University of Southern Queensland

Faculty of Health, Engineering and Sciences

FEASIBILITY ASSESSMENT OF USING WATER STORAGE SOURCES TO IMPROVE THE EFFICIENCY OF AIR-CONDITIONING SYSTEMS

A dissertation submitted by

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ENG4111 and ENG4112 RESEARCH PROJECT 1 & 2

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Abstract

The efficiency of air conditioning systems has improved but is fundamentally limited by the ambient conditions of the air outside the building (to which heat is dumped for cooling, or from which heat is extracted for heating). This research project investigated the feasibility of improving the efficiency of a reverse cycle air conditioning unit by using water from a solar hot water system and a residential pool as an additional source/sink of low-grade heat.

Reverse cycle air conditioning use for cooling and heating is becoming ever more prevalent worldwide. This is driven by the growing population and need for comfortable environments in households and workplaces to ensure employees are performing at high standards. This project extends on existing knowledge in using water as a heat sink/source in air-conditioning systems and assesses the feasibility of using such a design.

Baseline data was collected on a household reverse cycle ducted air conditioner (a/c), solar hot water system and residential pool by recording refrigerant temperatures, heat sink/source temperatures and power consumption over all seasons. This was used to ascertain a reasonable energy consumption without modifications throughout the year. Calculations were then made in MATLAB to extrapolate the energy consumption with the heat sink/source utilised on the system. Validation of the calculations made on the household system was performed by collecting the same data on a reverse cycle box a/c and performing similar MATLAB calculations with and without a water heat sink/source modification. The validation of the household a/c system's MATLAB calculations was consistent with the results of the box a/c system and the design was feasible using two different flow rate water pumps for heating and cooling respectively.

Results were a 2.74% reduction in power use for the household a/c system with a flowing water loop from a heat sink (residential pool) in heating mode and a 16.11% reduction in power use for the household a/c system with a flowing water loop from a heat source (solar hot water system) in cooling mode. These results used the highest performing water pump with a mass flow rate of 1.33kg/s, however, a 0.83kg/s water pump was feasible in cooling mode with a breakeven time of 1.5 years and a 0.42kg/s water pump was feasible in heating mode with a breakeven time of 4.1 years.

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Limitations of Use

Certification

I certify that the ideas, designs and experimental work, results, analyses and conclusions set out in this Progress Report are entirely my own effort, except where otherwise indicated and acknowledged.

I further certify that the work is original and has not been previously submitted for assessment in any other course or institution, except where specifically stated.

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Nomenclature

a/c	air-conditioning
α (alpha)	co-efficient of thermal expansion
c_p	heat capacity at constant pressure
ε	emissivity
ERV	energy recovery ventilator
h/H	specific enthalpy/enthalpy - the measure of energy in a thermodynamic system - the sum of the internal energy **E** plus the product of the pressure **p** and volume **V** ([J/kg]/[J])
HFC	hydrofluorocarbon
K	kelvin - SI unit of thermodynamic temperature
k	thermal conductivity
latent heat	heat that, when supplied to or removed from a body or thermodynamic system, results in a change in moisture content; the temperature of the air is not changed
ρ	density
s/S	specific entropy/entropy – the heat or energy change per degree Kelvin temperature (J/K)
sensible heat	the dry heat causing change in temperature but not in moisture content
WHS	Workplace Health and Safety
Heat Pumps	
COP_{HP}	Coefficient of Performance – efficiency of a heat pump
Q_L	magnitude of the heat absorbed from a low-temperature source
Q_H	magnitude of the heat supplied to the desired output
T_L	temperature of the heat absorbed from a low-temperature source (heat sink)
T_H	temperature of the heat supplied to the desired output (heat source)
$W_{net,in}$	net work input to the heat pump
Refrigerators	
COP_R	Coefficient of Performance – efficiency of a refrigerator
Q_L	magnitude of the heat removed from the refrigerated space
Q_H	magnitude of the heat rejected to the warm environment
T_L	temperature of the heat removed from the refrigerated space (heat source)
T_H	temperature of the heat rejected to the warm environment (heat sink)
$W_{net,in}$	net work input to the refrigerator

1. Introduction

1.1 Background

Air-conditioning use for both cooling and heating is becoming ever more prevalent in Australia and around the world. The growing population and need for comfortable environments in households and workplaces to ensure employees are performing at high standards have led to companies and residential households investing thousands in reverse-cycle air-conditioning systems. The last 10-15 years has seen an exponential growth in energy prices and with that several new designs and technologies created to increase system efficiencies and supplement these rising costs. Furthermore, most refrigerants used in these air-conditioning systems contain dangerous hydrofluorocarbons (HFCs) that are toxic to the environment, therefore another goal with these new designs and technologies is to reduce the harmful effects to the environment.

There are still several areas and designs that have not been appropriately assessed and researched to increase these efficiencies in air-conditioning systems and this project endeavours to perform further research in these areas and assess the feasibility of some of these designs.

1.2 Idea Development

The efficiency of air-conditioning systems has improved but is fundamentally limited by the ambient conditions of the air outside the building (to which heat is dumped for cooling, or from which heat is extracted for heating). This project will investigate the feasibility of improving the efficiency of a reverse-cycle air-conditioning unit. The method/hypothesis is to increase the temperature difference between the refrigerant and the source/sink, thus increasing the heat transfer rate of the refrigerant, reducing the mass flow rate of the refrigerant and therefore reducing the amount of energy required to process the refrigerant inside the air-conditioning unit.

This project will have to compare any improvements in efficiency within the air-conditioning system against any extra energy costs and capital costs of installing the additional heat-exchanger system(s).

1.3 Aim

The aim of this research project is to carry out a feasibility study of improving the efficiency of reverse-cycle air-conditioning systems using water from a solar hot-water system and a pool as a source/sink of low-grade heat.

1.4 Objectives

The objectives outlined below were the basis of reaching defined goals throughout the research and design process of the project. To ensure these objectives were carried out to a high standard and in a timely manner, a project plan utilising a Gantt Chart was formed to ensure the goals were met and that an outcome of value was produced. The objectives were designed to be specific, measurable, achievable and realistic in the given time frame and included added objectives to work towards if time permitted. Objectives for the research project of carrying out a feasibility study of improving the efficiency of reverse-cycle air-conditioning systems using water from a solar hot-water system and a pool as a source/sink of low-grade heat are as follows:

1. Review existing household reverse-cycle air-conditioning systems and evaluate any current use of heat sinks/sources in this area.

2. Research existing heat exchanger systems relating to reverse cycle air-conditioning systems to evaluate the feasibility of integrating heat sinks/sources into existing systems.

3. Perform data collection over all seasons on a pool, solar hot water system and reverse-cycle air-conditioning system to assess how temperatures and refrigerant mass flow rate changes under a variety of environmental conditions.

4. Design, build and test an air-conditioning system using pool water as a heat sink and solar hot water as a heat source.

5. Compare improvements in efficiency between the designed and built air-conditioning system using a heat sink/source and a standard air-conditioning system.

6. Investigate any additional energy/capital costs associated with installing the designed heat-exchanger system to an existing air-conditioning system.

