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THERMODYNAMIC AND ELECTRICAL PERFORMANCE MONITORING OF A DOMESTIC SPLIT-TYPE AIR CONDITIONER AND DEVELOPMENT OF A SIMULATION BASED R22 PERMANENT REPLACEMENT.

By

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DECLARATION

I certify that this dissertation has not been submitted anywhere for any award of any sort before now.

DEDICATION

I dedicate this work firstly to the LORD GOD ALMIGHTY for wisdom, to my Mom, Mrs. Gumon Bantan Theresia, my elder brother, Mr. Bantan Yambuin Fritz and to my fiancé, Mr. Mbounda Yongkah Gilford for their financial, spiritual and moral support.

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GLOSSARY

AC	Air Conditioner
CFC	Chlorofluorocarbon
COP	Coefficient of Performance
GWP	Global Warming Potential
HC	Hydrocarbon
HCFC	Hydro-chlorofluorocarbon
HFC	Hydrofluorocarbon
HTC	Heat Transfer Coefficient
LHV	Latent Heat of Vaporisation
MM	Molar Mass
NEES	National Energy Efficiency Strategy
ODP	Ozone Depletion Potential
PFC	Perfluorocarbon
TEWI	Total Equivalent Warming Potential
VRC	Volumetric Refrigerating Capacity

NOMENCLATURE

COP_c	Cooling mode COP
COP_h	Heating mode COP
E_c	Cooling mode energy consumed
E_h	Heating mode energy consumed
E_m	Compressor energy consumed
h_g	Indoor heating mode enthalpy
h_l	Ondoor cooling mode enthalpy
h_o	Outdoor enthalpy
$\dot{m}$	Refrigerant Mass Flow rate
Q_{evp}	Evaporator Thermal Energy
Q_{con}	Condenser Thermal Energy
Q_{com}	Compressor Thermal Energy
T_a	Ambient temperature
T_{cp}	Outdoor unit periphery temperature
T_e	Evaporator inlet/outlet temperature difference
T_m	Compressor inlet/outlet temperature difference
T_n	Condenser inlet/outlet temperature difference
T_r	Room temperature

SUMMARY

The difficulty that exists in accurately monitoring the performance of air conditioners has made performance prediction an arduous task. Nevertheless, the performance still needs to be monitored and predicted as it helps solve a lot of problems resulting from this technology like effect of the technology on the grid, energy consumption, water utilisation and GHGs emission. With the introduction of regression modelling as a means of system monitoring and prediction some years ago, the accuracy was still a call for concern. It is worth realising that increasing the number of predictors will enhance this method's accuracy. As such, this document intends to increase the accuracy of this method's monitoring and predicting ability by increasing the number of predictors to cut across system thermal, environmental and human behavioural variation. These predictors experimentally gotten are used to build an environ-behavioural model that monitors the coefficient of performance and energy consumption of a domestic split-type air conditioner with higher accuracy.

Refrigerants have undergone evolution in the past decades in a bid to come up with a refrigerant that has zero ODP, lower – than – R22 GWP and much better than R22 thermodynamic performance. No pure refrigerant has been found to possess these qualities as such mixtures or blends are the best shot at the moment. R410A could stand the test of time to be the long term R22 replacement but for the fact that besides R410A's higher GWP than that of R22, the former's system performance is lower than that of the latter's due to the lower thermodynamic performance of the former. This means the search continues. In this document, a combination of carefully chosen refrigerant components are carefully blended to come up with a simulation based R22 long term replacement, which will be referred to in this document as BTEP.

1. CHAPTER ONE: INTRODUCTION

1.1. BACKGROUND AND RATIONALE

2015 was the last of the 10 years energy reduction life span anticipated in the National Energy Efficiency Strategy (NEES) with intended 12% general reduction in electricity demand in South Africa (15% reduction in industry and mining, 15% reduction in public and commercial buildings, 10% in residential buildings and 9% reduction in transport) and a vision of "reducing the energy intensity of the economy through energy efficiency" (Modise, 2014). In order for Eskom to achieve this reduction in demand and reduction in GHG emission, since 91.7% of the country's electricity is generated from coal (Eyetwa, et al., 2010), they embarked on Demand Side Management (DSM) programmes and later Integrated Demand Management (IDM) programmes to ensure energy efficiency, which is intended to contribute 34% towards the estimated 2015 12% energy demand reduction (Department, 2004). As such it is required of every device that uses electricity to utilise this energy efficiently.

Heating, Ventilation and Air Conditioning (HVAC) systems account for about 26% of energy consumption in the commercial sector, 20% in the industrial sector and 10 – 16% in the residential sector in South Africa (Eskom, 2015) (Eskom, 2013) (Tonde, et al., 2001). These systems are responsible for controlling the temperature, humidity and air quality in a defined space where they are installed to make the air healthy, conducive and comfortable at specified thermal and humidity limits (Eskom, 2015). Their aim is achieved by humidifying and pumping heat into or dehumidifying and cooling the space as need be.

One of the major types of HVAC systems used in South Africa is the air conditioning (AC) unit for both space heating and cooling with the split-type AC gaining more grounds and penetrating the South African market faster than any other type of AC (Covary, et al., 2015).

Human temperature and relative humidity comfortability range is respectively between 23 – 27^{o}C and 25 – 55% irrespective of the season in question (Bhatia & B, 2012). Most often, the outside temperature and humidity fall out of these ranges leading to discomfort, hence, an AC is needed to restore human comfort.

In 2002, about 358 million ACs were installed worldwide (Dariusz, et al., 2005). This number is predicted to double to about 700million by 2030 and 1.6billion by 2050 (Lawrance, 2016).In 2010, ~2.6 million electric ACs were installed in South Africa with each AC having an annual consumption of ~3.1MWh and total AC annual consumption of about 8.1TWh (Covary, et al., 2015). With about 73% potential growth in the purchase, installation and utilisation of AC especially the split-type AC in South Africa between 2010 and 2020, it depicts increase in energy consumption and increase in the national grid constrain. The energy consumption for these ACs can be reduced by 7% but this reduction and even more can only be effected if the current actual efficiency or performance of the system is known (Covary, et al., 2015). Knowledge of the current performance of the AC system will serve as a baseline. This idea will provide an inside on the type of AC efficient enough to replace the current technology as well as facilitate the quantification of energy savings that will be achieved when the replacement is eventually effected.

Though performance monitoring and prediction is very crucial for AC systems, there is no universally acceptable method of monitoring and predicting the performance of air conditioners due to the complexity surrounding the operation of the system. The complexity in the system operation is as a result of the human behavioural dynamics as well as environmental and system thermal variation. As such, this dissertation seeks to develop an easier and faster means to ascertain and predict the performance of an installed domestic split-type AC. This means is via a mathematical model. Mathematical modelling is all about representing complex real life situation in simple mathematical equations. It is a much appreciable idea to use it for monitoring and prediction of the performance of systems as complex as ACs. The characteristic of an AC system used for portraying the thermodynamic performance is known as the Coefficient of Performance (COP), while the electrical performance is characterised by the electrical energy consumption of the system. COP as a function of exergy, is the ratio of useful output thermal energy to the input electrical energy. Besides thermal energy, the thermodynamic and electrical performance of an AC system is affected by other measurable environmental and thermal or system behavioural factors, which include ambient temperature (T_a), relative humidity(RH), room temperature(T_r), outdoor unit periphery temperature (T_{cp}), refrigerant inlet and outlet temperature difference at the level of the compressor, condenser and evaporator. Hence, development of a robust multiple linear regression model that cuts across environmental, human and system behavioural properties correlating them with COP and energy consumption is the first driving factor of this dissertation.

The major heat transfer medium inside an AC is the refrigerant. Its performance affects the electrical performance of the AC as well as contributes directly and indirectly to some unhealthy environmental impacts. For the refrigerant to be able to afford optimal performance, it needs to possess some peculiar properties like low boiling and freezing points, eco-friendly: meaning not a greenhouse gas (GHG) or ozone depletion layer catalyst, able to favourably harbour a lubricant (mineral oil in case of CFCs). Before the invention of chlorofluorocarbons (CFC), an ideal refrigerant needed: a normal boiling point range of -40^o – 0^{o}C, to be non-toxic, non-flammable and chemically stable (Bolaji, et al., 2011). None of the available refrigerants before the 30s possessed any of these qualities until when CFCs were discovered to possess all these properties by Midgley and associates. CFCs when released into the atmosphere liberate chlorine, which reacts with the ozone layer oxygen thus, destroying the ozone layer thereby exposing earth occupants to UV rays. These rays are cancerous and a great contributor to global warming and greenhouse effect.

According to the Montreal Protocol, CFCs were phased out by January 1996 and hydro-chlorofluorocarbons (HCFCs) were the next chemicals to serve as refrigerants. R22 is one of the lowest Ozone Depletion Potential (ODP) chlorine-contained HCFC with a value of about 0.055 reasons why it was the most commonly used refrigerant especially in ACs (Bolaji, et al., 2015). Yet HCFCs is supposed to be completely phased out between 2020 and 2030 due to its environmental hazard. It became primordial to get a refrigerant with similar and better qualities to replace R22. Researchers realised that it is close to impossible to get a pure refrigerant that is ideal or that can strike an excellent balance between good thermo-physical properties and reduced or no negative environmental side effects much better than what R22 offers. This landed researchers with the idea of coming up with blends and mixtures that can possibly strike the balance needed. They ended up with a wide range of possible R22 alternatives including: R134a blends, hydrocarbons (HCs), hydrofluorocarbons (HFC) blends and R744 (Air-conditioning, 1992-1997). Yet all of these have a good number of limitations.

R410A, which is an HFC azeotropic mixture of R32 and R125 is the current acceptable refrigerant blend found in ACs especially split-type ACs in the South African market and worldwide. Though it is approved in most countries as a substitute for R22 systems because of its zero ODP value and insignificant recommended change in the AC component build up during substitution, it is remarked and proven to negatively influence the thermodynamic and electrical performance of AC systems as compared to AC systems habouring R22 (Devotta, et al., 2000), (Payne & Domanski, 2002) (Zaghdoudi, et al., 2010), (Bolaji & Huan, 2012), (Bukola, 2012), (Abhishek & Gupta, 2013), (Venkataiah & Venkata, 2014). This explains why, R410A is a temporal R22 replacement refrigerant. Hence, the search for a long term replacement continues.

For that reason, the second part of this dissertation seeks to embark on the journey of this search by commencing with the use of software to develop a new mixture or blend from some pure refrigerants in order to come up with a simulation based mixture that possesses thermo-physical and environmental properties better than that of R22 and R410A. This new blend, which will be referred to in this dissertation as BTEP is to serve as the best software based refrigerant that strikes a compromise between environmental and thermo-physical properties better than that of R22.

1.2. OVERVIEW OF HVAC TECHNOLOGY

Due to the dynamics that exist in demand for air quality control, there exist a wide variety of HVAC systems. HVAC systems can be classified into the following categories; District Network HVAC systems, Heaters, Hot Water Heating Systems, Warm Air Systems, Dual Duct Systems, Single Duct systems, Dehumidifiers, Free Cooling Systems and Air Conditioning Systems (Eskom, 2015). The above mentioned HVAC systems are just a handpicked few amongst others.

According to (Wang & Shan, 2000), AC systems can further be categorised into: Individual Room Air Conditioning System, Evaporative-Cooling Room Air Conditioning System, Desiccant-Based Air Conditioning System, Thermal Storage Air Conditioning System, Clean Room Air Conditioning System, Space Conditioning Air Conditioning System, Unitary Packaged Air Conditioning System, Central Hydronic Air Conditioning System.

Split-type AC (a type of individual room air conditioning system) was initially used mostly for residential purposes, which happens to be the second largest energy consuming sector in South Africa but is now gaining grounds in the commercial and public buildings sector and is dominating the South African market with prospects of increase in demand for this AC type in the imminent years (Eskom, 2015) (Covary, et al., 2015). With regards to this observation, it can be said beyond reasonable doubt that energy demand in the domestic sector will significantly rise since space conditioning accounts for about 10 - 16% of the total energy utilised in this sector. In order to ensure energy efficiency and calculate achievable energy savings accompanying this technology, it is paramount to monitor the performance of the system. For this reason, the Domestic Split-Type AC System will be the case study AC type for this write-up. Split-type AC is either wall or window mounted with a separate outdoor unit containing mainly the compressor, condenser and the fan and an indoor unit composed of the evaporator, expansion valve and the blower.

1.3. PROBLEM STATEMENT

Eskom being faced with the challenge of meeting the electrical energy need of the ever growing population, combating global warming caused by energy production from coal as well as efficient energy transmission has decided to embark on different strategies to remedy the situation. One of these strategies that is short live and produces significant result is DSM programme to ensure demand reduction and IDM that guarantees energy efficiency (Modise, 2014) (Department, 2004). HVAC systems especially AC systems are highly infiltrating the global market and South African community as well. They are contributing greatly to energy consumption especially in the second highest energy consuming sector, i.e. the domestic sector, with anticipated increase in purchase and utilisation of these systems in imminent years (Covary, et al., 2015). This invariably means that energy efficient techniques must be employed in this area to reduce energy misuse, wastage and grid constrain. To efficiently manage the energy used in this system as well as prevent further installation of energy inefficient technologies, the current technology's performance must firstly be monitored both thermodynamically and electrically to ascertain if the system is efficient or not as well as establish the system's baseline to facilitate energy savings and its accessories' computation. The complexity of the system operation due to environmental, human behavioural as well as the system's behavioural influence makes performance monitoring and prediction using solely environmental variation or system dynamics or qualitative or quantitative or deterministic modelling less accurate as has been previously done by other researchers. This explains why no universal performance monitoring and predicting method exists. Moreover, there already exist millions of installed split-type ACs nation and worldwide, with different manufacturers and different specifications contributing to the existing performance monitoring strain. As such, this study entails the development of a robust environ-behavioural multiple linear regression model for a domestic split-type AC system to facilitate performance monitoring and prediction. This model will combine the environmental, human and system thermal behavioural variations as predictors. The analysis will facilitate the monitoring and prediction of the thermodynamic (COP) and electrical performance (energy consumption) of a domestic split-type AC with an increased degree of accuracy. The environmental predictors include outdoor unit periphery temperature (T_{cp}), ambient temperature (T_a) and relative humidity (RH). Human behaviour and activities in a home can be depicted by the temperature in the room. This explains why the human thermal behavioural variable that directly affects the AC operation and hence serves as a model predictor is the room temperature (T_r). For the system's thermal behaviour, the predictors for the model will be the refrigerant temperature difference at the inlet and outlet of the evaporator (T_e), condenser (T_n) and compressor (T_m).

Without refrigerants in ACs, there will be no vapour compression refrigerant cycle (VCRC), hence, no air conditioning. This is to say that refrigerants have a vital role to play in an AC as their performance influences the thermodynamic and electrical performance of the system as well as contribute directly or indirectly to unhealthy environmental hazards. This explains why they have been a point of focus for researchers and manufacturers for many decades now to determine safe, environmental friendly, thermodynamically high performant and chemically stable refrigerants. This has been a daunting task owing to the fact that no refrigerant has been discovered to successfully possess all the characteristics of an ideal refrigerant. For instance natural refrigerants like NH_3, CO_2 and HCs, though provide favourable thermodynamic and environmental AC performance (zero ODP and low GWP), are highly flammable, toxic and have high pressures depending on the refrigerant in question. This has led to the phasing out and anticipated phasing out of refrigerants like CFCs and HCFCs (R22) by international protocols like the Montreal and the Kyoto protocols by 2040 and has driven researchers to source for blends and mixtures that will provide a good compromise refrigerant thermo-physical and environmental influence. It is known that R22 brings out the best AC thermodynamic and electrical performance and ensures that the AC is energy efficient but the reason for its phase out is its ozone depletion ability. R410A meant to replace R22 causes the AC to underperform reason why it is termed as a temporal R22 replacement refrigerant. As such, there is still a frantic need for an R22 permanent replacement with much better thermo-physical and environmental properties than R22 and R410A as this will improve on the thermodynamic and electrical performance of the AC as well. This dissertation intends to commence the realisation of this great refrigerant need by developing a simulation or software based new refrigerant blend referred to as BTEP from the refrigerant constituents R32, R152a, R134a and R152a. This new blend, BTEP will serve as a permanent replacement for R22 with better thermo-physical and environmental properties than that of the latter.

1.4. RESEARCH QUESTIONS

This research intends to answer the following questions:

- Is it possible to combine environmental, system and human thermal behavioural predictors in one model?
- Can this environ-behavioural model accurately monitor and predict the performance in terms of COP and energy consumption of a domestic split-type AC?

- What is the individual influence of each of predictors on the COP and energy consumed both in winter and summer?
- How much electrical energy is required to heat up or cool down air in the room from temperature different from the set heating or cooling temperature to the set temperatures?
- How much thermal energy is given off or gained during cooling or heating?
- What is the COP during heating and cooling?
- Is the COP of cooling greater or less than the COP of heating?
- What is the percentage impact of AC system on domestic consumption?
- Can a blend or mixture composing of R32, R152a, R134a and R143a be arrived at with an optimum balance between thermo-physical and environmental properties on REFPROP?
- How thermodynamically and environmentally compliant is BTEP as compared to R22 and R410A?

1.5. RESEARCH OBJECTIVES

The foremost aim of this research is to develop a robust environ-behavioural multiple linear regression model that will be used to conveniently monitor as well as predict the thermodynamic and electrical performance of a domestic split-type AC as a function of some environmental, human and system thermal factors.

The next aim is to develop software or simulation based long term or permanent R22 refrigerant blend or mixture replacement with much better thermo-physical and environmental properties than the latter for domestic split-type AC application.

The objectives of this research include:

- Verify the effect and weight of influence of each predictor on the COP and energy consumption.
- Model the heating and cooling COP and the energy consumption of the installed AC for both seasons.
- Determine the individual percentage consumption of the compressor, fan and blower.
- Determine software based permanent thermo-physical and environmentally more compliant R22 replacement.

1.6. HYPOTHESIS

Monitoring as well as predicting the performance of a domestic split-type AC can be accurately effectuated by building and utilising a robust environ-behavioural multiple linear regression model for the Coefficient of Performance and the energy consumed with a determination coefficient of about 90 – 94%. This is the case for predictors namely: ambient temperature, relative humidity, room temperature, outdoor unit periphery temperature and refrigerant temperature difference at the inlet and outlet of the evaporator, compressor and condenser whose respective impacts are evident in the model with a correlation coefficient of about 95 – 97%.

The first and most important step in achieving an excellent refrigerant that can strike the balance between thermodynamic as well as environmental optimum performance can be obtained by carefully analysing and mixing R134a, R152a, R143a and R32 on the NIST software. This excellent software based refrigerant thermo-physically and environmentally outperforms R22 and R410A by more than 50%.

1.7. DELINEATION AND LIMITATIONS

Though AC systems have so much yet to be discovered and investigated and refrigerants have so many unanswered questions and unsettled minds, the research focuses mainly on the building of a robust environ-behavioural multiple linear regression model to predict the COP (heating and cooling) and energy consumption of a domestic split-type AC using ambient temperature, relative humidity, room temperature, outdoor unit periphery temperature and refrigerant temperature difference at the inlet and outlet of the evaporator, compressor and condenser as predictors. The set heating temperature for the experiment is 27^oC and the set cooling temperature is 25^oC. Secondly, it concerns itself with finding software based long term best fit replacement for R22 and R410A on REFPROP simulation application by blending R32, R152a, R134a and R143a based on the following properties: LHV, ρ_{vap}, k_{liq}, μ_{liq}, C_p, MM, T_{crit}, GWP.

1.8. ASSUMPTIONS

For this research, the following major assumptions are made:

- It is assumed that the volume of air in the room is constant and equal to the product of the length, width and height of the room.
- Since the pipe containing the refrigerants cannot be bored so as to insert the temperature sensors, the temperature sensors will be externally attached to the pipes harbouring the refrigerants with the assumption that the pipes and the refrigerant are at thermal equilibrium at the various measuring points.
- Thermal energy lost by the refrigerant in the indoor heat exchanger during space heating is equal to the thermal energy gained by the space being conditioned and vice versa during cooling mode.
- Heat exchanger temperature is equal to the average of the inlet and outlet temperature.

1.9. CHAPTER OVERVIEW

Chapter two is a review on some basic principles pertaining to major parts of the dissertation. It elaborates on the technology underlying AC systems together with the functional principle of split-type ACs. The choice of model is also deliberated upon. It throws light on the evolution of refrigerants, the different classes, their strengths and limitations and enunciates on the various aspects of thermodynamics that will be used to analyse and blend the various components of BTEP.

In chapter three, the methodology involved in setting up the data acquisition system (DAS) is elaborately enunciated. Each of the instruments that make up the DAS are described, their functional principle, their use, reason for the choices made and a broad description of the experimental setup. The software based refrigerant constituents blending process is outlined.

Chapter four reveals the results from the DAS and dissevers the four obtained models. It also gives explanations and justifications for the obtained results linking the results to the literature behind the research and elaborates on the effect of each predictor on the responses.

Chapter five exhibits and dissects the results obtained from the REFPROP analysis. It explains the thermodynamic and environmental behaviour of each of BTEP and its components. It also points out the outstanding performance of this new blend over R22 and R410A.

This dissertation ends with conclusion and recommendations. This section summarises all the findings of the research laying emphasis on the contributions made by this research.

1.10. SUMMARY

- Eskom anticipated a 12% reduction in demand throughout the country by 2015 in order to drop the percentage of energy generated especially from coal and hence reduce GHG emission.
- DSM and IDM programs are meant to contribute 34% towards the achievement of this 2015 goal.

