuous
    .ColorIndex = 0
    .TintAndShade = 0
    .Weight = xlThin
```

```
    End With

'Remove all element cycles
Range("A4:K4").Select
    Range(Selection, Selection.End(xlDown)).Select
    ActiveWorkbook.Worksheets("Data").Sort.SortFields.Clear
    ActiveWorkbook.Worksheets("Data").Sort.SortFields.Add Key:=Range("H5:H" & LastRow) _
        , SortOn:=xlSortOnValues, Order:=xlAscending, DataOption:=xlSortNormal
    With ActiveWorkbook.Worksheets("Data").Sort
        .SetRange Range("A4:K" & LastRow)
        .Header = xlYes
        .MatchCase = False
        .Orientation = xlTopToBottom
        .SortMethod = xlPinYin
        .Apply
    End With

'Delete all element values
Dim ELowRow As Excel.Range
Dim i As Long
v = 0.028
Set ELowRow = sht.Range("H:H").Find(what:=v, lookat:=xlWhole)
If Not ELowRow Is Nothing Then
i = ELowRow.Row
```

```
With Sheets("Data")
    .Rows(i & ":" & .Rows.Count).Delete
End With
Else
MsgBox (v & "Not Found")
End If
Application.ScreenUpdating = True
Range("A1").Select
End Sub
```

Button 3:

```
Sub Macro4()
```

// Confidential

Button 4:

```
Sub Macro2()
' Macro2 Macro
'Sort according to humidity
Application.ScreenUpdating = False
Dim LastRow As Long, sht As Worksheet, F As Variant

 Set sht = ThisWorkbook.Worksheets("Data")
 LastRow = sht.Cells(sht.Rows.Count, "A").End(xlUp).Row
    Range("A4:K13").Select
    Range(Selection, Selection.End(xlDown)).Select
```

```
   ActiveWorkbook.Worksheets("Data").Sort.SortFields.Clear
   ActiveWorkbook.Worksheets("Data").Sort.SortFields.Add Key:=Range("B5:B" & LastRow) _
      , SortOn:=xlSortOnValues, Order:=xlAscending, DataOption:=xlSortNormal
   With ActiveWorkbook.Worksheets("Data").Sort
      .SetRange Range("A4:K" & LastRow)
      .Header = xlYes
      .MatchCase = False
      .Orientation = xlTopToBottom
      .SortMethod = xlPinYin
      .Apply
   End With

'Cut data for 90% humidity

Dim ELowRow As Excel.Range
v = 85
Set ELowRow = sht.Range("B:B").Find(what:=v, lookat:=xlWhole)
If Not ELowRow Is Nothing Then
i = ELowRow.Row
With Sheets("Data")
   .Rows(i & ":" & .Rows.Count).Cut
End With
Else
MsgBox (v & "Not Found")
```

```
End If
Sheets("90%").Select
Range("A4").Select
    ActiveSheet.Paste
Sheets("Data").Select
Range("A1").Select

'Cut data for 80% humidity
Dim ELowRow80 As Excel.Range
v = 75
Set ELowRow80 = sht.Range("B:B").Find(what:=v, lookat:=xlWhole)
If Not ELowRow80 Is Nothing Then
i = ELowRow80.Row
With Sheets("Data")
    .Rows(i & ":" & .Rows.Count).Cut
End With
Else
MsgBox (v & "Not Found")
End If
Sheets("80%").Select
Range("A4").Select
    ActiveSheet.Paste

Sheets("Data").Select
```

```
Range("A1").Select


Application.ScreenUpdating = True

End Sub
```

Button 5:

```
Sub Macro5()

' Macro5 Macro

Application.ScreenUpdating = False

Dim LastRow As Long, sht As Worksheet, F As Variant


 Set sht = ThisWorkbook.Worksheets("Data")

 LastRow = sht.Cells(sht.Rows.Count, "A").End(xlUp).Row

'Cut data for 70% humidity


Dim ELowRow70 As Excel.Range

v = 65

Set ELowRow70 = sht.Range("B:B").Find(what:=v, lookat:=xlWhole)

If Not ELowRow70 Is Nothing Then

i = ELowRow70.Row

With Sheets("Data")

    .Rows(i & ":" & .Rows.Count).Cut

Sheets("70%").Select

Range("A4").Select
```

```
    ActiveSheet.Paste
End With
Else
MsgBox (v & "Not Found")
End If
Sheets("Data").Select
Range("A1").Select

'Cut data for 60% humidity

Dim ELowRow60 As Excel.Range
v = 55
Set ELowRow60 = sht.Range("B:B").Find(what:=v, lookat:=xlWhole)
If Not ELowRow60 Is Nothing Then
i = ELowRow60.Row
With Sheets("Data")
    .Rows(i & ":" & .Rows.Count).Cut
Sheets("60%").Select
Range("A4").Select
    ActiveSheet.Paste
End With
Else
MsgBox (v & "Not Found")
End If
```

```
Sheets("Data").Select
Range("A1").Select

'Cut data for 50% humidity

Dim ELowRow50 As Excel.Range
v = 45
Set ELowRow50 = sht.Range("B:B").Find(what:=v, lookat:=xlWhole)
If Not ELowRow50 Is Nothing Then
i = ELowRow50.Row
With Sheets("Data")
    .Rows(i & ":" & .Rows.Count).Cut
Sheets("50%").Select
Range("A4").Select
    ActiveSheet.Paste
End With
Else
MsgBox (v & "Not Found")
End If
Sheets("Data").Select
Range("A1").Select

'Cut data for 40% humidity
```

```
Dim ELowRow40 As Excel.Range

v = 35

Set ELowRow40 = sht.Range("B:B").Find(what:=v, lookat:=xlWhole)

If Not ELowRow40 Is Nothing Then

i = ELowRow40.Row

With Sheets("Data")

   .Rows(i & ":" & .Rows.Count).Cut

Sheets("40%").Select

Range("A4").Select

   ActiveSheet.Paste

End With

Else

MsgBox (v & "Not Found")

End If

Sheets("Data").Select

Range("A1").Select


'Cut data for 30% humidity


Dim ELowRow30 As Excel.Range

v = 25

Set ELowRow30 = sht.Range("B:B").Find(what:=v, lookat:=xlWhole)

If Not ELowRow30 Is Nothing Then

i = ELowRow30.Row
```

```
With Sheets("Data")
    .Rows(i & ":" & .Rows.Count).Cut
Sheets("30%").Select
Range("A4").Select
    ActiveSheet.Paste
End With
Else
MsgBox (v & "Not Found")
End If
Sheets("Data").Select
Range("A1").Select

'Cut data for 20% humidity

Dim ELowRow20 As Excel.Range
v = 15
Set ELowRow20 = sht.Range("B:B").Find(what:=v, lookat:=xlWhole)
If Not ELowRow20 Is Nothing Then
i = ELowRow20.Row
With Sheets("Data")
    .Rows(i & ":" & .Rows.Count).Cut
Sheets("20%").Select
Range("A4").Select
    ActiveSheet.Paste
```

```
End With
Else
MsgBox (v & "Not Found")
End If
Sheets("Data").Select
Range("A1").Select

'Cut data for 10% humidity
 Range("A4").Select
    Range(Selection, Selection.End(xlToRight)).Select
    Range(Selection, Selection.End(xlDown)).Select
    Selection.Cut
    Sheets("10%").Select
    Range("A4").Select
    ActiveSheet.Paste
    Sheets("Data").Select

Application.ScreenUpdating = True
End Sub
```

Button 6:

```
Sub Macro6()
' Macro6 Macro
Application.ScreenUpdating = False
```

```
'Calculate Data For 90%

Dim LastRow90 As Long, sht90 As Worksheet

 Sheets("Data").Select

    Range("A1:K3").Select

    Selection.Copy

    Sheets("90%").Select

    Range("A1").Select

    ActiveSheet.Paste

 Set sht90 = ThisWorkbook.Worksheets("90%")

 LastRow90 = sht90.Cells(sht90.Rows.Count, "A").End(xlUp).Row

    Range("L4").Select

    Application.CutCopyMode = False

    ActiveCell.FormulaR1C1 = "=RC[-4]*60"

    Range("M4").Select

    ActiveCell.FormulaR1C1 = "=(RC[-3]/60)*4.184*(RC[-7]-RC[-8])"

    Range("N4").Select

    ActiveCell.FormulaR1C1 = "=RC[-9]-RC[-7]"

    Range("O4").Select

    ActiveCell.FormulaR1C1 = "=RC[-2]/RC[-3]"

    Selection.AutoFill Destination:=Range("O4:O" & LastRow90)

