Demonstration of residential air source heat pump water heaters performance in South Africa: Systems monitoring and modelling

By

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DECLARATION

I, Stephen L. Tangwe declare that the thesis for the award of a Doctor of Philosophy Degree by existing published or creative works at the University of Sunderland, Faculty of Engineering and Advanced Manufacturing, hereby submitted by me, has not been previously submitted for a degree at this or any other university and that it is my original work in design and execution. Above all, the co-authors in the published works that are consolidated to form this thesis were my promoters and did not provide any technical intellectual contributions. Furthermore, all the reference materials contained therein have been duly acknowledged.

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Stephen Loh Tangwe

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Dr David Baglee

(Director of studies)

....

Prof Alan Wheatley

(Co-Supervisor)

DEDICATION

This work is dedicated to my:

Beloved wife, Dr Mrs Manyi-Loh, C.E

Lovely and wonderful kids, Loh Alexia-Augusta, Loh Gad-Manuel, Loh Faith-Simon and Loh Lois-Caleb

Precious mother, Mrs. Martha Achale and late father, Mr. Peter Mbeb Tangwe

ACKNOWLEDGEMENTS

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Nomenclature

AC	Alternating current
ANOVA	Analysis of variance
ASHP	Air source heat pump
c	Specific heat capacity of water
COP	Coefficient of performance
COP_{cal}	Calcaluted Coefficient of performance
COP_{mod}	Modelled Coefficient of performance
DAS	Data acquisition system
DC	Direct current
DSM	Demand Side Management
E	Electrical energy
Eg	Electrical energy consumed by geyser
Es	Electrical energy consumed by heating Systems
EF	Electrical energy factor
Esco	Energy service company
Eskom	South African electricity supply utility
f(n)	User defined function

GSHP	Geothermal source heat pump
GWh	Giga Watt hour
IDM	Integrated Demand Management
kg	Kilogramme
kVA	Kilo Volt ampere kVAR
	Reactive kilo volt ampere kWh
	Kilo Watt hour
L	Liters
LF	Load factor
m	Mass in kg
MW	Mega Watt
p	Pressure P
	Power
Ps	Power consumed by systems
PF	Power factor
Q	Thermal energy
RH	Relative humidity
S	Enthropy
SPP	Simple payback period
t	Time taken
T_a	Ambient temperature
T_{cm}	Difference in refrigerant temperature between the compressor outlet and inlet
T_{cmi}	Refrigerant temperature at compressor inlet
T_{cmo}	Refrigerant temperature at compressor outlet
T_{cn}	Difference in refrigerant temperature between the condenser inlet and outlet
T_{cni}	Refrigerant temperature at condenser inlet
T_{cno}	Refrigerant temperature at condenser outlet
T_{in}	Split type ASHP inlet average water Temperature

T_{out}	Split type ASHP outlet average water Temperature
T_s-T_a	Difference in hot water set point temperature and the ambient temperature
V	Volume
VCRC	Vapour compression refrigeration cycle
V_d	Volume of hot water drawn off - from geyser or ASHP

Symbol

Greek symbol	Full name	Representation
□	Beta	Scaling notation for the multiple linear regression in Chapter Seven
□	Kappa	Scaling notation for the surface fitting regression in Chapter Eight
□	Gamma	Scaling notation for the multi variant regression in Chapter Nine
□	Lambda	Product of ambient temperature and relative humidity
□	Delta	Difference
□	Sigma	Summation
□		Change in enthropy
□Q		Work done due to gained of thermal energy

□W□ Carnot's efficiency

□carnot

GENERAL ABSTRACT

The purpose of the research was to evaluate the coefficient of performance (COP) of both 150 L split and integrated type air source heat pump (ASHP) water heaters via experimental analysis, statistical tests and mathematical modelling. The ASHP water heaters are used as a potential replacement of inefficient geyser for the production of sanitary hot water due to the excellent efficiency of COP ranging between 2 and 4 and also the capability of reducing the electrical energy consumption by 50-70%. Both types of ASHP water heaters together with a 150 L geyser that served as the control experiment were set up such that distinctive real-time simulated volumes of hot water (100, 50 and 150 L) were drawn off from each of the storage tanks per day over a full year. A data acquisition system (DAS) was designed and built comprising of power meters, flow meter, temperature sensors, ambient temperature and relative humidity sensors in order to monitor the electrical, thermo-physical and environmental contributions of the various hot water heating devices. The hot water set point temperature on each of the technologies was 55^{o}C and the volume drawn off corresponded to the demand during the morning, afternoon and evening, respectively. This mimic the profile of a typical middle or highincome family (3-4 adults) in South Africa. The results depicted that the average annual COP, load factor, and energy saving of the split and integrated type systems was 2.95 and 2.45; 10.2 and 16.7% and 2.770 and 2.499 MWh while the simple payback period was 3.9 and 5.2 years, respectively. The reliefF test revealed that the predictors (ambient temperature and relative humidity) were secondary factors while the electrical energy consumed, the difference in the temperature of the refrigerant at the inlet and outlet of the compressor and condenser were the primary factors to the COP. The derived multiple linear regression models exhibited an excellent determination coefficient of over 90% between the calculated and modelled COP of both types of ASHP water heaters. Finally, the 2D multi-contour plots simulation was accurately used to show the variation of each predictors to the COP. Also, a simulation application to simultaneously compare the COP of both types of ASHP water heaters was developed in the Simulink environment utilising the derived mathematical models. Heat pump manufacturers and energy service companies can employ both the 2D multi-contour plots simulation and the simulation application to show the variation of the specific predictors with the COP and to predict the COP of both types of ASHP water heaters. Conclusively, the research provides substantial evidence for both policy makers and home owners to justify the techno-economic and social benefits of retrofitting a geyser with an ASHP water heater.

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Chapter One

Introduction

1.1 Background of the study

In South Africa, electricity generation by the electricity supply utility (Eskom) is mainly from coal thermal power plant. Sanitary hot water production in the residential sector constitutes 40-60% of the average monthly electrical energy consumption and is achieved by the use of inefficient geysers (Meyer and Tshimankinda, 1998). It is worth mentioning that the geysers are among the residential energy consuming utilities responsible for peak demand and daunting energy consumption, which is forcing the national grid to experience constraint (Eskom, 2010). As a consequence, air source heat pump (ASHP) is being used as a replacement for the geyser, serving as a potential solution for both demand and energy consumption due to its excellent efficiency and coefficient of performance (COP) of range 2 to 4 (Bodzin, 1997; Levins, 1982; Tangwe *et al*., 2014).

The COP of ASHP water heater is defined as the ratio of the useful output thermal energy gained by stored water to the input electrical energy consumed during the vapour compression refrigeration cycle (Sinha and Dysarka, 2008). Eskom targeted rolling out 65,580 residential ASHP systems by 2013 in a bid to achieve an evening peak demand reduction of 54 MW and an annual energy saving of 80.86 GWh (Eskom, 2011). Furthermore, in order to justify the anticipated demand and energy saving through the retrofitting of geysers with ASHP systems, there is a need to experimentally determine the COP as well as mathematically model the dynamic performance of the ASHP water heaters.

This study focused on addressing these underline goals through experimentation, mathematical modelling and simulation.

1.2 Rationale behind this study

A considerable section of the residential sector in both developing and developed countries is utilising electrical energy for the production of sanitary hot water (hot water at the set point temperature greater than or equal to 55^{o}C) from geysers. It eventually results in the consumption of an enormous quantity of electrical power and energy, which calls for an integrated demand management (IDM), environmental, economic and social concerns. Precisely, in South Africa, geysers are strategically controlled through the demand side management (DSM) under the residential load management (RLM) programme, which results only in load reduction at specific times of use period where load shifting occurred (Rankin and Rousseau, 2008). This initiative is often termed energy neutral intervention since the total daily energy consumption remains unchanged.

Furthermore, the RLM programme only provides a temporary solution wherein shifting loads out of the peak hours by switching off geysers during this peak period and allowing them to come on during the off-peak hours. Without the loss of generality, sanitary hot water production from geysers is associated with one of the electrical energy utility responsible for significant constraints on the national grids of the South Africa electricity supply utility (Eskom). Hence, resulting in increases in the global warming and ozone layer depletion potential (Tangwe *et al*., 2015). Interestingly, a permanent solution to both demand and energy consumption reduction could be achieved via the retrofitting of installed electric geysers with residential ASHP (air source heat pump) units. The configuration of the ASHP water heater can either be presented with the split type ASHP unit or an integrated type ASHP water heater for new installation without existing geyser.

The residential ASHP water heater is a mature technology and an efficient and renewable energy device for the production of sanitary hot water (Morrison *et al*., 2004). The unique characteristic associated with the excellent performance of heat pump water heaters is known as the coefficient of performance (COP) (De Swardt and Meyer, 2001). A better and concrete definition of the COP of an ASHP water heater involves, the useful thermal energy gained and the input electrical energy to operate the vapour compression refrigeration cycles (VCRC); as described by Ashdown (2004) and Sinha and Dysarka, (2008). Also, the COP of the ASHP water heater is in the range of 2 to 4 (Levins, 1982; Bodzin, 1997) and can be modelled and simulated with some degree of confidence using the TRNSYS software (KLEIN-TRNSYS, 1990). Moreover, the COP of the split type residential ASHP water heater during the first-hour heating rating can be determined from experimental data-driven simulation model (Tangwe *et al*., 2014).

In addition, the techno-economic analysis of this technology in the residential sector also justifies the potential viability for the mass roll-out of ASHP water heaters in South Africa (Tangwe *et al*., 2014). The multi-purpose benefits of installing ASHP water heater and the complexity of the modelling and simulation of the COP, even with the powerful TRNSYS software offered further sufficient reasons that necessitated an elaborate in-depth research in the field of ASHP refrigeration technology.

The study involved the quantitative analyses to ascertain potential viabilities of both split and integrated type ASHP water heaters over geysers. It also dealt with the comparison of the COP of both types of ASHP water heaters based on the employment of statistical tests, the development of multiple regression models and the use of simulation application. It could be articulated that the COP of the split type ASHP water heater was better than that of the integrated type based on critical assumption that the former closed loop circuit design was more efficient and also its refrigerant exhibited a higher heat transfer coefficient than the latter.

1.3 Problem Statement

Fossil fuels, e.g. coal, oil and natural gas are conventional sources of energy that provided electricity for developing and maintaining the technologically advanced modern world. Fossil resources are finite, and their recovery and use appreciably impact our environment and affect the global climate. Shortening of oil and gas are predicted to occur within our lifetimes or those of our children (Nasi *et al*., 2008). Also, in the residential sector, sanitary hot water production devices are one of the intense electrical energy-consuming utilities and account for the daunting cost of energy consumption and the high level of greenhouse gas (carbon dioxide) emission to the environment (Lemmon *et al*., 2002). Although, the ASHP water heater is an energy efficient device whose efficiency can be enhanced by proper installation, the COP is dynamic and is governed by the ambient conditions, the system design and volume of hot water drawn off (Douglas, 2008; Baxter *et al*., 2005). Nevertheless, there is no accredited standard as well as an international performance measurement and verification protocol guidelines to determine the COP of either the integrated or split type ASHP water heaters (Ye and Zhang, 2012).

In South Africa, Eskom embarked in a mass rollout of 65,580 residential ASHP units to retrofit existing geysers with a goal of achieving a demand reduction of 54 MW and an annual energy saving of 80.6 GWh during the Eskom evening peak (18:00-20:00) (Eskom, 2011). The Eskom's residential ASHP water heater simulation application employed to compute the performance of the ASHP water heaters was subjected to significant limitation. As a consequence, the COP prediction was below 70% confidence level and was ascribed to the accuracy of the type and class of power and energy meter used for the collection of energy consumption data. Also, the ambient temperature and relative humidity data that were obtained from the meteorological weather station of the major cities that were considered due to their close proximity to the location of the installed ASHP water heaters (Eskom, 2011). These were possible because of the lack of involvement of experts on heat pump technologies in the initial contracted agreement between Eskom and service provider of the designed simulation application. Notwithstanding, qualitative studies have demonstrated that the integrated type ASHP water heater performed better than split type ASHP water heater irrespective of the heat pump configurations but provided both are of the same tank size (Marrison *el at*., 2004; Ye and Zhang, 2012).

Against this background, a core challenge in this area of research is to size the ASHP unit correctly with a storage tank capacity based on the volume of hot water (on an average daily hot water drawn off and the average daily hot water usage profiles) and power consumed. Also, there is no rigorous research conducted that quantitatively measured and modelled the COP of integrated and split type ASHP water heaters with the underlining emphasis on performing a comparative analytical study (Tangwe *et al*., 2018).

Therefore, in this research, a great depth of comparative analysis based on the quantitative determination of the COP with the aid of developing and building mathematical models was conducted for a 150 L split type ASHP water heater (without an electric backup element) and a 150 L integrated type ASHP water heater (with an electric backup element) under the different volume of hot water drawn off scenarios. The comparison was focused on the Eskom's ASHP water heaters categorisation that was based on the volume of the tank and input power range (Eskom, 2011).

1.4 Research questions

The research sought to answer the following questions:

i. Can a reliable and accurate data acquisition system be designed and built to monitor the performance of both the geyser and the ASHP water heaters?
ii. Can the retrofitting of geysers with ASHP units provide permanent load and energy consumption reduction?
iii. Can the ASHP water heaters be considered as a potentially viable investment option in the domain of sanitary hot water production? iv. Can the impact generated by the installation of the isotherm blanket on the

hot water storage tanks for sanitary hot water production devices be quantitatively measured?

v. Can the electrical, thermo-physical properties of the refrigerants, and the ambient condition parameters be used as diagnostic predictors to compare the performance of split and integrated type ASHP water heaters?

vi. Can the predictors of an integrated type ASHP water heater with an electric backup and a split type ASHP water heater without an electric backup be ranked according to the weight of contribution to their COP based on real-time controlled volume of hot water drawn off under varying ambient conditions?

vii. Can the coefficient of performance of the split and integrated type ASHP water heaters be quantitatively measured during VCRC?

viii. Can simple but reliable mathematical models be developed and built to predict the COP of the residential split and integrated type ASHP water heaters?

ix. Can a 2D multi contour plots simulation be utilised in showing the variation of each of the predictors to the COP for both types of ASHP water heaters while the others are held constant?

x. Can a simulation application be designed on the Simulink of MATLAB to forecast the COP of both types of ASHP water heaters?

1.5 Research aims

The overall aims of the research were to conduct a comparative and a quantitative analysis of the COP of a 150 L integrated type ASHP water heater with an electric backup and a 150 L split type ASHP water heater without an electric backup as well as to develop mathematical models together with simulation application that can be used to predict the performance of the systems.

1.6 Objectives of the study

To accomplish the overall aims, the following specific objectives were outlined:

i. To design and build a data acquisition system (DAS) that guarantees a better recording and storing of the measurement data that were collected and further used in the research analysis.

ii. To determine the power and energy consumption of the ASHP water heaters and intended geyser proposed to be retrofitted.

iii. To conduct a techno-economic analysis of both types of ASHP water heaters whereby the life cycle cost analysis was used to justify the potential viability of the ASHP water waters.

iv. To analytically evaluate standby thermal energy losses of the geyser and the ASHP water heaters and the impact upon installing isotherm blankets on the hot water cylinders.

v. To use critical predictors such as electrical, thermo-physical properties of the refrigerants, and ambient condition parameters to compare the performance of split and integrated type ASHP water heaters?

vi. To conduct a statistical test which enabled the ranking of the specific predictors by virtue of their importance to the contribution in the desired output (i.e. ReliefF algorithm).

vii. To perform a multiple comparison test to verify if any significant difference occurred in the group COP means of the ASHP water heaters under the different scenarios of controlled volume of hot water drawn off.

viii. To develop and build multiple linear regression models using the critical thermodynamic, electrical and ambient weather parameters to predict the COP of the ASHP water heaters.

ix. To use the two-dimensional, multi contour plots simulation to show the variation of each predictor with the COP of the two types of ASHP water heaters.

x. To design an architectural algorithm of a simulation application of both types of ASHP water heaters from the Simulink environment of MATLAB using the derived mathematical models.

1.7 Limitations

i. The research was conducted in one location which typically represented the ambient condition of one geographical region in South Africa due to the huge capital cost and cost involved in deploying the systems in multiple regions.

ii. The practical challenge encountered by running all the three systems under same scenarios in an actual home with occupants also forced the experiment to be conducted based on real-time simulated controlled volume of hot water drawn off but using an outdoor testing facility.

1.8 Delineations

i. The research focused on the simulated controlled volume of hot water draws which mimic the typical residential hot water profile for a middle or high-income family.

ii. The COP of the ASHP water heaters was determined from the experimental data obtained as well as the developed and built mathematical models and simulation application.

1.9 Assumptions

1.9.1 The temperature measurements on precise pipeline locations were equal to the primary (refrigerant) and secondary (water) fluid temperatures of the hot water heating devices. This assumption was supported by the following;

i. The pipes were made of copper, and at thermal equilibrium, the temperature of the installed temperature sensor in the pipe corresponded to the temperature of the fluid at that location.

ii. The temperature sensors were well insulated to ensure that only the temperature of the fluid (refrigerant or water) was sensed and recorded.

iii. The temperature sensors were incorporated with electronic input pulse adapters that converted analogue signals to digital and prevented errors due to noise interference.

iv. The uncertainty in the temperature measurements was negligible because of the accuracy of the temperature sensor and its response time.

1.9.2 The uncertainty of the recorded measurements obtained from the power meter, flow meter and ambient temperature and relative humidity sensors did not influence the actual measurements due to the high accuracy and the minuscule response time of the transducers and sensors. Also, electronic input pulse adapters were installed on the transducers and sensors cables which converted the analogue to digital signals. Hence, eliminated the errors from noise interference.

1.9.3 The uncertainty in the calculated COP of the residential ASHP water heaters was negligible and wouldn't affect the COP calculations as the uncertainty of its drivers (predictors) were also insignificant.

1.10 Hypotheses

i. The coefficient of performance of the residential split and integrated type ASHP water heaters can reliably be modelled with over 90% accuracy via the use of multiple linear regression models which harbour the following as predictors; change in the outlet and inlet refrigerant temperatures at the compressor and condenser, electrical energy consumed, ambient temperature and relative humidity.

ii. The two-dimensional, multi contour plots simulation employing the derived mathematical models can be used to predict the coefficient of performance of the ASHP water heaters with a 95% confidence bounds under the variation of any specific predictor while the others are held constant.

1.11 Chapter overview

This thesis comprises of ten chapters as follows;

Chapter one introduces the general overview of the topic of the thesis with primary emphasis on the rationale, problem statement, research questions, objectives and hypotheses.

Chapter two assembles information on the fundamental principles and the various heat pump technologies involve in hot water heating. In addition, a concise literature review was presented on the ASHP water heaters. Chapter three covers an overall research methodology, followed by an experimental set up of the installed hot water heating technologies as well as the data acquisition system used to monitor the performance of the various hot water heating devices. Also, an elaborate description of the design and construction of the data acquisition system was presented and detail configuration of the sensors and data loggers were also discussed. This is an in-depth chapter from published work (both co-authors were my promoters as presented in authorship letter in appendix III):

i. Tangwe, S.L., Simon, M. and Meyer, E.L., 2016. Design of a heat pump water heater performance monitoring system: To determine performance of a split type system. *Journal of Engineering, Design and Technology*, 14 (4), pp. 739-751.

Chapter four encompasses a fundamental methodology to quantitatively and qualitatively determine the benefits of using either an integrated or split type residential ASHP water heater over geyser for sanitary hot water production. It equally harbours information on the elucidation of the demand reduction and energy savings achieved from the implementation of both the residential split and integrated type ASHP water heaters. A conservative approach was implemented to determine the annual tonnage of carbon dioxide emission reduction, the volume of water saved and the payback period based on the retrofit or replacement of existing geyser with ASHP water heater.

This is a consolidated chapter from published works (both co-authors were my promoters as presented in authorship letter in appendix III):

i. Tangwe, S., Simon, M. and Meyer, E. 2014. A techno-economic viability of a residential air source heat pump water heater: Fort Beaufort, South

Africa. *International Journal of Engineering Science and Research Technology*, 3(10), pp 504-510.

ii. Tangwe, S., Simon, M. and Meyer, E., 2015, Quantifying residential hot water production savings by retrofitting geysers with air source heat pumps. 23rd *International Conference on the Domestic Use of Energy* (DUE), 2015. Pp. 235-241. Publisher: IEEE, IEEE Xplore Journal, ISSN: 978-0-9922-0419-8

iii. Tangwe, S, Michael Simon and Edson Meyer, 2017. Residential air source heat pump water heaters as renewable and energy efficient systems. 25th Southern African Universities Power Engineering Conference, University of Stellenbosch, South Africa. 30th Jan-01 Feb 2017. Pp 170-175, ISBN 978-0-620-74503-1.

Also, the proceeding papers were orally delivered at both the 23rd International Conference on Domestic Use of Energy, Cape Town, South Africa and the 25th Southern African Universities Power Engineering Conference, University of Stellenbosch, South Africa.

Chapter five comprises information on the evaluation of the standby thermal energy losses taking into consideration that the required input electrical energy from the ASHP water heaters and the geyser were equivalent to the compensated thermal energy losses. Furthermore, empirical and statistical methods were established to quantify the standby thermal energy losses of each of the hot water cylinders upon the installations of isotherm blankets. This is a consolidated chapter from published works (both co-authors were my promoters as presented in authorship letter in appendix III):

i. Tangwe, S., Simon, M. and Meyer, E., 2014. Analytical Evaluation of the Energy Losses of an Air Source Heat Pump Water Heater: A Retrofit type.

Journal of Energy and Power Engineering, 8(7), pp 1251-1257. ii. Tangwe, S., Simon, M. and Meyer, E., 2017. Impact of standby losses and potential reduction by installation of isotherm blanket on the hot water cylinders. 25th *International Conference on Domestic Use of Energy* (DUE), 2017. pp. 101-109. Publisher: IEEE, IEEE Xplore Journal, ISSN: 978-0-9946759-2-7.

Also, the proceeding paper was orally presented at the 25th International Conference on Domestic Use of Energy, Cape Town, South Africa.

Chapter six incorporates the comparative analysis of the performance of residential split and integrated types ASHP water heaters using diagnostic characterisation predictors such as ambient weather conditions, electrical and thermodynamic properties of both systems with respect to volumes of hot water drawn off.

Part of this chapter is published (with the co-authors, being a research candidate under my mentorship and my promoter as presented in authorship letter in appendix III):

i. Tangwe S, Rubengo F and Simon M. 2016. Comparative analysis of the performance of an integrated and retrofit type air source heat pump water heater by diagnostic characterization. 15th International Conference on Sustainable Energy Technologies–SET 2016 (19th– 22nd of July 2016), National University of Singapore, Singapore. http://set2016.chbe.nus.edu.sg. Paper id: #113.

Moreover, the conference proceeding manuscript was orally presented at the 15th International Conference on Sustainable Energy Technologies, Singapore. The full chapter is peer reviewed and published in May 2018, in the Journal of Energy in Southern Africa, 29(2), pp. 12-20.

Chapter seven deals with the development and building of simplified multiple linear regression models benchmarking the coefficient of performance of both the residential split and integrated type ASHP water heaters with the following predictors; the difference in hot water set point temperature and ambient temperature, and the relative humidity. In addition, the equivalent thermal energy gained was equated to the electrical energy consumed by the electric geyser which served as the control experiment.

Part of this chapter is published (co-authors were my promoters and two research colleagues as presented in authorship letter in appendix III):

i. Tangwe, S., Simon, M., Meyer, E.L., Mamphweli, S. and Makaka, G., 2015. Performance optimization of an air source heat pump water heater using mathematical modelling. *Journal of Energy in Southern Africa*, 26(1), pp.96105.

The full chapter is under review for publication consideration in the Journal of Energy Efficiency, submission date: June 2017, status: Under review.

Chapter eight encompasses the development and building of surface fitting regression models that correlated both electrical energy consumption and product of ambient temperature and relative humidity to the coefficient of performance of the residential split type ASHP water heater without electric backup and an integrated type ASHP water heater with an electric backup. It also demonstrates an in-depth correlation of both predictors to the coefficient of performance using the three-dimensional surface fitting mesh plots and twodimensional multi contour plots simulation.

This is a consolidated chapter from published works (co-authors were my promoters as presented in authorship letter in appendix III):

i. Tangwe, S.L., Simon, M. and Meyer, E.L., 2017. Prediction of Coefficient of Performance and Simulation Design of an Air Source Heat Pump Water Heater. *Journal of Engineering, Design and Technology*, 15(3), pp.378-394.

ii. Tangwe, S., Simon, M. and Meyer, E., 2015. Models based simulation of the coefficient of performance of a domestic heat pump water heater. 3rd Southern African Solar Energy Conference, South Africa, 11-13 May 2015, pp.353-358. ISBN: 978-1-77592-109-7. Available at: http://hdl.handle.net/2263/49520.

Similarly, the conference proceeding paper was orally presented at the 3rd Southern African Solar Energy Conference, Kruger National Park, South Africa.

The full chapter is peer reviewed and published in March 2018 in, Journal of Thermal Science and Engineering Progress, 5, pp. 516-523.

Chapter nine contains the development and building of robust and multivariate models of the coefficient of performance of both residential split type ASHP water heater without electric backup and an integrated type ASHP water heater with an electric backup using ambient temperature, relative humidity and change in the refrigerant temperatures at the inlet and outlet of the compressor and condenser as the predictors. It also demonstrates an in-depth relationship of all the predictors to the coefficient of performance using the two-dimensional multi contour plots simulation. The predictors were ranked according to their importance of weight contribution, and also a test was conducted to determine any significant difference in the group COP means for both types of ASHP systems under the different operational scenarios. Lastly, a simulation application was designed to predict the COP of both types of ASHP water heaters.

This is a consolidated chapter from published works (co-authors were my promoters as presented in authorships letter in appendix III):

i. Tangwe, S., Simon, M. and Meyer, E., 2014. Mathematical modelling and simulation application to visualize the performance of retrofit heat pump water heater under first-hour heating rating. *Renewable Energy*, *72*, pp. 203-211.

ii. Tangwe, S, Michael Simon and Edson Meyer, 2016. Dynamic system modelling as a robust tool to evaluate the performance of domestic integrated and split type air source heat pump water heaters. 4th Southern African Solar Energy Conference. (30 Oct – 01 st Nov 2016), University of Stellenbosch, South Africa. pp 87-93, ISBN: 978-0-7972-1658-7

In addition, the conference proceeding paper was orally delivered at the 4th Southern African Solar Energy Conference, Stellenbosch, South Africa. Chapter ten assembles information on the general discussion, originality of work, findings, concluding remarks and recommendations from the research conducted. It also highlights the recommendation of a proposed hybrid photovoltaic assisted ASHP water heater and future research. As a final point, a list of research publications associated with this study and other publications is herein presented.

1.12 Matrix table of the chapters, research questions and objectives

Table 1.1 shows the respective chapters and its corresponding research section, together with the associated research questions and objectives accomplished.

Table 1.1: Matrix table for the Chapters and the deliverables

Chapters	Research section	Research questions	Objectives
Chapter one	Commentary (General introduction)		
Chapter two	Commentary (Fundamental principles and literature review)		
Chapter three	Commentary (Methodology)	Question i	Objective i
Chapter four	Publications (Results and discussion)	Questions ii & iii	Objectives ii & iii
Chapter five	Publications (Results and discussion)	Question iv	Objective iv
Chapter six	Publications (Results and discussion)	Question v	Objective v
Chapter seven	Publications (Results and discussion)	Questions vii & viii	Objectives vii & viii
Chapter eight	Publications (Results and discussion)	Questions vii, viii & ix	Objectives vii, viii & ix
Chapter nine	Publications (Results and discussion)	Questions vii, viii, ix & x	Objectives vii, viii, ix & x
Chapter ten	Commentary (General discussion, findings, contributions, conclusions, recommendations and future works)		

Chapter Two

Fundamental principles and literature review

2.1 Heat pump water heater technology

The heat pump water heater is a conversion system comprising of a heat pump unit and a storage tank. It is of paramount importance to highlight that the geyser can function as a storage tank provided the heating element can be disabled or removed from the hot water cylinder. The heat pump operates on a vapour compression refrigeration cycle (VCRC) similar to the air conditioning unit (i.e. reverse Rankine cycle); although, in the air conditioning unit, the cycle is intended for air cooling purposes.

By induction, the heat pump water heater is named based on the source from which it is deriving its renewable energy (Hepbasli and Kalinci, 2009).
According to this criterion, if the renewable energy is from the ground (geothermal energy), it is called ground or geothermal source heat pump (GSHP) water heater. Also, if the energy source is from the air, it is called an air source heat pump (ASHP) water heater, and if the source of energy is directly from the sun (solar energy); thus, solar assisted heat pump (SAHP) water heater. Lastly, if the energy source is from water (hydrothermal); it is therefore called water source heat pump (WSHP) water heater.

Overall, the ASHP water heaters can further be classified as split and integrated types. In addition, the heat pump unit in an air source heat pump water heater transfers the renewable aero-thermal energy from the environment to the water stored in the tank. Hot water heating using the ASHP water heater is achieved by the VCRC taking place in the heat pump unit while the storage tank serves as a reservoir for the hot water (Cochran and Cochran, 1981). It is worth mentioning that both the split and integrated type ASHP water heaters were implemented and studied in this research.

2.2 Description of an ASHP water heater

2.2.1 Major components of an ASHP unit and how it works

ASHP (standalone system) constitutes of the following principal components:

i. An evaporator acting as a heat exchanger between the ambient air and the refrigerant (liquid and vapour coexist). Heat is transferred from the ambient air to the refrigerant.

ii. A compressor that compresses lower pressure and temperature refrigerant vapour to a high temperature and pressure super-heated refrigerant vapour. iii. A condenser which acts as a heat exchanger between high temperature and pressure refrigerant and circulating water inside the water pipes embedded in the condenser compartment.

iv. A thermal expansion valve which carries out the process of throttling thereby converting high pressure and high temperature saturated refrigerant liquid to low-temperature and low-pressure refrigerant (liquid and vapour coexist). In addition to these primary components, there are also:

v. A propeller axial fan or blower situated at the rear end of the evaporator which is responsible for the forceful convection of ambient air to enhance the rate of thermal energy transfer.

vi. An electrical induction motor to drive the crank shaft of the compressor during the VCRC.

vii. Refrigerant acting as the working (primary) fluid and undergoes phase changes during the compression and expansion cycles. The refrigerant (primary) fluid used in heat pumps must be able to possess very good thermo-physical properties to ensure efficiency in the expansion and compression cycles and also need to be non-toxic, non-flammable, with zero ozone depletion potential, minimal global warming potential and a very low boiling point etc.

viii. A water circulation pump (for split type) to enable the flow of water (secondary fluid) circulating between the tank and the condenser of the ASHP unit.

2.2.2 Operation and function of an ASHP water heater

An ideal ASHP water heater transfers thermal energy during its VCRC from ambient air to heat water in the storage tank and in turn causes cooling as well as to an extent, dehumidification of the air depending on the ambient condition. Figure 2.1 shows a block diagram of energy distribution in an ideal ASHP water heater and Figure 2.2 provides a schematic diagram of the components involve in the VCRC processes which occur in a typical ASHP unit.

A salient and better understanding of the refrigeration cycle of heat pump water heater was given by Ashdown (2004) and Sinha and Dysarkar, (2008). During a VCRC, aero-thermal energy gained at the evaporator end is absorbed by the pure refrigerant (liquid and vapour coexist) to change the phase of the liquid portion to vapour without any change in the refrigerant temperature (latent heat) and also the pure refrigerant gains negligible sensible thermal energy. The process is isothermal and occurs on stage (1- 2) as shown in Figure 2.2. Owing to the pressure difference between the suction line and the discharge line as shown in Figure 2.2, the pure refrigerant vapour (dry and low temperature and pressure refrigerant vapour) flows to the compressor, where the vapour is compressed to a super-heated vapour and exits along the discharge line. The process is isenthropic and occurs on stage (2 – 3) as illustrated in Figure 2.2. As the super-heated pure refrigerant vapour flows into the condenser, the refrigerant is condensed, and a saturated refrigerant liquid is formed, alongside. Thermal energy is dissipated to heat the water flowing inside the inner tube of the condenser. At this stage (3 – 4) as shown in Figure 2.2, the super-heated vapour temperature drops to form a sub-cool vapour, which in turn loses thermal energy to become a saturated refrigerant liquid. At the expansion valve, the pressure and temperature decrease and the saturated pure refrigerant liquid becomes a low-pressure liquid refrigerant. The process is an isenthalpic process and occurs on stage (4 – 1) as illustrated in Figure 2.2.

Similarly, in ASHP water heater, thermal energy is transferred from the air (cold reservoir) to heat water (hot reservoir) and this process can only be possible with the input of energy (electrical) into the heat pump (cyclic engine) in conformity with Clausius's statement which is in accordance with the second law of thermodynamics (Egbert and Rienk, 2013).

An efficiently installed ASHP water heater has a COP ranging between 2 and 4, whereas typical conventional water heaters (i.e. electric resistance element, coal, gas, kerosene stove, etc.) have a performance energy factor less than or equal to 1 (Levins, 1982; Bodzin, 1997).

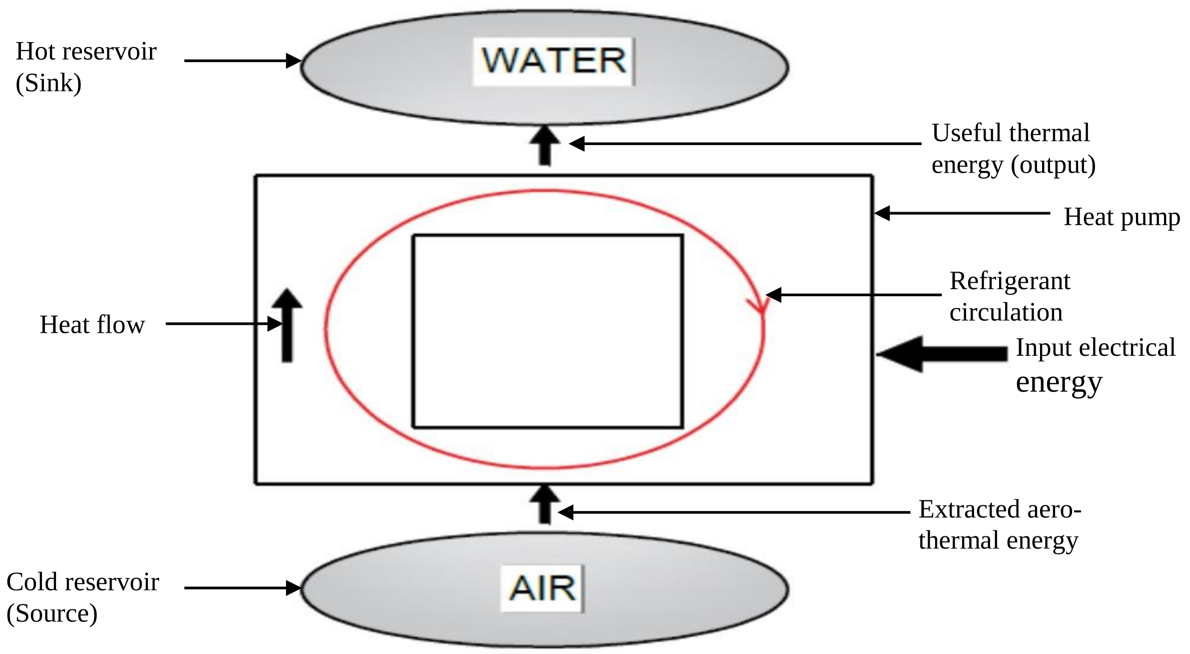


Figure 2.1: A block diagram of energy distribution for ideal ASHP water heater

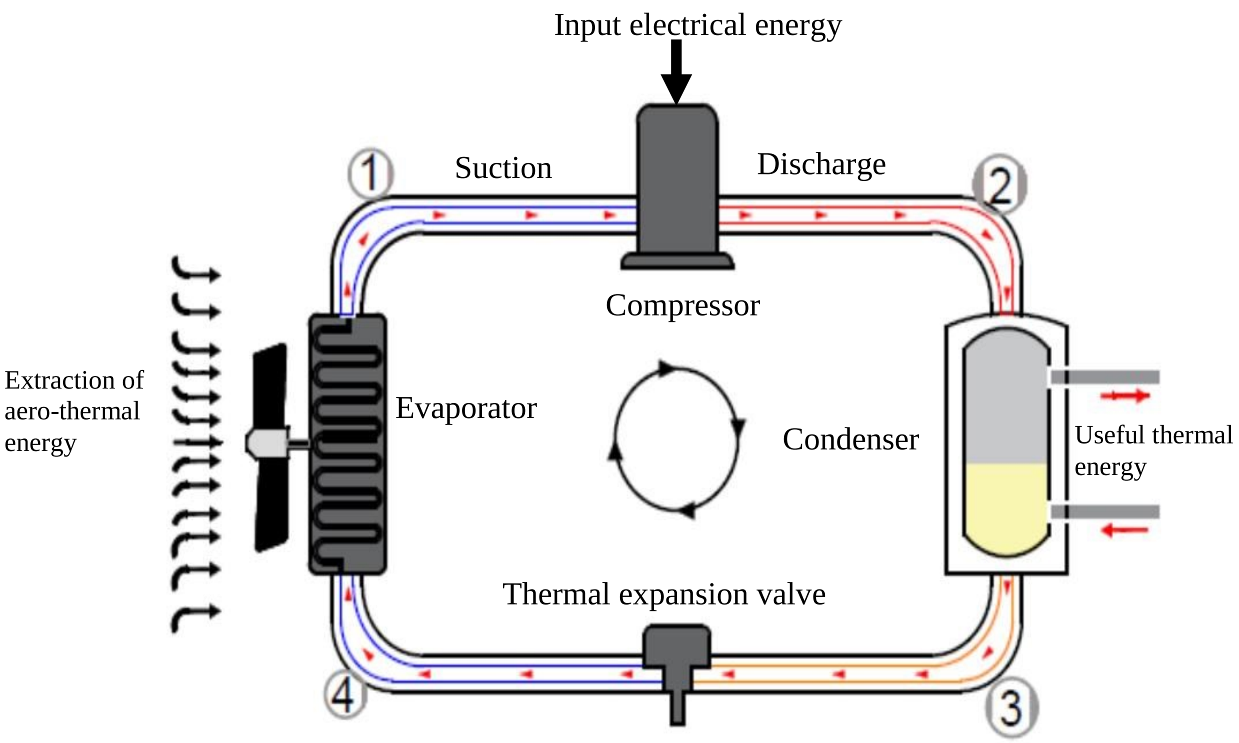


Figure 2.2: A schematic block diagram of the ASHP main components

2.3 Types of ASHP water heater in South Africa

Although there is a substantial growth in the technology of the ASHP water heater, it is not yet economically ascertained due to its market price, limitation of public awareness of the product added to a wrong conception of system durability (Douglas, 2008). Furthermore, poor installation and lack of routine maintenance can lead to inefficiency of the system (Douglas, 2008). Nevertheless, heat pump water heaters also render an extra benefit of dehumidification and space cooling during operation, wherein, it pulls warm vapour from the air (Baxter *et al*., 2005).

In Japan, there are already manufactured innovative heat pumps that exploit carbon dioxide as the refrigerant fluid and are more than 300% energy efficient. These became feasible due to the government and private partnership rebates initiatives (Hashimoto, 2006; Maruyama, 2008).

There are two common types of ASHP water heaters namely;

a) Integrated type ASHP water heater: It describes a heat pump water heater in which the condenser is immersed as an essential part of the tank or mounted inside the tank. Thermal energy is transferred to the water in the tank by free convection over the tank wall or by the condenser tubing inside the tank. It is also known as a hybrid or ‘drop-in' heat pump water heater. Figure 2.3 shows a schematic diagram of a residential integrated type ASHP water heater.

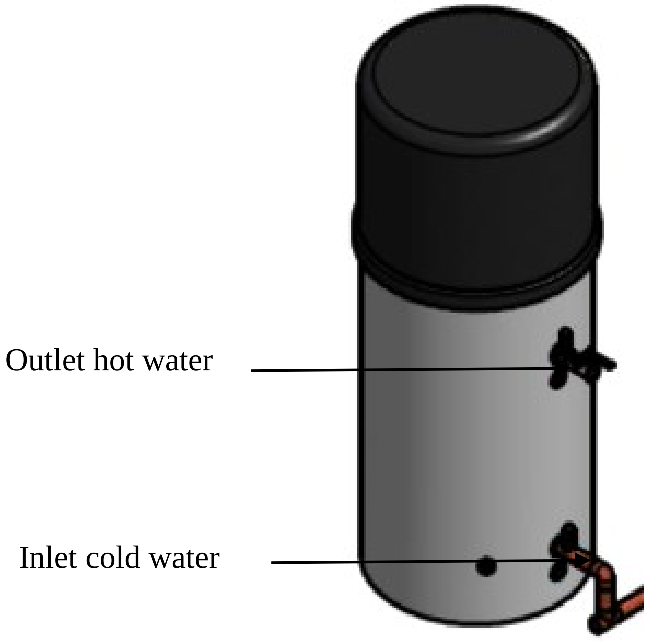


Figure 2.3: Residential integrated type ASHP water heater

b) Split type (Standalone) heat pump water heater: It is a heat pump water heater without the heat pump unit directly mounted together with the storage tank. Here, heat is delivered to water flowing through the condenser of the heat pump. It is also known as the retrofit type ASHP water heater. In addition, split systems can be grouped into re-circulating and once-through as described below;

i. Re-circulating split type heat pump water heater: It is a heat pump water heater that requires recirculation of water between the tank and the condenser unit of the heat pump before it attains the required set point temperature during the VCRC. This type of system is also known as a multipass system.

ii. Once-through split type heat pump water heater: In this type, the heat pump is capable of delivering water at the required set point temperature (usually 55°C or higher) in one pass through the condenser unit of the heat pump.

Figure 2.4 shows a diagram of a residential split type ASHP water heater.

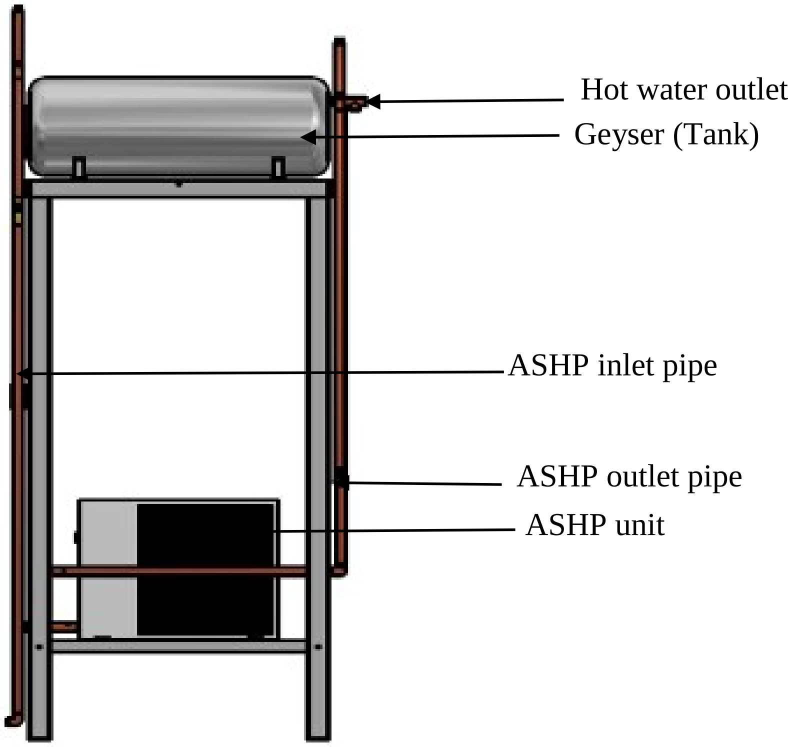


Figure 2.4: Residential split type ASHP water heater

Figure 2.5 shows a detailed chart of the classification of heat pump water heaters with great emphasis on the ASHP water heater which is critically monitored under this research.

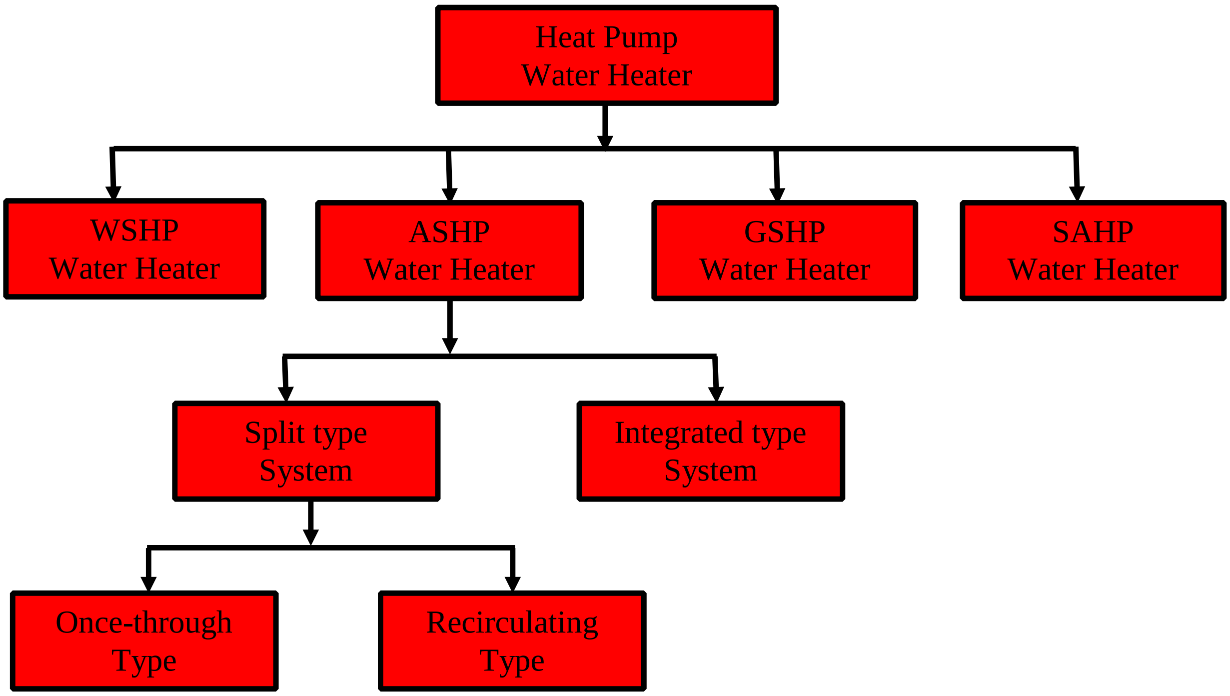


Figure 2.5: A detailed chart of the classification of heat pump water heaters

2.4 Control System of ASHP water heaters

The core components of the ASHP water heater for residential purposes include a single speed hermetic rotary compressor, single speed circulation water pump and at most two speed regimes for fan control. Based on this configuration, the ASHP unit is said to operate in the on/off control scheme whereby the main energy users (compressor, circulation water pump and fan) are turned on, only when the water temperature goes below a certain temperature differential (usually around 12°C) and turned off when the water reaches the set point temperature (usually 55°C to avoid growth of *Legionella* sp). Despite the design specifications to satisfy maximum load, these systems function at quasi-partial load throughout their life cycle. Such a conventional technique to cope with partial loading could degrade the compressor durability significantly (Saleh and Ayman, 2015). Also, the components of the system controlled under such scheme are being inefficiently utilised, energy-wise as they suffer big drawbacks of undesired current peaks during its state transitions (Orhan *et al*., 2012; Vinther *et al*., 2012).

Remarkably, one novel approach has been to introduce capacity control in the heat pump units in order to be able to match the heating load or working point to the consumption point. This is primarily driven by the fact that residential ASHP units operate under dynamic conditions like varying heat load (volume of water heated at a particular time) with ambient weather variations. Capacity control is therefore desirable to match operating conditions to the system's optimal performance by reducing power and energy consumption, reducing compressor cycling as well as decreasing starting load and possibly, good oil return. Except for on/off control which is the simplest form of capacity control, other control mechanisms exist like variable speed compressors, hot gas bypass with or without liquid injection, and digital control circuits for scroll type compressor (HWR, 2014). With the present circuitry configuration of residential ASHP water heaters, the main actuators (compressor, pump and fan) are mostly built from induction motors making it easy for a variable speed capacity control technique to be implemented.

However, capacity control by adding variable speed compressors in heat pump systems has been tackled both theoretically and experimentally by many researchers. Green *et al*. (1980) carried out some of the pioneer works on capacity control of heat pumps. They built an electrically-driven ASHP water heater which offered compressor control, motorized expansion valve and a variable speed air flow fan. The entire heat pump unit was fully instrumented by means of a suitable control algorithm through a microprocessor control unit.

Experimental validation of test data showed that the prototype operated with an improved COP compared to conventional systems (Green *et al*., 1980). Similarly, Wang *et al*. (1983) also worked on a novel heat pump control system using the classic variable speed compressor, motorized expansion valve and a variable-input air mass flow rate. All sensing and motorized functions were handled by a central microcomputer based control system which maintained the refrigerant pressure across the evaporator to ensure maximum heat transfer. In addition, the results obtained from the experiment revealed that the efficiency of the heat pump could be improved using the on-line system (Wang *el al*., 1983). A similar study was carried out in 1989 by Parnitzki who developed a digital control system based on a microcomputer to fully automate and entirely motorize a heat pump. Although, the system was able to operate under very much varying conditions than precedent technologies, the prototype could operate near optimum by regulating the temperature difference at the evaporator (Parnitzki, 1989).

Karlsson and Fahlen (2007) investigated the energy-saving potential of using variable-speed capacity control instead of the conventional intermittent operation mode in domestic ground source heat pumps (GSHP). Intermittent control and variable-speed capacity control were compared on a benchmark experiment using two capacity-controlled heat pumps and one standard heat pump with a single-speed compressor. Results showed that capacity-controlled technique primarily, depended on a correct relationship between refrigerant flow and heat transfer media flows. Despite the improved performance at part load, the variable-speed controlled heat pump did not improve the annual efficiency unlike the intermittently operated heat pump (Karlsson and Fahlen, 2007). Equally, Madani *et al.* (2010) studied capacity control with emphasis on the compressor and inverter loss behaviour in a variable speed controlled heat pump. The data obtained from experiments demonstrated that an increase in the compressor speed caused a reduction in the COP of the heat pump, of up to 30%. The inverter losses increased as the compressor speed was increased, although, the total compressor power decreased. Moreover, increasing the compressor speed alongside, the pressure ratio from 2.7 to 5.8, provoked increase in the loss due to the drastic pressure ratio mismatch. Finally, the highest total isenthropic efficiency of the compressor was obtained when the compressor frequency was close to 50Hz (Madani *et al*., 2010).

2.5 Comparison of performance of ASHP and GSHP water heater In general, geothermal source heat pump (GSHP) water heater can perform better than air source heat pump (ASHP) water heater both as a single or coupled system and with an excellent payback time. However, the capital cost of the design and construction of the system is enormous as opposed to the ASHP water heater.

Several studies conducted by other authors across the globe revealed and confirmed the high performances of GSHP system over ASHP system. Such studies included; A techno-economic analytical comparison of the performance of air coupled and horizontal-ground coupled air conditioners conducted in South Africa (Petit and Meyer, 1999). A payback assessment of heating and cooling GSHP system using carbon dioxide as the primary refrigerant was carried out in a high energy consumption area in Tokyo (Hepbashi, 2002). Hepbashi (2002) conducted a performance evaluation of a vertical groundcoupled heat pump system in Izmir, Turkey to justify the energy saving potential of the system. Also, a techno-economic comparison of ground-coupled heat pump system for space cooling was also demonstrated by Esen (2007). The author further investigated the parameters affecting the performance of a ground source coupled heat pump (Inalli and Esen, 2004).

In addition, a comparative study based on performance was carried out between an air-coupled heat pump and an air-coupled air conditioner in South Africa (Oerder and Meyer, 1997; Petit and Meyer, 1998). Furthermore, Bi and co-workers (2004) evaluated the performance of both solar and ground coupled heat pump systems.

2.6 Eskom's categorisation of the rebate ASHP water heaters

Eskom adopted simple criteria to group the list of accredited residential split and integrated type ASHP water heaters in South Africa. The necessary parameters for the grouping included a specific range of input electrical power consumption and the capacity of the storage tank. Table 2.1 shows the categories of both the split and integrated type ASHP water heaters (Eskom, 2013).

Table 2.1: Categories of split and integrated type ASHP water heaters

Category	Tank size (L)	Range of electrical input power (kW)
Small tank	100	0.5 – 1.0
Small tank	150	0.8 – 1.5
Small tank	200	0.9 -1.8
Small tank	300	1.2 – 2.0
Large tank	350	1.4 – 2.5
Large tank	400	1.8 – 2.7
Large tank	500	2.0 - 3.0

According to the categorisation, other key parameters including refrigerant charge, types of refrigerants, the design of the closed loop circuit and products manufacturer were not taken into account. Based on the uptake of the technology and from the Eskom database, both the 150 L split and integrated type ASHP water heaters have the largest market penetration (Eskom, 2013). The research focused on extensive performance monitoring using the small tank (150 L, 0.8-1.5 kW). Both the Airco integrated type ASHP water heater (Integrated type ASHP water heater of 150 L tank size and input power of 0.9 kW) and the SIRAC split type ASHP water heater (split type ASHP water heater of 150 L tank size and input power of 1.2 kW) according to manufacturer's specifications were selected and used for the comparative analysis.

2.7 Literature review

The literature review covers access of functional source of energy and its primary intended purpose, especially in the residential sector and with emphasis in South Africa. The core of the literature was on sanitary hot water heating using geyser and ASHP water heaters.

2.7.1 Literature introduction

Electricity is a functional form of energy and in the Africa continent, there exist a severe challenge whereby demand is exceeding the supply. Therefore, there is a crucial need of the implementation of integrated demand management and energy efficiency interventions. Above all, electricity access is of low levels within the Sub Saharan Africa region. Studies have demonstrated that owing to the deficiency in modern energy access, less than 17% of the region's population, and less than 5% of rural areas are electrified (Davidson and Sokona, 2002). Paramount to the highlighted energy crisis, Africa's energy need is expected to increase by 85% between 2010 and 2040 (EIA, 2016). Despite the new power generation, the associated infrastructure is critical in bridging the gap between energy supply and demand. As a consequence, the implementation of energy efficiency as a least-cost energy resource is fundamental. This helps in reducing overall demand, decrease potential energy peak load, and allows electricity supply to be optimally utilised to meet the increasing demand in a timely, low-cost, and sustainable way.

Precisely, energy efficiency initiatives have been effectively employed in Ghana and South Africa which resulted in significant peak energy savings of 120 MW and 3 GW, respectively, during their pilot projects (Eskom, 2014). However, the penetration of energy efficiency in Africa is still insignificant both at the industrial, transport and domestic level as a result of the combination of the following factors; poor institutional framework and infrastructure, poor baseline information, lack of energy engineers in conjunction with minimal incentives to promote energy efficiency technology (Karekezi *et al*., 2005). Sustainable energy regulation and policy-making for Africa (SERPA, 2015) identified three key strategies that could aid in overcoming the barriers faced in developing both renewable energy and energy efficiency systems in the region. These included; energy efficiency and renewable energy policy programmes; appropriate technology, technology transfer and building local capacity and lastly, innovative financing mechanisms.

The economy of South Africa is energy intensive with the industrial sector having the greatest demand compared to others like the residential, commercial, transport and agricultural sectors. South Africa is one of the countries with high dependence on coal, being used primarily, for local energy production. The country's coal reserve is estimated to be about 53 billion (about 92.8% of electricity coming from coal) (SSA, 2009). Figures 2.6 and 2.7 illustrate the energy distribution for both the generation and the demand-side, respectively. Even with the large renewable energy potential of South Africa, only about 1% is effectively utilised for electricity production. Furthermore, at the residential sector, energy consuming activities are largely dominated by the production of sanitary hot water via heating. In a typical residential setup, approximately 45% of the energy consumed is due to water heating (WH) followed by energy consumed by way of use of the washing machine (WM) and finally, energy consumed by small electrical rated devices like fridge, TV's and stoves as shown in Figure 2.8 (SSA, 2005).

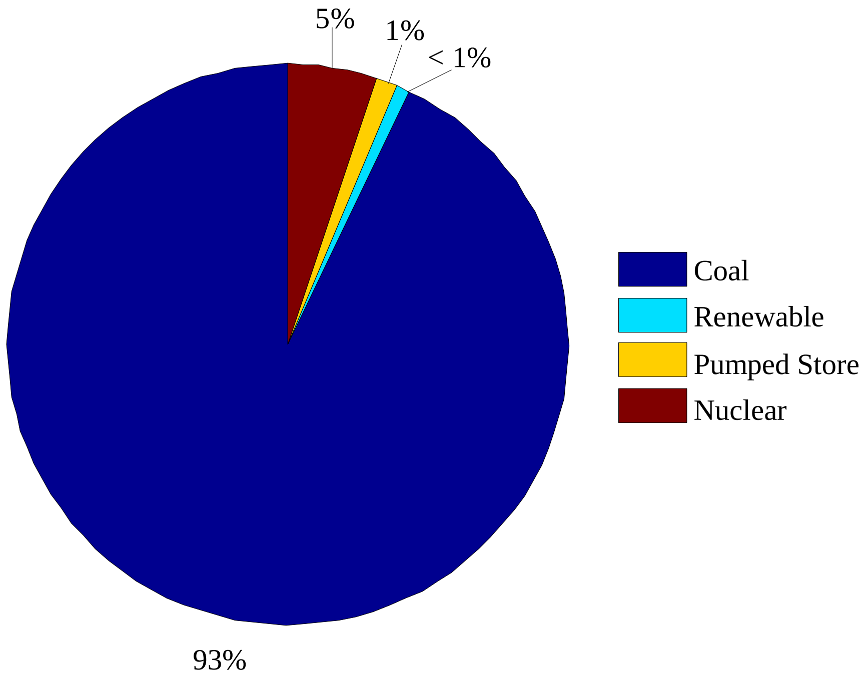


Figure 2.6: South Africa’s Local Energy Production Partition

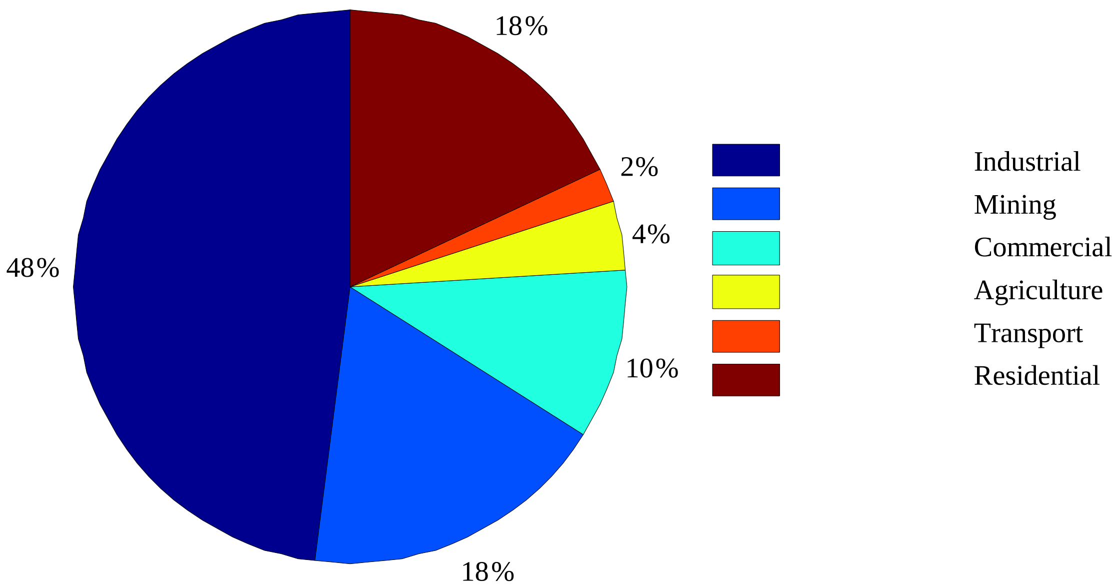


Figure 2.7: South Africa’s Electricity Demand Partition

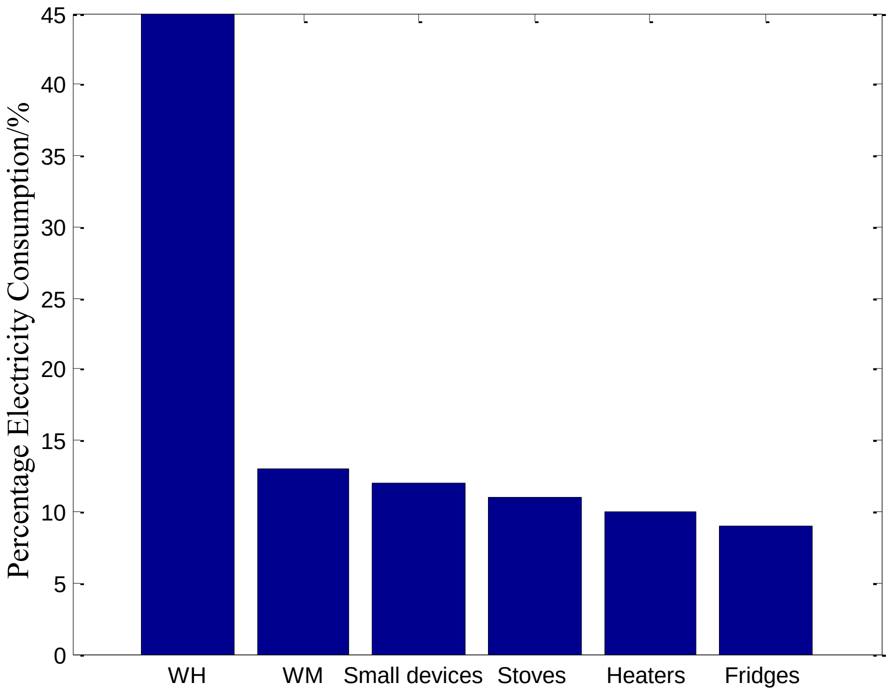


Figure 2.8: South Africa's residential electricity consumption

Following, the multi-purpose benefits which cuts across low-cost, sustainable and demand reduction potential achieved via energy efficiency initiatives, the government through Eskom in 2004, embarked on energy efficiency and Demand-Side Management funding programme with the target to promote the implementation of more energy-efficient technologies, processes and behaviours amongst all electricity consumers. A qualitative analysis depicted that South Africa achieved a demand reduction of over 2,770 MW from all Demand Side Management projects initiated in 2004 through to 2011 (Skinner, 2012).