If time and resources permit:

7. Field testing of the designed heat exchanger system on an existing reverse-cycle air-conditioning system.

8. Compare and evaluate results from design testing with results from field testing.

9. Make recommendations on refinement of current heat exchanger design for further efficiency gains.

2. Literature Review

The following review of literature will detail several key subjects relevant to the project area. Information from previously published works and reputable sources will be researched and critically reviewed to identify and understand the project area thoroughly. As the project is based around air-conditioning efficiencies, it will be imperative to highlight thermodynamic and heat transfer processes and the relevant factors relating to performance and efficiency. The over-arching goal of this review will be to identify and analyse what studies and findings have occurred in the area and ensure that the proposed methodology is original and will not be a replication of a previous study. Finding the knowledge gap or design gap is of the upmost importance and will aid in future works if required.

2.1 Heat Transfer Process

The heat transfer process occurs in nature when heat is transferred from the high temperature medium to the low temperature media; therefore, for this to occur there must be a temperature difference between the two media. When discussing this process and how it relates to air-conditioning and refrigeration it is important to understand the first and second law of thermodynamics, as a heat transfer process cannot take place unless both laws have been satisfied. The first law of thermodynamics is ‘energy cannot be created or destroyed during a process, only change forms; and the second law of thermodynamics is processes occur in a certain direction and energy has quality as well as quantity’ (Cengel & Boles 2015, p. 275). There are several variations to the second law of thermodynamics, however, the Clausius Statement is more relevant to refrigeration and heat pumps. It states that ‘it is impossible to construct a device that operates on a cycle and produces no effect other than the transfer of heat from a lower temperature body to a higher temperature body’ (Cengel & Boles 2015, p. 288).

These laws are based around the study of thermodynamics and are only concerned with the amount of heat transfer for a system undergoing a process. More specifically, thermodynamics deals with equilibrium states and changes from one form to another. Heat transfer deals with the determination of the rate of energy transfer, thus the laws of thermodynamics can be re-written considering this distinction as follows. Cengel & Ghajar (2015) describe how the first law requires the net rate of energy transfer into a system be equal to the rate of energy increase of that system and the second law requires that heat be transferred in the direction of decreasing temperature. The rate of energy transfer is subject to the magnitude of the temperature difference; whereby the larger the difference in temperature of the two media, the higher the rate of heat transfer.

2.2 Cooling in Air Conditioning Systems

Although there are several types of systems and designs, the most common air-conditioners typically use the vapour-compression refrigeration cycle to cool the inside air to a comfortable level. In this cycle the refrigerant is vaporised and condensed alternately and is compressed when in a gaseous state (Cengel & Boles, 2015). In performing these functions, the a/c system removes heat from the target space (source) and transfers it to the outside air (sink). Referring to figure 1, the phase changes of the refrigerant are caused by the heat transfer process of the refrigerant and ensure the cycle can continue by removing the heat Q_L (the desired output) from the cold inside environment T_L and transferring it to the warm outside environment T_H.

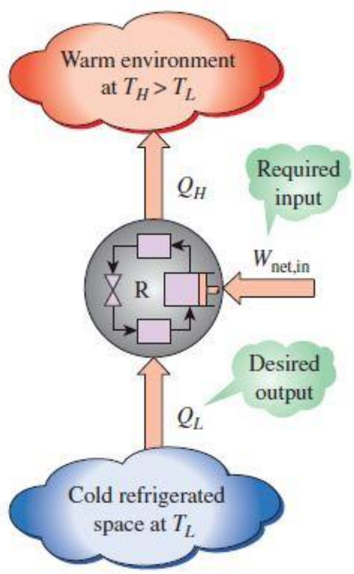

Figure 1. The objective of a refrigeration cycle (cooling): to remove Q_L from the cooled space (Cengel & Boles, 2015)

2.3 Heating in a Heat Pump

The basic heat pump uses the same vapour-compression refrigeration cycle as an air conditioner. The objective is to heat the space (supply the heat) instead of cooling the space (removing the heat) like an air-conditioner. To achieve this the flow of refrigerant is opposite when compared to an air conditioner, however, the flow direction does not change through the compressor as the compression of the refrigerant is still required, as discussed later in Section 2.4.1. Therefore, the target space for the desired output of the heat pump has been reversed and the system removes heat from the outside cooler air (sink) and transfers it to the target space (source). The cycle for the refrigerant is identical to a cooling system. Figure 2 shows how heat Q_L is drawn in through the condenser from the cold environment outside T_L and transferred to the warmer inside space T_H (Q_H being the desired output).

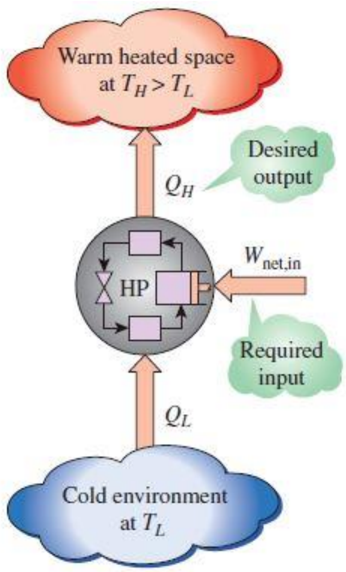

Figure 2. The objective of a heat pump: to supply heat Q_H into the warmer space (Cengel & Boles, 2015)

2.4 Components of an Air Conditioning/Heat Pump System

The main components are a compressor, condenser, throttling device and an evaporator. Depending on whether the system is an air-conditioner or a heat pump will determine what role the condenser and evaporator heat exchanger plays. These systems operate using a working fluid called refrigerant, which is an essential part of the system for absorbing and transferring the heat to the required area. The purpose of these systems is to extract heat from the outside ambient air and transfer it into the building if set to the heating mode (heat pump). In cooling mode, the system extracts heat from the inside of the building and transfers it to the outside, leaving cooler air on the inside (air conditioner). As the outside air is a sufficiently large thermal energy reservoir, there is no notable change in the outside air ambient temperature (See Section 2.6).

2.4.1 Compressors

The purpose of a compressor in an air conditioning system is to increase the pressure of the refrigerant saturated/superheated vapour by reducing its specific volume, thus compressing and changing the state of the refrigerant to a superheated vapour, which also has the effect of increasing the temperature of the refrigerant. As the refrigerant enters the compressor, the state of the refrigerant may not be possible to control so precisely, therefore by slightly superheating the vapour ensures the refrigerant is completely vaporised upon entry to the compressor. The compressor adds mechanical energy or work ($W_{net,in}$) to the system, in addition to circulating the flow of refrigeration through the cycle. The compressor is positioned between the evaporator and the condenser in the system and supplies refrigerant to the condenser at high pressure and relatively high temperature ready for condensing (Cengel & Boles, 2015).

Referring to figure 3, there are several types of refrigeration compressors including but not limited to reciprocating, screw, rotary, centrifugal and scroll type (Bright Hub Engineering, 2020). Reciprocating compressors are the most common type and have three sub-categories, being hermetically sealed, semi-hermetically sealed and open. This form of compressor uses the reciprocating motion of a piston arrangement to compress the refrigerant inside a cylinder.