    Range("O4:O" & LastRow90).Select

    Range("N4").Select

    Selection.AutoFill Destination:=Range("N4:N" & LastRow90)
```

```
Range("N4:N" & LastRow90).Select

Range("M4").Select

Selection.AutoFill Destination:=Range("M4:M" & LastRow90)

Range("M4:M" & LastRow90).Select

Range("L4").Select

Selection.AutoFill Destination:=Range("L4:L" & LastRow90)

Range("L4:L" & LastRow90).Select


'Move 90% data to Data Page

   Range("N4").Select

Range(Selection, Selection.End(xlToRight)).Select

Range(Selection, Selection.End(xlDown)).Select

Selection.Copy

Sheets("Data").Select

Range("A5").Select

ActiveSheet.Paste

Selection.PasteSpecial Paste:=xlPasteValues, Operation:=xlNone, SkipBlanks _

   :=False, Transpose:=False

Range("A4").Select

Application.CutCopyMode = False

ActiveCell.FormulaR1C1 = "90%"

Range("A1").Select


'Calculate Data For 80%
```

```
Dim LastRow80 As Long, sht80 As Worksheet
 Sheets("Data").Select
    Range("A1:K3").Select
    Selection.Copy
    Sheets("80%").Select
    Range("A1").Select
    ActiveSheet.Paste
 Set sht80 = ThisWorkbook.Worksheets("80%")
 LastRow80 = sht80.Cells(sht80.Rows.Count, "A").End(xlUp).Row
    Range("L4").Select
    Application.CutCopyMode = False
    ActiveCell.FormulaR1C1 = "=RC[-4]*60"
    Range("M4").Select
    ActiveCell.FormulaR1C1 = "=(RC[-3]/60)*4.184*(RC[-7]-RC[-8])"
    Range("N4").Select
    ActiveCell.FormulaR1C1 = "=RC[-9]-RC[-7]"
    Range("O4").Select
    ActiveCell.FormulaR1C1 = "=RC[-2]/RC[-3]"
    Selection.AutoFill Destination:=Range("O4:O" & LastRow80)
    Range("O4:O" & LastRow80).Select
    Range("N4").Select
    Selection.AutoFill Destination:=Range("N4:N" & LastRow80)
    Range("N4:N" & LastRow80).Select
    Range("M4").Select
```

```
Selection.AutoFill Destination:=Range("M4:M" & LastRow80)

Range("M4:M" & LastRow80).Select

Range("L4").Select

Selection.AutoFill Destination:=Range("L4:L" & LastRow80)

Range("L4:L" & LastRow80).Select


 'Move 80% data to Data Page

    Range("N4").Select

Range(Selection, Selection.End(xlToRight)).Select

Range(Selection, Selection.End(xlDown)).Select

Selection.Copy

Sheets("Data").Select

Range("C5").Select

ActiveSheet.Paste

Selection.PasteSpecial Paste:=xlPasteValues, Operation:=xlNone, SkipBlanks _

    :=False, Transpose:=False

Range("C4").Select

Application.CutCopyMode = False

ActiveCell.FormulaR1C1 = "80%"

Range("A1").Select


'Calculate Data For 70%

Dim LastRow70 As Long, sht70 As Worksheet

 Sheets("Data").Select
```

```
  Range("A1:K3").Select
  Selection.Copy
  Sheets("70%").Select
  Range("A1").Select
  ActiveSheet.Paste
Set sht70 = ThisWorkbook.Worksheets("70%")
LastRow70 = sht70.Cells(sht70.Rows.Count, "A").End(xlUp).Row
  Range("L4").Select
  Application.CutCopyMode = False
  ActiveCell.FormulaR1C1 = "=RC[-4]*60"
  Range("M4").Select
  ActiveCell.FormulaR1C1 = "=(RC[-3]/60)*4.184*(RC[-7]-RC[-8])"
  Range("N4").Select
  ActiveCell.FormulaR1C1 = "=RC[-9]-RC[-7]"
  Range("O4").Select
  ActiveCell.FormulaR1C1 = "=RC[-2]/RC[-3]"
  Selection.AutoFill Destination:=Range("O4:O" & LastRow70)
  Range("O4:O" & LastRow70).Select
  Range("N4").Select
  Selection.AutoFill Destination:=Range("N4:N" & LastRow70)
  Range("N4:N" & LastRow70).Select
  Range("M4").Select
  Selection.AutoFill Destination:=Range("M4:M" & LastRow70)
  Range("M4:M" & LastRow70).Select
```

```
  Range("L4").Select
  Selection.AutoFill Destination:=Range("L4:L" & LastRow70)
  Range("L4:L" & LastRow70).Select

  'Move 70% data to Data Page
    Range("N4").Select
  Range(Selection, Selection.End(xlToRight)).Select
  Range(Selection, Selection.End(xlDown)).Select
  Selection.Copy
  Sheets("Data").Select
  Range("E5").Select
  ActiveSheet.Paste
  Selection.PasteSpecial Paste:=xlPasteValues, Operation:=xlNone, SkipBlanks _
    :=False, Transpose:=False
  Range("E4").Select
  Application.CutCopyMode = False
  ActiveCell.FormulaR1C1 = "70%"
  Range("A1").Select

    'Calculate Data For 60%
Dim LastRow60 As Long, sht60 As Worksheet
 Sheets("Data").Select
  Range("A1:K3").Select
  Selection.Copy
```

```
    Sheets("60%").Select
    Range("A1").Select
    ActiveSheet.Paste
Set sht60 = ThisWorkbook.Worksheets("60%")
LastRow60 = sht60.Cells(sht60.Rows.Count, "A").End(xlUp).Row
    Range("L4").Select
    Application.CutCopyMode = False
    ActiveCell.FormulaR1C1 = "=RC[-4]*60"
    Range("M4").Select
    ActiveCell.FormulaR1C1 = "=(RC[-3]/60)*4.184*(RC[-7]-RC[-8])"
    Range("N4").Select
    ActiveCell.FormulaR1C1 = "=RC[-9]-RC[-7]"
    Range("O4").Select
    ActiveCell.FormulaR1C1 = "=RC[-2]/RC[-3]"
    Selection.AutoFill Destination:=Range("O4:O" & LastRow60)
    Range("O4:O" & LastRow60).Select
    Range("N4").Select
    Selection.AutoFill Destination:=Range("N4:N" & LastRow60)
    Range("N4:N" & LastRow60).Select
    Range("M4").Select
    Selection.AutoFill Destination:=Range("M4:M" & LastRow60)
    Range("M4:M" & LastRow60).Select
    Range("L4").Select
    Selection.AutoFill Destination:=Range("L4:L" & LastRow60)
```

```
    Range("L4:L" & LastRow60).Select


    'Move 60% data to Data Page
        Range("N4").Select
    Range(Selection, Selection.End(xlToRight)).Select
    Range(Selection, Selection.End(xlDown)).Select
    Selection.Copy
    Sheets("Data").Select
    Range("G5").Select
    ActiveSheet.Paste
    Selection.PasteSpecial Paste:=xlPasteValues, Operation:=xlNone, SkipBlanks _
        :=False, Transpose:=False
    Range("G4").Select
    Application.CutCopyMode = False
    ActiveCell.FormulaR1C1 = "60%"
    Range("A1").Select


    'Calculate Data For 50%
Dim LastRow50 As Long, sht50 As Worksheet
 Sheets("Data").Select
    Range("A1:K3").Select
    Selection.Copy
    Sheets("50%").Select
    Range("A1").Select
```

```
    ActiveSheet.Paste
Set sht50 = ThisWorkbook.Worksheets("50%")
LastRow50 = sht50.Cells(sht50.Rows.Count, "A").End(xlUp).Row
    Range("L4").Select
    Application.CutCopyMode = False
    ActiveCell.FormulaR1C1 = "=RC[-4]*60"
    Range("M4").Select
    ActiveCell.FormulaR1C1 = "=(RC[-3]/60)*4.184*(RC[-7]-RC[-8])"
    Range("N4").Select
    ActiveCell.FormulaR1C1 = "=RC[-9]-RC[-7]"
    Range("O4").Select
    ActiveCell.FormulaR1C1 = "=RC[-2]/RC[-3]"
    Selection.AutoFill Destination:=Range("O4:O" & LastRow50)
    Range("O4:O" & LastRow50).Select
    Range("N4").Select
    Selection.AutoFill Destination:=Range("N4:N" & LastRow50)
    Range("N4:N" & LastRow50).Select
    Range("M4").Select
    Selection.AutoFill Destination:=Range("M4:M" & LastRow50)
    Range("M4:M" & LastRow50).Select
    Range("L4").Select
    Selection.AutoFill Destination:=Range("L4:L" & LastRow50)
    Range("L4:L" & LastRow50).Select
```