- 26%, 20% and 10 – 16% of energy supplied to the commercial, industrial and residential sectors respectively is used by HVAC systems.
- HVAC systems especially ACs are used to condition air in a particular space by regulating the temperature and humidity ensuring human comfort.
- Human thermal and humidity comfort range is between 23°C – 27°C and 25% – 55% respectively.
- Between 2010 and 2020, ACs will experience a 73% increase in installation in South Africa from the ~2.6million already installed by 2010 hence increase in energy consumption.
- To go energy efficient in this technology, the COP and energy consumption of the system has to be monitored using a highly accurate means.
- This study intends to build a robust environ-behavioural multiple linear regression model with a determination and correlation coefficient in the ranges of 90 – 97%.
- Since R410A is just a temporal replacement for R22 due to its low performance, there is still a search for a long term replacement.
- This study will develop a simulation based long term replacement for R22 called BTEP with much better thermodynamic properties than R22.

2. CHAPTER TWO: LITERATURE REVIEW

2.1. INTRODUCTION

This chapter is a review on some basic principles pertaining to major parts of the dissertation. It elaborates on the technology underlying AC systems together with the functional principle of split-type air conditioners. It also throws light on the evolution of refrigerants, the different classes, their strengths, limitations and the various criteria used in refrigerant selection. It enunciates the various aspects of thermodynamics that will be used to analyse and blend the various components of BTEP.

2.2. OVERVIEW OF SPLIT-TYPE AIR CONDITIONING SYSTEM

In all the sectors in South Africa, from 1999 – 2013, split-type air conditioning systems have increasing gained grounds to the tune of 18% and is anticipating further increase in purchase in the coming years amongst. Its percentage and rate of increase is faster than the other existing air conditioning systems. Figure 2.1 is a basic block diagram of a split-type air conditioner.

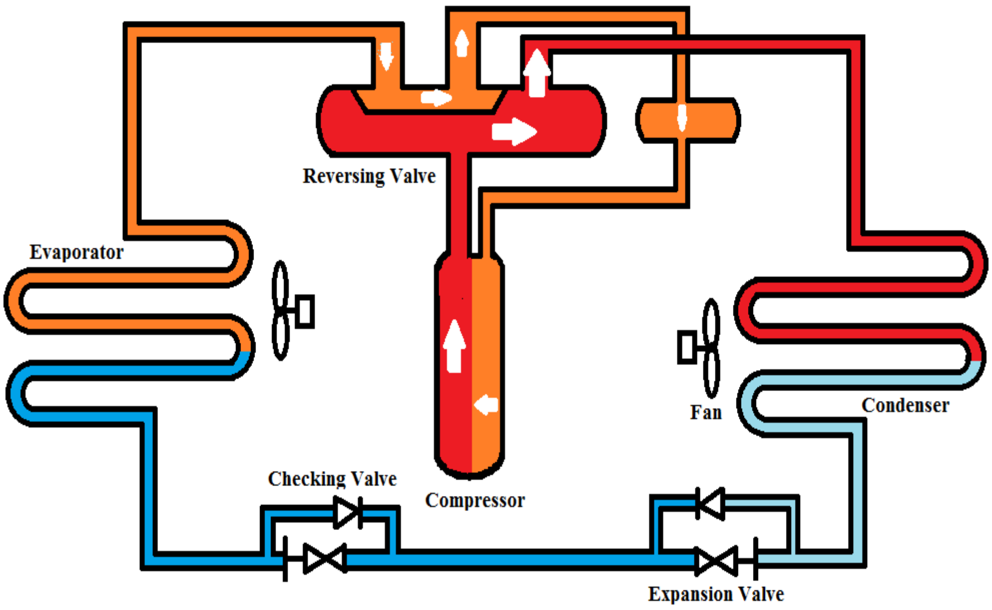


Figure 2.1: Basic Block diagram of a refrigeration cycle for an AC

This air conditioner has the following parts:

- Refrigerant: This is the major heat transport medium in an AC and is made from chemicals like CFC, HCFC, HFC, HFO and natural refrigerants (CO_2, NH_3 and HC). According to (Vandaarhuzhali, 2014), (Wang & Shan, 2000) (Environmental, n.d.) CFCs and HCFCs are highly responsible for Ozone layer depletion, Global warming and greenhouse effect as such Montreal and Kyoto protocols advocated for the phasing out of CFCs and HCFCs. R22 most especially must be kicked out of the refrigerant application by 2040 in South Africa. HFCs and blends of HFCs together with natural refrigerants are the focus for new AC systems being designed to replace R22.
- Compressor: This device, which is the heart of the AC system and consumes majority of the electrical energy supplied to the AC aids in the circulation of refrigerant in the system. Based on body structure, there exist three different types of compressors: Hermetic Compressors, Semi-hermetic Compressors and Open Type Compressors
- Condenser: This is a heat exchanger, which is found in the indoor unit during AC heating mode and in the outdoor unit during AC cooling mode. In this device, the refrigerant releases thermal energy and is cooled, its temperature and pressure reduced and the refrigerant partially converted into a liquid. There exist four types of condensers; air cooled, water cooled, evaporative (with both air and water as coolants) and static condenser.
- Sight Glass: Shows whether or not there is sufficient refrigerant just before the evaporator. A clear glass indicates that there is sub-cooled refrigerant but bubbles say there is refrigerant shortage, thus, need for recharge.
- Fan: It blows the cold air towards the condenser coil to enable the heat exchange that decreases the temperature of the refrigerant.

- Reversing Valve: This is a 4-way valve connected to the compressor and is responsible for effecting the change in operation mode of the device from heating to cooling mode.
- Check Valve: This is a one way valve that is used to bypass the expansion valve.
- Receiver: It is a metal tank connected on the high pressure line of the AC system, between the condenser and the expansion valve. It stores excess liquid refrigerant and oil that was not used during previous cycles in order to use it during low demand, contains a filter that traps dirt particles and has a desiccant that absorbs moisture. It also holds liquid refrigerant when the system is undergoing maintenance.
- Accumulator: This is a metal tank similar to the receiver but is found in AC systems without an expansion valve. It is connected on the low pressure line of the system between the evaporator and the compressor. It is used to trap liquid refrigerant that did not evaporate in the evaporator and warm the refrigerant for it to become vapour. It contains a desiccant to trap moisture and a filter to rid the refrigerant of any dirt.
- Expansion Valve: This is a throttling device also known as the metering device used to regulate the amount of refrigerant entering the evaporator by matching the flow rate with how quickly the refrigerant evaporates in the evaporator. It further decreases the temperature and pressure of the refrigerant.
- Evaporator: It is a heat exchanger and just like the condenser, it can be found either in the indoor or outdoor unit depending on the mode of operation of the device. Unlike the condenser the refrigerant extracts sensible as well as latent heat of vaporisation from the air and becomes a saturated vapour thereby cooling the air in question. This air could be the air of the room to be conditioned in the case of cooling mode or the outdoor air in the case of heating mode.

- Blower: The blower supplies hot air to the evaporator so that this air can be cooled by the evaporator. According to (UNEP, 2006), blowers have a higher discharge to suction pressure ratio than fans and the former can have pressures as high as $1.20 kg/cm^2$.
- Drain Pipe: This pipe serves as the exit route for water that originates from dehumidification of hot air by the evaporator.
- Louvres or fins: They enable the cool air supplied by the blower to get into the room.
- Filters: It is located just before the evaporator coil and it ensures clean air is supplied to the room.

2.3. FUNCTIONAL PRINCIPLE OF SPLIT-TYPE AIR CONDITIONING UNIT

Split-type air conditioners are so called because they contain two units; the indoor unit (composed of mainly the evaporator, expansion valve and blower), which is installed in the space to be conditioned and the outdoor unit (made up of mainly the compressor, condenser and fan), which is installed outdoor as shown on the block diagram in figure 2.4.

The split-type AC operation is referred to as the Vapour Compression Refrigeration Cycle (VCRC), which is a practical application of the first and second laws of thermodynamics. In this cycle, thermal energy is neither created nor destroyed by the AC system and there is spontaneous flow of heat from hot to cold sections. This AC has the ability to act both as a heater and cooler by simply reversing the functions of the evaporator and condenser depending on which mode is activated.

2.3.1. AC Valve Operation

For the AC to be able to perform both heating and cooling there is an additional valve connected to the compressor as shown in figure 2.1. This 4-way electromechanical valve is referred to as the reversing valve and is shown in the figure 2.2 for both heating and cooling cycle of the AC.

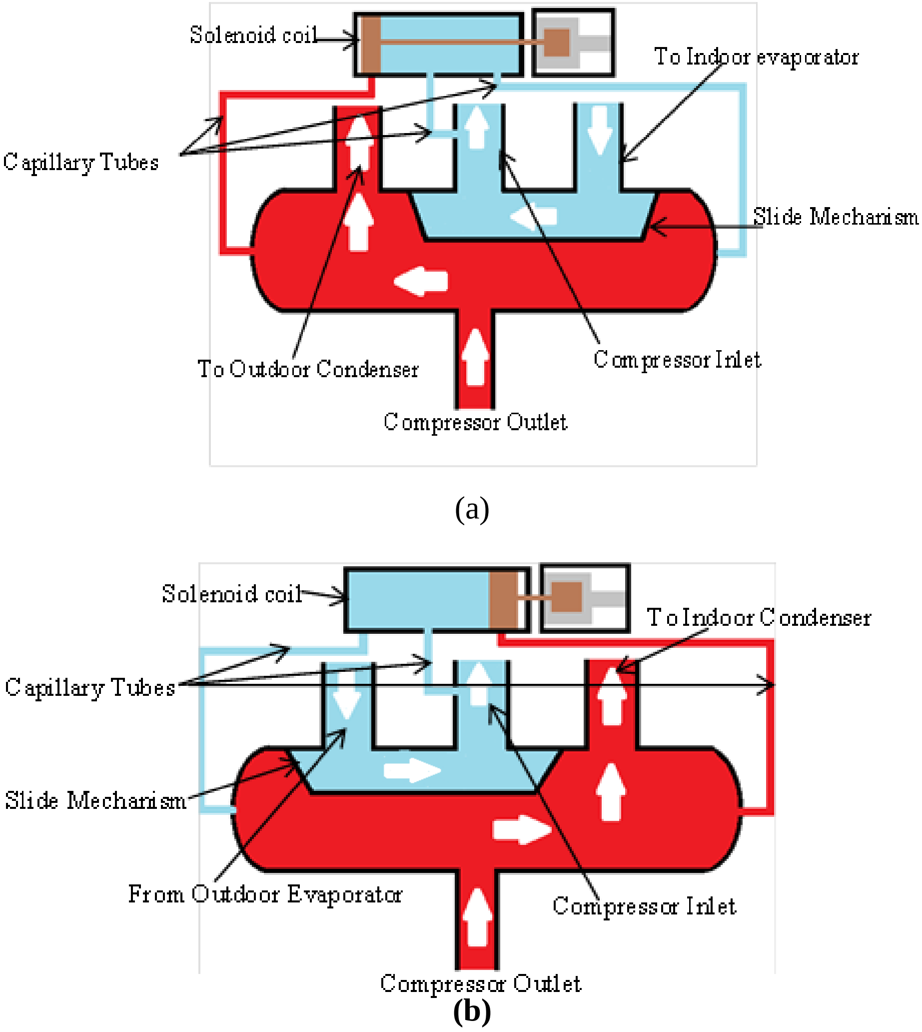


Figure 2.2: Refrigerant flow in the reversing valve for cooling and heating mode

During heating mode, the solenoid valve is de-energised and the solenoid coil shifts back as shown in figure 2.2a. As such, a high pressure from the coil head is built in the capillary tube connected to the indoor heat exchanger side of the 4 way valve. The pressure difference, which exists between the two capillary tubes, pushes the slide mechanism in a direction so as to cover the compressor inlet port and the outdoor heat exchanger port. This permits high temperature high pressure vapour refrigerant exiting the compressor to pass through the port leading to the indoor heat exchanger, which in this mode is acting as the condenser, while the low temperature low pressure vapour refrigerant from the outdoor heat exchanger, which acts as the evaporator, is sent to the compressor.

During cooling mode, the solenoid valve is energised and the solenoid coil moves forward. High pressure from the heat of the coil is built in the capillary tube located at the outdoor heat exchanger end of the reversing valve and the pressure difference between the capillary tubes move the slide mechanism to close the compressor inlet port and the indoor heat exchanger port. High temperature high pressure vapour refrigerant then moves from the compressor outlet port to the outdoor heat exchanger, which operates as the condenser while low temperature low pressure vapour refrigerant then moves from the indoor heat exchanger, which operates as the evaporator, to the compressor.

There exist two expansion valves for throttling process during the two AC operation modes; one after the outdoor heat exchanger and one before the indoor heat exchanger as shown in figure 2.1. Each of these expansion valves have a one way check valve connected in parallel for bypass purposes.

2.3.2. AC Operation

In summer when the AC is off, there is refrigerant already present in the liquid receiver. Once the system is activated on cooling mode, the solenoid coil of the reversing valve is energised and the valve takes the configuration shown on figure 2.2b. The outdoor heat exchanger, which functions as the condenser is ready for motion and the expansion valve in the indoor unit awaits appropriate signal from the thermostat. Once the thermostat senses an increase beyond the set temperature, it sends signals to the indoor expansion valve, which opens in preparation for the throttling process of the refrigerant while the outdoor expansion valve is bypassed by its corresponding checking valve. The compressor (also known as the heart of the refrigeration cycle or the vapour pump) is connected to the 4-way valve as shown in figure 2.1 where due to the pressure difference between the suction and the discharge lines the refrigerant is compressed through an adiabatic and isentropic process and its temperature and pressure increased making it suitable for the condenser. The thermodynamic equation for the isentropic process in the compressor is shown in equation 2.1 based on the pressure – enthalpy graph of figure 2.3

$$\Delta \boldsymbol{Q}_{\boldsymbol{com}} = \dot{\boldsymbol{m}}(\boldsymbol{h}_{\boldsymbol{2}} - \boldsymbol{h}_{\boldsymbol{1}}) \qquad \textbf{\textit{Eq. 2.1}}$$

The superheated refrigerant vapour flows into the condenser and loses thermal energy (both sensible and latent heat) in an isobaric process to the outdoor unit surrounding air. The saturated liquid refrigerant has its temperature and pressure abruptly dropped through a throttling process leading to a flash evaporation of some part of the refrigerant in the indoor expansion valve. This expansion valve also regulates the amount of refrigerant (partly liquid and vapour) entering into the evaporator. All these activities in the expansion valve occur via an isenthalpic and adiabatic process. The thermodynamic equations in the condenser and expansion valve are shown in equation 2.2 and 2.3.

$$\Delta \boldsymbol{Q}_{\boldsymbol{con}} = \dot{\boldsymbol{m}}(\boldsymbol{h}_{\boldsymbol{4}} - \boldsymbol{h}_{\boldsymbol{2}}) \qquad \textbf{\textit{Eq. 2.2}}$$

$$\boldsymbol{h}_{\boldsymbol{5}} \doteq \boldsymbol{h}_{\boldsymbol{4}} \qquad \textbf{\textit{Eq. 2.3}}$$

The low pressure low temperature saturated refrigerant (coexisting liquid and vapour) then gets into the evaporator where it extracts thermal energy (latent and sensible heat) from the room air blown over the evaporator coils by an isobaric process. This thermal energy gained causes the refrigerant to boil, evaporate into a saturated vapour and then into a low pressure low temperature superheated vapour suitable for compression and the process continues as shown in equation 2.4.

$$\Delta \boldsymbol{Q}_{\boldsymbol{evp}} = \dot{\boldsymbol{m}}(\boldsymbol{h}_{\boldsymbol{1}} - \boldsymbol{h}_{\boldsymbol{5}}) \qquad \textbf{\textit{Eq. 2.4}}$$

This is what occurs during summer and the pressure – enthalpy graph is shown in figure 2.3.

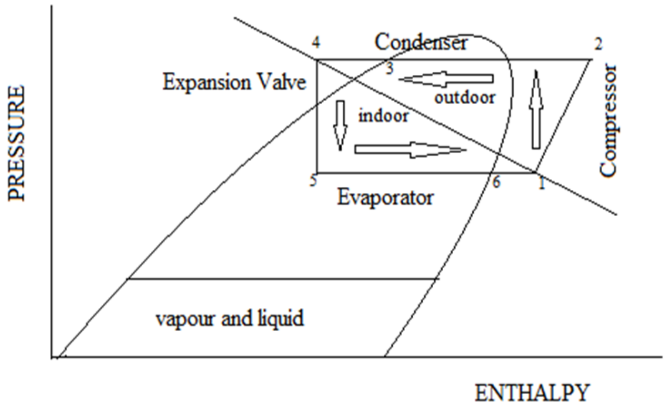


Figure 2.3: Pressure - Enthalpy graph for the refrigeration cycle **(Wang & Shan, 2000)**

During winter, the indoor expansion valve is bypassed by the check valve. The outdoor unit acts as the evaporator thus drawing heat from the outside air while the indoor unit acts as the condenser releasing the heat to the room air. The compressor is connected to the reversing valve as shown in figure 2.2a. During the winter season for the condenser to operate as the evaporator, its temperature has to be lower than that of the outside environment. This leads to ice formation on the outside unit coil walls. To eliminate this ice formation, the AC embarks on a defrost cycle wherein it takes heat from the inside room and sends it outside. During that period, the blower and the fan are turned off until the defrost cycle is over.

It is worth noting that for an ideal Carnot cycle, the processes taking place at the evaporator and condenser are supposed to be both isothermal and isobaric. This means that only latent heat of vaporisation is supposed to be gained and lost at these heat exchangers but this is not the case in a pragmatic system operation. Some sensible heat is gained and lost, hence, making the process strictly isobaric.

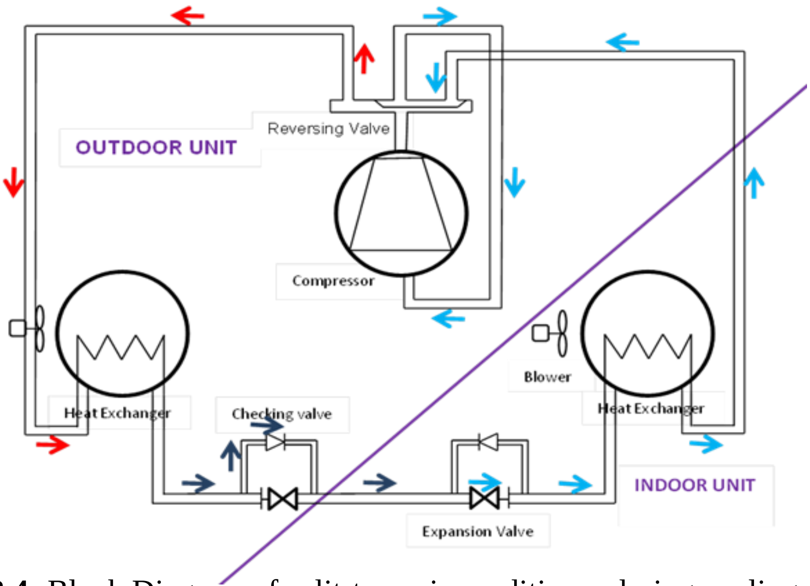


Figure 2.4: Block Diagram of split-type air conditioner during cooling mode

2.4. AC THEORETICAL PERFORMANCE REVIEW

The performance of this system during heating and cooling mode is characterised by the COP (COP_h and COP_c) and energy consumption. For the performance of an AC system to be optimal, it is required that the system's energy consumption be low while the COP be high. From an energy point of view and assuming that any energy gained by the space is equivalent to the thermal energy lost by the refrigerant and vice versa.

$$COP_h = \frac{Q_g}{E_m} \qquad \textit{Eq. 2.5}$$

$$COP_c = \frac{Q_l}{E_m} \qquad \textit{Eq. 2.6}$$

During heating, the thermal energy gained in the space to be heated (Q_g) is gotten from the thermal energy drawn from the outdoor air (Q_o) and electrical work done (E_m) by the compressor as illustrated by equation 2.7. Correspondingly, the thermal energy lost from the indoor (Q_l) during the cooling mode is the difference between the thermal energy gained by outdoor air (Q_o) and the compressor work done (E_m) as mathematically represented by equation 2.8.

$$Q_g = E_m + Q_o \qquad \textit{Eq. 2.7}$$

$$Q_l = Q_o - E_m \qquad \textit{Eq. 2.8}$$

Substituting equation 2.7 and 2.8 in 2.5 and 2.6 gives equation 2.9 and 2.10.

$$COP_h = \frac{Q_g}{(Q_g - Q_o)} \qquad \textit{Eq. 2.9}$$

$$COP_c = \frac{Q_l}{(Q_o - Q_l)} \qquad \textit{Eq. 2.10}$$

In terms of refrigerant mass flow rate (ṁ) and enthalpy,

$$COP_h = \frac{\dot{m}h_g}{(\dot{m}h_g - \dot{m}h_o)} \qquad \textit{Eq. 2.11}$$

$$COP_c = \frac{\dot{m}h_l}{(\dot{m}h_o - \dot{m}h_l)} \qquad \textit{Eq. 2.12}$$

Since the refrigerant mass flow rate of the refrigerant is constant,

$$COP_h = \frac{h_g}{(h_g - h_o)} \qquad \textit{Eq. 2. 13}$$

$$COP_c = \frac{h_l}{(h_o - h_l)} \qquad \textit{Eq. 2. 14}$$

2.5. REVIEW OF EXISTING WORK ON AC PERFORMANCE MONITORING

It has been an arduous task to conveniently and accurately monitor the performance of an AC due to the numerous environmental and human behavioural factors influencing the system operation. This explains why there is yet no universally acceptable method for AC performance monitoring. So far some work has been done in this light which includes: "Studying thermal performance of split-type air conditioners at building re-entrant via computer simulation" (Chow, et al., 1999). In this research performance of an air conditioning unit was monitored via monitoring the performance of the condenser during heat dissipation. This was done using simulation in the computational dynamic fluid analysis. The researchers used manufacturer's data to build a simple linear regression model that correlates COP and condenser on-coil temperature at constant room temperature. In 2007, the effects of air volume on refrigerating capacity, input power, COP, outlet temperature, discharge pressure, discharge temperature and suction pressure for ducted, digital scroll and conventional scroll ducted AC units was monitored and the results compared (Hu, et al., 2007). “Development of the performance evaluation method for a split air conditioning system using the compressor characteristic curve” whereby the refrigerant mass flow rate was obtained using a regression model and the compressor curve method. The two results were compared (Wakahara, et al., 2010). “Performance Evaluation of an AC according to different test standards” in which the cooling capacity and the power consumption of a split-type air conditioner installed in a psychrometric room (Air conditioner test room) set up based on ASHRAE standards, was monitored through calculations using data obtained from installed sensors (Ravi, et al., 2013). In 2011, the performance of a multi-split or variable refrigerant flow (VRF) AC with a digital scroll compressor was studied with the aid of a self-made monitoring software. The performance monitored here included the power consumption as well as the capacity of these AC types in relation to input parameters like ambient temperature, air flow, suction pressure, discharge pressure, etc. (Qiu, et al., 2011). In 2010, a room experiment was carried out to monitor the effect of ambient temperature and relative humidity on the cooling COP and energy consumption of a room split-type AC system with the aid of a regression model (Izham & Mahlia, 2010).