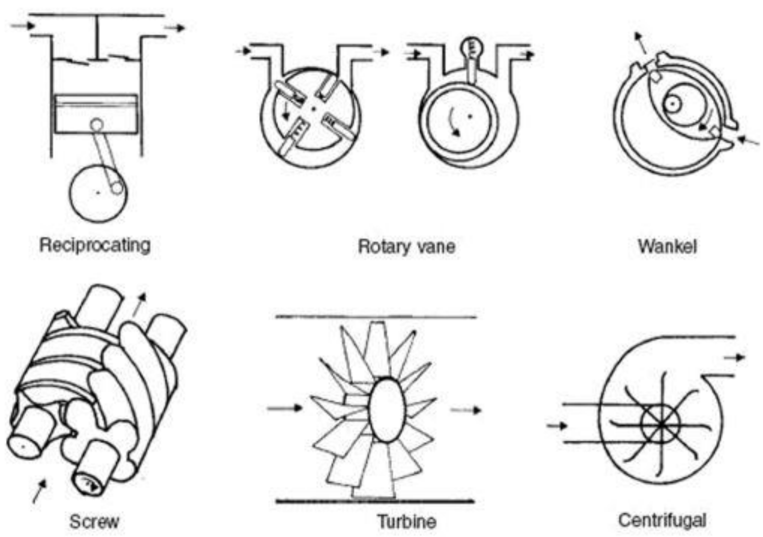

Figure 3. Common types of compressors available (Dincer, 2017)

2.4.2 Condensers

The condenser's function is to lower the temperature of the superheated vapour refrigerant as it flows through the coils of the condenser at constant pressure, thereby undergoing a phase change from superheated vapour to saturated or sub-cooled liquid (Dincer, 2017). The condenser is positioned between the compressor and the throttling device and comes in several types and configurations, each having a specific application.

The three types of condensers used in industry as are water-cooled, air-cooled and evaporative (Dincer, 2017). Sub-categories of the air- and water-cooled types include shell-and-tube, shell-and-coil and tube-in-tube. Air-cooled condensers are primarily used for domestic and commercial air-conditioning systems as outside air is used as the cooling medium (cooling mode). In cooling mode, these condensers have a fan that draws air past the coil containing the refrigerant and in turn the latent heat of the refrigerant is removed as sensible heat by this air flow. In heating mode, an air-cooled condenser acts more like an evaporator, whereby heat is extracted from the outside cold ambient air and past over the coils, transferring the heat through the system to the inside heated space.

2.4.3 Throttling Devices

A throttling device is used in the refrigeration process to reduce the condensing pressure of the refrigerant to the required evaporator pressure and to control the flow of refrigerant to the evaporator depending on loading characteristics (Dincer, 2017). As the throttling device is located between the condenser and the evaporator in the cycle, the refrigerant enters the device at high pressure and the "throttling" effect or expansion drastically reduces the pressure of the refrigerant, also reducing the temperature. Depending on what type of throttling device is used will determine the rate at which refrigerant enters the evaporator and in turn the amount of heat that is transferred to the refrigerant.

Some common types of throttling devices include capillary tubes, expansion valves and float valves, each of which are designed to be used for certain applications. There are sub-categories of each type, however the thermostatic expansion valve is the most common type, which controls the flow of liquid refrigerant to the evaporator based on system load and capacity by sensing the outlet temperature of the refrigerant vapour leaving the evaporator.

2.4.4 Evaporators

The function of the evaporator is to absorb the heat from the refrigerated space, thus providing the required cooling effect (Dincer, 2017). The refrigerant enters the evaporator from the throttling device as a low-temperature, low-pressure saturated liquid, however, this usually only occurs in an ideal cycle or if the system is carefully tuned. The actual cycle tends to have the refrigerant as a sub-cooled liquid as it enters the evaporator. During the heat transfer process in the evaporator, the liquid refrigerant evaporates as it is heated due to the refrigerant being at the saturation temperature for the given pressure, so any added energy (due to the heat transfer) will cause a phase change, resulting in a saturated vapour or slightly superheated vapour (Cengel and Boles, 2015). The vapour-compression cycle is then completed as the vapour refrigerant returns to the compressor to begin the process again.

In a heat pump, the function of the evaporator is to absorb heat from the heat source. In the case of a heat pump for space heating, heat energy from the outside cold ambient air is absorbed through the evaporator coil (serving as a condenser in this mode) and to transfer this heat energy to the condenser coil (serving as an evaporator in this mode) on the inside and release it into the heated space by a fan and ducting (Goodman Manufacturing Company, 2020) (refer figure 5).

Evaporators are divided into two main categories: direct cooler type and indirect cooler type. Direct cooler evaporators cool air, which in turn cools the required space and indirect cooler evaporators cool liquid, which in turn cools the required space (Dincer, 2017). These can further be divided up into liquid, air and gas coolers, which are each used for specific applications. Each of the components described above operate on a steady-state, steady-flow process and operate continuously during system operation.

2.5 Reverse Cycle Air Conditioning Systems

There are several types of reverse cycle air conditioning systems currently on the market, however all have the same basic principles. This being, the principles of the vapour-compression refrigeration cycle, which depending on the a/c setting will differ in its objectives. Refer to figures 4 and 5 for a schematic of the flow of refrigerant in a reverse cycle air conditioning system. As stated in Section 2.4, both air-conditioners and heat pumps individually have the same mechanical components, however, a reverse cycle air conditioner has the addition of a reversing valve, which essentially enables one system to function as a heat pump in winter and an air conditioner in summer by reversing the flow of refrigerant after the compressor.

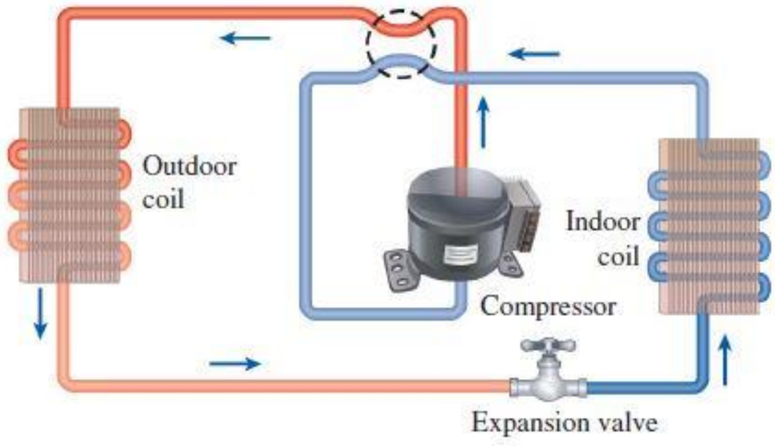

Figure 4. Reverse cycle a/c system operation – Cooling mode (Cengel & Boles, 2015)

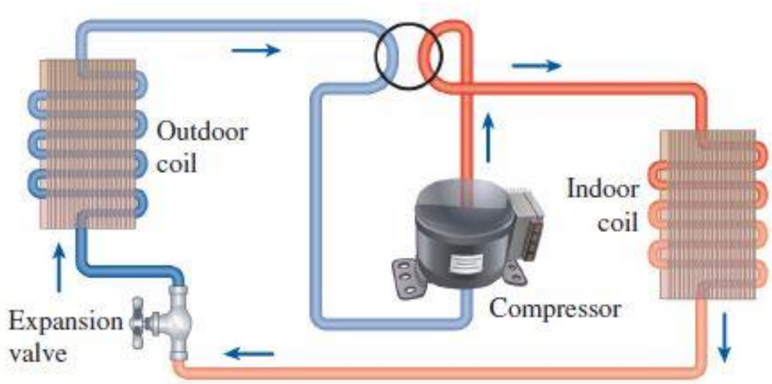

Figure 5. Reverse cycle a/c system operation – Heating mode (Cengel & Boles, 2015)

2.6 Thermal Energy Reservoirs

Cengel and Boles (2015, p. 277) describe a thermal energy reservoir as a 'body with relatively large thermal energy capacity (mass × specific heat) that can supply or absorb finite amounts of heat without undergoing any change in temperature'. Examples of such bodies are oceans, lakes and the atmosphere as megajoules of industrial by-products are dumped into these reservoirs without causing any notable change in the temperature. Thermal energy reservoirs come in two forms, which are heat sources and heat sinks. A reservoir that supplies energy in the form of heat is a source and a reservoir that absorbs energy in the form of heat is a sink (Cengel & Boles, 2015).