With the high energy demand during the Eskom morning and evening peaks, renewable and energy efficiency technologies rebate programmes targeting the water heating sector were introduced. Among them, was the solar water heating rebate programme which was meant to reduce 2,300 GWh of energy consumption between the pilot scheme periods from 2008 to 2013. The project however, claimed to have achieved in 2011, the total installation of over

156,000 solar water heaters (high pressure and low pressure) with energy savings of 60 GWh/annum (Eskom IDM, 2011). In addition to the solar water heating rebate programme, was the residential heat pump rebate programme which targeted the installation of 65,580 units of heat pumps between November 2010 and March 2013, across the six Eskom Distribution Regions and head offices. The rebate programme projected that a total of 54 MW (80.86 GWh) at a load factor of 17% will be saved from the installation of the 65,580 heat pumps (University of Pretoria, 2011). The heat pump rebate programme was primarily aimed at retrofitting existing geysers in residential homes with heat pumps. Therefore, it was envisaged that this strategy will go a long way to promote the use of this technology within the residential sector. However, the Eskom residential ASHP water heater rebate programme was discontinued in 2013 (Eskom, 2014) due to the inability of the National Energy Regulator to continue the funding scheme. This left the country without any comparative tests for residential ASHP water heaters. It is paramount to highlight that the discontinuation of the heat pump rebate scheme was concluded as a result of lack of funding to support the initiative eventhough, the systems demonstrated an excellent overall COP of over two, all year round (Tangwe *et al.*, 2014).

It is worthy to mention that all the ASHP water heaters contained an ASHP unit and a geyser in the form of a storage tank. Geysers vary with tank sizes, tank configurations and types of heating elements. Hence, the geyser tanks can either be vertical or horizontal and in which, a circular or a straight heating element is installed.

The South African market registers 14 Eskom accredited ASHP suppliers namely; South Africa heat pump engineers cc, Fourway air conditioning, Genergy Pty, ITS Solar, Kwikot Pty, M-Tech industrial, Thermo wise, Energy efficiency homes and business, SIRAC Southern African, Airco Pty, Powertech IST, Solatricity, Liquid heat, SIRAIR and Express Mining Supplies (Eskom, 2011).

Heat pump water heaters since their invention in the 1950's have been at the centre of all refrigerant processes (Cochran and Cochran, 1981). Yongoua *et al*. (2016) summarised some of the most prominent works recently carried out to assess the performance of ASHP water heaters both at the macroscopic level as well as the individual system components. Some of the important environmental and uncontrollable parameters that affect the performance of ASHP water heaters are; the volume of hot water drawn by the user, ambient temperature, relative humidity and the degree of insulation of the storage tanks. However, the most vital system components that largely influence the performance of ASHP water heaters are the compressor types and the choice of heat exchangers. An in-depth research on the system performance, taking into consideration, system components and their combined influence on the overall performance of ASHP water heaters had been conducted. Zhang *et al*. (2007) worked on the possibilities of optimising the performance of ASHP water heaters by considering capillary tube length, the filling quantity of refrigerant, the condenser coil tube length and system matching.

2.7.2 The performances of hot water technologies and standby losses The characteristic feature which gives the heat pump water heater an efficiency of more than 300% is known as the coefficient of performance (De Swardt and Meyer, 2001). The instantaneous, seasonal or annual COP can be determined using the TRNSYS simulation software package (KLEIN-TRNSYS, 1990). Itoe *et al*. (1999) presented an analytic, mathematical model of the performance of solar assisted heat pump water heater correlating ambient temperature and hot water set point temperature to COP. A dynamic performance of water heater driven by heat pump was proposed and designed to model the coefficient of performance of heat pump water heater (Kim *et al*., 2004). It was demonstrated that the coefficient of performance of heat pump water heater could be enhanced by using R11 (Chlorofluorocarbon compound) and R12 (Hydrochlorofluorocarbon compound) as the thermo-physical refrigerant in the heat pump unit (Zhen-Hao *et al*., 2005). However, both R11 and R12 have been phased out due to their high ozone depletion and global warming potentials.

Notwithstanding, most modern and acceptable ASHP water heater are using either zeotropic, azeotropic or alkanes as refrigerants. In details, an azeotrope could be defined as a mixture consisting of two or more refrigerants with similar boiling points that act as a single fluid. The components of azeotropic mixtures will not separate under normal operating conditions and can be charged as a vapour or liquid while a zeotrope is a mixture made up of two or more refrigerants with different boiling points. Zeotropic mixtures are similar to nearazeotropic mixtures except that they have a temperature glide greater than 12^{o}C. In addition, zeotropic mixtures should be charged in the liquid state most preferably (http://www.refrigerants.com/terminology.htm, 2012). Also, the refrigerants used as the primary fluid in the split and integrated type ASHP water heaters were R417A and R407C (Zeotropic refrigerants) with almost the same critical temperatures and pressures. It should however, be emphasised that the heat transfer coefficient of

R417A is better than in R407C (Aprea *et al*., 2008). Furthermore, a heat pump water heater with dual tanks gives a better performance than the corresponding system with a single tank and the hot water usually attains a much higher temperature (Hiller, 1996).

The performance of hot water heating devices is adversely impacted by the standby thermal energy losses of the systems. Furthermore, the average energy factor of a geyser is 0.92 owing to the standby thermal energy losses in the hot water cylinder (Haung and Lin, 1997, Tangwe *et al*., 2017). The hot water cylinder or geyser standby losses are the thermal energy losses from the stored water as the temperature drops below its set point over a 24-hour period without any hot water drawn off. The geyser standby thermal energy losses were determined in the multi-level expert modelling, evaluation of geyser load management opportunities in South Africa (Deport and Van Harmelen, 1999). Moreover, an experimental method was conducted to determine the geyser standby losses (Beute, 1993), as well as an optimised geyser control switching method was used to minimise the geyser standby losses (Zhang *et al*., 2007). Nevertheless, a laboratory benchmark approach was employed to evaluate the standby losses of an integrated heat pump water heater (Sparn *et al*., 2011).

2.7.3 Techno-economic potential of ASHP water heaters

The techno-economic analysis of a technology is a measure of the payback period. Vividly, the payback period is an economic analysis of a technology in a bid to assess its viability in retrospect to its capital cost and to some extent, the maintenance cost (Tangwe *et al*., 2014). A technology can be considered viable provided both the lifespan and payback period are favourable. The payback period could also be greatly impacted by the increase in electrical energy tariff over the years. The ASHP water heater is an energy-efficient device for sanitary hot water production. It is capable of using 1 unit of input electrical energy to provide 3 units of useful thermal output energy during vapour compression refrigeration cycles due to its coefficient of performance of 3 (Bodzin, 1997; Tangwe *et al*., 2014).

It is worth mentioning that most hot water devices are the conventional heater (electric geysers) with an average energy factor of 0.92 (Haung and Lin, 1997). The ASHP water heater is a renewable energy device capable of heating water with the majority of the useful thermal output energy derived from the ambient aero-thermal energy (Morrison *et al*., 2004). It can provide energy saving in the range of 50-70%, as the ASHP unit has a coefficient of performance ranging from 2 to 4 (Levins, 1982; Bodzin, 1997). The type of hot water storage tank utilised in the ASHP water heater is a real challenge to the hot water temperature inside the tank. A similar volume of water heated by an ASHP is said to be at a much higher temperature in a dual tank than in a single tank system, but the thermal energy losses are lower for the latter (Hiller, 1996). Tangwe *et al*. (2014) demonstrated that the residential split type ASHP water heater is a viable and renewable energy technology for sanitary hot water production with a favourable techno-economic potential.

2.7.4 Mathematical modelling of ASHP water heater

A mathematical model is the use of mathematical language or equations to describe the dynamic behaviour of a system or process, taking into considerations some predictors to forecast the response. It can be of great benefit in optimisation and control of the system under different scenarios. Different regression models have been developed and built to model the performance of residential air source heat pump water heater in the separate heating cycles. Specifically, the multiple linear regression models were used as the mathematical models to predict the performance of split type ASHP water heater under the first-hour heating rating (Tangwe *et al*., 2014). In addition, mathematical models embedded in the multi-dimensional contour plots simulation in the MATLAB statistical tool were used to illustrate how each of the predictors (ambient temperature, relative humidity and the COP of heat pump unit) varied with the COP of a split type ASHP water heater while all the other predictors were held constant (Montgomery and Myers, 1995; Tangwe *et al*., 2013; Tangwe *et al*., 2018).

It must be alluded that a pocket of dynamic models of heat pump water heaters have been developed. More so, the bulk of the established mathematical models were developed from first principles whereby the integrated model of the heat pump water heaters was derived from the combination of the subsystem models that made up the VCRC closed loop circuit. Fardoun *et al*. (2011) developed a dynamic model of ASHP water heater based on independent heat transfer, thermodynamics, fluid mechanics and empirical correlations of the evaporator, compressor, condenser and expansion valve of the system. The results confirmed that the rate of heating increased with a decrease in the capacity of the hot water storage tank and also the performance of the integrated system increased with an increase in ambient temperature.

MacArthur and Grald (1989) designed and built a model of vapour-compression heat pumps. The evaporator and the condenser were modelled with in-depth heat distribution equations, while the expansion valve was modelled as a capillary tube. Fu *et al*. (2003) presented a dynamic model of air-to-water dualmode heat pump with a screw compressor having four step capacities. The dynamic models developed with the introduction of additional compressor capacity in a stepwise manner were studied. Kima *et al*. (2004) presented a dynamic model of a water heater system driven by a heat pump and applied a finite volume method to describe the heat exchangers. Furthermore, the lumped parameter models were employed to analyse the compressor and the storage tank, where dynamic simulations were carried out for various reservoir sizes. Techarungpaisan *et al*. (2007) presented a steady state simulation model to forecast the performance of a small split type air conditioner comprising of a rotary compressor and a capillary tube but integrated with water heater. Despite, the complexity of the dynamic models of the various heat pump water heaters, the determination coefficient of the predicted and measured COP was slightly above 0.9.

Furthermore, a multiple comparison test was performed to evaluate any significant mean difference upon comparing the interval between the difference of the 95% mean confidence interval and the true mean of the particular heating cycle with respect to the COP while employing the analysis of variance approach (Goodall, 1993; Hochberg and Tamhane, 1987; Tangwe *et al*., 2018). Interestingly, an extensive review of the literature has been undertaken on the performance assessment and optimisation of residential ASHP water heaters to justify the year-round efficiency of the technology (Yongoua *et al*., 2016). The authors further confirmed through a thorough presentation of facts that the coefficient of performance of the residential ASHP water heater could be accurately modelled using ambient weather predictors and system design of the components that form the closed loop of the VCRC circuit. Also, the study demonstrated that the performance of the ASHP water heater could be optimised through effective sizing of the length and diameter of the heat exchangers of the ASHP unit.

2.7.5 Research overview

Hot water heating contributes to a significant percentage of residential energy consumption, worldwide. In South Africa, more than 50% of the residential monthly energy consumption is from sanitary hot water production (Meyer and Tshimankinda, 1998). This research entailed the characterisation and mathematical modelling of the COP of residential integrated and split type ASHP water heaters using critical thermodynamic, electrical and ambient weather parameters as predictors. The ASHP water heater is capable of harnessing the ambient waste thermal energy in the form of aero-thermal energy and processed as high-grade thermal energy that is utilised for sanitary hot water heating during the vapour compression refrigeration cycle (Tangwe *et al*., 2014).

The COP of residential ASHP water heater ranges from 2 to 4 and depends on ambient conditions and the design of the major components (evaporator, compressor, condenser and expansion valve as well as the primary refrigerant) that make up the closed loop circuit of the VCRC. The focus of the research incorporated a detailed design and building of a DAS to determine the thermodynamic, ambient weather conditions and electrical parameters (volume of water heated, the amount of hot water drawn off, VCRC main component temperature profiles, ASHP inlet and outlet water temperature profiles; and also ambient temperature, relative humidity, power factor and electrical power and energy consumption profiles). These measured data were used in the diagnostic characterisation to benchmark the performance of the ASHP water heaters. Also, the thermodynamics and ambient weather parameters were used as predictors in the development of mathematical models to compute the COP of the ASHP water heaters under different scenarios (firstly, under the first-hour heating rating and secondly, under the controlled simulated volume of hot water drawn off). More so, a techno-economic analysis of the two types of ASHP water heaters was performed. Furthermore, real-time standby losses of both types of ASHP water heaters were statistically evaluated under two scenarios (without isotherm blankets and with isotherm blankets on the hot water cylinders).

Chapter Three

Research Methodology

3.1 Introduction

This chapter covers the complete research methodology employed in this study to achieve the supposed objectives. We explored the dynamic performance of the installed residential air source heat pump water heaters and the geyser under investigation as well as we justified the choice of these hot water systems in this experiment among other commercialized counterparts. The geographical location in terms of seasonal and annual variations of ambient weather parameters is also described. Finally, the instruments employed to collect the data, including methods implemented to preserve the validity and reliability of the metering instruments are described.

3.2 Research approach and design

The goals of this research were centered on the ten objectives as outlined in chapter one.

In this regard, a quantitative research was designed based on the objectives and systematic approach in gathering experimental data to describe variables like COP and determine its impact and interactive effect with other variables under a simulated controlled volume of hot water drawn-off. Specifically, this project sought to investigate the COP with ambient conditions as well as standby losses and payback period through measurable quantities like ambient temperature, relative humidity, electrical power consumption, temperatures of the refrigerant at the inlet and outlet of the compressor and condenser for both split and integrated type ASHP water heaters.

Due to the exploratory aspect of this work, a qualitative research component was incorporated through the development of mathematical models and designed simulation application to eventually compare the performance of both types of ASHP water heaters. This qualitative analysis, therefore, served as a benchmark to test for significant difference in COP of both ASHP water heaters based on controlled volume of hot water drawn-off and system variables such as energy consumption (E), ambient temperature (Ta), relative humidity (RH), set point temperature of hot water (Ts), change in temperature of refrigerant at the discharge and suction points of the compressor (Tcomo – Tcomi) and change in temperature of the refrigerant at the inlet and outlet of the condenser (Tconi – Tcono). Figure 3.1 shows the schematic layout of the research design.

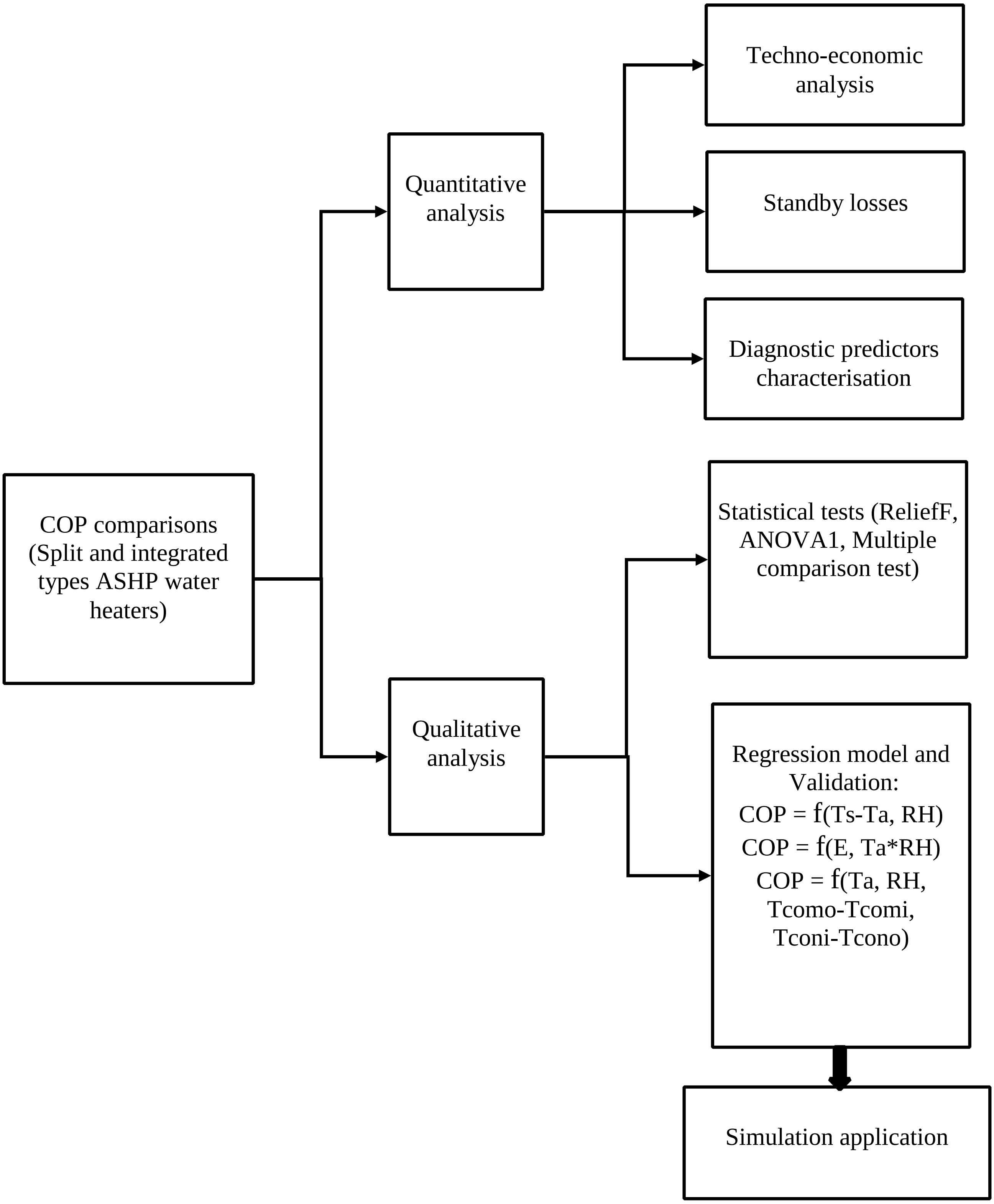


Figure 3.1: Layout of research approach

The experimental set up was built in the Fort Hare Institute of Technology research center, University of Fort Hare, Alice campus. The University of Fort Hare is a public university in the Eastern Cape Province, South Africa. It has three campuses of which, Alice is the main campus. The Alice main campus is situated near the Tyhume river about 50 km west of king William's town. Figure 3.2 shows the location with the GPS, and is found along Latitude: -32°46'59.99" S and Longitude: 26°52'59.99" E.

Figure 3.2: Satellite Map of University of Fort Hare, Alice Campus

Of other regions in South Africa, the Eastern Cape Province was chosen because of its legendary temperature records and wide annual temperature variations. For example, in November 1918, South Africa experienced the highest ever recorded temperature of 50.0°C at Dunbrody along the Sundays River Valley in the Eastern Cape Province and its ever-coldest recorded temperature of -18.6°C on the 28th of June, at Buffelsfontein near Molteno (Eastern Cape Province). Still, the coldest place in South Africa is Buffelsfontein near Molteno, with a mean annual temperature of 11.3°C and an average annual minimum temperature of 2.8°C (SAWS, 2016). However, the annual weather profile of Alice shows average monthly maximum ambient temperatures of around 27°C in the months of January, February and March while June and July register the lowest average temperatures. Additionally, Alice experiences the highest precipitation and consequently, the highest rainfall days around November and December while reaching a minimum around June and July (WWO, 2016).

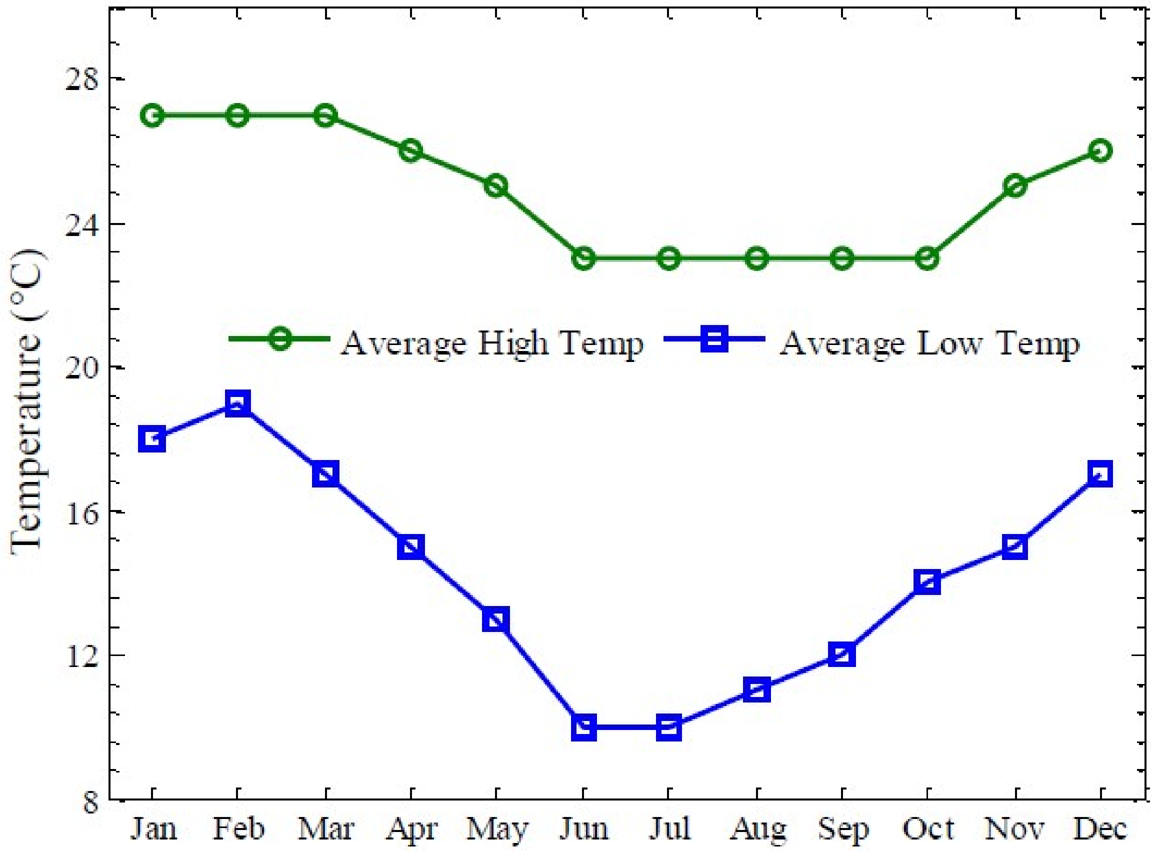


Figure 3.3: Monthly average temperature for Alice, South Africa

With sufficient evidence on the influence of hot water usage profile on the performance of ASHP water heaters (Yongoua *et al*., 2016), the experiment was designed to cover the entire range of hot water usage profile of a typical residential user (both middle and high-income families with 4 or 5 adults). Table

3.1 and Figure 3.4 show the hot water technologies and sensors used in the study. With a tank size of 150 L, for the geyser, split and integrated type ASHP water heaters, the experiment was designed to perform control draws of 50, 100 and 150 L throughout a full year (12 months' period: October 2015 to September 2016).

3.3 Materials and methods

A 150 L high-pressure geyser, I50 L integrated type ASHP water heater with a backup element and a 150 L split type ASHP water heater without an auxiliary backup were installed at the research center of the Fort Hare Institute of Technology, University of Fort Hare, Alice campus. A DAS was designed and built to accommodate the relevant sensors and transducers required to monitor the performances of the three hot water heating devices. The temperature sensors (12 bits S-TMB temperature sensors) were installed at the VCRC closed-loop circuits for both types of ASHP water heaters. Temperature sensors were also installed in proximity to the inlet and outlet of the geyser and the ASHP units. A flow meter (T-Minol 130 flow meter) was installed at the inlet of the split type ASHP unit. Power and energy meters (Quality track power meters) were connected to all the hot water heating devices. Ambient temperature and relative humidity sensor (12 bits S-THB ambient temperature and relative humidity sensor) enclosed within a solar radiation shield was installed in the vicinity of the hot water heating systems.

A full description of the sensors is contained in the published article titled "Design of a heat pump water heater performance monitoring system: to determine performance of a split type system" (Tangwe *et al*., 2016). All the recorded measurements obtained by the sensors and transducers were stored in the data loggers (U30-NRC Hobo data logger) (Tangwe *et al*., 2016). All the temperature sensors and the ambient temperature and relative humidity sensor were integrated with electronic input pulse adapters (S-UCC electronic input pulse adapters) to eliminate errors due to noise interference. The flow meter was incorporated with an electronic input pulse adapter (S-UCD electronic input pulse adapter). All the electronic input pulse adapters converted the analogue signals to digital. The U30-NRC Hobo data logger was powered by a 4.5 V DC battery.

One hundred (100)-ampere current transformers and voltage cables were installed on each of the power and energy meters to enable the measurement of the power factor, electrical demand and energy consumption for each of the hot water heating devices. The power and energy meter was endowed with an inbuilt data logging capability. The data logger was configured to log at every one-minute interval throughout the performance monitoring period of these hot water heating systems. Finally, it is of crucial importance to highlight that all the sensors and transducers used in the study were of class A and of very high accuracies such that their determined uncertainties were negligible to the actual measurements (Tangwe *et al*., 2014). Across, the different hot water drawn off scenarios, the hot water set point temperature was set at 55^{o}C and was viewed as the threshold for sanitary hot water temperature which also guaranteed the maximum COP of the ASHP water heater during operation.

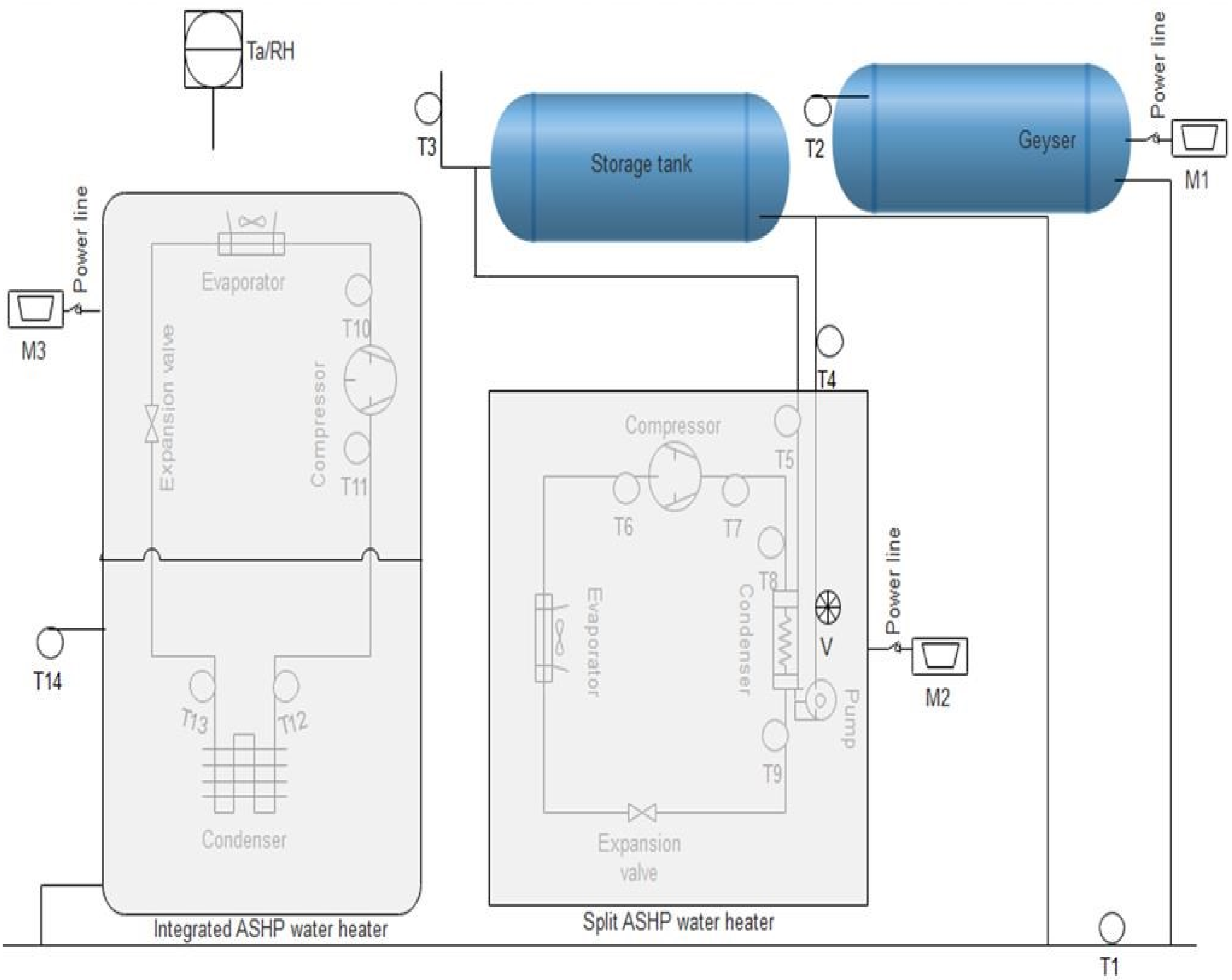
The experimental duration spans a full year to cater for seasonal changes (summer and winter periods). A full cycle of summer months and winter months were subjected to conducting the first-hour heating rating (150 L hot water drawn off), 100 L and finally 50 L of hot water drawn off from each of the storage tanks. This procedure was executed three times daily, and the sessions were designated as; the morning period, between 7:00 – 10:00, afternoon period, between 13:00 – 15:00 and evening period, between 18:00 – 21:00. The data stored in the data loggers were downloaded and analysed with the purpose to perform a quantitative and qualitative comparative analysis.

The analysed data for the input parameters (ambient temperature, relative humidity, temperatures of the refrigerant at the inlet and outlet of the compressor and condenser, and electrical energy consumption) and the desired output (COP) for the two types of ASHP water heaters were utilised in performing the statistical test (ReliefF test) to rank the predictors according to function as either primary or secondary factors. It was also used to verify if any significant mean difference occurred in the group COP means between the split and integrated type ASHP water heaters from the achievable COPs of the different hot water drawn off scenarios. The reliefF algorithm was used to rank the predictors into primary and secondary factors and to predict the contribution by weight of importance to the COP (Robnik-Sikonja and Kononenko, 2003). The multiple comparison procedure tests were used to test for any significant mean difference between the COP of the two types of ASHP water heaters under the described hot water drawn off scenarios (Hochberg and Tamhane, 1987). The multiple linear regression models were used to correlate the predictors to the desired response. These models were developed and built from a sample of the collected and analysed dataset known as the trained data. Subsequently, the models were validated using another sample of the dataset called the test data. In addition, the mathematical models were used to forecast the COP of the two types of ASHP water heaters for the different seasons taking into consideration, all the drawn off scenarios. A simulation application of the COP of the two types of ASHP water heaters was designed and built in the Simulink environment using the Simulink library (Chapoutot and Martel, 2008, Tangwe *et al*., 2014). Table 3.1 shows the materials used for the research.

Table 3.1: Sensors and hot water devices used in the research

Parameter	Equipment	Quantity
Temperature	12 bits S-TMB temperature Sensor	14
Volume	T-Minol 130 flow meter	1
Power factor, power and energy	Quality track Power Meter (Single phase)	3
100-ampere current transformer	Split core current transformer	3
Voltage cables	Live, Neutral and Earth voltage cables	3
Ambient temperature and relative humidity	12 bits S-THB ambient temperature and relative humidity	1
Electronic input pulse adapter	S-UCC electronic input pulse adapter	14
Electronic input pulse adapter	S-UCD electronic input pulse adapter	1
Data logger	U-30 NRC Hobo 15 channels data logger	1
System Enclosure	Water proof and radiation shield enclosure	2
Water calibrated drum	100 L water calibrated container	1
conventional water heating	Electric geyser	1
Split type ASHP water heating	Retrofit ASHP water heater with element disable	1
Integrated type ASHP water heating	Integrated ASHP water heater with electric backup	1

Figure 3.4 shows a layout of the block diagram of the hot water heating technologies and the metering sensors used in the study.



Key

Ta/RH = Ambient temperature & relative humidity sensor, T1 = In-line cold water temperature sensor T2 = Geyser hot water outlet temperature sensor, T3 = Split ASHP water heater hot water outlet temperature sensor,

T4 = Split ASHP inlet water temperature sensor, T5 = Split ASHP outlet water temperature sensor

T6 = Split ASHP compressor 's suction refrigerant temperature sensor , T7 = Split ASHP compressor 's discharge refrigerant temperature sensor , T8 = Split ASHP condenser's inlet refrigerant temperature sensor , T9 = Split ASHP condenser 's outlet refrigerant temperature sensor , T10 = Integrated ASHP compressor 's suction

refrigerant temperature sensor , T11 = Integrated ASHP compressor 's discharge refrigerant temperature sensor , T12 = Integrated ASHP condenser's inlet refrigerant temperature sensor , T13 = Integrated ASHP condenser 's outlet refrigerant temperature sensor, T14 =Integrated ASHP outlet water temperature sensor, V =Split ASHP

inlet flow meter, M1 = Geyser's power meter, M2= Split ASHP water heater's power meter, M3= Integrated ASHP water heater's power meter

Figure 3.4: Schematic of the full experimental set up

A full diagram of the installed geyser, split and integrated type ASHP water heaters as well as the DAS is shown in Figure 3.5.

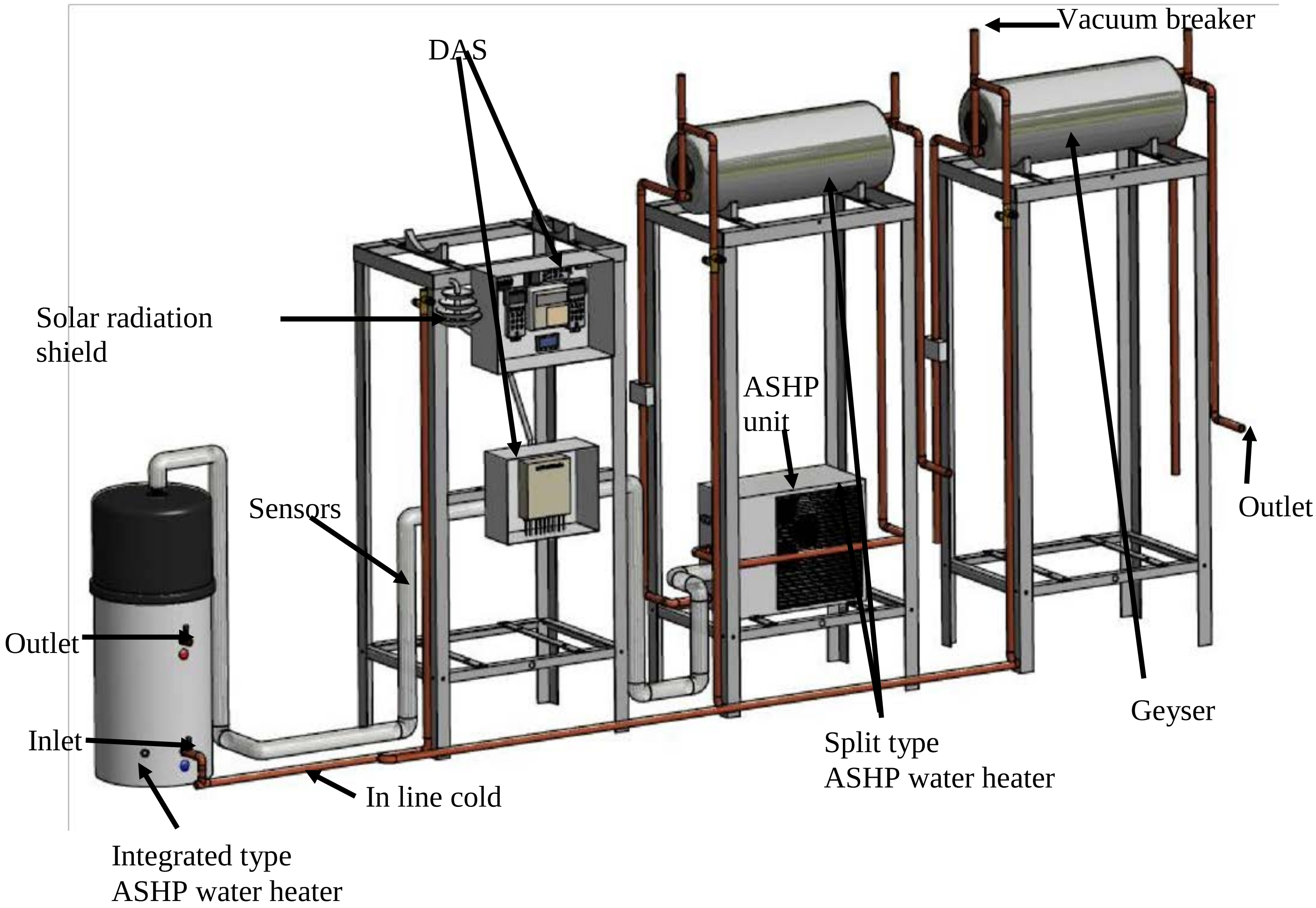


Figure 3.5: Schematic of installed hot water technologies and DAS

Also, Figure 3.6 shows the design and built DAS employed in the study for recording of the measured data from each of the installed sensors throughout the performance monitoring period.

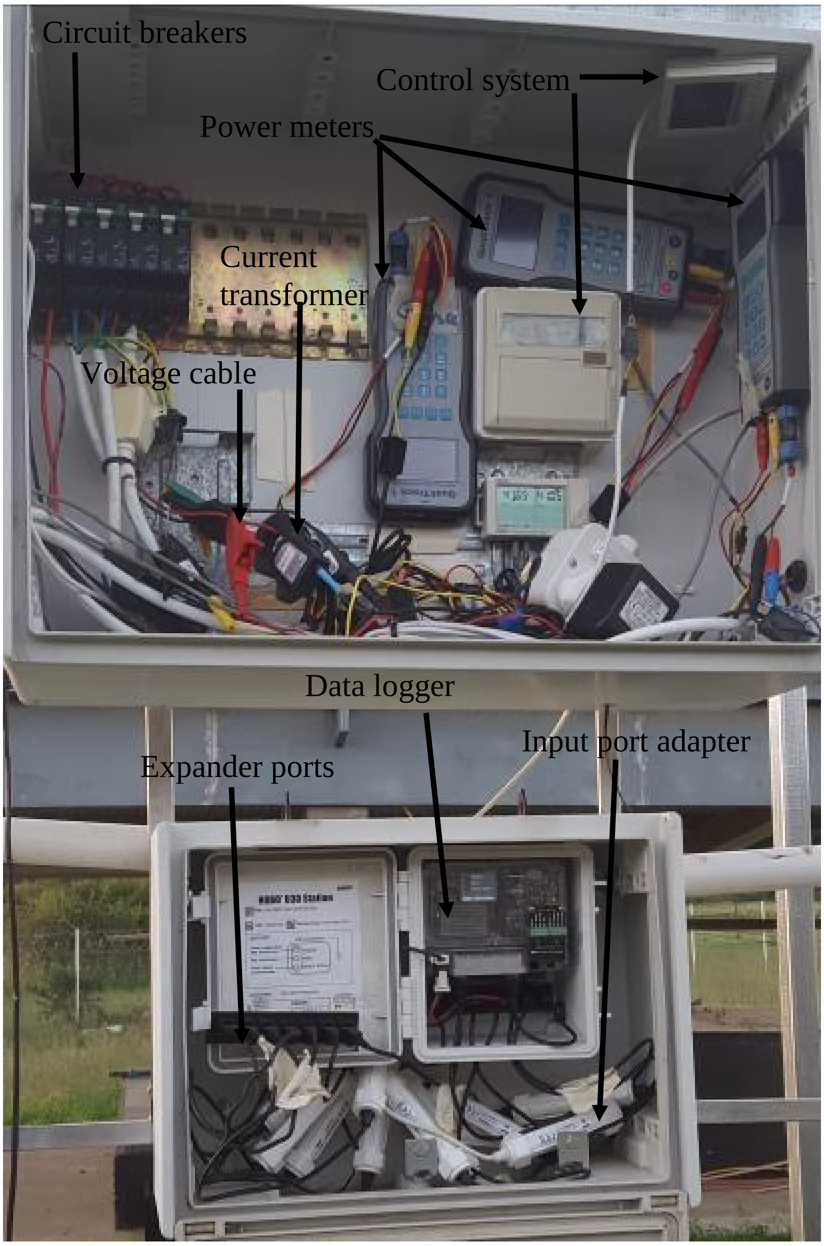


Figure 3.6: Design and built DAS used in the study

3.3.1 Data Collection metering sensors

The variables of interest in this experiment both environmental and at system level were recorded by the following transducers and sensors shown in Figure 3.4. These included:

i. Geyser power consumption measured by power meter M1
ii. Split type ASHP water heater power consumption measured by power meter M2 iii. Integrated type ASHP water heater power consumption measured by power meter M3 iv. In-line cold water temperature measured by sensor T1

v. Geyser hot water outlet temperature measured by sensor T2 vi. Split type ASHP water heater hot water outlet temperature measured by sensor T3 vii. Split type ASHP unit inlet water temperature measured by sensor T4 viii. Split type ASHP unit outlet water temperature measured by sensor T5 ix. Split type ASHP compressor suction refrigerant temperature measured by sensor T6

x. Split type ASHP compressor discharge refrigerant temperature measured by sensor T7 xi. Split type ASHP condenser inlet refrigerant temperature measured by sensor T8 xii. Split type ASHP condenser outlet refrigerant temperature measured by sensor T9 xiii. Integrated type ASHP compressor suction refrigerant temperature measured by sensor T10 xiv. Integrated type ASHP compressor discharge refrigerant temperature measured by sensor T11 xv. Integrated type ASHP condenser inlet refrigerant temperature measured by sensor T12

xvi. Integrated type ASHP condenser outlet refrigerant temperature measured by sensor T13 xvii. Integrated type ASHP water heater outlet water temperature measured by sensor T14 xviii. Split type ASHP unit inlet water flow rate measured by transducer V

For each variable, a specific sensor or transducer was assigned and placed at specific positions in order to capture its dynamics as shown in Figure 3.4.

3.4 Governing equations

The energy conversion in heat pump system is provided by the two laws of thermodynamics. The thermodynamic variables of most importance are enthropy (S), pressure (p), volume (V) and temperature (T). Since the refrigerant fluid exists in more than one state (liquid, gas), the moles (ni) of chemical (i) in phase (φ).

In general the first law can be expressed as in Equation 3.1 to 3.4.

$$\delta Q = dU + pdV \qquad (3.1)$$

Or in integral form

$$Q = U_2 - U_1 + \int_1^2 pdV \qquad (3.2)$$

Where the subscript $1 \rightarrow 2$ is omitted in the added heat Q. Since, the system also performs some electromechanical work, that work is expressed by adding δW to Equation 3.1 and its integral to Equation 3.2 to obtained Equations 3.3 and 3.4.

$$\delta Q = dU + pdV + \delta W \qquad (3.3)$$

$$Q = U_2 - U_1 + \int_1^2 pdV + \int_1^2 \delta W \qquad (3.4)$$

The second law of thermodynamics can be expressed in Equation 3.5.

$$dS = \frac{\delta Q}{T} \quad (3.5)$$

More generally, the process is irreversible and leads to Clausius inequality given in Equation 3.6.

$$dS > \frac{\delta Q}{T} \quad (3.6)$$

Thermal efficiency η of the heat pump engine can be expressed as shown in Equation 3.7.

$$\eta = \frac{\text{Work output}}{\text{Heat input}} = \frac{W}{Q_H} \quad (3.7)$$

The Equation 3.7 can be reduced to the Carnot efficiency given in Equation 3.8.

$$\eta_{Carnot} = 1 - \frac{T_C}{T_H} \quad (3.8)$$

The coefficient of performance of heat pumps (COP) is given in Equation 3.9 and 3.10

$$COP = \frac{\text{Useful output}}{\text{Required input}} \quad (3.9)$$

$$COP = \frac{Q_H}{W} = \frac{Q_H}{Q_H - Q_C} = \frac{1}{1 - \frac{Q_C}{Q_H}} \approx \frac{1}{1 - \frac{T_C}{T_H}} = \frac{1}{\eta_{Carnot}} \quad (3.10)$$

The multiple linear regression models used in the study exhibited typical workflows that involved: import the data, fit a regression, test its quality, modify it to improve the quality and share it. The following procedures were implemented in agreement to the workflows;

i. Step 1: Import the data into a data array.

ii. Step 2: Create a fitted model.

iii. Step 3: Locate and remove outliers. iv. Step 4: Simplify the model.

v. Predict the response.

The main concept of the multiple linear regression model is the fact that it included more than one independent variables. The principles of least squares and maximum likelihood are used for the estimation of parameters. We present the algebraic, geometric, and statistical aspects of the problem, each of which has an intuitive appeal. Let y denotes the dependent (or study) variable that is linearly related to K independent (or explanatory) variables $X_1, \ldots, X_K$ through the parameters $\beta_1, \ldots, \beta_K$ and we write as shown in Equation 3.11

$$y = X_1\beta_1 + \ldots + X_K\beta_K + e \qquad (3.11)$$

This is known as the multiple linear regression model. The parameters $\beta_1, \ldots, \beta_K$ are the regression coefficients associated with $X_1, \ldots, X_k$, respectively, and e is the difference between the observed and the fitted linear relationship. We have T sets of observations on y and $(X_1, \ldots, X_k)$, which we represent as shown in Equation 3.12.

⋯ Xk1

(y,X)= (⋮ ⋮ ⋮) (( )) (y1 X'1 ⋮ ⋮) y1 X11 = y,x 1

,⋯⋯x(k) =

(3.12) yT X1T⋯ XkT y1 X'T

The sets of multiple linear regression models employed in the study are given in Equations 3.13, 3.14 and 3.15, having the predictors as hot water set point temperature (Ts), ambient temperature (Ta), relative humidity (RH), electrical energy consumed (E), the temperature of refrigerant at the compressor inlet (Tcmi), the temperatureof the refrigerant at the compressor outlet (Tcmo), the temperature of the refrigerant at the condenser inlet (Tcni), the temperature of the refrigerant at the condenser outlet (Tcno).

COPmod □□0 □□1(Ts □Ta) □□2RH (3.13)

COPmod □ □0 □□1(TaRH) □□2E (3.14)

COPmod □ □0 □□1Ta □□2RH □□3(Tcmo □Tcmi) □□4(Tcni □Tcno) (3.15)

3.5 Uncertainty in the measurement and calculation

Measurement uncertainty is a non-negative induce value that characterises the dispersion of the values attributed to a measured quantity. All measurements are subject to uncertainty and a measurement result is satisfactory only if linked to uncertainty (Bich and Cox, 2006). The accuracy of an experimental data and the calculations using such data depended on the uncertainty in the sensors and transducers that formed the experimental set up (Coleman and Steel, 199).

The uncertainty of the temperature and relative humidity was ±0.2, the uncertainty of volume of water in liters was ±0.03. Similarly, the uncertainty of power measurement was ±0.005. The uncertainty of the COP derived from the calculation was ±0.203.

3.6 Reliability and Validity

These are two concepts that are of crucial relevance in defining and measuring bias and distortion. These are imperative to any scientific method and therefore need to be explained with high level of precision.

3.6.1 Reliability

Reliability is the degree to which a test consistently measures and repeatedly produces the same results for the same given input predictors or better still, the extent to which the same measurements can be obtained using the same instrument more than one time. In this regard, the collected dataset was analysed for consistency and the various temperature sensors, water flow meter and power meters showed similar readings with minimum variance for the same ambient conditions. But it should be alluded that there existed slight variance in the power consumption profiles for the various hot water heating technologies under similar ambient conditions. However, the variance could not be attributed to random errors as the air temperature is not the only crucial parameter influencing the system performance of geyser and ASHP water heaters. Thus, it could be an evidence of consistency in measurements. Reliability can also be ensured by minimising the sources of error, for example, data collection bias. Error minimisation during the data collection process was handled by restricting the access to the DAS only to two research fellows. The two research fellows had a good mastery of the operation of a domestic geyser and the ASHP water heater and the controlled hot water draws were carried out only during the prescribed morning, afternoon and evening periods.

3.6.2 Validity

The validity of an instrument is the degree to which an instrument measures what it is intended to measure. The validity of a research can also be explained as the extent to which requirements of a scientific research method have been followed during the process of generating research findings. As mentioned earlier, the controlled-simulated drawn off was implemented over 12 months from October 2015 to September 2016. These months were selected to represent effectively the weather profiles during the summer and winter seasons. Additionally, the experiment was conducted three times a day and mimic a typical morning, afternoon and evening hot water usage profile. The hot water drawn off were carried out within the specified time interval as outlined below;

- Morning: 7:00 – 10:00
- Afternoon: 13:00 – 15:00
- Evening: 18:00 – 21:00

Also, the ASHP unit was installed in an open space, although, the performance was not adversely impacted by shading.

Chapter Four

Potential viability of residential air source heat pump water heaters

Abstract

Inefficient geysers still stand as the most popular and conventional modes of hot water production in South Africa. The air source heat pump (ASHP) water heater is an energy-efficient technology for sanitary hot water production. This research employed the built data acquisition system (DAS) housing various temperature sensors, power meters, flow meter, ambient temperature and relative humidity sensor, to determine electrical energy consumption and useful thermal energy gained by the hot water in a geyser and storage tanks of residential ASHP water heaters. The load factors, average power and electrical energy consumptions for the 150 L high-pressure geyser, a 150 L split and integrated type ASHP water heaters were evaluated based on the controlled volume (150, 50 and 100 L) of daily hot water drawn off. The results depicted that the average electrical energy consumed and load factors of the summer months for the geyser, split and integrated type ASHP water heaters were 312.3, 111.7 and 121.1 kWh and 17.9, 10.2 and 16.7%, respectively. Finally, the simple payback period for both the split and integrated type ASHP water heaters were determined to be 3.9 years and 5.2 years, respectively.
Keywords: Air source heat pump, Geyser, Global warming potential, Load factor, Payback period

4.1 Introduction

Coal is the primary source of electricity generation in South Africa. The utilisation of coal for electricity generation from the thermal power plant is associated with greenhouse gasses emissions and global warming potential. The ASHP water heater is an energy-efficient device for sanitary hot water production. It is capable of using 1 unit of input electrical energy to provides 3 units of useful thermal output energy during vapour compression refrigeration cycles due to its coefficient of performance of 3 (Bodzin, 1997). The sanitary hot water is set at a threshold temperature of 55^{o}C to prevent the growth of the bacteria (*L*egionella).

The South Africa electricity supply utility (Eskom) is the sole supplier of electricity in South Africa with more than 90% of its generation coming from coal. The global warming potential because of greenhouse gases, primarily carbon dioxide, is 510 Mts of which 45% emanates from the generation of electricity from coal (Bryson, 2011). In South Africa, domestic electrical energy consumption is typically allocated according to the proportions of various residential energy devices (water heating; 43%, washing machine; 12.3%, stove; 10.2%, heater; 9.9%, fridge ;8.6% and small appliances; 11.2%) (www.Waterlite.co.za, 2013). It can be depicted without loss of generality but based on in-depth research that the contribution of electrical energy consumption by sanitary hot water production in the residential sector ranges from 40 to 60% depending on climatic conditions. Sanitary, water heating in the country is the largest residential consumer of electrical energy with up to

50% of the monthly consumption used for this purpose (Meyer and Tshimankinda, 1998).

It is worth mentioning that most hot water devices are the conventional heater (electric geysers) with an average energy factor of 0.92 (Haung and Lin, 1997). Interestingly, the ASHP water heater is a renewable energy device capable of heating water with the majority of the useful thermal output energy derived from ambient aero-thermal energy (Morrison *et al*., 2004). It can provide energy saving in the range from 50-70%, as the ASHP unit has a coefficient of performance ranging from 2 to 4 (Levins, 1982; Bodzin, 1997). The type of hot water storage tank for the ASHP water heater is a real challenge to the hot water temperature inside the tank. Heated water by ASHP of a similar volume is at a much higher temperature in a dual tank than a single tank system, but the thermal energy losses are lower for the latter (Hiller, 1996). An ASHP unit comprises of an evaporator, compressor, condenser and thermal expansion valve connected in a closed circuit by copper pipes with refrigerant as the heat transfer medium. The thermo-physical properties of the refrigerant are of priority in ASHP. Extensive research has exploited eco-friendly fluid, replacing R22 (Dichlorodifluoromethane) and R12 (Chlorodifluoromethane) because of their high ozone depletion potential (Zhang *et al*., 2012). The special characteristics that present the heat pump with excellent efficiency are its coefficient of performance (De Swardt and Meyer, 2001). In this regard, it is noteworthy that series of researchers have effectively evaluated heat pump water heater performance. Also, a dynamic model of an ASHP water heater was designed to achieve optimal energy management in a test room (Gao *et al*., 2009).

In a bid to avoid constraint on the national grid during peak hours, Eskom targeted rolling out more than 65,500 ASHP up to March 2013 under a residential rebate scheme to achieve a demand reduction of 54 MW (Ye and Zhang, 2012). The projected annual cost saving by the implementations of ASHP water heaters as retrofits to existing geysers were determined using the Eskom mega flex (flat rates) tariff (Van Eeden *et al*., 2016). Tangwe et al. (2014) demonstrated that the residential split type ASHP water heater is a reliable and renewable energy technology for sanitary hot water production with a viable techno-economic potential. The avoided annual water and carbon dioxide emissions reduction by the energy efficiency intervention whereby the ASHP water heaters are intended to replace the geysers were evaluated using the South African national energy regulator (NERSA) and Eskom statistical conversion factors (Van Eeden *et al*., 2016).

This research ultilised the designed and built DAS in Figure 3.6 to monitor the power and energy consumption of the installed hot water technologies shown in Figure 3.5 of Section 3.3 of chapter three wherby simulated controlled volume of hot water are drawn off from each cylinder. The three technologies were a convectional electrical heater (150 L geyser) and a 150 L split and integrated type ASHP water heaters installed at the research center of Fort Hare Institute of Technology, University of Fort Hare. The emphasis of the research was on the demand reduction, energy and cost savings achieved by the implementation of both types of ASHP water heaters as a replacement for an existing geyser.

4.1.1 Types and categories of ASHP units in South Africa

All 65,580 ASHP water heaters targeted in the rollout were classified into two broad categories; integrated (add-on) and split types (retrofit). Both exist in two modes; with an auxiliary element as a backup, or without a backup. The split type ASHP water heater could be grouped as single passed or recirculation. The single passed type ensures ASHP inlet water reach a set point temperature before exiting the ASHP outlet. The recirculation type is a multiple-passed system where ASHP inlet water undergoes continuous circulation before reaching set point temperature. Again, research conducted so far demonstrated that the integrated type has better and higher COP than the split type due to larger parasitic losses in the latter provided both types do not make use of a backup electric element. The split type ASHP water heaters are more reliable and stable. Figure 4.1 shows the distribution of the 65,580 residential ASHP intended to be rollout by Eskom according to the types and categories. It is of great importance to mention that the category 1 which constituted of small tank size systems made up 51,186 of the total systems. The total number of small tank size residential split and integrated type ASHP systems were 46,067 and 5,117 respectively. Furthermore, 55% of this number was allocated to the 150 L tank size ASHP systems. The 150 L ASHP systems were divided into residential split type ASHP units, and the integrated type ASHP water heaters and the allocated intended number to be installed were 25,337 and 2,815, respectively. The research focused on the 150 L ASHP systems because of the huge potential of demand and energy saving anticipated to be achievable by replacing or retrofitting the existing geysers with these energyefficient ASHP systems. Finally, the research also provides justifiable reasons for the viability of the ASHP water heaters base on the payback period.