2.6. REFRIGERANTS EVOLUTION

Refrigerants have undergone so much evolution in the past decades due difficulties in obtaining a pure refrigerant that strikes a balance between thermodynamic, electrical and environmental benefits. Figure 2.5 is a block diagram that summarises the evolution of refrigerants so far.

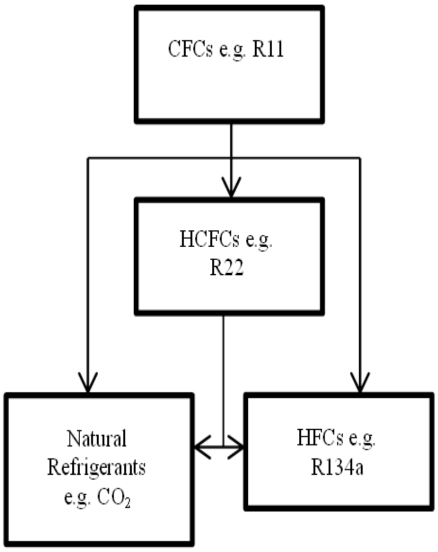


Figure 2.5: Refrigerants Evolution **(Bantan, et al., 2016)**

Looking down memory lane, there has been so much evolution in the history of refrigerants from "first generation" refrigerants (like SO_2 and NH_3) in the 1830s, which were also called "whatever worked", through "second generation" chlorofluorocarbons (CFCs) refrigerants in the 1930s as shown in figure 2.5 (ASHRAE, 2012). Due to its high Ozone Depletion Potential (ODP), Montreal Protocol of September 1987 adopted that CFCs be kicked out of the refrigerant business and hydro-chlorofluorocarbons (HCFCs) with lower ODP act as interim Refrigerants (ASHRAE, 2001). Though they too are at the verge of being phased out since they still destroy the ozone to a certain level and possess a high Global Warming Potential (GWP), HCFC – 22 commonly known as R22 is the present day most used refrigerant in ACs due to the fact that it is one of the HCFCs with the lowest ODP of about 0.055. Nevertheless, it still destroys the ozone layer though at a slower rate than others and according to the Montreal Protocol must be wiped out by 2040. This gave room for urgent need to find a best fit for R22, which could either be a mixture or a blend since it is close to impossible to get a pure refrigerant that strikes a balance between thermodynamic, electrical and environmental AC performance. Worldwide as well as in South Africa, R410A among the available HFCs is the chosen interim solution for ACs though research carried out indicates that COP, compressor power, Pressure ratio amidst other characteristics are poorer in systems running with R410A than those running with R22, (Bantan, et al., 2016). One of the most outstanding HCs is R600, which was proven to have a higher COP than R22, yet it is flammable which rolls it out as a pure refrigerant replacement for R22 (Zaghdoudi, et al., 2010) (Prapainop & Suen, 2012). For shortlisted natural refrigerants like ammonia (R717) though it has higher COP and refrigerating effect than R22, its pressure ratio, toxicity and flammability makes it unsuitable for small systems like a domestic split-type AC system (Dalkiliç & Selim, 2010) (Boumaza, 2015) (Stanciu, et al., 2011) (Stanciu, et al., 2011). For carbon dioxide (R744), it was discovered that its systems have a lower COP than that of R22 and its compressor consumes more power than R22, hence it cannot solely act as an R22 replacement (Douglas, et al., 1997) (Brown, et al., 2002). Due to the difficulty in getting a pure refrigerant that will possess much better thermodynamic, environmental and electrical properties than R22, it was resorted to try mixtures or blends since it is possible to arrive at a compromise by blending refrigerants that excel in different properties to get a single blend that out performs R22.This gave rise to the worldwide acceptance of R410a in domestic split-type ACs. R410A is replacing R22 for a short while. This is as a result of its poor thermodynamic and electrical AC output performance, which accounts for increase in the system's energy consumption. Hence, there is a need for further research to be carried out to get a blend or mixture that will ensure optimal performance of AC systems and be environmental friendly as well, which is the second aim of this study.

2.7. SIGNIFICANCE OF STUDY

The complexity in system operation as well as difficulty in obtaining a universally acceptable method of performance monitoring is simultaneously caused by variation in three major influencers. These are human, system thermal behavioural and environmental variation. Depending solely on one of these influencing factors to monitor the performance just like the researchers enumerated in section 2.5 and many others, cannot adeptly give a true reflection of what the system performance is. Intrinsically, parameters from the three major system performance influencers must be taken into consideration in the process of system performance monitoring and prediction. With modelling, it is easy to correlate predictors to response irrespective of the degree of system complexity. This explains why in this study, an environ-behavioural multiple linear regression model is built with predictors spanning across the three major system performance influencers. This will result system in a long list of predictors but will also ensure performance monitoring and prediction of higher accuracy.

The movement from CFCs through HCFCs to HFCs is as a result of the desperate need to obtain a refrigerant with no damaging environmental effect coupled with outstanding thermodynamic properties. Due to the inability to get a pure refrigerant that complies perfectly to the above demands, researchers resulted to blends and mixtures to replace the best performing R22. Though resorting to R410A has environmental benefits due to its 0ODP and little or no system structural change needed upon refilling R22 systems with R410A, the drop in systems' thermodynamic performance due to the latter refrigerant cannot be under looked as pointed out by a good number of researchers in section 2.6. As such the search for a better mixture continues. The second part of this study will develop a new simulation based long term replacement for R22 and R410A called BTEP that strikes the required balance between environmental friendliness and thermodynamic excellence.

2.8. MODELLING AND REASON FOR MODEL CHOICE

Models describe how a system in real life behaves while mathematical model further quantitatively explains the behaviour of that real life system using mathematical equations or computer code to correlate the system's parameters, variables and responses. Mathematical models are advantageous owing to the precise and concise nature of the mathematical language. It is beneficial in the development of scientific understanding of the real life system, system changes follow up and proper decision making.

According to (Marion & Lawson, 2008), mathematical models can be classified based on the type of outcome they are to predict and hierarchical organisation of the system to be modelled. Based on the outcome to be predicted, we have: Deterministic and Stochastic Models while classification based on the hierarchical organisation of the system to be modelled is as follows: Mechanistic and Empirical Models.

The validity of a model is ascertained based on its usefulness. This explains why there is no perfect model for a particular real life phenomenon and each real life phenomenon can be described by two or more models depending on the angle from which the model's usefulness is perceived. Regression models are one of the simplest forms of mathematically representing real life scenario. This was used by Izham et al. though they had just two predictors (Izham & Mahlia, 2010). Increasing the number of predictors increases model accuracy and model performance reason why in this thesis the predictors cut across environment, system thermal variation and human behavioural variation to four build environ-behavioural models. This explains why a multiple linear regression model brings about further improvement on the prediction of thermodynamic and electrical performance of a domestic split-type AC system.

2.9. REFRIGERANT THERMODYNAMIC AND ENVIRONMENTAL PROPERTIES

For a refrigerant to replace another, the thermodynamic as well as environmental performance of the AC system harbouring this replacement refrigerant has to be better than the system harbouring the old refrigerant. In this dissertation, the focus is a long term replacement for R22. There are a number of properties that define AC system performance. These properties are influenced by the refrigerant type and are both thermodynamic and environmental as shown in the tree diagram in figure 2.6. These are the properties that serve as the measuring rod for the choice of an ideal refrigerant.

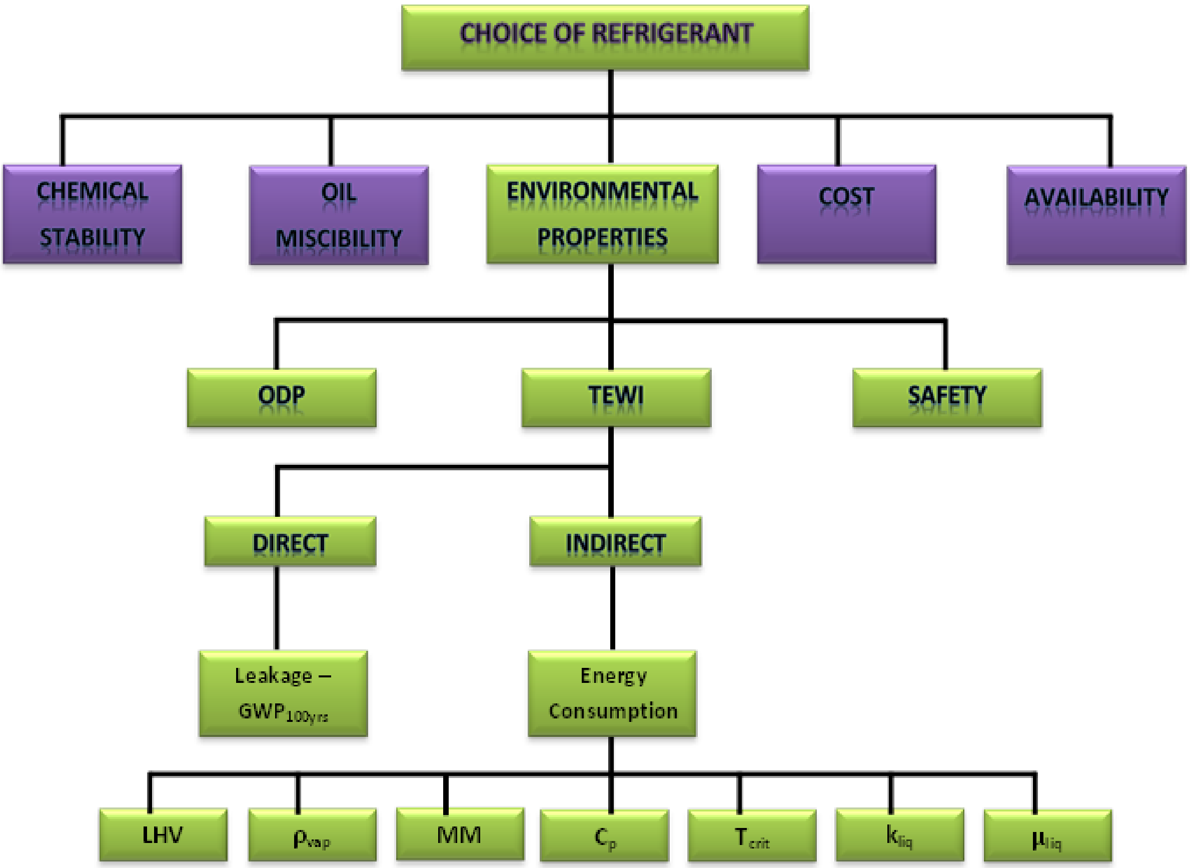


Figure 2.6: Refrigerant selection criteria

How harmful a refrigerant is lies in its impact on human lives, hence, it has to do with the environment. This explains why for a refrigerant to be considered as a candidate for R22 replacement, its environmental impacts have to be evaluated first. It is in this light that this research seeks to develop a refrigerant blend that is environmentally friendly; hence, the blend should be satisfactory with respect to the green boxes in figure 2.6.

First and foremost, for any blend to be considered in any refrigeration system, it must have 0ODP. Secondly, in terms of safety, it should be non-flammable and nontoxic. Thirdly, its TEWI should be lower than the refrigerant it is to replace. Direct contribution to TEWI contributes to GWP and originates from refrigerant leaks. Its contribution to TEWI is just about 5% since for split-type ACs, annual refrigerant leakage is about 4 – 5% of the original factory charge (UNEP, 2003). It has been realised that more than 90% of global warming effect from refrigerating systems originates from indirect influence via the energy used to run the system

(Lommers, et al., 2003). How much energy the system will consume or will be generated from the power utility company to feed this system depends on the thermodynamic performance of the system, which in turn depends on the type of refrigerant. As such, it is recommended that the system's latent heat of vaporisation, critical temperature, vapour density, specific heat, liquid thermal conductivity, liquid viscosity as well as the refrigerant's molecular weight and GWP be much better than that of R22 (ASHRAE, 2001) (Prapainop & Suen, 2012). Afore mentioned thermodynamic properties have an influence on the COP and VRC, hence they influence the amount of electrical energy that is being consumed and generated. For a country like South Africa with primary energy source being the burning of coal, these thermodynamic properties influence how much CO_2 is release into the atmosphere and hence contributing to GWP and indirectly contributing to TEWI. The subsequent subsections clearly relate these properties to the performance of the AC.

2.9.1. Latent Heat of Vaporisation

LHV with unit being kJ/kg is the amount of heat required to convert a unit mass of liquid into vapour. For an ideal refrigerant, an increase in the LHV is advantageous. This is because high LHV means a low mass flow rate per cooling capacity. More thermal energy will be carried from the outdoor air by a small amount of refrigerant to the indoor air during heating and more thermal energy will be taken away from the indoor air during cooling by a small amount of refrigerant. Hence, increase in volumetric refrigerating effect with a consequently small compressor size and increase in COP (Prapainop & Suen, 2012). As such in a bid to replace R22 and/or R410A, BTEP is expected to have a higher latent heat of vaporisation than that of R22 and R410A as this will promote decrease in power consumption, decrease in compressor displacement resulting into a small and compact unit as observed by B Boladji et al., 2014.

2.9.2. Liquid Thermal Conductivity

High liquid thermal conductivity means high HTC. High HTC contributes to a high performant refrigerant as such high COP. Therefore for a refrigerant to confidently replace R22 and/or R410A, it has to have a liquid thermal conductivity higher than that of R410A and R22. According to (Hogberg & Vamling, 1996), the vapour thermal conductivity has no significant influence on the COP of the system whereas 10% increase in the liquid thermal conductivity amounts to 0.5 – 0.6% increase in COP.

2.9.3. Liquid Viscosity

Increase in the liquid viscosity leads to increase pressure drop at the level of the heat exchangers (evaporator and condenser). it also increases pumping power and reduces heat transfer. Hence increase in liquid viscosity decreases both performance and system capacity. According to Hogberg and Vamling 15% increase in liquid viscosity leads to 0.4 – 0.5% decrease in COP (Hogberg & Vamling, 1996).

2.9.4. Vapour Specific Heat

Specific heat affects the slope of the temperature – entropy (T-S) graph both on the liquid and vapour line as demonstrated by (Didion & D, 1999). Increase in specific heat will lead to wet compression, which is undesirable for the compressor performance. A very low value of specific heat entails increase in the compression work, increase in discharge temperature and decrease in compressor capacity, hence, decrease in thermal efficiency and COP. As such, there is a need for a trade-off for the value of heat capacity. (Rotchana & Suen, 2012) analysed that for a higher COP to be obtained, it is required that vapour specific heat be high. As such for a refrigerant to serve as a replacement for R22 and R410A, it is required that BTEP should have a vapour specific heat that is higher than that of the two old refrigerants but not too high to avoid increase in compression work.

2.9.5. Vapour Density or Vapour Volume

For one refrigerant to replace another, the new refrigerant must have a better vapour pressure and specific volume. Monitoring the specific volume or vapour density brings us to same effect since one is the reciprocal of the other. Eisuke T et al., 2007 revealed that vapour density is directly proportional to mass flow rate. This entails more compressor work, hence, a higher refrigerating capacity at a cost, which is increase in energy consumption by the compressor. For a given compressor speed and size, increase in vapour density will increase compressor capacity and reduce pressure drop throughout the refrigeration system circuit (Rotchana & Suen, 2012). A 20% decrease in pressure drop at the level of the evaporator will result in 0.1% increase in compressor work during the cooling mode, which is countered by a 0.1% increase in heating COP (Domanski & Didion, 1987) (Hogberg & Vamling, 1996). As such, it is expected that BTEP have a vapour density higher than that of R22 and R410A or a lower specific volume in order to ensure that compressor size for the new refrigerant will be smaller.

2.9.6. Molecular Weight

Refrigerants with low molecular weight have been notice not only to have high enthalpy of evaporation, but to exhibit no energy losses across the compressor valve. This is not the case for higher molecular weight refrigerants that experience energy losses across the compressor valve and have low enthalpy of evaporation (Gosney, 1982) (Woollatt, 1982). As such a low molecular weight refrigerant will have higher efficiency.

2.9.7. Critical Temperature

Lower vapour density refrigerants have been realised to be associated with high critical temperature. As a result of the low vapour density of these refrigerants, their VRC will be low caused by low mass flow rate especially in constant speed compressor systems. On the other hand, for a variable speed compressor system, a larger compressor sweep (meaning more electrical energy) is required to provide same refrigerating capacity for high critical temperature refrigerants. This will result in reduction in flash gas losses, hence, an increase in COP. It was realised that a 10% increase in critical temperature, which is accompanied by a 10% decrease in vapour density will eventually give rise to an overall 9% increase in COP and a 6% decrease in compressor work, hence decrease in the amount of space swept by the compressor and corresponding decrease in VRC (Hogberg & Vamling, 1996) (Domanski PA, 1987). Decrease in compressor work means decrease in electrical energy consumed, which further contributes to increase in COP. As such a trade-off must be made between high capacity and high system performance (Didion & D, 1999) (McLinden & Didion, 1987). For a new blend to be arrived at, it is imperative that there be a refrigerant component peculiar for its high vapour density and another refrigerant noted or peculiar for its high critical temperature so that a compromised is arrived at.

2.9.8. Ozone Depletion Potential (ODP) and Global Warming Potential (GWP)

The ozone layer, which is a protective layer of gas about 10 – 50km above the earth surface shades the earth and its occupants from harmful UV-B radiation from the sun and completely prevents lethal UV-C radiation from reaching the earth's surface. Chlorine when released into the atmosphere is held in inactive compounds like hydrochloric acid, chlorine nitrate, etc. during winter. With the coming of spring, the UV radiation from the sun catalyses reactions giving rise to active chlorine monoxide, which is responsible for about 1% daily destruction of the ozone layer. As such, ODP is a reflection of the combination of percentage by weight of chlorine atoms and the lifetime of their compounds in the atmosphere (Lommers, et al., 2003). It is recommended that the ODP of BTEP be zero since it has been proven that annual refrigerant leakage for split-type ACs is in the order of 4 – 5% (UNEP, 2003).

Under normal operation, the earth is supposed to radiate some heat back to space. Due to the numerous thin films of gases surrounding the earth, some of these gases actually trap the heat and prevent it from going back into space thereby making the earth warmer and causing distortion to the original climatic pattern of the earth. This effect is referred to as the Global Warming Effect and the gases responsible for this phenomenon are referred to as greenhouse gases (GHG). Some of these GHGs include carbon dioxide, methane, nitrous oxide, HFCs, PFCs and SF_6. From pre-industrial times up to now, it has been realised that carbon dioxide, methane and nitrous oxide concentration has increased by 31 – 32%, 145% and 13% respectively while the concentration of the fluorinated compounds is relatively small though with a longer life span in the atmosphere (Lommers, et al., 2003). GWP is the climatic influence of a substance over a time horizon and it is evaluated relative to CO_2, which has a GWP of 1. In this thesis, the time horizon is 100yearss. As such, the GWP for any refrigerant intended to replace R22 permanently must have a lower GWP than that of R22, which is approximately 2088.

2.10. SUMMARY

- In the South African market, split-type ACs have experienced an 18% increase in purchase between the years1999 – 2013.
- Split-type ACs are made up of mainly the indoor and outdoor unit harbouring numerous parts including the compressor, condenser, evaporator, valves, motors, accumulator, refrigerant, louvres, fans and blowers
- They are capable of performing both heating and cooling with the aid of a four way reversing and a check valve.
- Their operation is based on VCRC or Carnot cycle and frost formation is prevented via the defrost cycle especially during heating.
- Performance monitoring is characterised by system COP and energy consumption for both modes of operation.