Air-conditioning systems, including heat pumps use ambient air as a thermal energy reservoir. Air-conditioners use outside ambient air at a higher temperature as a heat sink, whereby the system removes heat from the inside and transfers it to the warmer outside environment, maintaining a low temperature on the inside. Heat pumps conversely use the low temperature outside ambient air as a heat source by absorbing heat in the lower temperature environment and transferring it to the high temperature space on the inside of the building.

2.7 Modifications and Developments in Air Conditioning Systems

2.7.1 Cascade, Multistage and Gas Refrigeration Systems

Over the past few decades there have been many new developments and modifications of existing air conditioning and reverse cycle systems to improve efficiencies and increase performance. One example of these modifications is cascade refrigeration systems. These systems still use the vapour-compression cycle, however, they do so in two or more stages operating in series (Cengel & Boles, 2015) (refer to figure 6 showing a two-stage cascade system and the applicable T-s diagram). The stages of the system are connected through a heat exchanger which acts as a condenser for one stage and evaporator for the next stage. Each stage can use different refrigerants depending on the requirements of the system, as different refrigerants have different characteristics. The advantage of these systems is that it requires less compressor work and the amount of heat absorbed from the refrigerated space increases, thus increasing the coefficient of performance (COP). These systems are usually used in industrial applications where the temperature range and thus pressure range may be large, and efficiency is the main priority.

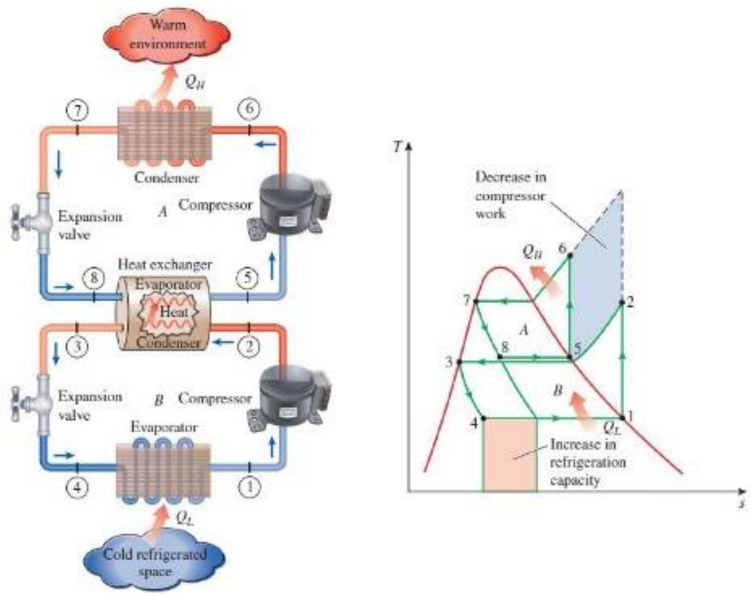

Figure 6. Two-stage cascade system and applicable T-s diagram
(Cengel & Boles, 2015)

Similar to the cascade refrigeration system, is the multistage compression refrigeration system referring to figure 7. If the same refrigerant can be used throughout multiple stages, instead of connecting the stages by a heat exchanger, a flash chamber can be used. The flash chamber is positioned between two expansion valves and the condenser and evaporator and has excellent heat transfer properties. A portion of the liquid refrigerant vaporises in the flash chamber forming a saturated vapour and is routed to and mixed with the superheated vapour from the low-pressure compressor. The mixture then enters the high-pressure compressor creating a regeneration process.

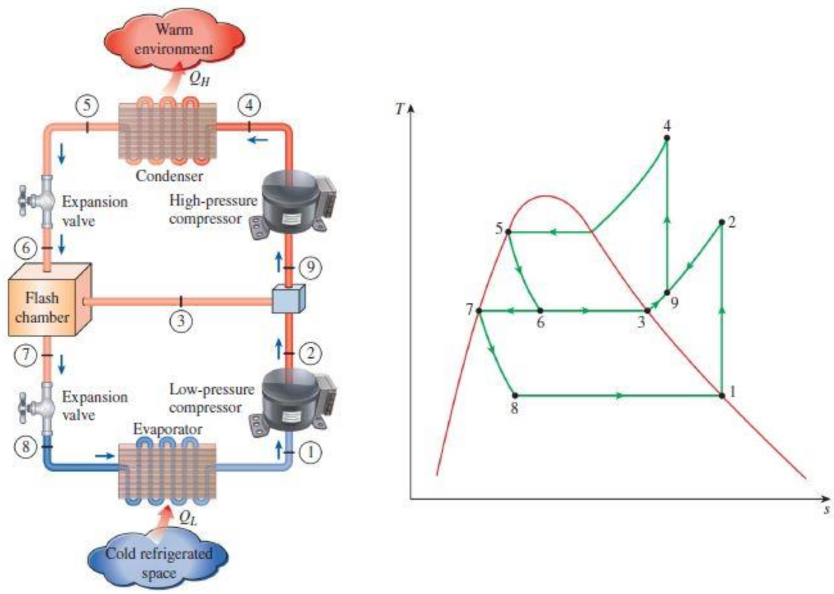

Figure 7. Two-stage compression refrigeration system with a flash chamber and applicable T-s diagram

Gas refrigeration cycles or the reversed Brayton cycle is similar to the reversed Carnot cycle, however, due to the heat transfer process not being isothermal these processes differ (Cengel & Boles, 2015). Additionally, this cycle is not practical in industrial and household applications as the COP is relatively low due to the temperature variation during the heat transfer process and is only suitable for aircraft cooling, due to the lightweight, simple and regenerative design.

2.7.2 Absorption Refrigeration and Solar Thermal Systems

Absorption refrigeration systems are becoming increasingly utilised and developed as the systems use naturally occurring solar, gas, waste heat and geothermal as a source of energy (Cengel & Boles, 2015). These systems function by absorption of the refrigerant by a transport medium, usually water. The refrigerant can range in type depending on the application such as ammonia or water and transport mediums such as lithium bromide or lithium chloride.

The most common absorption refrigeration systems have the same components as the vapour-compression cycle except the compressor is replaced by an absorption mechanism consisting of a generator, regenerator, rectifier, absorber, pump and expansion valve (refer figure 8 showing an ammonia (NH_3) absorption refrigeration system with solar as the thermal energy source). The sole purpose of the absorption mechanism is to raise the pressure of the refrigerant. The main difference and advantage of absorption systems over vapour-compression systems is that a liquid is compressed instead of a vapour. Also, the work input is very low as the heat transfer is from an external source, such as solar energy and a liquid is being compressed instead of a gas. These systems are very expensive, complex, require a large space and are less efficient (usually COP less than 1), however, if the cost of the thermal energy source is relatively low compared to electricity than the system could be considered for commercial and industrial applications.