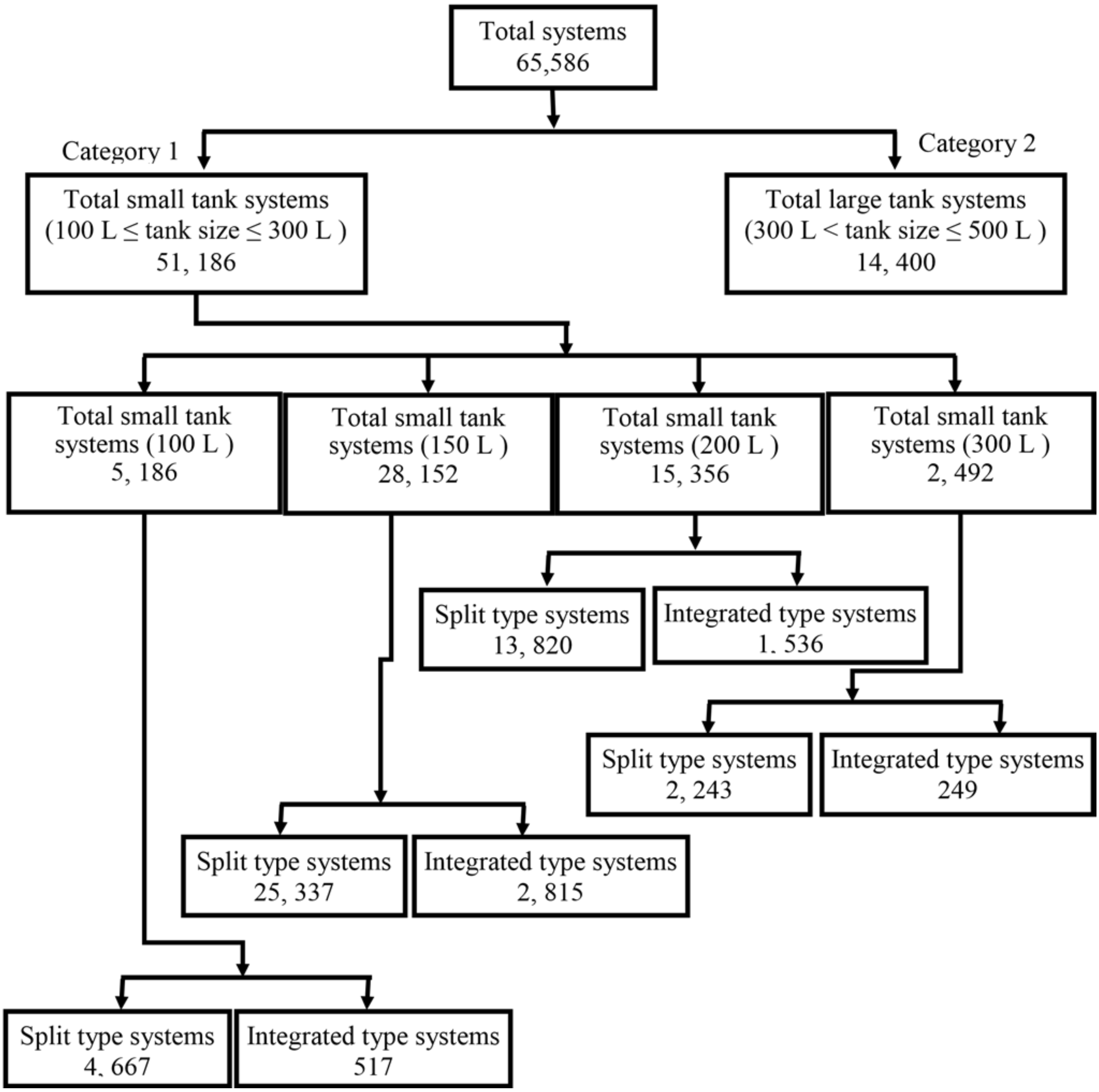


Figure 4.1: Allocations of the intended Eskom ASHP under the rebate scheme

4.2 Theory and calculations

The electrical energy consumed via heating of water in the geyser and the tanks of the ASHP water heaters was given by Equation 4.1.

$$E = Pt \quad (4.1)$$

Where

P = Active electrical power in kW

E = Electrical energy consumption in kWh

Power factor (PF) of the geyser and the ASHP water heaters was given by Equation 4.2.

$$PF = \frac{\text{Active power}}{\text{Apparent power}} \quad (4.2)$$

Where

Active power was measured in kW

Apparent power was measured in kVA

The coefficient of performance of the ASHP water heaters was given by Equation 4.3.

$$COP = \frac{Q_{out}}{E_{in}} \quad (4.3)$$

Where

Q_{out} = Output useful thermal energy gained

E_{in} = Input electrical energy

The energy factor or the performance energy factor of the heating technologies was given by Equation 4.4.

$$PEF = \frac{\sum Q}{\sum E} \quad (4.4)$$

Where

PEF = Performance energy factor

∑Q = total thermal energy over 24 hours

∑E = total electrical energy over 24 hours

The load factor of the hot water devices (high-pressure geyser and ASHP water heaters) was given by Equation 4.5.

$$LF = \frac{E}{P_{max} \times 24} \qquad (4.5)$$

Where

LF = Load factor

P_{max} = Maximum active power over a 24 hour period

The simple payback period of the ASHP water heaters was given by Equation 4.6.

$$SPP = \frac{\text{Capital cost}}{(\text{Annual energy saving}) \times (\text{tarrif rate})} \qquad (4.6)$$

Where

SPP = Simple payback period

4.3 Results and discussion

4.3.1 Typical summer and winter daily consumptions and ambient conditions

Table 4.1 shows the typical summer and winter daily average power (P), electrical energy consumptions (E), mean ambient temperature (T_a) and relative humidity (RH) for the 50, 100 and 150 L hot water drawn off (V_d) scenarios of each hot water technology.

Table 4.1: Typical daily consumptions and ambient conditions

System	Season	Time	V_d (L)	P (kW)	E (kWh)	Ta (°C)	RH (%)
Geyser	summer	morning	150	2.40	4.42	22.95	57.00
Integrated ASHP	summer	morning	150	0.93	1.40	22.97	57.10
Split ASHP	summer	morning	150	1.32	1.34	23.40	55.66
Geyser	summer	afternoon	50	2.40	1.78	29.60	36.88
Integrated ASHP	summer	afternoon	50	0.92	0.86	29.60	36.88
Split ASHP	summer	afternoon	50	1.13	0.81	29.53	36.73
Geyser	summer	evening	100	2.40	3.67	23.27	67.66
Integrated ASHP	summer	evening	100	0.91	1.57	23.27	67.66
Split ASHP	summer	evening	100	1.25	1.43	22.55	71.53
Geyser	winter	morning	150	2.40	6.06	16.14	78.85
Integrated ASHP	winter	morning	150	0.84	2.28	16.14	78.85
Split ASHP	winter	morning	150	1.25	1.94	17.29	72.26
Geyser	winter	afternoon	50	2.40	2.48	12.13	76.44
Integrated ASHP	winter	afternoon	50	0.91	1.56	12.13	76.44
Split ASHP	winter	afternoon	50	1.11	1.06	12.64	73.66
Geyser	winter	evening	100	2.40	4.64	15.07	84.04
Integrated ASHP	winter	evening	100	0.87	2.00	15.07	84.04
Split ASHP	winter	evening	100	1.25	1.54	13.55	88.58

Vd, Hot water drawn off; P, Average electrical power consumed; E, Average electrical energy consumed; Ta, Average ambient temperature; RH, Average relative humidity

It can be depicted from Table 4.1 that during the summer period, the average ambient temperature (25.24°C) was higher while the average relative humidity

(54.12%) was lower compared to the winter period with an average ambient temperature (14.46^{o}C) and relative humidity (79.24%) during the period of the heating cycles after the hot water was drawn off. The favourable average ambient condition during the summer season was responsible for the lesser electrical energy consumed by each of the hot water heating devices compared to the average winter performance. The typical average daily power consumption of the summer period for the geyser was 2.4 kW and was practically equal to that of the average daily power consumption for winter. The average daily energy consumption of the geyser was higher in the winter (13.59 kWh) compared to that of the summer period (10.31 kWh). This could be accounted for by the high rate of standby losses during the winter as opposed to the summer. The typical summer daily power consumption of the integrated and split type ASHP water heaters (0.90 and 1.27 kW) were higher than that of the winter power consumption (0.87 and 1.20 kW), respectively. It should be alluded that the typical average daily energy consumption of the split and integrated type ASHP water heaters for the summer period (3.69 and 3.99 kWh) were much lower to that of the winter period (4.66 and 6.00 kWh), respectively. This was due to better COP achieved in summer when compared to that in winter.

4.3.2 Daily energy consumptions, load factor and coefficient of performance

Table 4.2 shows the average daily energy consumptions (electrical energy (E) and thermal energy (Q)), load factors (LF) and the coefficient of performance (COP) of the three hot water heating devices. It should be noted that the total daily volume of hot water drawn off for both the summer and winter seasons was 300 L. The drawn off was controlled such that a volume of 150, 50 and

100 L were drawn in the morning, afternoon and evening period.

Table 4.2: Daily energy consumptions, load factors and COP

Systems	Season	P (kW)	E (kWh)	Q (kWh)	LF (%)	COP
Geyser	Summer	2.40	10.31	10.31	18.0	1.00
Integrated ASHP	Summer	0.92	3.99	10.31	16.6	2.69
Split ASHP	Summer	1.25	3.69	10.31	10.2	3.04
Geyser	Winter	2.40	13.59	13.59	23.6	1.00
Integrated ASHP	Winter	0.87	6.00	13.59	26.3	2.26
Split ASHP	Winter	1.20	4.66	13.59	13.8	2.86

P, Average electrical power consumed; P, Average electrical energy consumed; Q, Total thermal energy gained; LF, Load factor; COP, Coefficient of performance

Table 4.2 shows that in all the scenarios, the load factor (LF) of the split type ASHP water heater was better than that of the integrated type ASHP water heater and geyser. This can be accounted for by fact that the average daily electrical energy (E) of summer and winter periods were minimum for the split type ASHP water heater (3.69 and 4.66 kWh) compared to the integrated type ASHP water heater (3.99 and 6.00 kWh) and the geyser (10.31 and 13.59 kWh), respectively. The COP of the split type ASHP water heater had a better year-round performance of 2.95 as opposed to the COP of 2.48 for the integrated type ASHP water heater. The maximum power consumption during the heating cycles for the summer and winter periods as per the geyser was same (2.5 kW), and for the split type ASHP water heater was 1.50 and 1.20 kW, respectively. Also, the maximum power consumption in the summer and winter seasons for the integrated type ASHP water heater was 0.92 and 0.87 kW, respectively. These three hot water heating technologies had an excellent power factor of 0.98 all year-round.

4.3.3 Daily demand profiles of the different hot water technologies Figure 4.2 illustrates the subplots of the morning 150 L, afternoon 50 L and evening 100 L hot water drawn off of the average daily summer profiles. All the three subplots showed that the average power consumption of the geyser was highest (2.4 kW) in comparison to the split and integrated type ASHP water heaters of 1.25 and 0.92 kW, respectively. The total time used for the heating interval of the replacement water to set point temperature (55^{o}C) for the entire daily hot water drawn off were 310, 295 and 195 minutes for the geyser, integrated and split type ASHP water heaters, respectively. The average daily energy consumed was lowest for the split type ASHP water heater (3.69 kWh) by virtue of the least time required for the heating cycles. The split type ASHP water heater experience the least heating duration because of its excellent COP of 3.04 as opposed to 2.69 for the integrated type ASHP water heater.

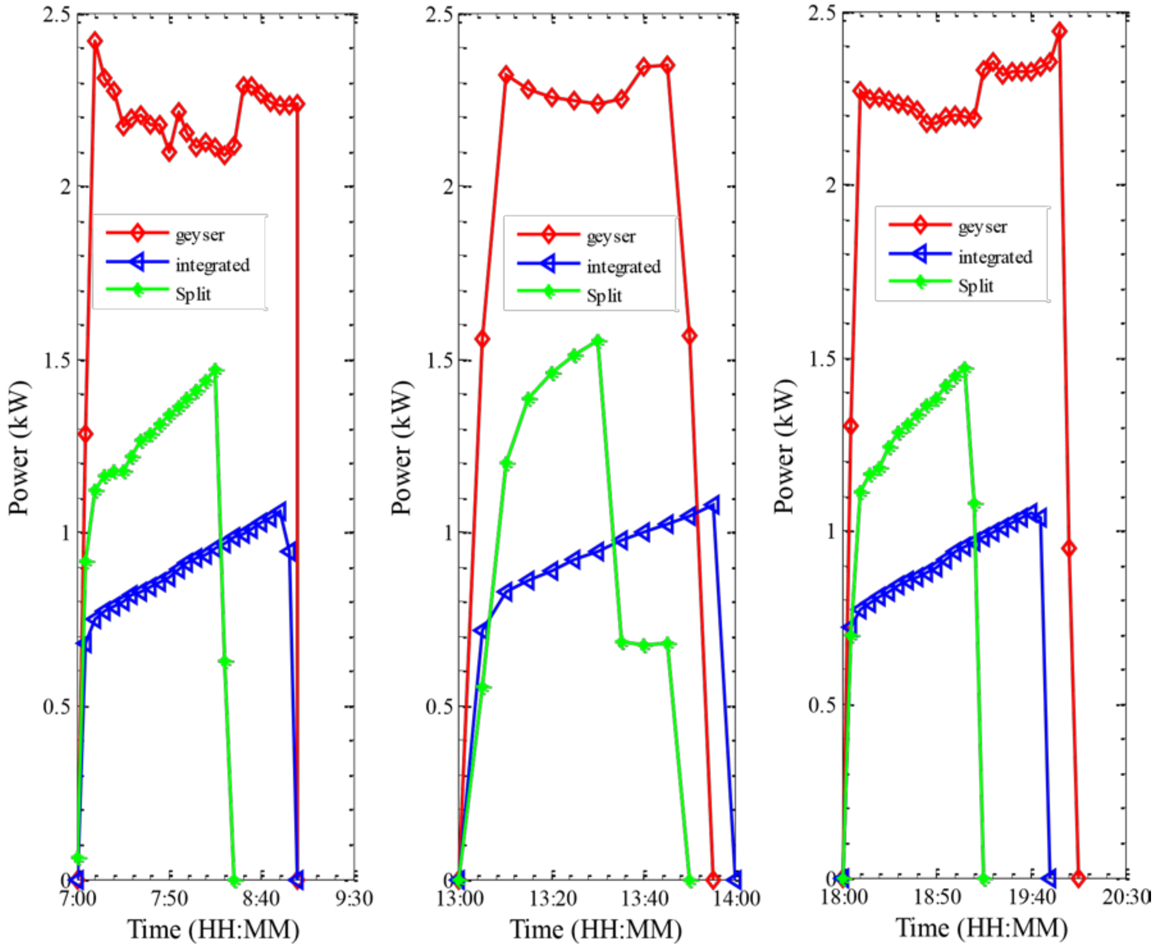


Figure 4.2: Summer daily subplots of power consume profiles of the three technologies

Figure 4.3 shows the subplots of the morning 150 L, afternoon 50 L and evening 100 L hot water drawn off of the average daily winter profiles. All the three subplots demonstrated that the average power consumption of the geyser was highest (about 2.4 kW) in contrast to that of the split and integrated type ASHP water heaters of values 1.2 and 0.87 kW, respectively. The total time used for the heating duration of the replacement water to set point temperature (55^{o}C) for the entire daily hot water drawn off were 510, 420 and 270 minutes for the geyser, integrated and split type ASHP water heaters, respectively. The average daily energy consumed was lowest for the split type ASHP water heater (4.66 kWh) owing to the least time required for the heating cycles. The least heating duration exhibited by the split type ASHP water heater was due to its better COP of 2.86 while that of the integrated type ASHP water heater was 2.26. The decrease in both COP of the split and integrated type ASHP water heaters during the winter season was due to a decreased in the ambient temperature and the initial cold water temperature.

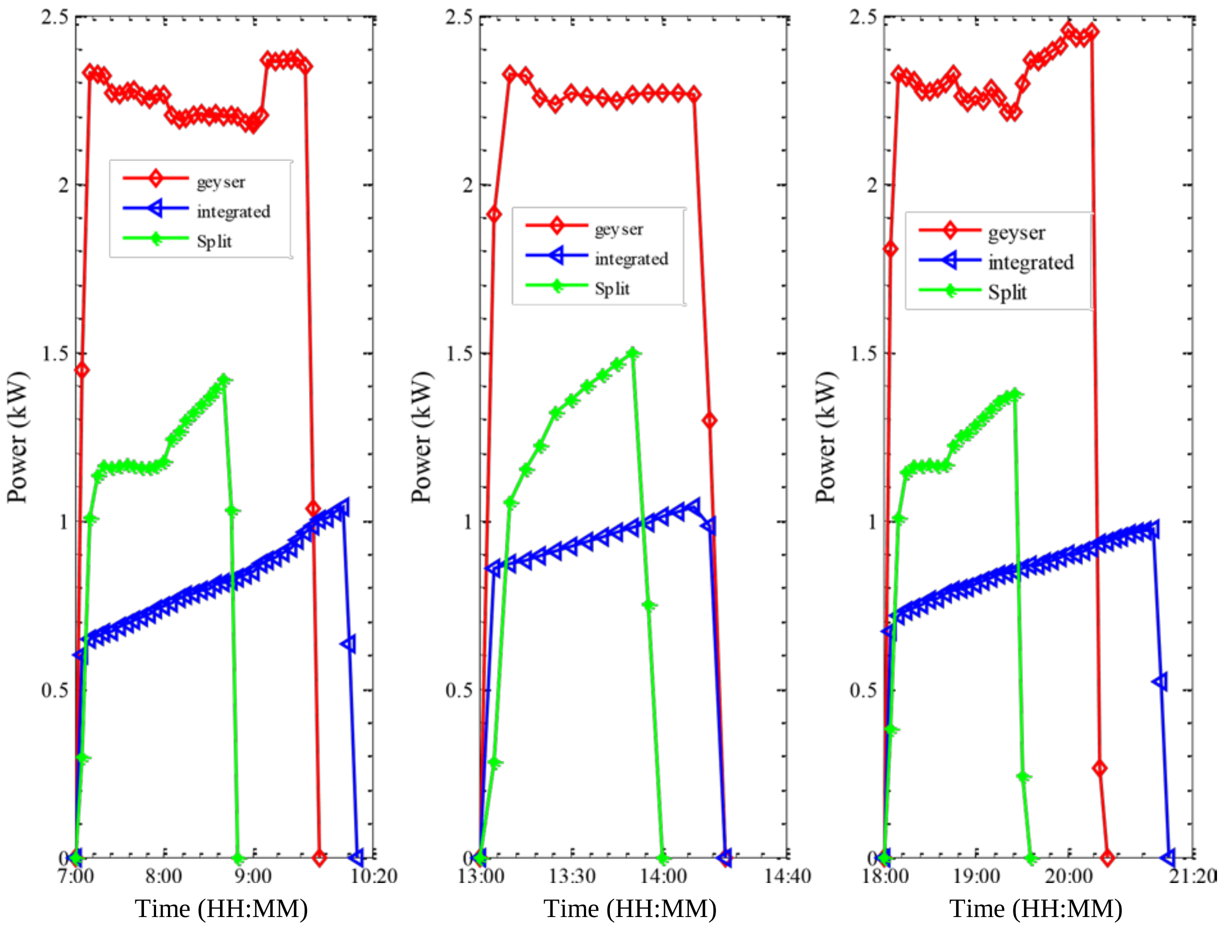


Figure 4.3: Winter daily subplots of power consume profiles of the three technologies

4.3.4 Annual electrical energy consumption and avoided water and gas emission

Figure 4.4 shows the monthly energy consumptions throughout the monitoring period (from October 2015 to September 2016) of the geyser, the split and integrated type ASHP water heaters. The monthly electrical energy consumption of the geyser ranged from 299.08 to 421.22 kWh. The minimum electrical energy consumption occurred during the summer season, and the maximum electrical energy consumed occurred during the winter periods. In addition, the total annual energy consumption of the geyser was 4.27 MWh. The minimum monthly energy consumption of the integrated and split type ASHP water heaters was 115.94 and 106.92 kWh, respectively and these also occurred during the summer month (February 2016). It can also be deduced from the bar plots that the maximum electrical energy consumption of the integrated and split type ASHP water heaters occurred during the winter month (May 2016) and was 186.14 and 144.61 kWh, respectively. The annual electrical energy consumption of the integrated and split type ASHP water heaters was 1.766 and 1.495 MWh. The annual electrical energy saving by replacing the geyser with the integrated type ASHP water heater would be 2.499 MWh. The annual electrical energy saved by retrofitting of the geyser with the split type ASHP unit would be 2.770 MWh. The projected combined annual electrical energy saving for the 25,337 of the 150 L split type ASHP water heaters and the 2,815 of the 150 L integrated type ASHP water heaters would be 77.22 GWh with a potential demand reduction of 33.94 MW. Clearly, the application of the emission factor of carbon dioxide of 1.07 kg and water saving factor of 1.46 kL, revealed that the avoided carbon dioxide reduction and water saving of the both integrated and split type ASHP water heaters would be 82620.79 kg of avoidance carbon dioxide emission and 112,734.90 kL of water saved from the power generation.

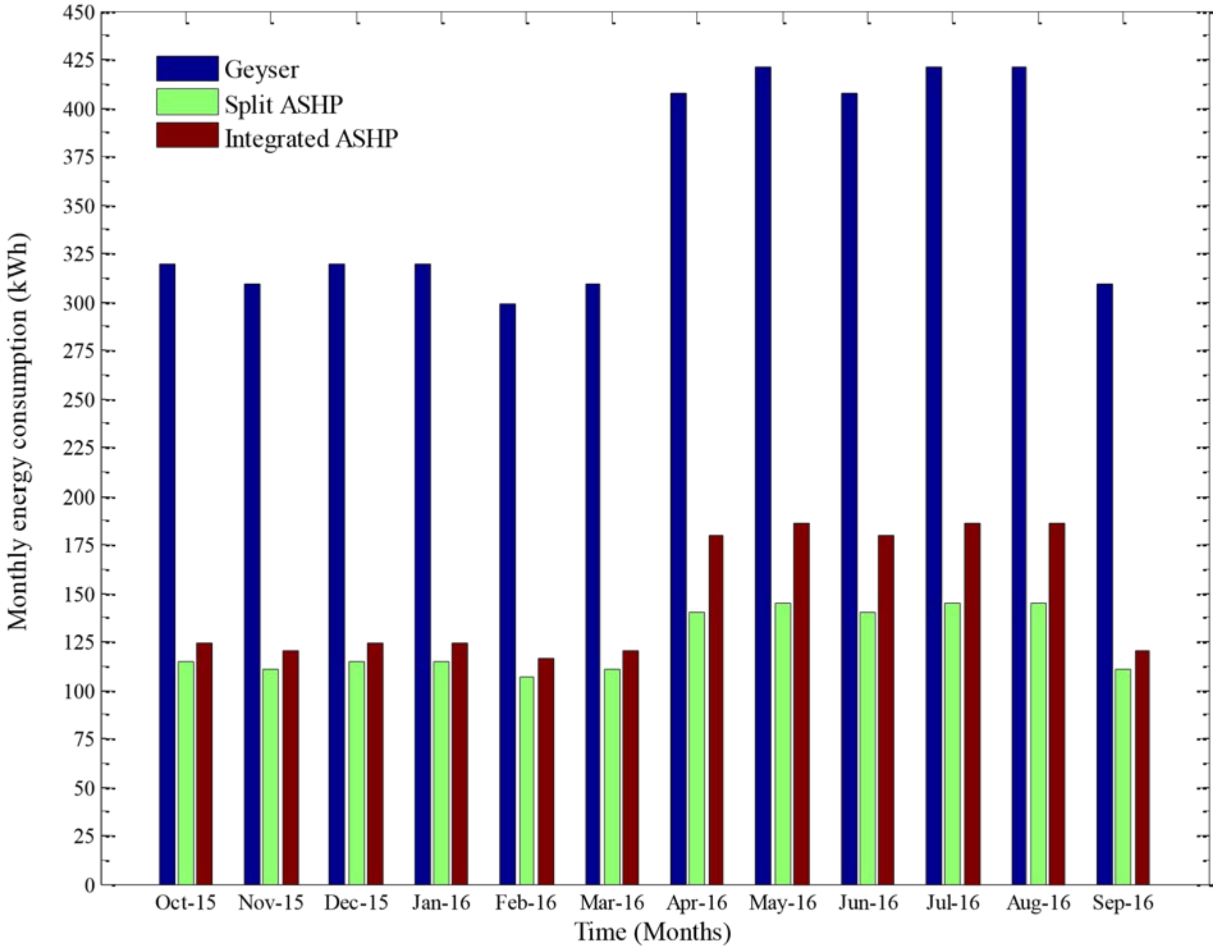
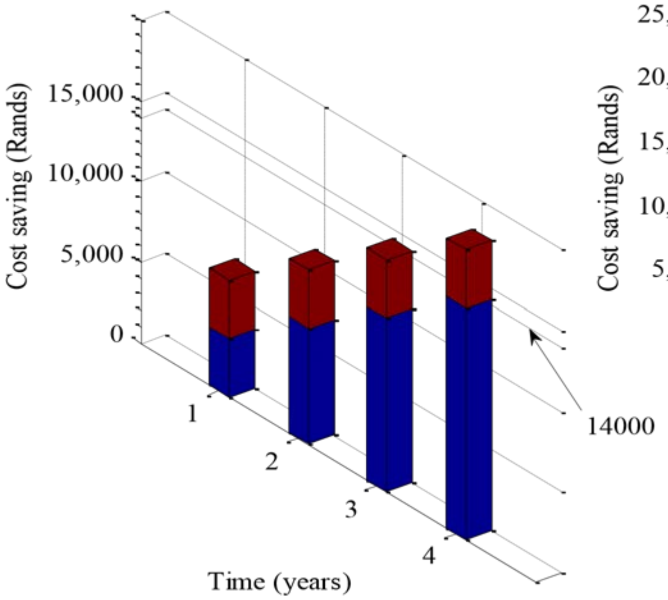
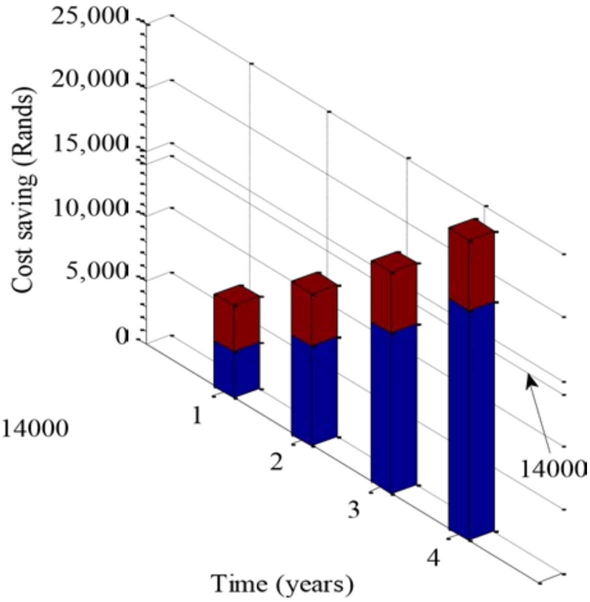


Figure 4.4: Bar plots of the three technologies monthly annual electrical energy consumed

4.3.5 Determination of the payback period of the residential ASHP water heaters

Vividly, the payback period is an economic analysis of a technology in a bid to assess its viability in retrospect to its capital cost and in some extent, the maintenance cost (Tangwe *et al*., 2014). A technology can be considered viable provided both the lifespan and payback period are favourable. The payback period could also be greatly impacted by the increase in electrical energy tariff over the years. The residential ASHP water heaters have a lifespan of close to 15 years with negligible once off cost of maintenance of the filters (strainer) and capacitors after 5 years or more depending on the water quality (Tangwe *et al*., 2016). The simple payback period for both types of ASHP water heaters was evaluated using the Eskom mega flex tariff of R 1.30 for 1 kWh of the electrical energy saved. The payback period for the split and integrated type ASHP water heaters were also calculated using an annual increase in the tariff rate of 15% as per Eskom projection (Eskom, 2012). It's very important to highlight that the capital cost of the split type ASHP unit and the integrated type ASHP water heater together with the installations was R 14,000.00 and R 17,000.00, respectively. The annual electrical energy saving by retrofitting the geyser with the split type ASHP unit was 2.77 MWh, and the cost saving was R 3,600.00. The simple payback period and the payback period inclusive of electricity tariff hikes was 3.9 and 3.3 years, respectively. Figure 4.5 demonstrates the simple payback period derived from the analytical calculation and the payback period due to tariff hikes from the computational economic analytic methodology. The individual stacks bar plots on Figure 4.5 shows both the cumulative annual cost saving (bottom – blue colour bar) and the consecutive yearly cost saving (top – brown colour bar). The cost-saving labelled by the text arrow (14000) corresponds to the capital cost of the residential split type ASHP unit.





20,000 Simple

paybackPayback with tarrif hikes

Figure 4.5: Payback analysis of the residential split type ASHP water heater

The annual electrical energy saving by replacing the geyser with the integrated type ASHP water heater was 2.50 MWh, and the cost saving was R 3,248.00. The simple payback period and the payback period taking electricity tariff hikes into consideration was 5.2 and 4.1 years, respectively. Figure 4.6 illustrates the simple payback period determined from the analytical calculation and the payback period due to tariff hikes by the computational approach analysis. The cost-saving labelled by the text arrow (17000) correspond to the capital cost of the residential integrated type ASHP water heater.

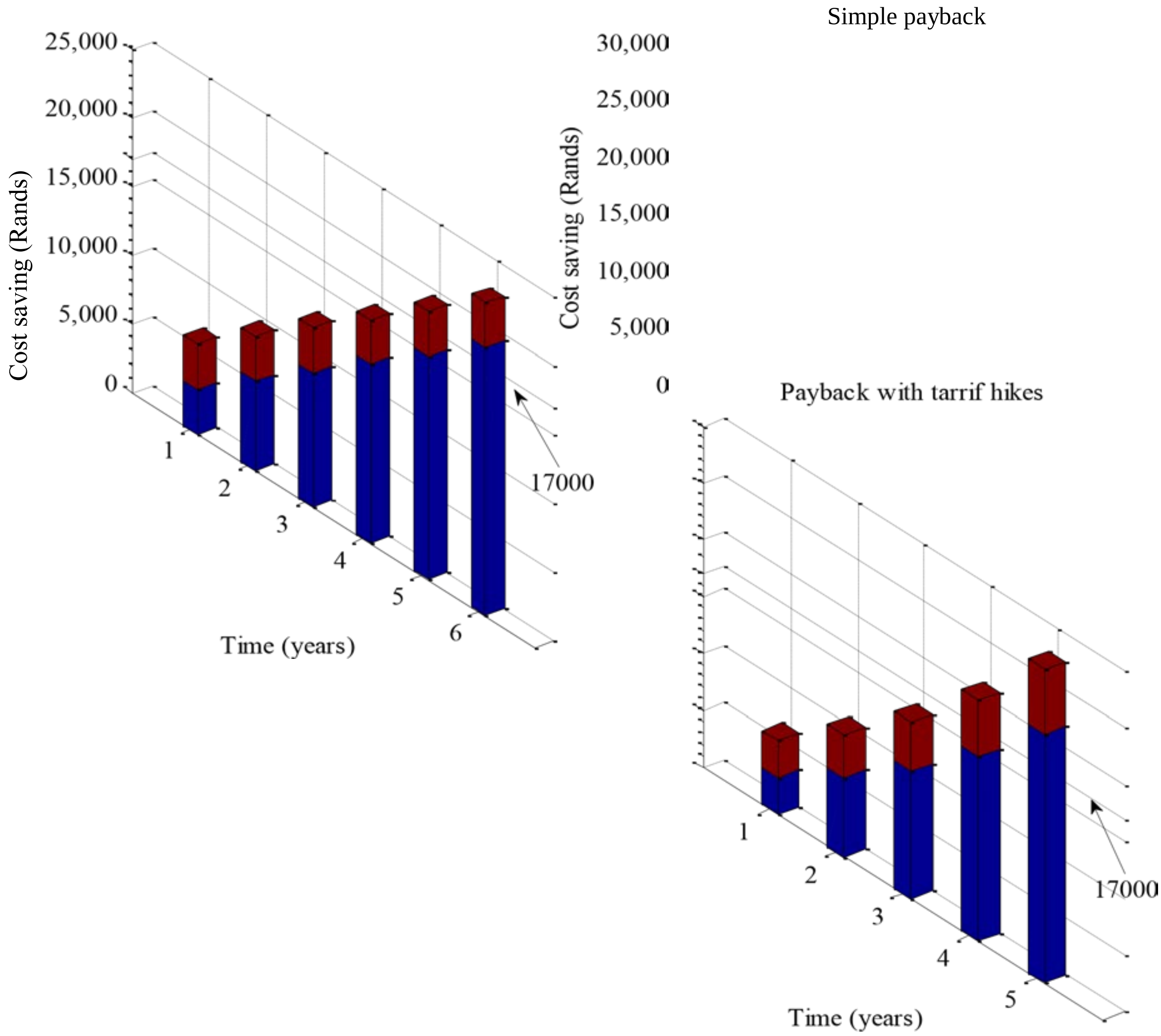


Figure 4.6: Payback analysis of the residential integrated type ASHP water heater

4.4 Summary

It could be affirmed that retrofitting or replacing of existing geyser with ASHP system (split or integrated type) can provide a permanent hot water solution on potential demand and energy reduction. Hence, contributing in minimising the constraint on the Eskom national grids. Apparently, both types of ASHP water heaters are viable technologies for sanitary hot water heating with a favourable payback period. The ASHP water heater could perform with a COP of over two all-round the year, but with a better performance during the summer period.

Although, the ASHP water heater COP was lower during the winter season, both the amount of electrical energy consumed and the projected electrical energy saving was higher during the winter period. Furthermore, the implementation of the ASHP water heaters could lead to both water saving and carbon dioxide emission reduction from the power generation and which can be determined from the achievable electrical energy saved. In addition to the energy and cost saving achieved by the retrofitting or replacing of geysers with ASHP systems, the technology also provides a very good power factor, load factor and favourable payback period.

Chapter Five

Impact of both standby losses and the isotherm blanket on hot water cylinders

Abstract

The performance of hot water heating devices is adversely impacted by the standby thermal energy losses of the systems. The study focused on monitoring the electrical energy consumed to compensate for the standby losses of three hot water cylinders without and with isotherm blankets. Accordingly, the analysis of the standby thermal energy losses was performed using 150 L highpressure geyser and 150 L split and integrated type air source heat pump (ASHP) water heaters without the withdrawal of hot water throughout the entire monitoring period. The results demonstrated that the average electrical energy consumed to compensate for the standby losses of the geyser, the split and integrated type ASHP water heaters without the isotherm blankets was 2.71, 1.33 and 0.94 kWh, respectively. The introduction of a 40 mm thick isotherm blanket on each of the hot water cylinders resulted in the electrical energy reduction by 18.5, 15.8 and 3.2% with respect to the geyser, the split and integrated type ASHP water heaters. A multiple

comparison test showed no significant difference in the mean of the group electrical energy consumed, that was required to compensate for the standby losses of both types of ASHP water heaters without and with the isotherm blankets installed.

Keywords: Air source heat pump (ASHP), Multiple comparison test, Isotherm blanket, and Standby thermal energy losses.

5.1 Introduction

Across the globe, sanitary hot water production constitutes a significant percentage of the monthly electrical energy consumption in the residential sector. Specifically, in South Africa, residential hot water heating can contribute to more than 50% of the monthly electrical energy utilisation (Meyer and Tshimankinda, 1998). An in-depth research conducted in South Africa to justify the electrical energy usage revealed that the hot water contribution in the residential sector was between 40% to 60% on a monthly average basis. Figure 5.1 demonstrates that 45% of the total energy consumption in a typical South African residence is from hot water heating (www. Waterlite.co.za, 2013).

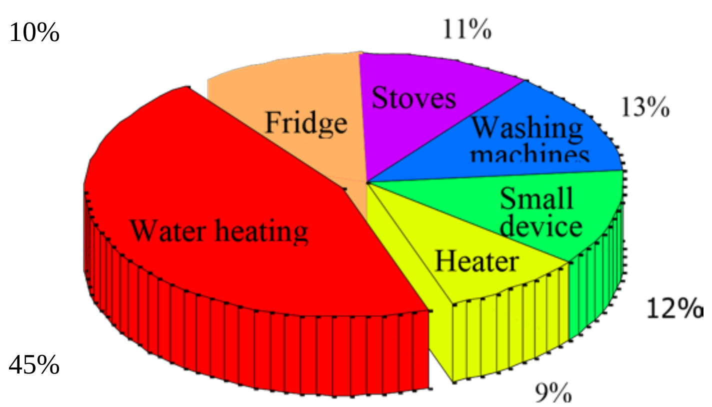


Figure 5.1: Percentage of energy consumption in a typical South African residence

It is worth mentioning that despite the daunting electrical energy consumed by hot water production, not all the thermal energy gained by the hot water is effectively utilised. There are always standby thermal energy losses which are responsible for 20% to 30% of the total thermal energy gained by hot water contained in a storage tank (Van Tonder and Holm, 2001).

Alternatively, the performance of the ASHP water heater is described by a unique factor known as the coefficient of performance (COP). The COP can range in value from 2 to 4 but it is crucial to emphasise that the COP depends on the primary (components used in the closed circuit design of the heat pump, volume of water heated, hot water set point temperature and mains supply cold water temperature) and secondary factors (ambient temperature and relative humidity) (Douglas, 2008; Baxter *et al*., 2005). Clearly, the COP could be defined as the ratio of the useful thermal energy gained when water is heated to the set point temperature and the electrical energy used by the system during the vapour compression refrigeration cycle. A salient and better understanding of refrigeration cycle of heat pump water heater was given by Ashdown (2004) and Sinha and Dysarkar, (2008). However, the performance can be severely affected by standby thermal energy losses (Douglas, 2008).

To the best of our knowledge, research has been conducted on standby thermal energy losses, emphatically on the geyser, solar water heater and the integrated heat pump water heater. More elaborately, standby thermal energy losses of the geyser were determined in the multi-level expert-modelling and evaluation of geyser load management opportunities in South Africa (Delport and Van Harmelen, 1999). Also, an experimental methodology was adopted to determine the standby thermal energy losses of the geyser (Beute, 1993) and an optimised geyser control switching method was used to minimise the standby thermal energy losses (Zhang and Xia, 2007). In addition, the standby losses of the solar water heater were computed via an experimental and numerical method (Kenjo *et al.,* 2007) while the standby thermal energy losses of the integrated heat pump water heater were also evaluated but on a laboratory benchmark study (Sparn *et al*., 2011). Thus, there is rear information with regards to the standby thermal energy losses of the split type ASHP water heater. Interestingly, the research focused on the analytical evaluation of electrical energy consumed to compensate for the standby thermal energy losses of a 150 L geyser, a 150 L split and integrated type ASHP water heaters under the scenarios described; wherein the hot water cylinders were without and with installed isotherm blankets as shown in appendix V. Finally, the multiple comparison test was employed to determine if a significant mean difference exists in the electrical energy consumed to compensate for the standby losses without and with the isotherm blankets installed on the different hot water heating technologies. The p-value was also used to test for significant difference in the mean electrical energy consumed under the two configurations of the different hot water heating devices. The p-value is a statistical technique that can be used to compare two or more groups means to test for a significant difference. If the p-value was very small (less than 0.05), there was a significant mean difference without requiring a further test. But if the p-value was very large (greater than 0.95), there was no significant mean difference (Tangwe *et al*., 2018).

5.1.1 Description of the installation of the three hot water heating devices

The both Figures in appendix V show the installation of the geyser, the split and integrated type ASHP water heaters without isotherm blankets and with isotherm blankets at the Fort Hare Institute of Technology Research Center,

University of Fort Hare, South Africa.
The geyser and ASHP water heaters were set to produce hot water at 55℃ with a temperature differential of 5℃. This implied that both the geyser and ASHP units started the heating cycles once the hot water inside the storage tank was 5℃ or more below the set point temperature. The systems were allowed to operate in an uninterrupted mode and without any hot water withdrawal from the 07 th of April 2015 to the 14 th of April 2015. The electrical energy consumptions, the ambient temperature and the relative humidity during the standby losses heating cycles of the systems were evaluated over four consecutive days with and without the isotherm blankets. The Section 3.3 in chapter three described the methodology employed in the research.

The research procedure was divided into two;

i. Monitoring of the performance of the electrical energy consumptions and the ambient weather conditions of the hot water heating technologies without the installed isotherm blankets.

ii. Monitoring of the performance of the electrical energy consumptions and the ambient weather conditions of the hot water heating technologies with the installed isotherm blankets.

5.2 Theory and calculations

The set of Equations 5.1 and 5.2 were used to calculate the electrical energy (kWh) and the electrical energy factor during the respecting heating modes involved in the standby thermal energy losses.

$$E = P \times t \qquad (5.1)$$

Where;

E = Electrical energy consumption of the hot water heating device in kWh

P = Power consumption of the hot water heating device in kW

t = Time intervals of 5 minutes

$$EF = \frac{\text{Electrical energy used by geyser over 24H}}{\text{Electrical energy used by ASHP over 24H}} \qquad (5.2)$$

Where;

EF = Electrical energy factor

The standby thermal energy losses were experimentally determined for each system, every 24 hour period.

5.3 Results and discussion

5.3.1 Performance analysis of hot water devices without the isotherm blankets

The electrical energy consumed to compensate for standby thermal energy losses, and the ambient temperature and the relative humidity for the different hot water heating devices were monitored from the 7 th to the 10 th of April 2015.

5.3.1.1 Performance analysis of geyser without the isotherm blanket The electrical power consumed, the ambient temperature and the relative humidity were averaged into 5-minute intervals throughout the standby thermal energy losses monitoring periods. Table 5.1 shows the electrical energy consumed and the ambient conditions over 24 hours of the specified days for which monitoring were conducted.

Table 5.1: Electrical and ambient evaluations for geyser without blanket

Day	Average relative humidity (%)	Average energy consumed (kWh)	Total electrical cycles	No of ambient temperature(℃)
07 April	13.96	81.21	2.81	9
08 April	16.04	82.74	2.67	8
09 April	18.20	74.31	2.47	7
10 April	14.38	73.96	2.91	11
Average	15.64	78.05	2.71	9

It can be shown that the heating cycles per day due to standby thermal energy losses ranged from 7 to 11 and the electrical energy consumed to compensate for these standby losses were between 2.47 and 2.91 kWh. It was observed that the lowest heating cycles per day (7) corresponded to the least electrical energy consumed (2.47 kWh) and the average ambient temperature was maximum (18.2℃). Overall, the average electrical energy consumed was 2.71 kWh and the average ambient temperature and relative humidity were 15.64℃ and 78.05%, respectively. The geyser daily average heating cycles was 9 under the scenario without the installation of the isotherm blanket.

5.3.1.2 Analysis of the split type ASHP water heater without the isotherm blanket

The electrical power consumed, the ambient temperature and the relative humidity were averaged into 5 minute intervals throughout the standby thermal energy losses monitoring periods. Table 5.2 represents the electrical energy consumed and the ambient conditions over 24 hours of the specified days for which monitoring were conducted.

Table 5.2: Electrical and ambient evaluations for split system without blanket

Day	Average ambient temperature (ºC)	Average relative humidity (%)	Total electrical energy consumed (kWh)	No of cycles
07 April	12.85	88.65	1.54	3
08 April	16.35	82.48	1.40	3
09 April	14.38	86.61	0.97	2
10 April	15.69	68.99	1.42	3
Average	14.82	81.72	1.33	3

From Table 5.2, it is observed that the standby thermal energy losses heating cycles per day ranged from 2 to 3 and the electrical energy consumed to compensate for the standby losses was between 0.97 and 1.54 kWh. Also, it was deduced that the lowest heating cycles per day (2) also corresponded to the least electrical energy consumed (0.97 kWh). In a nutshell, the average electrical energy consumed per day was 1.33 kWh, and the average ambient temperature and relative humidity were 14.82℃ and 81.72%, respectively. The average electrical energy factor was 2.04. The average heating cycles per day of the split type ASHP water heater without the installation of the isotherm blanket was 3.

5.3.1.3 Analysis of integrated type ASHP system without the isotherm blanket

The electrical power consumed, the ambient temperature and the relative humidity were averaged into 5 minute intervals throughout the standby thermal energy losses monitoring periods. Table 5.3 shows the electrical energy consumed and the ambient conditions over 24 hours of the specified days for which monitoring were conducted.

Table 5.3: Electrical and ambient vales for integrated ASHP without blanket

Day	Average ambient temperature (℃)	Average relative humidity (%)	Total electrical energy consumed (kWh)	No of cycles
07 April	13.77	84.74	1.33	2
08 April	17.05	83.58	0.61	1
09 April	14.48	86.64	1.24	2
10 April	20.82	49.30	0.58	1
Average	16.53	76.07	0.94	1-2

It can be alluded in Table 5.3 that the standby thermal energy losses heating cycles per day ranged from 1 to 2 and the electrical energy consumed to compensate for the losses was between 0.58 and 1.33 kWh. It was justified that the lowest heating cycles per day (1) were equal to the least electrical energy consumed (0.58 or 0.61 kWh). Summarily, the average electrical energy consumed per day was 0.94 kWh and the average ambient temperature and relative humidity were 16.53℃ and 76.07%, respectively. The average electrical energy factor was 2.88. The average heating cycles per day of the integrated type ASHP water heater without the installation of the isotherm blanket can either be 1 or 2.

5.3.2 Performance analysis of hot water devices with the isotherm blankets

In order to compare to the counterpart technologies without the isotherm blanket, the electrical energy consumptions to compensate for the standby thermal energy losses and the ambient temperature and the relative humidity of the different hot water heating devices with installed isotherm blankets were equally monitored from the 11 th to the 14 th of April 2015.

5.3.2.1 Performance analysis of geyser with the isotherm blanket

The electrical power consumed, the ambient temperature and the relative humidity were averaged into 5 minute intervals throughout the monitoring periods of the standby thermal energy losses. Table 5.4 shows the electrical energy consumptions and the ambient conditions over 24 hours of performance monitoring with respect to the specified days.

Table 5.4: Electrical and ambient evaluations for geyser with blanket

Day	Average ambient temperature (℃)	Average relative humidity (%)	Total electrical energy consumed (kWh)	No of cycles
13 April	16.49	85.74	2.24	8
14 April	19.74	66.28	2.14	6
15 April	14.86	77.70	2.02	6
16 April	12.45	74.96	2.25	7
Average	15.89	76.17	2.18	7

As presented in Table 5.4, the standby thermal energy losses heating cycles per day ranged from 6 to 8 and the electrical energy consumed to compensate for the losses was between 2.02 and 2.25 kWh. It was observed that the lowest heating cycles per day (6) corresponded to the least electrical energy consumed (2.02 and 2.14 kWh). The average electrical energy consumed throughout the monitored period was further reduced to 2.18 kWh, and the average ambient temperature and relative humidity were 15.89℃ and 76.17 %, respectively. The average heating cycles per day of the geyser with the isotherm blanket was also reduced to 7.

5.3.2.2 Analysis of the split type ASHP water heater with the isotherm blanket

The electrical power consumed, the ambient temperature and the relative humidity were averaged into 5 minutes intervals throughout the monitoring periods evaluating the standby thermal energy losses. Table 5.5 provides the electrical energy consumptions and the ambient conditions over 24 hours of performance monitoring with respect to the specified days.

Table 5.5: Electrical and ambient evaluations for split type ASHP with blanket

Day	Average ambient temperature (℃)	Average relative humidity (%)	Total electrical energy consumed (kWh)	No of cycles
	15.51	88.35	0.92	2
	15.97	78.70	1.20	3
	16.03	71.58	0.85	2
16 April	13.30	73.94	1.48	3
13 April				
14 April				
15 April				
Average	15.39	78.08	1.12	2-3

Table 5.5 shows the heating cycles per day due to the standby thermal energy losses and ranged from 2 to 3. The electrical energy consumed to compensate for the losses was between 0.85 and 1.48 kWh. It was established that the lowest heating cycles per day (2) corresponded to the least electrical energy consumed (0.85 or 0.92 kWh). The average electrical energy consumed was 1.12 kWh, and the average ambient temperature and relative humidity were

15.39℃ and 78.08%, respectively. The average electrical energy factor was 1.95. The average heating cycles per day of the split type ASHP water heater with the isotherm blanket installed was either 2 or 3.

5.3.2.3 Analysis of integrated type ASHP system with the isotherm blanket

The electrical power consumed, the ambient temperature and the relative humidity were averaged into [1] minute intervals throughout the monitoring periods of the standby thermal energy losses. Table 5.6 shows the electrical energy consumptions and the ambient conditions over 24 hours of performance monitoring with respect to the specified days.

Table 5.6: Electrical and ambient evaluations for integrated ASHP with blanket

Day	Average ambient temperature (℃)	Average relative humidity (%)	Total electrical energy consumed (kWh)	No of cycles
13 April	16.77	87.09	1.15	2
14 April	11.85	91.82	0.68	1
15 April	13.03	85.00	1.31	2
16 April	28.28	17.28	0.51	1
Average	17.48	70.30	0.91	1-2

It can be delineated from Table 5.6 that the heating cycles per day as reason of the standby thermal energy losses ranged from 1 to 2 and the electrical energy consumed to compensate for the losses was between 0.51 and 1.31 kWh. It was justified that the lowest heating cycles per day (1) were equal to

[1] **.3.3 Box plots comparisons between the two configurations**

The box plots analysis was used to compare the standby losses of each hot water heating devices without and with the isotherm blanket based on the

the least electrical energy consumed (0.51 or 0.68 kWh). In general, the average electrical energy consumed was 0.91 kWh and the average ambient temperature and relative humidity were 17.48℃ and 70.30%, respectively. The average electrical energy factor was 2.40. The average heating cycles per day of the integrated type ASHP water heater with the isotherm blanket installed can either be 1 or 2.

electrical energy consumptions per day over the four successive days of monitoring.

5.3.3.1 Box plot analysis of standby losses of the geyser

The electrical energy consumed over 24 hour periods based on the specified monitoring days for both configurations (without the installed isotherm blanket and with the installed isotherm blanket) were compared using the box plots. Figure 5.2 shows the box plots of the daily electrical energy that were required to compensate for the standby losses of the two geyser configurations over the entire monitoring period.

It can be depicted from the Figure 5.2 that the electrical energy distributions of the geyser under both configurations (without an isotherm blanket and with an isotherm blanket) were normally distributed. The mean daily electrical energy to compensate for the standby thermal energy losses of the geyser without the isotherm blanket and with the isotherm blanket was 2.71 and 2.18 kWh, respectively. The average relative humidity and ambient temperature during both monitoring periods showed no significant difference. The reduction of electrical energy due to the installation of the isotherm blanket was 18.5%.

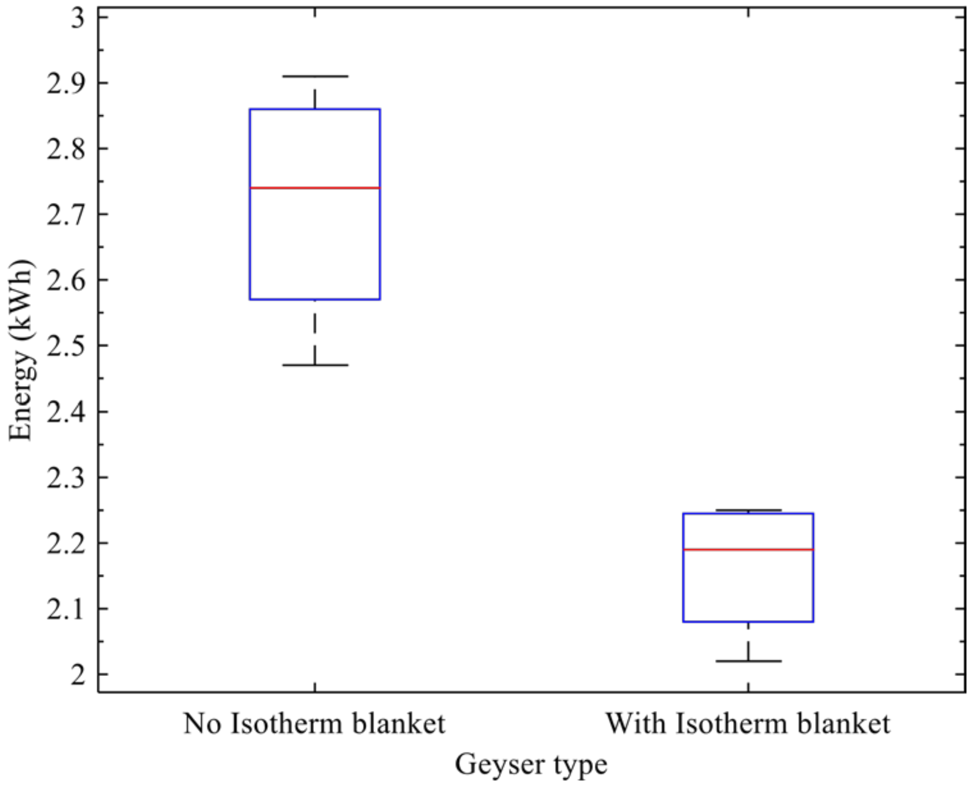


Figure 5.2: Box plots of the two geyser configurations

The y-axis range of the box plot in Figure 5.2 shows the average daily standby thermal energy losses, while the horizontal red line in the box plots corresponded to the overall average daily standby losses for the entire monitoring period. The box plots with their lower and upper horizontal bars show the distributions of average daily electrical energy consumed to compensate the standby losses (usually in the form of normal distribution).

5.3.3.2 Box plot analysis of standby losses of the split type ASHP water heater

The electrical energy consumed over a 24 hour period based on the specified monitoring days for both configurations (without the installed isotherm blanket and with the installed isotherm blanket) was compared using the box plots.

Figure 5.3 illustrates the box plots of the daily electrical energy required to compensate for the standby thermal energy losses of the split type ASHP water heater under the two configurations over the entire monitoring duration. It can be affirmed from the Figure 5.3 that the electrical energy distributions of the split type ASHP water heater without the isotherm blanket and with the isotherm blanket were normally distributed. The mean electrical energy to compensate for the standby thermal energy losses of the split type ASHP water heater without the isotherm blanket and with the isotherm blanket was 1.33 and 1.12 kWh, respectively. The average relative humidity and ambient temperature during both monitoring periods exhibited no significant difference. The reduction of electrical energy due to the installation of the isotherm blanket was 15.8%.

The y-axis range of the box plot in Figure 5.3 shows the average daily standby thermal energy losses, while the horizontal red line in the box plots corresponded to the overall average daily standby losses for the entire monitoring period. The box plots of both configurations of the split type ASHP water heaters demonstrated a skew normal distribution with most of the average daily electrical energy consumed above the box plot normal mean probably due to the prevailing ambient conditions that influences the daily standby losses. Hence, in the configuration with no isotherm blanket installed in the split type ASHP water heater the daily electrical energy consumed are more above the normal mean of the box plot. Also, in the configuration with the isotherm blanket installed in the split type ASHP water heater, the daily electrical energy consumed was more below the normal mean of the box plot.

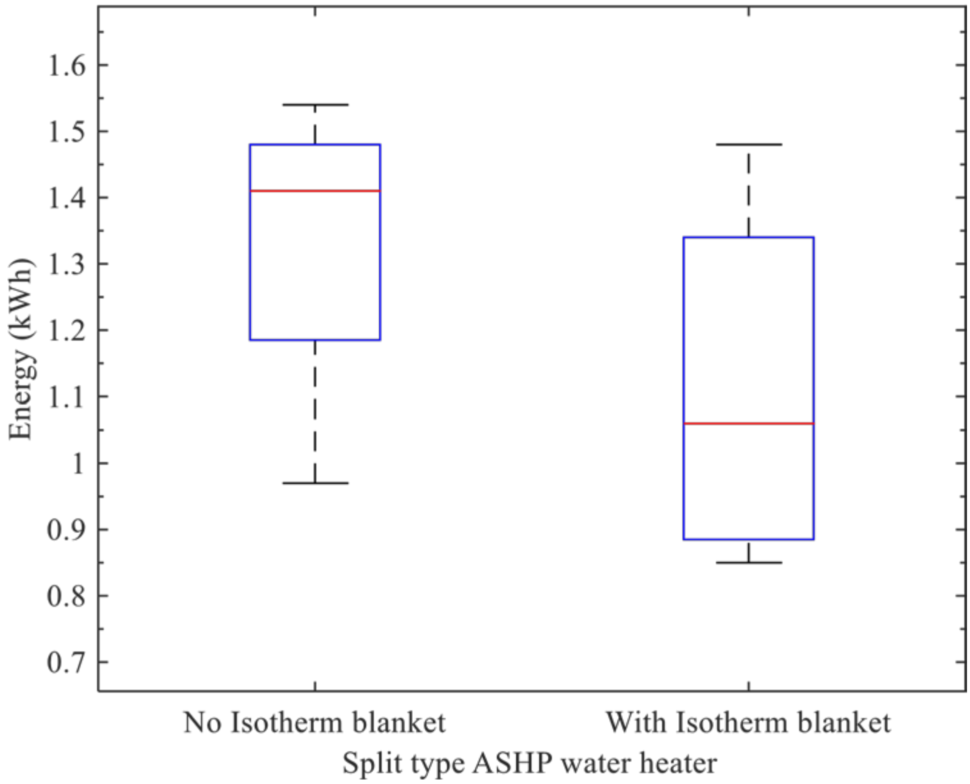


Figure 5.3: Box plots of the two split type ASHP configurations

5.3.3.3 Box plot analysis of standby losses of the integrated type ASHP system

The electrical energy consumed over a 24 hour period of performance monitoring for the specified days under both configurations of without the installed isotherm blanket and with the installed isotherm blanket was compared using the box plots. Figure 5.4 provides the box plots of the daily electrical energy required to compensate for the standby thermal energy losses of the integrated type ASHP water heater under the different configurations during the monitoring periods.

It can be depicted from Figure 5.4 that both electrical energy distributions for the integrated type ASHP water heater without the isotherm blanket and with the isotherm blanket were normally distributed. The mean electrical energy to compensate for the standby losses of the integrated type ASHP water heater without the isotherm blanket and with the isotherm blanket was 0.94 and 0.92 kWh, respectively. The average relative humidity and ambient temperature during both monitoring periods revealed no significant difference. The reduction of electrical energy due to the installation of the isotherm blanket was 3.2%. This negligible impact on electrical energy consumption was attributed to the orientation (vertical position) and the initial double outer walls constructed in the integrated type ASHP system to eliminate thermal energy losses at the storage tank.

The y-axis range of the box plot in Figure 5.4 shows the average daily standby thermal energy losses, while the horizontal red line in the box plots corresponded to the overall average daily standby losses for the entire monitoring period. The box plots of both configurations of the integrated type ASHP water heaters demonstrated an almost perfectly normal distribution with most of the average daily electrical energy consumed within the box plot normal mean probably due to the addition double insulations on the tank which prevented the prevailing ambient conditions from influences the daily standby losses. Hence, in the configuration with no isotherm blanket installed and with isotherm blanket installed in the integrated type ASHP water heater showed that the daily electrical energy consumed are within the normal mean of the box plot.

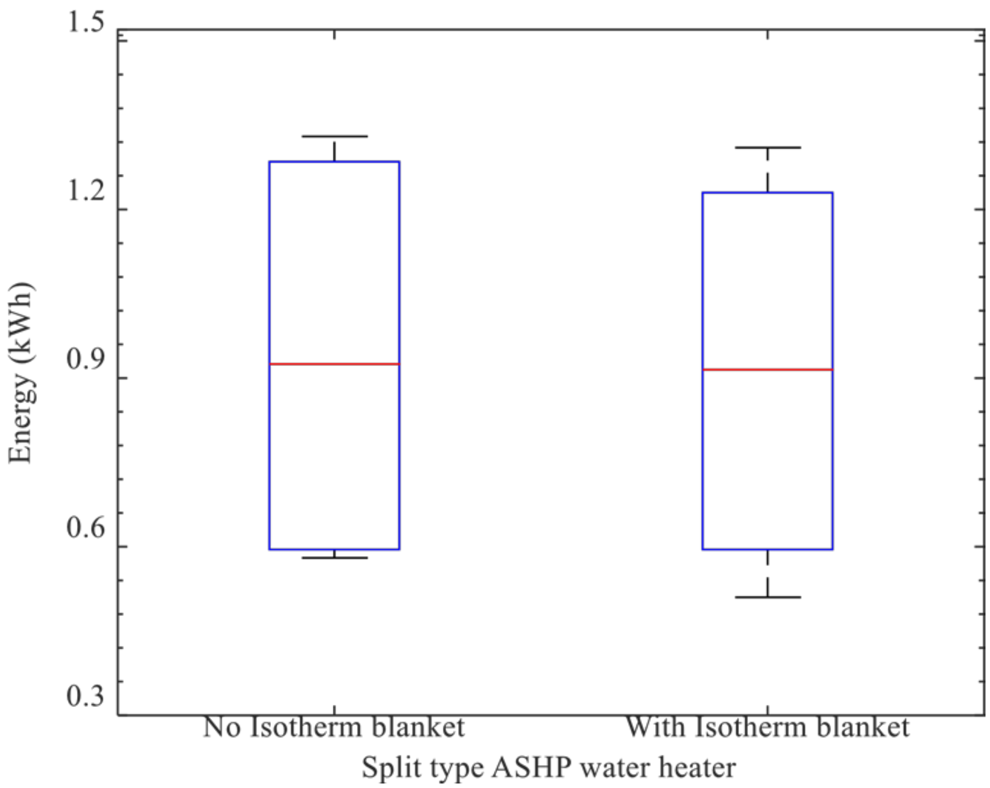


Figure 5.4: Box plots of the two integrated type ASHP configurations

5.3.4 Multiple comparison test between the two configurations

The multiple comparison statistical tests were used to determine if there was a significant mean difference of electrical energy consumptions due to the standby thermal energy losses of each hot water heating device without and with the isotherm blanket.