- AC operation is affected by three main influencers namely: environmental, system thermal and human behavioural variation. It is primordial to take all of them into consideration if adequate monitoring and prediction is to be achieved.
- Predictors taken into consideration from each of the system performance influencers include environmental variation – ambient temperature (T_a), relative humidity (RH) and outdoor unit periphery temperature (T_{cp}), system thermal variation – compressor and heat exchangers inlet/outlet temperature difference (T_m, T_e and T_n), human behavioural thermal variation – room temperature (T_r).
- Mathematical models are a more pragmatic means of representing complex real life situations with regression models being one of the simplest.
- Since R410A is a temporal replacement for R22 there is a need for a long term replacement.
- The choice of refrigerant depends primordially on environmental properties, which encompasses safety, TEWI and ODP.

3 CHAPTER THREE: METHODOLOGY

3.1. INTRODUCTION

This chapter elaborates on the AC system sizing procedure, the assumptions taken into consideration during the system sizing and installation, the experimental set up, each component making up the DAS and the location where the AC was installed. It also lists out all the data collected from the system and how the data was processed, analysed and modelled to arrive at the required results.

3.1.1. AIR CONDITIONING UNIT SIZING

In order for the unit to be properly sized, the heating and cooling load of the room in question needed to be estimated. This entailed knowing the dimension of the room, daily occupancy, number and type of heat generating and heat consuming appliances available and daily operational period.

The house into which the AC system was installed is the living room of a residential home in Alice, Eastern Cape Province, South Africa with geographical coordinates (-32.79^o, 26.84^o). It accommodates two adults and two children. Its dimension is 8.43 x 4.25 x 3.00 m^3 thus a floor area of ~35m^2 = ~377ft^2 as shown on the sketched floor plan in figure 3.1.

For this dissertation, the cooling and heating load calculations and equations enunciated in ASHRAE 2001 Fundamental Standards was used with some assumptions. The following assumptions were made during the load estimation.

- Since the heat loss and heat gain of children is less than that of adults as indicated in ASHRAE 2001 fundamental standards, for analysis purposes, the two children are considered to be one adult hence, an average of 3persons per day in the house.

- The heat gained and lost as a result of infiltration and exfiltration through door is ignored as the door in this particular home is always closed. The few times when the door is opened is for a very short space of time, most often at most a minute.
- At least one person in the living room at all times for all the 18hours a day, from 06:00 – 00:00.

Based on ASHRAE handbook of 2001, the cooling and heating loads for this room were calculated taking indoor summer temperature to be 78^{o}F (~25^{o}C) and outdoor temperature to be 85^{o}F (29^{o}C).

3.1.2. For cooling load estimation,

For the cooling load, it is worth noting that the "rule of thumb", which is a wide industrial means to estimate the cooling load states that for residential purposes, "1ton (12000btu/hr) = 400 – 600ft^2 (37.1 – 55.7m^2)" (Bhatia & B, 2012). This approximation is an over statement as not all rooms have same characteristics, orientation and building materials. The number of people in the house and the total number of hours they spend in the cooling space a day is needed. The entire house accommodates 3 adults and there is always at least one person in the living room at any particular time within the 18hours. Humans generate both sensible and latent heat, which will contribute in increasing the temperature of the cooling space, hence increasing the cooling load. As such, the total heat that is generated by one adult is given by equation 3.1

$$\boldsymbol{Q_{ppl} = Q_{sen} + Q_{lat}} \qquad \textbf{\textit{Eq. 3.1}}$$

$$\boldsymbol{Q_{sen} = N + SHG + CLF} \qquad \textbf{\textit{Eq. 3.2}}$$

$$\boldsymbol{Q_{lat} = N + LHG} \qquad \textbf{\textit{Eq. 3.3}}$$

Where:

CLF = cooling load factor = 1.0

LHG = latent heat gained per person per hour

N = number of people

Q_{lat} = latent heat gained

Q_{ppl} = heat gained due to presence of people

Q_{sen} = sensible heat gained

SHG = sensible heat gained per person per hour

Based on chapter 28 of (ASHRAE, 2001), an adult (male or female) person emits 400 – 600btu/hr of thermal heat meaning 225 – 275btu/hr SHG and 105 – 475btu/hr latent heat for sitting, light walking and working. On average, an adult person has SHG = 250btu/hr/per and LHG = 290btu/hr/per. This means that;

Q_{sen} = 3 x 250 x 1 = 750btu/hr

Q_{lat} = 3 x 290 = 870btu/hr

Q_{ppl} = 750 + 870 = 1,620btu/hr

The number of bulbs, laptops and TV are of importance because they generate heat. In this room in question, there are 2 bulbs, of 60W rating, 2 laptops each 65W and 1 – 100W TV. Therefore, we had 120W for the two bubs. The heat generated by electrical appliances is given by equation 3.4.

$$\boldsymbol{Q_{app} = 3.14 \times W \times F_U \times F_R \times CLF} \qquad \textbf{\textit{Eq. 3.4}}$$

$F_U = F_R$ = CLF = 1.0 (Bhatia & B, 2012)

Q_{light} = 3.14 x 120 = 376.8btu/hr

Q_{tv} = 3.14 x 100 = 314btu/hr

Q_{laptop} = 2 x 3.14 x 65 = 443.2btu/hr

Some heat is gained in the room due to the presence of windows. In this living room as seen in figure 3.1, there are 2 windows with one window facing west and the other facing east. For transparent windows, the amount of heat contributed due to its presence is given by equation 3.5.

$$Q_{win} = A \times maxSHG \times SC \times CLF \qquad \textit{Eq. 3.5}$$

Where:

SHGF = solar heat gain factor

Window area, A, is $1.5m^2 = 16.15ft^2$

$SHGC_{win1}$ facing west = 150 = $SHGC_{win2}$ facing east = 150 (Bhatia & B, 2012)

SC=CLF=1.0

For one of the windows, Q_{win1} = 2422.5btu/hr and for the second window, Q_{win2} = 2422.5btu/hr. For both windows, the sum is given in equation 3.6.

$$Q_{win} = Q_{win1} + Q_{win2} \qquad \textit{Eq. 3.6}$$

Q_{win} = 2422.5 + 2422.5 = 4845.0btu/hr

Some heat is gained via the roof and walls as such

$$Q_{wall} = A \times CLTD \times U \qquad \textit{Eq. 3.7}$$

For the wall facing west, average CLTD for 5 hours from midday = 17.2 while for wall facing east, average CLTD for 5 hours from 8am up till midday = 14. Considering an 8-in common brick,

Where:

U = 0.302

Q_{sw} = 8.23 x 4.23 x 17.2 x 0.302 = 181.69btu/hr

Q_{SE} = 147.88btu/hr

Q_{wall}=0.59+1.4 = 330.57btu/hr

Therefore, total cooling load is the sum of all the above contributors, which is 7930btu/hr. The cooling capacity of the purchased AC is expected to be at most 15% higher than the calculated cooling capacity while heating load should not be more than 125% of cooling load (Burdick,

2012) (EnerGuide, 2004). This means the unit to be purchased should be 9120btu/hr. The closest available unit was a 9000/12000 Btu/hr wherein the cooling capacity was 9000Btu/hr, which is about 13.5% higher than the calculated load and the heating capacity was 12000Btu/hr. This will cater for peak load scenario. The position of installation of the AC both indoor and outdoor unit is shown on the floor diagram in figure 3.1.

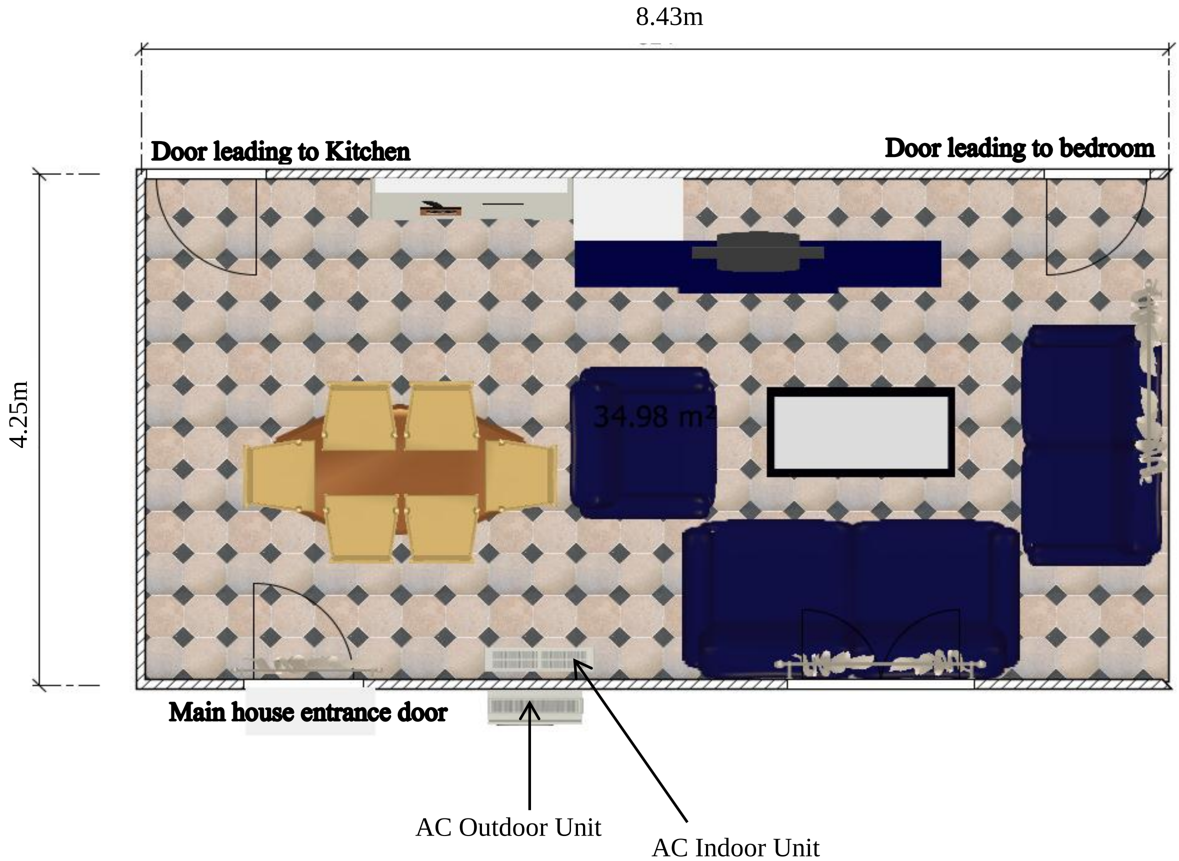


Figure 3.1: Floor plan including split-type AC

3.2. SYSTEM DESCRIPTION

The AC used is a SAMSUNG digital inverter-embedded domestic split-type AC with R410A as refrigerant. It has a cooling temperature range of 16 – 32°C and heating temperature range of any temperature less than 27°C. The cooling and heating input electrical power rating is 735W and 910W respectively while the cooling and heating output thermal power is 2.5kW and 3.3kW respectively. The indoor unit comprises of a control board, air flow blade and motor (stepper motor: 100pps, 12VDC, 1.47rpm), a fan and motor (single phase induction motor: 17W, 230V, 1300r/min), temperature sensor, evaporator coil and fins. The outdoor unit has a hermetic digital scroll compressor, which employs the services of a solenoid valve that regulates the compressor output and is run by an electronically commutated motor also known as a brushless DC motor (BLDC), a modulating drive or inverter which adjust the drive frequency to control the speed of the motor and vary the compressor capacity, reversing valve, fan and motor (single phase induction motor: 38W, 230V), condenser coil and fins as shown in figure 3.2.

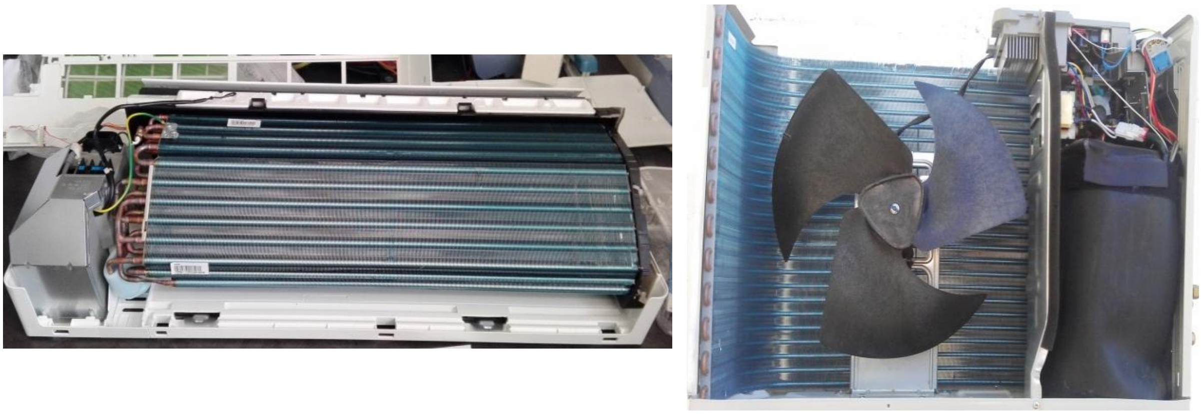


Figure 3.2: Photos of the opened indoor and outdoor unit respectively.

3.3. DAS

The DAS for this experiment comprised of TMC6-HE and TMC20-HD temperature sensors, Landys+Gyr Energy Meters, Current Transformers (CTs), data loggers and a relative humidity/ambient temperature sensor. Figure 3.3 is a block diagram of the AC showing the positions where the equipment that makes up the DAS was placed.

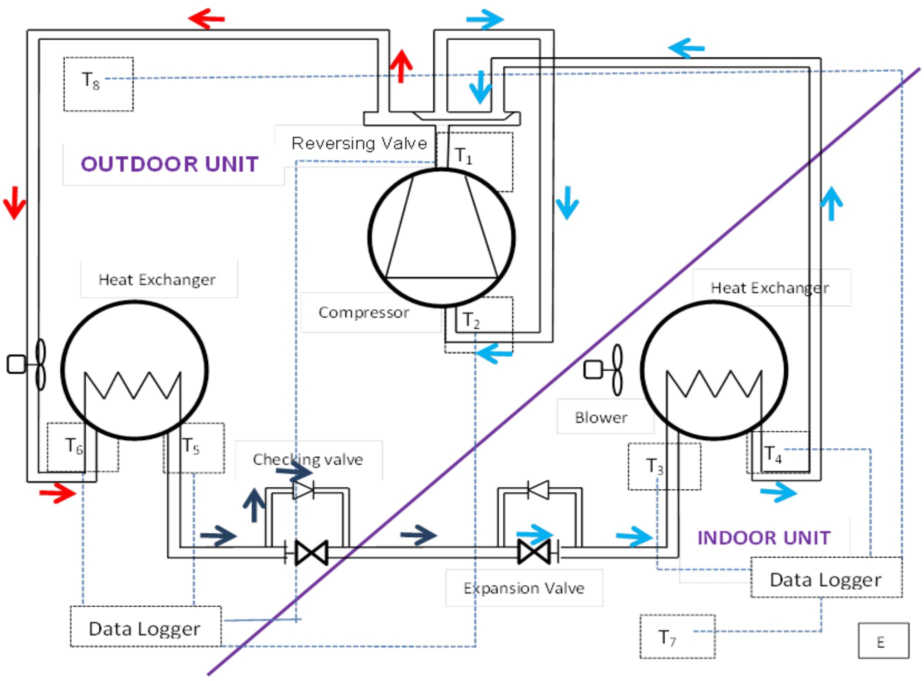


Figure 3.3: Block diagram of split-type AC including the positions of DAS equipment

3.3.1. Temperature Sensors

The temperature sensors used to build the DAS were TMC6-HE and TMC20-HD. TMC20-HD is a copper-plated-tip temperature sensor that can measure temperature in water, soil and air and can be connected to a pipe or left on a surface. It is 6.1m long and measures temperature of air and water/soil in the range of -40 to 100°C and -40 to 50°C respectively with a 2min response time in air (Onset, 2013-2015). This is what was used in position T_1, T_2, T_5 and T_6 of the DAS as shown in figure 3.3. Figure 3.4 is a photo of this sensor

Figure 3.4: TMC20-HD photo

TMC6-HE is also a copper-plated-tip temperature sensor that can measure temperature of air and other liquids; hence, it can be mounted on a surface or on a pipe. It has a temperature range of -40 to 100°C and it response two times faster on pipe than TMC20-HD. It is about

1.8m long and has a response time of 2min in air. On the DAS, it is T_3, T_4, T_7 and T_8. These temperature sensors are compatible with ONSET hobo loggers. Figure 3.5 is a photo of this sensor.

Figure 3.5: TMC6-HE Temperature Sensor

3.3.2. Energy Meter

The meter used to obtain energy consumed by the AC is the Landys+Gyr E650 Energy meter. This is a three phase class 0.5S/1.0 meter compatible with 5A, 1A CTs or both that has the ability to measure, display using LCD (8 digit display) and log voltage, current, power factor, active, apparent and reactive power and energy for each phase. It has an accuracy of 0.5% and 1% for both active and reactive energy respectively at full load and measures energy in all the four possible quadrants or directions. It has 11 input terminals with terminal 1, 3, 4, 6, 7 and 9 for current input and terminal 2, 5, 8 and 11 for the various phase voltages and neutral respectively. Its logger has many logging intervals but for this research, an interval of 5mins was used. Data from this meter can be downloaded through the dlms (Device Language Message Specification) readout point and an optical interface to a read out device like a laptop. This meter was connected to a CT. This RISH Xmer 50/30(30) class 1.0 CT has a ratio of 100/5A and an accuracy of 2.5VA. Figure 3.6 is a photo of the energy meter and the compatible CT used.

Figure 3.6: Photo of Landys+Gyr meter and CT

3.3.3. Data Logger

The data logger to which the temperature sensors were connected is an Onset HOBO 4-channel analogue data logger. It has four external inputs, can record numerous parameters and has different logging intervals but for this research, 5mins interval was used. It contains an LCD that displays current temperature readings for all the activated channels as well as the current battery status. It also contains a port to which a computer can be connected via a USB cable for data to be downloaded with the aid of the HOBOware. Figure 3.7 is a photo of the data logger connected to temperature sensors

Figure 3.7: Hobo data logger

3.4. EXPERIMENTAL SETUP

For this research, 4 TMC20-HD and 1 TMC6-HE temperature sensors were connected at some locations in the outdoor unit. The TMC6-HE temperature sensor was connected at the periphery of the outdoor unit while the 4 TMC20-HD temperature sensors were each connected at the input and output of the condenser and compressor respectively. In the outdoor unit was also connected a landys+gyr energy meter at the input to the outdoor fan. In the indoor unit were connected 3 TMC6-HE temperature sensors: one in the space to be conditioned, the other two at the input and output of the evaporator. A landys+gyr energy meter was also connected to the main input of the AC as show in the figure 3.8. All the eight temperature sensors were connected to two 4-channel analogue data loggers. Table 3.1 is a summary of the total number of equipment used to build up the DAS

The accuracy of an experimental data and DAS depends on the accuracy of the sensors and transducers (Coleman & Steel, 2010). The accuracy of the temperature sensors is ±0.15 that of the relative humidity and ambient temperature is ±0.2 while that of the Landys+Gyr energy meter is ±0.005. As such, the accuracy of the DAS is ±0.355.

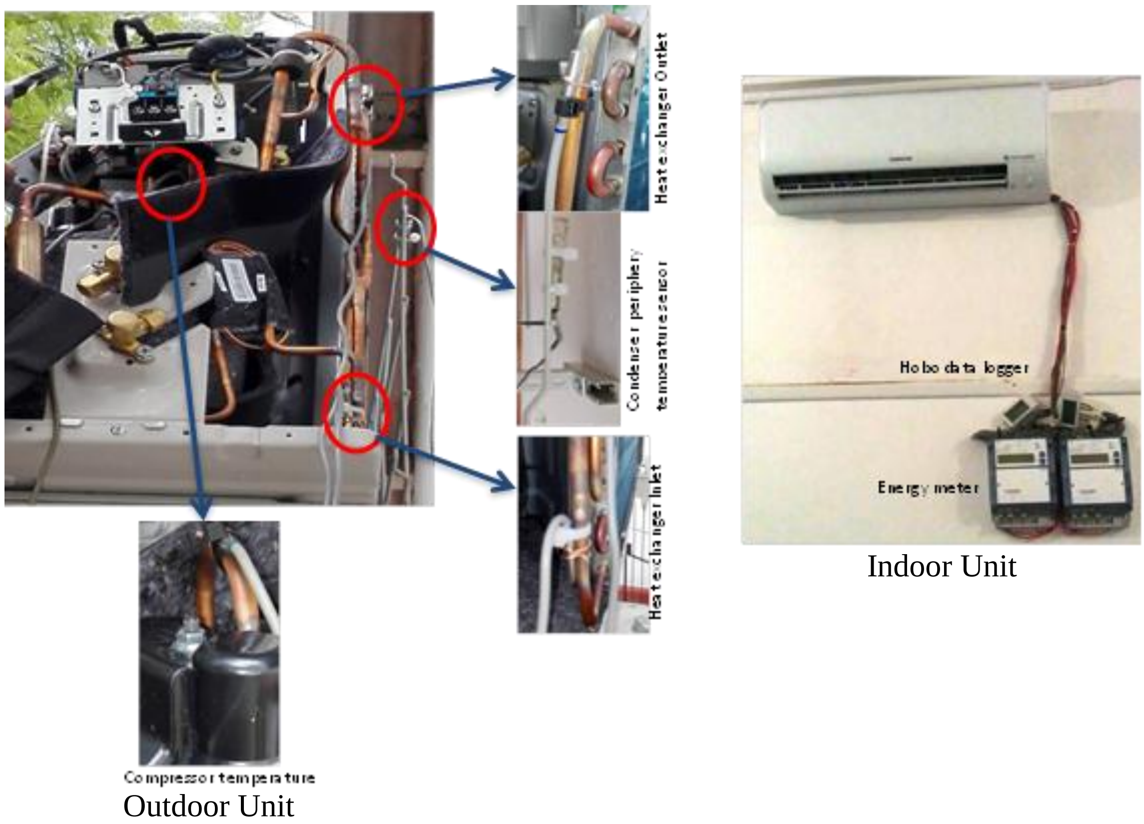


Figure 3.8: Picture of the Data Acquisition System

Table 3.1: Summary of the DAS

Equipment	Number	Purpose
E650 Landys+Gyr Energy meter and CT	02	Measure Energy consumed by the AC and the outdoor unit fan
TMC20-HD Temperature sensors	04	Measure temperature
TMC6-HE Temperature sensors	04	Measure temperature
Relative Humidity and Ambient Temperature Sensor	01	Measure relative humidity and ambient temperature
4-channel Data Loggers	02	To log temperature data during the experiment period

3.5. DATA ANALYSIS

The data obtained from the DAS includes energy consumed by the AC, temperature at the input and output of the compressor, condenser and evaporator, temperature of the room, temperature of the outdoor unit periphery, ambient temperature and relative humidity. The experiment was run for 8 months, from May 2016 to December, 2016. System heating performance was monitored from May to October while the system cooling performance was monitored from November to December. Both temperature and energy meters logged in five minutes interval and data analysis was carried out for an average day in thirty minutes interval for both seasons. The first step of the data analysis was data integrity check.