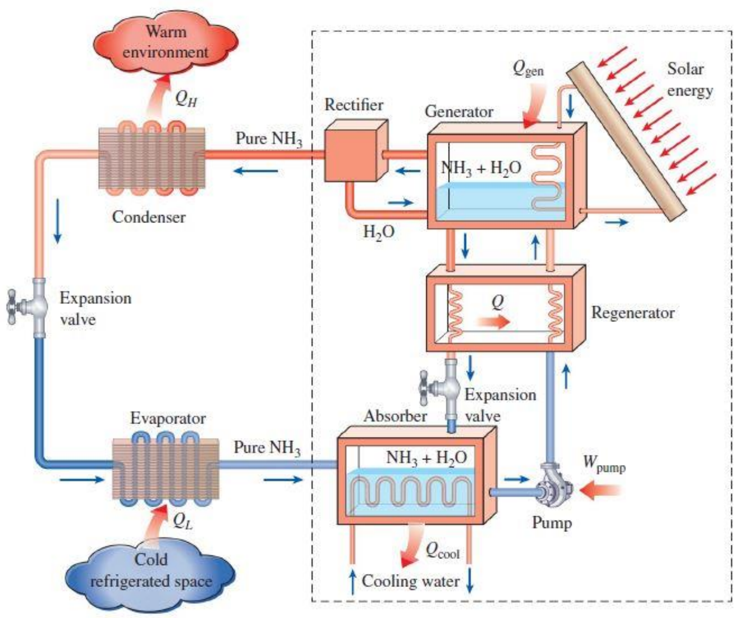

Figure 8. Ammonia (NH_3) absorption refrigeration system with solar as the thermal energy source (Cengel & Boles, 2015)

Referring to figure 9, the absorption cooling method via solar thermal technology is a process whereby the refrigerant (most commonly a LiBr/$H_2$0 or Ammonia- $H_2$0 solution) is evaporated in an evaporator, absorbed in the absorber and diluted (Siddiqui, M. U. and S. A. M. Said, 2015). This completes the pressurisation of the refrigerant by dissolving the refrigerant (H_2O) in the absorbent (LiBr). From here, the diluted solution in pumped into the generator but must be cooled to optimise the absorption process. The refrigerant is then condensed in a condenser and is fed back into the evaporator via an expansion valve.

The cooling effect from the system is then fed to either a chilling water flow or air flow, which passes through the evaporator (Solar Thermal Energy for Utilities and Industry, 2019). This is known as a closed continuous or intermittent operation system and the main benefit of the system is that the pressurisation of the refrigerant requires very little electrical power. Solar thermal collectors are used to regenerate the refrigerant vapour in the generator and can produce several hundred kW of cooling capacity (Al-Zubaydi, 2011).

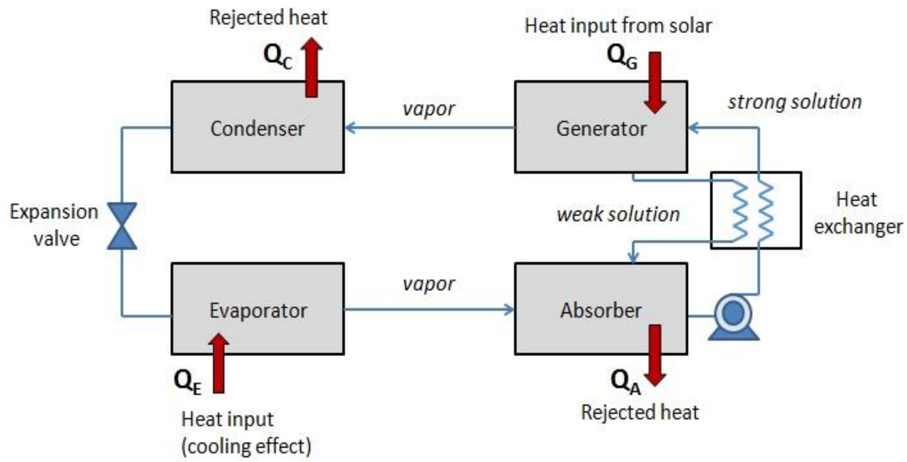

Figure 9. Basic solar thermal absorption cycle for cooling and refrigeration (Solar Thermal Energy for Utilities and Industry, 2019)

2.7.3 Solar Thermal Adsorption Systems

The adsorption process is similar to the absorption process, whereby it does not require a compressor (refer figure 10). While absorption is a chemical process which uses the attraction between a liquid and a surface, the adsorption cycle uses the attraction between a gas and a surface and uses an adsorption bed, a condenser and an evaporator and is usually a closed system but can be open. Referring figure 8, the process of adsorption involves the refrigerant solution boiling in the evaporator and the vapour produced during this phase is collected on the adsorption bed, which is made of Carbon (Kuri, et al., 2007). The refrigerant is depleted after a short time (usually occurring at the peak solar intensity when cooling is most required) at which point the adsorbent is heated by solar power and regenerated, thus removing the vapour, cooling the refrigerant and condensing it for the required purpose. Due to this regeneration cycle, the adsorption process is intermittent, which can be overcome with the use of multiple adsorption beds. The evaporation of the adsorbent occurs in the evaporator and the evaporator simultaneously absorbs the heat from ambient air, which causes the refrigeration process. This solar thermal cooling process is mainly used in refrigeration systems more so than air-conditioning systems and is usually not as efficient as absorption systems.

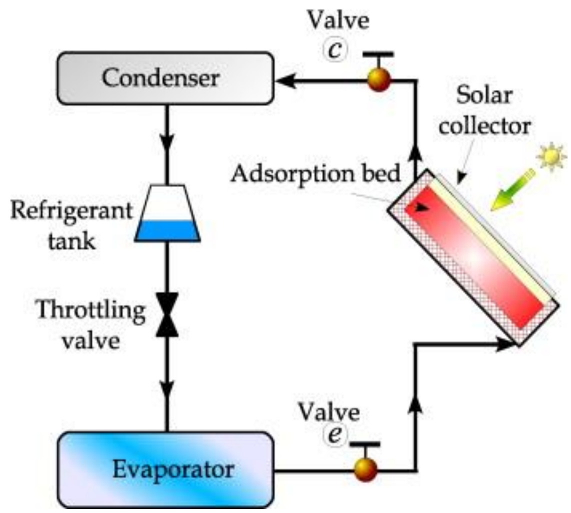

Figure 10. Basic schematic diagram of the solar adsorption cooling system (Hassan, H. Z. & A. A. Mohamad, 2012)

2.7.4 Solar Thermal Ejector Systems

Ejector systems in general are not a new technology, but with the integration of solar thermal energy to power and provide heat for the systems, the technology has seen interesting recent developments. Referring to figure 11, this process functions by a high-pressure vapour being produced in a generator, which is the primary fluid. The liquid refrigerant is evaporated by the heat source at state 5 and the primary fluid moves to the ejector at state 1, where it increases velocity and reduces pressure, in turn inducing the vapour in the evaporator (secondary fluid). Both the primary and the secondary fluids then mix in the mixing chamber creating the refrigerant fluid where it flows into the diffuser. Here the fluid’s velocity is reduced, and pressure increases once again. As the fluid enters the condenser at state 2, it rejects the heat and part of the liquid in the condenser is routed to the generator during state 3 and state 6 and the other half of the liquid is expanded through the expansion device during state 4 and enters the evaporator as a mixture of vapour and liquid. This mixture is then completely evaporated in the evaporator, which creates the cooling effect fluid (Al-Zubaydi, 2011). Solar thermal ejector systems generally have a low coefficient of performance depending on the temperature of the generator and condenser when compared with absorption and adsorption systems.