```
'Move 50% data to Data Page
    Range("N4").Select
Range(Selection, Selection.End(xlToRight)).Select
Range(Selection, Selection.End(xlDown)).Select
Selection.Copy
Sheets("Data").Select
Range("I5").Select
ActiveSheet.Paste
Selection.PasteSpecial Paste:=xlPasteValues, Operation:=xlNone, SkipBlanks _
    :=False, Transpose:=False
Range("I4").Select
Application.CutCopyMode = False
ActiveCell.FormulaR1C1 = "50%"
Range("A1").Select

'Calculate Data For 40%
Dim LastRow40 As Long, sht40 As Worksheet
Sheets("Data").Select
Range("A1:K3").Select
Selection.Copy
Sheets("40%").Select
Range("A1").Select
ActiveSheet.Paste
Set sht40 = ThisWorkbook.Worksheets("40%")
```

```
LastRow40 = sht40.Cells(sht40.Rows.Count, "A").End(xlUp).Row
  Range("L4").Select
  Application.CutCopyMode = False
  ActiveCell.FormulaR1C1 = "=RC[-4]*60"
  Range("M4").Select
  ActiveCell.FormulaR1C1 = "=(RC[-3]/60)*4.184*(RC[-7]-RC[-8])"
  Range("N4").Select
  ActiveCell.FormulaR1C1 = "=RC[-9]-RC[-7]"
  Range("O4").Select
  ActiveCell.FormulaR1C1 = "=RC[-2]/RC[-3]"
  Selection.AutoFill Destination:=Range("O4:O" & LastRow40)
  Range("O4:O" & LastRow40).Select
  Range("N4").Select
  Selection.AutoFill Destination:=Range("N4:N" & LastRow40)
  Range("N4:N" & LastRow40).Select
  Range("M4").Select
  Selection.AutoFill Destination:=Range("M4:M" & LastRow40)
  Range("M4:M" & LastRow40).Select
  Range("L4").Select
  Selection.AutoFill Destination:=Range("L4:L" & LastRow40)
  Range("L4:L" & LastRow40).Select


    'Move 40% data to Data Page
    Range("N4").Select
```

```
    Range(Selection, Selection.End(xlToRight)).Select
    Range(Selection, Selection.End(xlDown)).Select
    Selection.Copy
    Sheets("Data").Select
    Range("K5").Select
    ActiveSheet.Paste
    Selection.PasteSpecial Paste:=xlPasteValues, Operation:=xlNone, SkipBlanks _
        :=False, Transpose:=False
    Range("K4").Select
    Application.CutCopyMode = False
    ActiveCell.FormulaR1C1 = "40%"
    Range("A1").Select

    'Calculate Data For 30%
Dim LastRow30 As Long, sht30 As Worksheet
 Sheets("Data").Select
    Range("A1:K3").Select
    Selection.Copy
    Sheets("30%").Select
    Range("A1").Select
    ActiveSheet.Paste
 Set sht30 = ThisWorkbook.Worksheets("30%")
 LastRow30 = sht30.Cells(sht30.Rows.Count, "A").End(xlUp).Row
    Range("L4").Select
```

```
Application.CutCopyMode = False
ActiveCell.FormulaR1C1 = "=RC[-4]*60"
Range("M4").Select
ActiveCell.FormulaR1C1 = "=(RC[-3]/60)*4.184*(RC[-7]-RC[-8])"
Range("N4").Select
ActiveCell.FormulaR1C1 = "=RC[-9]-RC[-7]"
Range("O4").Select
ActiveCell.FormulaR1C1 = "=RC[-2]/RC[-3]"
Selection.AutoFill Destination:=Range("O4:O" & LastRow30)
Range("O4:O" & LastRow30).Select
Range("N4").Select
Selection.AutoFill Destination:=Range("N4:N" & LastRow30)
Range("N4:N" & LastRow30).Select
Range("M4").Select
Selection.AutoFill Destination:=Range("M4:M" & LastRow30)
Range("M4:M" & LastRow30).Select
Range("L4").Select
Selection.AutoFill Destination:=Range("L4:L" & LastRow30)
Range("L4:L" & LastRow30).Select


'Move 30% data to Data Page
   Range("N4").Select
Range(Selection, Selection.End(xlToRight)).Select
Range(Selection, Selection.End(xlDown)).Select
```

```
    Selection.Copy
    Sheets("Data").Select
    Range("M5").Select
    ActiveSheet.Paste
    Selection.PasteSpecial Paste:=xlPasteValues, Operation:=xlNone, SkipBlanks _
        :=False, Transpose:=False
    Range("M4").Select
    Application.CutCopyMode = False
    ActiveCell.FormulaR1C1 = "30%"
    Range("A1").Select

        'Calculate Data For 20%
Dim LastRow20 As Long, sht20 As Worksheet
 Sheets("Data").Select
    Range("A1:K3").Select
    Selection.Copy
    Sheets("20%").Select
    Range("A1").Select
    ActiveSheet.Paste
 Set sht20 = ThisWorkbook.Worksheets("20%")
 LastRow20 = sht20.Cells(sht20.Rows.Count, "A").End(xlUp).Row
    Range("L4").Select
    Application.CutCopyMode = False
    ActiveCell.FormulaR1C1 = "=RC[-4]*60"
```

```
Range("M4").Select

ActiveCell.FormulaR1C1 = "=(RC[-3]/60)*4.184*(RC[-7]-RC[-8])"

Range("N4").Select

ActiveCell.FormulaR1C1 = "=RC[-9]-RC[-7]"

Range("O4").Select

ActiveCell.FormulaR1C1 = "=RC[-2]/RC[-3]"

Selection.AutoFill Destination:=Range("O4:O" & LastRow20)

Range("O4:O" & LastRow20).Select

Range("N4").Select

Selection.AutoFill Destination:=Range("N4:N" & LastRow20)

Range("N4:N" & LastRow20).Select

Range("M4").Select

Selection.AutoFill Destination:=Range("M4:M" & LastRow20)

Range("M4:M" & LastRow20).Select

Range("L4").Select

Selection.AutoFill Destination:=Range("L4:L" & LastRow20)

Range("L4:L" & LastRow20).Select

Application.ScreenUpdating = True


  'Move 20% data to Data Page

  Range("N4").Select

Range(Selection, Selection.End(xlToRight)).Select

Range(Selection, Selection.End(xlDown)).Select

Selection.Copy
```

```
    Sheets("Data").Select
    Range("O5").Select
    ActiveSheet.Paste
    Selection.PasteSpecial Paste:=xlPasteValues, Operation:=xlNone, SkipBlanks _
        :=False, Transpose:=False
    Range("O4").Select
    Application.CutCopyMode = False
    ActiveCell.FormulaR1C1 = "20%"
    Range("A1").Select


        'Calculate Data For 10%
Dim LastRow10 As Long, sht10 As Worksheet
 Sheets("Data").Select
    Range("A1:K3").Select
    Selection.Copy
    Sheets("10%").Select
    Range("A1").Select
    ActiveSheet.Paste
 Set sht10 = ThisWorkbook.Worksheets("10%")
 LastRow10 = sht10.Cells(sht10.Rows.Count, "A").End(xlUp).Row
    Range("L4").Select
    Application.CutCopyMode = False
    ActiveCell.FormulaR1C1 = "=RC[-4]*60"
    Range("M4").Select
```

```
ActiveCell.FormulaR1C1 = "=(RC[-3]/60)*4.184*(RC[-7]-RC[-8])"
Range("N4").Select
ActiveCell.FormulaR1C1 = "=RC[-9]-RC[-7]"
Range("O4").Select
ActiveCell.FormulaR1C1 = "=RC[-2]/RC[-3]"
Selection.AutoFill Destination:=Range("O4:O" & LastRow10)
Range("O4:O" & LastRow10).Select
Range("N4").Select
Selection.AutoFill Destination:=Range("N4:N" & LastRow10)
Range("N4:N" & LastRow10).Select
Range("M4").Select
Selection.AutoFill Destination:=Range("M4:M" & LastRow10)
Range("M4:M" & LastRow10).Select
Range("L4").Select
Selection.AutoFill Destination:=Range("L4:L" & LastRow10)
Range("L4:L" & LastRow10).Select


    'Move 10% data to Data Page
  Range("N4").Select
Range(Selection, Selection.End(xlToRight)).Select
Range(Selection, Selection.End(xlDown)).Select
Selection.Copy
Sheets("Data").Select
Range("Q5").Select
```