The temperature at the input and output of the condenser and evaporator were used to calculate the cooling and heating COP of the AC on REFPROP based on equations 2.13 and 2.14 in chapter 2. These temperatures were entered into the NIST software in order to get the corresponding enthalpy values, which were then substituted into afore mentioned equations. For COP_h, h_g was obtained from the vapour enthalpy of the indoor heat exchanger (acting as the condenser) inlet temperatures since the refrigerant at this point is a saturated vapour. On the other hand, h_o was obtained from the liquid enthalpy of the outdoor heat exchanger (acting as the evaporator) inlet temperatures since the refrigerant at this point is a saturated liquid. For COP_c, h_l was gotten from the vapour enthalpy of the indoor heat exchanger (acting as the evaporator) outlet temperatures whereas h_o was gotten from the liquid enthalpy of the outdoor heat exchanger (acting as the condenser) outlet temperatures.

On the ambient temperature, relative humidity, room temperature, outdoor unit periphery temperature, inlet and outlet temperature difference of the compressor, condenser and evaporator was applied the reliefF algorithm test relative to heating and cooling COP and energy consumption. This was done in order to ascertain the weight of influence of each of the predictors on the individual responses. The range of weight accorded to each predictor is within the range -1 to 1. A predictor with a weight of 1 has a very strong influence or correlation to the response while a predictor with a weight of -1 has no influence at all. Predictors with positive weights are primary factors while predictors with negative weights are secondary factors (Robnik-Sikonja & K, 2003). This test is done to avoid situations in which a predictor with very little significance is included in a model. The weight of the predictors was further normalised and converted into percentage, hence a percentage weight range of -100% to 100%. The computed COP and measured energy together with afore mentioned predictors that had more than 5% influence on the responses were used to build and develop four robust stochastic models on MATLAB.

For winter mode, five months (May to September) data was used to develop the model while the sixth month data (October) was used to assess the validity of the developed models. For summer, 6weeks data (November and first half of December) was used to develop the model while its validity was ascertained by the data of the last two weeks of December.

3.6. BTEP BUILD UP METHODOLOGY AND DATA ANALYSIS

As earlier mentioned, BTEP is composed of R32, R134a, R143a and R152a. Table 3.2 is a summary of the physical and safety properties of these refrigerants including that of R22 and R410A. A comparison is done between the four constituent refrigerants with respect to the following properties: latent heat of vaporisation, liquid thermal conductivity, liquid viscosity, specific heat, vapour density, molecular weight (MM), GWP and critical temperature. For each of these properties, the percentage contribution of each of the constituent refrigerants in BTEP for the respective properties is computed based on normalisation of their respective average values as shown in equation 3.8.

$$\% \boldsymbol{composition\ per\ constituent\ per\ property} = \frac{\boldsymbol{value\ of\ the\ property\ of\ that\ constituent}}{\sum \boldsymbol{values\ of\ same\ property\ for\ all\ constituents}} \times \mathbf{100} \qquad \textbf{\textit{Eq. 3.8}}$$

The total percentage composition of each of the refrigerant constituent in the new blend, BTEP, was computed by averaging the obtained individual refrigerant components' percentages for all respective properties as indicated in equation 3.9.

$$\% \boldsymbol{composition\ of\ per\ constituent\ in\ BTEP} = \frac{\sum \boldsymbol{constituent\ percentage\ per\ property}}{\mathbf{8}} \qquad \textbf{\textit{Eq. 3.9}}$$

The data used for comparing these thermodynamic properties of each of the refrigerant components was obtained from REFPROP software within temperature range of -5^{o}C to 60^{o}C. This is the operating temperature range for the condenser and evaporator of a typical split-type domestic AC. The method of obtaining this data was validated by comparing the obtained data with that of DuPont and ASHRAE 2001 fundamental handbook data. MATLAB software was used to plot the various graphs. Based on the performance of each of the individual refrigerants as regards the different thermo-physical properties, the proportion or percentage contribution of each of the components to make up the new refrigerant was determined. The thermodynamic properties of BTEP is compared to that of R22 and R410 because if this blend is supposed to serve as a long term replacement for R22, its thermodynamic and environmental as well as safety properties must be much better than that of R22 and R410A. For GWP of BTEP, it is obtained by summing the GWP of the components as a function of their percentage composition in the sample.

Table 3.2: Thermodynamic properties of the various components **(ASHRAE, 2001) (Lemmon, et al., 2013)**

Code	Name	Molecular Weight	T_{crit}	GWP	ODP	ASHRAE safety
R32	Difluoromethane	52	78	650	0	A2
R152a	Difluoroethane	66.1	114	140	0	A2
R134a	Tetrafluoroethane	102	101	1300	0	A1
R143a	Trifluoroethane	84	73	3800	0	A2
R22	Chlorodifluoromethane	86	96	1810	0.05	A1
R410A	R32/R125 (50/50)	73	71	2088	0	A1

3.5. SUMMARY

- The living room of a residential home with dimensions, 107m^3, needs a 9000/12000BTU/hr inverter split-type AC as demonstrated by the heating and cooling load computations taking into consideration human occupancy, electrical appliances, windows and walls.
- The performance monitoring and prediction DAS comprised of 8 temperature sensors, 1 relative humidity and ambient temperature sensor with inbuilt logger, 2 energy meters and 2 4-channel data loggers installed in the AC system.
- Based on the accuracy of each of the DAS components, the accuracy of the entire DAS was computed to be ±0.355.
- The experiment was run for 8 months from May to December and the data integrity checked used to develop four multiple linear regression models for both heating and cooling mode of operation.
- Data was treated both on REFPROP and MATLAB.

- BTEP constituent pure refrigerants to be blended are R134a, R143a, R152a and R32 and the properties being tested for are molecular weight, critical temperature, GWP, ODP, specific heat, vapour density, latent heat of vaporisation, liquid viscosity and liquid thermal conductivity.
- Simulation was run on each of the BTEP components to ascertain their percentage contribution in BTEP based on comparison of their individual properties. This simulation was carried out on NIST – REFPROP.

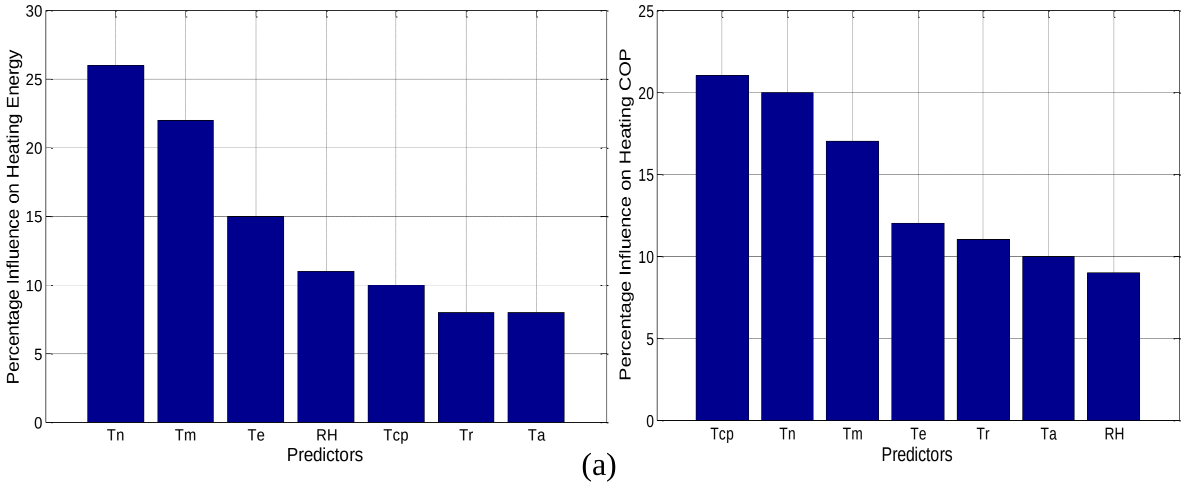
4. CHAPTER FOUR: AC PERFORMANCE MONITORING AND PREDICTION

4.1. INTRODUCTION

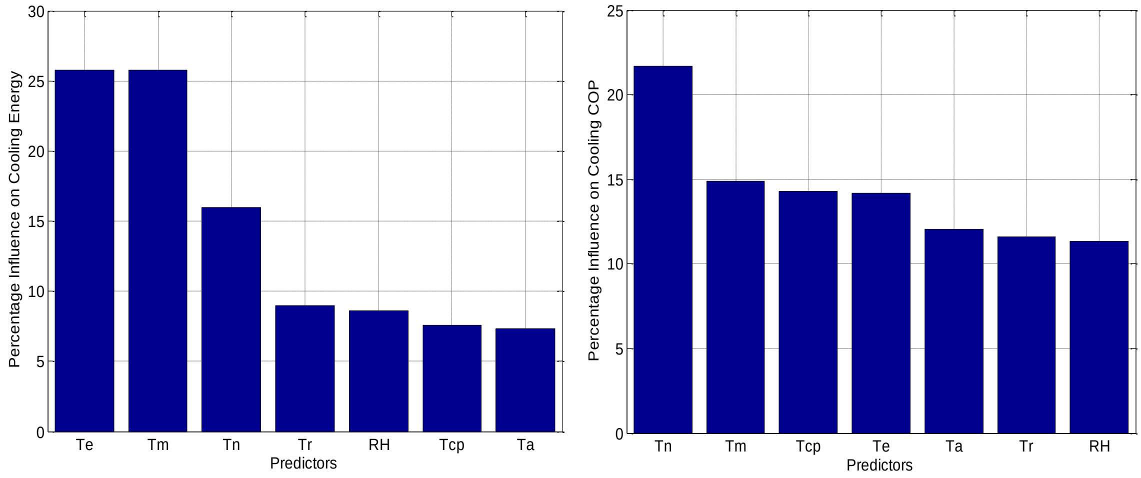
In this chapter is presented in details the result of the reliefF algorithm test carried out on the environmental predictors (ambient temperature, relative humidity), human behavioural predictors (room temperature) and system thermal variation predictors (Temperature difference at the inlet and outlet of the heat exchangers and the compressor) relative to the thermodynamic (COP) and electrical (electrical energy) performance of the split-type AC for both seasons. It also graphically highlights the influence of key predictors on the response and presents the four multiple linear regression models and their respective coefficient of performance. Finally, it presents the model confidence level, predictors' confidence boundary and percentage error for each of the models.

4.2. PREDICTOR SELECTION CONFIRMATION

The choice of predictors to be used for the models was confirmed using the reliefF algorithm test to ascertain the percentage weight of influence of each predictor on the responses for both heating and cooling mode. Figure 4.1a and b are bar charts displaying the reliefF algorithm test results for both heating and cooling mode performance respectively.



(a)



(b)

Figure 4.1: Variation of heating response with predictors for an average day

From figure 4.1, it can be observed that all predictors contribute more than 5% to the responses, hence can be used to monitor and predict system performance. These predictors are averagely about 5% more influential on the COP than on the energy.

Though the percentage contribution of each of the predictors differs for each of the responses, there is a remarkable difference in the case of T_m and T_{cp} with regards to the heating performance responses and T_e and T_m with regards to the cooling performance responses.

The response of E_h to T_{cp} is 10% less than the response of COP_h to T_{cp}. This is because T_{cp} is related and directly proportional to the amount of useful thermal energy picked up from the outdoor air by the refrigerant in the outdoor heat exchanger and made available to the space to be conditioned. Hence, it is related to the useful thermal energy gained and to the COP_h.

On the other hand, the response of E_h and E_c to T_m is respectively 5% and 11% higher than the response of COP_h and COP_c to T_m. How large or how small the compressor outlet temperature is required to be, determines how much work is needed by the compressor. Compressor work depends on the electrical energy input. This explains why there is a stronger relationship between T_m and both energies than with COP_h and COP_c.

Response of E_c to T_e is about 11% higher than COP_c response to T_e. A high difference in temperature between the outlet and inlet temperature of the evaporator (T_e) during cooling indicates much thermal energy was gained from the space to be cooled; hence, much work has to be done by the compressor to condition the refrigerant for condensing process. This means more electrical energy will be needed at the level of the compressor.

Response of COP_c to T_n is higher than that of E_c. This is because the amount of thermal energy gained from the space to be conditioned depends on how low the refrigerant temperature at the outlet of the condenser is. As such, the higher T_n, the higher thermal energy drawn from the space and hence, high COP_c.

This analysis also reveals that the most influential predictors on system performance are the environmental and system thermal variation predictors: T_{cp}, T_e, T_n and T_n.

4.3. PERFORMANCE MONITORING

4.3.1. Predictors' Influence on System Performance

The heating and cooling COP and energy consumed for both seasons were monitored using graphical analysis. With graphical analysis, the rate of variation of each predictor relative to each response is monitored bearing in mind that the other predictors are varying as well. Two regression models were built correlating room temperature with ambient temperature, relative humidity and outdoor unit periphery temperature in the absence of a space conditioning device for both winter and summer. These models, with an error margin of ±1^{o}C and ±0.7^{o}C respectively as shown in equation 4.1 and 4.2, were used to generate the baseline of what the room temperature would have been supposing the AC was not there.

$$\boldsymbol{T_{rc} = 3.91 - 0.0.06T_a + 0.57T_{cp} + 0.11RH} \qquad \textit{Eq. 4.1}$$

$$\boldsymbol{T_{rh} = 27.52 + 0.28T_a - 0.20T_{cp} - 0.03RH} \qquad \textit{Eq. 4.2}$$

Figure 4.2 shows two graphs depicting the baseline room temperature in the absence of an AC and the actual temperature in the presence of an AC for an average day in winter and summer.

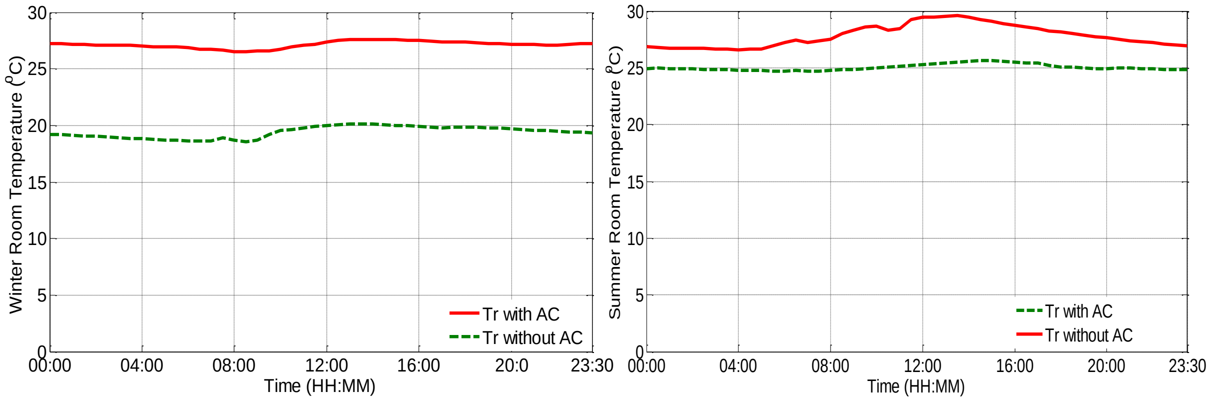
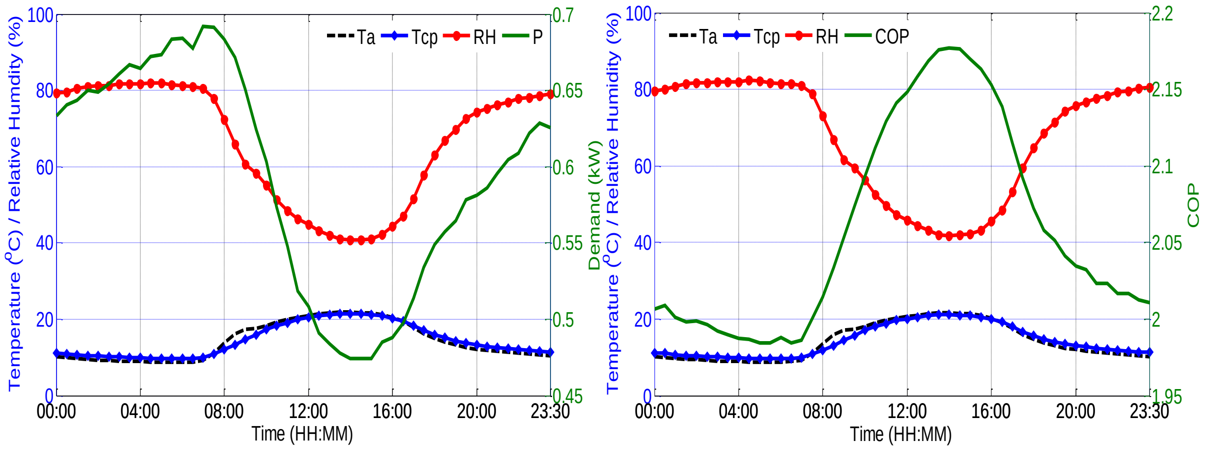


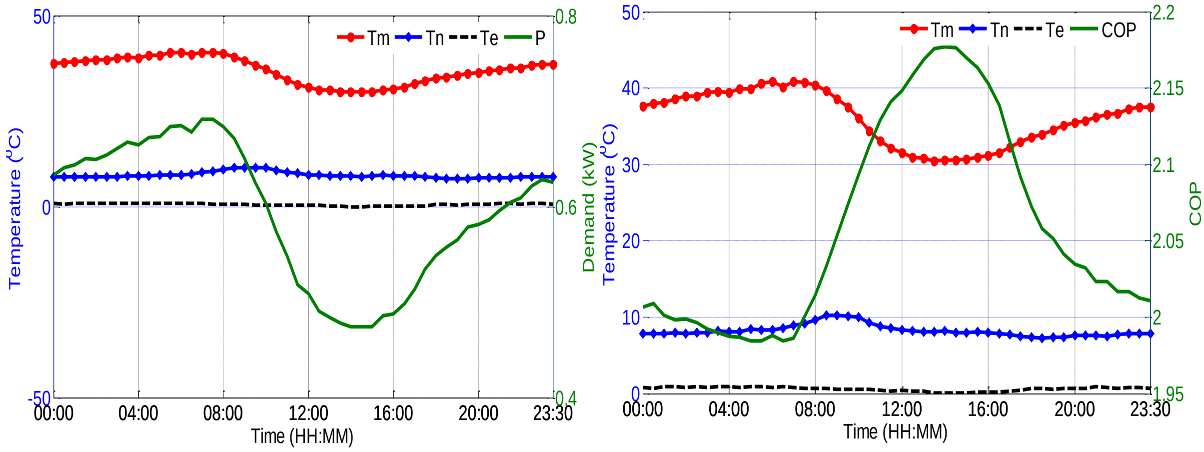
Figure 4.2: Graph of baseline average day room temperature and actual room temperature

It was observed that without an AC, the average daily room temperature in winter is about 19.40°C and in summer is about 27.82°C, while with an AC, the average room temperature in winter is 27.14°C and for summer it is 25.04°C.