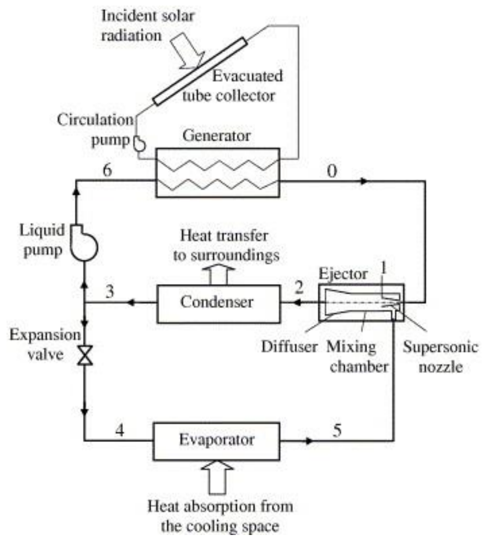

Figure 11. Schematic diagram of solar thermal ejector cooling system (Ersoy, H. K., et al., 2007)

2.7.5 Solar Thermal Desiccant Systems

Desiccant cooling systems are not a new technology much the same as ejector systems, however with the addition of solar power to run the system, these systems can be very effective under certain conditions. There are two types of desiccant systems, solid and liquid and each of these systems have several configurations. Al-Zubaydi, (2011) stated that this system "consists of a combination of drying process by absorption of water vapour in the air and evaporative cooling of water in the air to be treated. The drying is done on the surface of an adsorbent material such as Silica gel, activated Alumina, Zeolite, Lithium Chloride and Lithium Bromide".

Referring to figure 12 for the solid desiccant system, two revolving wheels including a dehumidification wheel and a heat exchange wheel make up the first stage of the system. Initially the warm ambient air from the environment passes through the dehumidification wheel, whereby the wheel rotates and absorbs the moisture in the air. The air is then heated as it passes through the heat exchange wheel, which is then pre-cooled. From that point, depending on the temperature and humidity requirements of the system, the air then passes through a humidifier and is supplied to the application. The return air is humidified until it reaches the saturation point to ensure maximum cooling potential and makes sure the heat exchanger operates efficiently. On this return cycle the air then passes through the heat exchange wheel, whereby the humid air is heated by a heating coil that is fed from the solar heated water or air. For solid solar thermal desiccant systems low temperature solar thermal collectors result in higher efficiencies and the high ventilation rates associated with these systems provides comfortable indoor conditions with respect to temperature and humidity.

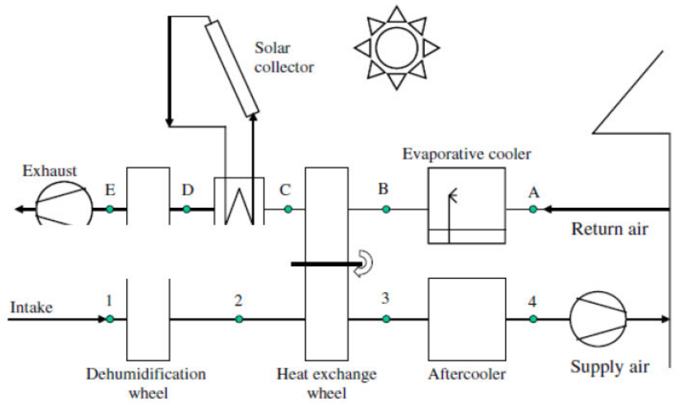

Figure 12. Schematic diagram of a solid desiccant cooling system with a solar collector (Al-Zubaydi, 2011)

Liquid desiccant cooling systems are a recent development in the industry and very similar to solid desiccant systems, although the desiccant is in liquid form, usually a lithium chloride solution. Referring to figure 13, the desiccant solution is circulated between the absorber and the regenerator and is sprayed over the cooling coil in the opposite direction to the return air (Al-Zubaydi, 2011). The desiccant air absorbs the moisture from the air and is cooler after passing the cooling coil. If further cooling is required at this point, an aftercooler is used to achieve the lower temperatures. The desiccant solution is sprayed over the heating coil against the flow of the ambient air, which rejects the water from the solution and reconcentrates it for the cooling cycle. The solar collectors are used here to heat the heating coil via air flow or water flow. Liquid desiccant solar thermal systems offer a higher level of dehumidification at the same temperatures when compared to the solid desiccant systems and have been most effective in hot, humid environmental conditions.

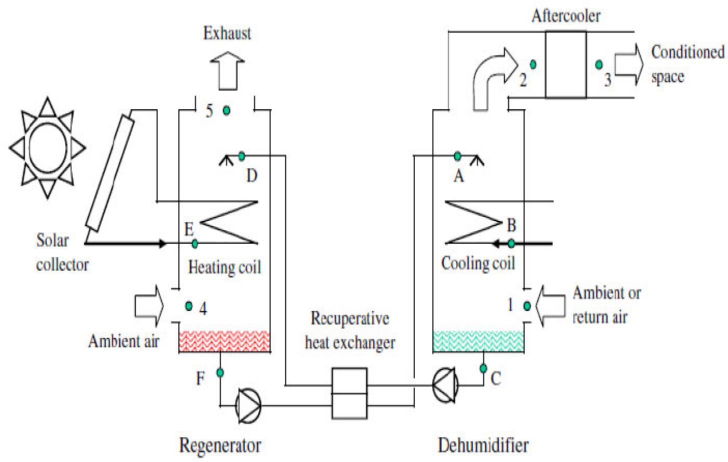

Figure 13. Basic schematic diagram of a liquid desiccant cooling system with a solar collector (Al-Zubaydi, 2011)

2.7.6 Energy Recovery Ventilators (ERV)

Energy Recovery Ventilators (ERV's) are devices engineered to transfer energy between the exhaust air of a commercial or residential building and the outdoor air supply to reduce energy consumption of the system by recycling/pre-conditioning the air, reducing load requirements and therefore requiring lower capacity a/c systems (Yang, et al., 2015). These devices recover otherwise wasted energy during the ventilation process in vapour-compression systems by removing humidity from the incoming outdoor air and transferring it to the exhaust air in summer and by removing humidity from the exhaust air and transferring it to the incoming outdoor ventilation air in winter (Hoger, 2009). It is important to note that an ERV can recover both sensible and latent heat energy and a Heat Recovery Ventilator (HRV) can recover sensible heat only. Latent heat energy is the energy absorbed or released from a substance during a phase change from gas to solid or liquid and vice versa and sensible heat energy is the energy required to change the temperature of a substance without a phase change (Latent and Sensible Heat, n.d.). ERV's are usually the preferred device due to the added benefit of humidity control and reducing indoor air pollution, which can enhance the quality of air of the conditioned space and improve work/living conditions.

There are several types of ERV's of which are selected on location and size of the a/c or HVAC system and the climate conditions in the region. Rotary (wheel) heat exchanger ERV's are the most popular type due to size, low cost, ease of maintenance and effectiveness. The device uses either a plastic or metal wheel that rotates between an exhaust and outdoor incoming air stream, transferring the heat from one air stream to the other (Hoger, 2009). The plastic wheels are usually made with an impregnated desiccant that can absorb and release moisture or latent energy, while the metal wheels can only transfer heat or sensible energy.

Plate (fixed core) heat exchanger ERV's utilise a larger footprint but have no moving parts. This type of ERV directs air though a series of channels, which heat or cool the fixed plate material and transfer the heat energy to the other air stream (Hoger, 2009). As with rotary ERV's, the metal core type can only transfer sensible energy, however, the plastic and paper types can transfer sensible and latent heat energy.