5.3.4.1 Multiple comparison test of standby losses of the geyser The electrical energy consumed over a 24 hour period for both configurations without the installed isotherm blanket and with the installed isotherm blanket was compared using the multiple comparison test over the four successive days of monitoring. Figure 5.5 shows the multiple comparison plots of the daily electrical energy to compensate for the standby thermal energy losses of the geyser under the two configurations. It can be deduced from the Figure 5.5 that the mean daily electrical energy required to compensate for the standby losses of the geyser configured without the isotherm blanket (blue line) and with the isotherm blanket (red line) do not overlap. The electrical energy consumed group means difference between these two scenarios was 0.56 kWh. The pvalue of the electrical energy consumed over the period where the geyser was without and with the isotherm blanket was 0.002. The very small p-value indicated that there was a significant difference under the two monitoring configurations. The difference in electrical energy consumed between the true mean in the configuration without the isotherm blanket and at 95% confidence level was 0.28 kWh. The difference in electrical energy consumed between the true mean in the configuration with the isotherm blanket and at 95% confidence level was 0.82 kWh. Therefore, there exists a significant mean difference; since, in traversing between the two intervals (without the isotherm and with the isotherm blanket), the value zero would not be included and also the fact that the two horizontal line plots (daily electrical energy consumed under the both configurations) do not overlap.

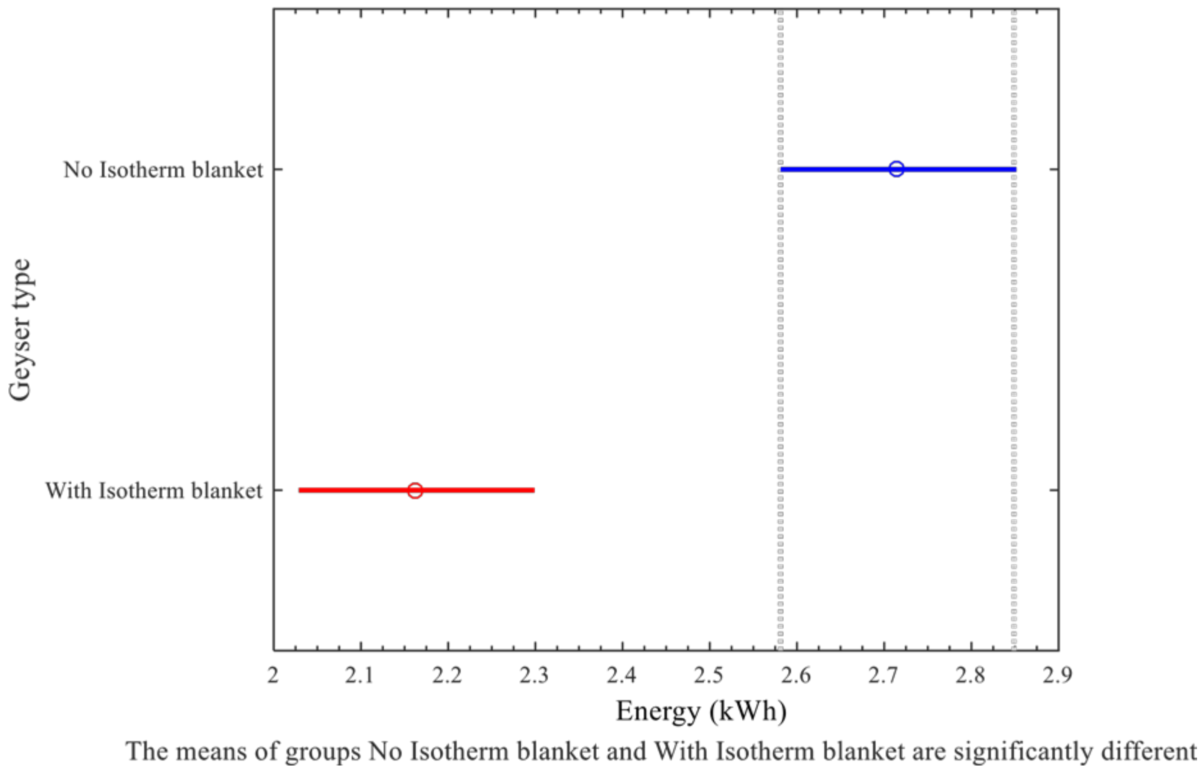


Figure 5.5: Multiple comparison simulation plots of the two geyser configurations

5.3.4.2 Multiple comparison test of standby losses of the split type system

The electrical energy consumed over a 24 hour period for both configurations; without the installed isotherm blanket and with the installed isotherm blanket was compared using the multiple comparison test over the four successive days of monitoring. Figure 5.6 shows the multiple comparison plots of the daily electrical energy to compensate for the standby thermal energy losses of the split type ASHP water heater under the two configurations.

It can be delineated from the Figure 5.6 that both mean daily electrical energy of the split type ASHP water heater without the isotherm blanket (blue line) and with the isotherm blanket (red line) does overlap. The electrical energy consumed group means difference between both cases was 0.21 kWh. The pvalue of the electrical energy consumed over the period where the split type ASHP system was without and with the isotherm blanket was 0.29. The p-value showed that there was no significant difference between the two monitoring configurations. The difference in electrical energy consumed between the true mean in the configuration without isotherm blanket and at 95% confidence level was   0.24 kWh. The difference in electrical energy consumed between the true mean in the configuration with the isotherm blanket and at 95% confidence level was 0.67 kWh. Therefore, there exists no significant mean difference; since, in traversing between the two intervals (without the isotherm and with the isotherm blanket), the value zero would be included and also the fact that the two horizontal line (daily electrical energy consumed under the both configurations) plots do overlap.

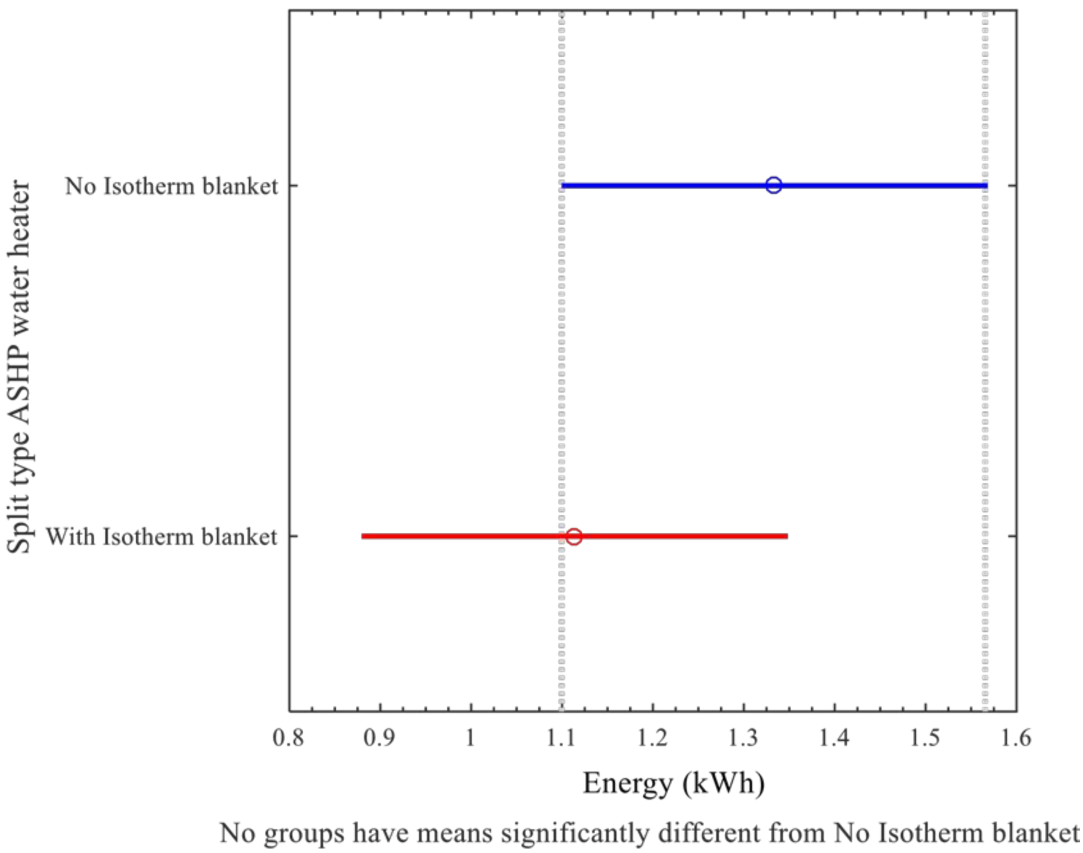


Figure 5.6: Multiple comparison simulation plots of the split type ASHP configurations

5.3.4.3 Multiple comparison test of standby losses of the integrated type system

The electrical energy consumed over a 24 hour period for both configurations that is without the installed isotherm blanket and with the installed isotherm blanket was compared using the multiple comparison test over the four successive days of monitoring. Figure 5.7 provides the multiple comparison plots of the daily electrical energy to compensate for the standby thermal energy losses of the two configurations of the integrated type ASHP water heater.

It can be depicted from the Figure 5.7 that both mean daily electrical energy of the integrated type ASHP water heater without the isotherm blanket (blue line) and with the isotherm blanket (red line) does overlap. The electrical energy consumed group means difference between the two cases was 0.03 kWh. The p-value of the electrical energy consumed over the period where the integrated type ASPH water heater was without and with the isotherm blanket was 0.92. The very large p-value showed that there was no significant difference between the two monitoring configurations. The difference in electrical energy consumed between the true mean in the configuration without the isotherm blanket and at 95% confidence level was -0.65 kWh. The difference in electrical energy consumed between the true mean in the configuration with the isotherm blanket and at 95% confidence level was 0.70 kWh. Therefore, there exists no significant mean difference; since, between the two intervals (without the isotherm and with the isotherm blanket) the value zero is included and also because the two horizontal line plots (daily electrical energy consumed under the both configurations) do overlap.

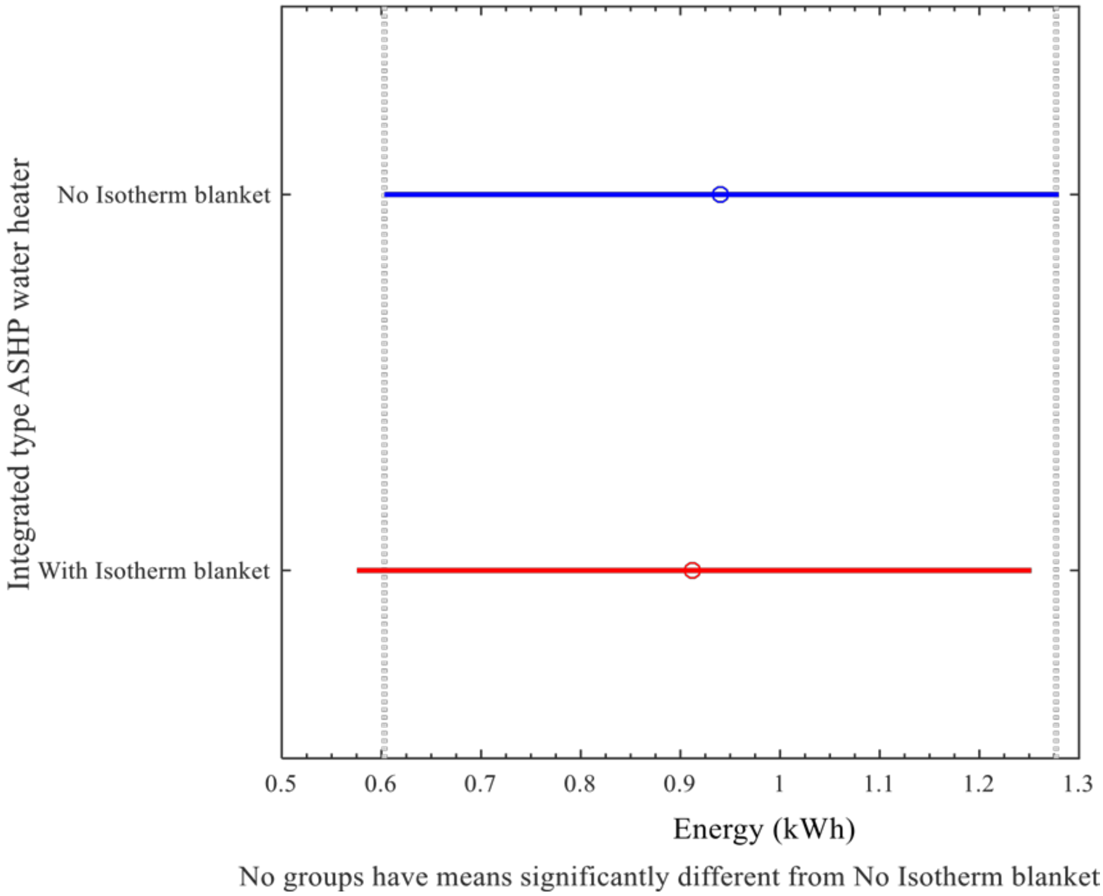


Figure 5.7: Multiple comparison simulation plots of the integrated type configurations

5.4 Summary

From the results, the following conclusions can be reached; despite the average standby thermal energy losses of over 2.5 kWh of a horizontally placed 150 L high-pressure geyser, which is in conformity with the South African Board Standard (SABS) for measurement and verification rating of the storage tank, the standby losses could be reduced by 18.5% by the installation of an isotherm blanket on the hot water cylinder. Also, there exists a significant mean difference in the electrical energy consumption to compensate for the standby thermal energy losses of the geyser without and with an isotherm blanket. On the contrary, there exists no significant difference in the electrical energy consumption in the case of ASHP water heaters (with and without the installed isotherm blanket). The standby thermal energy losses were lower with the integrated type than the split type ASHP water heaters in all the configurations. Also, the electrical energy factor was higher with the integrated type than the split type in all the configurations. The results can be of great significance to manufacturers and Energy Service companies of hot water heating devices in order to influence their decision whether to incorporate the isotherm blanket on hot water cylinders or otherwise.

Chapter Six

The performance of split and integrated type air source heat pump water heaters

Abstract

Renewable energy technologies that can provide optimum and cost-effective energy savings to mitigate global warming, energy crisis and to achieve energy efficiency continue to be of paramount importance. The present study focused on identifying critical parameters such as the volume of hot water drawn off; ambient temperature; relative humidity; refrigerant temperatures at the inlet and outlet of the compressor and condenser; and deterministic quantities such as time used, power consumption and coefficient of performance as indicators to benchmark the performance of both the split and integrated types of air source heat pump water heaters. The basis for analysis was on two predominant scenarios: first-hour heating rating and the heating cycle due to controlled volume of hot water drawn off wherein both the integrated and split type ASHP water heaters experienced vapour compression refrigeration cycles. A data acquisition system was employed to monitor the performance of both systems. The results obtained during summer season showed that, under the scenario of 150 L hot water withdrawal, the average COP of the systems was 3.18 and 2.85 for the split and integrated types, respectively. The average power consumed was 1.29 (split type) and 0.85 kW (integrated type). The duration of operation were 84 minutes (split type) and 138 minutes (integrated type).

Keywords: Air source heat pump, Coefficient of performance, Vapour compression refrigeration cycle, and Renewable energy technologies.

6.1 Introduction

Residential hot water heating offers an opportunity for energy savings, and the heat pump water heater provides a promising technology. The vapour compression refrigeration cycles is a process whereby refrigerant in the closed circuit loop of the heat pump undergoes phase change between the evaporator and condenser unit in a bid to transfer useful thermal energy. It can generate sanitary hot water by harnessing the aero-thermal energy during the vapour compression refrigeration cycle (VCRC). In South Africa, more than 90% of electrical energy is generated from coal and is solely supplied by Eskom (Van Eeden *et al*., 2016). The global warming potential caused by greenhouse gases, primarily carbon dioxide, is 510 Mt, of which 45% emanates from coalfired power plants (Bryson, 2011; Van Eeden *el al*., 2016).

Producing hot water accounts for up to 50% of domestic electricity use (Meyer and Tshimankinda, 1998; Tangwe *et al*., 2015). The energy factor for a geyser is the ratio of useful stored thermal energy in the cylinder to the input electrical energy consumed. The conventional heater (electric geyser) predominates, with an average energy factor of 0.92 (Huang and Lin, 1997; Tangwe *et al*., 2014). A possible alternative is the more energy-efficient air source heat pump (ASHP) water heater (Morrison *et al*., 2004), which can provide energy savings in the range of 50-70%, as it has a coefficient of performance (COP) that ranges from 2 to 4 (De Swardt and Meyer, 2001; Bodzin, 1997). The ASHP operates on the principle of VCRC and is a reverse air conditioner process (Marrison *et* *al.*, 2004). The thermo-physical properties of the refrigerant also contributes to the performance of the ASHP unit. The refrigerants used as the primary fluid for both the split and integrated types of ASHP water heaters were R417A and R407C of the zeotropic type with almost equal critical temperatures and critical pressures. The heat transfer coefficient of R417A is better than for R407C (Aprea *et al*., 2008).

In a bid to reduce demand on the national grid during peak hours, Eskom targeted rolling out 65,580 ASHP units up to March 2013 under a residential rebate scheme (Zhang *et al*., 2012). This strategy was expected to reduce annual demand by 54 MW, with savings of about 80.86 GWh during morning and evening peak hours. Having real-time data on the COP of ASHP water heaters was necessary, as any reliable mathematical model and simulation application to compute savings depended on the accuracy of data employed in the algorithm.

There are two categories of ASHP water heaters: the integrated and the split types (Marrison *et al*., 2004). The integrated type comprises an ASHP unit and a storage tank as a compact system, with the tank below the heat pump unit. It is commonly configured in two forms: one with an auxiliary backup heating element and the other without any backup element. Similarly, the split type also is in two groups: the single passed or 'once passed' type, and the recirculation system type. It can also operate with or without a backup element. The investigation reported on here was conducted with a split type ASHP water heater without an auxiliary backup element, and an integrated type ASHP water heater with a backup element. Both had a capacity of 150 L. The full methodology was provided in chapter three in section 3.3 and Figure 3.4 shows the schematic layout with the exclusion of the geyser and its metering sensors. The major goal was to use identified predictors such as ambient temperature, relative humidity, and the refrigerant temperatures at the inlet and outlet of the compressors and condensers, to analytically determine which of the two systems demonstrated a better performance in terms of COP.

The underlined deliverables were to:

i. determine the COPs of both split and integrated types of ASHP water heaters under different heating cycle scenarios, with controlled volumes of hot water drawn off; ii. evaluate the performance of the two types of ASHP water heaters, based on the average COP, power and energy consumption under the different heating cycle scenarios; and iii. ascertain the performance of the two types of heat pump water heaters by the predictors (power consumed, power factor, ambient temperature, relative humidity, inline cold water temperature, refrigerant temperatures at the inlet and outlet of the compressor and condenser) during the

VCRC.

6.2 Theory and calculations

The useful output thermal energy gained by the stored water is given by Equation 6.1.

$$Q = mc\delta T \quad (6.1)$$

Where;

Q = Useful thermal energy gained in kWh

m = Mass of water heated in kg c = Specific heat capacity of water in kJ/kg℃

□T = Temperature change in ℃

The input electrical energy consumed by the ASHP water heater is given by Equation 6.2.

$$E □ Pt \qquad (6.2)$$

Where;

P = Electrical power consumed in kW

t = Time taken for the VCRC in h E = Electrical energy consumption in kWh

The COP of the ASHP water heater is also given by the Equation 6.3.

$$COP = \frac{Q}{E} \qquad (6.3)$$

Where;

COP = Coefficient of performance of the ASHP water heater

6.3 Results and discussion

The analysis used performance data of the two types of ASHP water heaters for the full year from October 2015 to September 2016.

6.3.1 Summer performance of the two systems when 50 L of hot water is drawn off

The split and integrated type ASHP water heaters were switched off and 50 L hot water was drawn from each tank and replaced with cold water from the inline pipe feeding both tanks via the inlet pipe of each tank. After the withdrawal, the systems were switched on, at a common circuit breaker. The analysis was based on the morning (from 08:00), afternoon (from 13:00) and evening (from 18:00) data for a week in March 2016. The performance of the two systems on each of the operation times was analytically evaluated, with all the relevant predictors examined: power consumed, power factor, relative humidity, ambient temperature, inline cold water temperature, and refrigerant temperatures at the inlet and outlet of the compressor and condenser, as shown in Table 6.1. The average power consumed and the duration to complete the VCRC by both split and integrated type water heaters in the morning drawn off scenario was

1.30 and 0.86 kW, with the VCRC durations of 40 and 70 minutes, respectively. Average power consumed and time taken was 1.5 kW and 45 minutes for the split type system, while for the integrated type system it was 0.9 kW and 65 minutes during the afternoon drawn off scenario. Table 6.1 shows the evening drawn-off average power consumed, and the duration for the VCRC, for the split type system as 1.35 kW and 40 minutes, as opposed to 0.87 kW and 70 minutes for the integrated type system. The higher input power consumption of the split system aided the completion of the VCRC in a shorter time, when compared with the integrated type. The average power consumption for both systems was highest during the afternoon drawn off scenario because of the corresponding increase in ambient temperature and inline cold water temperature experienced during this period. Also, the input power during VCRC is strongly ambient temperature dependant.

Table 6.1 shows that both systems had an excellent power factor of 0.98 in all three periods. There were negligible variations in the relative humidity, ambient temperature and initial cold water temperature during the VCRC that occurred in the two systems in the morning period, and their averages were respectively 72%, 19.4℃ and 18.7℃. The averages of the relative humidity, ambient temperature and initial cold water temperature during the VCRC that occurred in the two systems in the afternoon period were respectively 36%, 29.5℃ and 26.5℃. The respective averages of the relative humidity, ambient temperature and initial cold water temperature during the VCRC encounter by the two systems in the evening period were 86%, 18.6℃ and 19.5℃. The significantly increased in the ambient temperature and also the inline cold water temperature in the afternoon period during the VCRC due to the 50 L drawn off were responsible for the increase in average power consumption for the both systems.

Although the average refrigerant temperature at the compressor inlet of the integrated system was lower than that of the split system in the morning scenario, 10.7℃ and 25.2℃, more thermal energy was gained by the refrigerant as it entered the suction end and exited the discharge end of the compressor in the split type, contrary to what happened in the integrated type. Moreover, the amount of the thermal energy gained was a function of the change in the refrigerant temperature at the inlet and outlet of the compressor and was 47.8 ℃ and 40.9℃ for the split and integrated types respectively. The average temperature of the refrigerant at the inlet of the compressor in the integrated type system was lower than that of the split type in the afternoon scenario, with respective temperatures of 12.3℃ and 22.5℃. The amount of the thermal energy gained was proportional to the change in the temperature of the refrigerant at the inlet and outlet of the compressors, at 43.1℃ and 40.5℃ for the split and integrated type ASPH water heaters respectively. The results showed that the average temperature of the refrigerant at the inlet of the compressor in the integrated type system was lower than that of the split type in the evening scenario, at 12.3℃ and 22.5℃. The amount of the thermal energy gained was proportional to the change in the temperature of the refrigerant at the inlet and outlet of the compressors and was 43.1 and 40.5℃ for the split and integrated types respectively. In all three scenarios, the refrigerant temperature at the inlet of the compressor was higher in the split type than in the integrated type, but the difference in the outlet and inlet temperature of the compressors was higher in the split type.

The amount of useful thermal energy gained by the hot water was a function of change in refrigerant temperature between the inlet and outlet of the condensers. The morning period average showed a difference in the change of the refrigerant temperature at the inlet and outlet of the condensers in the split type system (30.9℃), and the integrated type system (2.9℃) of 28.0℃. The refrigerants used in the two systems (R407C and R417A) were zeotropic, so the temperature gliding occurred at the condensers as well as at the evaporators during the VCRC. The afternoon drawn off shows a difference of 29.0℃ in the refrigerant temperature at the inlet and outlet of the condensers in the split and integrated systems (from 35.0℃ to 6.0℃). The evening averages show a difference of 26.1℃ at the inlet and outlet of the condensers in the split and integrated systems (from 30.0℃ to 3.9℃).

Analysis, supported by theory, thus showed that the split type had a better performance than the integrated type in all the scenarios, with a higher refrigerant temperature difference between the condenser inlet and outlet.

Table 6.1: Averages of the critical parameters when 50 L is drawn off

Parameter	Morning period		Afternoon period		Evening period	
	SIRAC	AIRCO	SIRAC	AIRCO	SIRAC	AIRCO
P (kW)	1.30	0.86	1.50	0.90	1.35	0.87
PF	0.98	0.98	0.98	0.98	0.98	0.98
RH (%)	72.00	72.00	36.00	36.00	86.00	86.00
AT (°C)	19.40	19.40	29.50	29.50	18.60	18.60
Ticw (°C)	18.70	18.70	26.50	26.50	19.50	19.50
Tcomi (°C)	25.20	10.70	22.50	12.30	22.50	12.30
Tcomo (°C)	73.00	51.60	65.60	52.80	65.60	52.80
Tconi (°C)	70.00	50.00	64.00	51.00	64.00	51.00
Tcono (°C)	39.10	47.10	29.00	45.00	34.00	47.10

P =average power, PF = power factor, RH = average relative humidity, AT = average ambient temperature, Ticw = inline cold water temperature, Tcomi = average refrigerant temperature at compressor inlet, Tcomo = average refrigerant temperature at compressor inlet, Tconi = average refrigerant temperature at condenser inlet, Tcono = average refrigerant temperature at condenser inlet.

6.3.2 Summer performance of both systems when 100 L of hot water is drawn off

The procedure described in Section 6.3.1 was repeated, but this time with 100 L of hot water drawn off. Table 6.2 shows the averages of the nine parameters examined.

The morning average power consumption of the split type system was 1.20 kW as opposed to 0.86 kW for the integrated type, with VCRC durations of 70 and 110 minutes respectively. The afternoon drawn off showed an average power consumption for the split type system of 1.30 kW, and 0.89 kW for the integrated type system, with VCRC durations of 60 and 100 minutes respectively. The evening drawn off showed an average power consumption for the split type system of 1.29 kW, and 0.89 kW for the integrated system, with VCRC durations of 65 and 110 minutes respectively. The higher input power consumption of the split system comparatively facilitated its completion of the VCRC.

The power factor of both systems in all three time scenarios was an excellent 0.98. The averages for the relative humidity, ambient temperature and the inline cold water temperature were negligible. The morning averages were respectively 69%, 22.0 and 20.0℃; with afternoon averages of 64.0%, 23.0 and 24.0℃; and the evening averages of 88%, 17.3 and 18.7℃ .

Table 6.2 shows the averages of the refrigerant temperature at the inlet and outlet of the compressors in the two systems in the three scenarios when 100 L was drawn off. In the morning, although the average temperature of the refrigerant at the inlet of the compressor in the integrated system, at 13.0℃, was lower than that of the split system, at 27.7℃, for the split type greater thermal energy was gained by the refrigerant as it entered the suction end and exited the discharge end. The amount of the thermal energy gained was a function of the change in the temperature of the refrigerant at the inlet and outlet of the compressor and was 45.8℃ and 39.6℃ for the split and integrated types respectively. Afternoon averages show that the refrigerant temperature at the inlet of the compressor in the integrated system was 12.8℃, compared to 28.4 ℃ for the split type system. There was a greater thermal energy gained by the refrigerant in the split type, with the difference in temperature of the refrigerant at the inlet and outlet of the compressor being 48.0 and 40.1℃ for the split and integrated types respectively. The evening drawn off showed the average refrigerant temperature at the inlet of the compressor in the integrated type system at 10.7℃, compared with 23.5℃ for the split type system. The corresponding difference in temperature of the refrigerant at the inlet and outlet of the compressor was 48.2℃ and 39.0℃ for the split and integrated types respectively.

Table 6.2 shows the averages of refrigerant temperature at the inlet and outlet of the condensers in both systems. The amount of useful thermal energy gained by the water strongly correlated with change in the refrigerant temperature between the inlet and outlet of the condenser. This difference was 29.5℃ for the morning drawn off (from 34.3℃ to 4.8℃) in the split and integrated systems. The difference in the afternoon was 33.5℃ (from 37.4 to 3.9℃). The evening difference was 32.0℃ (from 34.8℃ to 2.8℃) between the change in refrigerant temperature at the inlet and outlet of the condensers.

Analysis, supported by theory, thus showed that the split type had a better performance than the integrated type, with a higher refrigerant temperature difference between the condenser inlet and outlet.

Table 6.2: Averages of the nine critical parameters when 100 L is drawn off

Parameter	Morning period		Afternoon period		Evening period	
	SIRAC	AIRCO	SIRAC	AIRCO	SIRAC	AIRCO
P (kW)	1.20	0.86	1.50	0.89	1.29	0.87
PF	0.98	0.98	0.98	0.98	0.98	0.98
RH (%)	69.00	69.00	64.00	64.00	88.00	88.00
AT (°C)	22.00	22.00	23.00	23.00	17.30	17.30
Ticw (°C)	18.70	18.70	24.00	24.00	18.70	18.70
Tcomi (°C)	27.70	13.00	28.40	12.80	23.50	10.70
Tcomo (°C)	73.50	52.60	76.40	52.90	71.70	49.70
Tconi (°C)	71.50	51.50	75.20	51.40	70.50	48.70
Tcono (°C)	37.20	46.70	37.80	47.50	36.50	45.90

P =average power, PF = power factor, RH = average relative humidity, AT = average ambient temperature, Ticw = inline cold water temperature, Tcomi = average refrigerant temperature at compressor inlet, Tcomo = average refrigerant temperature at compressor inlet, Tconi = average refrigerant temperature at condenser inlet, Tcono = average refrigerant temperature at condenser inlet

6.3.3 Summer performance of both systems when 150 L of hot water is drawn off

The procedure described in Section 6.3.1 was repeated, but this time with 150 L of hot water drawn off. Table 6.3 shows the averages of the nine parameters examined.

The average power consumption of the split type system was 1.25 kW, compared with 0.83 kW for the integrated type system, with VCRC durations respectively 85 and 145 minutes during the morning session. In the afternoon, average power consumption of the split and integrated systems were 1.33 and

0.86 kW respectively, with VCRC durations of 75 and 125 minutes. In the evening, average power consumption was 1.28 and 0.86 kW for the split and integrated systems respectively, with VCRC durations of 90 and 145 minutes.

The higher input power consumption of the split system allowed for a shorter time taken for completing the VCRC. The power factor average for both systems was an excellent 0.98 in all three periods. There were no clear differences for the two systems in relative humidity, ambient temperature and the initial cold water temperature. The averages of the relative humidity, ambient temperature and in-line cold water temperature for both systems were 70%, 22.8℃ and 23.2℃ in the morning period; in the afternoon they were 35.0%, 27.0℃ and 25.0℃ , and in the evening they were 67%, 18.2℃ and 21.2℃.

Table 6.3 shows that the average temperature of the refrigerant at the inlet of the compressor in the integrated system was lower than that of the split system in the morning, at 11.1℃ and 23.2℃ ; afternoon at 11.1℃ and 35.6℃; and evening at 9.7℃ and 22.6℃ . The change in the temperature of the refrigerant at the inlet and outlet of the compressors was 48.7℃ and 37.5℃ for the split and integrated systems, respectively, in the morning; the difference in the afternoon was 48.2℃ and 41.9℃ , and in the evening it was 48.4℃ and 38.6℃ . The amount of useful thermal energy gained by water was a function of the change in the refrigerant temperature between the inlet and outlet of the condenser. The difference in the change of the refrigerant temperature at the inlet and outlet of the condensers in the split and integrated systems in the morning was 32.6℃ (from 37.4℃ to 4.8℃); in the afternoon 38.1℃ (from 43.1.4℃ to 5.0℃), and in the evening 33.6℃ (from 36.9℃ to 3.3℃).

Analysis, backed with the theoretical formulation of COP based on temperature lift, shows that the split type performed better than the integrated system.

Table 6.3: Averages of the nine critical parameters when 150 L is drawn off

Parameter	Morning period		Afternoon period		Evening period	
	SIRAC	AIRCO	SIRAC	AIRCO	SIRAC	AIRCO
P (kW)	1.25	0.83	1.33	0.86	1.28	0.86
PF	0.98	0.98	0.98	0.98	0.98	0.98
RH (%)	70.00	70.00	35.00	35.00	67.00	67.00
AT (°C)	22.80	22.80	27.00	27.00	18.20	18.20
Ticw (°C)	23.20	23.20	25.00	25.00	21.20	21.20
Tcomi (°C)	23.20	11.10	35.60	11.10	22.60	9.70
Tcomo (°C)	71.90	48.60	83.80	53.00	71.00	48.30
Tconi (°C)	70.40	47.50	82.50	52.00	70.00	47.50
Tcono (°C)	33.00	42.70	39.10	47.00	33.10	44.20

P = average power, PF = power factor, RH = average relative humidity, AT = average ambient temperature, Ticw = inline cold water temperature, Tcomi = average refrigerant temperature at compressor inlet, Tcomo = average refrigerant temperature at compressor inlet, Tconi = average refrigerant temperature at condenser inlet, Tcono = average refrigerant temperature at condenser inlet

6.3.4 Summary of the two systems' performance

Table 6.4 summarises the average performance of the split type and integrated type ASHP water heaters. In all scenarios, the average COP was more than 2, in line with previous research (Levins, 1982; Bodzin, 1997; Tangwe *et al*., 2014). The energy consumption of the integrated system was greater than that of the split system because of the backup element that switched on and in conjunction with the input electrical power delivered during the VCRC as well as the lengthy period of heating cycles. The average power consumed by the integrated system after withdrawals of 50, 100 and 150 L was respectively 0.85, 0.87 and 0.84 kW, compared with 1.27, 1.26 and 1.28 kW for the split system. Throughout the process of hot water withdrawal, the two systems showed negligible variation in power consumption. Despite this, the split system had a higher power consumption in all the scenarios, with the average electrical energy consumption lower at 0.81, 1.35 and 1.75 kWh, compared with 0.95, 1.55 and 1.96 kWh for the integrated system. Furthermore, the average COPs of the split type, at 2.88, 3.01 and 3.17, were consistently higher than those for the integrated system, at 2.44, 2.65 and 2.84. Finally, the duration of the VCRC that occurred in all scenarios was longer in the case of the integrated heat pump water heater, because of its lower electrical input power and COP.

Table 6.4: Comparisons of the two systems based on energies and COP

ASHP system	Drawn-off (L)	Time (min)	Power (kW)	Electrical energy (kWh)	Thermal energy (kWh)	COP
Integrated	50.0	69.88	0.85	0.99	2.19	2.19
Split	50.0	34.81	1.31	0.76	2.19	2.87
Integrated	50.0	60.71	0.85	0.86	2.32	2.70
Split	50.0	40.00	1.14	0.76	2.32	3.04
Integrated	50.0	70.47	0.86	1.01	2.45	2.42
Split	50.0	40.00	1.35	0.90	2.45	2.72
Integrated	100.0	110.82	0.85	1.57	4.20	2.68
Split	100.0	67.56	1.19	1.34	4.20	3.01
Integrated	100.0	100.91	0.88	1.48	3.92	2.64
Split	100.0	60.00	1.30	1.30	3.92	3.01
Integrated	100.0	111.03	0.87	1.61	4.23	2.63
Split	100.0	65.12	1.29	1.40	4.23	3.02
Integrated	150.0	146.02	0.83	2.02	6.16	3.05
Split	150.0	85.44	1.25	1.78	6.16	3.46
Integrated	150.0	126.35	0.85	1.79	4.78	2.67
Split	150.0	74.88	1.33	1.66	4.78	2.87
Integrated	150.0	145.41	0.85	2.06	5.79	2.80
Split	150.0	85.98	1.27	1.82	5.79	3.19

ASHP = Air source heat pump, COP = coefficient of performance

6.3.5 Comparative analysis of the two systems' overall performance

Tables 6.5 and 6.6 show the average COPs, power and energy consumptions of the two types of ASHP water heaters achieved for the typical summer and winter monitoring durations. The electrical and thermal energies of both systems under specific volumes of hot water drawn off were lower in summer than in winter periods, which can be accounted for by the lower ambient temperature during winter. The initial in-line cold water temperature as well as the water temperature into the inlet of the ASHP are also lower in winter. The average COPs of the two types of ASHP water heaters were better in summer than in winter. In addition, there was an increase in the COPs when large volumes of hot water were withdrawn. Lastly, the average power consumptions of both types, with the corresponding specific volumes of hot water drawn off, were lower in winter because of ambient temperature. Above all, it should be noted that both systems operated simultaneously. The average ambient temperature, relative humidity and the initial in-line cold water temperature were practically equal for the different scenarios of specific volumes of hot water drawn off.

Table 6.5: Summer comparison based on average energy and COP

ASHP system	Drawn off L	Power kW	Electrical energy kWh	Thermal energy kWh	COP
Split	50.0	1.1667	0.8067	2.3200	2.8767
Integrated	50.0	0.8533	0.9500	2.3200	2.4367
Split	100.0	1.2600	1.3600	4.1167	3.0133
Integrated	100.0	0.8667	1.567	4.1167	2.6500
Split	150.0	1.2833	1.7467	5.5767	3.1733
Integrated	150.0	0.8433	1.9543	5.5767	2.8400

ASHP=Air source heat pump, COP = coefficient of performance

Table 6.6: Winter comparison based on average energy and COP

ASHP system	Drawn off L	Power kW	Electrical energy kWh	Thermal energy kWh	COP
Split	50.0	1.1407	1.1564	2.6541	2.499
Integrated	50.0	0.9128	1.5635	2.6540	2.093
Split	100.0	1.2151	1.5994	4.9141	2.923
Integrated	100.0	0.8673	2.1612	4.9141	2.294
Split	150.0	1.2314	1.9091	6.0196	3.155
Integrated	150.0	0.8370	2.2798	6.0196	2.403

ASHP=Air source heat pump, COP = coefficient of performance

6.4 Summary

A residential air source heat pump water heater is an energy-efficient technology for sanitary hot water production irrespective of the type being employed or utilised. In this study, the split type heater without an electric backup had a better COP than the integrated type with an electric backup. The COP was also impacted by the input electrical energy consumption. There was a significant difference between the refrigerant temperature of the inlet and outlet of the condenser in the split system to that of the integrated system. Although the increase in the difference in refrigerant temperatures at the condenser units could account for the split system having a higher COP, the higher temperatures of the refrigerant at the inlet and outlet of the condenser unit in the split system could lead to it having a shorter lifespan. Based on the analysis, better COP was achieved when the difference between the refrigerant temperature of the inlet and outlet of the condenser was large. Another conclusion is that the COP of both types of ASHP water heaters performed better in summer than winter, thanks to favourable ambient conditions.

Chapter Seven

Simplified benchmark models to predict the coefficient of performance of air source heat pump water heaters

Abstract

A critical mathematical model can lead to reliable prediction of the dynamic behaviour of a system. In this study, a robust and accurate data acquisition system was employed to monitor the electrical energy consumption of a 150 L geyser and 150 L split and integrated type air source heat pump water heaters. This study equally focused on using the multiple linear regression models to correlate the coefficient of performance of the split and the integrated type ASHP water heaters to the difference between the hot water set point temperature and the ambient temperature ($T_s - T_a$) and the relative humidity (RH). The models derived for both the split and integrated type ASHP water heaters had good determination coefficients of 0.900 and 0.901, respectively. The reliefF algorithm tests showed that in either of the systems the RH was a secondary factor while the ($T_s - T_a$) was a primary factor. The cost of the DAS used in obtaining the data required for the model derivation was relatively low but of high measurement accuracy.

Keywords: Geyser, Air source heat pump, Coefficient of performance, ReliefF algorithm test, Multiple linear regression models and Data acquisition systems.

7.1 Introduction

In South Africa, there is an ongoing constraint on the electricity supply from the national grid to meet the demand. The South Africa electricity supply utility (Eskom) is implementing various measures such as; the Integrated Demand Management and the promotion and encouragement of the use of energyefficient devices like an air source heat pump (ASHP) water heater to replace the high electrical energy consuming conventional geysers in sanitary hot water production.

Hot water heating constitutes a significant percentage of electrical energy consumption in industrial, commercial and residential sectors, worldwide. Seemingly, water heating is the largest residential user of energy, with up to 50% of monthly electricity consumption being used for this purpose in South Africa (Meyer and Tshimankinda, 1998). The Eskom strategic plan outlook for 2010 to 2030 envisages over 20% reduction of electricity production from coal (Cooper and Prinsloo, 2002) as shown in Figure 7.1. One way to achieve this energy conservation measure is the implementation of an energy-efficient technology such as the heat pump for sanitary hot water production. Figure 7.1 illustrates the statistical outlook for sources of electrical energy generation in South Africa.

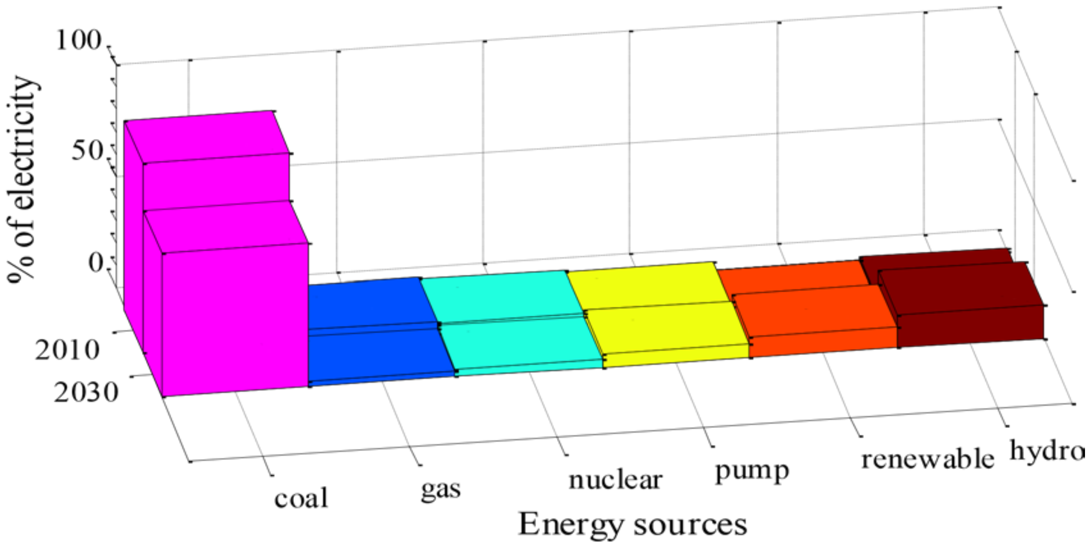


Figure 7.1: Eskom's energy outlook for sources of electricity production

In order to execute the aforementioned energy-efficient technology, Eskom embarked in rolling out a rebate programme of approximately 65,580 units of residential ASHP to retrofit existing geysers until March 2013 (Eskom report, 2011). Consequently, this strategy will go a long way to promote the use of this technology within the residential sector. However, the Eskom residential ASHP water heater rebate programme was discontinued in 2013 (Eskom, 2014) due to the inability of the National Energy Regulator of the country to continue the funding scheme. This left the country without any comparative tests for residential ASHP water heaters. It is paramount to highlight that the discontinuation of the heat pump rebate scheme was concluded as a result of lack of funding to support the initiative eventhough, the systems demonstrated an excellent annual COP of over two.

Considering the fact that the ASHP technology has been recommended and accepted for demand and energy reduction, it is therefore, imperative at this juncture to give an overview of the ASHP technology. The ASHP water heater is an electro-mechanical `closed circuit system comprising of a heat pump and a water storage tank; which operates on the principles of a vapour compression refrigeration cycle (VCRC). The key components of the heat pump unit are the evaporator coil, compressor, heat rejection condenser and an expansion valve. The ASHP water heaters can be categorised into integrated and split types. In the integrated type, both the ASHP unit and the storage tank exist as a single system and the ASHP is laid on top of the tank whereas in the split type, the ASHP unit is situated below the storage tank and connected to it by pipes (Tangwe *et al.,* 2016). Generally, the split type can further be classified as one passed circulation system and recirculation system. Studies have documented that the ASHP water heater could provide hot water at a quicker or same rate as an electrical resistance units (40 to 100%) and gas units (30 to 50%), but required warm ambient temperatures and a large heat pump or storage tank so as to provide a constant flow of hot water (Bodzin, 1997; Aguilar *et al*., 2005; Goswami and Kreith, 2007).

The characteristic of the heat pump that enabled it to provide such a very high efficiency of 300% is called the COP (De Swardt and Meyer, 2001). The COP of ASHP water heater is dependent on various parameters including component design, the load capacity cycle, thermo-physical properties of the working fluids, relative humidity and air speed through the duct space. The instantaneous, seasonal or annual COP can be calculated using simulation with the TRNSYS software package (KLEIN-TRNSYS, 1990). An analytical mathematical model that correlated the COP and the temperature of solar assisted ASHP water heater has also been developed (Itoe *et al*., 1999). It must be alluded that pocket of dynamic models of heat pump water heaters have been developed. More so, the bulk of the established mathematical models were developed from first principles whereby the integrated model of the heat pump water heaters is derived from the combination of the subsystem models that make up the VCRC closed loop circuit. Fardoun *et al*. (2011) developed a dynamic model of ASHP water heater based on independent heat transfer, thermodynamics, fluid mechanics and empirical correlations of the evaporator, compressor, condenser and expansion valve of the ASHP. The results confirmed that the rate of heating increased with a decrease in the capacity of the hot water storage tank and also the performance of the integrated system increased with an increase in ambient temperature.

MacArthur and Grald (1989) designed and built a model of vapour-compression heat pumps. The evaporator and the condenser were modelled with in-depth heat distribution equations, while the expansion valve was modelled as a capillary tube. Fu et al. (2003) presented a dynamic model of air-to-water dualmode heat pump with a screw compressor having four step capacities. The dynamic models developed with the introduction of additional compressor capacity in stepwise manner were studied. Kima et al. (2004) presented a dynamic model of a water heater system driven by a heat pump and applied a finite volume method to describe the heat exchangers. Furthermore, the lumped parameter models were employed to analyse the compressor and the storage tank, where dynamic simulations were carried out for various reservoir sizes.

Techarungpaisan et al. (2007) presented a steady state simulation model to forecast the performance of a small split type air conditioner comprising of a rotary compressor and a capillary tube but integrated with water heater.

Despite, the complexity of the dynamic models of the various heat pump water heaters, the determination coefficient of the predicted and measured COP was slightly above 0.9.

The focus of the study was to derive simple mathematical models to predict the COP of the residential ASHP water heaters, which could be of high accuracy and with the employment of a low cost DAS. The present research, therefore, focused on benchmarking the performance of a 150 L split type ASHP water heater without an electric element as a backup and a 150 L integrated type ASHP water heater with an electric element as a backup to the performance of a 150 L geyser under different heating cycle scenarios. In the various controlled volumes of hot water drawn off, the thermal energy gained by the water in the storage tanks of all three heating devices was equal to the electrical energy consumed by the geyser. Multiple linear regression models were developed and built for the two types of ASHP water heaters using the predictors [(Ts-Ta) and RH] and the desired response (COP). The hot water set point temperature (Ts) was set at 55℃, since sanitary hot water at this set point temperature is free from bacteria growth. The predictors were ranked according to their weight of importance to the COP using the reliefF algorithm test (Robnik-Šikonja and

Kononenko, 2003). The derived mathematical models of the COP of the both types of ASHP water heaters could be used to identify the system with a better performance.

The full methodology was provided in Chapter three in section 3.3 and Figure 3.4 with the exclusion of the temperature sensors installed in the closed loop circuit of the both types of ASHP water heaters.

7.2 Theory and calculations

The thermal energy gained by stored water in the tanks of the hot water heating devices as a result of the specific controlled volume of hot water drawn off was equal to the electrical energy consumed by the geyser. The impact of stand by losses was neglected because before each scenario of hot water drawn off, the hot water set temperature of each storage tank was adjusted to 55℃. The Equation 7.1 shows that the electrical energy consumed by the geyser was equal to the thermal energy gained by water in all the hot water heating devices.

$$Q_s = E_g \qquad (7.1)$$

Where;

Q_s = Thermal energy gained by stored water in the hot water device in kWh

E_g = Electrical energy consumed by geyser in kWh

The electrical energy consumed by the three technologies was given by Equation 7.2.

$$E_s = P_s t \qquad (7.2)$$

Where;

Ps = Average electrical power consumed by the hot water device in kW

t =Time taken in h

Es = Electrical energy consumed by the hot water device in kWh

The theoretical COP of the ASHP water heater was given by the ratio of the useful thermal energy gained to the input electrical energy consumed during the VCRC as shown in Equation 7.3.

$$COP_{cal} = \frac{Q_s}{E_s} \qquad (7.3)$$

Where;

COP_{cal} = Calculated coefficient of performance of the ASHP water heaters

The mathematical modelled COP of the ASHP water heater was given by Equation 7.4.

$$COP_{mod} = \beta_0 + \beta_1(T_s - T_a) + \beta_2 RH \qquad (7.4)$$

Where;

COP_{mod} = Modelled COP of the ASHP water heater

$T_s - T_a$ = Difference in hot water setpoint temperature and the ambient temperature in ℃

RH = Relative humidity in %

β_0 = Forcing constant

β_1 = Scaling constant of the predictor ($T_s - T_a$) in /℃

β_2 = Scaling constant of the predictor (RH) in /%

7.3 Results and discussion

7.3.1 Comparative analysis of the performance of the hot water devices

The performance of the three hot water heating devices was compared based on the average electrical power consumption, the total electrical energy consumption and the average COPs after the specific volumes (50, 100 and 150 L) of hot water drawn off scenarios for both the summer and the winter periods.

7.3.1.1 Summer comparison of the performance of the hot water technologies

Table 7.1 shows the summer crucial parameters that were monitored and measured under the average specific volumes (50, 100 and 150 L) of hot water drawn off scenarios.

Table 7.1: Summer parameters measured under the controlled drawn off

Heating Systems	Drawn off (L)	Power (kW)	Electrical energy (kWh)	Ambient temperature (℃)	Relative humidity (%)	COP
Split	50.0	1.215	0.860	22.54	65.48	3.00
Integrated	50.0	0.922	0.957	22.74	64.86	2.54
Geyser	50.0	2.500	1.830	22.74	64.86	-----
Split	100.0	1.270	1.416	21.08	73.92	3.01
Integrated	100.0	0.912	1.545	21.13	73.48	2.69
Geyser	100.0	2.500	4.054	21.13	73.48	-----
Split	150.0	1.293	1.411	23.69	58.88	3.10
Integrated	150.0	0.917	1.496	23.36	58.88	2.83
Geyser	150.0	2.500	4.390	23.36	58.88	-----

It can be observed from Table 7.1 that in all the scenarios of the controlled volume of hot water drawn off, the average electrical power and total energy consumption of the geyser was the largest in contrast to the ASHP water heaters. Although the average electrical power consumption of the split type ASHP water heater was higher than that of the integrated type ASHP water heater, it always had a lower total electrical energy consumption. It could also be depicted without loss of generality that the average COP of both types of ASHP water heaters increased with an increase in the volume of hot water drawn off and average ambient temperature. The average COP of the ASHP water heaters could also be influenced by the average relative humidity as changes in the relative humidity also affected the COP. The average COP of the ASHP water heaters was above 2 in all the heating cycles of controlled volume of hot water drawn off (Bodzin, 1997). Furthermore, the split type performed better than the integrated type ASHP water heater.

7.3.1.2 Winter comparison of the performance of the hot water technologies

Table 7.2 shows the crucial winter parameters that were monitored and measured under the average specific volumes (50, 100 and 150 L) of hot water drawn off scenarios.

Table 7.2: Winter parameters measured under the controlled drawn off

ASHP system	Drawn off (L)	Power kW	Electrical energy (kWh)	Ambient temperature (°C)	Relative humidity (%)	COP
Split	50.0	1.141	1.156	15.61	67.95	2.499
Integrated	50.0	0.912	1.564	15.69	67.98	2.093
Geyser	50.0	2.500	2.640	15.69	67.98	-----
Split	100.0	1.215	1.599	14.99	71.60	2.923
Integrated	100.0	0.867	2.161	15.24	70.04	2.294
Geyser	100.0	2.500	4.914	15.24	70.04	-----
Split	150.0	1.231	1.909	19.50	57.55	3.155
Integrated	150.0	0.837	2.280	19.28	59.73	2.403
Geyser	150.0	2.500	6.017	19.28	59.73	-----

Table 7.2 shows that the average power consumption of the ASHP water heaters was slightly lower during winter, but the total electrical energy consumption was higher as compared to the summer period with respect to the

corresponding controlled volume of hot water drawn off. In addition, the average COP of the ASHP water heaters also dropped in comparison to the summer performance owing to the decrease in ambient temperatures. It could also be demonstrated that during the winter season, a decrease in the average ambient temperature resulted in a corresponding decrease in the average temperature of the in-line mains cold water. Again, despite the drop in the average ambient temperatures, the average COP of both ASHP water heaters were still over 2 as depicted by Levin (1982).

7.3.1.3 Comparison of average crucial parameters of both systems under partial load condition

The overall performance of the split and integrated type ASHP water heaters can be assessed based on the average power consumed, the average COP, the average ambient temperature and relative humidity of each of the systems for both the winter and summer seasons under partial load (50 L and 100 L) hot water drawn off. It can be deduced from both Tables 7.1 and 7.2, that the average ambient temperature and relative humidity recorded during a VCRC obtained due to a specific volume of hot water drawn off were practically equal despite the significant difference in the duration for the particular heating cycle. Furthermore, an increase in the volume of hot water drawn off was associated with an increase in the average COP for either type of ASHP water heaters couple with an increase in the average ambient temperature and average power consumption (Tangwe *et al*., 2014). In all scenarios of the hot water drawn off, the average COP for the summer and winter periods was above 2. Nevertheless, but of the same volume of hot water drawn off, it was observed that the average COP for the summer period was greater than that achieved during the winter.

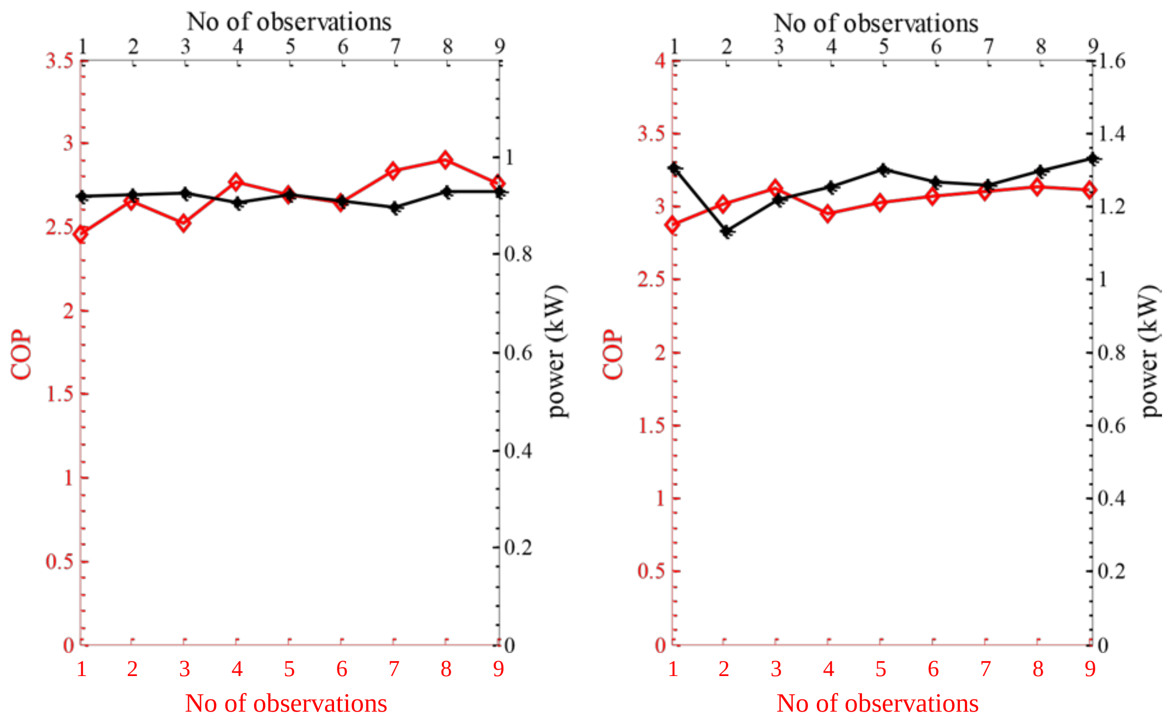
7.3.1.4 Comparison of average crucial parameters of both systems under full load condition

The performance of both types of ASHP water heaters was evaluated under full load condition which corresponded to 150 L hot water drawn off, during the summer and winter seasons. Tables 7.1 and 7.2 show the determined average power consumed, the average COP and the average ambient temperature and relative humidity for both the split and integrated type ASHP water heaters. It can be shown that the average COP was again better in the split type than in the integrated type under the full load condition. In addition, without loss of generality, the average COP of either systems, regardless of the season was better under a full load operation mode than in a partial load operation mode without any simultaneous feeding of cold water into the storage tanks. Also, the total electrical energy saved by retrofitting geyser with ASHP water heaters was greater during the full load operation condition as opposed to the partial load condition.

7.3.2 Variation of COP, electrical and weather parameters with observations

7.3.2.1 Summer variation of COP and power consumption with observations

Figure 7.2 shows the dataset of some average determined COP and average power consumption of some observations obtained by the specific volumes of hot water drawn off from each of the ASHP water heaters. The results depicted that throughout the observations, there occurred minimal fluctuation in the average COP and the average power consumption of both the integrated and split type ASHP water heaters. The average COPs and the average power consumption over the number of observations were about 2.6 and 3.0 beside 0.91 kW and 1.2 kW, for the integrated and split type ASHP water heaters, respectively. It should be emphasised that the observations were obtained from different controlled volumes of hot water drawn off.