```
ActiveSheet.Paste
Selection.PasteSpecial Paste:=xlPasteValues, Operation:=xlNone, SkipBlanks _
    :=False, Transpose:=False
Range("Q4").Select
Application.CutCopyMode = False
ActiveCell.FormulaR1C1 = "10%"
Range("A1").Select
Sheets("Data").Select
Application.ScreenUpdating = True
```

A.4 VBA macro results

The following figures gives the figures used to generate the linear regression lines at each of the relative humidity values as defined within the study.

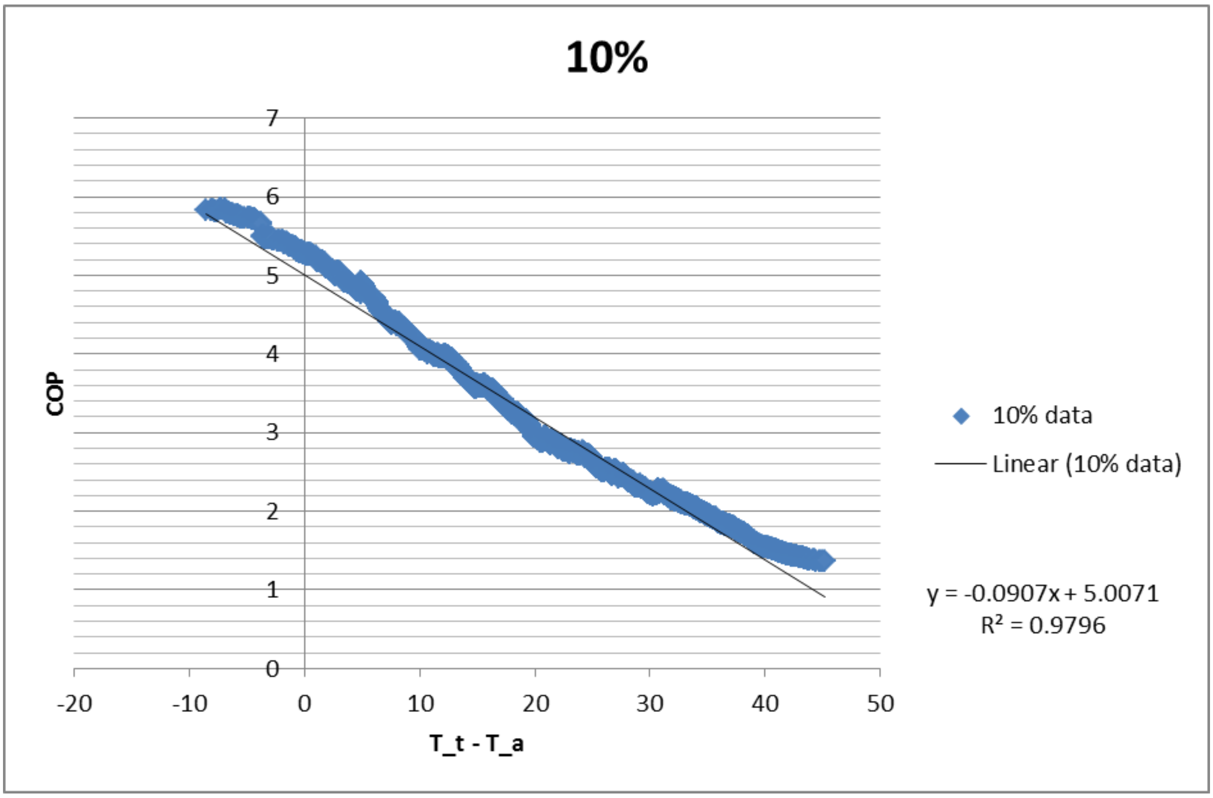

Figure A2: 10% Performance regression line

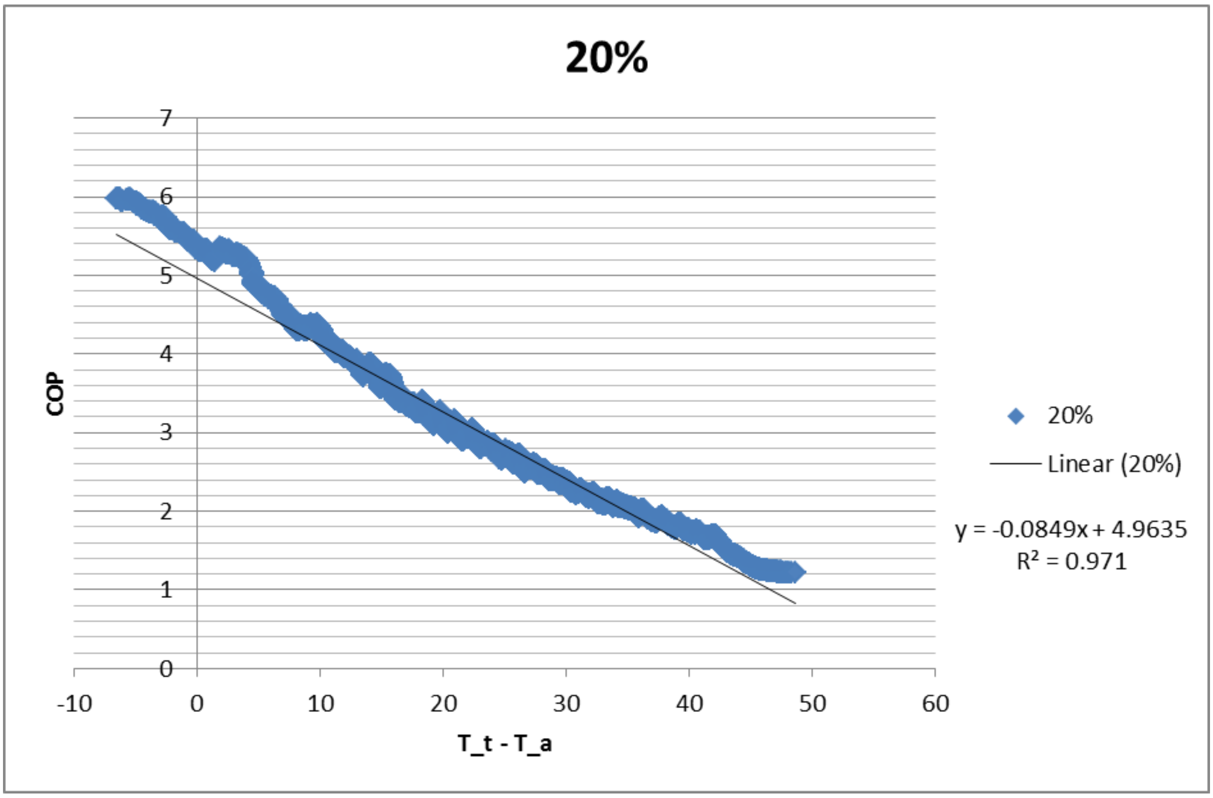

Figure A3: 20% Performance regression line

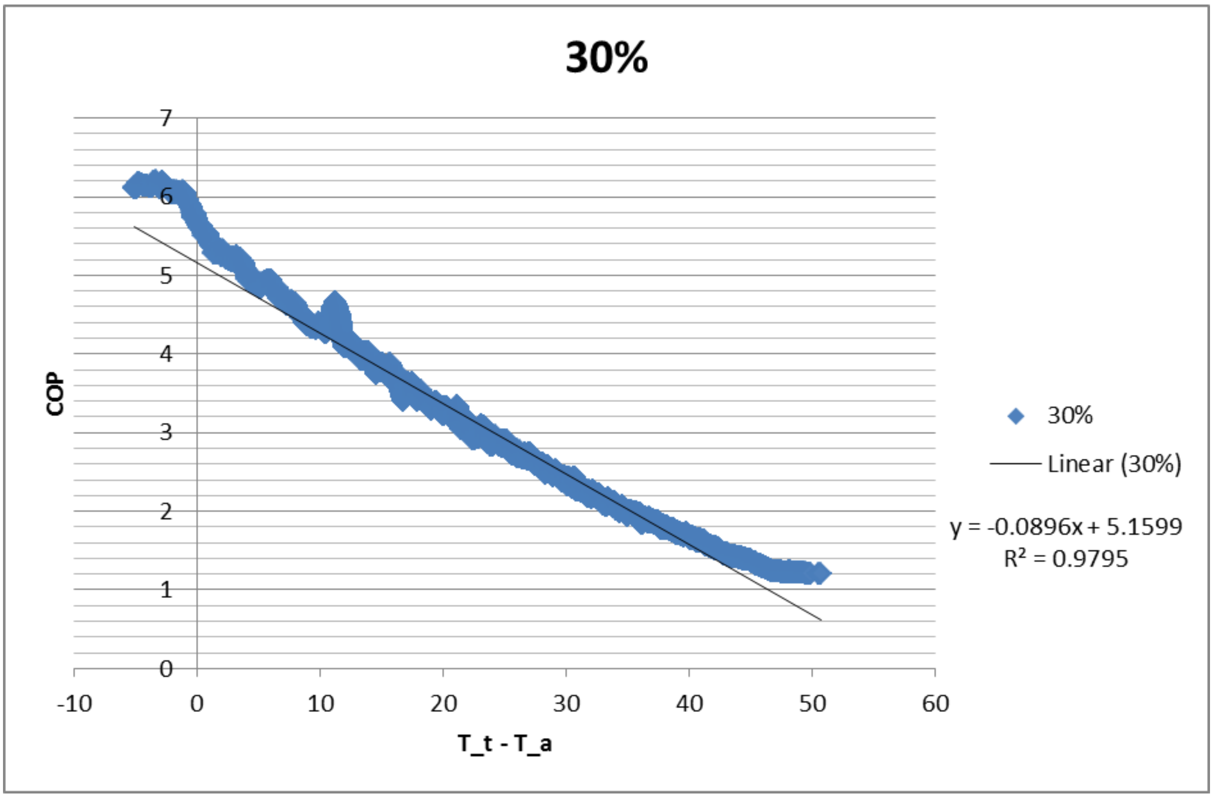

Figure A4: 30% Performance regression line