Heat-pipe heat exchangers are HRV's as only sensible heat can be transferred due to copper pipes being used containing refrigerant. These tubes are routed between the exhaust and outside incoming air streams, where one air stream heats the refrigerant, causing evaporation which is then routed to the other air stream which cools the pipe, condensing the refrigerant and warming the cooler air stream (Hoger, 2009). The system then routes the condensed refrigerant back to the warmer air stream to start the cycle again.

Run around Coils are also HRV's as only sensible heat can be transferred due to the water coil being unable to transfer latent heat. These systems have one water coil in the exhaust air stream and one water coil in the incoming outdoor air stream. The two coils are plumbed together and use either a water and/or glycol mixture. Like the heat-pipe, heat is absorbed in one air stream and transferred to the other (Hoger, 2009). These systems are usually used in large systems where the exhaust and outdoor incoming air streams are large distances apart.

Energy Recovery Ventilators offer a great energy benefit for several climates/environmental conditions and can further increase the efficiency of systems with large differences between indoor and outdoor air temperatures. These devices can be used in heating and cooling mode and could be used in conjunction with a system using water as a sink/source to increase the temperature differential of the refrigerant and sink/source and therefore further reduce energy consumption.

2.8 Performance Comparison of Solar Thermal Cooling/Heating Systems

The coefficient of performance (COP) of all the solar thermal technologies are relatively low when compared with current vapour-compression refrigeration systems on the market (Al-Zubaydi, 2011). The COP's are approximately 0.2-0.33 for ejector systems, 0.6 for adsorption systems, 0.86 for absorption systems and 1.2 for double stage absorption systems. Desiccant cooling and adsorption systems are currently quite expensive as well, which poses significant restrictions on wider commercialisation for these technologies. At this stage, absorption solar thermal systems seem to be the most efficient, effective and likely to penetrate the market of a wider scale, but significant progress and developments must be made to surpass the average COP of innovative vapour-compression systems of approximately 3.5.

2.9 Use of Water as a Sink in Air Conditioning Systems

Air conditioning systems that utilise water to increase the temperature difference between the refrigerant and the source and sink is a relatively new area of research, however, these designs have been used in some small applications since 2009, but not widely adopted. This is mainly due to the lack of research, documentation and standardisation of the systems. The main reason for using water as the sink or source is because the standard reverse cycle vapour-compression a/c system uses ambient air as a sink or source depending on whether cooling or heating mode is selected. Water storage devices such as a water tank or pool are generally at a lower temperature than the ambient temperature at any one time. From this, the work required to transfer the heat in a vapour-compression a/c system increases as the temperature difference between the cooler heat source and the warmer heat sink increases (Woolley, et al., 2010). Therefore, by using a water storage device to reduce this temperature difference could ultimately increase cooling/heating efficiencies and reduce the overall energy consumption.

2.9.1 Air Conditioning System Innovative Systems

Ma, et al. (2017) utilised an efficient water-cooled a/c using a heat pipe and bus-type water loop. In designing this system, low grade energy was used as the heat source or sink depending on the a/c setting by circulating water through the system in a bus-type setup to recover or recycle the internal heat otherwise lost to the atmosphere. The system combined a vapour-compression system with a separate heat pipe and provided a COP of 3.58 in heating mode and 3.65 in cooling mode, providing efficient and reliable performance. Other articles provided similar findings with using heat pipes to cool air prior to reaching the condenser with slight increases in COP and reductions in energy consumption.

Liu, et al. (2013) carried out a performance analysis on a multi-functional heat pump in cooling mode whereby four combinations of heat sink were studied. These combinations were using an air sink only, a water sink only, both air and water sink in parallel, and an air sink and water sink in series. The experiments were carried out at various ambient temperatures and found varying levels of cooling capacity and COP depending on the combination. The water sink used was a grey water tank and a hot water tank used to supply/supplement hot water to the household. The study found that the performance of the air sink and water sink in series was superior, however, the performance of all combinations reduced as ambient air temperature increased. This study was worth noting due to the findings of how performance of the system changed considerably depending on the temperatures of the water sink and air sink. For the proposed design, it may not be feasible to see efficiency gains in all seasons and environmental conditions.

2.9.2 Air Conditioners and Heat Pumps with Integrated Water Heaters

Ji, et al. (2003) published an article on the use of a water heater design in an air conditioner. This was achieved by immersing the condensing coils of the heat pump or air conditioner in a water tank, thereby the rejected heat from the system would heat up the water surrounding the condenser coils and supply hot water to the household, albeit supplementing primary hot water supply depending on demands. Due to the increased temperatures of the refrigerant in the condenser coils as the water temperature in the tank increases, the performance of the system was reduced. This was alleviated by connecting another air-cooled condenser in series after the immersed condenser coils. There have been several studies that investigate the incorporation of water heaters with air conditioning systems and all show that as water temperature in the tank increases, the performance and hot water making potential of the system decreases. A pool or water tank as the media for heat sink in a similar system to this via a pump and flowing water through an immersed tank surrounding the condenser coils could see further performance increases.

2.9.3 Swimming Pools as Heat Sinks for Air Conditioning Systems

Harrington, C & Modera, M (2013) suggest that using swimming pools as a sink for waste heat from air conditioning systems can reduce electricity consumption of the appliance by as much as 35% during peak demand. The paper also shows by using a swimming pool as a heat source for heating could increase the electricity saving for the appliance to 45%. These findings and energy savings are dependent on pool temperatures experienced throughout the year as well as the climate and location of the system, however, the use of a solar hot water system as a heat source for a heat pump during heating could be feasible to incorporate into this type of system. Harrington, C & Modera, M (2013) also found that by using a pool as a heat sink for air conditioners, the rate of evaporation of the pool water increased. The results found that for every degree Celsius increase in maximum hourly pool temperature there was a 2% increase in the rate of evaporation, therefore increasing water usage by at least 0.4 m^3 over the cooling season. However, once again this figure depended on climate, size of pool and whether the pool was located indoors or outdoors. Woolley, et al. (2010) also described this method of heat rejection from household cooling systems as a way to reduce or completely displace the requirement of direct pool heating, further reducing energy costs. This study validated a design model through experimentation on the thermal behaviour of pools as heat sinks, which included how several variables such as conductive heat exchange with the ground, radiative heat exchange with the sky, convective heat exchange with the air and pool shading would have an impact on the overall performance of the system.

2.9.4 Geothermal Heat Sinks in Air Conditioning Systems

The use of large bodies of water or the ground as a heat sink has been occurring for many years. Several powerplants around the world use lakes and oceans as a heat sink due to the lower than ambient temperatures experienced to reduce energy consumption and increase system efficiencies. Little research has been conducted into the efficiency comparisons between using ground coupled heat exchangers or geothermal heat pumps and conventional vapour-compression systems, however, these systems have been utilised increasingly in Europe and the US since the 1990's. Khedari, et al. (2001) performed experiments and provided comparison between a normal a/c system and a modified condenser a/c system. The modified condenser system consisted of a copper pipe of length 67m buried in a shaded area at a depth of 1.5m near a water reservoir to maintain moist soil. This required much more refrigerant but still reduced the energy consumption of the system due to the relatively constant temperature of the ground at that depth year-round. The COP calculated in this experiment was very high when compared to the normal a/c system, however, several considerations were raised in the conclusion of the article, which led to a less than specific outcome.