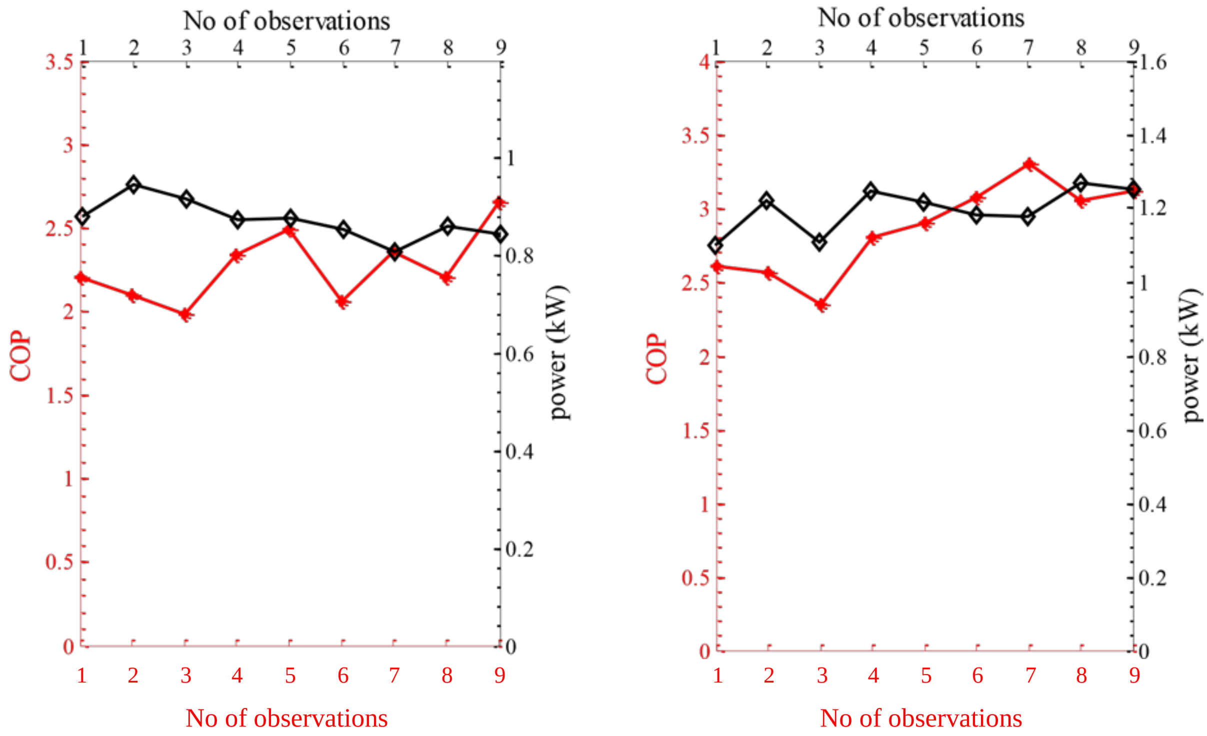
Integrated type Split type

Figure 7.2: Summer COPs and power with observations for both ASHP water heaters

7.3.2.2 Winter variation of COP and power consumption with observations

Figure 7.3 shows the dataset of some average determined COP and average power consumption of some observations obtained by the specific volumes of hot water drawn off from each of the ASHP water heaters. The results demonstrated that throughout the observations, there existed a lower average COP and average power consumption for both the integrated and split type ASHP water heaters as opposed to the performance in the summer period. The average COP over the number of observations was about 2.3 and 2.8 for the integrated and split type ASHP water heaters, respectively; alongside the average power consumption of about 0.87 kW and 1.10 kW for the respective types of the ASHP water heaters.

It is very important to mention that the observations were obtained from different controlled volume of hot water drawn off.

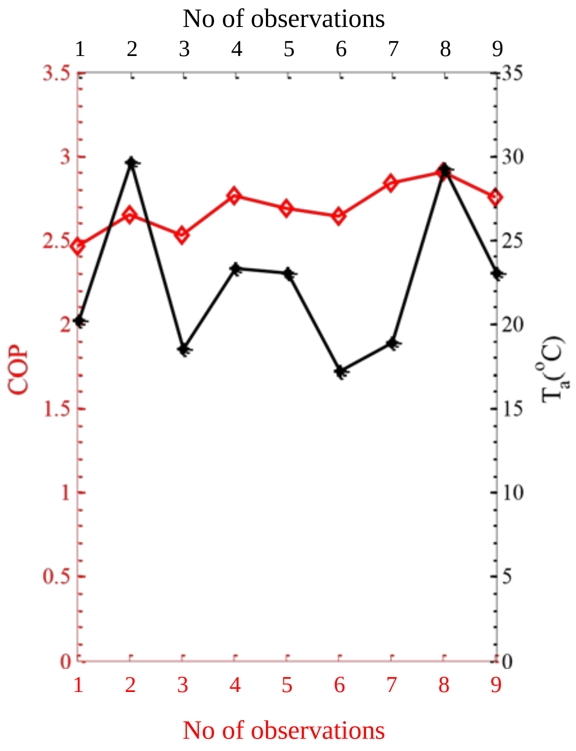


Integrated type Split type

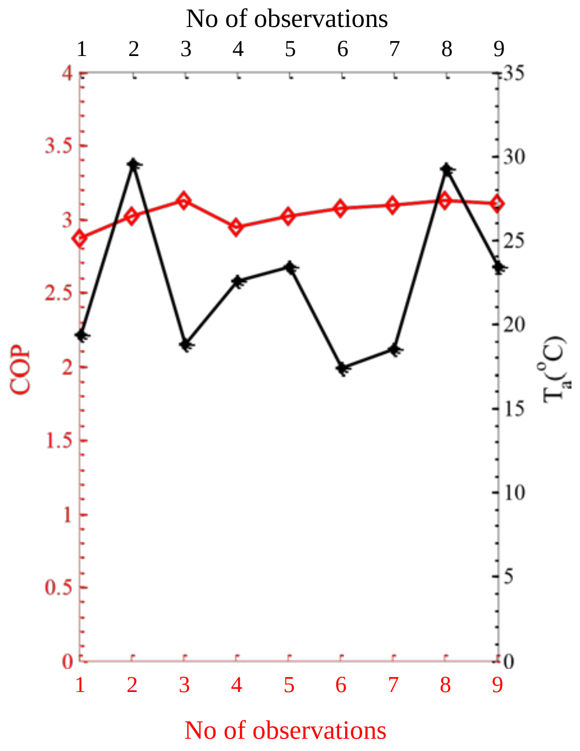
Figure 7.3: Winter COPs and power with observations for both ASHP water heaters

7.3.2.3 Summer variation of COP and ambient temperature with observations

Figure 7.4 shows the dataset of some average determined COP and the average ambient temperature of some observations obtained by the specific volumes of hot water drawn off from each of the ASHP water heaters. The results showed that throughout the observations, there were very small changes in the COP and the average ambient temperature for both the integrated and split type ASHP water heaters. The average COP over the number of observations was approximately 2.6 and 3.0 for the integrated and split type ASHP water heaters, respectively; whilst the average ambient temperature ranged from 18℃ to 29℃ for the respective heating systems. Despite the fact that the ambient temperature had influence on the COP, it should be alluded that it was not a primary factor as demonstrated by Tangwe et al. (2014). The statistical test obtained from the model revealed that both the ambient temperature and the relative humidity were secondary factors affecting the COP of the ASHP water heaters while the refrigerant temperatures of the evaporator and condenser as well as the volume of water heated were primary factors.



Integrated type

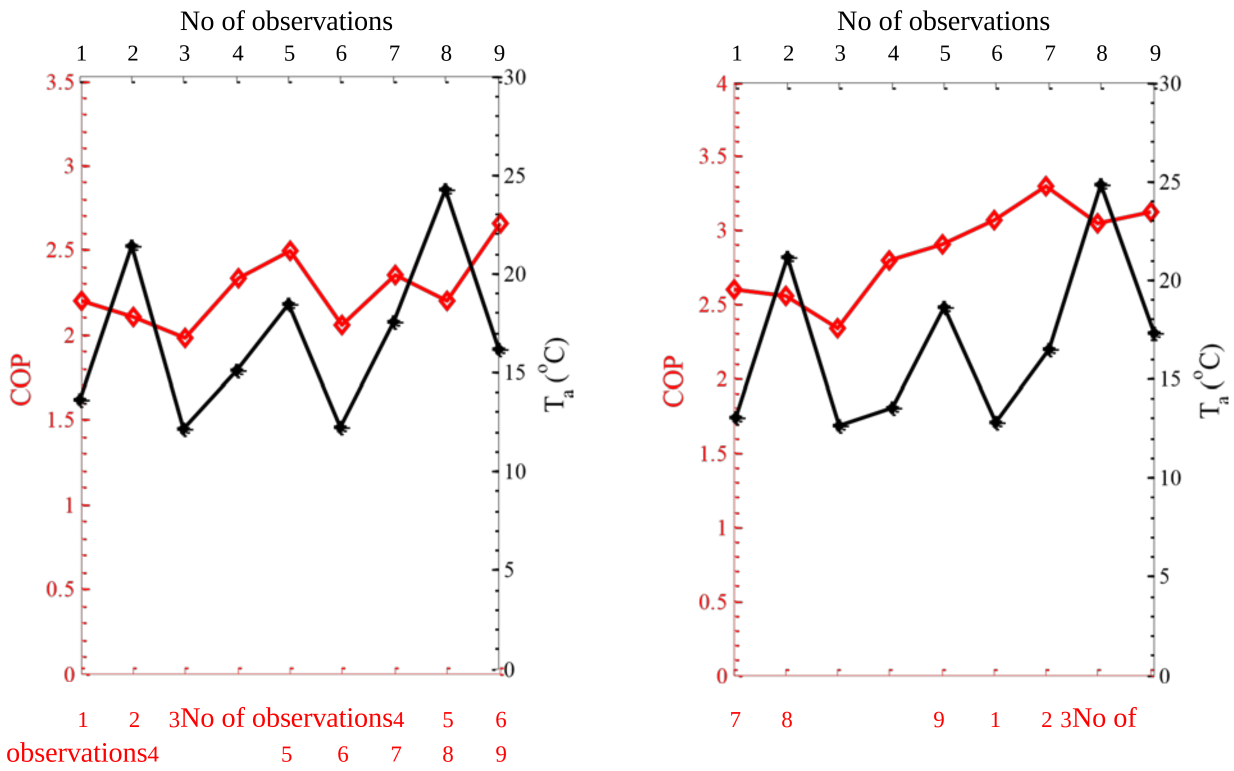


Split type

Figure 7.4: Summer COP and ambient temperature with observations for both systems

7.3.2.4 Winter variation of COP and ambient temperature with observations

Figure 7.5 shows the dataset of some average determined COP and the average ambient temperature of some observations obtained by the specific volumes of hot water drawn off from each of the ASHP water heaters. The results showed that throughout the observations, there were slight changes in the COP and the average ambient temperature for both the integrated and split type ASHP water heaters. The average COP over the number of observations was approximately 2.3 and 2.8 for the integrated and split type ASHP water heaters, respectively, whereas the average ambient temperature ranged from 14℃ to 24℃ for the both heat pump devices.



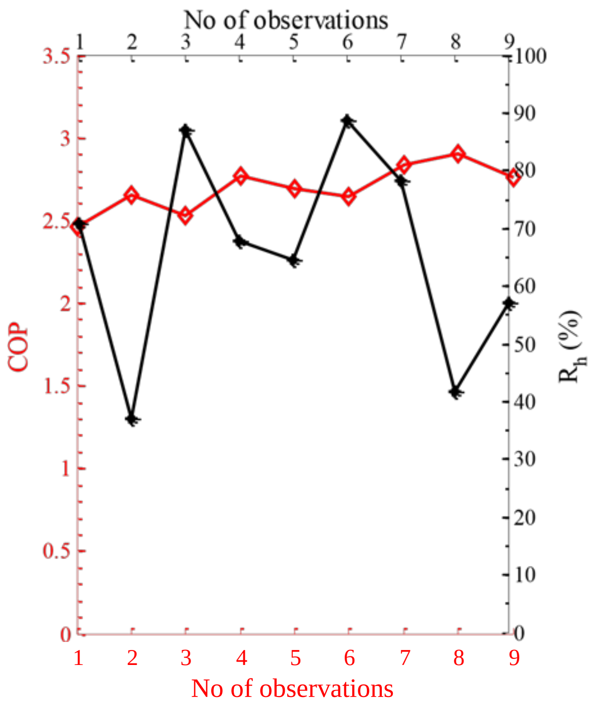
Integrated type

Split type

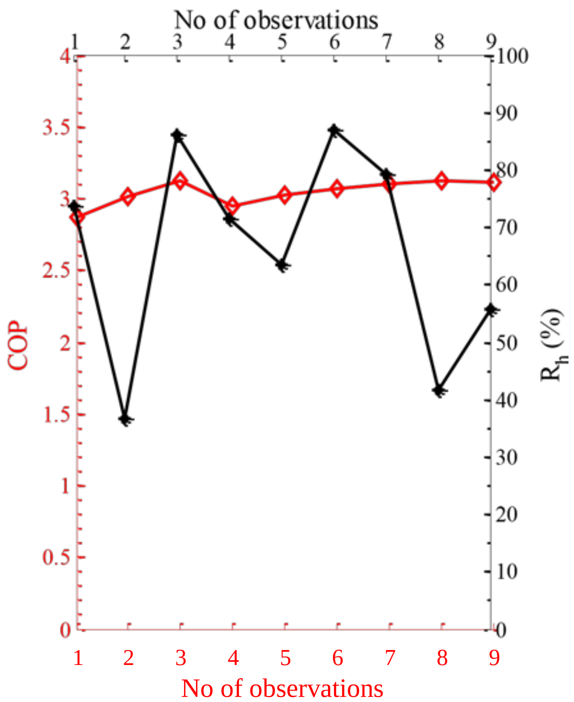
Figure 7.5: Winter COP and ambient temperature with observations for both systems

7.3.2.5 Summer variation of COP and relative humidity with observations

Figure 7.6 shows the dataset of some average determined COP and average relative humidity of some observations obtained by the specific volumes of hot water drawn off from each of the ASHP water heaters. Figure 7.6 demonstrated that throughout the observations, there were very marginal changes in the COP while changes in the average relative humidity were significant for both the integrated and split type ASHP water heaters. The average COP over the number of observations was about 2.6 and 3.0 for the integrated and split type ASHP water heaters, respectively. In addition, the average relative humidity over the number of observations ranged from 35% to 88% for the integrated and split type ASHP water heaters. In spite of the wide variation in the average relative humidity, the impact on the average COP was not significant. This revealed that relative humidity was also a secondary factor affecting the COP of the systems.



Integrated type

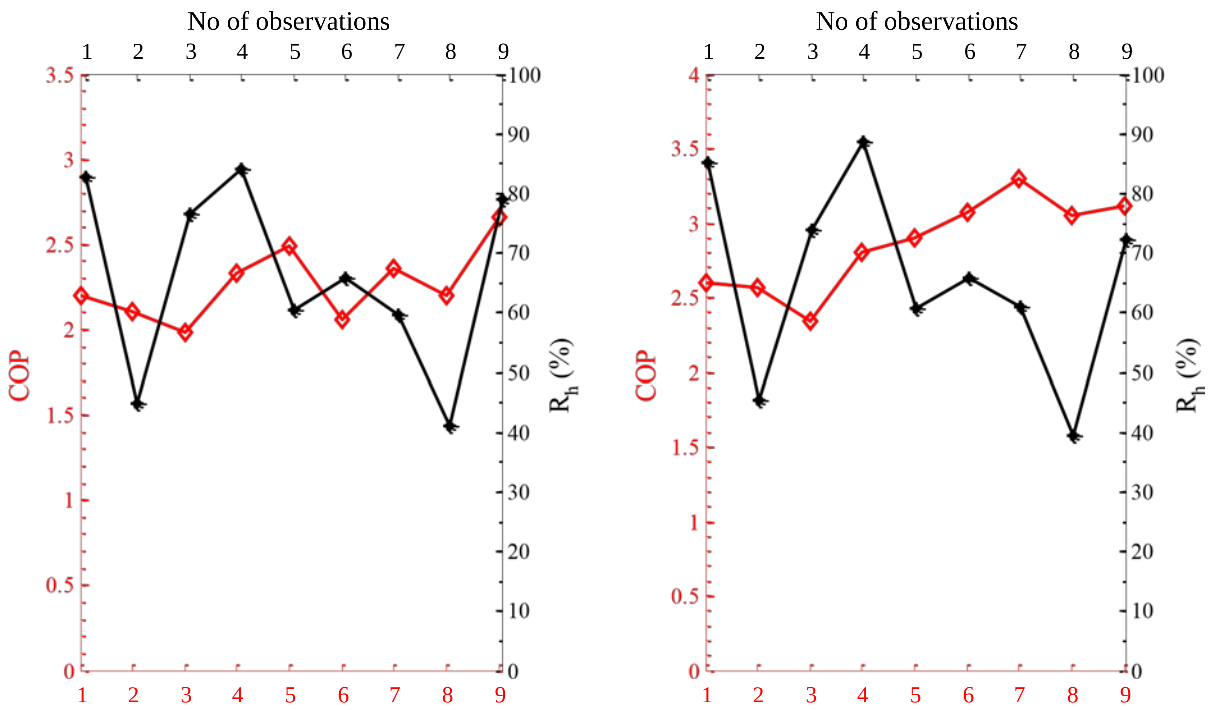


Split type

Figure 7.6: Summer COP and relative humidity with observations for both systems

7.3.2.6 Winter variation of COP and relative humidity with observations

Figure 7.7 shows the dataset of some average determined COP and the average relative humidity of some observations obtained by the specific volumes of hot water drawn off from each of the ASHP water heaters. Figure 7.7 showed that throughout the observations, there were very small fluctuations in the COP while changes in the average relative humidity were substantial for both the integrated and split type ASHP water heaters. The average COP over the number of observations was about 2.3 and 2.8 for the integrated and split type ASHP water heaters, respectively. Also, the average relative humidity over the number of observations ranged from 40% to 85% for the integrated and split type ASHP water heaters. Although there was a wide range in fluctuation that occurred in the average relative humidity, the impact on the average COP was not significant.



No of observations

No of observations

Integrated type

Split type

Figure 7.7: Winter COP and relative humidity with observations for both systems

7.3.3 Development of the mathematical models of the COP of the systems

More than 100 datasets of the predictors [($T_s - T_a$), and RH] and the calculated COP for each of the systems were used to develop and build the multiple linear regression models. This was to establish a correlation between the inputs and the output parameters. These datasets spanned the full winter and summer periods from October 2015 to September 2016. The derived multiple linear regression model used is as shown in Equation 7.4. Table 7.3 shows the forcing and scaling values of the mathematical model developed for the split type ASHP water heater. The model equation of the split type ASHP water heater depicted that the predictor ($T_s - T_a$) made a significant contribution to the COP. It could also be predicted that a decrease in ($T_s - T_a$) resulted in a corresponding increase in the COP provided the relative humidity (RH) was kept constant. Furthermore, an increase in relative humidity could lead to a marginal rise in the COP with the assumption that the predictor ($T_s - T_a$) was held constant.

Table 7.3: Model's scaling and forcing constants of the split system

Predictors	Symbols	Scaling notations	Scaling Values	Output
Forcing constant		□$_0$	3.632	
Difference in set point and of ambient temperatures (T_s, T_a)	T_s □ T_a	□$_1$	-0.0266	COP
Relative humidity	RH	□$_2$	0.0039	

From the model equation scaling constants shown in Table 7.3, it can be visualised that an increase in (T_s □ T_a) might have likely resulted in a decrease in COP at a rate of 0.0266 /℃. An increase in RH led to a corresponding increase in the COP of the split type ASHP water heater at a rate of 0.0039 /%. The forcing constant (3.632) is the arbitrary lump constant that accommodated the contribution offered by other predictors to the output, though not included in the derived model.

The modelled and calculated average COP of the split type ASHP water heater had a determination coefficient of 0.900, and there exists a good fit between the calculated average COP dataset and the predicted COP modelled curve fit. Figure 7.8 shows the sample dataset of the calculated average COP and the modelled COP best curve fit of some observations depicted from the different scenarios of hot water drawn off.

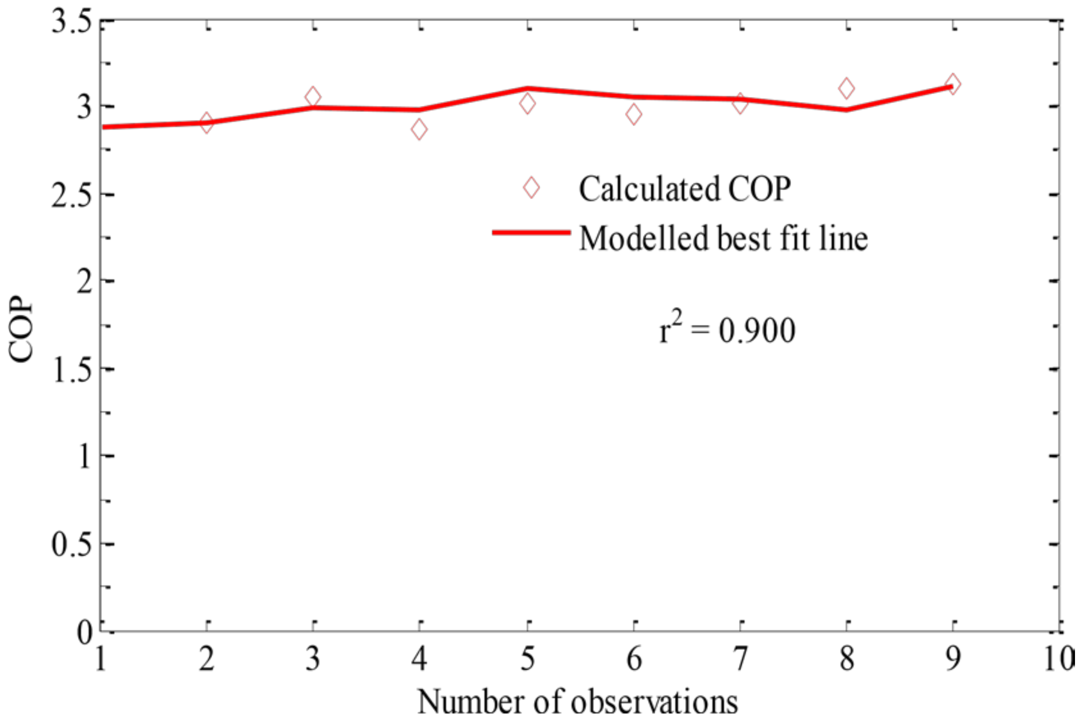


Figure 7.8: Determined and modelled COP of the split type ASHP water heater

Table 7.4 shows the forcing and scaling values of the mathematical model developed for the integrated type ASHP water heater. The model equation of the integrated type ASHP water heater justified that the predictor ($T_s \square T_a$) made a significant contribution to the average COP. It can also be shown that an increase in ($T_s \square T_a$) resulted in a corresponding decrease in the average COP, provided the relative humidity was unchanged.

Table 7.4: Model's scaling and forcing constants of the integrated system

Predictors	Symbols	Scaling notations	Scaling Values	Output
Forcing constant		$\square_0$	3.9311	
Difference in set point and of ambient temperatures (T_s , T_a)	$T_s \square T_a$	$\square_1$	-0.0697	COP

Relative humidity	RH	□2	0.0153

The scaling constants of the model equation as shown in Table 7.4, suggested that an increase in ($T_s \square T_a$) might have resulted in a decrease in the average COP at a rate of -0.0697 /℃. In addition, an increase in the average RH resulted in a corresponding increase in the average COP of the ASHP water heater at a rate of 0.0153 /%. The forcing constant of the average COP of the integrated type ASHP water heater was 3.931.

The modelled and calculated average COP of the integrated type ASHP water heater had a determination coefficient of 0.901, and there exists a good fit between the calculated average COP dataset and the modelled best curve fit. Figure 7.9 shows the sample dataset of the calculated average COP and the modelled COP best curve fit of some observations depicted in the different scenarios of hot water drawn off.

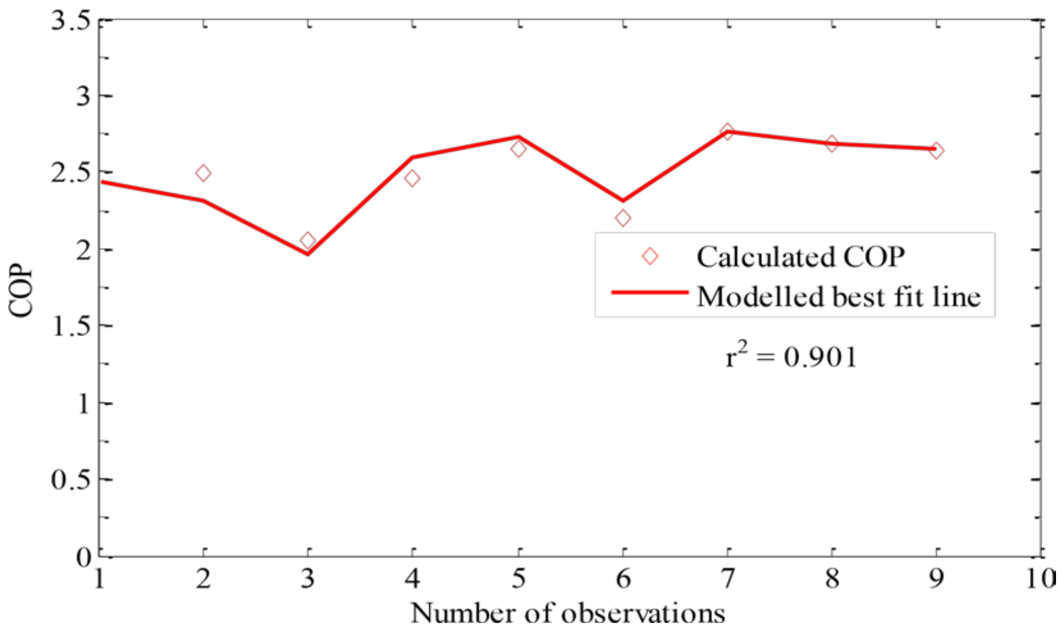


Figure 7.9: Determined and modelled COP of the integrated type ASHP water

Heater

7.3.4 Ranking of predictors by weight contribution to the output using ReliefF test

The two predictors [($T_s \square T_a$) and RH] and the output (COP) from the processed data of the split and integrated type ASHP water heaters were used in the ReliefF algorithm test to rank the predictors according to their importance of weight contribution. The ReliefF test is a statistical analysis that uses the regression method to rank predictors with respect to their importance of weight contribution to the output (Robnik-Sikonja and Kononenko, 2003). The weighted rank of a particular predictor can be between -1 and 1. Therefore, a positive weight rank of a predictor indicated that it was a primary factor while a negative weight rank insinuated that it was a secondary factor. Figure 7.10 shows the reliefF bar plots of the predictors and the importance of weight contributions to the COP for both the split and integrated type ASHP water heaters. The weight ranking showed that for both types of ASHP water heaters, the difference in hot water set point temperature and ambient temperature ($T_s \square T_a$) was a primary factor while relative humidity (RH) was a secondary factor. It could also be determined from the statistical algorithm that both the primary and secondary predictor weight contribution of the integrated type ASHP water heater were ($T_s \square T_a$) = 0.070 and RH = - 0.001 and those of the split type ASHP water heater were ($T_s \square T_a$) = 0.034 and RH = - 0.021.

The impact of the predictor (T_s □ T_a) via its contribution due to the weight of importance to the COP was the most significant. The contribution by weight of the predictor (T_s □ T_a) in the integrated type ASHP water heater was over twice than that of the split type ASHP water heater.

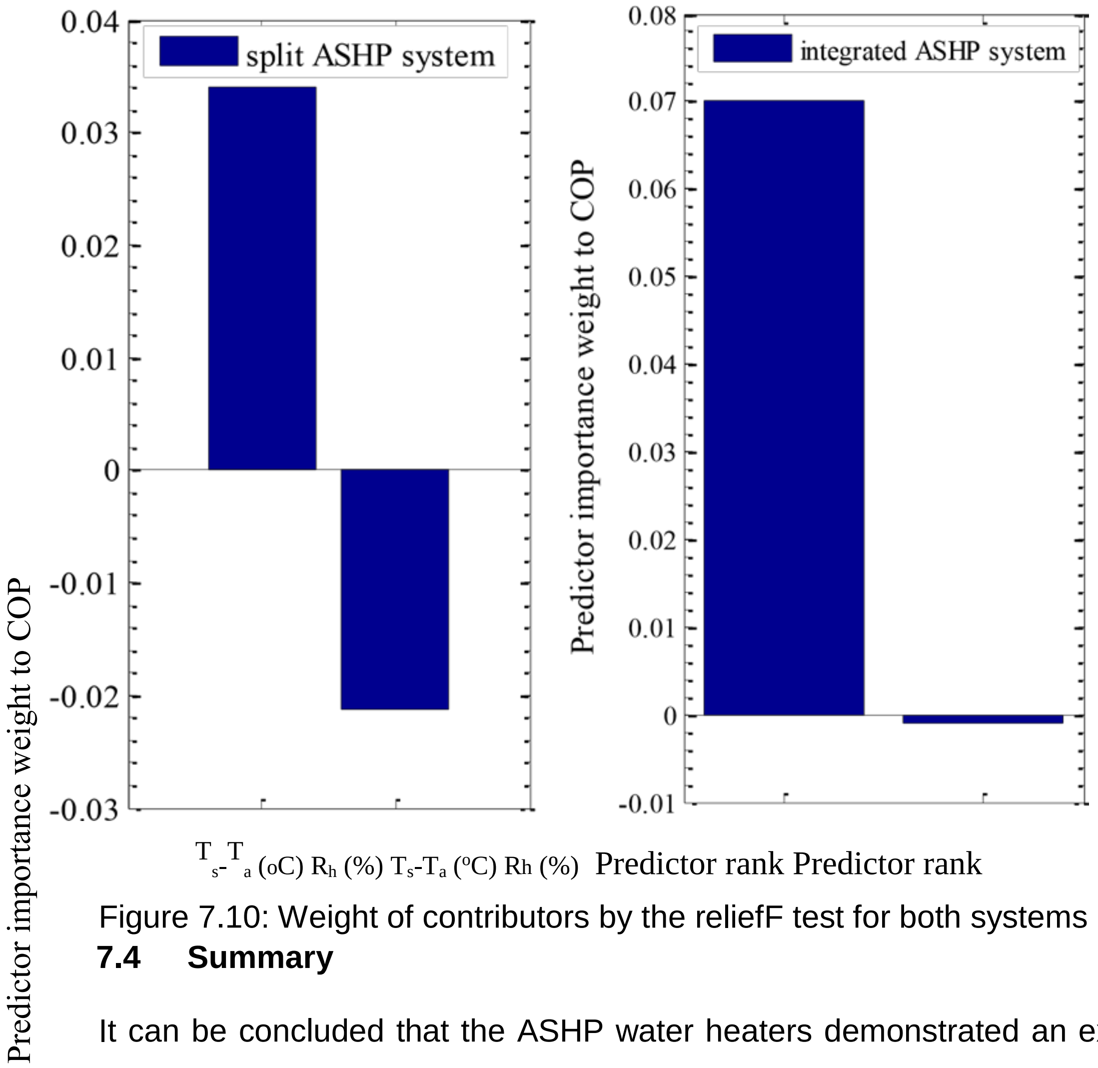


Figure 7.10: Weight of contributors by the reliefF test for both systems

7.4 Summary

It can be concluded that the ASHP water heaters demonstrated an excellent COP. The ASHP water heaters were capable of also using lesser or almost the same time in heating water to its set point temperature, but with an average power consumption in the range of 30% to 50% relative to that of an electric geyser. It can also be affirmed that the average COP of the split type ASHP water heater without an electric backup was better than that of the integrated type ASHP water heater with an electric backup. Overall, the performance of both systems was higher in the summer than in the winter season. The established multiple linear regression models had good determination coefficients and exhibited good fits with the calculated COP of both types of ASHP water heaters. The models were simple to apply and weather data from a nearby meteorological station which was obtained by logging at five-minutes interval could be used to predict the COP of both the installed ASHP water heaters in that location. Finally, using the reliefF algorithm test, it could be demonstrated that the predictor ($T_s - T_a$) contributed the most by weight of importance to the COP of the split and integrated type ASHP water heaters.

Chapter Eight

Evaluation of performance of air source heat pump water heaters via the surface fitting models

Abstract

Modelling of the coefficient of performance of an air source heat pump water heater can lead to optimisation and prediction of its performance. The study focused on the utilisation of surface fitting models to predict the COPs of a 150 L split type ASHP water heater without an electric backup element and a 150 L integrated type ASHP water heater with an electric backup element. A robust and accurate data acquisition system (DAS) was employed to measure the predictor parameters [E (electrical energy consumed) and □ (product of ambient temperature and relative humidity)] as well as the thermal properties to enable the computation of the COP during the vapour compression refrigeration cycles (VCRC) of the ASHP unit. It was observed that for both systems, the two predictors were primary factors. The surface fitting models for both systems showed that the COP increases with an increase in E by a rate of 0.30 and 0.28 /kWh for the split and integrated type systems, respectively. The models were simple and can be used to predict the COP of both systems with over 95% confidence level, and the determination coefficient of the split and integrated systems were 0.917 and 0.902, respectively. It was also depicted that the COP variation with the predictors in the controlled volume of hot water drawn off (50, 100 and 150 L) under different ambient conditions can be accurately predicted with either the 3D mesh plots or the 2D multi contour plots simulation. **Keywords:** Coefficient of performance, 3D mesh plot, 2D multi contour plots simulation.

8.1 Introduction

The ASHP water heater is an efficient and a renewable energy device for sanitary hot water production (Morrison. *et al*., 2004). The excellent efficiency for an ASHP water heater is due to its performance characteristics known as COP (De Swardt and Meyer, 2001). The COP of an ASHP water heater can range from 2 to 4 and depends on the component design of the system, ambient weather conditions (ambient temperature, relative humidity, etc.), duct space and the speed of the cold expelling air (Levins, 1982; Bodzin, 1997). The optimal COP of an ASHP water heater can be attained by an effective installation of the system (Douglas, 2008). But, it can be explained that the optimal COP of ASHP water heater could even be predicted from the utilisation of an accurate mathematical model. Notwithstanding, the COP can also be enhanced by the use of a primary refrigerant of an excellent thermo-physical property (Hashimoto, 2006; Maruyama, 2008). Salient and thorough exposition and analysis regarding the refrigeration cycle of heat pump water heaters has been presented by Ashdown (2004) and Sinha and Dysarkar (2008).

It is crucial to emphasise that extensive research has been conducted on the mathematical modelling of the performance of heat pump water heaters, but on either of the types of ASHP water heaters and not on both simultaneously. More elaborately, the performance of a heat pump water heater was simulated using the TRNSYS simulation software package (Klein, 1976). However, it was noted that the TRNSYS simulation application could not effectively model the performance of an ASHP water heater as a result of the complexity of the metal fins encapsulating the evaporator. An analytic mathematical model was also presented to predict the COP of a solar assisted heat pump water heater in correlation to temperatures (Ito *et al.*, 1999). A quantitative method can be used to compute the COP of an ASHP water heater based on the quantity of electrical energy consumed by the ASHP system and the thermal energy gained by the stored water (Tangwe *et al.*, 2015). Precisely, Tangwe et al*.* (2013) developed and built surface fitting regression models to predict the performance of a residential split type ASHP water heater under first-hour heating rating, standby losses and heating cycles due to hot water drawn off. Modelling of the residential air source heat pump (ASHP) water heater performance can provide an in-depth analysis of the dynamic behaviour of the coefficient of performance (COP). A mathematical model often employs the use of mathematical equations or a computational algorithm to correlate predictor to desired response (Bush and Mosteller, 2006). These mathematical models were developed and built with the electrical energy consumption and the ambient conditions (product of ambient temperature and relative humidity) data as the predictors. The multiple linear surface fitting model is an advanced regression model that ensures that predictors are forcefully fit to the desired response. The ASHP water heater optimal COP can be achieved from the efficient installation and the mathematical modelling perspective.

The residential ASHP water heater technology is fast gaining maturity in the market and can be classified into two categories; namely, the split and the integrated types. A survey conducted on the COP of the integrated and split type ASHP water heaters both without backup electric element revealed that the former performed better than the latter (Hepbasli and Kalinci, 2009). The study concentrated on the development and building of linear surface fitting models of the COP of ASHP water heaters (split type comprising of an ASHP unit of 1.2 kW power input and a 150 L kwikot high-pressure geyser with its 3 kW element disabled and an integrated type with a backup electric element of 0.5 kW and of 0.9 kW power input with a storage tank of 150 L). Nevertheless, the ASHP water heater technologies were among those accredited residential systems approved and roll-out by South Africa electricity supply utility (Eskom) during the residential ASHP rebate scheme (Zhang and Huan, 2013). The COP of both ASHP water heaters under the different controlled volume of hot water drawn off were mathematically modelled using the derived multiple linear surface fitting response models correlating the predictors and response [product of ambient temperature and relative humidity (□), electrical energy consumption (E) and the COP] during the vapour compression refrigeration cycles (Coleman and Li, 1996). Two-dimensional multi contour plots simulation on the MATLAB statistical toolbox were used to further illustrate the graphical observation of the COP variation to a specific predictor with the others held constant (Chapoutot and Martel, 2008; MathWorks Inc, 2012). The derived models could be used to predict the COP of the two types of ASHP water heaters under the different operation condition of the input parameters. The COP of the heating cycles of both types of ASHP water heaters under these scenarios has never been compared from the perspective of mathematical modelling. In addition, by application of the built and developed surface fitting response models, it can be deduced that the split type ASHP water heater without an electric backup was performing better than the integrated type with an electric backup. Finally, due to the better COP of both types of ASHP water heaters during the summer period, additional analyses such as surface fitting 3D plots and 2D multi contour plots simulation were also conducted for this specific season. These analyses would provide more insight into the correlation of the predictors to the COP.

The research designed and method is provided in chapter three in section 3.3 and Figure 3.4 showed the experimental set up with the exemption of the geyser and its associated metering sensors. The research methods were grouped into three sections; namely, experimental, development of the multiple linear surface fitting models and employing of 2D multi contour plots simulation to show variation of predictors with COP.

8.1.1 Development of the surface fitting models to compare performance

All the measured data were averaged into five-minute interval during the heating cycles of each of the ASHP water heaters. The stored data for the parameters (average ambient temperature, average relative humidity, average power consumption and average time of operation) associated with the predictors and the volume of the water heated during the different heating up scenarios were determined. The multiple linear surface fitting model was derived to correlate the inputs to the output parameters (Chatterjee and Hadi, 1986; Robnik-Sikonja and Kononenko, 2003). The developed and built multiple linear surface fitting models for each of the ASHP water heaters were used to predict both the summer and winter modelled COPs of the specific system. The results of the modelled COPs were compared to that of the calculated COPs, to test for the model's accuracy.

8.1.2 2D multi contour plots simulation to show variation of inputs with the COP

The 2D multi contour plots simulation also termed the two-dimensional linear simulation plots from the statistics toolbox of MATLAB was invoked and utilised as the platform to show how specific independent predictor changed with COP of the different type of ASHP water heater while the other input parameters were held constant for the summer periods. The two-dimensional multi contour plots simulation can be used to illustrate the variation of specific input parameter and the desired response for up to 13 predictors (Math Works Inc, 2012).

8.2 Calculations and theory

The total electrical energy consumed during heating cycle is given by Equation 8.1.

$$E = \sum_{i=1}^{n} P_i t \qquad (8.1)$$

Where;

E = Electrical energy consumed in kWh over the heating cycle

P_i = Average power consumption every 5 minutes during VCRC in kW

t = Time interval of 5 minutes n = Number of successive 5 minutes interval over a period of VCRC

The total thermal energy gained by the hot water in the storage tank is given by Equation 8.2.

$$Q = \sum_{i=1}^{n} c m_i (T_{out(i)}\ T_{in(i)}) \qquad (8.2)$$

Where;

Q = Thermal energy gained in kWh over the heating cycle

m_i = Mass of water heated every 5 minutes during VCRC c = Specific heat capacity of water in kJ/kg℃

$T_{out(i)}$ = Split type, ASHP outlet average water temperature every 5 minutes in ℃

$T_{in(i)}$ = Split type, ASHP inlet average water temperature every 5 minutes in ℃

n = Number of successive 5 minutes interval over a period of VCRC

The parameter □ (average of the product of the ambient temperature and relative humidity every 5 minutes interval) over a heating cycle is given by Equation 8.3.

$$\square = \frac{1}{n}\sum_{i}^{n} \left(T_a\right)_i (RH)_i \qquad (8.3)$$

Where;

$T_{a(i)}$ = Average ambient temperature every 5 minutes in ℃

$RH_{(i)}$ = Average relative humidity every 5 minutes in % n = Number of successive 5 minutes interval over a period of VCRC

The ASHP water heater calculated COP is defined as the ratio of the useful output thermal energy gained (Q) by the heated water and the input electrical energy consumed (E). The Equation 8.4 shows the determination of COP for an ASHP water heater.

$$COP_{cal} = \frac{Q}{E} \qquad (8.4)$$

Where;

COP_{cal} = Calculated COP

The multiple linear surface fitting model of the COP correlating E and □ is given by Equation 8.5. The parameters E and □ are the predictors.

$$COP_{mod} = \square_0 + \square_1 \square + \square_2 E \qquad (8.5)$$

Where;

COP_{mod} = Modelled COP of the ASHP water heaters

$\square_0$ = Forcing constant

$\square_1$ = Scaling constant for $\square$ in (℃%)$^{-1}$

$\square_2$ = Scaling constant for E in (kWh)$^{-1}$

The thermal energy gained by stored water in the split type ASHP water heater was considered to be equal to that gained by the integrated type ASHP water heater. This was based on the fact that both hot water systems were set to the same temperature and were of equal tank capacity.

8.3 Results and discussion

The performance of the residential split and integrated type ASHP water heaters were monitored for the period, October 2015 to September 2016. The results were critically analysed under three scenarios; where the heating cycle occurred due to 150, 100 and 50 L of hot water drawn off from each of the systems. The two systems were forced to start their heating cycles simultaneously.

8.3.1 Summer experimental comparisons of energies and COP

Table 8.1 shows the average thermal energy generated, the average electrical energy consumed and the COP during the entire 50, 100 and 150 L of hot water drawn off from the two types of ASHP water heaters in the summer period (October 2015-April 2016).

Table 8.1: Summer comparisons of the two systems based on energy and COP

ASHP system	Volume of water drawn off L	Electrical energy kWh	Thermal energy kWh	COP
Split	50.0	0.8067	2.1200	2.6280
Integrated	50.0	0.9500	2.1200	2.2316

Split	100.0	1.3600	4.1167	3.0270
Integrated	100.0	1.5670	4.1167	2.6271
Split	150.0	1.7467	5.5767	3.1950
Integrated	150.0	1.9543	5.5767	3.0013

It was realised that for a specific corresponding volume of hot water drawn off, the consumed electrical energy for the split system was lower than that of the integrated system. This could be due to the longer time taken by the integrated system during the heating cycle. The average electrical energy consumed at 50 L hot water drawn off were 0.8067 and 0.9500 kWh, while the average time taken was 40.34 and 67.06 minutes for the split and integrated systems, respectively. The average electrical energy consumed for the 100 L hot water drawn off and the average duration was 1.3600 kWh and 68.00 minutes for the split type system and 1.5670 kWh and 110.61 minutes for the integrated system. In the 150 L hot water drawn off scenario, the average electrical energy consumed and time for the heating cycle were also 1.7467 kWh and 87.33 minutes for the split type system and 1.9543 kWh and 137.95 minutes for the integrated system. The average COP of the split type in the entire heating cycles was 2.9500 while the integrated type system recorded a COP of 2.6200. The COP of the two systems under the different scenarios were above 2 on average and increased as the volume of hot water drawn off was increased (Levins, 1982; Bodzin, 1997).

8.3.2 Winter experimental comparisons of energies and COP

Table 8.2 shows the average thermal energy gained, the average electrical energy consumed and the COP during the entire 50, 100 and 150 L hot water drawn off from the two types of ASHP water heaters in the winter period (May 2016-September 2016).

Table 8.2: Winter comparisons of the two systems based on energy and COP

ASHP system	Volume of water drawn off L	Electrical energy kWh	Thermal energy kWh	COP
Split	50.0	1.1564	2.6541	2.4990
Integrated	50.0	1.5635	2.6540	2.0930
Split	100.0	1.5994	4.9141	2.9230
Integrated	100.0	2.1612	4.9141	2.2940
Split	150.0	1.9091	5.9144	3.0980
Integrated	150.0	2.2798	5.9144	2.5943

It was observed that at a specific corresponding volume of hot water drawn off, the electrical energy consumed by the split system was lower than that of the integrated system just like in the summer period. Also, during the winter period, both the electrical and thermal energies for the two types of ASHP water heaters were higher compared to the summer scenarios with regards to the same volume of hot water drawn off. The average COPs of the two types of ASHP water heaters were lower in the winter periods owing to the drop in ambient temperatures. The average COP of the split type in the entire heating cycles for the winter season was 2.840 in contrast to 2.330 noted in the integrated type system.

8.3.3 Development of the mathematical models of the systems COP for summer

More than 100 datasets of the predictors and COP for each of the systems were used to develop and build a multiple linear surface fitting model to establish a correlation between the inputs and output parameter for the summer period. Equation 8.5 is the derived multiple linear surface fitting response equation used. Table 8.3 shows the mathematical model forcing and scaling values for the split type ASHP water heater. The modelled equation of the split type ASHP water heater indicates that the electrical energy consumption contributed significantly to the COP. It can also be predicted that increase in E would result in a corresponding increase in the COP.

Table 8.3: Summer scaling and forcing constants of the split system

Predictors	Symbols	Scaling notations	Scaling Values	Output
Forcing constant		$\square_0$	3.36800	
Product of ambient condition (T_a, RH)	□	$\square_1$	-0.00050	COP
Electrical energy consumption	E	$\square_2$	0.29900	

From the modelled equation scaling constants shown in Table 8.3, it can be shown that increase in □ may likely result in decrease in COP at a rate of 0.0005 /℃%. An increase in E would lead to a corresponding increase in the COP of the split type ASHP water heater at a rate of 0.299 /kWh. The forcing constant (3.368) is the arbitrary lump constant that catered for the contribution made by other predictors to the output, although not included in the derived model.

The modelled and calculated COPs of the split type ASHP water heater had a determination coefficient of 0.917 and there exists a good fit between the calculated COPs dataset and the predicted modelled curve. Figure 8.1 shows the sample dataset of the calculated COPs and the modelled COP curve for 25 observations involving all the three scenarios of hot water drawn off.

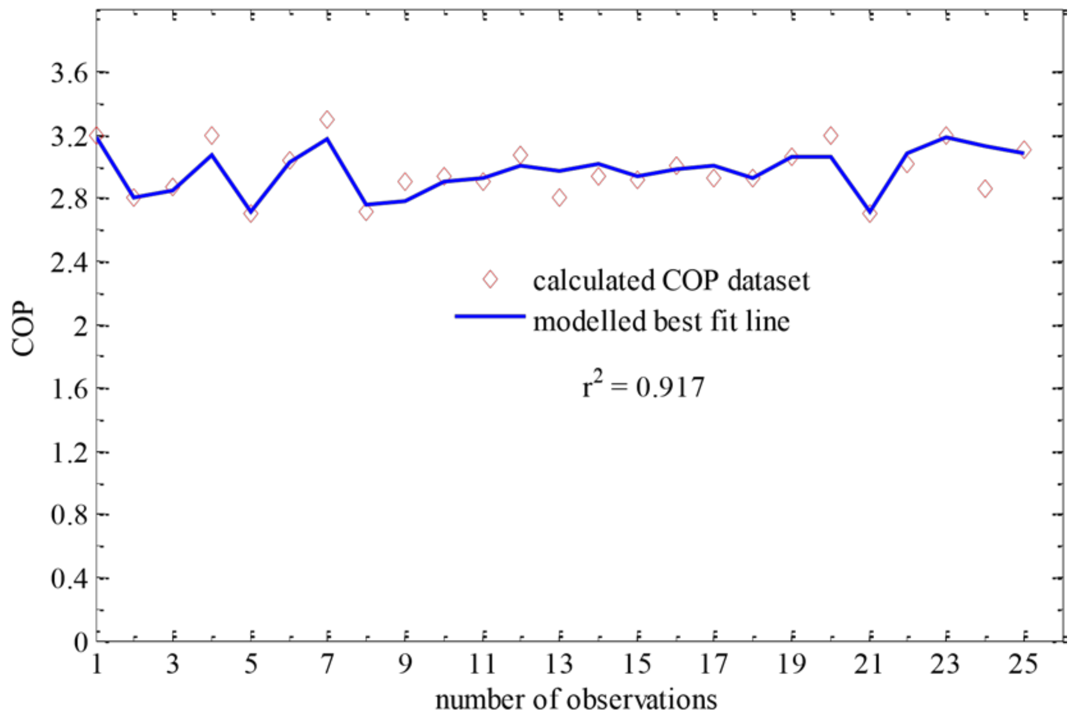


Figure 8.1: Calculated dataset and modelled curve fit for the split type COP

Table 8.4 shows the mathematical model forcing and scaling values for the integrated type ASHP water heater. The modelled equation of the integrated type ASHP water heater justified that the electrical energy consumption had a significant contribution to the COP. It can also be shown that increase in E would result in a corresponding increase in the COP.

Table 8.4: Summer scaling and forcing constants of the integrated system

Predictors	Symbols	Scaling notations	Scaling Values	Output

Forcing constant		$\square_0$	2.31800	COP
Product of ambient condition (T_a ,RH)	$\square$	$\square_1$	-0.00005	
Electrical energy consumption	E	$\square_2$	0.28000	

The modelled equation scaling constants shown in Table 8.4, imply that increasing $\square$ may likely result in a decrease in COP at a rate of -0.00005 /℃%. Similarly, an increase in E would result in a corresponding increase in the COP of the ASHP water heater at a rate of 0.280 /kWh. The forcing constant (2.318) took care of the contribution by other predictors to the output, though the predictors were not included in the derived model.

The modelled and calculated COPs of the integrated type ASHP water heater had a determination coefficient of 0.902 and there exists a good fit between the calculated COPs dataset and the predicted modelled curve. Figure 8.2 shows the sample dataset of the calculated COPs and the modelled COP curve for 22 observations involving all the three scenarios of hot water drawn off.

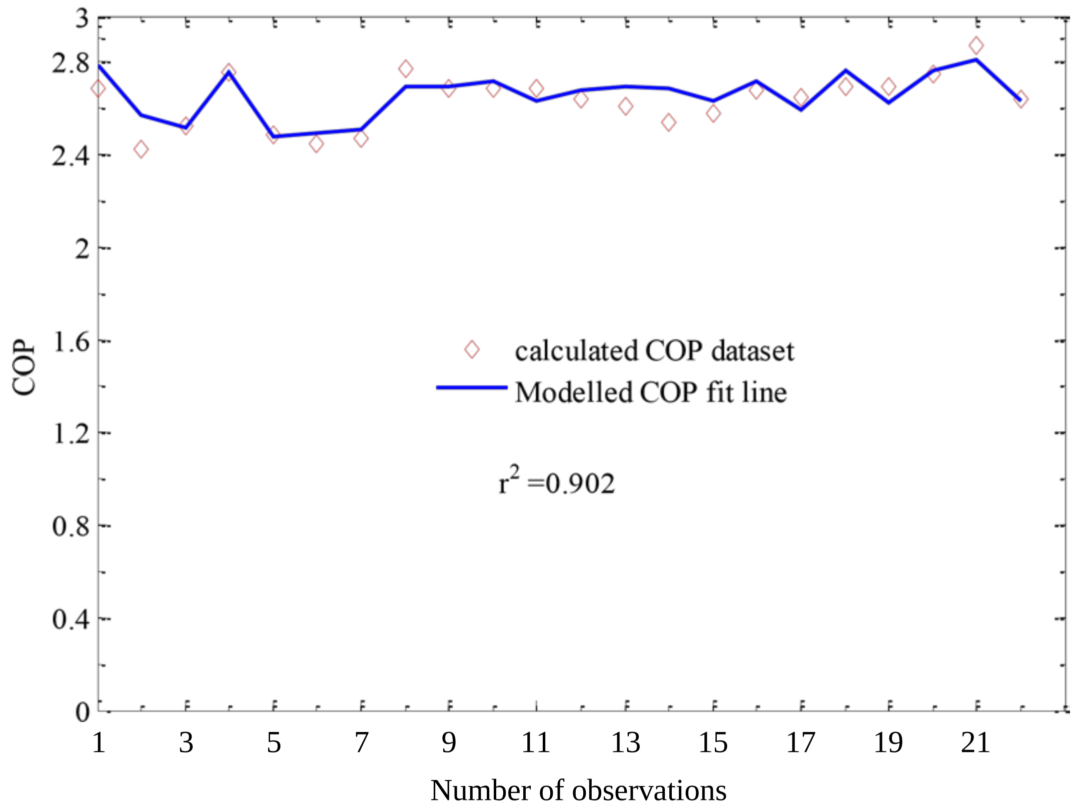


Figure 8.2: Calculated dataset and modelled curve for the integrated type's COP

8.3.4 Development of the mathematical models of the systems COP for winter

More than 100 datasets of the predictors and COP for each of the systems were used to develop and build a multiple linear surface fitting model to establish a correlation between the predictors and response for the winter period. The derived multiple linear surface fitting response model in Equation 8.5 was used to determine the forcing and scaling constants. Table 8.5 shows the mathematical model forcing and scaling values for the split type ASHP water heater. The modelled equation of the split type ASHP water heater demonstrated that the electrical energy consumption had a significant contribution to the COP. It can also be predicted that increase in E would result in a corresponding increase in the COP due to the associated positive scaling constant.

Table 8.5: Winter scaling and forcing constants of the split type system

Predictors	Symbols	Scaling notations	Scaling Values	Output
Forcing constant		$□_0$	1.81254	COP
Product of ambient condition (T_a, RH)	□	$□_1$	-0.00034	
Electrical energy consumption	E	$□_2$	0.90242	

From the modelled equation scaling constants shown in Table 8.5, it can be alluded that increase in □may likely result in a decrease in COP at a rate of 0.00034 /℃%. Equally, an increase in E would lead to a corresponding increase in the COP of the split type ASHP water heater at a rate of 0.90242 / kWh. The forcing constant (1.8125) handled the contribution offered by the other predictors to the output (COP) even though they are not included in the derived model. The determination coefficient and the root mean bias errors of the modelled and calculated COPs for the split type ASHP water heater was 0.912 and 0.044, respectively.

Table 8.6 shows the mathematical model forcing and scaling values for the integrated type ASHP water heater. The modelled equation of the integrated type ASHP water heater demonstrated that the electrical energy consumption had a significant contribution to the COP. It can also be shown that increase in E would most probably result in a corresponding increase in the COP due to its attributed positive scaling constant.

Table 8.6: Winter scaling and forcing constants of the integrated system

Predictors	Symbols	Scaling notations	Scaling Values	Output

Forcing constant		□0	0.81155	
Product of ambient condition (T_a ,RH)	□	□1	0.00106	COP
Electrical energy consumption	E	□2	0.16261	

It can be observed from the modelled equation scaling constants shown in Table 8.6 that increase in □could result in an increase in COP at a rate of 0.00106 /℃%. Also, an increase in E would lead to a corresponding increase in the COP of the integrated type ASHP water heater at a rate of 0.16261 /kWh. The forcing constant (0.8116) accommodated for the contribution offered by other predictors to the COP, although the predictors were not considered in the derived model. The determination coefficient and the root mean bias errors of the modelled and calculated COPs for the integrated type ASHP water heater was 0.901 and 0.047.

8.3.5 Summer surface 3D plots derived by fitting of dataset and modelled COP

Over 100 datasets of the predictors (data values of each predictor within the experimentally determined ranges) for both the integrated and split type ASHP water heaters were generated and used to forecast the predicted modelled COP. The mesh plot of the generated predictors and modelled COP was established on a 3D plot. The actual samples of dataset of the determined predictors and the calculated COP of the two systems were plotted on the same 3D plots. Figure 8.3 shows the 3D plot of the surface fitting mesh plot of the modelled COP and the sample calculated COP for the split type ASHP water heater.

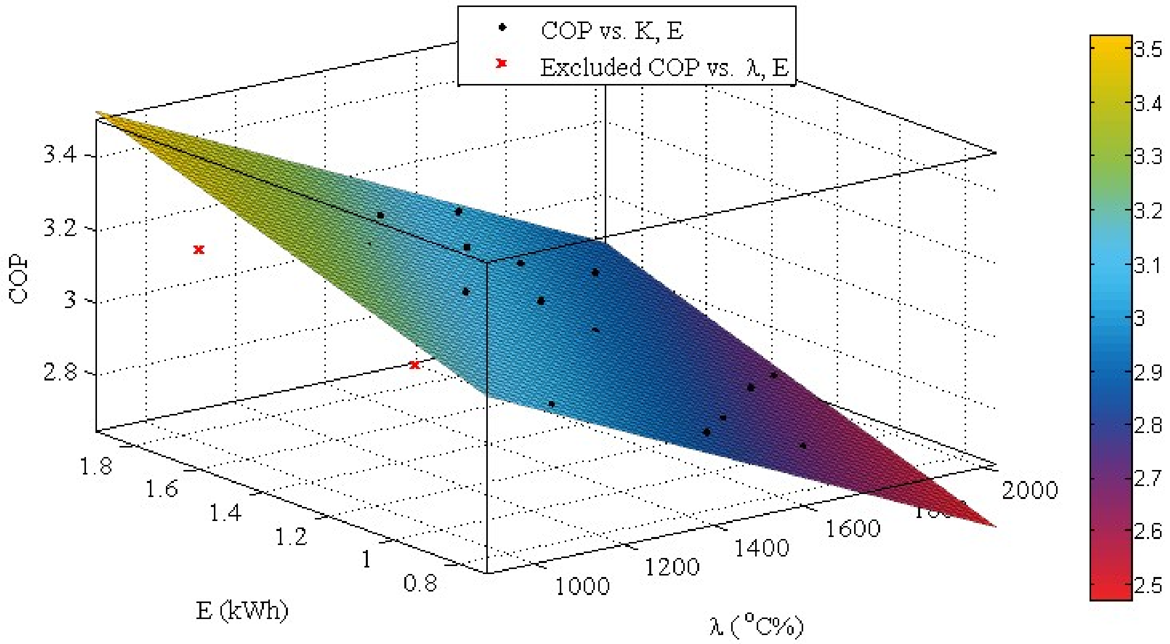


Figure 8.3: 3D mesh modelled and calculated COP for the split type ASHP water heater

As shown in Figure 8.3, □, E and COP are placed on the x-axis, y-axis and zaxis, respectively. The visual representation shows the actual calculated COP and the best fit of the surface mesh of the modelled COP. It should be noted that the black dotted points represent the data for both predictors and determined COP that fitted with the modelled surface mesh. The red cross markers were outlier data points and these were excluded from the derivation of the determination coefficient. It can also be depicted that at constant □, any increase in E was followed by an increase in COP at the rate of 0.3 /kWh. The potential decrease in □could result in an increase on the COP at a rate of -0.0005 /℃% provided E was held constant.

Figure 8.4 shows the 3D plot of the surface fitting mesh plot of the modelled COP and sample calculated COP for the integrated type ASHP water heater. It harbours □, E and COP on the x-axis, y-axis and z-axis respectively.