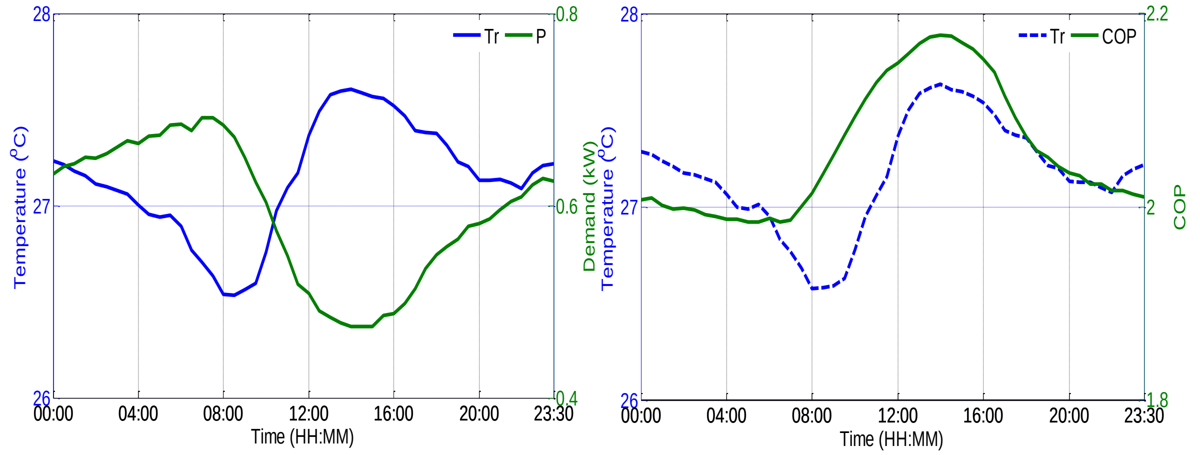
Figure 4.3 represents the result of the graphical analysis portraying the variation in influence of predictors on the heating mode responses for an average winter day while figure 4.4 is for the cooling mode responses for an average summer day. Graphs 4.3a are for environmental predictors, 4.3b are system thermal predictors and 4.3c for human behavioural predictor.



(a)



(b)



(c)

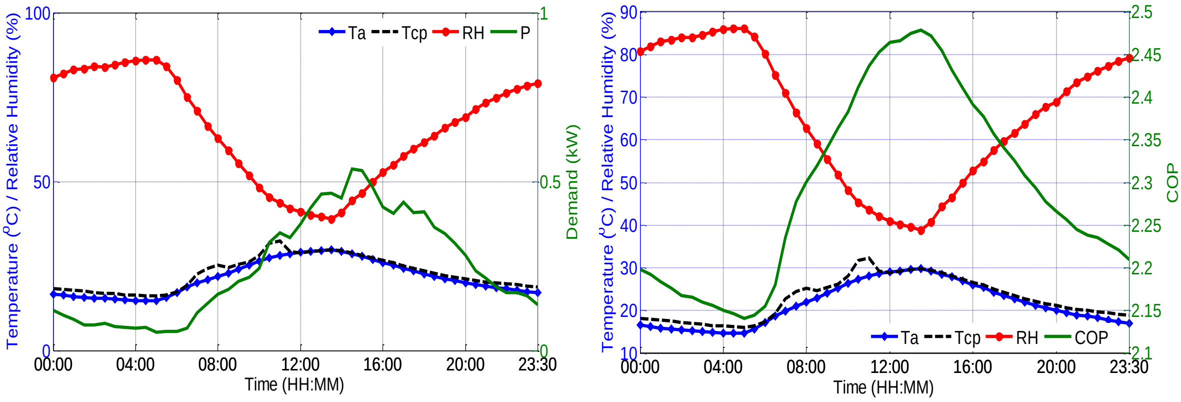
Figure 4.3: Variation of predictors with COP_h and E_h

Ambient temperature increases in winter for the time interval between about 08:00 and 17:00 as seen in figure 4.3a. During this time interval, the amount of thermal energy in the room increases with or without the AC, thus increasing T_r as shown in figure 4.2 and 4.3c. As such, the amount of work needed to be done in the compressor to step up the temperature of the refrigerant reduces since the refrigerant coming from the evaporator has gained a relatively high amount of thermal energy from the increased temperature outdoor air, T_{cp}. This means a decrease in compressor temperature difference, T_m, hence a decrease in the amount of electrical energy needed by the compressor and the entire AC as seen in figure 3b. Decrease in electrical energy and increase in thermal energy made available to the space to be heated means an increase in the COP_h of the system as demonstrated by the graphs.

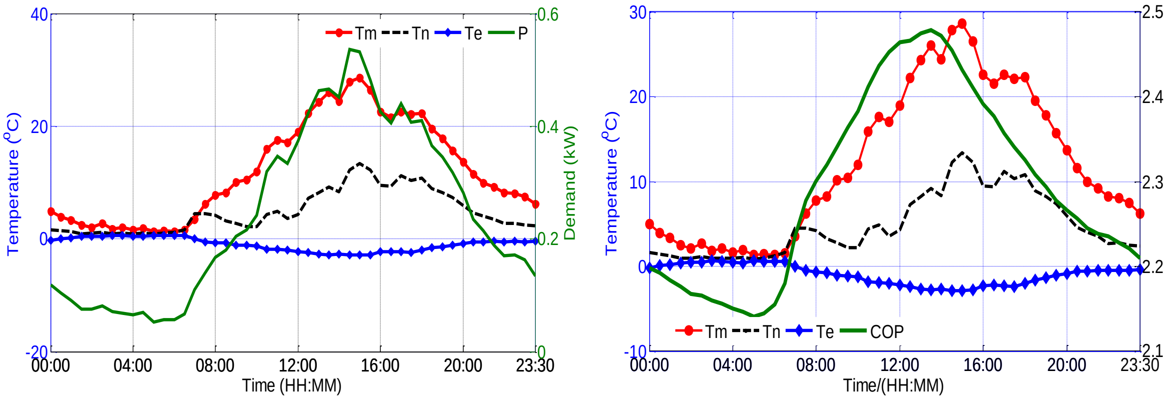
The reverse occurs when the ambient temperature becomes low as seen from about 17:30 to 07:30 of figure 3a. The low ambient temperature is responsible for decrease in room temperature, hence, increase in T_m and thus E_h. As such, figure 3 goes to show that E_h is directly proportional to RH, T_e, T_n and T_m. It is inversely proportional to T_a, T_{cp} and T_r. On the other hand, COP_h is directly proportional to T_a, T_{cp} and T_r and is inversely proportional to RH, T_e, T_n and T_m, which is expected since COP_h is inversely proportional to E_h.

For an average winter day, COP_{hmax} is 2.17 experienced between 12:00 and 16:00, when ambient temperature is highest whereas the COP_{hav} is 2.06. On the side of the energy, total daily energy is 2.36kWh with a maximum demand of 0.68kW experienced at about 07:30am when ambient temperature is lowest. When the AC is turned on for the first time or the room temperature reduced to about 25^oC from the set 27^oC, it was observed that the inverter type AC demand ramped to as high as 1.3kW depending on how low the ambient temperature was at the moment. Therefore, a 2^oC room temperature difference from the set temperature is enough to trigger the compressor to operate at a higher power, though this was not observed very often. As soon as the set temperature is attained, because the AC is an inverter type AC, it endeavours to maintain the temperature with a daily average demand of about 0.61kW.

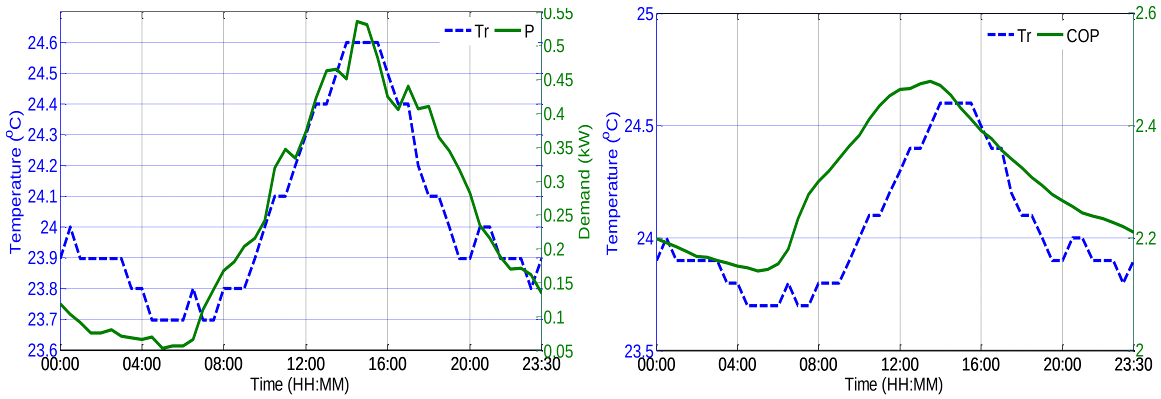
Figure 4.4 displays graphs of predictor variation influence on the cooling mode system responses for an average summer day



(a)



(b)



(c)

Figure 4.4: Variation of predictors with COP_c and Demand

It was realised that between the time interval of 05:00 and about 15:00, the cooling energy increases and so does the cooling COP. This time interval is accompanied by increase in ambient temperature as seen in figure 4.4a hence, increase in room temperature with or without an AC. As a result of the increase in T_a, it becomes difficult for the AC to dump the heat drawn from the room outdoors. For this to be achieved, the refrigerant temperature at the input of the condenser has to be higher than the outdoor temperature for heat transfer to be effected. This means more work needs to be done by the compressor, hence, more electrical energy required to achieve a suitably higher condenser inlet temperature to facilitate heat transfer to the surrounding. As such, for high T_n, more work has to be done by the compressor, hence, increase in E_c.

A high T_e means increase in the evaporator outlet temperature, hence little compressor work needed to be done by the compressor to get suitable condenser inlet temperature, thus decrease in E_c. This explains why, E_c is directly proportional to T_r, T_m, T_a, T_n, T_{cp} and inversely proportional to RH and T_e.

On the other hand, as T_a increase, T_r increases. With more work being done to facilitate removal of heat from the space to be cooled, more thermal energy will be transferred from the room to the outdoor air. A unit increase in electrical energy is accompanied by more than a unit increase in amount of thermal energy removed from the space to be cooled. This explains why though electrical energy increases due to the nature of variation of the predictors, COP_c increases as well.

COP_{cav} for an average summer day is 2.29 while COP_{cmax} is 2.48. Maximum demand is 0.54kW while total daily energy is 1kWh. This goes to confirm the fact that COP_h is always lower than the COP_c as a result of low ambient temperatures, which retards the performance of the split-type AC. It also revealed that only about 57.63% of total daily energy used for heating is used for daily cooling.

4.4. PERFOMANCE PREDICTION

Performance prediction was achieved by building multiple linear regression models correlating environmental, system and human behavioural thermal variables to system performance in terms of COP and electrical energy consumed. As a result of the fact that not all the possible predictors that affect the responses were taken into consideration, the introduction of a forcing constant in all of the models catered for this limitation.

4.4.1. Heating Mode Performance Prediction

The models for heating performance and their respective forcing and scaling constants are shown in the equations and table below.

$$\boldsymbol{E_h = \alpha_0 + \alpha_1 T_{cp} + \alpha_2 T_r + \alpha_3 T_n + \alpha_4 T_m + \alpha_5 T_e + \alpha_6 T_a + \alpha_7 RH} \qquad \textit{\textbf{Eq. 4.3}}$$

$$\boldsymbol{COP_h = \delta_0 + \delta_1 T_{cp} + \delta_2 T_r + \delta_3 T_n + \delta_4 T_m + \delta_5 T_e + \delta_6 T_a + \delta_7 RH} \qquad \textit{\textbf{Eq. 4.4}}$$

Table 4.1: Predictors' forcing and scaling constants for both winter responses

Response	Constants	Values	Response	Constants	Values
E_h, $r^2 = 0.90$	α_0	-1.15×10^{-2}	COP_h, $r^2 = 0.94$	δ_0	1.76
	α_1	-1.25×10^{-3}		δ_1	2.19×10^{-2}
	α_2	1.64×10^{-3}		δ_2	-2.11×10^{-3}
	α_3	9.38×10^{-4}		δ_3	7.18×10^{-4}
	α_4	6.62×10^{-4}		δ_4	2.47×10^{-3}
	α_5	2.04×10^{-3}		δ_5	-7.38×10^{-3}
	α_6	3.29×10^{-4}		δ_6	-3.35×10^{-3}
	α_7	-4.81×10^{-5}		δ_7	1.24×10^{-4}

The r value for E_h and COP_h is respectively 0.95 and 0.97, thereby indicating a strong correlation between these predictors and the heating performance response. The determination coefficient is as high as 0.90 and 0.94, hence a high degree of accuracy. This is further demonstrated on the graph of modelled and actual heating COP and energy consumption shown in figure 4.6.

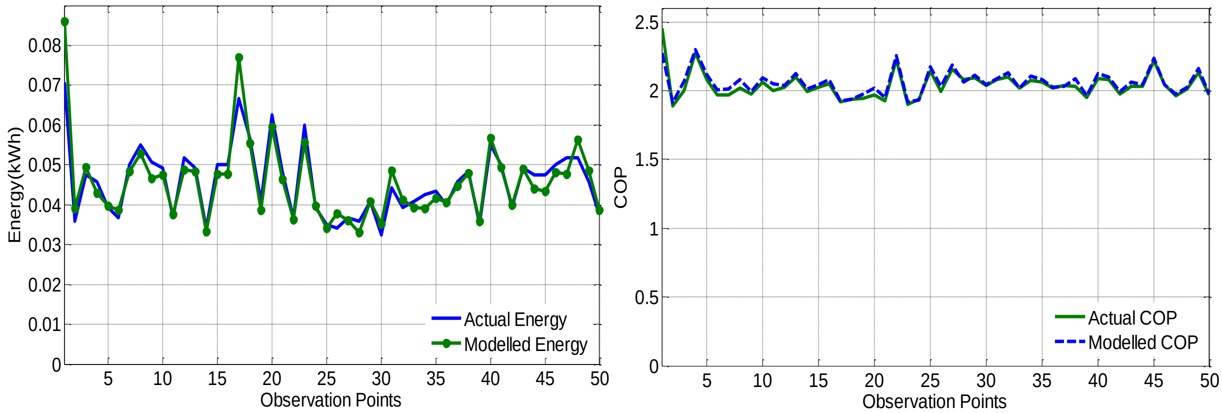


Figure 4.5: Graph of heating mode modelled and actual response

4.4.2. Cooling Mode Performance Prediction

The models for cooling performance and their respective forcing and scaling constants are shown in the equations and table below.

$$\boldsymbol{E_c = \gamma_0 + \gamma_1 T_{cp} + \gamma_2 T_r + \gamma_3 T_n + \gamma_4 T_m + \gamma_5 T_e + \gamma_6 T_a + \gamma_7 RH} \qquad \textbf{\textit{Eq. 4.5}}$$

$$\boldsymbol{COP_c = \partial_0 + \partial_1 T_{cp} + \partial_2 T_r + \partial_3 T_n + \partial_4 T_m + \partial_5 T_e + \partial_6 T_a + \partial_7 RH} \qquad \textbf{\textit{Eq. 4.6}}$$

Table 4.2: Predictors' forcing and scaling constants for both summer responses

Response	Constants	Values	Response	Constants	Values
	γ_0	-6.78×10^{-3}		∂_0	1.89
	γ_1	03.03×10^{-4}		∂_1	9.72×10^{-3}
	γ_2	3.72×10^{-4}		∂_2	-5.21×10^{-3}
E_c	γ_3	4.29×10^{-4}	COP_c	∂_3	-4.99×10^{-3}
$r^2 = 0.92$	γ_4	1.13×10^{-3}	$r^2 = 0.91$	∂_4	2.52×10^{-4}
	γ_5	5.33×10^{-4}		∂_5	7.12×10^{-4}
	γ_6	-2.69×10^{-4}		∂_6	1.22×10^{-2}
	γ_7	$-\ 9.12\times10^{-6}$		∂_7	$3.01\times10^{--4}$

The r value for E_c and COP_c is respectively 0.96 and 0.95, thereby indicating a strong correlation between these predictors and the cooling performance response. The determination coefficient is as high as 0.92 and 0.91, hence a high degree of accuracy. This is further demonstrated on the graph of modelled and actual cooling COP and energy consumption shown in figure 4.7.

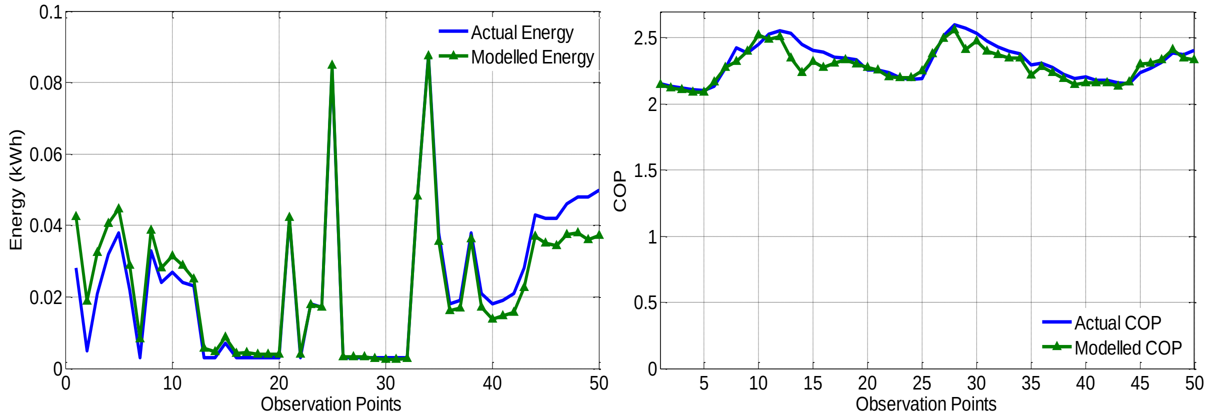


Figure 4.6: Graph of cooling mode modelled and actual response

4.5. MODEL VALIDATION

The heating mode models were tested using a two months test data (September and October), while the cooling mode models were tested using a two weeks data (last two weeks of December). After model validation, it was realised that the heating mode models have an average percentage error of ±4.27% and ±1.24% for heating energy and COP respectively as long as the predictors are within the confidence boundary defined in table 4.3.

Table 4.3: Heating and cooling mode predictors confidence boundaries

Predictors /AC Operation Mode	**Lower Limit**		**Upper Limit**	
	Heating	**Cooling**	**Heating**	**Cooling**
T_{cp}	2	11	31	38
T_n	-9	-3	29	47
T_m	0	-1	72	62
T_e	-15	-8	4	6
T_a	1	10	31	39
RH	1	16	100	98

For the AC cooling mode operation, model validation depicted an average error percentage of ±4.7 and ±1.24% for both cooling energy consumed and COP as long as the predictors are in the boundary listed in table 4.3.

4.6. SUMMARY

- All predictors have more than 5% influence on each of the responses as demonstrated by the reliefF algorithm; hence, they are all suitable for system performance monitoring and prediction.
- T_n influences COP than energy consumed while T_m influences energy more than COP for both seasons.

- Most influential predictors were system thermal variation predictors (T_m, T_n, T_e) and T_{cp}.
- Average daily winter room temperature was observed to be 19.40^{o}C while for summer was 27.14^{o}C.
- E_h is directly proportional to T_m, RH, T_e, T_n and inversely proportional to T_a, T_{cp} and T_r while the reverse is the case for COP_h.
- Average and maximum COP_c was respectively 10% and 12.5% higher than that of COP_h.
- Total daily energy used by the AC for an average day in winter was about 57.63% higher than that needed for an average day in summer based on the study analysis and data monitoring period.
- Determination and correlation coefficient for all the four models developed for performance prediction was above 0.90.
- Average percentage error for all the models ranged between ±1.24 and ±4.7%.

5. CHAPTER FIVE: BTEP

5.1. INTRODUCTION

This chapter concerns the simulation based development of a new environmental friendly long term replacement for R22 on REFPROP NIST software. It portrays the results obtained each step along the way from the thermodynamic and environmental properties of each of the constituent refrigerants, to the thermodynamic and environmental properties of the final blend

5.2. ENVIRONMENTAL PROPERTIES OF INDIVIDUAL COMPONENTS

Environmentally, based on table 3.2, all the refrigerant components have 0ODP. This means any blend composed of these refrigerants will definitely have 0ODP. Secondly, R152a, R32 and R143a are slightly flammable, the introduction of a non-flammable component, R134a, will render BTEP non-flammable. Among the refrigerant components that are to be used to make up BTEP, R152a has the lowest GWP. GWP value of R32, R134a and R143a is respectively 78%, 89% and 96% higher than that of R152a.

5.3. THERMODYNAMIC PROPERTIES ANALYSIS OF THE INDIVIDUAL COMPONENTS

The ensuing subsections elaborate on the performance of each of the individual refrigerant constituents with respect to the various properties that have been looked at in this research as generated on the NIST software. This is because as earlier mentioned, these same properties are the most important basis or measuring rod and criteria upon which a close to ideal R22 long term refrigerant replacement selection is based.

5.3.1. Latent Heat of Vaporisation

Figure 5.1 is a graph of latent heat of vaporisation against system saturation operation temperatures for each of the components used to build up the BTEP. The range of the saturation temperatures used in this graph covers the operating condenser and evaporator temperature of a typical domestic split-type AC from -5°C to 60°C as earlier declared. On a general note, the latent heat decreases with increase in operating temperatures. It is worth recalling that a refrigerant with a high latent heat of vaporisation enhances system performance. For system operating temperatures below 15°C, R152a has an average 2% lower latent heat of vaporisation than R32, hence portraying a decrease in system performance in that temperature range. Tables turn at temperatures higher than 15°C where R152a exhibits an average 10% higher latent heat of vaporisation than R32. This makes R152a to averagely have the highest latent heat of vaporisation with its average value being 5%, 37% and 45% higher than that of R32, R134a and R143a respectively when looking at the entire operating temperature range. The average latent heat of vaporisation of R152a and R32 is respectively 35% and 32% higher than that of R22. This goes to say that their presence in the new blend will ensure that the latent heat of vaporisation of that new blend is higher than that of R22.