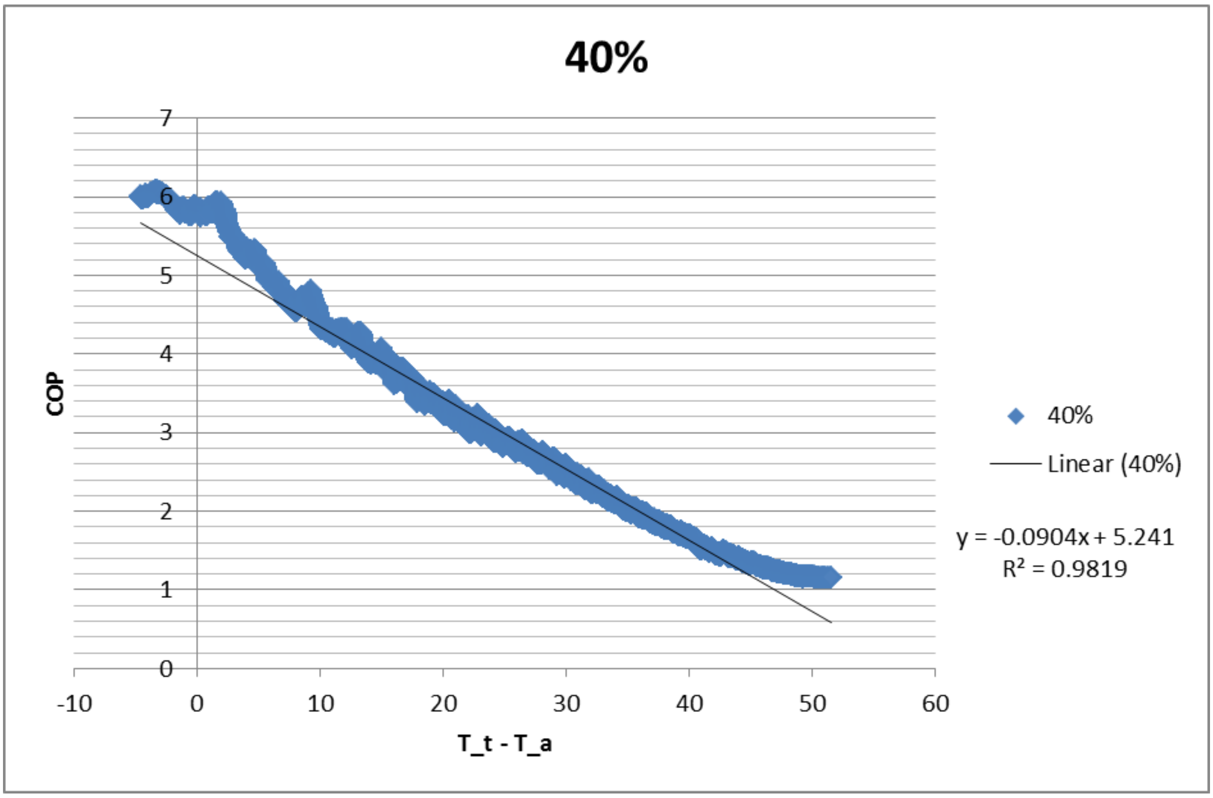

Figure A5: 40% Performance regression line

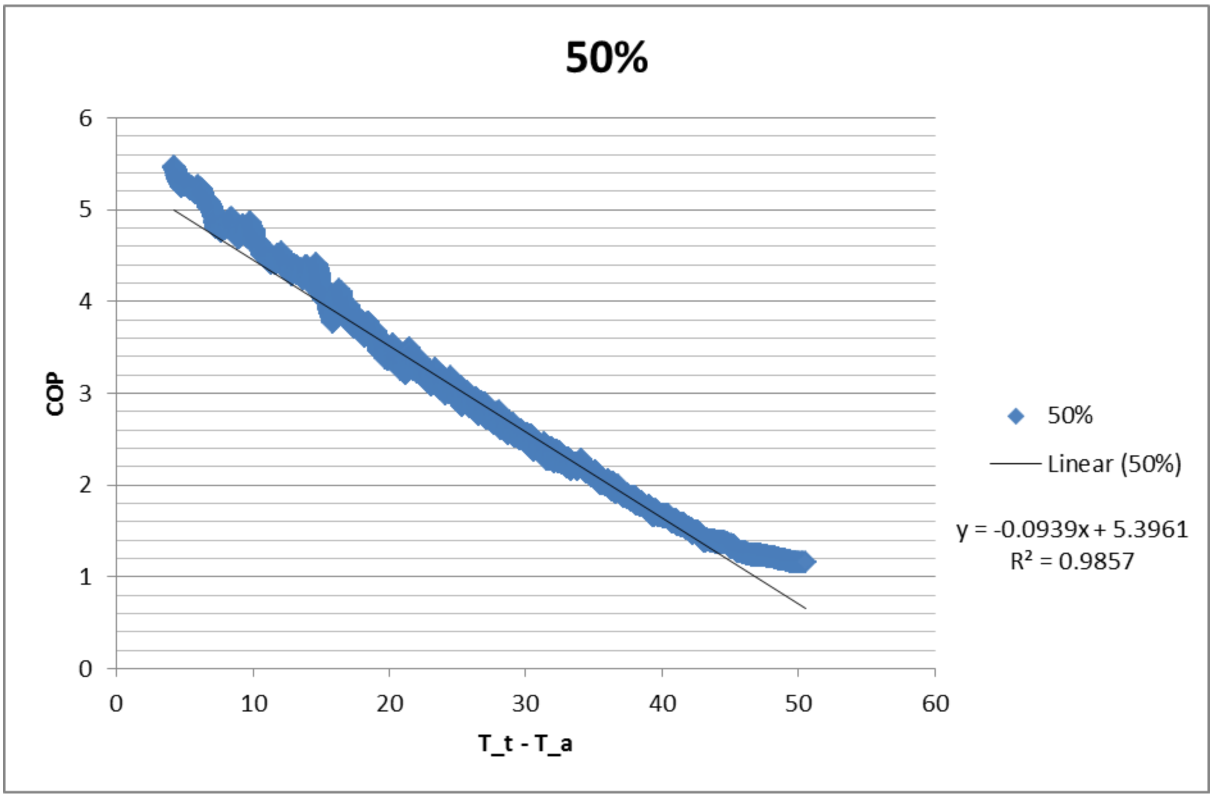

Figure A6: 50% Performance regression line

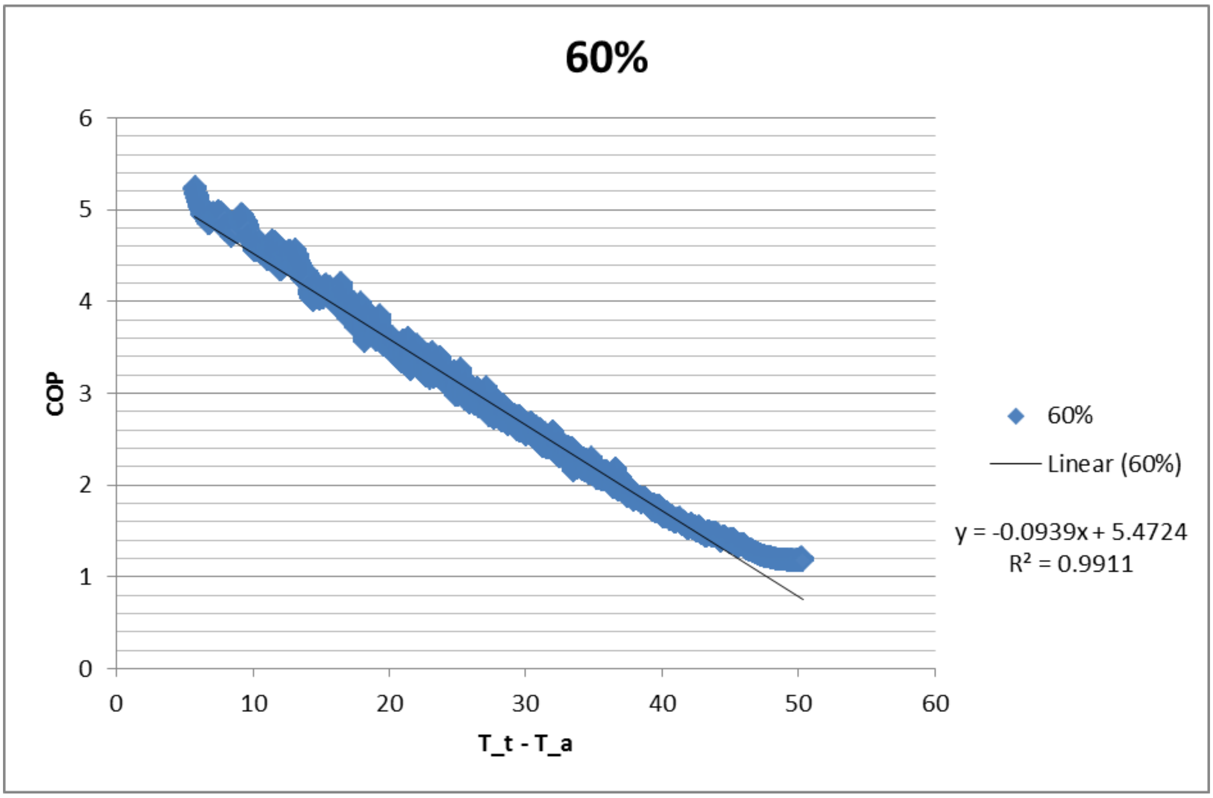

Figure A7: 60% Performance regression line

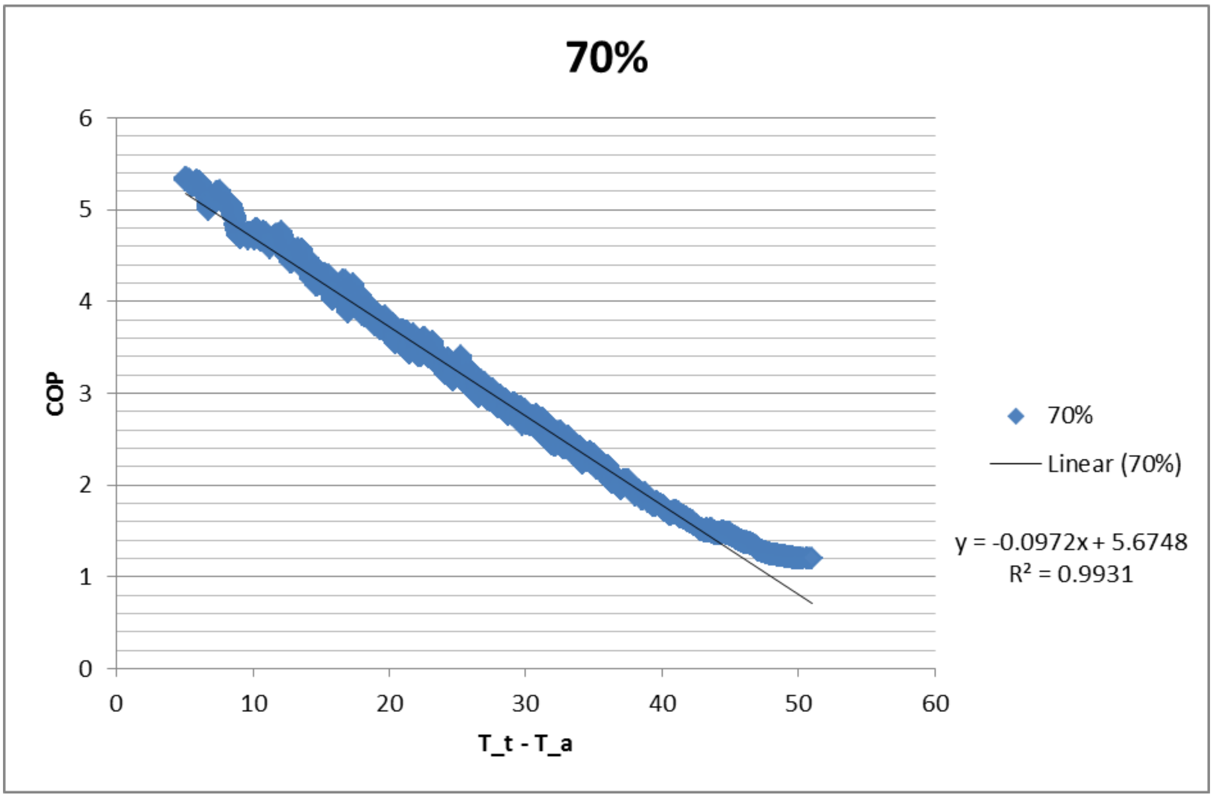

Figure A8: 70% Performance regression line

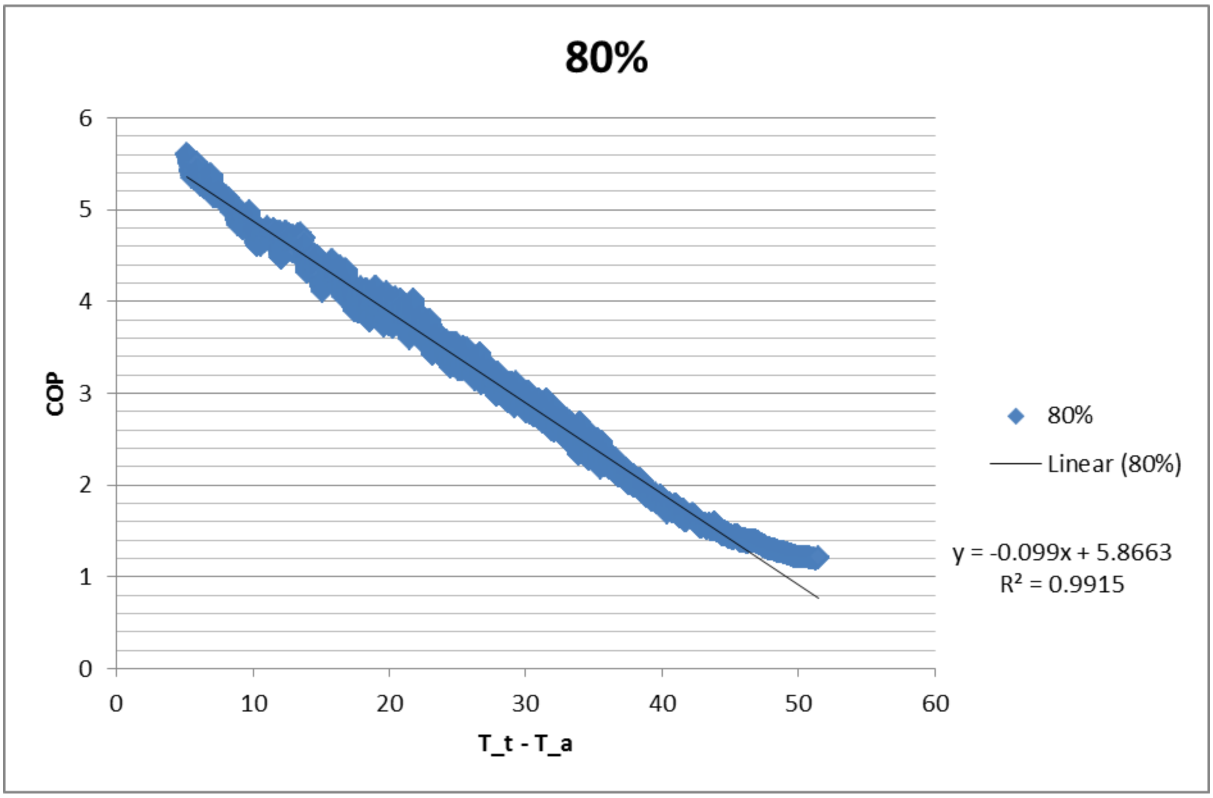

Figure A9: 80% Performance regression line

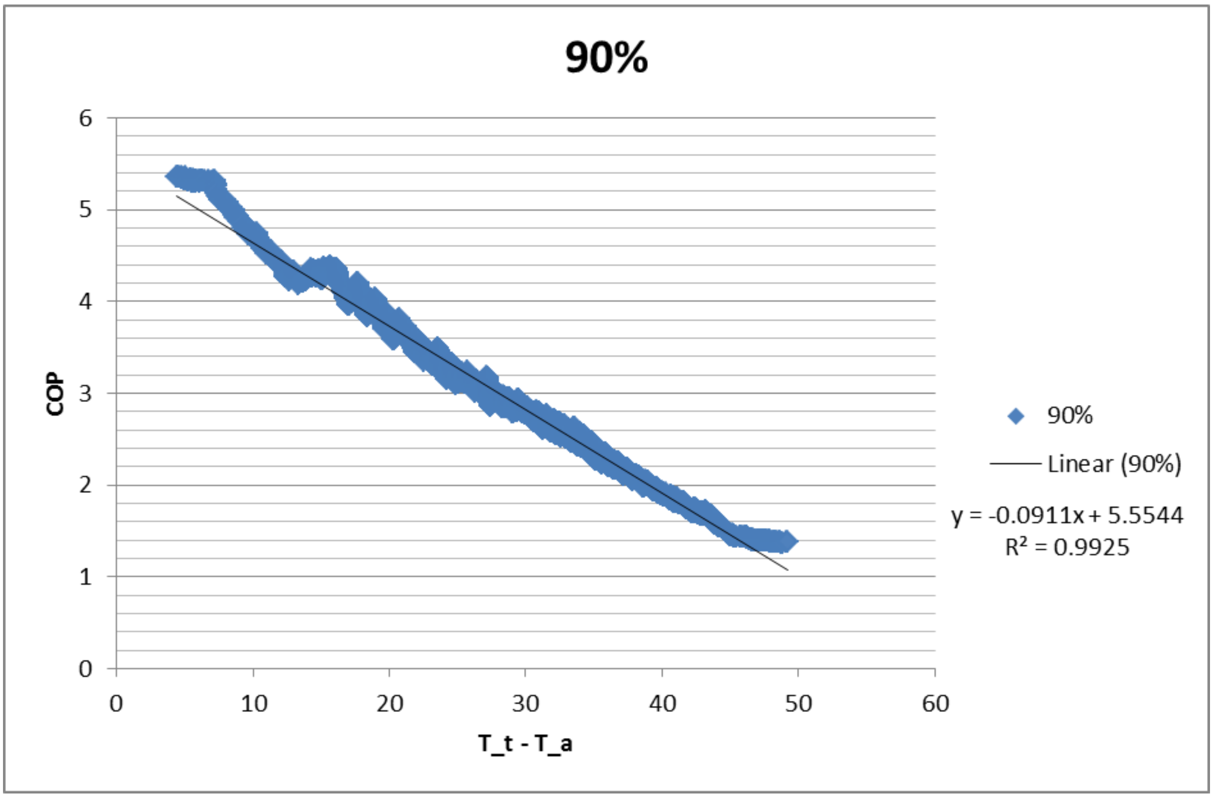

Figure A10: Performance regression line

Annexure B: Laboratory test

B.1 Example of raw data

Table B1 below shows a section of the laboratory test's recorded data.

Table B1: Laboratory test results example

	HP OUT TEMP	Tank Temp	HP Flow	Power usage	Ambient
Time	°C	°C	l/min	W	°C