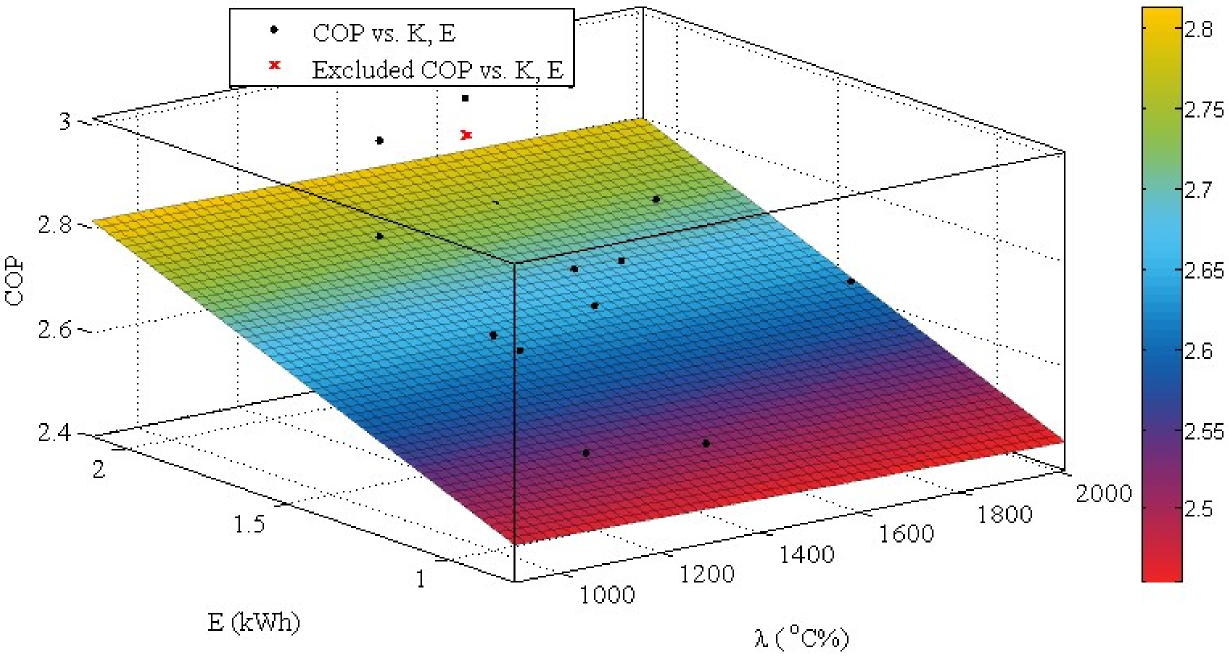


Figure 8.4: 3D mesh modelled and calculated COP for the integrated ASHP water heater

From the Figure 8.4, it can be depicted that at constant □, any increase in E was followed by an increase in COP at the rate of 0.288 /kWh. There was minimal rate of change of -0.00005 /℃% of □to the COP provided E was held constant. Furthermore, it could be alluded that a decrease in □could result in a minimal increase in the COP because of the negligible negative slope between □and COP.

8.3.6 Summer models multi contour plots simulation for the ASHP water

heaters

The multi contour plots simulation are a multi-two-dimensional plots used to model the variation of a specific predictor with the output using any given multiple linear regression models while the other predictors are held constant (MATLAB, 2012). These 2D multi contour plots simulation can be employed for up to thirteen predictors. In this study, the 2D multi contour plots simulation were used to visualise the variation of the electrical energy consumed (E) with the calculated COP for a constant □, for both split and integrated type ASHP water heaters. Likewise, to show how the predictor (□) varied with the COP while E was kept constant. Figure 8.5 shows the multi contour plots simulation for the split type ASHP water heater. The positive slope of E indicated that increase in predictor could result in an increase in the COP. The green lines on both plots show the linear relationship between the predictors and the COP while both broken red curves defined the 95% confidence bound. The slopes of the modelled COP with respect to □and E were -0.0005 /^{o}C% and 0.300 /kWh as determined from the derived mathematical model.

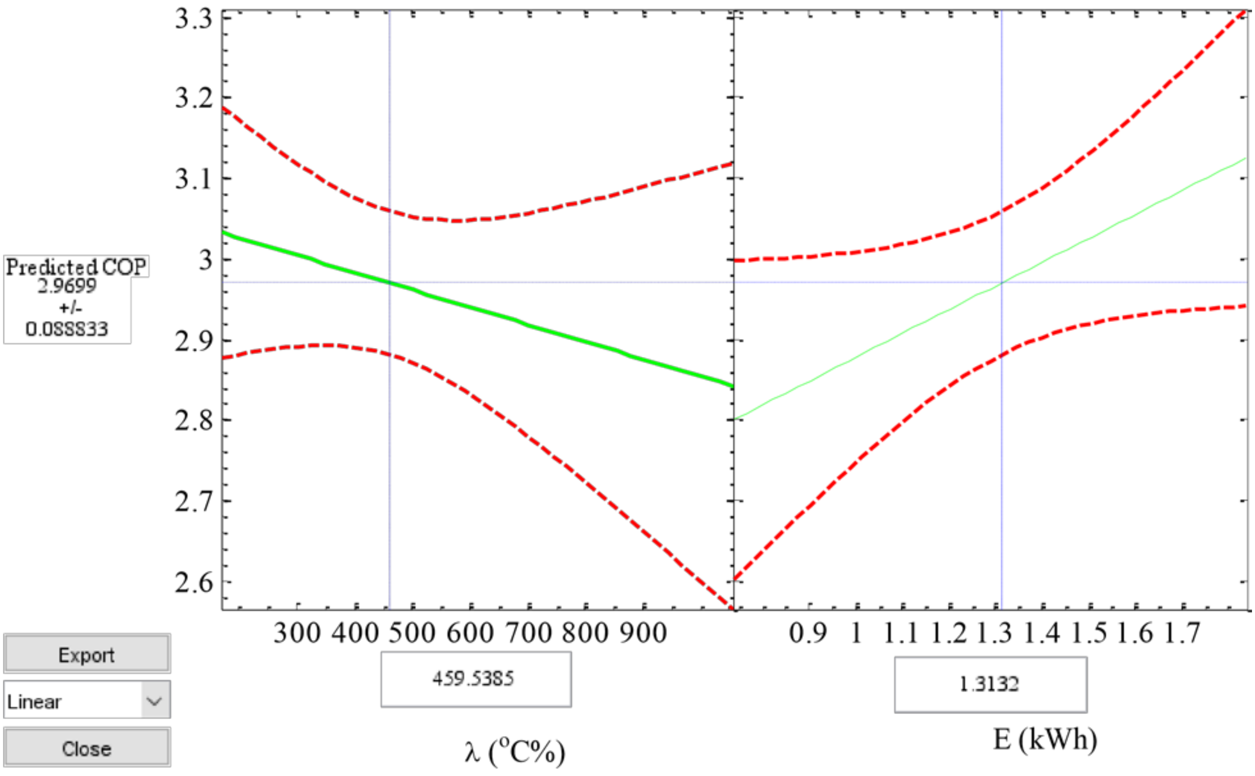


Figure 8.5: 2D multi contour plots simulation of predictors and COP for the split type ASHP water heater

Figure 8.6 demonstrates that under the drawn off scenarios, the predictor (□) increase with a decrease in the modelled COP of the integrated type ASHP water heater provided E was held constant. This is in agreement with the scaling coefficient obtained from the derived mathematical model represented in Equation 8.5. The calculated slopes for the modelled COP of the integrated system with respect to □and E were -0.00005 /°C% and 0.288 /kWh, respectively.

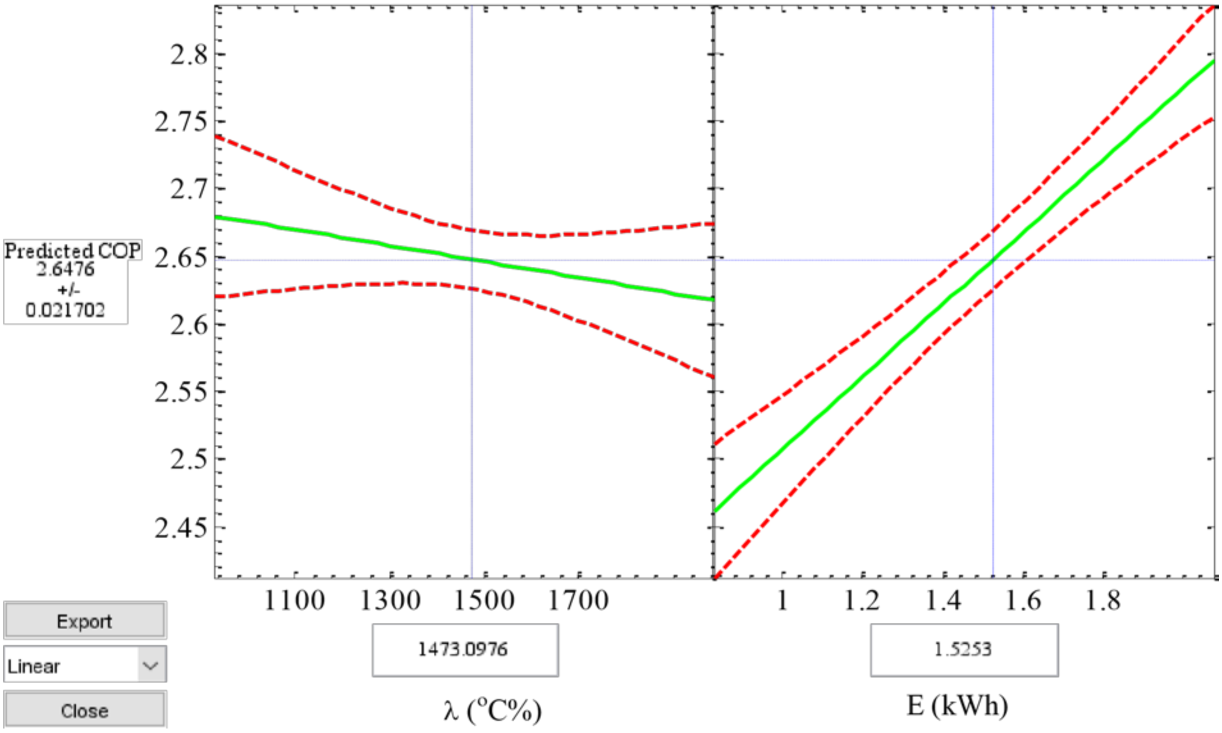
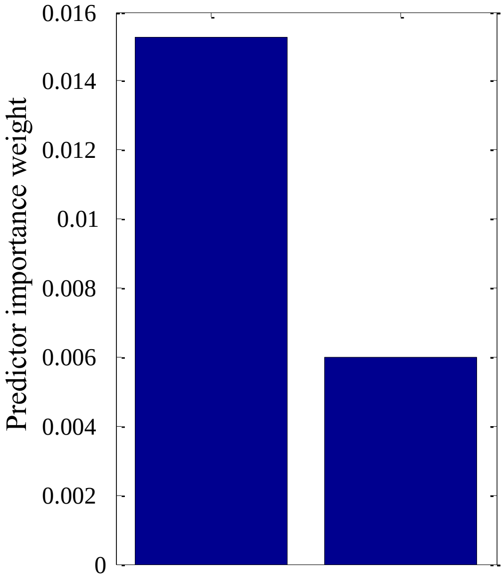


Figure 8.6: 2D multi contour plots simulation of predictors and COP for the integrated type ASHP water heater

8.3.7 Predictors ranking using ReliefF test for the summer period The two predictors (□ E) and the output (COP) from the processed data of the split and integrated type ASHP water heaters were used in the ReliefF algorithm to rank predictors according to their importance of weight contribution to the desired response. The ReliefF test is a statistical tool that uses the regression method to rank predictors with respect to their importance of weight contribution to the output (Robnik-Sikonja and Kononenko, 2003). The weighted rank for a particular predictor can be between -1 and 1. A positive weight rank of a predictor shows that it is a primary factor while a negative weight rank depicts that it is a secondary factor. Figure 8.7 shows the reliefF bar plots for the predictors and the importance of weight contributions to the COP as per the split and integrated type ASHP water heaters. The weight ranking showed that for both types of ASHP water heaters, the electrical energy consumption (E) and the product of ambient temperature and relative humidity (□) were primary factors. It can also be determined from the analysis that both primary predictors weight contributions with regard to the split type system (E= 0.015 and □= 0.006) were lower than those of the integrated type system (E = 0.043 and □ = 0.023). The impact of the electrical energy consumption contribution owing to the weight importance was the most significant. The contribution by weight of the predictor E is three times more in the integrated type to the split type.



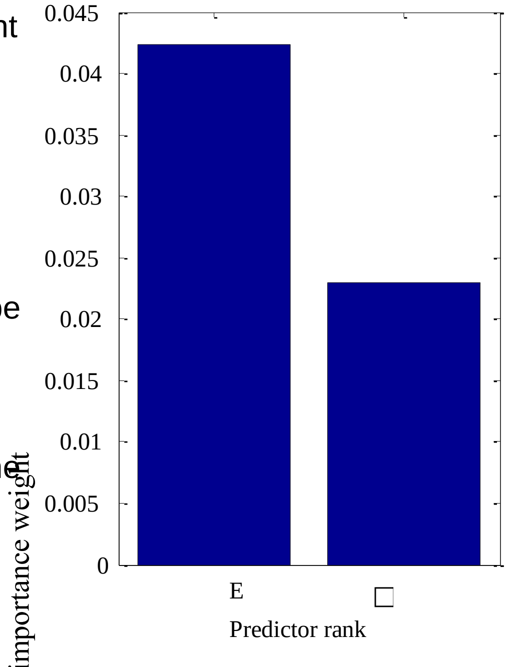


Figure 8.7: ReliefF bar plots of the predictors weight of the ASHP systems

8.4 Summary

It is worth concluding that surface fitting modelling of COP of an ASHP water heater with the aid of 2D multi contour plots simulation can give an in-depth analysis into the performance since it can be visually automated. The increase in the electrical energy consumed for both split and integrated type ASHP water heaters resulted in a corresponding increase in the COP in summer as well as winter. Furthermore, in all the scenarios of hot water drawn off from both systems, both predictors were determined to be the primary factors from the reliefF algorithm test. The weight of importance by the contribution of the predictor (E) to the COP was about 3 times more in the integrated type ASHP water heater compared to that of the split type ASHP water heater. This was so, by virtue of the backup electrical energy consumption of the integrated type system during the heating cycles. The derived determination coefficient from the surface fitting models over a 95% confidence bound was more than 0.9000 and with an excellent fitness between the calculated and modelled COPs for both the split and integrated type ASHP water heaters. The established multiple linear surface fitting models demonstrated that the COP of the split type ASHP water heater (without a backup electric element) was better than that of the integrated type ASHP water heater (with a backup electric element).

Chapter Nine

Dynamic multivariate models and simulation application to predict coefficient of performance of the air source heat pump water heaters

Abstract

Modelling and simulating the performance of air source heat pump (ASHP) water heaters can provide an in-depth understanding of the dynamic behaviour of the coefficient of performance (COP). The primary data used in the building and development of the models were collected from a data acquisition system that was designed and employed to monitor the COP of installed 150 L integrated and split type ASHP water heaters under three scenarios of controlled volume of hot water withdrawal. The study presents both statistical simulation and robust mathematical models developed for the COP of both systems; using the temperature difference of the refrigerant at the compressor suction and discharge ends, the temperature difference of the refrigerant at the inlet and outlet of the condenser, the ambient temperature and the relative humidity as predictors. The results revealed that the split type without electric backup element performed better than the integrated type incorporated with an electric backup element. In addition, all the predictors were important drivers of the COP, and the reliefF algorithm tests depicted that both the ambient temperature and the relative humidity were secondary factors. Furthermore, the predicted COP from the derived mathematical models of both systems demonstrated a significant difference among the COP means of the two types of ASHP water heaters under the operating scenarios.

Keywords: Air source heat pump (ASHP), Coefficient of performance (COP), Mathematical model, ReliefF algorithm test and significant difference.

9.1 Introduction

The dynamic behaviour of the performance of the residential air source heat pump (ASHP) water heater can be determined via mathematical modelling. Traditionally, in a mathematical model, input parameters are correlated to desired output parameter(s) through mathematical equations or by the use of a computational algorithm (Bush and Mosteller, 2006). Modelling of the COP of ASHP water heaters can provide an in-depth analysis of its dynamic behaviour. The unique characteristic of ASHP water heater responsible for its high efficiency, which exemplifies its performance or behaviour is known as COP (De Swardt and Meyer, 2001). The COP of an ASHP water heater ascribes to, the ratio of the quantity of electrical energy consumed to the useful thermal energy gained by the stored water (Tangwe *et al*., 2015). Apparently, the following factors, including the ambient weather conditions, the design of the components that constitute the VCRC closed loop circuit as well as the duct space, are salient parameters noted to influence the COP of an ASHP (Levins, 1982; Bodzin, 1997). In this study, mathematical models were developed and built which involved the temperatures of the refrigerant at critical locations in the closed loop circuit of the vapour compression refrigeration cycle (VCRC) and the data on ambient weather conditions as predictors.

An efficient COP of an ASHP water heater can be achieved by way of conducting experiments and the development of mathematical models (Douglas, 2008). Alternatively, an accurate mathematical model developed under different system operating conditions can be utilised to obtain an optimal COP of an ASHP water heater. Also, the COP of the system can further be increased by the use of a primary refrigerant characterised with an excellent thermo-physical property (Hashimoto, 2006; Maruyama, 2008). Accordingly, Ashdown (2004) and Sinha and Dysarkar (2008), demonstrated in their respective studies, findings that presented a better understanding of the refrigeration cycle that takes place in a heat pump water heater. Above all, research has been conducted on the modelling and simulation of ASHP water heaters with emphasis on only one type of the ASHP water heaters, without the simultaneous monitoring of both systems (Tangwe *et al*., 2017). It is crucial to highlight that there is a dearth of information regarding the mathematical modelling of the performance of both the split and integrated type ASHP water heaters under simultaneous investigation and monitoring (Tangwe and Simon, 2018).

The TRNSYS simulation software was indicated as one of the methods that can be employed in simulating the performance of a heat pump water heater (Klein, 1976). However, it is covered with a fundamental challenge based on the complexity of the auxiliary design of the metal fin enclosing the evaporator that is anticipated for the enhancement of the performance of the system COP. Therefore, the TRNSYS simulation cannot effectively model the performance of an ASHP water heater. The prediction of the COP of ASHP water heaters using the TRNSYS software was of determination coefficient of about 0.9. On the other hand, an analytical, mathematical model was also employed to predict the COP of a solar assisted heat pump water heater (Ito *et al.,* 1999). Specifically, in South Africa, Tangwe and colleagues (2013) developed and built surface fitting multiple linear regression models to predict the performance of a residential split type ASHP water heater under various scenarios of operation of the VCRC.

Based on categories, there exist two residential types of ASHP water heaters; namely, the split and the integrated types. A survey conducted on the COP of the two types of ASHP water heaters demonstrated that the integrated type had a better performance as opposed to the split type ASHP water heater, as long as both systems were without backup electric element and were of the same tank size. A survey study conducted on the COP of the two types of ASHP water heaters revealed that the integrated type performed better than the split type ASHP water heater, wherein both systems were without backup electric element and were of the same tank capacity (Hepbasli and Kalinci, 2009). The study was geared toward the development and building of multiple linear regression models of the COP of ASHP water heaters (split type comprised of an ASHP unit of 1.2 kW power input and a 150 L kwikot high-pressure geyser with its 3 kW element disabled and an integrated type with a backup electric element of 0.5 kW and a storage tank of 150 L). These systems were among the accredited domestic systems approved and roll-out by the South Africa electricity supply utility (Eskom) during the residential ASHP rebate scheme (Eskom, 2011; Zhang and Huan, 2013). The COP of both ASHP water heaters under the different controlled volumes of hot water withdrawal was mathematically modelled using the derived multiple linear regression models which correlated the predictors and the response during the VCRC (Coleman and Li, 1996).The predictors included; ambient temperature, relative humidity, temperature difference of the refrigerant between the compressor discharge and suction ends, temperature difference of the refrigerant between the condenser inlet and outlet.

A two-dimensional multi contour plots simulation on the MATLAB statistical tools were used to further illustrate the graphical observation of the COP variation to a specific predictor with the others held constant (Chapoutot and Martel, 2008; Tangwe and Simon, 2018). The derived models could be used to effectively predict the COP of the two types of ASHP water heaters under different operation conditions of the input parameters. The COP of both types of ASHP water heaters under these scenarios has never been simultaneously compared through mathematical modelling and simulation. Hence, showcased the significant contribution and novelty underlining this study. In addition, a multiple comparison procedure test was also performed to identify any significant difference in the average group COPs for both types of ASHP water heaters under the controlled volumes of hot water drawn off (Hochberg and Tamhane, 1987).

The research design and method implemented is described in chapter three in Section 3.3 and Figure 3.4 showed the schematic layout of the experimental set up with the geyser and its associated installed sensors excluded.

9.1.1 Development and building of mathematical models to compare performance

All the obtained data was averaged into five-minute intervals during the heating cycles of each of the ASHP water heaters. The stored data for the predictors (average ambient temperature, average relative humidity, average of the difference in temperature of the refrigerant at the compressor discharge and suction ends and the average of the difference in temperature of the refrigerant at the condenser inlet and outlet ends), volume of the water heated and electrical energy consumed during the different heating scenarios were determined. The multiple linear regression models were derived to correlate the inputs to the output as per the methods of Chatterjee and Hadi (1986) and Robnik-Sikonja and Kononenko (2003). The developed and built multiple linear regression models for each of the ASHP water heaters were used to predict the modelled COP of the specific systems and the outcomes were compared to those of the calculated COP in order to test for the accuracy of the models.

9.1.2 Simulation plots and statistical analysis to compute the performance

The two-dimensional multi contour plots simulation from the statistics toolbox of MATLAB was invoked and utilised as the platform to show how specific independent predictor changed with the COP of the different types of ASHP water heaters while the other input parameters were kept constant. The twodimensional multi contour plots simulation can be used to illustrate the variation of the specific input parameter and the desired response for up to 13 predictors (MathWorks, 2012, Tangwe and Simon, 2018). The one-way analysis of variance (ANOVA) was employed to determine any significant difference in the average group COP of the different heating scenarios using the ANOVA plots and the p-value according to the method of Hogg and Ledolter (1987). In conclusion, a multiple comparison procedure test was applied to show if the difference in the average group COPs of the two types of ASHP water heaters was of significance (Hochberg and Tamhane, 1987).

9.2 Theory and calculations

The total electrical energy consumed during a heating cycle is given by Equation 9.1.

$$E = \sum_{i=1}^{n} P_i t \qquad (9.1)$$

Where;

E = Electrical energy consumed (kWh)

P_i = Average power consumption in every 5 minutes intervals during VCRC (kW)

t = Time interval of 5 minutes

n = number of successive 5 minutes intervals during VCRC

The total thermal energy gained by the hot water in the storage tank of the split type ASHP water heater is given by Equation 9.2.

$$Q = \sum_{i=1}^{n} cM_i(T_{out} - T_{in})_i \qquad (9.2)$$

Where;

Q = Thermal energy gained by stored water (kWh)

c = Specific heat capacity of water (kJ/kg^oC)

M_i = Mass of water heated (kg)

n = number of successive 5 minutes intervals during VCRC

The parameter ΔT_{cm} (difference in refrigerant temperature between the outlet and inlet of the compressor) is given by Equation 9.3.

$$\Delta T_{cm} = T_{cmo} - T_{cmi} \qquad (9.3)$$

Where;

ΔT_{cm} = Difference in temperature of the refrigerant between the outlet and inlet of the compressor (^oC)

T_{cmo} = Temperature of the refrigerant at the outlet of the compressor (^oC)

T_{cmi} = Temperature of the refrigerant at the inlet of the compressor (^oC)

The parameter ΔT_{cn} (difference in temperature of the refrigerant between the inlet and outlet of the condenser) is given by Equation 9.4.

$$\Delta T_{cn} = T_{cni} - T_{cno} \qquad (9.4)$$

Where;

ΔT_{cn} = Difference in temperature of the refrigerant at the inlet and outlet of the condenser (^oC)

T_{cni} = Temperature of the refrigerant at the inlet of the condenser (^oC)

T_{cno} = Temperature of the refrigerant at the outlet of the condenser (^oC)

The calculated COP of the ASHP water heater is the ratio of the useful output thermal energy gained (Q) by the heated water to the input electrical energy consumed (E).

The Equation 9.5 represents the equation for the determination of the COP of an ASHP water heater.

$$COP_{cal} = \frac{Q}{E} \quad (9.5)$$

The multiple linear regression models of the predictors correlating the COP of the ASHP water heater is given by Equation 9.6.

$$COP_{mod} = \beta_0 + \beta_1 T_a + \beta_2 RH + \beta_3 \Delta T_{cm} + \beta_4 \Delta T_{cn} \quad (9.6)$$

Where;

COP_{mod} = Modelled COP of the ASHP water heaters

T_a = Average ambient temperature in ^oC

RH = Average relative humidity in %

β_0 = Forcing constant

β_1 = Scaling constant for T_a in $(^oC)^{-1}$

β_2 = Scaling constant for RH in $(\%)^{-1}$

□3 = Scaling constant for □T cm in ($^{o}C)^{-1}$

□4 = Scaling constant for □T cn in ($^{o}C)^{-1}$

Considering that the hot water set point temperatures (55^{o}C) and the tank capacity were the same for both systems; the thermal energy gained by storing water in the split type ASHP water heater was assumed to be equal to that gained by the integrated type ASHP water heater.

9.3 Results and discussion

9.3.1 Summer comparison of crucial parameters during operations The crucial parameters that could affect the performance of both types of ASHP water heaters undergoing vapour compression refrigeration cycle under the specific volume of hot water withdrawal were; ambient temperature, relative humidity, in-line cold water temperature, temperature of refrigerant at the inlet and outlet of the compressor and condenser. It was deduced that for specific controlled volume of hot water withdrawal from either tanks in both system, the duration and COP were influenced by changes in the following;

i. Average ambient temperature ii. Average relative humidity iii. Difference in the average temperature of the refrigerant at the outlet and inlet of the compressor iv. Difference in the average temperature of the refrigerant at the inlet and outlet of the condenser

In accordance with the summer period (October 2015-April 2016) during which the study was conducted, the results showed that during the 50, 100 and 150 L hot water drawn off scenarios, the following ranges and the average values of the different key parameters were obtained as represented in Tables 9.1 and

9.2. Table 9.1 shows the minimum and maximum values of each key parameter during the VCRC of the split and integrated type ASHP water heaters.

Table 9.1: Summer comparisons of the minimum and maximum parameter values

ASHP system	Vd L	P kW	Time mins	Tcw °C	Ta °C	RH %	Tcmi °C	Tcmo °C	Tcni °C	Tcno °C
Split-min	50	1.12	45	18.51	18.81	36.72	22.83	61.77	65.50	38.16
Split-max	50	1.30	55	28.53	29.53	86.05	36.56	80.50	82.50	45.22
Integrated-min	50	0.91	60	18.10	18.47	36.88	10.15	52.29	45.54	41.93
Integratedmax	50	0.92	70	28.50	29.60	86.84	14.56	58.71	52.52	46.50
Split-min	100	1.24	70	17.04	17.37	63.40	22.31	70.64	74.54	38.43
Split-max	100	1.29	75	22.73	23.33	86.84	27.26	75.41	78.84	39.69
Integrated-min	100	0.91	100	17.05	17.15	64.33	10.31	50.38	44.24	41.42
Integratedmax	100	0.92	110	22.27	23.27	88.46	12.43	53.46	46.39	41.88
Split-min	150	1.25	65	18.30	18.51	41.50	21.32	69.82	73.77	36.95
Split-max	150	1.32	80	28.67	29.18	79.18	34.29	81.19	84.03	42.51
Integrated-min	150	0.89	90	18.50	18.86	41.52	10.26	50.15	43.47	40.26
Integratedmax	150	0.93	120	28.63	29.23	78.02	15.98	57.84	50.53	43.71

Vd , Volume of water drawn off; P, Electrical power consumed; Tcw , In-line cold water temperature; Ta , Ambient temperature; RH , Relative humidity; Tcmi , Temperature of the refrigerant at the inlet of the compressor; Tcmo , Temperature of the refrigerant at the outlet of the compressor; Tcni , Temperature of the refrigerant at the inlet of the condenser; Tcno, Temperature of the refrigerant at the outlet of the condenser

It should be noted that at 50 L hot water withdrawal from the split type ASHP water heater, the difference between the maximum and minimum values of the power consumption, the time taken, the in-line cold water temperature, the ambient temperature, the relative humidity and the temperature of the refrigerant at the inlet of the compressor, the outlet of the compressor, the inlet of the condenser and the outlet of the condenser were 0.18 kW, 10 minutes, 10.02°C, 10.72°C, 49.33%, 13.73°C, 18.73°C, 17.00°C and 7.06°C, respectively. The difference between the maximum and minimum values of the aforementioned parameters for the counterpart integrated type ASHP system under the same heating cycles and start up time were 0.01 kW, 10 minutes, 10.40ºC, 11.13ºC, 49.96%, 4.41ºC, 6.42ºC, 6.98ºC and 4.57ºC, respectively. Clearly, there was no significant difference between the maximum and minimum values of the two systems during the vapour compression refrigeration cycles, with regards to the temperature at the inlet of the compressor, the outlet of the compressor, the inlet of the condenser and the outlet of the condenser. However, the corresponding values of all the measured parameters were much higher for the split type ASHP water heater. The data obtained during the 100 L hot water withdrawal, showed that the difference in the maximum and minimum values of the all nine parameters specified in the 100 L drawn off for the split type ASHP water heater were 0.05 kW, 5 minutes, 5.69ºC, 5.96ºC, 23.44%, 4.95ºC, 4.77ºC, 4.30ºC and 1.26ºC, respectively. On the other hand, the difference between the maximum and minimum values of the identical measured parameters for the integrated type ASHP system under the same 100 L withdrawal heating cycles with a common start up time were 0.01 kW, 10 minutes, 5.22ºC, 6.12ºC, 24.13%, 2.120ºC, 3.08ºC, 2.15ºC and 0.46ºC, respectively. A minimal difference between the maximum and minimum values was observed for the temperature at the inlet of the compressor, outlet of the compressor, inlet of the condenser and outlet of the condenser for both systems during the duration of the VCRC due to the 100 L hot water withdrawal. Again, the measured parameters were much higher for the split type system. Following the increase in the volume of hot water that was drawn off, there was a corresponding increase in the time used for the respective heating cycles.

The results achieved under the 150 L hot water withdrawal operation, demonstrated that the difference in the maximum and minimum values of the nine parameters into consideration for the split type ASHP water heater were

0.07kW, 10 minutes, 10.37°C, 10.67°C, 37.68%, 12.97°C, 11.37°C, 10.26°C and 5.56°C, respectively. The difference between the maximum and minimum values of the same parameters for the integrated type ASHP water heater under the 150 L withdrawal heating cycles operated under same start up time were 0.04 kW, 30 minutes, 10.13°C, 10.37°C, 36.50%, 5.72°C, 7.69°C, 7.06°C and 3.45°C, respectively.

Table 9.2 shows a comprehensive summary of the average values of each parameter during the three scenarios of hot water withdrawals from the both types of ASHP water heaters.

Table 9.2: Summer comparisons of the average values of key parameters

ASHP system	Vd L	P kW	Time mins	Tcw °C	Ta °C	RH %	Tcmi °C	Tcmo °C	Tcni °C	Tcno °C
Split	50.0	1.21	50.00	21.54	22.54	65.48	27.47	71.64	74.72	42.05
Integrated	50.0	0.92	66.66	21.74	22.74	64.86	12.18	54.71	48.04	43.69
Split	100.0	1.27	73.33	20.68	21.08	73.92	24.90	72.71	76.42	39.00
Integrated	100.0	0.91	106.66	20.79	21.13	73.48	11.67	52.34	45.49	41.58
Split	150.0	1.29	70.00	22.69	23.69	58.78	27.76	76.00	79.54	39.97
Integrated	150.0	0.92	101.66	22.68	23.68	58.88	13.03	54.15	46.84	41.82

Vd , Volume of water drawn off; P, Electrical power consumed; Tcw , In-line cold water temperature; Ta , Ambient temperature; RH , Relative humidity; Tcmi , Temperature of the refrigerant at the inlet of the compressor; Tcmo , Temperature of the refrigerant at the outlet of the compressor; Tcni , Temperature of the refrigerant at the inlet of the condenser; Tcno, Temperature of the refrigerant at the outlet of the condenser

It can be depicted that the average power consumption of the integrated system (0.92 kW) was lower as compared to that of the split system (1.26 kW) throughout the heating cycles by 0.34 kW. The average time difference through the entire heating cycle was 81 minutes, but the integrated system was operated for a longer period. The average temperature of the refrigerant at the inlet and outlet of the compressor in the split and integrated type ASHP water heaters were (26.71 and 73.45°C) and (12.29 and 53.73°C), respectively. The average difference in the temperature of the refrigerant at the inlet and outlet of the compressor in both systems (the difference in the refrigerant temperature at the outlet and inlet of the compressor) was 5.3^{o}C and was much higher in the split type (46.74^{o}C) than in the integrated type (41.44^{o}C). The average temperatures of the refrigerant at the inlet and outlet of the condenser in the split and integrated type ASHP water heaters were (76.89 and 42.36^{o}C) and (46.79 and 40.34^{o}C), respectively. The average difference in the temperature of the refrigerant at the inlet and outlet of the condenser in the two types of ASHP water heaters was 30.10 ^{o}C and was much higher in the split type (34.53^{o}C) than in the integrated type (4.43^{o}C). It could be observed from Table 9.2 that the average energy consumption increased as the volume of hot water withdrawn increased from 50 L to 100 L and to 150 L from both systems. Finally, the averages in the ambient temperatures (22.51 and 22.43^{o}C), the relative humidity (65.74 and 66.06%) and the in-line cold water temperatures (21.73 and 21.64^{o}C) were almost the same for the two systems during the entire VCRC scenarios.

9.3.2 Winter comparison of crucial parameters during operations The winter period results (May 2016-August 2016) during which the research was conducted depicted that during the 50, 100 and 150 L hot water withdrawal scenarios, the following range and the average values for the different key parameters were obtained as presented in Tables 9.3 and 9.4. Table 9.3 shows the minimum and maximum values of each key parameter during the VCRC obtained in the split and integrated types ASHP water heaters.

Table 9.3: Winter comparisons of the minimum and maximum parameter values

ASHP system	Vd L	P kW	Time mins	Tcw ^{o}C	Ta ^{o}C	RH %	Tcmi ^{o}C	Tcmo ^{o}C	Tcni ^{o}C	Tcno ^{o}C
Split-min	50.0	1.09	60.00	12.21	12.64	45.21	12.22	59.15	62.60	37.22
Split-max	50.0	1.21	80.00	20.50	21.17	85.08	28.81	74.91	77.33	42.52
Integrated-min	50.0	0.87	80.00	12.00	12.12	44.81	5.21	45.33	39.32	40.14
Integratedmax	50.0	0.94	140.0	20.20	21.36	82.60	11.71	55.05	48.63	44.89
Split-min	100.0	1.18	80.00	12.30	12.81	60.55	16.01	63.26	66.79	36.66
Split-max	100.0	1.24	95.00	18.10	18.62	88.58	23.13	69.49	72.77	37.85
Integrated-min	100.0	0.85	140.0	12.10	12.25	60.42	3.29	44.67	38.10	39.11
Integratedmax	100.0	0.87	185.0	17.80	18.40	84.03	8.47	49.28	41.76	39.57
Split-min	150.0	1.17	90.00	16.13	16.43	39.40	19.38	66.06	69.91	34.83
Split-max	150.0	1.26	110.0	24.29	24.79	72.25	30.77	76.86	79.71	39.03
Integrated-min	150.0	0.81	145.0	16.00	16.14	40.94	4.79	44.45	37.70	35.94
Integratedmax	150.0	0.86	190.0	23.50	24.21	78.84	9.86	51.77	44.92	39.98

Vd , Volume of water drawn off; P, Electrical power consumed; Tcw , In-line cold water temperature; Ta , Ambient temperature; RH , Relative humidity; Tcmi , Temperature of the refrigerant at the inlet of the compressor; Tcmo , Temperature of the refrigerant at the outlet of the compressor; Tcni , Temperature of the refrigerant at the inlet of the condenser; Tcno, Temperature of the refrigerant at the outlet of the condenser

It was noted that at 50 L hot water withdrawal, the difference between the maximum and minimum values of the nine parameters (as presented in the 50 L drawn off scenario during the summer period) of the split type ASHP water heater were 0.12 kW, 20 minutes, 8.29^{o}C, 8.53^{o}C, 39.87%, 16.59^{o}C, 15.76^{o}C, 14.73^{o}C and 5.30^{o}C, respectively. The difference between the maximum and minimum values of the mentioned parameters for the integrated type ASHP system under the same heating cycles and start up time were 0.07 kW, 60 minutes, 8.20^{o}C, 9.24^{o}C, 37.79%, 6.50^{o}C, 9.72^{o}C, 9.31^{o}C and 4.75^{o}C, respectively. There was no significant difference between the maximum and minimum values for the two systems during the VCRC, with regards to the temperature of the refrigerant at the inlet of the compressor, outlet of the compressor, inlet of the condenser and outlet of the condenser. The corresponding values of all the parameters were much higher in the split type ASHP water heater.

The data achieved during the 100 L hot water withdrawal, demonstrated that the difference in the maximum and minimum values of the nine parameters in the split type ASHP water heater were 0.06 kW, 15 minutes, 5.80^oC, 5.81^oC, 28.0%, 7.11^oC, 6.22^oC, 5.97^oC and 1.19^oC, respectively. The difference between the maximum and minimum values of the described parameters in relation to the integrated type ASHP system under the same 100 L withdrawal heating cycles with a common starting time were 0.02 kW, 45 minutes, 5.70^oC, 6.15^oC, 23.61%, 5.18^oC, 4.61^oC, 3.66^oC and 0.46^oC, respectively. A minimal difference between the maximum and minimum values of the desired measured parameters was observed for the two systems during the duration of the VCRC due to the 100 L hot water drawn off, at the inlet of the compressor, outlet of the compressor, inlet of the condenser and the outlet of the condenser. Also, much higher measurements were recorded for the split type with respect to the corresponding parameters previously highlighted for the integrated type. Moreover, increase in volume of hot water withdrawn led to a corresponding increase in time of operation of the respective heating cycles.

The results produced under the 150 L hot water drawn off, indicated that the difference between the maximum and minimum values of the desired nine parameters in the split type ASHP water heater were 0.09 kW, 20 minutes, 8.16^oC, 8.36^oC, 32.85%, 11.39^oC, 10.80^oC, 9.80^oC and 4.20^oC, respectively. The difference between the maximum and minimum values of the described critical parameters investigated in the integrated type ASHP water heater under the 150 L hot water drawn off heating cycles operated under the same starting time were 0.05 kW, 45 minutes, 7.50°C, 8.07°C, 37.90%, 5.07°C, 7.32°C, 7,2 °C and 4.04°C, respectively.

Table 9.4 shows a comprehensive summary of the average values of each parameter during the three scenarios of hot water withdrawal from both types of ASHP water heaters.

Table 9.4: Winter comparisons of the average values of key parameters

ASHP system	Vd L	P kW	Time mins	Tcw °C	Ta °C	RH %	Tcmi °C	Tcmo °C	Tcni °C	Tcno °C
Split	50.0	1.14	68.33	15.31	15.61	67.98	19.64	66.35	69.15	39.79
Integrated	50.0	0.91	110.00	15.40	15.69	67.95	7.66	49.49	43.25	42.42
Split	100.0	1.21	86.66	14.79	14.99	71.60	19.12	66.36	69.92	37.35
Integrated	100.0	0.86	156.66	14.90	15.24	70.04	6.354	47.00	40.20	39.40
Split	150.0	1.23	100.00	18.50	19.50	57.55	24.08	70.65	74.12	36.57
Integrated	150.0	0.83	168.33	18.89	19.28	59.73	7.56	47.75	40.97	37.69

Vd , Volume of water drawn off; P, Electrical power consumed; Tcw , In-line cold water temperature; Ta , Ambient temperature; RH , Relative humidity; Tcmi , Temperature of the refrigerant at the inlet of the compressor; Tcmo , Temperature of the refrigerant at the outlet of the compressor; Tcni , Temperature of the refrigerant at the inlet of the condenser; Tcno, Temperature of the refrigerant at the outlet of the condenser

It can be observed from Table 9.4 that the average power consumption of the integrated type system (0.87 kW) was lower relative to that of the split type system (1.19 kW) throughout the heating cycles with a difference of 0.33 kW. The average time difference of the heating cycle was 180 minutes, although the integrated system was operated for a longer period. The average temperature of the refrigerant at the inlet and outlet of the compressor in the split and integrated type ASHP water heaters were (20.95 and 67.78°C) and (7.19 and 48.08°C), respectively. The average difference in the temperature of the refrigerant at the compressor in both systems was 5.94°C, but it was higher in the split type (46.83°C) than in the integrated type (40.89°C). The average temperature of the refrigerant at the inlet and outlet of the condenser in the split and integrated type ASHP water heaters were (71.06 and 39.83°C) and (41.47 and 37.90°C), respectively. The average difference in the temperature of the refrigerant at the inlet and outlet of the condenser in the two types of ASHP water heaters was 27.66°C; however, it was much higher in the split type (31.23°C) in contrast to the integrated type (3.57°C). It could be observed from Table 9.4 that the average power consumed decreased as the volume of hot water drawn off increased from 50 L, to 100L and to 150 L for both systems. It was determined that the averages in the ambient temperature (16.73 and 16.70 °C), relative humidity (65.91 and 65.71%) and in-line cold water temperatures (16.39 and 16.20°C) were almost equal with regards to the split and integrated type ASHP water heaters for the entire VCRC scenarios.

9.3.3 Summer comparison of energies and COP of the both systems Table 9.5 shows the average thermal energy gained, the average electrical energy consumed and the COP achieved during the entire 50, 100 and 150 L hot water withdrawal from the two types of ASHP water heaters.

Table 9.5: Summer comparisons of the two systems based on energy and COP

ASHP system	Drawn off L	Power kW	Electrical energy kWh	Thermal energy kWh	COP
Split	50.0	1.1667	0.8067	2.3200	2.8767
Integrated	50.0	0.8533	0.9500	2.3200	2.4367
Split	100.0	1.2600	1.3600	4.1167	3.0133
Integrated	100.0	0.8667	1.5670	4.1167	2.6500
Split	150.0	1.2833	1.7467	5.5767	3.1733
Integrated	150.0	0.8433	1.9543	5.5767	2.8400

Clearly, it can be depicted from Table 9.6 that for a specific volume of hot water drawn off, the corresponding electrical energy consumed by the split type ASHP water heater was lower as opposed to that of the integrated type. This could be affirmed by the longer time taken during the heating cycle which occurred in the integrated type ASHP water heater. The average electrical energy consumed by the split and integrated type ASHP water heaters at 50 L hot water withdrawal was 0.807 and 0.950 kWh, while the average time taken was 41.67 and 73.33 minutes, respectively. The average electrical energy consumed during the 100 L hot water withdrawal and average duration spent for heating cycles was 1.360 kWh and 65 minutes for the split type system and 1.570 kWh and 108 minutes for the integrated type system. In the 150 L hot water drawn off scenario, the average electrical energy consumed and time taken for the heating cycles was 1.747 kWh and 82 minutes for the split type system and 1.954 kWh and 140 minutes for the integrated type system. The average COP of the split type ASHP water heater in the entire heating cycles was 2.95 and that of the integrated type system was 2.62. The COP of the two systems under the different scenarios was above 2 on average and increased with a corresponding increase in the volume of hot water drawn off, which is in agreement with the studies reported in literature (Levins, 1982; Bodzin, 1997).

9.3.4 Winter comparisons of energies and COP of both systems Table 9.6 shows the average thermal energy gained, the average electrical energy consumed and the COP during the entire 50, 100 and 150 L hot water drawn off from the two types of ASHP water heaters.

Table 9.6: Winter comparisons of the two systems based on energy and COP

ASHP system	Drawn off L	Power kW	Electrical energy kWh	Thermal energy kWh	COP
Split	50.0	1.1407	1.1564	2.6541	2.499
Integrated	50.0	0.9128	1.5635	2.6540	2.093
Split	100.0	1.2151	1.5994	4.9141	2.923
Integrated	100.0	0.8673	2.1612	4.9141	2.294
Split	150.0	1.2314	1.9091	6.0196	3.155
Integrated	150.0	0.8370	2.2798	6.0196	2.403

Apparently, from the data displayed on Table 9.6, it is shown that for a specific volume of hot water withdrawal, the corresponding electrical energy consumed by the split type ASHP water heater was lower than that of the integrated type. This could be ascertained by the longer time taken during the heating cycles experienced by the integrated type ASHP water heater. The average electrical energy consumed by the split and integrated type ASHP water heaters at 50 L hot water withdrawal were 1.156 and 1.563 kWh, while the average duration was 55.49 and 104.20 minutes, respectively. The average electrical energy consumed during the 100 L hot water withdrawal and average duration of the heating cycles were 1.599 kWh and 76.72 minutes for the split type system and 2.161 kWh and 144.07 minutes for the integrated type system. In the 150 L hot water drawn off scenario, the average electrical energy consumed and time for the heating cycles were 1.909 kWh and 91.63 minutes for the split type system and 2.279 kWh and 151.93 minutes for the integrated type system. The average COP of the split type ASHP water heater in the entire heating cycles was 2.86 relative to 2.26 for the integrated type ASHP water heater.

9.3.5 Development of the mathematical models of the system's COP for summer

The dataset of the predictors and COP for each of the systems were used to develop and build the multiple linear regression models which established the correlation between the inputs and the output parameters during the summer heating cycles. The derived multiple linear regression equation used is shown in Equation 9.6. Table 9.7 shows the mathematical model forcing and scaling values for the split type ASHP water heater. From the model equation of the split type ASHP water heater, it was revealed that the difference in the temperature of the refrigerant at the inlet and outlet of the condenser (ΔT_{cn}) contributed significantly to the COP. It could also be predicted that increase in ΔT_{cn} resulted in a corresponding increase in the COP. Also, both increase in ambient temperature and relative humidity can resulted in a corresponding decrease in COP provided other parameters were kept constant.

Table 9.7: Summer scaling and forcing constants for the split type system

Predictors	Symbols	Scaling notations	Scaling Values	Output
Forcing constant		β_0	7.1280	
Ambient temperature	Ta	β_1	-0.0890	
Relative humidity	RH	β_2	-0.0140	COP
Difference in compressor temperature	ΔT_{cm}	β_3	-0.0620	
Difference in condenser temperature	ΔT_{cn}	β_4	0.0470	

The modelled and calculated COP of the split type ASHP water heater had a strong determination coefficient of 0.945 and showed a perfect fit. Figure 9.1 shows the sample dataset of the calculated COP and the modelled COP curve fit for 27 observations that involved all the three scenarios of hot water drawn off.

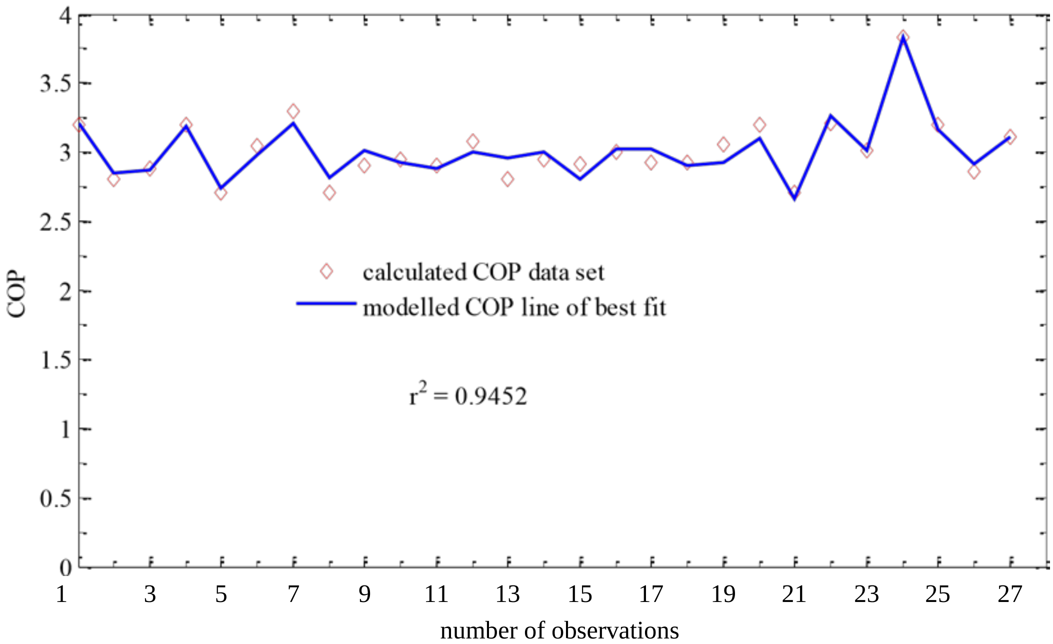


Figure 9.1: Summer calculated COP dataset and modelled COP curve of the split type

Also, Table 9.8 shows the mathematical model forcing and scaling values for the integrated type ASHP water heater. The model equation of the integrated type ASHP water heater equally emphasised that the difference in the temperatures of the refrigerant at the inlet and outlet of the condenser (ΔT_{cn}) contributed significantly to the COP. It could also be predicted that increase in ambient temperature (T_a) resulted in a corresponding increase in the COP.

But, an increase in relative humidity leads to a decrease in the COP.

Table 9.8: Summer forcing and scaling values for the integrated type system

Predictors	Symbols	Scaling notations	Scaling Values	Output
Forcing constant		$\square_0$	8.0990	
Ambient temperature	Ta	$\square_1$	0.0060	
Relative humidity	RH	$\square_2$	-0.0080	COP
Difference in compressor temperature	$\square T_{cm}$	$\square_3$	-0.1230	
Difference in condenser temperature	$\square T_{cn}$	$\square_4$	-0.0260	

The modelled COP and calculated COP of the integrated type ASHP water heater had a strong determination coefficient of 0.925 and exhibited a good fit. Figure 9.2 shows the sample dataset of the calculated COP and the modelled COP curve fit for 27 observations that involved all the three heating scenarios (50, 100 and 150 L hot water withdrawal)

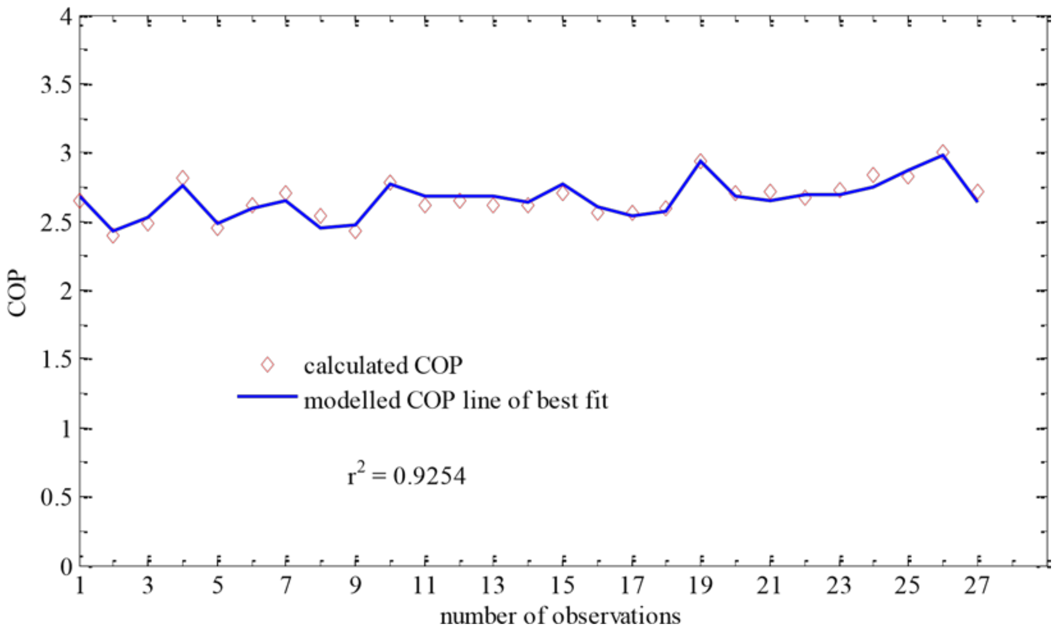


Figure 9.2: Summer calculated COP and modelled COP curve of the integrated type

9.3.6 Development of the mathematical models of the system's COP for winter

The dataset of the predictors and COP for each of the systems were used to develop and build a multiple linear regression models which established the correlation between the inputs and the output parameters for the winter season. The derived multiple linear regression models used is shown in Equation 9.6. Table 9.9 shows the mathematical model forcing and scaling values for the split type ASHP water heater. The model equation of the split type ASHP water heater revealed that the difference in the refrigerant temperatures at the inlet and outlet of the condenser (ΔT_{cn}) offered a great contribution to the COP. It could also be predicted that increase in ΔT_{cn} resulted in a corresponding increase in the COP. The modelled and calculated COP of the split type ASHP water heater had a strong determination coefficient of 0.935 and demonstrated a strong agreement from a visual representation.

Table 9.9: Winter forcing and scaling constants for the split type system

Predictors	Symbols	Scaling notations	Scaling Values	Output
Forcing constant		β_0	14.581	
Ambient temperature	T_a	β_1	-0.1295	COP
Relative humidity	RH	β_2	-0.0003	
Difference in compressor temperature	ΔT_{cm}	β_3	-0.2920	
Difference in condenser temperature	ΔT_{cn}	β_4	0.1187	

It could be noted without loss of generality, that since all the scaling constants were negative except that since the change in refrigerant temperature at the inlet and outlet of the condenser, any increase in those specific predictors is associated with a corresponding decrease in the COP for the split type system. Also an increase in the change in the refrigerant temperature between the inlet and outlet of the condenser is associated with an increase in the COP. In addition, Table 9.10 presents the forcing and scaling values for the mathematical model developed for the integrated type ASHP water heater. The modelled equation of the integrated type ASHP water heater equally laid credence to the significant contribution impacted by the difference in the temperature of the refrigerant at the inlet and outlet of the condenser (ΔT_{cn}) to the COP. It could also be predicted that increase in ΔT_{cn} resulted in a corresponding increase in the COP. The modelled and calculated COP of the integrated type ASHP water heater had a very good determination coefficient of 0.912 and demonstrated a strong agreement with negligible outliers.

Table 9.10: Winter forcing and scaling constants for the integrated type system

Predictors	Symbols	Scaling notations	Scaling Values	Output
Forcing constant		β_0	8.9377	
Ambient temperature	Ta	β_1	0.0046	COP
Relative humidity	RH	β_2	0.0011	
Difference in compressor temperature	ΔT_{cm}	β_3	-0.1700	
Difference in condenser temperature	ΔT_{cn}	β_4	0.0392	

Also, an increase in ambient temperature can result in a corresponding increase in COP as well as increase in relative humidity can also give rise to an increased in the COP provided other parameters were kept constant. Again, an increase in the changed in refrigerant temperature at the inlet and outlet of the condenser will lead to an increase in the COP.

9.3.7 Testing of the modelled and calculated COP of the systems by ANOVA using summer data

The dataset of over 27 averages of calculated COP of the split and integrated type ASHP water heaters that spanned the entire heating cycle scenarios was used in the one-way analysis of variance (ANOVA) test to determine any significant difference in the group COP. Furthermore, the one-way ANOVA test employed the regression analysis methods and the null hypothesis test that treated all group means to be equal. The critical parameter that determined the possibility of a significant difference among group means is known as the pvalue (Hogg and Ledolter, 1987). Clearly, a very small p-value (0.01, 0.05 etc.), indicated a significant difference among the group means. The group means had no significant difference if the p-value was close to 1. Figure 9.3 shows the ANOVA plots of the groups of calculated and modelled COP means of the split and integrated types ASHP water heaters.

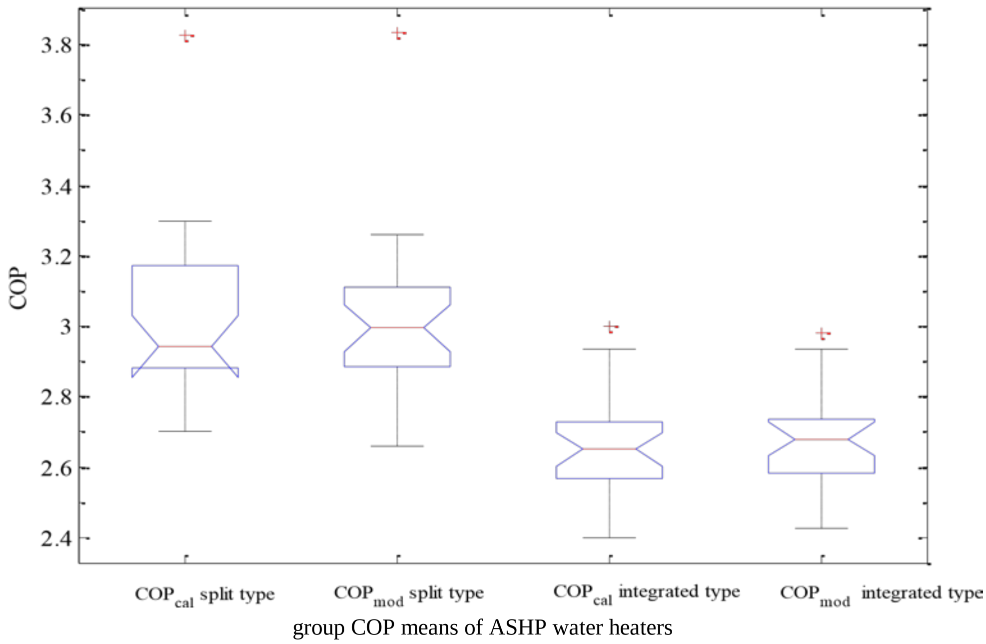
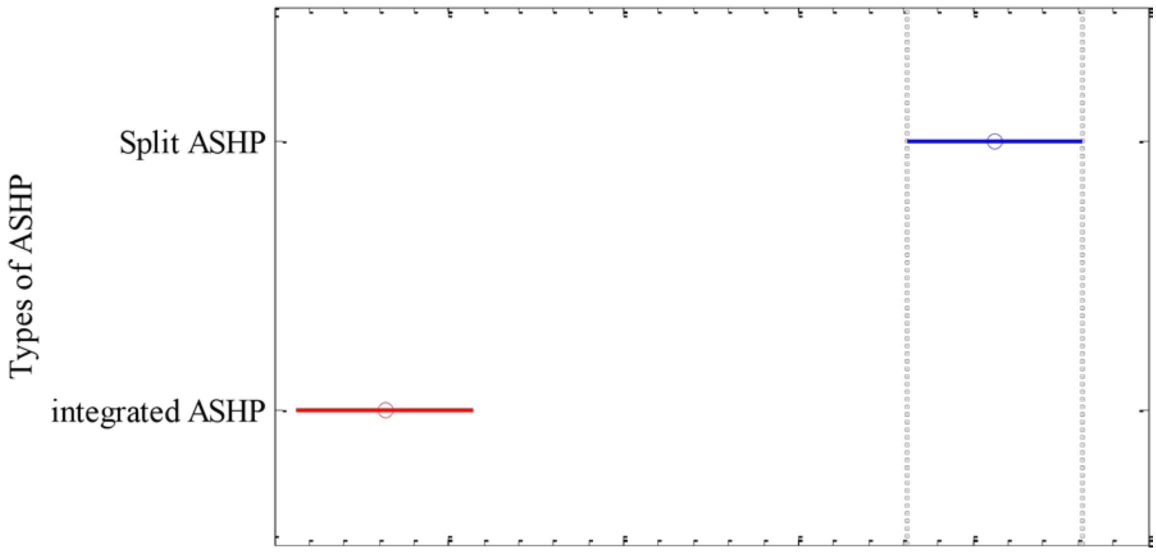


Figure 9.3: Summer ANOVA plots for the calculated and modelled group COP

From the Figure 9.3, it can be interpreted that there was no significant difference among the calculated and modelled group COP means of the split type ASHP water heater as the p-value was 0.998 and the dataset was normally distributed. It could also be illustrated that there was no significant difference between the group COP means of the modelled and calculated COP for the integrated type ASHP water heater as the p-value was 0.996. The p-value of the modelled COP means of the split and integrated type ASHP water heaters was $6x10^{-9}$. Hence, there was a significant difference between the two group COP means.