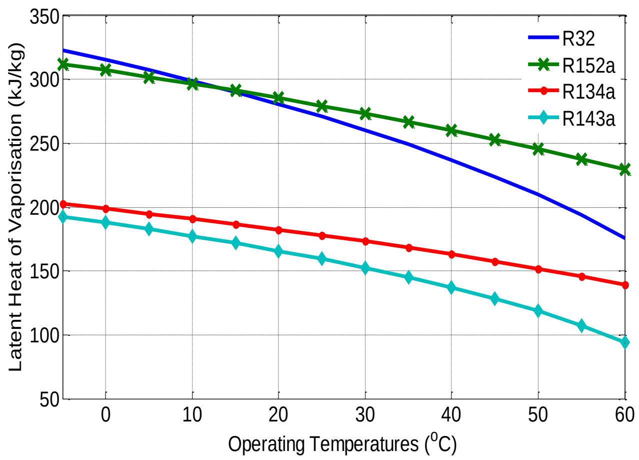


Figure 5.1: Graph of latent heat of vaporisation against operating temperature

5.3.2. Liquid Thermal Conductivity

Figure 5.2 is a graph of liquid thermal conductivity related to operating temperature for each of the refrigerant components. Liquid thermal conductivity decreases with increase in operating temperatures. As earlier mentioned in section 2.7.2 the effect of a high liquid thermal conductivity is high HTC and hence high system performance. R32 is topping the other refrigerant components. Its average liquid thermal conductivity is 124mW/m.K, which is 23%, 36% and 44% higher than that of R152a, R134a and R143a respectively. That of R32 and R152a is 58% and 15% higher than that of R22. Hence, their presence in BTEP will enormously and positively contribute to the HTC and hence COP of the system.

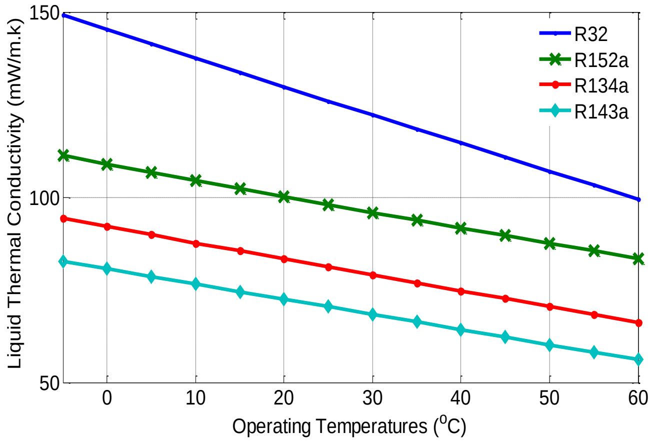


Figure 5.2: Liquid thermal conductivity against operating temperatures

5.3.3. Liquid Viscosity

Figure 5.3 is a graph of liquid viscosity against operating temperatures. Liquid viscosity decreases with increase in temperature. With the negative impact of high liquid viscosity on pressure drop, it is recommended that BTEP be made up of refrigerant components with low liquid viscosity. It was observed that for operating temperatures between -5°C and about 35°C, R32 averagely has a 5% lower liquid viscosity than R143a, hence its presence in BTEP in this temperature range will reduce pressure drop. From 35°C to 60°C, the liquid viscosity of R32 is 4% higher than that of R143a indicating the presence of R143a in this temperature range will enhance reduction in pressure drop. Averagely, looking at the entire operating temperature range, R32 has a liquid viscosity of 112uPa.s, which is 45%, 73% and 2% lower than that of R152a, R134a and R143a respectively. Average liquid viscosity of R32 and R143a are respectively 46% and 43% lower than that of R22 as such their presence in BTEP will ensure that the liquid viscosity is lower.

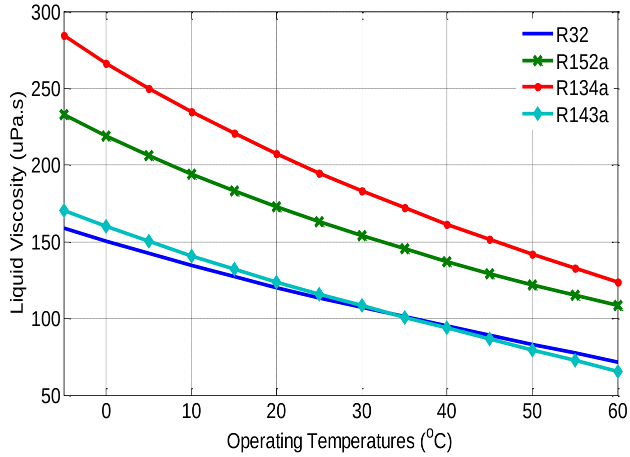


Figure 5.3: Graph of liquid viscosity against operating temperature

5.3.4. Specific Heat

Figure 5.4 is a graph of vapour specific heat against operating temperatures. As can be seen from the figure, vapour specific heat increases with increase in operating temperatures. In section 2.7.4, it was mentioned that a slightly high vapour specific heat enhances thermal efficiency and COP. Averagely, R32 has an average vapour specific heat of 1.873kJ/kgK, which is 31%, 42% and 14% higher than R152a, R134a and R143a respectively. The vapour specific heat of all these constituent refrigerants is higher than that of R22; hence, the presence of these refrigerants in BTEP will ensure that its vapour specific heat is higher than that of R22.

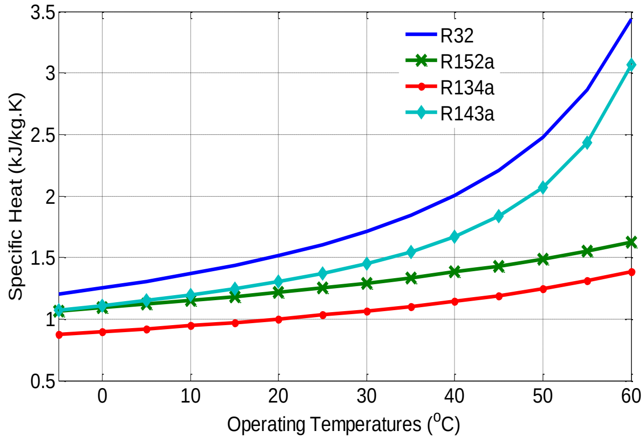


Figure 5.4: Graph of vapour specific heat against operating temperature

5.3.5. Vapour Density

Figure 5.5 is a graph of vapour density of the individual components. This property increases with increase in operating temperature. As elaborated in section 2.7.5, increase in vapour density will favour reduction in pressure drop hence, increase in VRC. From a comparison of the afore mentioned BTEP constituents, R143a has the highest vapour density with an average value of about 74kg/m^3, which is 50% higher than that of R134a, 25% higher than that of R32 and 72% higher than that of R152a. R143a and R32 have average vapour densities which are 27% and 10% higher than that of R22. As such it is possible that their presence in BTEP will further enhance the VRC of the system.

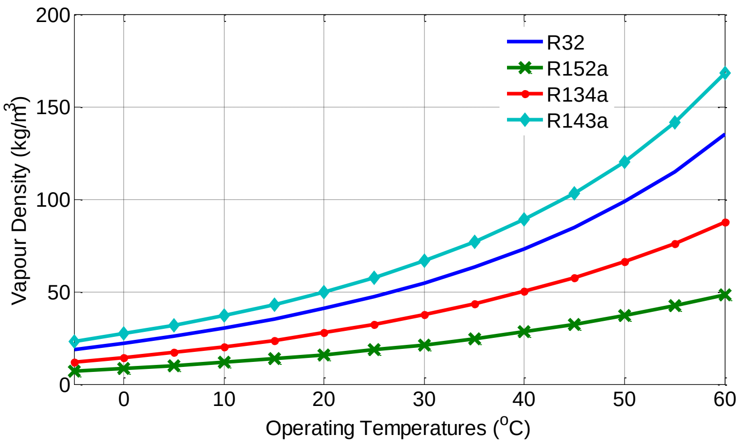


Figure 5.5: Graph of vapour density against operating temperature

5.3.6. Molecular Weight

Figure 5.6 is a chart representing molecular weight of each of the components. A high molecular weight means low latent heat of vaporisation reason why components of low molecular weight should have a higher percentage composition in the new mixture. From the comparison, it can be seen that R32 has the lowest molecular weight. Its value of 52kg/mol is 27%, 96% and 62% lower than that of R152a, R134a and R143a respectively. The molar mass of R32, R152a and R143a is 65%, 30% and 2% lower than that of R22, thus it can be said that their presence in the new blend will ensure that the molar mass of BTEP is lower than that of R22.

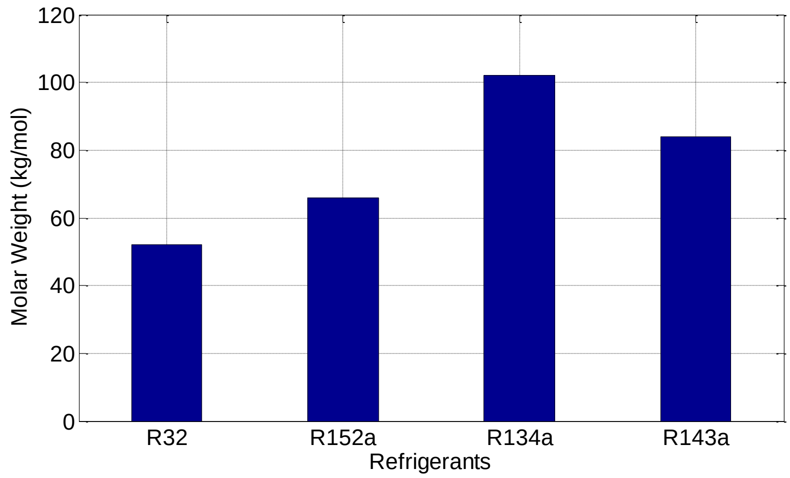


Figure 5.6: A bar chart of molecular weight

5.3.7. Critical Temperature

The critical temperature of the various refrigerant components is represented on the chart in figure 5.7. A high critical temperature refrigerant will increase the system COP. As such with R152a having the highest critical temperature its presence in the BTEP will ensure high COP and low compressor work. Its critical temperature is 32%, 11% and 36% higher than that of R32, R134a and R143a respectively.

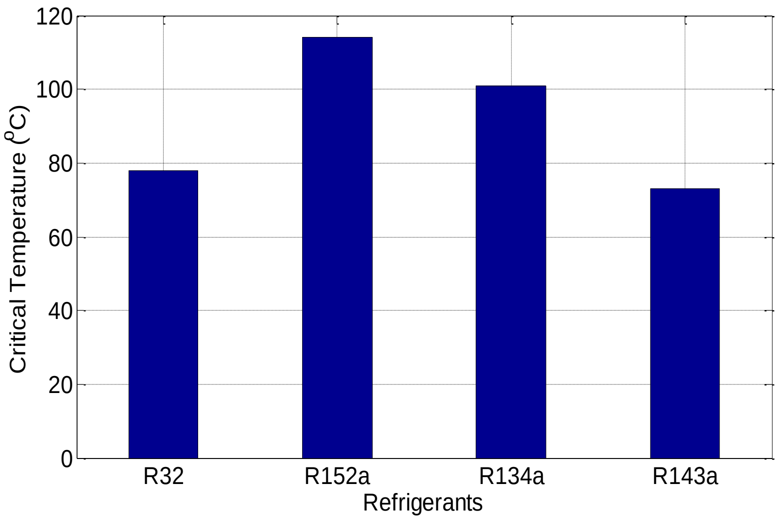


Figure 5.7: A bar chart of critical temperatures

A critical look at figure 5.5 and 5.7, it can be realised that R152a which has the highest critical temperature has the lowest vapour density whereas R143a with the highest vapour density has the lowest critical temperature. This confirms the research mentioned in section 2.7.7 about the antagonistic behaviour of these two properties, which is responsible for the antagonistic variation of both VRC and COP. As such, the presence of these two refrigerant constituents in the BTEP will ensure that there is a compromise between VRC and COP. As such, VRC and COP will both increase.

5.4. NEW BLEND RESULTS

From the analysis of the thermodynamic performance and environmental influence of each of the refrigerant components, it was realised that out of the eight properties being evaluated, R32 outperformed the other refrigerant constituents in 4 properties (liquid thermal conductivity, liquid viscosity, specific heat and molar mass), hence giving it a performance percentage of 50%. R152a also portrayed outstanding performance in 3 out of the 8 properties. These 3 include critical temperature, latent heat of vaporisation and GWP, thus a performance percentage of 38%. R143a also portrayed brilliant performance with respect to vapour density, thus keeping its percentage performance at 13%. Though R134a did not portray any outstanding or striking performance in the presence of afore mentioned refrigerant constituents for any of the evaluated properties, its presence in BTEP is not negotiable not only because of the fact that it has an overall good performance but also because its presence in the new blend will render the afore mentioned flammable constituents non-flammable when blended to generate the BTEP. The analysis of these individual components also confirmed the fact there is no pure component that can stand in as an ideal refrigerant replacement for R22. Nevertheless, blending refrigerants can give rise to a refrigerant mixture with properties much closer to an ideal refrigerant.

Table 5.1 contains the average value of the various thermodynamic properties and GWP for each of the constituent components in the operating temperature range -5^{o}C – 60^{o}C. It also contains the normalised percentage contribution of the individual component in the new blend for each of the respective properties.

Table 5.1: Percentage composition generation of each constituent in BTEP

Properties	**R32**	**R152a**	**R134a**	**R143a**
Av Latent Heat	260	274	174	152
% Composition	30%	32%	20%	18%
Av Conductivity	124	97	80	69
% Composition	33%	26%	22%	19%
Av Viscosity$^{(-)}$	112	163	195	114
% Composition	33%	20%	19%	28%
Av Specific Heat	1.87	1.30	1.08	1.60
% Composition	33%	22%	18%	27%
Av Vapour Density	60	23	40	74
% Composition	30%	12%	20%	38%
Molar Mass$^{(-)}$	52	66	102	84
% Composition	33%	28%	17%	22%
Av Critical Temp	78	114	101	73
% Composition	21%	31%	28%	20%
GWP$^{(-)}$	650	140	1300	3800
% Composition	22%	65%	11%	2%
BTEP	**29%**	**30%**	**19%**	**22%**

(-) The percentage contribution was not gotten from direct normalisation of the average values but was swapped such that the component with the highest and least percentage was swapped. This is because, refrigerant or system performance is inversely proportional to this property in question.

From the percentage normalisation summarised in table 5.1, the respective percentage of R32, R152a, R134a and R143a in BTEP is 29%, 30%, 19% and 22%.

5.4.1. Latent heat of vaporisation

Figure 5.8 is a graph showing the behaviour of the latent heat of vaporisation of BTEP together with that of R22 and R410A relative to operating temperature. As observed in earlier graphs, the latent heat of vaporisation generally decreases with increase in temperature and that of BTEP is highest. The average value of latent heat of vaporisation for BTEP over the temperature range is 225kJ/kg and it is 21% and 22% higher than that of R22 and R410A respectively. From -5 to 30°C, R410A has a higher latent heat of vaporisation than R22 but the table turns after 30°C. Averagely, latent heat of R22 is 1% higher than that of R410A.

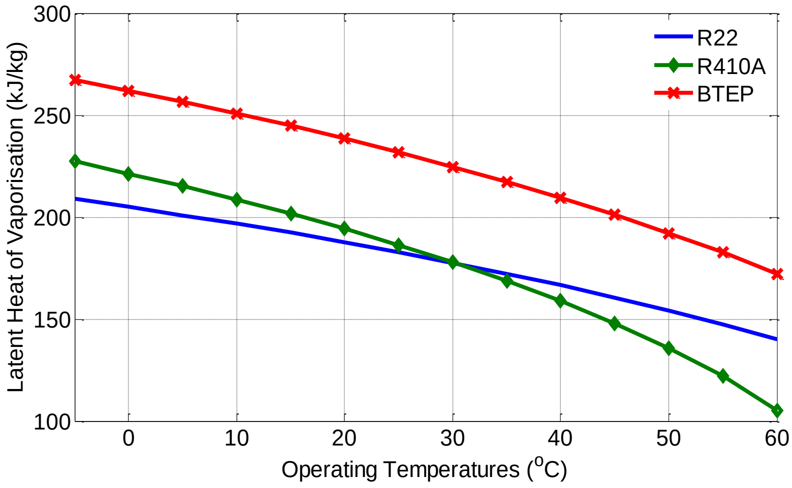


Figure 5.8: Graph of latent heat of vaporisation of the various blends

5.4.2. Liquid Thermal Conductivity

As seen in figure 5.9, liquid thermal conductivity decrease with increase in operating temperatures and that of the new blend is leading. As earlier mentioned in section 2.7.2, it is required that the liquid thermal conductivity of the refrigerant required to replace another be higher than the latter. The average liquid thermal conductivity of the new blend over the operating temperature range is 94mW/m.K. This average value for BTEP is 13% and 7% greater than that of R22 and R410A respectively.

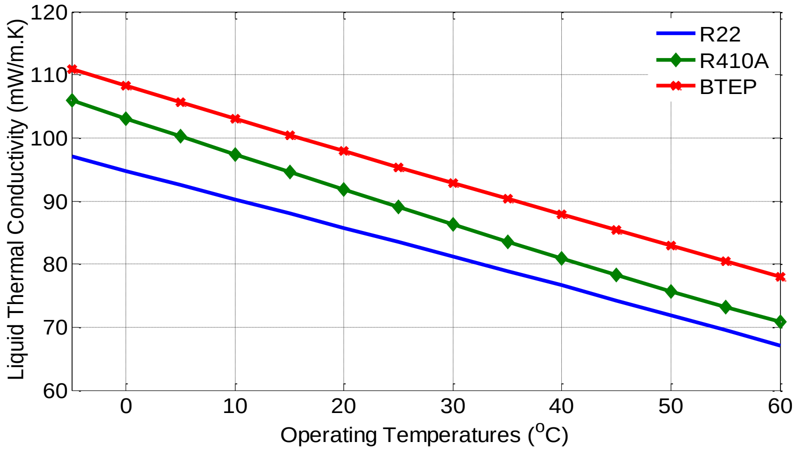


Figure 5.9: Graph of liquid thermal conductivity of the various blends

5.4.3. Liquid Viscosity

It is worth recalling that increase in liquid viscosity increases pressure drop across heat exchangers, which lowers the performance hence, it is better if the new refrigerant viscosity is lower than that of R22 and R410A. According to Figure 5.10, it is observed that liquid viscosity decreases with increase in system operating temperature. Secondly, the figure portrays that the average liquid viscosity of the BTEP is 22% lower than that of R22 but it is 13% higher than that of R410A. Its average liquid viscosity in that operating temperature range is 134uPas.

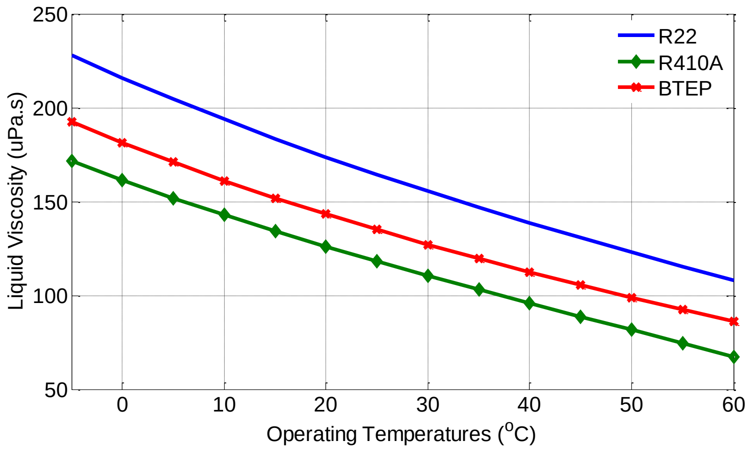


Figure 5.10: Graph of liquid viscosity of the various blends

5.4.4. Specific heat

Based on the result displayed on figure 5.11, it is observed that vapour specific heat increases with operating temperatures. The analysis carried out in this research revealed that at the refrigerant operating temperatures, the average vapour specific heat of the new blend, BTEP, is 1.339kJ/kg.K. This makes that of BTEP 31% higher than that of R22 but 34% lower than that of R410A. It is worth recalling that as expatiated upon in section 2.7.4, very high vapour specific heat possess both positive and negative effects. The advantage is that it enhances the AC thermal efficiency and COP. On the other hand, it causes wet compression, which is detrimental to compressor, hence there is need for a trade-off as concluded by Rotchana et al., 2012 (Rotchana & Suen, 2012). As such, the fact that the vapour specific heat of the new blend is slightly higher than R22 and lower than R410A is advantageous for BTEP and healthy for the system's compressor.

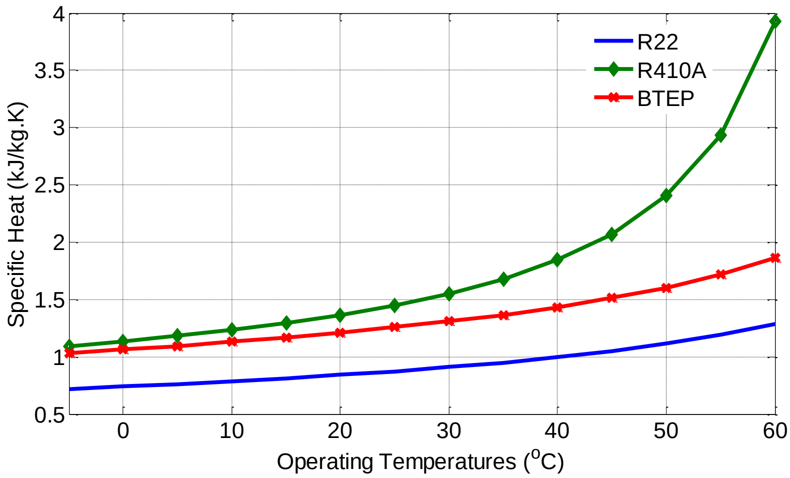


Figure 5.11: Graph of specific heat of the various blends

5.4.5. Vapour Density

The data analysis graphed on figure 5.12 demonstrates that the vapour density of R410A is highest and seconded by that of R22. The average value of BTEP is 41kg/m^3 and it is 32% and 108% lower than that of R22 and R410A respectively.