10:25:00 AM	26.9	26.0	0.080313	34.8646	42
10:25:01 AM	27.0	26.1	0.057843	34.8646	42
10:25:02 AM	27.0	26.1	-0.00586	34.8646	42
10:25:03 AM	27.1	26.1	2.73819	105.588	42
10:25:04 AM	26.9	26.0	11.8491	105.588	42
10:25:05 AM	26.4	25.5	12.2202	105.588	42
10:25:06 AM	25.9	25.3	12.2453	71.9999	42
10:25:07 AM	25.6	25.3	12.3035	62.9844	42
10:25:08 AM	25.5	25.4	12.367	62.9844	42
10:25:09 AM	25.5	25.5	12.3759	62.9844	42
10:25:10 AM	25.6	25.6	12.4159	92.1276	42
10:25:11 AM	25.6	25.6	12.4231	92.1276	42
10:25:12 AM	25.6	25.5	12.4143	93.3432	42
10:25:13 AM	25.6	25.5	12.384	93.3432	42
10:25:14 AM	25.6	25.4	12.3876	93.3432	42
10:25:15 AM	25.5	25.4	12.41	93.6136	42
10:25:16 AM	25.5	25.4	12.4225	93.6136	42
10:25:17 AM	25.5	25.4	12.4341	93.6136	42