9.3.8 Testing of the mean significant difference of the COP of both systems using summer data

Following the result obtained from the one-way ANOVA plots of the COP, a multiple comparison procedure algorithm was used to further test for a significant difference in the modelled COP means for the two systems under all the scenarios. A simulation plot of the multiple comparisons between the modelled COP means of the split and the integrated type ASHP water heaters is as shown in Figure 9.4. The horizontal lines show the range of the group COP means of the two systems while the marked circle on the line indicated the mean COP. Furthermore, it should be noted that if the lines overlapped, there exists no significant difference. The modelled group COP means of the split type ASHP water heater (blue line plot) and that of modelled group COP means of the integrated type ASHP water heater (red line plot) is as shown in Figure 9.4. Figure 9.4 shows there was a significant difference as they did not overlap. The mean difference in the group COP of the two systems was 0.349. The difference in the true average modelled group COP means, and at the 95% confidence level of the modelled group COP means of the split type system was 0.249. The difference in the true average modelled group COP means, and at the 95% confidence level of the modelled group COP means for the integrated type ASHP system was 0.449. Hence, there is no value of 0, between this interval [0.249 and 0.449]; therefore, there was a significant difference in the modelled group COP means of both systems.



2.6 2.7 2.8 2.9 3 3.1

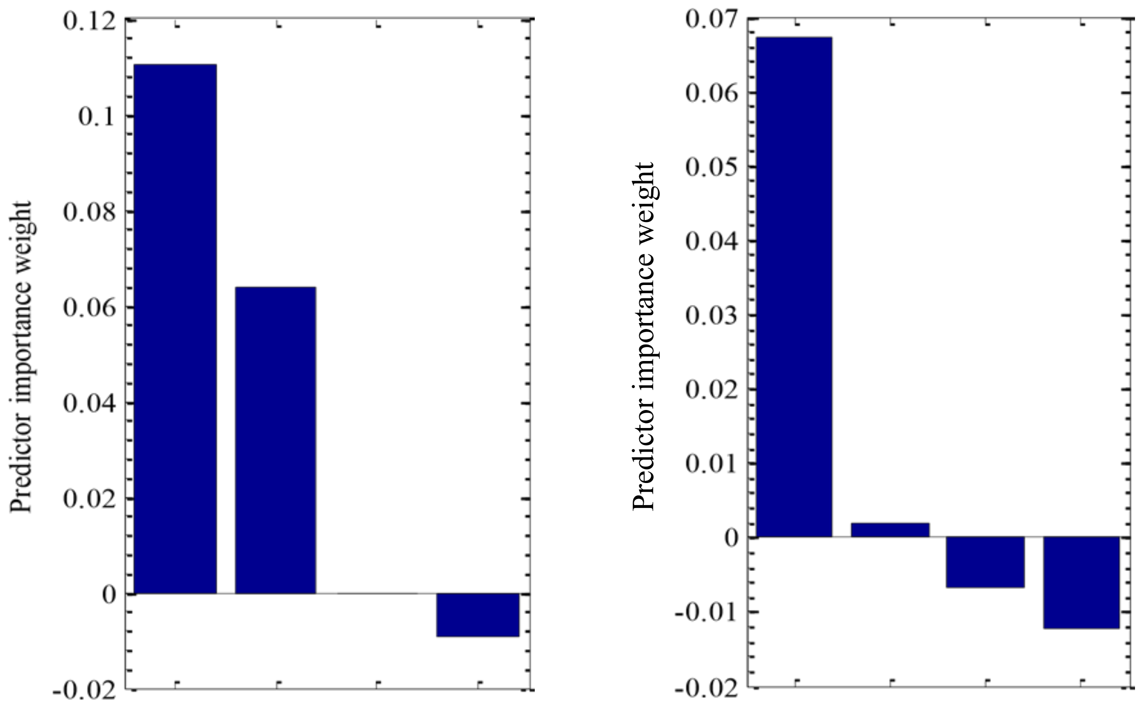
There was significantly different in the group mean COPs between the two types ASHP

Modelled COP

Figure 9.4: Summer, multiple comparison plots, to test group COP significant difference

9.3.9 Ranking of predictors by ReliefF test using the summer data of both systems

The four predictors (T_a, RH, $\square T_{cm}$, $\square T_{cn}$) and the output (COP) from the processed data of the split and integrated type ASHP water heaters were used in the ReliefF algorithm to rank predictors according to their importance of weight contribution. Figure 9.5 shows the reliefF bar plots for the predictors and the importance of weight contributions to the COP for both the split and integrated type ASHP water heaters. The weight ranking showed that for both types of ASHP water heaters, the difference in the temperature of the refrigerant at the outlet and inlet of the compressor ($\square T_{cm}$) and the difference in the temperature of the refrigerant at the inlet and outlet of the condenser ($\square T_{cn}$) were primary factors. It could also be determined from the analysis that both primary predictors weight contributions to the COP of the split type system ($\square T_{cm}$ = 0.111 and $\square T_{cn}$ = 0.064) were higher than their contributions to the COP of the integrated type system ($\square T_{cm}$ = 0.067 and $\square T_{cn}$ = 0.002). The ambient temperature (T_a) and relative humidity (RH) were categorised as secondary factors with regards to both systems. The impact of the ambient temperature contribution due to the weight of importance was almost negligible for the split type ASHP water heater (T_a = -0.004) but was 1.75 times higher in comparison to the integrated system (T_a = -0.007). Both T_a and RH were secondary factors, but changes in either or both could affect the COP.



□T_{cm} □T_{cn} T_a R_h

Predictor rank for split type

□T_{cm} □T_{cn} T_a R_h

Predictor rank for integrated type

Figure 9.5: Summer reliefF bar plots for the predictors and contributions of both systems

9.3.10 2D multi contour plots simulation of the ASHP systems using summer data

The 2D multi contour plots simulation is a multiple two-dimensional plot used to model the variation of a specific predictor with the output in any given multiple linear regression model while the other predictors are held constant. The twodimensional multi contour plots simulation can be employed for up to thirteen predictors (MathWorks, 2012; Tangwe and Simon, 2018). In this study, the 2D multi contour plots simulation was used to visualise the variation of the ambient temperature (T_a) with the predicted COP at a constant RH , □T_{cm} and □T_{cn} for both the split and integrated type ASHP water heaters. Similarly, each of the other predictors was varied, and the change in the modelled COP was determined using the simulation model plots while the rest predictors were kept constant. Figure 9.6 shows the two-dimensional multi contour plots simulation of the split type ASHP water heater. The positive slope of □T_{cn} in the split type system suggested that increase in predictor led to an increase in the COP. The green lines on these plots represent the linear relationship between the predictors and the COP and both red broken curves defined the 95% confidence bound. The slopes of the modelled COP and the T_a , RH ,

δT_{cm} and δT_{cn} were -0.089 /°C, -0.014 /%, -0.062 /°C and 0.047 /°C, respectively as determined from the derived mathematical model.

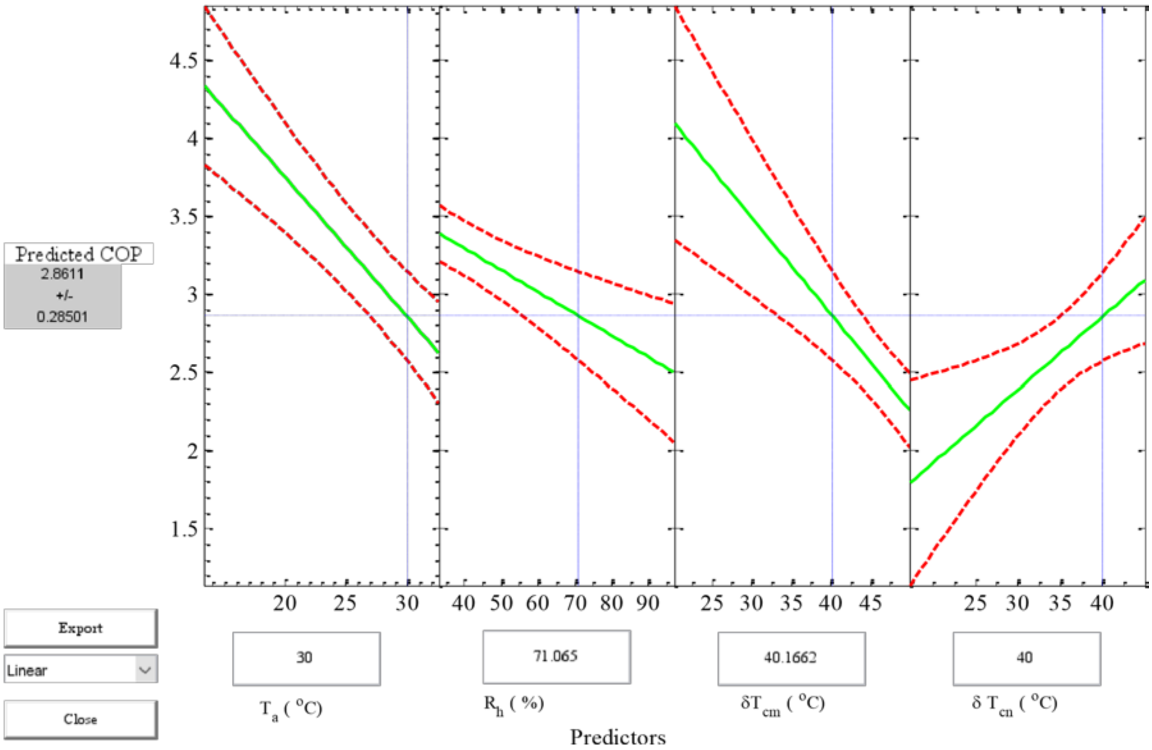


Figure 9.6: Summer 2D simulation plot of predictors and COP for the split type

Figure 9.7 demonstrates that under these drawn off scenarios, the predictor (T_a) increased with the modelled COP of the integrated type ASHP water heater provided the others were kept constant. This is in agreement with the scaling coefficient obtained from the derived mathematical model. The determined slopes for the modelled COP means of the integrated system with respect to T_a, RH, δT_{cm} and δT_{cn} were 0.006 /°C, -0.008 /%, -0.123 /°C and -0.026 /°C.

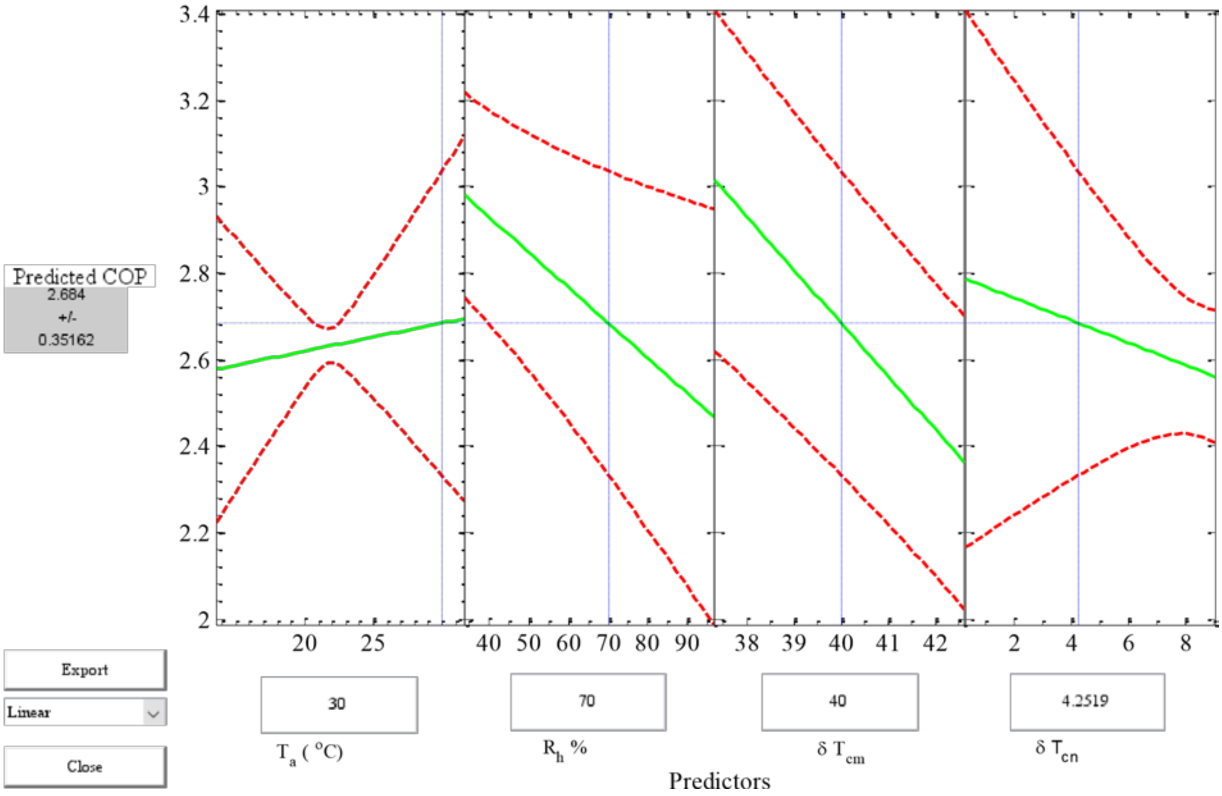


Figure 9.7: Summer 2D simulation plot of predictors and COP for the integrated type

9.3.11 Validation of the developed mathematical models of both systems using summer data

The exclusive dataset of the predictors and the response from the two types of ASHP water heaters obtained under the same controlled volume of hot water drawn off scenarios were employed to test the validity of the developed models. The determination coefficient and the p-value of the predicted COP and the calculated COP of the test dataset were determined. The determination coefficient and the p-value of the COP of the split type ASHP water heater was 0.915 and 0.967, respectively. Table 9.11 shows the sample of test datasets critical measured parameters of the split type ASHP water heater, the calculated COP (COP_{cal}) and the predicted COPs from the derived mathematical model (COPmod). The predicted COP (COPmod) and the calculated COP (COPcal) from the test dataset showed a strong correlation and therefore, justified the used of the derived multiple linear regression model for the COP prediction. The root mean square bias error of the calculated and modelled COP was 0.0024 and was much smaller than the minimum calculated COP (2.7) obtained from the test dataset. Hence, the very small root mean square bias error further confirmed the accuracy of the derived mathematical model.

Table 9.11: Test dataset of key parameters, and COPs of the split system

Time mins	P kW	Ta ºC	RH %	Tcmi ºC	Tcmo ºC	Tcni ºC	Tcno ºC	Q kWh	COP_{cal}	COP_{mod}
85	1.26	21.37	59.55	23.00	71.15	72.76	36.57	5.72	3.20	3.11
70	1.26	18.3	85.18	22.69	70.58	72.99	38.09	4.44	3.01	2.98
65	1.28	34.92	45.80	39.38	84.28	85.76	40.05	3.83	2.75	2.75
80	1.30	19.78	76.12	22.719	71.00	73.50	35.81	5.28	3.05	3.08
60	1.34	23.35	55.79	28.61	76.73	79.07	40.30	4.17	3.11	3.11
65	1.30	17.27	91.38	22.46	70.92	74.23	38.70	4.21	2.98	2.98
65	1.27	23.48	77.86	26.98	74.42	77.60	39.94	3.92	2.84	2.78
60	1.32	23.52	77.58	28.22	76.12	79.23	40.52	3.69	2.80	2.80
70	1.26	19.27	70.44	24.85	71.87	73.77	38.60	4.56	3.10	3.16
40	1.14	29.47	36.47	38.40	81.45	80.91	45.87	2.32	3.04	2.97
35	1.31	19.27	73.72	25.19	73.02	73.74	42.94	2.19	2.87	2.86
40	1.37	20.35	85.61	26.59	75.75	78.27	43.00	2.47	2.70	2.73
40	1.14	29.47	36.47	38.40	81.45	80.91	45.87	2.32	3.04	2.97
70	1.25	13.95	86.13	17.71	65.70	69.32	37.36	4.70	3.21	3.21
60	1.34	23.09	75.05	27.87	76.31	78.12	40.17	3.89	2.89	2.80

P, Average power consumed; Ta, Average ambient temperature; RH,Average relative humidity; Tcmi, Average temperature of the refrigerant at the inlet of the compressor; Tcmo, Average temperature of the refrigerant at the outlet of the compressor; Tcni, Average temperature of the refrigerant at the inlet of the condenser; Tcno, Average temperature of the refrigerant at the outlet of the condenser; Q, Useful thermal energy gained; COPcal, Calculated COP of the system; COPmod, Modelled COP of the system

Also, Table 9.12 shows a sample of test dataset critical measured parameters of the integrated type ASHP water heater, the calculated COP (COPcal) and the predicted COP (COPmod). The determination coefficient and the p-value of the COP of the integrated type ASHP water heater was 0.925 and 0.970, respectively. The predicted COP (COPmod) and the determined COP (COPcal) from the test dataset produced by the integrated type ASHP water heater demonstrated a very good correlation and therefore provided adequate reasons to use the model for prediction of COP. The root mean square bias error of the calculated and modelled COP was 0.0047 and was significantly negligible than the least calculated COP (2.45) obtained from the test dataset. In this regard, the accuracy of the derived mathematical model was considered to be very good and could be seconded by the very small root mean square bias error and minimal deviation between the calculated and modelled COP.

Table 9.12: Test dataset of key parameters, and COPs of integrated system

Time mins	P kW	Ta ºC	Rh %	Tcmi ºC	Tcmo ºC	Tcni ºC	Tcno ºC	Q kWh	COP_{cal}	COP_{mod}
135	0.86	19.36	78.72	11.33	49.53	43.15	38.34	5.44	2.80	2.75
110	0.88	19.27	81.19	10.44	50.32	44.05	41.22	4.27	2.64	2.57
90	0.88	35.05	45.47	25.08	62.02	55.79	43.98	4.05	3.04	3.09
135	0.86	19.36	78.72	11.33	49.53	43.15	38.34	5.46	2.81	2.75
95	0.88	23.07	56.70	13.25	54.01	46.26	41.40	3.80	2.71	2.63
125	0.86	18.46	87.84	10.67	49.73	43.73	40.24	4.61	2.57	2.60
100	0.88	23.87	77.01	15.32	54.67	48.41	42.06	3.93	2.67	2.61
95	0.87	22.99	78.26	15.86	55.90	48.48	41.92	3.72	2.67	2.50
125	0.84	19.08	71.40	10.48	49.90	42.80	38.78	4.72	2.67	2.67
60	0.85	29.56	36.83	15.50	57.99	51.75	45.79	2.32	2.70	2.60
70	0.85	19.88	71.46	10.69	51.66	45.13	42.28	2.50	2.50	2.52
65	0.85	20.36	85.62	13.68	53.81	47.28	43.33	2.27	2.44	2.48
60	0.86	29.56	36.83	15.50	57.99	51.75	45.79	2.23	2.60	2.60
135	0.84	13.93	86.60	7.39	46.40	39.67	38.76	5.05	2.65	2.65
95	0.86	22.95	75.52	15.12	54.75	46.50	40.99	3.66	2.67	2.60

P, Average power consumed; Ta, Average ambient temperature; RH,Average relative humidity; Tcmi, Average temperature of the refrigerant at the inlet of the compressor; Tcmo, Average temperature of the refrigerant at the outlet of the compressor; Tcni, Average temperature of the refrigerant at the inlet of the condenser; Tcno, Average temperature of the refrigerant at the outlet of the condenser; Q, Useful thermal energy gained; COPcal, Calculated COP of the system; COPmod, Modelled COP of the system

9.3.12 Simulation application developed to compare COP of the two systems

The COP of the split and integrated type ASHP water heaters was simulated in the Simulink environment using the developed and built mathematical models. Figure 9.8 shows the schematic architectural algorithm of the design simulation application. The simulation application also aided in the automated calculation and visualisation of the COP of both ASHP water heaters under simultaneous heating cycles.

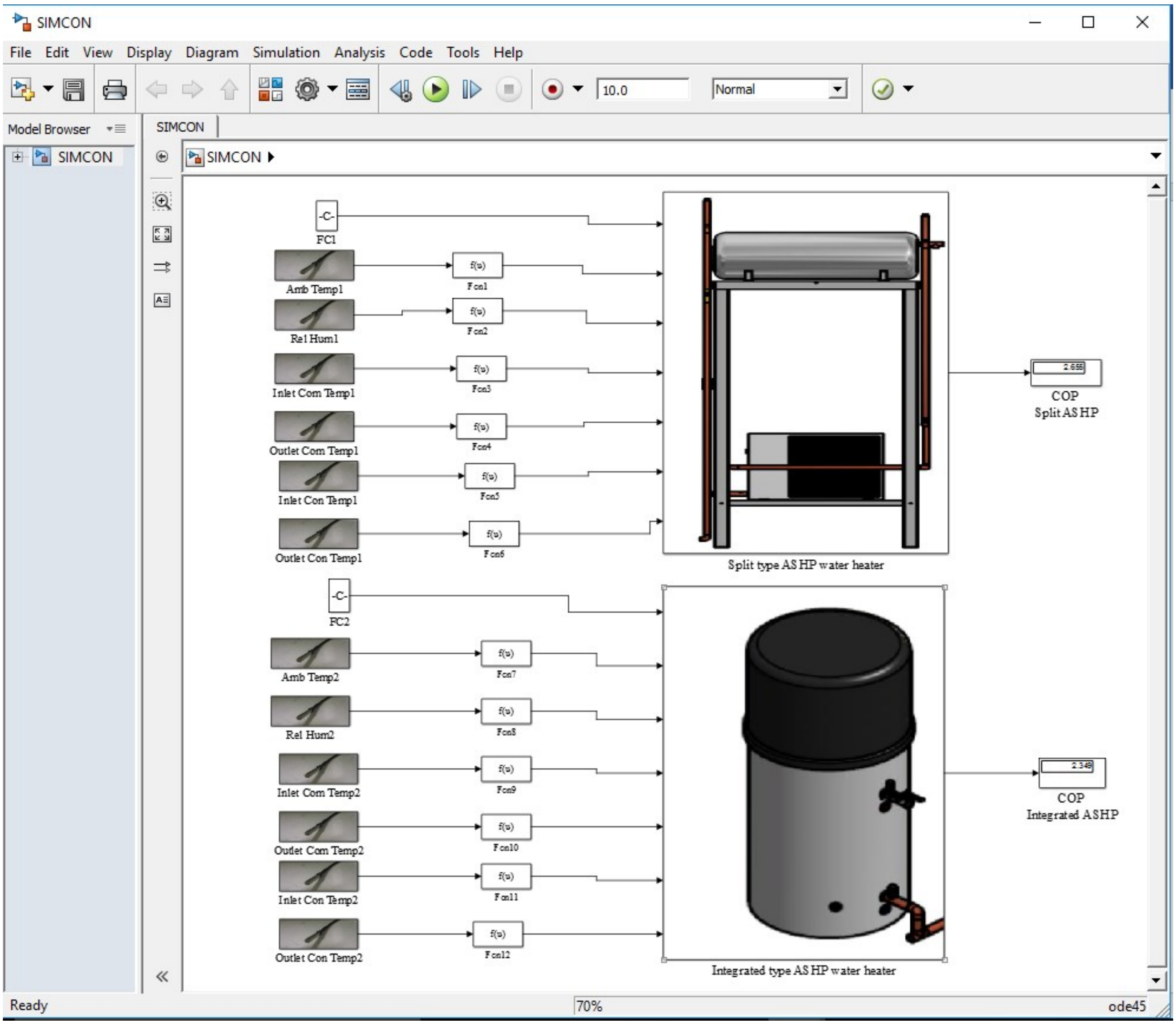


Figure 9.8: Schematic of the simulation application uses to compare COP

The simulation application used the constant block in the source library for the input of predictors dataset, and the user defines function (fn) for the determination of the average value of the predictors during the VCRC. The two subsystems were block masked with the image of the split and integrated type ASHP water heaters embedded with the derived mathematical models of both systems. The respective subsystem block consisted of a summation and gained blocks which adequately accommodated all the required input parameters. It was noted that the predictors (difference in the temperature of the refrigerant at both inlet and outlet of the compressor or condenser) were each handled by a summation block with both plus and minus sign to cater for the difference. The calculated COP of the two type ASHP water heaters was shown on the display blocks that was obtained from the Simulink sink library. Furthermore, the simulation application was set to run by clicking on the start button on the Simulink environment. The dataset of all the crucial inputs (T_a, RH, T_{cmi}, T_{cmo}, T_{cni}, T_{cno}) obtained at the specific time interval during the VCRC operated by the split and integrated type ASHP water heaters were loaded into the respective source blocks. The function block (fn) was also adjusted to compute the average of each of the input parameters by using the notation ((u(1)+…u(n))/n), whereby u represented the dataset and n was the number of data values. It was observed that for any particular scenario of hot water withdrawal, using the same logging interval for both the split and integrated type ASHP water heaters, the time taken to complete that VCRC was much higher for the integrated type ASHP water heater than the split type ASHP water heater. Hence, the number of data values in the case of the integrated system during the VCRC was more than that of the split type ASHP water heater. After input dataset was loaded and the user define functions (fn) adjusted, the start button was then clicked to run the application, and the results of the calculated COPs were shown on the display blocks. The calculated COPs shown on the display blocks could be used from a real-time perspective to compare the performance of both types of ASHP water heaters.

9.4 Summary

It can be concluded that mathematical modelling and simulation provided a rapid and in-depth approach for the evaluation and comparison of the performance of both split and integrated type ASHP water heaters. The interpretation of the results confirmed to a great extent that the difference in the temperatures of the refrigerant between the inlet and outlet of the condenser was the strongest predictor and a primary factor which influenced the COP of both types of ASHP water heaters. The COP of the split type ASHP water heater without an electric backup performed better to that of the integrated type ASHP water heater with an electric backup element. In addition, the average COP of both ASHP water heaters during the VCRC, irrespective of the volume of hot water drawn off was above two, but there exists a mean significant difference in the group COP of the two systems based on a multiple comparison procedure test. The COP of either or both systems could be predicted by the design simulation application employing the derived mathematical models. It was equally confirmed that the thermodynamic predictors (difference in the temperature of the refrigerant at the inlet and outlet of the compressor or condenser) contributed more to the COP of the both types of ASHP water heaters as opposed to ambient condition predictors (ambient temperature and relative humidity). Finally, the simulation application can further be used to compare the COP of both split and integrated type ASHP water heaters with high reliability and accuracy.

Chapter Ten

10.1 General discussion

Residential ASHP water heater is a renewable and energy-efficient device utilised for sanitary hot water production. The ability of the system to substantially explore the indirect solar energy in the form of aero-thermal energy during vapour compression refrigeration cycles necessitated its classification as a renewable energy device (Marrison *et al*., 2004). Although, the COP of ASHP water heater can range between 2 and 4 (Levins, 1982; Bodzin, 1997), there often exists a significance difference in the COP of the system during the standby losses heating cycles due to the first hour heating rating and other distinctive volumes of hot water drawn off scenarios (Tangwe *et al*., 2014). It is of absolute importance to note that the COP of the residential ASHP water heater is higher when operated under all possible hot water drawn off scenarios in the summer than in the winter periods. This can be attributed to the favourable ambient temperature and relative humidity under which the ASHP water heater would be operating during the summer period. The energy saving potential of the residential ASHP water heater was strongly governed by the capacity of hot water drawn off from the tank and the COP of the system.

It should be greatly emphasised that an efficient installation of the ASHP water heater could guarantee an excellent performance of the system (Douglas, 2008). Comprehensively and without any loss of generality, it is of huge benefits both on the demand consumption and energy conservation measures to encourage the use of residential ASHP water heaters as an efficient technology for sanitary hot water heating (Tangwe *et al.,* 2015).

The COP of the system could be accurately predicted by mathematical modelling provided the ASHP unit, the tank volume and number of occupants in the building were correctly sized. Mathematical modelling of the COP of the residential ASHP water heaters using different multiple linear regression methods depicted that ambient temperature and relative humidity were secondary predictor drivers affecting the system's COP. It could be affirmed that the prediction accuracy of the mathematical model increased as the number of predictors increased and provided they were actively influencing the desired response.

Overall, the performance of the residential ASHP water heater could effectively be improved by ensuring that the connected pipes between the ASHP unit (split type) and the tank were thoroughly insulated. More so, periodic cleaning of the evaporator, the fan and the duct space of the ASHP should be performed. Despite the robustness of the designed and built DAS, there were also high levels of confidence in the various parameters measured by the precision accuracy of the sensors and the data logger that were used (Tangwe *et al*., 2016a). The DAS also consumed very minimal electricity as it was powered by a 4.5 V DC battery which once fully charged, the data logger was capable of storing data for over six months in a minute logging interval without recharging the battery.

Furthermore, from the chapters containing results and discussion, the following potential findings are hereby outlined to showcase their implications in the field of engineering and science. The implications are geared toward engineering innovations, but also included the applied physical sciences applications, applied systems application, environmental and social science impacts.

In chapter three, the design and building of the reliable and accurate DAS was used to monitor the electrical, thermal and climatic performance of both types of residential ASHP water heaters and was the first of its kind to be deployed in South Africa (Tangwe *et al.*, 2016). The DAS was enclosed with a waterproof enclosure and was designed using smart sensors and data loggers, which, were all compatible (Hobo cooperation, 2013). The DAS was portable and capable of withstanding extreme outdoor conditions. It was powered by a 4.5 V DC battery and once fully charged either by electricity from the national grid or solar panels had the potential to sustain the DAS to log for over 6 months in 1minute logging interval. The DAS is easy to configure and does not require any high level of expertise to install. The stored data in the DAS can also be analysed both from a statistical and graphical plots with the aid of the hoboware pro software.

In addition, engineers, heat hump manufacturers and Energy Service Company are able to evaluate the performance of an ASHP by installing the DAS and performing the analysis from the hoboware pro without exporting the data to other data analysis software package (Excel, MATLAB, etc). The data downloaded from the DAS with the hoboware pro software can easily be exported to excel and hence MATLAB for further analysis. As a final point, the installation of the DAS on the ASHP water heaters provided sound and scientific assurance to justify the COP of the systems and a potential payback period as well as cause reduction in environmental pollutions (Tangwe *et al.*, 2014; Bryson, 2011; Van Eeden *et al*., 2016).

Chapter four demonstrated that by retrofitting or replacing a geyser with either a split or an integrated type ASHP water heater resulted in potential permanent demand and energy reduction between 50 and 70% per annum (Morrison *et al*., 2004, Tangwe *et al*., 2017). Both, the split type and integrated type ASHP water heater exhibited a favourable simple payback period of 3.9 and 5.2 years, respectively (Tangwe *et al*., 2017). Tangwe and co-authors (2017) also confirmed that the payback period could be further reduced by taking into account the net return on investment as well as the annual Eskom projected tariff rate hikes. In addition, chapter four also justify the multi-purpose benefits of using residential ASHP water heaters for sanitary hot water heating over the geyser in accordance with the load factors, avoidance carbon dioxide emission and potential water saved in the generation of electricity at the thermal coal power plant. Also, heat pump manufacturers and engineers can determine quantitatively, the saving and make an informed decision on whether the ASHP water heater is performing according to its manufacturing specifications. In conclusion, chapter four also provides credible seasonal and annual data to compute the COP of the ASHP water heaters, which can be used as historical data for future research on the development of advanced ASHP water heater, with a much improved performance.

Chapter five demonstrated that the standby losses adversely impacted the COP of the ASHP water heaters. The average electrical energy consumed to compensate for the standby losses in the geyser was over twice that of either the split and integrated type ASHP water heaters. The installation of the isotherm blanket on the geyser and storage tanks of the split and integrated types ASHP water heaters was responsible for an average daily percentage of 18.5, 15.7 and 3.2%, respectively, in the reduction of the electrical energy consumption required to compensate for the standby losses (Tangwe *et al*., 2017). Both, the box plots, and multiple comparison procedure analysis were employed to show that there was a significant difference in the group electrical energy consumed in a bid to compensate for the standby losses of the geyser with and without the installation of an isotherm blanket. On the other hand, standby losses exhibited no significant difference in the group daily electrical energy to compensate for the standby losses in both scenarios wherein, the ASHP water heaters were without or with an installed isotherm blanket. Hence, it is paramount for heat pump experts, heat pump suppliers, heat pump installers and home owners of ASHP water heaters to understand that although, installation of isotherm blanket on the cylinder of the ASHP water heaters led to a reduction in the standby losses, the contribution was of no significant difference. Also, it was shown that the COP of the ASHP water heaters was lower in comparison to the system performance under the other vapour compression refrigeration cycles (e.g. First-hour heating rating and all the various controlled volumes of simulated hot water drawn off). In addition, increase in the average daily standby losses was associated with a corresponding increase in the number of heating cycles and was a function of the ambient climatic conditions and the degree of insulation of the cylinder which acted as an integral component of the hot water heating devices.

Chapter six established the diagnostic comparison of the COP of the split and the integrated type ASHP water heaters based on critical parameters such as electrical power consumption, power factor, ambient temperature and relative humidity and the temperatures of the refrigerant at the inlet and outlet of the compressor and condenser. The analysis was conducted from three distinctive volumes of hot water drawn off scenarios (50, 100 and 150 L). In all the scenarios of hot water drawn off, it was depicted that both types of ASHP water heaters had an excellent power factor of about 0.98. Although, the average power consumption during the vapour compression refrigeration cycles were lowered in the integrated type in contrast to the split type, the energy consumption has always been greater for the integrated type in comparison to the split type ASHP water heater. Besides, the temperature of the refrigerant at the suction of the compressor was higher in the split type as opposed to the integrated type ASHP water heater (Tangwe *et al*., 2016). Above all, the average of the difference in temperature of the refrigerant between the suction and discharge ends of the compressor for both types of ASHP water heaters showed no significant difference, but that of the split type was higher than that of the integrated type.

There exists a significant difference in the change between the temperature of the refrigerant at the inlet and the outlet of the condenser of both the split and integrated type ASHP water heater, but the former was much higher than the latter. The significant difference in the temperature lift at the condenser was also responsible for the better performance of the split type compared to the integrated type (Tangwe and Simon, 2017). The reasonable high temperature lift at the condenser of the split type ASHP water heater can be attributed to the thermo-physical properties of the refrigerant and the closed loop circuit design (Douglas, 2008; Marrison *et al*., 2004). Again, the high temperature of the refrigerant recorded at the inlet and outlet of the condenser can be responsible for the potential lowering of the lifespan of the split type system as opposed to the integrated type.

Chapter seven covers the establishment of a benchmark simplified mathematical models to compare the COP of a split and integrated type ASHP water heaters using predictors as relative humidity and the difference between hot water set point temperature and the ambient temperature. The ranking of the predictors based on the importance of weight contributions to the COP using the reliefF test revealed that the difference between the hot water set point temperature and the ambient temperature was a primary factor while the relative humidity was a secondary factor.

The derived mathematical models for the COP of both types of ASHP water heaters had very good determination coefficients of over 90% and there existed a strong visual correlation between the actual determined COP and the predicted COP for both types of ASHP water heaters. Furthermore, the increase in relative humidity increased with an increase in the COP provided the temperature parameter was held constant. Also, the COP increased with a lower hot water set point temperature and an increase in ambient temperature. Although, the increase in ambient temperature led to an increase in COP, it is worth mentioning that as an independent quantity, it is not a primary factor (Tangwe *et al*., 2014). Finally, the derived regression models were low cost to develop as meteorological data could be utilised from the nearest weather station provided the logging interval was in 5 minutes.

Chapter eight provides surface fitting multiple regression models to evaluate the COP of the split and the integrated type ASHP water heaters and which incorporated the electrical energy consumed by the system and the product of ambient temperature and relative humidity as the predictors. In-depth analysis was conducted to demonstrate the variation of each predictor with the COP for both the split and integrated type ASHP water heaters using the two dimensional, multi-contour plots simulation and the three dimensional surface mesh plots (Tangwe and Simon, 2018). The results revealed that the split type performed better unlike the integrated type and both predictors were primary factors as depicted by the ReliefF algorithm test. It was further deduced from the ReliefF test that the electrical energy consumption, contribution by weight of importance to the COP was greater than that of the product of ambient temperature and relative humidity for both systems. The two derived models of the types of ASHP water heaters for both the summer and winter performance were capable of predicting the desired response, with over 90% determination coefficient and less than 2% mean square bias error.

Finally, the derived models showed that the increase in electrical energy consumption by both types of ASHP water heaters was accompanied by a subsequent increase in COP provided the other predictor was kept constant. The variation of COP with each predictor (electrical energy consumed and product of ambient temperature and relative humidity) was demonstrated with a confidence bound of 95% using the 2D multi-contour plots simulation. Finally, in all the seasons of performance monitoring of both types of ASHP water heaters, the built and developed mathematical model of the split type system outperformed the integrated type counterpart.

Chapter nine details the results achieved from the development and building of multivariate regression models to predict the COP of both types of ASHP water heaters with input parameters including ambient temperature, relative humidity, the difference between the temperature of the refrigerant at the discharge and suction ends of the compressor and the difference between the temperature of the refrigerant at the inlet and outlet ends of the condenser. The majority of developed models used to predict the COP of the ASHP water heaters were built from first principles and applying the thermodynamic laws, heat transfers and fluid mechanics concepts of individual components of the closed loop circuit of the vapour compression refrigeration cycles (Tangwe and Simon, 2018). It should be noted that the established models employed the holistic system approach and the ambient conditions as the predictors. The statistical ranking of the predictors by the reliefF method showed that the thermo-physical properties of the refrigerant (i.e. difference in the temperature of the refrigerant at the inlet and outlet of the compressor and condenser) were primary factors while the ambient conditions (ambient temperature and relative humidity) were secondary factors. The accuracy of the derived model was higher relative to the previous models discussed in chapter seven and eight, respectively. This could be accounted for by the increase in the number of contributing predictors in the developed mathematical models in chapter nine.

The two-dimensional multi-contour plots simulation was also used to show the variation of each predictors with the COP while the others were held constant for both types of ASHP water heaters. It was also depicted that the difference in the temperature of the refrigerant at the inlet and outlet of the condenser contributed the most to the COP, for the two types of ASHP water heaters. More so, the multiple comparison procedure test demonstrated that there was a significant difference in the COP and the split type outperformed the integrated type ASHP water heater. Finally, a real-time simulation application to predict the COP of both the split and integrated type ASHP water heater was developed in the Simulink environment. The simulation application can be of great assistance in enabling heat pump manufacturers and heat pump engineers as well as installers to predict the COP from the simulating perspective. Again, via the utilisation of the simulation application, the maximum COP during VCRC of the ASHP water heaters can be achieved with the optimal operating conditions predicted.

10.2 Originality of research

The research novelty can be captured at both national and international levels on the following merits;

i. The DAS employed in the performance monitoring of the two types of ASHP water heaters were the first of its kind to be developed and built in South Africa.
ii. The utilisation of the two-dimensional multi contour plots simulation to demonstrate the variations of the predictors to the COP of the two types of ASHP water heaters stand out so classic, as no research conducted on the COP of residential ASHP water heaters has analysed the predictors influencing it using such a technique.
iii. The classification of relevant predictors of the COP of the ASHP water heaters into primary and secondary factors based on the deterministic multiple linear regression models were very unique and have never be analysed statistically.
iv. The research is the first of its kind to use a full year data from the performance monitoring of the two types of ASHP water heaters to develop mathematical models and also to design a simulation application to predict the COPs.

10.3 Research findings

The following strong and generalise findings were depicted from the research;

i. There existed no mean significant difference in the average ambient conditions under which both types of ASHP water heaters were operated, based on the different scenarios of hot water drawn off, but the COP of the split type without an electric backup performed better and with a significant mean difference to that of the integrated type with an electric backup.

ii. The difference in the temperature of the refrigerant at the inlet and outlet of the condenser of the split type system was higher than that of the integrated type system and the superheated refrigerant vapour temperature at the inlet as well as the refrigerant at the outlet of the condenser were both higher in the split type relative to the integrated type.

iii. The accuracy of the mathematical models for the two types of ASHP water heaters increased as the number of contributing predictors to the COP also increased.

iv. There was a strong agreement between the scaling constants for each of the input parameters with respect to the desired response for the split and integrated type ASHP water heaters in comparison to the slopes of each of the predictors to the output in the 2D multi contour plots simulation.

v. The implementation of isotherm blankets on the storage tanks of both types of ASHP water heaters do not offer a significant reduction in the standby losses.

vi. Irrespective of the difference in the COP of both types of ASHP water heaters, the both systems demonstrated to be of potential viability based on the overall year round performance and payback period.

10.4 Conclusions

The following concluding statements are worth putting forth;

i. The DAS is the first of its kind to be designed in South Africa that could monitor the instantaneous and average thermal and electrical properties of the ASHP water heaters with more than 90% confidence level. This conclusion was obtained from chapter three and provided the solution for research question i and objective i as shown in the matrix table 1.1.

ii. The retrofitting of ASHP units to existing geysers could provide a permanent solution on electrical demand and energy consumption reduction. Hence, it can assist in minimising the constraint on the Eskom national grids. Both types of ASHP water heaters are viable technologies for sanitary hot water heating with favourable payback period. This conclusion was obtained from chapter four and provided the solution for research questions ii & iii and objectives ii & iii, as presented in the matrix table 1.1.

iii. The COP was also impacted by the input electrical energy consumption. There was a significant difference between the temperature of the refrigerant at the inlet and outlet of the condenser located in the split type ASHP water heater when compared to the integrated type. This conclusion was obtained from chapter nine and provided the solution for research question vii and objective vii as shown in the matrix table 1.1. iv. There exists no significant mean difference in the electrical energy consumption to compensate for the standby thermal energy losses of the ASHP water heaters without and with an isotherm blanket on the storage tanks. This conclusion was obtained from chapter five and provided the solution for research question iv and objective iv as stated in the matrix table 1.1.

v. The established multiple linear regression models had good determination coefficients and exhibited good fits with the actual calculated COPs for both types of ASHP water heaters. This conclusion was obtained from chapters seven, eight and nine and provided the solution for research question viii and objective viii as indicated in the matrix table 1.1.

vi. The surface fitting modelling of the COP of the split and integrated type ASHP water heaters aided by the 2D multi contour plots simulation can easily be used to visualise the system performance. This conclusion was obtained from chapters eight and nine and provided the solution for research question ix and objective ix as specified in the matrix table 1.1.

vii. The difference in the temperature of the refrigerant between the inlet and outlet of the condenser was the strongest predictor and a primary factor to the COP in both the split and integrated type ASHP water heaters. Also, the simulation application can be used to simultaneously compare the COP of both types of ASHP water heaters. This conclusion was obtained from chapter nine and provided the solution for research questions vi & x and objectives vi & x as specified in the matrix table 1.1.

10.5 Future works

i. To monitor the performance of all the categories of residential split and integrated type ASHP water heaters installed in actual homes with occupants and in all the regions of South Africa. ii. To compare the performance of identical types of ASHP water heaters in both inland and coastal region of South Africa.

iii. To conduct a full techno-economic analysis of both split and integrated type ASHP water heaters installed in homes with occupants.

iv. To develop a system analysis model of residential ASHP water heaters with input parameters including the environmental conditions, refrigerant thermo-physical properties and the volume of hot water drawn off by the occupants.

v. To assess the impact of the various design configurations of the heat exchangers (evaporator and condenser) of the residential ASHP water heaters in South Africa markets.

10.6 Recommendations

i. There is a need for training heat pump water heater installers who can take up the responsibility of the installation, maintenance and repairs of the system since at the moment the technology is fairly new in South Africa.

ii. During installation of the split type system, it should be ensured that the ASHP unit and the storage tank contain an isolating gate valve on the connected pipes, so that in case of any fault developed in the ASHP unit, it can be isolated from the tank with relative ease.

iii. Policy makers should encourage the promotion of this technology as an energy conservation measure to reduce global warming potential and environmental pollutants by providing incentives to offset the daunting capital cost of the ASHP water heater.

iv. Widening of the scope of campaign in order to sensitise the masses and create awareness of this technology can go a long way to increase the number of house owners willing to retrofit their existing geysers with the ASHP unit. Otherwise, the installations of a split or integrated type ASHP water heater in new buildings as the performance of the system does not depend primarily on the building design or orientation as is the case with solar water heater installation.

v. The design of a prototype hybrid photovoltaic assisted air source heat pump unit for sanitary hot water heating. The proposed innovative heat pump unit will be used to retrofit geyser for sanitary hot water production. The required electrical energy to operate both motors of the compressor and fan will be provided by the photovoltaic panel. The electrical energy of the photovoltaic panel will be stored in battery bank house by the heat pump unit. During the VCRC, the DC electricity from the battery along with the power electronic integrated circuit board embedded in the heat pump unit will power the compressor, water circulating pump and the fan. Nevertheless, the system will be designed such that the grid electricity will be on standby and can be utilised to run the heat pump in a scenario wherein, the battery electricity is insufficient or completely discharged.

vi. ESCO (Energy service company) and installers of ASHP water heaters should carefully check the water quality in the area where the intended ASHP is going to be installed as hardness of water has an adverse effect on the lifespan of the system.

vii. Except insisted by home owners', installers and ESCO should not recommend the introduction of an isotherm blanket on system's tank.

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Appendices

Appendix I: MATLAB codes for regression analysis and statistical tests

A.) regress

Multiple linear regression

Syntax

```
b = regress(y,X) [b,bint]
= regress(y,X)
[b,bint,r] = regress(y,X)
[b,bint,r,rint] = regress(y,X)
[b,bint,r,rint,stats] = regress(y,X)
[...] = regress(y,X,alpha)
```
Description

b = regress(y,X) returns a *p*-by-1 vector b of coefficient estimates for a multilinear regression of the responses in y on the predictors in X. X is an *n*-by*p* matrix of *p* predictors at each of *n* observations. y is an *n*-by-1 vector of observed responses. regress treats NaNs in X or y as missing values, and ignores them.

If the columns of X are linearly dependent, regress obtains a basic solution by setting the maximum number of elements of b to zero.

[b,bint] = regress(y,X) returns a *p*-by-2 matrix bint of 95% confidence intervals for the coefficient estimates. The first column of bint contains lower confidence bounds for each of the *p* coefficient estimates; the second column contains upper confidence bounds.

If the columns of X are linearly dependent, regress returns zeros in elements of bint corresponding to the zero elements of b.

[b,bint,r] = regress(y,X) returns an *n*-by-1 vector r of residuals.

[b,bint,r,rint] = regress(y,X) returns an *n*-by-2 matrix rint of intervals that can be used to diagnose outliers. If the interval rint(i,:) for observation i does not contain zero, the corresponding residual is larger than expected in 95% of new observations, suggesting an outlier.

In a linear model, observed values of y are random variables, and so are their residuals. Residuals have normal distributions with zero mean but with different variances at different values of the predictors. To put residuals on a comparable scale, they are "Studentized," that is, they are divided by an estimate of their standard deviation that is independent of their value. Studentized residuals have *t* distributions with known degrees of freedom. The intervals returned in rint are shifts of the 95% confidence intervals of these *t* distributions, centered at the residuals.

[b,bint,r,rint,stats] = regress(y,X) returns a 1-by-4 vector stats that contains, in order, the R^2 statistic, the *F* statistic and its *p* value, and an estimate of the error variance.

Note: When computing statistics, X should include a column of 1s so that the model contains a constant term. The *F* statistic and its *p* value are computed under this assumption, and they are not correct for models without a constant. The *F* statistic is the test statistic of the F-test on the regression model, for a significant linear regression relationship between the response variable and the predictor variables.

The R^2 statistic can be negative for models without a constant, indicating that the model is not appropriate for the data.

[...] = regress(y,X,alpha) uses a 100*(1-alpha)% confidence level to compute bint and rint. **References**

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B.) anova1

One-way analysis of variance

- p = anova1(y)
- p = anova1(y,group)
- p = anova1(y,group,displayopt)
- [p,tbl] = anova1(___)
- [p,tbl,stats] = anova1(___)

Description p = anova1(y) returns the *p*-value for a balanced one-way ANOVA. It also displays the standard ANOVA table (tbl) and a box plot of the columns of y. anova1 tests the hypothesis that the samples in y are drawn from populations with the same mean against the alternative hypothesis that the population means are not all the same.
p = anova1(y,group) returns the *p*-value for a balanced one-way ANOVA by group. It also displays the standard ANOVA table and a box-plot of the observations of y by group.

p = anova1(y,group,displayopt) enables the ANOVA table and box plot displays when displayopt is 'on' (default) and suppresses the displays when displayopt is 'off'.

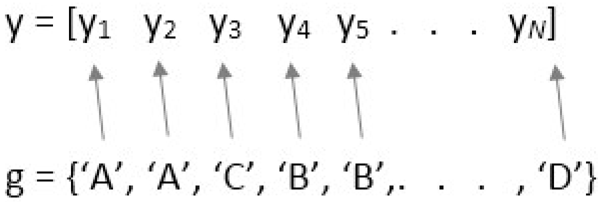
[p,tbl] = anova1(___) returns the ANOVA table (including column and row labels) in the cell array tbl. To copy a text version of the ANOVA table to the clipboard, select **Edit > Copy Text**.

[p,tbl,stats] = anova1(___) returns a structure, stats, which you can use to perform a multiple comparison test. A multiple comparison test enables you to determine which pairs of group means are significantly different. To perform this test, use multcompare, providing the stats structure as an input argument. □ Perform One-Way ANOVA **Input Arguments**

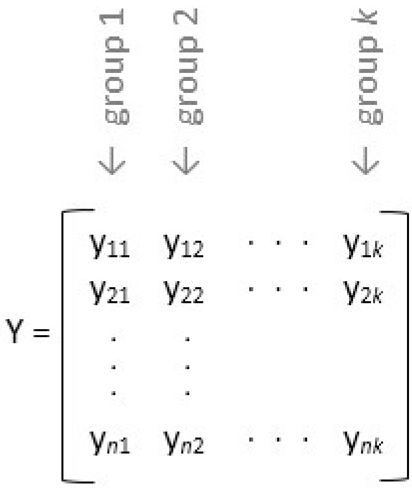
y — sample datavector | matrix

Sample data, specified as a vector or a matrix.

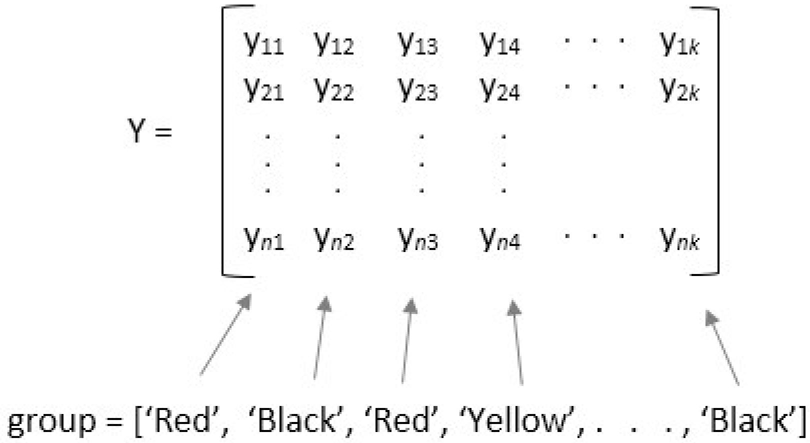
- If y is a vector, you must specify the group input argument. group must be a categorical variable, numeric vector, logical vector, string array, or cell array of strings, with one name for each element of y. The anova1 function treats the y values corresponding to the same value of group as part of the same group. Use this design when groups have different numbers of elements (unbalanced ANOVA).



- If y is a matrix and you do not specify group, anova1 treats each column of y as a separate group. In this design, the function evaluates whether the population means of the columns are equal. Use this design when each group has the same number of elements (balanced ANOVA).



- If y is a matrix and you specify group, then group must be a character array or cell array of strings, with one name for each column of y. The anova1 function treats the columns that have the same group name as part of the same group.

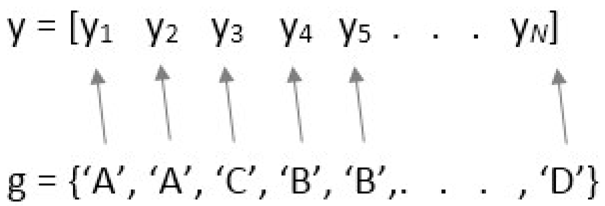


If group contains empty or NaN valued cells or strings, anova1 disregards the corresponding observations in y. **Data Types:** single **|** double

group — Grouping variablenumeric vector **|** logical vector **|** character array **|** cell array of strings

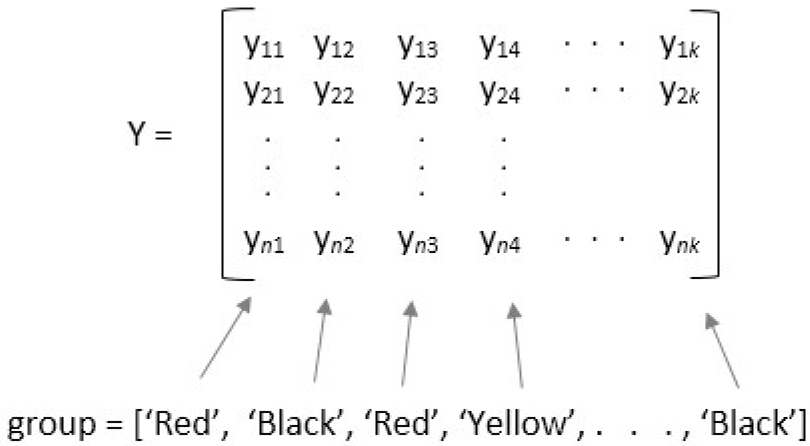
Grouping variable, specified as a numeric or logical vector, character array, or a cell array of strings, containing group names.

- If y is a vector, group must be a categorical variable, numeric vector, logical vector, string array, or cell array of strings, with one name for each element of y. The anova1 function treats the y values corresponding to the same value of group as part of the same group.



N is the total number of observations.

- If y is a matrix, then group must be a character array or cell array of strings, with one group name for each column of y. The anova1 function treats the columns of y that have the same group name as part of the same group.



If you do not want to specify group names, enter an empty array ([]) or omit this argument.

If group contains empty or NaN valued cells or strings, the corresponding observations in y are disregarded.

For more information on grouping variables, see Grouping Variables. For example, if y is a vector, with observations categorized into groups 1, 2, and 3, then you can specify the grouping variables as follows.

Example: 'group',[1,2,1,3,1,...,3,1]

For example, if y is a matrix, with six columns categorized into groups red, white, and black, then you can specify the grouping variables as follows.

Example: 'group',{'white','red','white','black','red'} **Data Types:** single | double | logical | char | cell displayopt — Indicator to display ANOVA table and box plot'on' (default) | 'off' **Output Arguments**

---

p — *p*-value for the *F*-testscalar value

p-value for the *F*-test, returned as a scalar value. *p*-value is the probability that the *F*-statistic can take a value larger than the computed test-statistic value. anova1 tests the null hypothesis that all group means are equal to each other against the alternative hypothesis that at least one group mean is different from the others. The function derives the *p*-value from the cdf of the *F*distribution. A*p*-value that is smaller than the significance level indicates that at least one of the sample means is significantly different from the others. Common significance levels are 0.05 or 0.01.

tbl — ANOVA tablecell array

stats — Statistics for multiple comparison testsstructure

More About

Box-Plot

anova1 returns box plots of the observations in y, by group. Box plots provide a visual comparison of the group location parameters.

If y is a vector, then the plot shows one box for each value of group. If y is a matrix and you do not specify group, then the plot shows one box for each column of y. On each box, the central mark is the median and the edges of the box are the 25th and 75th percentiles (1st and 3rd quantiles). The whiskers extend to the most extreme data points that are not considered outliers. The outliers are plotted individually. The interval endpoints are the extremes of the notches. The extremes correspond to $q2 – 1.57(q3 – q1)/\text{sqrt}(n)$ and $q2 + 1.57(q3 – q1)/\text{sqrt}(n)$, where $q2$ is the median (50th percentile), $q1$ and $q3$ are the 25th and 75th percentiles, respectively, and n is the number of observations without any NaN values.

Two medians are significantly different at the 5% significance level if their intervals do not overlap. This test is different from the *F*-test that ANOVA performs, but large differences in the center lines of the boxes correspond to large *F*-statistic values and correspondingly small *p*-values. For more information about box plots, see boxplot.

- One-Way ANOVA
- Multiple Comparisons

References

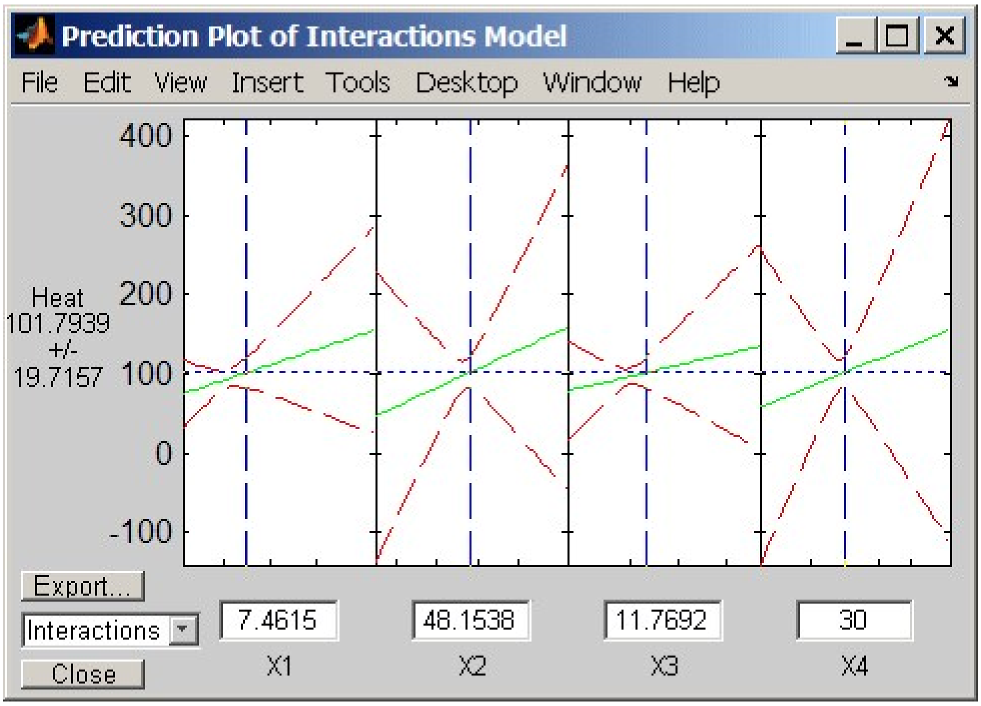
[1] Hogg, R. V., and J. Ledolter. *Engineering Statistics*. New York: MacMillan, 1987.

C.) rstool

Interactive response surface modeling

Syntax

rstool rstool(X,Y,*model*) rstool(x,y,*model*,alpha) rstool(x,y,*model*,alpha,xname,yname) **Description** rstool opens a graphical user interface for interactively investigating onedimensional contours of multidimensional response surface models.

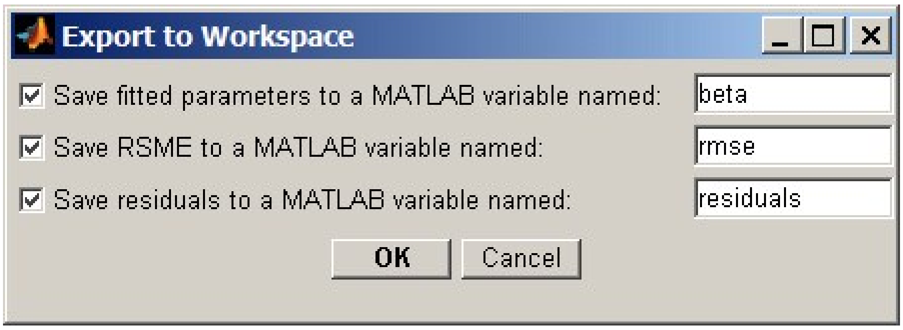


By default, the interface opens with the data from hald.mat and a fitted response surface with constant, linear, and interaction terms.