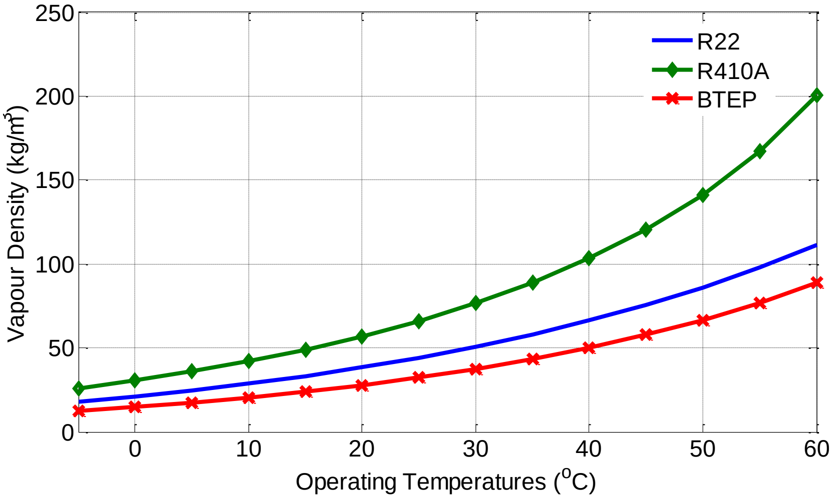


Figure 5.12: Graph of vapour density of the various blends

The low vapour density performance of BTEP can be explained based on the fact that the constituent with the lowest vapour density percentage of 12%, which is R152a, happened to be the highest refrigerant present in BTEP in terms of percentage composition as shown on table 5.2, hence, reducing the vapour density of the new mixture drastically.

5.4.6. Molecular Weight

From the bar chart in figure 5.13, it is observed that the molecular weight of BTEP, which is 69kg/mol, is the lowest, which is favourable as it means it has higher enthalpy of evaporation. Its weight is 6% and 25%, lower than that of R22 and R410A respectively; hence its latent heat of evaporation is higher than the others by the same percentage as shown on figure 5.8.

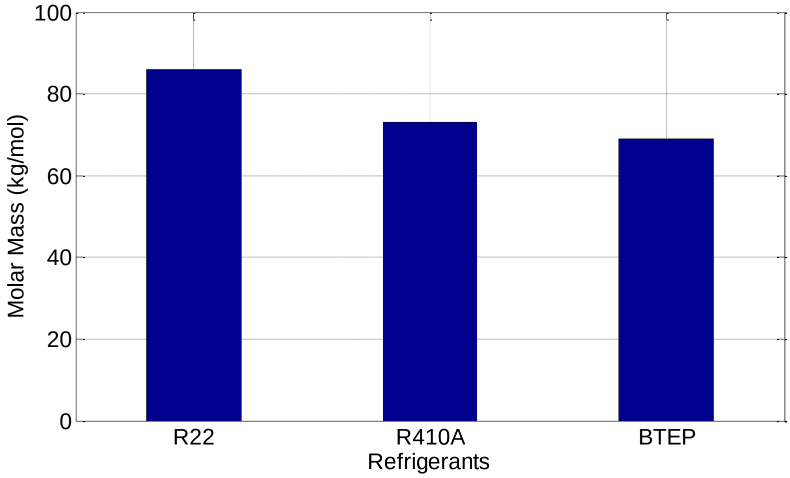


Figure 5.13: A bar chart of molecular weight of the various blends

5.4.7. Global Warming Potential (GWP)

The GWP_{100yrs} of the new blend was computed by summing the GWP_{100yrs} of the individual components in same ratio as their percentage presence in the blend. Based on this computation, it was realised that the GWP_{100yrs} of the new blend is 1314. This GWP_{100yrs} is 38% lower than that of R22 and 59% lower than that of R410A. This goes to say that BTEP is more environmentally friendly than R22 and R410A.

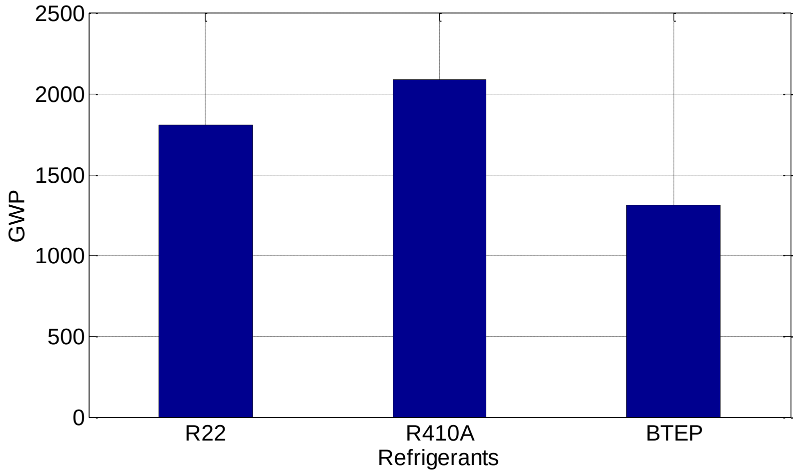


Figure 5.14: A bar chart of GWP of the various blends

5.4.8. Critical Temperature

The critical temperature of BTEP is 91°C. This is very close to that of R22, which is 96°C. As such, it can be observed that the critical temperature of BTEP is 5% lower than that of R22 but it is 30% higher than that of R410A.

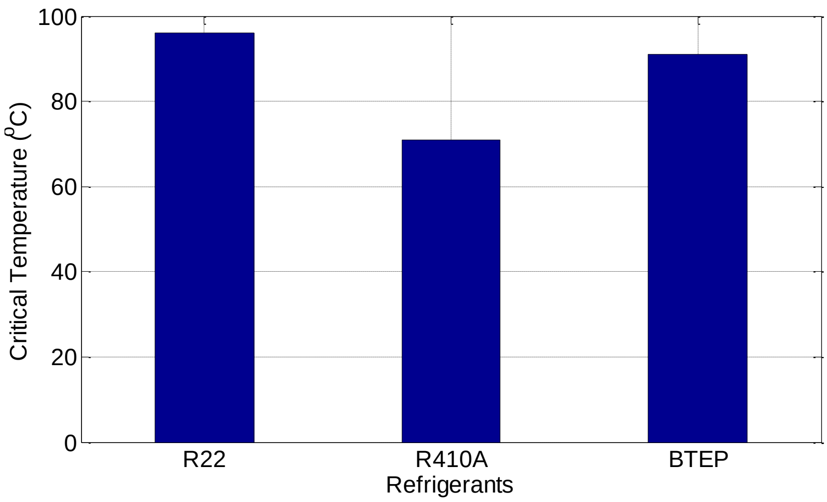


Figure 5.15: A bar chart of critical temperature of the various blends

5.5. SUMMARY

- For each of the thermodynamic properties being considered in this study, at least two of the BTEP constituents possessed properties higher than that of R22.
- All the constituents have a zero ODP, which fulfils the fundamental rule for choosing an R22 replacement.
- BTEP outperforms R22 in six out of the eight properties except for critical temperature and vapour density giving it a performance percentage of 75% relative to R22.
- BTEP outperforms R410A in five out of the eight properties giving it a performance percentage of 63% relative to R410A.

6. CHAPTER SIX: CONCLUSION AND RECOMMENDATION

6.1. AC PERFORMANCE PREDICTION

From this study, the reliefF algorithm statistical test showed that all the predictors have a significant influence on the four responses, hence, were suitable to be used for performance monitoring and prediction. Secondly, it was realised that most influential predictors were system thermal predictors and T_{cp}. Thirdly, the study proved that for the entire monitoring period, cooling COP is higher than heating COP, which goes to confirm the fact that low ambient temperature causes the system to underperform during the heating mode relative to the cooling mode. More energy is needed in a day to heat than to cool with average daily heating thermal energy being 57.63% higher than average daily cooling energy. Lastly, based on the determination and correlation coefficient of more than 90% achieved for the four models, it is sufficient to say that using the system, human behavioural as well as environmental variation predictors in a multiple linear regression model is suitable to predict the performance of the system with a higher degree of accuracy. This is also evident in the strength of the relationship observed between the predictors and each model. Hence, the environ-behavioural model can be adeptly used to improve on the prediction of the performance of a domestic split-type AC.

It is recommended that other more sophisticated models like probabilistic or stochastic models be used to monitor and predict the performance of a domestic split-type AC while varying the number of predictors and the performance compared to that of the regression model developed in this document.

6.2. BTEP

From the research carried out, a confirmation is made on the fact that it is not possible to have a pure refrigerant that will perfectly replace R22 yet it is possible to bring together and blend pure refrigerants that are outstanding in different properties to come up with an excellent blend that can perfectly replace R22. Secondly, it has been realised that, though BTEP was not outstanding in all its thermo-physical properties, out of the eight properties, it outperformed R22 in six of those properties except for critical temperature and vapour density where its value was 5% and 32% lower than that of R22 respectively. It is most important to note that BTEP has its 0ODP to give it an upper hand over R22. As such it goes to confirm the fact that there is yet no mixture or blend that can possess all the excellent thermo-physical or thermodynamic and environmental properties required of an ideal refrigerant as such a compromise needs to be arrived at. BTEP outperformed R410A in five out of the eight properties tested above, hence, a percentage performance of 63%.

It can be said that BTEP will have a higher performance in terms of COP supposing the old system is maintained and R22 is simply replaced with BTEP. On the other hand, in case it is desired to build an entirely new AC system with BTEP as refrigerant, the BTEP system will require a smaller capacity compressor with a high VRC than R22 and R410A to have same output as R22 and R410A, hence, decrease in electrical energy consumed. In both scenarios, there will be no effect on the ozone layer and a lower contribution to the global warming effect over 100years. As such from the simulations and analysis carried out it shows that BTEP outperforms R22 by 75% taking into consideration both thermo-physical and environmental conditions. It will thus make a better long term replacement for R22 on simulation basis. This new blend is recommended to undergo testing for chemical stability so as to ascertain its behaviour with heat transfer, metals and lubricating oils.

RESEARCH OUTPUTS

Full Paper Publications:

- Bantan Mafor Glory, Stephen Loh Tangwe and Michael Simon (2016) '*A review on the thermodynamic and electrical properties of South African acceptable refrigerants as r22 substitutes in in domestic air conditioners in South Africa*', 2016 International Conference of Domestic Use of Energy (DUE), Cape Town, South Africa, pp: 222 – 228.
- B.L. Glory, S.L. Tangwe and M. Simon (2017) '*Energy impact quantification of air conditioning as a replacement for traditional space conditioning devices*' 2017 South African Universities Power Engineering Conference (SAUPEC), Cape Town, South Africa.

Reviewed and Published Abstracts:

- Bantan Mafor Glory, Stephen Loh Tangwe and Michael Simon (2016) '*Performance evaluation of a domestic split-type air conditioner in South Africa, a case study of Alice*' 61st Annual Conference of South African Institute of Physics, Cape Town, South Africa

Non reviewed Accepted Abstracts:

- B Mafor G, S Tangwe and M Simon (2016) '*Comparative analysis on the techno-economic characteristics of renewable energy space conditioning devices: A review*' 7th Renewable Energy Post Graduate Symposium, Alice, South Africa.

Publication under Review:

- Glory M. Bantan, Stephen L. Tangwe and M. Simon (2017). '*Performance monitoring and prediction of a domestic split-type air conditioner*', 2017 International Conference of Domestic Use of Energy (DUE), Cape Town, South Africa.

REFERENCES

1. Abhishek, K. & Gupta, R. C., 2013. A Performance of a Window Air Conditioner Using Alternative Refrigerants R22. *International Journal of Engineering Sciences & Research.*

2. Air-conditioning, a. R. I., 1992-1997. R22 and R502 Alternative Refrigerants Evaluation Program.

3. ASHRAE, 2001. *HVAC 2001 Fundamentals Handbook.*

4. ASHRAE, 2001. *HVAC 2001 Fundamentals Handbook.*

5. ASHRAE, 2012. ASHRAE Position Document on Refrigerants and their Responsible Use.

6. Bantan, M. G., Stephen, L. H. & Simon, M., 2016. *A Review on the Thermodynamic and Electrical Properties of Domestic Air Conditioners Based on South African R22 Substitutes.* Cape Town, DUE, pp. 222-228.

7. Bhatia, A. & B, E., 2012. *HVAC Made Easy: A Guide to Heating & Cooling Load Estimation.*

8. Boladji, B., Huan, Z. & Francis O, B., 2014. Energy efficient performance of eco-friendly R152a and R600a refrigerants as alternatives to R134a in vapour compression refrigeration systems. *UEM*, 1(1453-7397), pp. 354-367.

9. Bolaji, B., Akintunde, M. & Falade, T., 2011. *Comparative Analysis of the Performance of three Ozone-Friendly HFC Refrigerants in a Vapour Compression Refrigerator.*

10. Bolaji, B. & Huan, Z., 2012. Computational Analysis of the Performance of Ozone-Friendly R22 Alternative Refrigerants in Vapour Compression Air-Conditioning Systems.

11. Bolaji, B., Komolafe, D. & Ajayi, F., 2015. *Performance Assessment of three Eco-Friendly Hydro-fluorocarbon and Hydrocarbon refrigerant mixtures as R22 alternatives in Refrigeration Systems.*

12. Boumaza, M., 2015. Thermodynamic Performance Assessment of Ozone Layer Friendly Natural Refrigerants as Potential Substitutes to HCFC's. *International Journal of Chemical Engineering and Applications.*

13. Brown, J. S., Kim, Y. & Domanski, P., 2002. Evaluation of Carbon DIoxide as R22 SUbstitute for Air conditioning.

14. Bukola, B., 2012. *Performance of A R22 split-air-conditioner when retrofitted with ozone friendly refrigerants (R410A and R417A).*

15. Burdick, A., 2012. *Strategy Guideline: HVAC Equipment Sizing*, Springfield: US Department of energy.

16. Chow, T. T., Lin, Z., Wang, W. Q. & Lu, J. W. Z., 1999. *Studying thermal performance of split-type air conditioners at building re-entrant via computer simulation.*

17. Coleman, H. W. & Steel, W. G., 2010. *Experimentation and Uncertainty Analysis for Engineers.* New York.

18. Covary, T., Preez, K. D. & Gotz, T., 2015. *Energy Efficient Air Conditioners.*

19. Dalkiliç, A. & Selim, 2010. Theoretical Analysis on the Prediction of Coefficient of Performance of two tage cascade refrigeration system using various refrigerant alternatives.

20. Dariusz, B. et al., 2005. *Safeguarding the Ozone Layer and the Global Climate System.*

21. Department, o. M. a. E., 2004. *Draft Energy Strategy of the Republic of South Africa.*

22. Devotta, S., Waghmare, A. V., Sawant, N. N. & Domkundwar, B. M., 2000. Alternatives to HCFC-22 for air conditioners. *Applied Thermal Engineering.*

23. Didion & D, A., 1999. The Influence of Thermo-physical fluid properties of the new ozone safe refrigerant on performance. *International Journal of Appliend Thermodynamics,* Volume 2(1), pp. 19-35.

24. Domanski PA, D. D., 1987. *Impact of refrigerant property uncertainties on prediction of vapour compression cycle performance,* Gaithersburg: U.S. Department of Commerce, National Bureau of Standards.

25. Domanski, P. A. & Didion, D. A., 1987. *Impact of refrigerant property uncertainties on prediction of vapour compression cycle performance,* Gaithersburg: U.S. Department of Commerce, National Bureau of Standards.

26. Douglas, M., Robinson & Eckhard, A. G., 1997. Efficiencies of transcritical CO2 cycles with and. *ELSEVIER.*

27. Eisuke, T., Shin-ichi, T. & Tomohiro, A., 2007. *Development of fast calculation method for ammonia refrigeration cycle and parameter adjustment with genetic algorithm.* Tokyo, Building Simulation.

28. EnerGuide, 2004. *Heating and cooling with a heat pump,* s.l.: Natural Resources Canada's Office of Efficiency.

29. Environmental, A. South African Hydrochlorofluorocarbons Phas out ManagementPlan Newsletter.

30. Eskom, 2015. *Heating, Ventilation and Air Conditioning (HVAC) Systems: Energy-Efficient Usage and Technologies.*

31. Eskom, I., 2013. *Make the switch that saves you 10% or more,* s.l.: Eskom.

32. Eyetwa, M. M., Lindiwe, M. & Philip, G., 2010. *South African Energy Synopsis 2010,* Pretoria: Department of Energy (DoE).

33. Gosney, W. B., 1982. *Principles of refrigeration.* Cambridge: Cambridge Unversity Press.

34. Hogberg, M. & Vamling, L., 1996. Impact of uncertainties on estimates of heat pump cycle performance. *International Journal of Refrigeration,* Volume 1, pp. 34-42.

35. Hu, H. et al., 2007. An experimental study on variable air volume operation of ducted air conditioning with digital scroll compressor and conventional scroll compressor. *ELSEVIER,* pp. 761-766.

36. Izham, M. N. & Mahlia, T. M. I., 2010. Effect of ambient temperature and relative humidity on COP of a split room air conditioner. *Journal of Energy and Environment,* pp. 35-38.

37. Lawrance, B. N. L., 2016. *Wealth, Warm Climates Driving Worldwide AC Boom,* Washington DC: ASHRAE Journal.

38. Lemmon, E. W., Huber, M. L. & McLinden, M. O., 2013. *NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 9.1*. Gaithersburg: National Institute of Standards and Technology, Standard Reference Data Program.

39. Lommers, C. A., AIRAH, F. & ASHRAE, M., 2003. *Air Conditioning and Refrigeration Industry Refrigerant Selection Guide,* Melbourne: The Australian Institute of refrigeration Air Conditioning and Heating (AIRAH).

40. Marion, G. & Lawson, D., 2008. *An Introduction to Mathematical Modelling.* Scotland.

41. McLinden, M. O. & Didion, D. A., 1987. Quest for alternatives. *ASHRAE,* Volume 29(12), pp. 32-42.

42. Modise, M., 2014. *Department of Energy NEES Parliamentary Portfolio Committee.*

43. Onset, C. C., 2013-2015. *TMCx-HD Water/Soil Temperature Sensor.*

44. Payne, W. V. & Domanski, P. A., 2002. A Comparison Of An R22 And An R410A Air Conditioner Operating At High Ambient temperatures. *International Refrigeration and Air Conditioning Cnnference.*

45. Prapainop, R. & Suen, K. O., 2012. Simulation of Potential Refrigerants for Retrofit Replacement. *ARPN Journal of Engineering and Applied Sciences.*

46. Qiu, T., Kaijun, D., Deqiu, Z. & Yongman, L., 2011. Experimental study on multi-split air conditioner with digital scroll compressor. *ELSEVIER,* pp. 2449-2457.

47. Ravi, K. K., Ganesh, B. K. & Udaya, B. S., 2013. Performance Evaluation of a Air conditioner accorfing to different test standards. *IJARET,* pp. 178-186.

48. Robnik-Sikonja, M. & K, L., 2003. Theoretical and Empirical Analysis of ReliefF: Machine Learning.

49. Rotchana, P. & Suen, K. O., 2012. Effects of refrigerant properties on refrigerant performance comparison: A review. *IJERA,* 2(2248-9622), pp. 486-493.

50. Stanciu, C., Gheorghian, A., Stanciu, D. & Dobrovicescu, A., 2011. Exergy Analysis And Refrigerant Effect On the Operation And Performance Limits Of A One Stage Vapour Compression And Refrigeration system. pp. 36-42.

51. Tonde, V., J, C. & Holm, D., 2001. *Measurement Based Quantification of storage water heater losses and Benefit of Addition Thermal Insulation.* Cape Town, Domestic Use of Energy.

52. UNEP, 2003. *2002 Report of the Refrigeration, Air Conditioning and Heat Pumps Technical Options Committee – 2002 Assessment,* Nairobi: UNEP Ozone Secretariat.

53. UNEP, 2006. *Energy Efficiency Guide for Industry in Asia.*

54. Vandaarhuzhali, S., 2014. Investigation on R152A as a substitute for R22 in Coommercial Air Conditioning System - A Review. *Singaporean Journal of Scientific research,* p. 15.

55. Venkataiah, S. & Venkata, R. G., 2014. Analysis of Alternative Refrigerants to R22 for Air-Conditioning Applications at Various Evaporating Temperatures.

56. Wakahara, T., Imanari, T., Nobe, T. & Kametani, S., 2010. *Development of the performance evaluation method for a split air conditioning system using the compressor characteristic curve.*

57. Wang & Shan, K., 2000. *Handbook of air conditioning and refrigeration.*

58. Woollatt, D., 1982. *Estimating valve losses when dynamic effects are important.* Purdue, Purdue Compressor technology conference.

59. Zaghdoudi, M. C., Maalej, S., Saad1, Y. & Bouchaala, M., 2010. A Comparative Study on the Performance and Environmental Characteristics of Alternatives to R22 in Residential Air Conditioners for Tunisian Market. *Journal of Environmental Science and Engineering,.*