10:25:18 AM	25.5	25.4	12.4401	93.8835	42
10:25:19 AM	25.5	25.4	12.4205	93.8835	42
10:25:20 AM	25.5	25.4	12.4242	93.8835	42
10:25:21 AM	25.5	25.4	12.4286	93.9629	42
10:25:22 AM	25.5	25.4	12.407	93.9629	42
10:25:23 AM	25.5	25.4	12.4152	93.9629	42
10:25:24 AM	25.5	25.4	12.4193	94.054	42
10:25:25 AM	25.6	25.4	12.4184	94.054	42
10:25:26 AM	26.0	25.4	12.405	94.054	42
10:25:27 AM	26.6	25.4	12.4056	94.0864	42
10:25:28 AM	27.2	25.4	12.4205	94.0864	42
10:25:29 AM	27.8	25.4	12.4275	94.0864	42
10:25:30 AM	28.3	25.4	12.42	93.9717	42

B.2 Laboratory test results

The following figures gives the figures used to generate the linear regression lines at each of the temperatures recorded within the laboratory tests.

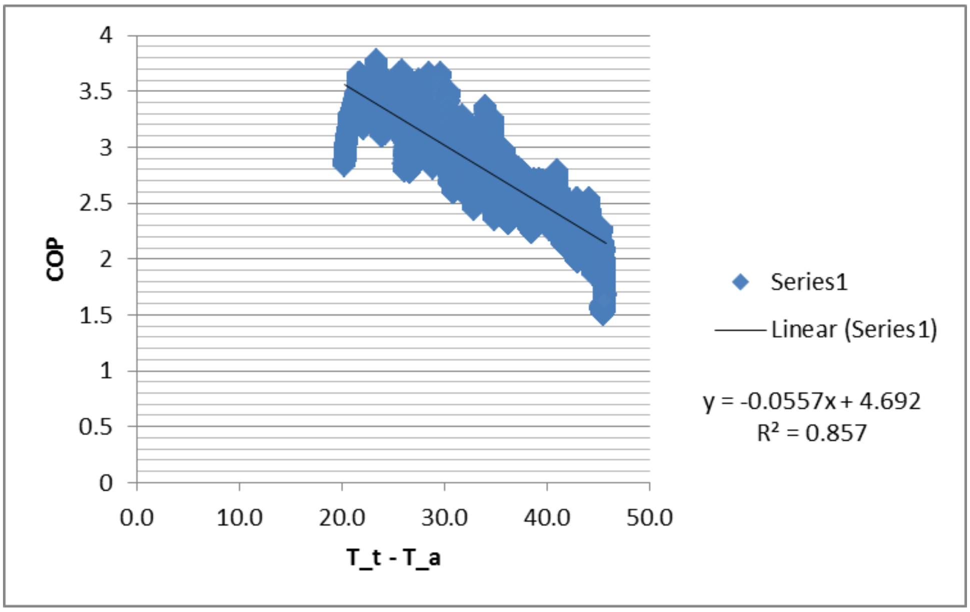