A sequence of plots is displayed, each showing a contour of the response surface against a single predictor, with all other predictors held fixed. rstool plots a 95% simultaneous confidence band for the fitted response surface as two red curves. Predictor values are displayed in the text boxes on the horizontal axis and are marked by vertical dashed blue lines in the plots. Predictor values are changed by editing the text boxes or by dragging the dashed blue lines. When you change the value of a predictor, all plots update to show the new point in predictor space.

The pop-up menu at the lower left of the interface allows you to choose among the following models:

- Linear — Constant and linear terms (the default)
- Pure Quadratic — Constant, linear, and squared terms
- Interactions — Constant, linear, and interaction terms
- Full Quadratic — Constant, linear, interaction, and squared terms Click **Export** to open the following dialog box:



The dialog allows you to save information about the fit to MATLAB® workspace variables with valid names.

rstool(X,Y,*model*) opens the interface with the predictor data in X, the response data in Y, and the fitted model *model*. Distinct predictor variables should appear in different columns of X. Y can be a vector, corresponding to a single response, or a matrix, with columns corresponding to multiple responses. Y must have as many elements (or rows, if it is a matrix) as X has rows.

The optional input *model* can be any one of the following strings:

- 'linear' — Constant and linear terms (the default)
- 'purequadratic' — Constant, linear, and squared terms
- 'interaction' — Constant, linear, and interaction terms
- 'quadratic' — Constant, linear, interaction, and squared terms

To specify a polynomial model of arbitrary order, or a model without a constant term, use a matrix for *model* as described in x2fx.

rstool(x,y,*model*,alpha) uses 100(1-alpha)% global confidence intervals for new observations in the plots.

rstool(x,y,*model*,alpha,xname,yname) labels the axes using the strings in xname and yname. To label each subplot differently, xname and yname can be cell arrays of strings.

Appendix II: Publications

A.) List of short listed publications in peer review journals and conference proceedings from the field of residential air source heat pump water heaters with emphasis on energy efficiency and heat pump technology considered in the PhD by existing published works.

1. Tangwe, S.L., Simon, M., Simon, M., Meyer, E.L. and Meyer, E.L., 2016. Design of a heat pump water heater performance monitoring system: To determine performance of a split type system. Journal of Engineering, Design and Technology, 14(4), pp.739-751. (Part of chapter 3)

2. Tangwe, S., Simon, M. and Meyer, E. 2014. A techno-economic viability of a residential air source heat pump water heater: Fort Bueafort, South Africa. International Journal of Engineering Science and Research Technology, 3(10): pp.504-510 October, 2014, ISSN: 2277-9655. (Part of chapter 4)

3. Tangwe, S., Simon, M. and Meyer, E., 2015, March. Quantifying residential hot water production savings by retrofitting geysers with air source heat pumps. 23rd International Conference on the Domestic Use of Energy (DUE), 2015 (pp. 235-241). IEEE Xplore, Publisher: IEEE, ISSN: 978-0-9922-0419-8. (Part of chapter 4)

4. Tangwe, S, Michael Simon and Edson Meyer, 2017. Residential air source heat pump water heaters as renewable and energy efficient systems. 25th Southern African Universities Power Engineering Conference, University of Stellenbosch, South Africa. 30th Jan-01 Feb 2017. Pp 170-175, ISBN 978-0-620-74503-1. (Part of chapter 4).

5. Tangwe, S., Simon, M. and Meyer, E., 2014. Analytical Evaluation of the Energy Losses of an Air Source Heat Pump Water Heater: A Retrofit Type. Journal of Energy and Power Engineering, 8(7), pp.1251-1257. (Part of chapter 5)

6. Tangwe, S., Simon, M. and Meyer, E., 2017, April. Impact of standby losses and potential reduction by installation of isotherm blanket on the

hot water cylinders. 25th International Conference on Domestic Use of Energy (DUE), 2017 (pp. 101-109). IEEE Xplore, Publisher: IEEE, ISSN: 978-0-9946759-2-7. (Full chapter 5)

7. Tangwe S, Rubengo F and Simon M. 2016. Comparative analysis of the performance of an integrated and retrofit type air source heat pump water heaters by diagnostic characterization. 15th International Conference on Sustainable Energy Technologies – SET 2016 (19th – 22nd of July 2016), National University of Singapore, Singapore. (Part of chapter 6)

8. Tangwe, S., Simon, M. and Mhundwa, R., 2018. The performance of split and integrated types air-source heat pump water heaters in South Africa. Journal of Energy in Southern Africa, 29(2), pp.12-20. (Entire chapter 6)

9. Tangwe, S., Simon, M., Meyer, E.L., Mwampheli, S. and Makaka, G., 2015. Performance optimization of an air source heat pump water heater using mathematical modelling. Journal of Energy in Southern Africa, 26(1), pp.96-105. (Part of chapter 7)

10. Tangwe, S., Simon, M. and Meyer, E., 2015. Models based simulation of the coefficient of performance of a domestic heat pump water heater. 3rd Southern African Solar Energy Conference, South Africa, 11-13 May, 2015. pp.353-358. ISBN: 978-1-77592-109-7, Available at: http://hdl.handle.net/2263/49520. (Part of chapter 8)

11. Tangwe, S.L., Simon, M. and Meyer, E.L., 2017. Prediction of Coefficient of Performance and Simulation Design of an Air Source Heat Pump Water Heater. Journal of Engineering, Design and Technology, 15(3). pp.378-394 (Part of chapter 8)

12. Tangwe, S.L., Simon, M. and Meyer, E.L., 2018. Evaluation of performance of air source heat pump water heaters using the surface fitting models: 3D mesh plots and 2D multi contour plots simulation.

Thermal Science and Engineering Progress, 5, pp.516-523. (Entire chapter 8)

13. Tangwe, S., Simon, M. and Meyer, E., 2014. Mathematical modeling and simulation application to visualize the performance of retrofit heat pump water heater under first hour heating rating. *Renewable Energy*, *72*, pp.203-211. (Part of chapter 9)

14. Tangwe, S, Michael Simon and Edson Meyer, 2016. Dynamic system modelling as a robust tool to evaluate the performance of domestic integrated and split type air source heat pump water heaters. 4 th Southern African Solar Energy Conference. (30 Oct – 01 st Nov 2016), University of Stellenbosch, South Africa. pp.87-93, ISBN: 978-0-79721658-7 (Part of chapter 9)

B.) List of publications in peer review journals and conference proceedings from the field of residential air source heat pump water heaters with emphasis on energy efficiency and heat pump technology but not considered in the PhD by existing published works

15. Simon, M., Tangwe, S. and Meyer, E. 2014. The impact of solar aerothermal energy in energy efficiency sector, Fort Hare Papers, Multidisciplinary Journal of the university of Fort Hare, Volume 21, No 1,2014, pp.82-92, ISSN: 0015-8054.

16. Tangwe, S., Simon, M. and Meyer, E. 2015. The influence of the primary refrigerant thermo-physical properties on the performance of a domestic air source heat pump water heater. *14 th International Conference on Sustainable Energy Technology*. (25 – 28 th August 2015), Nottingham, UK. Nottingham, UK. University of Nottingham: Architecture, Energy & Environment Research Group. Volume 1, pp853-863. Available from: eprints.nottingham.ac.uk. ISBN 9780853583134

17. Tangwe, S., Simon, M. and Meyer, E. 2013. Computational approach to evaluate performance of split type residential air source heat pump water heater at different operational state, *12 th International Conference on Sustainable Energy Technology*. (26 – 29 th August 2013), Hongkong.

18. Tangwe, S., Simon, M. and Meyer, E. 2014. The impact of solar aerothermal energy in energy efficiency sector. *2 nd Southern African Solar Energy Conference*. (27 – 29 th January 2014), Port Elizabeth, South Africa. Published in Fort Hare Multi-Disciplinary Journal (Vol 21, No 1, 2014).

19. Tangwe, S., Simon, M. and Meyer, E., 2014, April. Empirical model to determine electricity demand reduction in the South Africa residential sector by retrofitting geysers with air source heat pump. 22nd International Conference on the Domestic Use of Energy (DUE), 2014 (pp. 229-235). IEEE Xplore, Publisher: IEEE, ISSN: 978-0-9922041-5-0

20. Tangwe, S., Simon, M. and Meyer, E. 2013, July. Modelling and real time simulation of the instantaneous performance of residential air source heat pump water heater. In South African Institute of Physics, 58th Annual Conference (SAIP 2013). Pp496-501, ISBN: 978-0-620-62819-8

21. Tangwe, S., Simon, M. and Meyer, E. 2012, July. Mathematical modelling of the performance of a Carnot's air source heat pump water heater. In South African Institute of Physics, 57th Annual Conference (SAIP 2012). pp 542-549, ISBN: 978-1-77592-070-0

22. Tangwe, S., Simon, M. and Meyer, E. 2015. A preliminary study to ascertain the techno-economic viability of residential air source heat pump water heater: Fort Beaufort, South Africa, *23 rd Southern African Universities of Power Engineering*, 28 – 30 th January 2015, University of Johannesburg. Pp 164-169, ISBN 978-0-86970-786-9.

23. Tangwe S., Simon M. and E Meyer. Design and construction of a data acquisition system and its implementation in the monitoring of the performance of a domestic air source heat pump water heater. 13th International Conference on Sustainable Energy technologies, 25-28th August, 2014, Geneva. Paper ID: SET2014-E40134

24. Tangwe, S., Simon, M. and Meyer, E. 2013. Experimental investigation to quantify the benefits of residential air source heat pump water heater in South Africa. 5 th International Conference on Applied Energy, (1 – 4 th July 2013), South Africa.

25. Tangwe, S., Simon, M., 2018, April. Comparison of different types air source heat pumps water heaters in South Africa. 26th International Conference on the Domestic Use of Energy (DUE), 2018 (pp. 82-89) IEEE, ISBN 978-0-9946759-4-1.

C.) List of publications in peer review journals and conference proceedings from other research

1. Nwokolo, N., Mamphweli, S., Meyer, E. and Tangwe, S., 2014. Electrical performance evaluation of Johansson biomass gasifier system coupled to a 150 KVA generator. Renewable Energy, 71, pp.695-700.

2. Mhundwa, R., Simon, M. and Tangwe, S.L., 2017. Modelling of an onfarm direct expansion bulk milk cooler to establish baseline energy consumption without milk pre-cooling: A case of Fort Hare Dairy Trust, South Africa. African Journal of Science, Technology, Innovation and Development, pp.1-7.
3. Tangwe, S., Simon, M. and Meyer, E., 2015, August. An innovative optimization technique on performance efficiency verification in a coal thermal power plant unit. 12th International Conference on the Industrial and Commercial Use of Energy (ICUE), 2015 (pp. 325-331). IEEE Xplore, Publisher: IEEE, ISSN: 978-0-6206-5913-0

4. Simon, M., Tangwe, S. and Meyer, E., 2014, August. Model base simulation application to compute achievable increment in a benson coal power plant owing to the implementation of energy efficiency activities.11th International Conference on the Industrial and Commercial Use of Energy (ICUE), 2014 (pp. 1-7). IEEE Xplore, Publisher: IEEE, ISSN: 978-0-9922-0417-4

5. Tangwe, S., Simon, M. and Meyer, E., 2014, August. A quantitative and qualitative methodology to evaluate performance improvement of a coal power plant in South Africa by the virtue of energy efficiency intervention. 11th International Conference on the Industrial and Commercial Use of Energy (ICUE), 2014 (pp. 1-8). IEEE Xplore, Publisher: IEEE, ISSN: 978-0-9922-0417-4

6. Mhundwa, R., Simon, M. and Tangwe, S., 2017, August. Comparative analysis of the coefficient of performance of an on-farm direct expansion bulk milk cooler. In Industrial and Commercial Use of Energy (ICUE), 2017 International Conference on the (pp. 1-7). IEEE

7. Tangwe, S., Simon, M. and Meyer, E., 2016, March. An experimental study to determine the performance and potential saving of swimming pool air source heat pump water heater. 23th International Conference on the Domestic Use of Energy (DUE), 2016 (pp. 1-6). IEEE Xplore, Publisher: IEEE, ISSN:978-0-9946-7590-3.

8. Yongoua, J., Tangwe, S. and Simon, M., 2016, March. A review on the performance assessment and optimization techniques of air source heat pump water heaters used in South Africa. 23th International Conference on the Domestic Use of Energy (DUE), 2016 (pp. 1-8). IEEE Xplore, Publisher: IEEE, ISSN:978-0-9946-7590-3.

9. Mhundwa, R, Simon, M, Tangwe, S. and Meyer, E. 2015. 'A comparative analysis of the dairy milk cooling systems in selected farms in the

Eastern Cape province of South Africa'. In: Rodrigues, L. ed., Sustainable Energy for a Resilient Future: Proceedings of the 14th International Conference on Sustainable Energy Technologies. (25-27 August 2015), Nottingham, UK. University of Nottingham: Architecture, Energy & Environment Research Group. Volume 1, pp 570-575. Available from: eprints.nottingham.ac.uk. ISBN 9780853583134.

10. Tangwe, S., Mzolo, N., Simon, M. and Meyer, E., 2014, August. Modeling the demand of a Calorifier to establish the baseline before retrofitting it with a commercial air source heat pump. 11th International Conference on the Industrial and Commercial Use of Energy (ICUE), 2014 (pp. 1-8). IEEE Xplore, Publisher: IEEE, ISSN: 978-0-9922-0417-4

11. Tangwe, S., Mzolo, N., Simon, M. and Meyer, E., 2016, August. Qualitative and quantitative methods for the prediction of sanitary hot water consumption of a Calorifier and the justification of proposal of an air source heat pump unit as a retrofit: A case study, University of Fort Hare residence. 13th International Conference on the Industrial and Commercial Use of Energy (ICUE), 2016 (pp. 170-177). IEEE Xplore, Publisher: IEEE, ISSN: 978-0-9946-7591-0.

12. Bantan, M, G., Tangwe, S. and Simon, M. 2016. A review on the thermodynamic and electrical properties of South African acceptable refrigerants as R22 substitutes in domestic air conditioners in South Africa. 23th *International Conference on the Domestic Use of Energy (DUE),* 2016 2016 Mar (pp. 222-228). IEEE Xplore, Publisher: IEEE, ISSN:978-0-9946-7590-3

13. Mhundwa, R, Simon, M. and Tangwe, S. 2016. 'Modeling the electrical energy consumption of an on-farm direct expansion bulk milk cooler in the Eastern Cape Province of South Africa: A case study of Fort Hare Dairy Trust'. Accepted in the Proceedings of the 15th International Conference on Sustainable Energy Technologies, 19-22 July 2016, National University of Singapore, Singapore.

14. Mhundwa, R., Simon, M. and Tangwe, S., 2016, August. Low-cost empirical modelling to determine cooling savings in a dairy plant using a pre-cooler. 13th International Conference on the Industrial and Commercial Use of Energy (ICUE), 2016 (pp. 57-62). IEEE Xplore, Publisher: IEEE, ISSN: 978-0-9946-7591-0.

15. Nothando N., M. Simon and Tangwe S. 2016. Evaluation of an empirical model for a flat plat solar collector. *24 th Southern African Universities of Power Engineering*. (26 – 28 th January 2016), Vaal University of Technology.pp564-p569, ISBN: 978-1-77012-386

16. Yongoua, J., Tangwe, S. and Simon, M. 2017, Best operation time of air source heat pump water heaters for optimal energy management. 25th Southern African Universities Power Engineering Conference, University of Stellenbosch, South Africa. 30th Jan-01st Feb 2017. Pp176-181, ISBN 978-0-620-74503-1

17. Mhundwa, R, Simon, M. and Tangwe, S. 2017, Energy management opportunities in raw milk cooling in a dairy farm: the potential of using a raw milk pre-cooler, 25th Southern African Universities Power Engineering Conference, University of Stellenbosch, South Africa. 30th Jan-01st Feb 2017. Pp 164-169, ISBN 978-0-620-74503-1

18. Bantan, G.M., Tangwe, S.L. and Simon, M., 2017, April. Performance monitoring and prediction of a domestic split-type air conditioner. 24th International Conference on the Domestic Use of Energy (DUE), 2017 (pp. 220-226). IEEE Xplore, Publisher: IEEE, ISSN: 978-0-9946759-2-7.

19. Yongoua, J., Tangwe, S. and Simon, M., 2017, April. Computational design of an optimized control system for a split-type air source heat pump water heater. 24th International Conference on Domestic Use of Energy (DUE), 2017 (pp. 248-255). IEEE Xplore, Publisher: IEEE, ISSN: 978-0-9946759-2-7.

20. Bantan, G.M., Tangwe, S.L. and Simon, M., 2017, Energy Impact Quantication of Air Conditioning as a Replacement for Traditional Space Conditioning Devices. 25th Southern African Universities Power Engineering Conference, University of Stellenbosch, South Africa. 30th Jan-01st Feb 2017Pp182-pp187, ISBN 978-0-620-74503-1

21. Mandlenkosi, S., Tangwe, S. and Simon, M., 2018, April. Statistical analysis of split type air source heat pump water heaters. 26th International Conference on the Domestic Use of Energy (DUE), 2018 (pp.136-143). ISBN 978-0-9946759-4-1.

22. Tangwe, S., Simon, M. and Qayi, S., 2018, April. An innovative benchmark testing to quantify saving by retrofitting geyser with Hot spot device. 26th International Conference on the Domestic Use of Energy (DUE), 2018 (pp.144 -149). ISBN 978-0-9946759-4-1.

23. Bantan, G.M., Tangwe, S.L. and Simon, M., 2018, A review on the energy and environmental impact of three generations of space conditioning devices. 26th Southern African Universities Power Engineering Conference, University of Witwatersrand, Johannesburg, South Africa. 24th -26th Jan 2018. Paper no 119

24. Yongoua, J., Tangwe, S.L. and Simon, M., 2018, Analytical evaluation of the thermal energy losses of a residential air source heat pump water heater. 26th Southern African Universities Power Engineering Conference, University of Witwatersrand, Johannesburg, South Africa. 24th -26th Jan 2018. Paper no 110

D.) List of publications in scientific magazines

1. Tangwe, S., Simon, M. and Meyer, E. 2014. Model based simulation application: Coal power plant. Published in Electricity + Control magazine: Publisher Crown Magazine, pages 26 – 30.

2. Tangwe, S., Simon, M. and Meyer, E. 2014. Evaluation of the performance of a coal power plant: Publisher: Energize.

E.) List of publications of full book

1. Tangwe, S., 2014, Performance monitoring and mathematical modelling of an air source heat pump water heater. ISBN number: 978-3-65954375-3. Publisher: Lambert Academic Publisher (LAP), July 2014.

Appendix III: Letter of confirmation of authors' contributions

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15 th May 2018

The Administration
University of Sunderland
UK

Dear Sir/Madam

A letter of confirmation of authors' contributions to published works that make up the PhD thesis by publications of Mr Stephen Loh Tangwe

We, the undersigned author and co-authors of the list of scientific and engineering accredited peer- reviewed manuscripts that the above named candidate (**Stephen Loh Tangwe**) did opt for a PhD in Engineering by existing published or creative works in your reputable university, hereby confirm that are his research work. **Mr. S L Tangwe** has been responsible for the conceptualization, implementation of the research, development of the draft and final manuscripts while Prof Michael Simon and Prof Edson Meyer were his promoters under the Fort Hare Institute of Technology at the University of Fort Hare, Alice Campus, South Africa. Hence, his position as the author and corresponding author of those manuscripts is without contention.

Following the viva voce examination and the request to confirm **Mr. S L Tangwe** and the co-authors contributions to the selected published manuscripts, we hereby, wish to inform your prestigious University that the candidate contribution stands at 100%, while that of Prof Michael Simon and Prof Edson Meyer were 0% from the perspective of technical inputs. This implies, as promoters, our role was to provide financial supports for the acquisition of research equipment as the funding grant holders of Eskom and Department of Science and Technology, which were the funders of this research.

The three other co-authors (Prof Mamphweli Sampson, Prof Golden Makaka and Dr Russell Mhundwa) who appeared in one of the selected publications were colleagues under the institute and provided moral and non-technical supports throughout the conceptualization and formation of the manuscript.

Precisely, there is no conflict of interest and the consensus was reached between the candidate (author) and all his co-authors, for him to use the existing published works to apply for PhD by publications. We hope this letter will serve the purpose for which it is written.

Kind Regards

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Appendix IV: Title and abstract pages for the publications considered in the thesis

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Design of a heat pump water heater performance monitoring system

To determine performance of a split type system

Stephen Loh Tangwe, Michael Simon and Edson L. Meyer
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Abstract

Purpose – This paper aims to show that by using air source heat pump (ASHP) water heater in the residential sector, the energy consumption from sanitary hot water production can be reduced by more than 50 per cent. Hence, this study quantitatively and qualitatively confirms that domestic ASHP water heater is a renewable and energy efficient device for sanitary hot water production.

Design/methodology/approach – Design and building of a data acquisition system comprises a data logger, power meters, flow meters, temperature sensors, ambient and relative humidity sensor and an electronic input pulse adapter to monitor the ASHP water heater performance. All the sensors are accommodated by the U30-NRC data logger. The temperature sensors are installed on the inlet pipe containing a flow meter and the outlet pipe of the ASHP unit, the vicinity of both evaporator and expel cold air. An additional temperature sensor and a flow meter that cater for hot water drawn off measurements are incorporated into the data acquisition system (DAS).

Findings – The result from a specific monitoring split type ASHP water heater gives an average daily coefficient of performance (COP) of 2.36 and the total electrical energy of 4.15 kWh, and volume of hot water drawn off was 273 L. These results were influenced by ambient temperature and relative humidity.

Research limitations/implications – The cost involved in purchasing the entire sensors and data logger limits the number and categories of ASHP water heaters whose performance were going to be monitored. Pressure sensors were excluded in the data acquisition system.

Practical implications – The data acquisition system can easily be designed and the logger can also be easily programed. Hence, no high technical or computer skills are needed to install the DAS and to be able to read out the results.

Social implications – Hence, the data acquisition system can be installed on the entire domestic Eskom roll out air source heat pump water heaters to effectively determine the coefficient of performance and demand reductions.

Originality/value – This DAS is the first of its kind to be built in South Africa to be used to determine the performance of an ASHP water heater with high accuracy and precision. DAS is also robust.

Keywords Design, Energy efficiency, Innovation, Heating, Implementation, Alternative and new technologies

Paper type Research paper

The authors are delighted to acknowledge the financial supports from South African electricity supply utility (Eskom) and the Fort Hare Institute of Technology (FHIT) in a bid to enable the authors to purchase the equipment required to design and construct the data acquisition system and also the geyser and the ASHP unit.

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A Techno-Economic Viability of a Residential Air Source Heat Pump Water Heater: Fort Bueafort, South Africa

Stephen L. Tangwe*, Michael Simon and Edson L. Meyer
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Abstracts

sThe utilization of an air source heat pump (ASHP) to retrofit geyser can significantly reduce electricity consumption for sanitary hot water production. Furthermore, optimal operation of the system based on ambient conditions and capacity of hot water usage would enhance both achievable performance and payback time. The study focus on using a data acquisition system to evaluate the performance of an efficiently installed ASHP water heater and hence determine the payback period of the system. Preliminary results depict that during the four months of performance monitoring of the system, the average month-day input energy, coefficient of performance and volume of hot water usage was 3.0 kWh, 260 L and 2.2 respectively. An average monthly energy saved of 125 KWh was achieved while the average ambient temperature and relative humidity of 24.6 oC and 64.2% were recorded for the entire time of operation of the system. Finally, using a multiple comparison test, it was demonstrated that no mean significant difference occurred in both the average week electrical energy and COP for each of the different months throughout the observations. The payback period of the ASHP unit was determined to be less than 6 years from a conservative approach method.

Keywords: Air source heat pump (ASHP); Sanitary hot water; Coefficient of performance (COP); Payback, multiple comparison test, Data acquisition system (DAS).

Introduction

The commonly applicable type of heat pump heaters employ for sanitary hot water heating are the air source and the geothermal or ground source systems. These systems operate on the principle of vapor compression refrigerant cycle. The geothermal air source heat pump water heater possesses a better techno-economic potential to an ASHP water heater by virtue of its relatively constant and higher COP [1, 2]. Both systems can be classified as a renewable energy device, as they all use a given form of renewable energy from their immediate surroundings where the evaporator is located during the vapor compression cycle. The ground source heat pump water heater extract waste heat from underground in the form of geothermal energy while ASHP water heater utilized the heat from the air as aero-thermal energy. The capital cost of ground source heat pump water heater is much higher as compared to an ASHP water heater. ASHP water heat is fast gaining maturity in the market as sanitary hot water production constitutes a significant percentage of monthly energy consumption in the residential sector worldwide. In South Africa, residential hot water heating can contribute to more than 50% of the monthly energy utilization [3]. A far-reaching research conducted to justify in terms of energy usage revealed that the hot water contribution in the domestic sector of South Africa is between 40% to 60% on an average monthly basis [4, 5]. It is worth mentioning that despite the daunting electrical energy consumed for hot water production, not all the thermal energy gained by the hot water is effectively utilized. There are always standby losses which are responsible for 20% to 30% of the total thermal energy gained by hot water contained in a storage tank [6]. Although, ASHP water heater coefficient of performance (COP) value can range from 2 to 4 [7, 8]; it is crucial to note that the system COP depends on the COP of the ASHP unit and the ambient climatic condition [9]. Clearly, the COP could be defined as the ratio of the useful thermal energy gained when water is heated to set point temperature and the electrical energy used by the system during the vapor compression refrigerant cycle. A salient and better understanding of refrigeration cycle of heat pump water heater was given by Ashdown *et al.* (2004) and Sinha and Dysarkar, (2008) [10, 11]. Moreover, the performance can be severely affected by standby losses. Heat pump water heaters also render an extra benefit of dehumidification and space cooling because they pull warm vapor from the air [12]. An efficiently installed residential ASHP water heater can guarantee an improvement on the system performance [13]. The study deal with an in depth performance monitoring of a residential split type ASHP water heater installed in a middle class home (compose of 2 adults and a child) in

Quantifying residential hot water production savings by retrofitting geysers with air source heat pumps

Abstract—Inefficient geysers remain South Africans' most popular and conventional modes of hot water production Today, the air source heat pump (ASHP) water heater is used in the residential sector as energy efficiency technology for sanitary hot water production. It is capable of supplying sanitary hot water with 30 to 40% of the total thermal energy generated from electrical energy; the rest emanates from ambient aero-thermal energy. Sanitary hot water is set at a threshold temperature of 55°C to prevent growth of legionella (bacteria).Therefore, employing this technology could result in a substantial reduction in Eskom peak demand and global warming hazards. This research focused on designing and building (DAS housing) various temperature sensors, power meters, flow meters, relative humidity and ambient temperature sensors, to determine electrical consumption and useful thermal energy gained by the hot water in the geyser and the storage tank of the ASHP water heater. In addition, an empirical calculation of the coefficient of performance of the ASHP water heater was reached. Furthermore, the amount of aero-thermal energy extracted was evaluated based on the temperature difference between the warm air in the vicinity of the evaporator and the cold, dehumidified air expelled from the duct space of the ASHP unit. Finally, results of the electrical energy consumption from sanitary hot water production showed a reduction from 60% to 31% by the retrofitting of the geyser with ASHP.

Keywords: ***ASHP - air source heat pump; geyser; global warming potential reduction; energy efficiency technology; DAS - Data acquisition system, COP-coefficient of performance.***

List of abbreviations

DAS = data acquisition system
COP = coefficient of performance
ASHP = air source heat pump
p = active power in kW
E = active energy in kWh
t = time in hour
PF = power factor
Q_{out} = output useful thermal energy gained
E_{in} = input electrical energy
$\sum Q$ = total thermal energy over 24 hours
$\sum E$ = total electrical energy over 24 hours
pmax = maximum power consumption over the 24 hour
AW=average week
Vd=volume of hot water drawn off
Es= Electrical energy consumed by the system
Eb= Electrical energy used by entire building
Es=Electrical energy consumption of the system
TE= Thermal energy gained
Ec= Electrical energy used by controller

1 INTRODUCTION

Eskom is the sole supplier of electricity in South Africa; more than 90% is generated from coal. The global warming potential because of greenhouse gasses, primarily carbon dioxide, is 510 million tons of which 45% emanates from the generation of electricity from coal [1].In South Africa, domestic energy consumption is typically allocated according to the proportion of various residential energy utilities (water heating, 43%, washing machine, 12.3%, stove, 10.2%, heater, 9.9%, fridge 8.6% and small appliances, 11.2%) [2]. It can be depicted without loss of generality, but based on further research the contribution of energy consumption by sanitary hot water production in the domestic sector ranges between 40 to 60% depending on climatic conditions. Sanitary, water heating in SA is the largest residential use of electrical energy with up to 50% of monthly consumption used for this purpose [3]. It is worth mentioning most hot water devices are traditionally convectional heater (electric geysers) with an average energy factor of 0.92 [4]. The ASHP water heater is a renewable energy device capable of heating water with the majority of the useful thermal output energy derived from ambient aero-thermal energy [5]. It can provide energy saving in the range from 50-70%, as the ASHP unit has a co-efficient of performance ranging from 2 to 4 [6], [7]. The type of hot water storage tank for the ASHP water heater is a real challenge to the hot water temperature inside the tank. Heated water by ASHP of similar volume is at much higher temperature in a dual tank than a single tank system, but heat losses are higher [8]. An ASHP unit comprises evaporator, compressor, condenser and thermal expansion valve connected in a closed circuit by copper pipes with refrigerant as the heat transfer medium. The thermo-physical properties of the refrigerant are priority in ASHP. Extensive research has exploited eco-friendly fluid, replacing R12 (Dichlorodifluoromethane) and R22 (Chlorodifluoromethane) because of their high ozone depletion potential [9]. The special characteristics that present the heat pump with excellent efficiency are it coefficient of performance [10]. In this regard, it is noteworthy a series of researches have effectively evaluated

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RESIDENTIAL AIR SOURCE HEAT PUMP WATER HEATERS AS RENEWABLE AND ENERGY EFFICIENT SYSTEMS

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Abstract: Inefficient geysers still stand as the most popular and conventional modes of hot water production in the country. This study emphasized the used of the data acquisition system housing various temperature sensors, power meters, flow meter, relative humidity and ambient temperature sensor, to determine electrical energy consumption and useful thermal energy gained by the hot water in a 150 l geyser and 150 l storage tanks of the air source heat pump (ASHP) water heaters. The results depicted that the average electrical energy consumptions of the summer months for the geyser, split and integrated types ASHP water heaters were 312.3 kWh, 111.7 kWh and 121.1kWh, respectively. The electrical energy consumption for sanitary hot water production showed an annual reduction of 65% and 58.5% by attempting to assess the viability of the split and integrated type ASHP water heaters, respectively. Finally, the simple payback period for both the split and integrated types ASHP water heater were determined to be 3.9 years and 5.2 years respectively.

Keywords: Split type air source heat pump, Integrated type air source heat pump water heater, Geyser, Data acquisition system, Payback period

1. INTRODUCTION

The ASHP water heater is an energy efficient device for sanitary hot water production. It is capable of using 1 unit of input electrical energy to provide 3 units of useful thermal output energy, assuming a COP of 3 during vapor compression refrigerant cycles. The rest of the useful thermal energy emanates from ambient aero-thermal energy. Sanitary hot water is set at a threshold temperature of 55°C to prevent growth of the bacteria (*L*egionella). Water should be kept at a temperature of a minimum of 55°C (optimally 60°C) so that water at the outlet points of the hot water storage tank can be above 50°C within a minute [15].

Eskom is the sole supplier of electricity in South Africa with more than 90% of its generation coming from coal. The global warming potential because of greenhouse gases, primarily carbon dioxide, is 510 million tons, of which 45% emanates from the generation of electricity from coal [3]. In South Africa, domestic energy consumption contributed to 15-18% of total energy generations and is typically allocated according to the proportions of various residential energy devices (water heating, 43%, washing machine, 12.3%, stove, 10.2%, heater, 9.9%, fridge 8.6% and small appliances, 11.2%) [10]. It can be depicted without loss of generality, but based on further research that the contribution of energy consumption by sanitary

hot water production in the domestic sector ranged from 40 to 60% depending on climatic conditions. Sanitary, water heating in the country is the largest residential consumer of electrical energy with up to 50% of the monthly consumption used for this purpose [8].

It is worth mentioning that most of the hot water devices are the traditional convectional heater (electric geysers)

with an average energy factor of 0.92 [6]. Interestingly, the ASHP water heater is a renewable energy device capable of heating water with the majority of the useful thermal output energy derived from ambient aerothermal energy [9]. It can provide energy saving in the ranged from 50-70%, as the ASHP unit has a coefficient of performance ranging from 2 to 4

[7; 1]. The type of hot water storage tank for the ASHP water heater is a real challenge to the hot water temperature inside the tank. Heated water by ASHP of similar volume is at much higher temperature in a dual tank than a single tank system, but the heat losses are lower for the latter [5]. An ASHP unit comprises of evaporator, compressor, condenser and thermal expansion valve connected in a closed circuit by copper pipes with refrigerant as the heat transfer medium. The thermo-physical properties of the refrigerant are a priority in ASHP. Extensive research has exploited eco-friendly fluid, replacing R12 (Dichlorodifluoromethane) and R22 (Chlorodifluoromethane) because of their high ozone depletion potential [14]. The special characteristics that present the heat pump with excellent efficiency are its coefficient of performance [2].

In this regard, it is noteworthy that series of researches have effectively evaluated heat pump water heater performance. Also, a dynamic model of an ASHP water heater was designed to achieve optimal energy management in a test room [4]. In a bid to avoid constraint on the national grid during peak hours, Eskom targeted rolling out more than 65 500 ASHP up to March 2013 under a residential rebate scheme to achieve a demand reduction of 54 MW [11]. The projected annual cost saving by the implementations of ASHP water heaters as retrofits to existing geysers were determined using the Eskom mega flex tariff [12]. Eskom

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Analytical Evaluation of the Energy Losses of an Air Source Heat Pump Water Heater: A Retrofit Type

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Abstract: ASHP (air source heat pump) water heater is a renewable and energy efficient device used for sanitary hot water production. The system comprises of a storage tank and heat pump connected by pipes. These major units can either be compact as in the integrated model or split as in the retro-fit model. In this research, the analysis of energy losses was performed using SIRAC (the Southern African refrigeration and air conditioning) residential split type heat pump of 1.2 kW input power to retrofit a 200 liter high pressure kwikhot storage tank without hot water being drawn off for the entire monitoring period. Likewise to experimentally determine the losses DAS (data acquisition system) was designed and built to measure T_a (ambient temperature; RH-relative humidity), *RH*, T_o (ASHP outlet water temperature), T_i (ASHP inlet water temperature) and V_h. (volume of water heated by ASHP unit).The results showed that the heat energy gain to compensate standby losses could range from 1.8 kWh to 2.1 kWh with the corresponding electrical energy used by ASHP water heater ranging from 0.55 kWh to 0.66 kWh. The standby losses depend primarily on the V_h, the T_a and the RH while the influence of (T_o − T_i) is secondary. The results can be of valuable interest to manufacturer of retrofit ASHP unit for hot water production when matching the electrical energy required to compensate for the standby losses.

Key words: ASHP, DAS, T_a, V_h, T_o, T_i.

1. Introduction

Sanitary hot water production constitutes a significant percentage of monthly energy consumption in the residential sector worldwide. In South Africa, residential hot water heating can contribute to more than 50% of the monthly energy utilization [1]. A far-reaching research conducted to justify in terms of energy usage revealed that the hot water contribution in the domestic sector of South Africa is between 40% and 60% on an average monthly basis. Fig. 1 shows that 45% of the total energy consumption in a typical South African residence is from hot water heating [2].

It is worth mentioning that despite the daunting electrical energy consumed owing to hot water production, not all the thermal energy gained by the hot water is effectively utilized. There are always standby losses which are responsible for 20% to 30% of the total thermal energy gained by hot water contained in a storage tank [3, 4]. Although, COP (coefficient of performance of ASHP (air source heat pump) water heater can range from 2 to 4 [5, 6]; It is crucial to note that the COP depends primarily on (components used in the close circuit design of the heat pump, volume of hot water heated, hot water set point temperature and mains cold water supply temperature). The secondary factors that influence COP include the ambient temperature and relative humidity. Clearly, the COP could be defined as the ratio of useful thermal energy gained when water is heated to set point temperature and the electrical energy used by the system during the vapour compression refrigerant cycle. A salient and better understanding of refrigeration cycle of heat pump water heater was

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IMPACT OF STANDBY LOSSES AND POTENTIAL REDUCTION BY INSTALLATION OF ISOTHERM BLANKET ON THE HOT WATER CYLINDERS

Stephen Tangwe, Michael Simon, and Edson Meyer

Abstract— Air source heat pump (ASHP) water heater is an energy-efficient device for sanitary hot water production. The study focused on monitoring the electrical energy consumed to compensate for the standby losses of three hot water cylinders without and with isotherm blankets. Accordingly, the analysis of thermal energy losses was performed using 150 L high-pressure geyser and 150 L split and integrated types ASHP water heaters without hot water being drawn off throughout the entire monitoring period. Likewise, to experimentally determine the thermal losses, a data acquisition system (DAS) was constructed to measure the average ambient temperature and relative humidity as well as the cumulative electrical energy to compensate for the standby losses. The results on average electrical energy consumed to compensate for the standby losses of the geyser, split and integrated types ASHP water heaters without the isotherm blankets were 2.71 kWh, 1.33 kWh and 0.94 kWh, respectively. The introduction of a 40 mm thick isotherm blankets on the hot water cylinders resulted in the electrical energy reduction by 18.5%, 15.8% and 3.2 % for the geyser, split and integrated types ASHP water heaters, respectively. The multiple comparison tests revealed a significant difference on the geyser standby losses under the two configurations.

Key words: *ASHP-Air source heat pump, Multiple comparison tests, Isotherm blanket and standby thermal energy losses.*

1 INTRODUCTION

Across the globe, sanitary hot water production constitutes a significant percentage of monthly energy consumption in the residential sector. Specifically, in South Africa, residential hot water heating can contribute to more than 50% of the monthly energy utilisation [11]. A far-reaching research conducted to justify the energy usage revealed that the hot water contribution in the domestic sector of South Africa is between 40% to 60% on a monthly average basis. Figure 1 demonstrates that 45% of the total electrical energy consumption in a typical South African residence is from hot water heating [14].

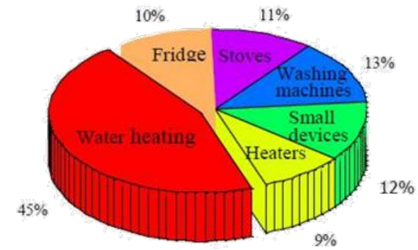

Figure 1: energy consumption in a typical South African residence

It is worth mentioning that despite the daunting electrical energy consumed for hot water production, not all the thermal energy gained by the hot water is effectively utilised. There are always standby thermal losses which are responsible for 20% to 30% of the total thermal energy gained by hot water contained in a storage tank [15]. In light to ASHP water heater, its performance is described by a unique factor known as the coefficient of performance. The COP can range in value from 2 to 4 but it is crucial to emphasis that the COP depends on primary factors (components used in the close circuit design of the heat pump, volume of hot water heated, hot water set point temperature and mains supply cold water temperature) and secondary factors (ambient temperature and relative humidity) [6; 2; 5;10]. Clearly, the COP could be defined as the ratio of useful thermal energy gained when water is heated to set point temperature to the electrical energy used by the system during the vapour compression refrigeration cycle. A salient and better understanding of refrigeration cycle of heat pump water heater were given by Ashdown et al. (2004) and Sinha and Dysarkar, (2008) [1; 13]. Moreover, the performance can be severely affected by standby thermal energy losses [6].

To the best of our knowledge, enormous research has been conducted on standby thermal energy losses, but with emphasis on the geyser, solar water heater and the integrated heat pump water heater. The geyser's standby thermal energy losses were determined in the multi-level expert-modelling, evaluation of geyser load management

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Title: Comparative analysis of the performance of split and integrated type air source heat pump water heater by diagnostic characterization

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Abstract:

The air source heat pump (ASHP) water heater generate sanitary hot water by harnessing the aerothermal energy during the process of vapour compression refrigerant cycle (VCRC). The study focuses on the identification of critical parameters (volume of hot water drawn off, ambient temperature, relative humidity, inlet and outlet temperatures of compressor and condenser) as well as deterministic quantities (time used, power consumption and coefficient of performance (COP) as the indicators to benchmark the efficiencies of the split and integrated ASHP water heaters. The analysis was performed based on two predominant scenarios (first hour heating rating and the heating up cycle due to controlled volume of hot water drawn off) whereby both the integrated and retrofit type ASHP water heaters were undergoing vapour compression refrigerant cycle. A robust and accurate data acquisition system (DAS) was designed and constructed to monitor the performance of both the systems. In all the VCRC scenarios, the average COP was more than 2 with the retrofit type performing better than the integrated type as could be deduced from the higher COP of the retrofit type.

Key words: Air source heat pump (ASHP); Coefficient of performance (COP); Vapour compression refrigerant cycle (VCRC); Data acquisition system (DAS); First hour heating rating.

1. INTRODUCTION

Eskom is the sole supplier of electricity in South Africa with more than 90% of the electrical energy generated coming from coal. The global warming potential because of greenhouse gasses, primarily carbon dioxide, is 510 million tons, of which 45% emanates from the generation of electricity from coal (Bryson, 2011). Sanitary, water heating in South Africa is the largest residential consumer of electrical energy with up to 50% of monthly consumption used for this purpose (Meyer and Tshimankinda, 1998). It is worth mentioning that most hot water devices are traditionally convectional heater (electric geysers) with an average energy factor of 0.92 (Haung and Lin, 1997). Today, the air source heat pump (ASHP) water heater is used in the residential sector as a renewable and energy efficient technology for sanitary hot water production (Morrison,

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The performance of split and integrated types air-source heat pump water heaters in South Africa

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Abstract

Renewable energy technologies that can provide optimum and cost-effective energy savings to mitigate global warming, energy crisis and to achieve energy efficiency continue to be of paramount importance. The present study focused on identifying critical parameters such as the volume of hot water drawn off; ambient temperature; relative humidity; refrigerant temperatures at the inlet and outlet of the compressor and condenser; and deterministic quantities such as time used, power consumption and coefficient of performance (COP) as indicators to benchmark the performance of both the split and integrated types of air-source heat pump (ASHP) water heaters. The basis for analysis was on two predominant scenarios: first-hour heating rating and the heating cycle due to controlled volume of hot water drawn-off wherein both the integrated and split types ASHP water heaters experienced vapour compression refrigeration cycles. A data acquisition system was constructed and implemented to monitor the performance of both systems. The results obtained during summer season showed that, under the scenario of 150 L hot water withdrawal, the average COP of the systems was 3.18 and 2.85 for the split and integrated types respectively. The average power consumed was 1.29 (split type) and 0.85 kW (integrated type). The times of operation were 84 minutes (split type) and 138 minutes (integrated type).

***Keywords**: coefficient of performance; vapour compression refrigeration cycle; renewable energy technologies*

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Performance optimization of an air source heat pump water heater using mathematical modelling

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Abstract
In South Africa, there is an ongoing constraint on the electricity supply at the national grid to meet the demand. Eskom is implementing various measures such as the Integrated Demand Management and the promotion and encouragement of the use of energy efficient devices like an Air Source Heat pump (ASHP) water heater to replace the high electrical energy consuming conventional geysers for sanitary hot water production. The ASHP water heater market is fast gaining maturity. A critical mathematical model can lead to performance optimization of the systems that will further result in the conservation of energy and significant reduction in global warming potential. The ASHP water heater comprises of an ASHP unit and a hot water storage tank. In this study, a data acquisition system (DAS) was designed and built which monitored the energy used by the geyser and the whole building, the temperature at the evaporator, condenser, tank outlet hot water, tank inlet cold water, the ambient temperature and relative humidity in the vicinity of the ASHP evaporator. It is also worthy to mention that the DAS also included to a flow meter and two additional temperature sensors that measured the volume of water heated and inlet and outlet water temperature of the ASHP. This work focused on using the mathematical equation for the Coefficient of Performance (COP) of an ideal Carnot's heat pump (CHP) water heater to develop basic computation in M-file of MATLAB software in order to model the system based on two reservoir temperatures: evaporator temperatures (T_{eap}) of 0°C to 40°C (approximated to ambient temperature, T_a) and condenser temperatures (T_{Con}) set at 50°C, 55°C and 60°C (approximated to the hot water set temperature of 50°C, 55°C and 60°C) respectively. Finally, an analytical comparison of a CHP water heater to the practical ASHP water heater was conducted on a hot water set point temperature of 55°C. From the modelling results, it can be deduced that at 0°C T_{eap}, the COP was 5.96 and 2.63 for CHP and ASHP water heater respectively, at a hot water set temperature of 55°C. Above 20°C T_{eap}, the rate of change of COP increased exponentially for the ideal CHP system, but was constant at 0.01/°C for the practically modelled ASHP water heater.

Keywords: Air source heat pump; coefficient of performance; data acquisition system; mathematical model; Carnot's heat pump

1. Introduction
Hot water heating constitutes a significant percentage of energy consumption in the industrial, commercial and residential sectors worldwide. In South Africa, water heating is the largest residential use of energy, with up to 50% of monthly electricity consumption being used for this purpose (Meyer and Tshimankinda, 1998). The Eskom strategic plan outlook for 2010 to 2030 envisages over 20% reduction of electricity production from coal (Digest of SA Energy statistics, 2009) as shown in Figure 1. One way to achieve this energy conservation could

Models Based Simulation of the Coefficient of Performance of a Domestic Heat Pump Water Heater

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ABSTRACT

Sanitary hot water production make up to over 50 % of the monthly electrical energy consumption in South Africa residential sector. Employing an effective and efficient mechanism for sanitary hot water heating can lead to a substantial energy saving and demand reduction as well as creating a benign environment owing to the decrease of the carbon dioxide emission. Most of the renewable energy devices for sanitary hot water heating utilized the free and abundant solar energy. Domestic air source heat pump (ASHP) water heater is one of the renewable energy device used for sanitary hot water production. In this study, a data acquisition system (DAS) was constructed to measure the predictors (E, electrical energy consumed) and (λ, average product of ambient temperature and relative humidity) and relevant parameters to compute the COP during the vapor compression refrigerant cycle (VCRC) of the ASHP unit. The coefficient of performance (COP) of an ASHP water heater under two different heating up cycle scenarios were critically examined. Modelling and simulation of the COP of the system provided a distinctive opportunity for optimization and prediction of its performance under different operational conditions. It was depicted that the mean COP in the both scenarios of the heating up cycles (firstly, where there was no successive hot water draw off and secondly, with simultaneous hot water draw off) was on average 2 and above. Finally, using the mathematical models in the both scenarios, it was revealed that increases in both predictors (E and λ) can result to decrease in the COP. The one way analysis of variance (ANOVA) test showed, the mean COP of 1.96 and 2.14 for the heating up scenario of simultaneous hot water drawn off and without a successive hot water drawn off. Predictors' weight ranking demonstrated that the contribution from the input parameters was 100 times more during the heating up scenario whereby successive hot water drawn off occurred. The modelled equations were used in the mathematical blocks of the Simulink to design the simulation application of the COP of an ASHP water heater. We therefore concluded that the COP of an ASHP water heater during simultaneous hot water drawn off was higher than without successive hot water drawn off.

Key words: Air source heat pump (ASHP); Coefficient of performance (COP); Data acquisition system (DAS); Models; Simulation application; Vapor compression refrigerant cycle (VCRC).

INTRODUCTION

The residential ASHP water heater is an efficient and a renewable energy device for sanitary hot water production [1]. The COP of an ASHP water heater can range from 2 to 4 and depends on the component design of the system, ambient weather conditions, duct space and the speed of the cold and dehumidify expelling air [2; 3]. The optimal COP of an ASHP water heater can be achieved by an efficient installation of the system [4]. The system COP can also be enhanced by the use of a primary refrigerant of an excellent thermo-physical properties [5]. It is crucial to allude that extensive research has been conducted on the simulation and mathematical modelling of the performance of heat pump water heaters. More elaborately, the performance of a heat pump water heater was simulated using the TRYSYN simulation software package [6]. However, it should be noted that the TRYSYN simulation cannot effectively model the performance of an ASHP water heater owing to the complexity of the metal fins embodying the evaporator. Furthermore, an analytical, mathematical model was also presented to predict the COP of a solar assisted heat pump water heater in correlation to temperatures [7]. A quantitative method can be used to compute the COP of an

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Prediction of coefficient of performance and simulation design of an air-source heat pump water heater

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Abstract

Purpose – The purpose of this study was to build and develop mathematical models correlating ambient conditions and electrical energy to the coefficient of performance (COP) of an air-source heat pump (ASHP) water heater. This study also aimed to design a simulation application to compute the COP under different heating up scenarios, and to calculate the mean significant difference under the specified scenarios by using a statistical method.

Design/methodology/approach – A data acquisition system was designed with respect to the required sensors and data loggers on the basis of the experimental setup. The two critical scenarios (with hot water draws and without hot water draws) during the heating up cycles were analyzed. Both mathematical models and the simulation application were developed using the analyzed data.

Findings – The predictors showed a direct linear relationship to the COP under the no successive hot water draws scenario, while they exhibited a linear relationship with a negative gradient to the COP under the simultaneous draws scenario. Both scenarios showed the ambient conditions to be the primary factor, and the weight of importance of the contribution to the COP was five times more in the scenario of simultaneous hot water draws than in the other scenario. The average COP of the ASHP water heater was better during a heating cycle with simultaneous hot water draws but demonstrated no mean significant difference from the other scenario.

Research limitations/implications – There was a need to include other prediction parameters such as air speed, difference in condenser temperature and difference in compressor temperature, which could help improve model accuracy. However, these were excluded because of insufficient funding for the purchase of additional temperature sensors and an air speed transducer.

Practical implications – The research was conducted in a normal middle-income family home, and all the results were obtained from the collected data from the data acquisition system. Moreover, the experiment was very feasible because the conduction of the study did not interfere with the activities of the house, as occupants were able to carry out their activities as usual.

Social implications – This paper attempts to justify the system efficiency under different heating up scenarios. Based on the mathematical model, the performance of the system could be determined all year round and the payback period could be easily evaluated. Finally, from the study, homeowners could see the value of the efficiency of the technology, as they could easily compute its performance on the basis of the ambient conditions at their location.

Originality/value – This is the first research on the mathematical modeling of the COP of an ASHP water heater using ambient conditions and electrical energy as the predictors and by using surface fitting

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Evaluation of performance of air source heat pump water heaters using the surface fitting models: 3D mesh plots and 2D multi contour plots simulation

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A R T I C L E I N F O

Keywords:
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Surface fitting model
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A B S T R A C T

Modelling of the coefficient of performance (COP) of an air source heat pump water heater can lead to optimisation and prediction of its performance. The study focused on the utilisation of surface fitting models to predict the COPs of a 150 L split type ASHP water heater without an electric backup element and a 150 L integrated type ASHP water heater with an electric backup element. A robust and accurate data acquisition system was employed to measure the predictor parameters (electrical energy consumed (E) and product of ambient temperature and relative humidity (λ)) as well as the thermal properties to enable the computation of the COP during the vapour compression refrigeration cycles of the ASHP unit. It was observed that for both systems, the two predictors were primary factors. The surface fitting models for both systems showed that the COP increases with an increase in E by a rate of 0.30/kWh and 0.28/kWh for the split and integrated type systems, respectively. The models were simple and can be used to predict the COP of both systems with over 95% confidence level, and the determination coefficient of the split and integrated systems were 0.917 and 0.902, respectively. It was also depicted that the COP variation with the predictors in the controlled volume of hot water drawn off (50 L, 100 L and 150 L) under different ambient conditions can be accurately predicted with either the 3D mesh plots or the 2D multi contour plots simulation.

1. Introduction

The air source heat pump (ASHP) water heater is an efficient and a renewable energy device for sanitary hot water production [17]. The excellent efficiency for an ASHP water heater is due to its performance characteristics known as coefficient of performance (COP) [7]. The COP of an ASHP water heater can range from 2 to 4 and depends on the component design of the system, ambient weather conditions (ambient temperature, relative humidity, etc.), duct space and the speed of the cold expelling air [14,2]. The optimal COP of an ASHP water heater can be attained by an effective installation of the system [8]. But, it can be explained that the optimal COP of ASHP water heater could even be predicted from the utilisation of an accurate mathematical model. Notwithstanding, the COP can also be enhanced by the use of a primary refrigerant of an excellent thermo-physical property [9,16]. Salient and thorough exposition and analysis regarding the refrigeration cycle of heat pump water heaters has been presented by Ashdown (2004) [1] and Sinha and Dysarkar [19].

It is crucial to emphasise that extensive research has been conducted on the mathematical modelling of the performance of heat pump water heaters, but on either of the types of ASHP water heaters and not on both simultaneously. More elaborately, the performance of a heat pump water heater was simulated using the TRNSYS simulation software package [22]. However, it was noted that the TRNSYS simulation application could not effectively model the performance of an ASHP water heater as a result of the complexity of the metal fins encapsulating the evaporator. An analytic mathematical model was also presented to predict the COP of a solar assisted heat pump water heater in correlation to temperatures [13]. A quantitative method can be used to compute the COP of an ASHP water heater based on the quantity of electrical energy consumed by the ASHP system and the thermal energy gained by the stored water [20]. Precisely, Tangwe et al. (2013) [21] developed and built surface fitting regression models to predict the performance of a residential split type ASHP water heater under first-hour heating rating, standby losses and heating cycles due to hot water drawn off. Modelling of the residential air source heat pump (ASHP) water heater performance can provide an in-depth analysis of the dynamic behaviour of the coefficient of performance (COP). A mathematical model often employs the use of mathematical equation or computational algorithm to correlate predictors to desired response

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Mathematical modeling and simulation application to visualize the performance of retrofit heat pump water heater under first hour heating rating

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A B S T R A C T

Air source heat pump unlike water and geothermal source heat pumps are widely used in sanitary hot water production by virtue of the relative ease of harvesting, low grade aero-thermal energy and less complexity in operating and maintaining the heat pump unit. This research focused on performance monitoring of 1.2 kW air source heat pump, retrofitting a 200 L high pressure storage tank and operating under first hour heating rating for seven different climatic conditions. A data acquisition system was designed and built to measure the desired predictors of the power consumption and coefficient of performance of the ASHP water heater. A robust mathematical multiple linear regression models were built and were used in the modeled blocks in the simulation application developed in Simulink of MATLAB. Results indicated that the maximum coefficient of performance was associated with the maximum power consumption of the ASHP water heater. Finally, the simulation application could also be used by ASHP manufacturers and the energy saving company to quantify the energy reduced when geysers are retrofitted by ASHP. Optimization using constrained linear least squares solver in the optimization toolbox was also used to eliminate errors in the measurement from inclusion into the mathematical models.

1. Introduction

The heat pump technology was invented in 1950 and it basically operates on the principles of refrigeration. The two commonly utilized refrigeration technologies of heat pumps are the vapor compression refrigerant cycle and vapor absorption refrigerant cycle. The first heat pump water heater intended for mass production was designed and built by Calm in 1984 [1]. ASHP water heater is one of the heat pump types that operates on a vapor compression refrigerant cycle and has the capability of providing over two units of useful thermal energy just with one unit input of electrical energy when the system is in the heating up cycle [3,9]; Oak Ridge National Laboratory, 2009. A salient and better understanding of refrigeration cycle of heat pump water heater was given by Ashdown et al. (2004) and Sinha and Dysarkar, (2008). Heat pump water heaters also render an extra benefit of dehumidifying and space cooling because during operation, it pulls warm vapor from the air. The performance of heat pump can be significantly enhanced by the use of a refrigerant of very excellent thermo-physical properties. In Japan, there are already manufactured innovative heat pump that exploits carbon dioxide as the refrigerant fluid and is more than 300% energy efficient and this was made possible due to the government and private partnership rebates initiatives. The coefficient of performance of heat pump water heater can also be increased by using R11 (Chlorofluorocarbon compound) and R22 (Hydrochlorofluorocarbon compound) as the thermo-physical refrigerant in the heat pump unit [15]. More generally, geothermal heat pump has a better performance than air source heat pump water heater both as a single or coupled system and with an excellent payback-time. However, the capital cost of the design and construction of this system is huge and therefore limits its viability as compared to the air source heat pump unit in the field of sanitary hot water production. The following studies confirmed the high performances of ground or geothermal heat pump system over an air source heat pump system. These studies include; A techno-economic analytical comparison of the performance of air coupled and horizontal-ground coupled air conditioners conducted in South Africa [14]. A payback assessment of heating and cooling ground source heat pump system using carbon dioxide as the primary refrigerant carried out in a high energy

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Dynamic system modelling as a robust tool to evaluate the performance of domestic integrated and split type air source heat pump water heaters

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Abstract

Air source heat pump (ASHP) water heater is a renewable and energy efficient technology for sanitary hot water heating. The common categories of ASHP water heaters are the integrated and the split types. Modelling the system performance can provide an in depth of the dynamic behaviour of the coefficient of performance (COP) with the refrigerant temperatures at critical locations on the close loop circuit of the vapor compression refrigerant cycles and the ambient weather data as predictors. Primary data used in the building and development of the models were collected from a data acquisition system that was designed to store these data from the installed 150 l integrated and split type ASHP under three scenarios of controlled volume of hot water drawn off. The paper present both statistical simulation and robust mathematical models of COP of both integrated and split types ASHP water heaters using the refrigerant temperature difference of the compressor suction and discharge ends, the refrigerant temperature difference of the condenser inlet and outlet, the ambient temperature and relative humidity. The result revealed that the split type performs better than the integrated type ASHP water heater and there exists in the group COPs mean a significant difference between the two systems.

Keywords: Air source heat pump, Coefficient of performance, Vapor compression refrigerant cycles, Mathematical modelling, statistical simulation, Mean significant difference

1. Introduction

Modelling the residential air source heat pump (ASHP) water heater performance can provide an in depth analysis of the dynamic behaviour of the coefficient of performance (COP). A mathematical model often employed the use of mathematical equations or a computational algorithm to correlate predictors to desired response [20]. In this paper, mathematical models were developed and built that used the refrigerant temperatures at critical locations in the close loop circuit of the vapor compression refrigerant cycle and the ambient weather data as predictors. ASHP water heater is an efficient and a renewable energy device for sanitary hot water production [1]. The COP of an ASHP water heater can range from 2 to 4 and depends on the component design of the system, ambient weather conditions (ambient temperature, relative humidity, etc.), duct space and the speed of the cold and dehumidified expelling air [2;3]. The excellent efficiency for an ASHP water heater is due to its performance characteristics known as COP [4]. The ASHP water heater efficient COP can be achieved from the installation and the mathematical modelling perspective. The optimal COP of an ASHP water heater can be attained by an effective installation of the system [5]. But, it can be explained that the optimal COP of ASHP water heater could even be predicted from an accurate mathematical model under different system operating conditions. The system COP can also be enhanced by the use of a primary refrigerant of an excellent thermo-physical property [6;7]. A salient and better understanding of refrigeration cycle of heat pump water heater was presented by Ashdown et al., [8] and Sinha and Dysarkar [9]. It is crucial to justify that extensive research has been conducted on the simulation and mathematical modelling of the performance of heat pump water heaters, but on either of the types of ASHP water heater and not on both simultaneously. More elaborately, the performance of a heat pump water heater was simulated using the TRNSYS simulation software package [10]. However, it should be noted that the TRNSYS simulation cannot effectively model the performance of an ASHP water heater owing to the complexity of the metal fins encapsulating the evaporator. An analytical mathematical model was also presented to predict the COP of a solar assisted heat pump water heater in correlation to temperatures [11]. A quantitative method can be used to compute the COP of an ASHP water heater based on the quantity of electrical energy

Appendix V: Experimental installed hot water heating devices

A. Installed hot water heating technologies without isotherm blankets

B. Installed hot water heating technologies with isotherm blankets