Figure B1: 5°C Laboratory test result

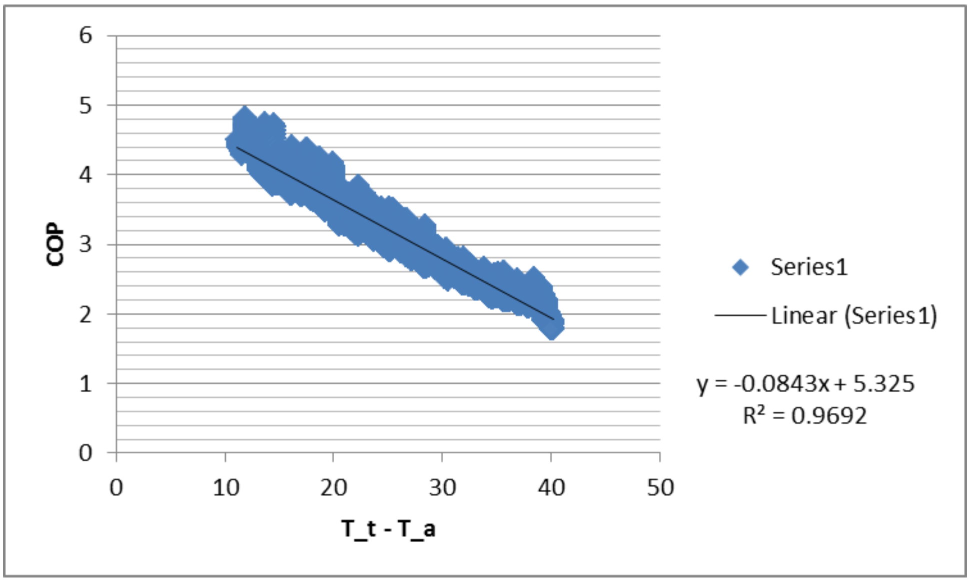

Figure B2: 15°C Laboratory test result

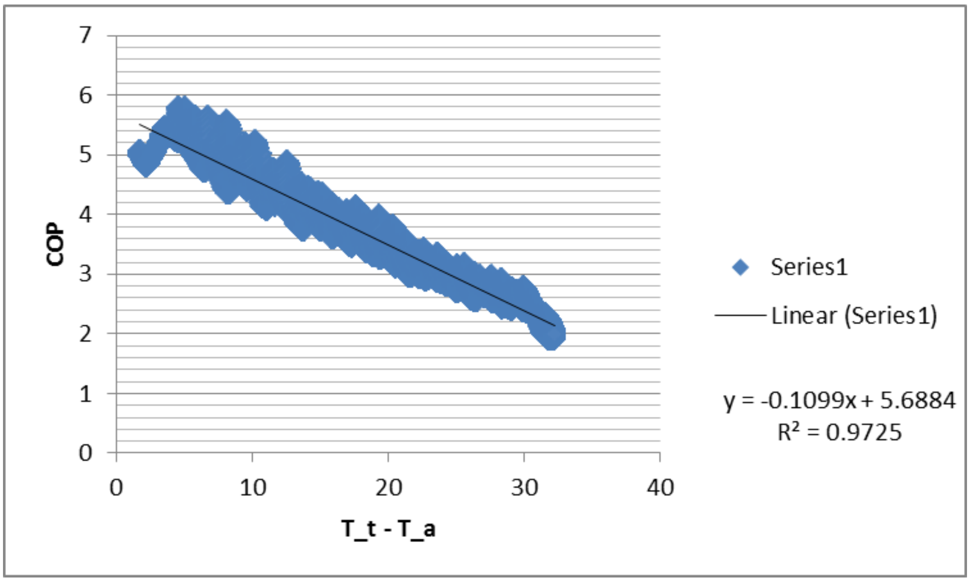

Figure B3: 25°C Laboratory test result

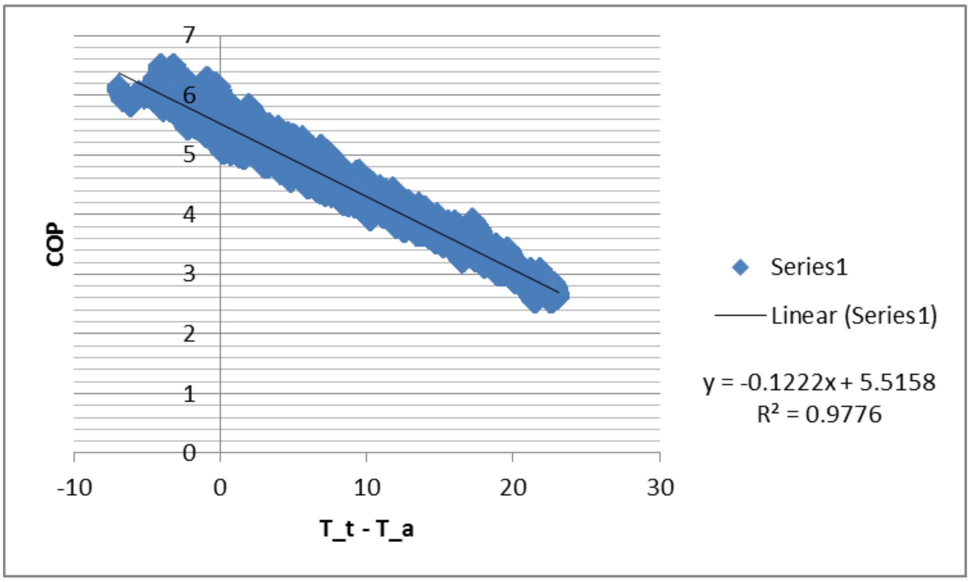

Figure B4: 35°C Laboratory test result

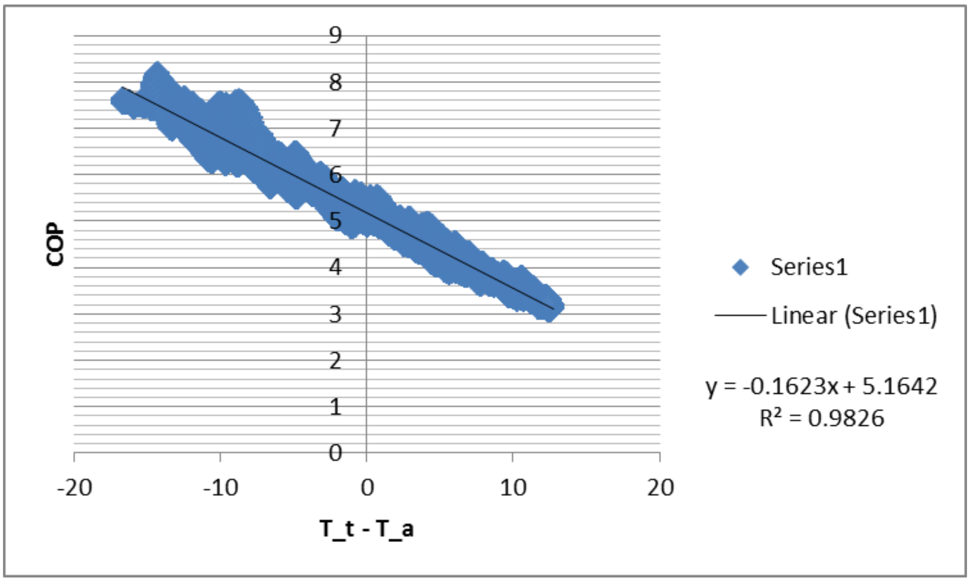

Figure B5: 42°C Laboratory test result

Annexure C: Simulation model

C.1 Example of component input tab

The following figure gives an example of the input tab found for each component listed within Flownex® SE's component library. This input tab requires all specifications to fully define the component.

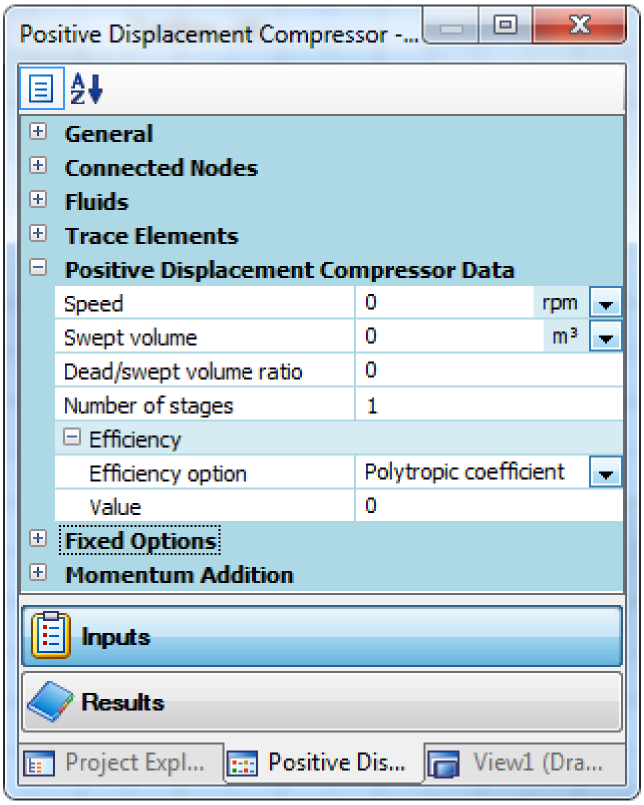

Figure C1: Component input tab for Flownex® SE