A more recent study by Speerforck, A & Schmitz, G (2016) used an open cycle desiccant assisted air conditioning system with three double U-tube borehole heat exchangers (BHX) at a depth of 80m to generate the cool water for the system. This study has caveats regarding the climate and soil conditions but finds the performance of the system is dramatically increased with the combination of a heat pump in the winter months, however, further economic evaluation would be necessary. The findings from these articles indicate using large thermal reservoirs such as water storage devices and the ground as heat sinks in a/c systems can produce large efficiency gains under the appropriate environmental conditions. Further research is required to isolate what these environmental conditions are and if the associated economic implications of using such a system in a residential household is feasible.

3. Methodology

3.1 Description of Works

The project was designed with six phases of work throughout the year. Each phase had defined tasks and goals to achieve within the given timeframe as per the project schedule and a phase review that included consultation/feedback from the project supervisor and any further work to complete the phase requirements. Phase 1 involved collecting data from the start of semester 1 on ambient air temperatures, pool temperatures, hot water system temperatures and household a/c refrigerant temperatures, which continued on a daily basis throughout the project year. Phase 2 was the design stage whereby a way to test the use of the heat sinks and sources to check for efficiency gains was researched and implemented. Phase 3 was testing the chosen design and collecting data for calculations and comparison to see where in the system would see power reductions depending on whether hot packs or cold packs were used and at what location these were applied. Phase 4 was focused around performing calculations utilising MATLAB to iterate and find refrigerant temperatures, mass flow rates and heat transfer at the required points in the household and box a/c system. Phase 5 was the comparison stage at which the results of the calculations on the household system were compared to the findings and calculations on the box a/c system. Phase 6 was the final stage where the findings from the testing, calculations and comparison were assessed for feasibility in application for residential use and determining if additional energy/capital costs were associated with incorporating such a design on a residential scale.

3.1.1 Phase One – Data Collection

Phase One of the project involved taking temperature and power measurements of a household reverse cycle ducted air-conditioner to ascertain a baseline for refrigerant temperatures, heat sink (residential pool) and heat source (solar hot water system) temperatures and power consumption of a standard household air-conditioner. The household air-conditioner was a 20kW Ducted Inverter Reverse Cycle a/c model numbers: Outdoor unit - RZQ200LY1 and Indoor unit - FDYQN200LBV1, utilising 8.3kg of R410A refrigerant. The rated capacity for this unit was 20kW in cooling and 22.4kW in heating.

The temperature measurements were taken at the compressor inlet and outlet on the unit with an infrared thermometer that was accurate to ±0.2°C (refer figure 14). The power measurements were taken using a residential smart meter (refer figure 15 - EDMI Atlas Series Mk10D meter) on the house and performing the appropriate calculations for power (refer table 3 for excerpt of power readings and Appendix C). The household utilised 3-phase power, so the appropriate 3-phase formula for power was used: $P = I \times 3^{1/2} \times PF \times V$, where I = current in amps, PF = power factor and V = volts. These measurements could be used to find mass flow rate and temperature of the refrigerant for the system and be used later in the project to compare efficiencies to the designed system utilising the heat sink/source and extrapolated data for the household a/c system. The heat sink for the project was a residential pool as a water tank was not available and the heat source was a residential solar hot water system (although on cooler and overcast days, the electric booster was required to supplement the solar heating).

Figure 14. Infrared thermometer used for temperature measurements (eBay, 2020)

Figure 15. EDMI Atlas Series Mk10D meter (EDMI Limited, 2010)

3.1.2 Phase Two – Design Options

Phase two involved designing an air conditioning system that utilised the pool and hot water system as a heat sink/source to increase the temperature difference between the refrigerant in the air conditioning copper lines and the source/sink. The hypothesis and knowledge gap to be proven was that by applying the heat sink/source temperature to the refrigerant lines in a precise location of the vapour-compression cycle, the temperature difference between the refrigerant and the source/sink would increase, thus increasing the heat transfer rate, reducing the mass flow rate, and therefore reducing the overall power consumption of the compressor. The general goal of the experiment being to reduce the amount of heat energy required from the compressor and thus a reduction in the power consumption for the air conditioner to operate effectively and efficiently.

There were two options for the design during phase two of the project and these were investigated for feasibility before going ahead with the building/testing. The first option was using a car air conditioning system as the basis of the design. The system was to be either removed from a vehicle or parts procured from a wreckers to build the system separate from the vehicle. Using a water pump and insulated thin flexible lines routed around the refrigerant lines of the a/c system to the heat sink and source, the goal was to actively transfer heat either from the refrigerant lines to the heat sink for cooling mode or transfer heat from the heat source in the flexible lines to the refrigerant in the lines for heating mode. Being a 12V system, a 240V electric motor would had to have been acquired to drive the compressor of the system via a belt to create the flow of refrigerant through the system and a 12V battery used to control the thermostat or temperature of the system and change the mode from heating to cooling.

The second design option was to use a reverse cycle wall/box air conditioner (a/c) to perform the same procedure using a water pump and insulated thin flexible lines around the a/c refrigerant lines. The design would be simplified through one 240V power source and one compact system, using a standard power meter to measure the power consumption of the unit with and without the heat sink/source applied. As with the original measurements for the household air conditioner, the temperature of the refrigerant lines would need to be measured at the required locations in the system as well as a mass flow rate and temperature of the refrigerant calculated. After testing in a variety of environmental conditions and seasons, a comparison could be made between the designed systems efficiency with the heat sink/source and the household system with calculations made to extrapolate the energy consumption and mass flow rate with the heat sink/source temperature utilised on the system.

3.1.3 Design Option Constraints and Selection

The feasibility assessment of the designs was time consuming and challenging, requiring thorough research and information from subject matter experts. The vehicle air conditioning system design was troublesome in a variety of ways. The accessibility to a wreckers that would allow this sort of removal was an initial concern, which included the time-consuming nature of the removal. A car air conditioning system is routed throughout the engine bay with rigid and flexible lines and travels through the firewall of the engine to the cabin area where the evaporator and thermostat are located. Concern laid with the serviceability of the system prior to removal; being at a wreckers for an unknown length of time and with the possibility of releasing or disturbing the refrigerant in the lines inadvertently, causing HFCs to be released into the atmosphere and a possible WHS risk associated with a pressurised gas. Further issues surrounded the power source of the car system as to effectively compare the design and the household system; both would need to be on a 240V system. As discussed briefly in Section 3.1.2, this could be possible, but may cause issue with other components in the car a/c system, being originally designed for 12V power.

Design option two was troublesome initially as acquiring a reverse cycle box air conditioner in the modern day was quite difficult. One was eventually sourced; however, it did not operate. Tests were performed using a multimeter to find the source of damage/failure, which showed an issue with the main printed circuit board (PCB). Another a/c was eventually procured, which operated well in both cooling and heating mode. The item was a Fujitsu Reverse Cycle a/c model number AKT9RGS-W and operated using 680 grams of R22 refrigerant. It had a rated capacity of 2.5kW in cooling and 2.8kW in heating. This design option implemented the use heat packs and cold packs at an appropriate temperature to simulate the heat sink and source temperature around the refrigerant lines of the designed system followed by performing the applicable measurements and calculations. The hot/cold packs did not flow as such, so the temperature of the packs changed quite rapidly, however, served a similar purpose to the use of a water pump routed to the heat sink (residential pool) and source (solar hot water system).

Due to the problems with design option one highlighted above, the safest, most ethical and practical design option chosen was design option two utilising the reverse cycle wall/box air conditioner. Furthermore, utilisation of the hot and cold packs to simulate the heat sink/source temperature was chosen with this design for testing and data collection.

3.1.4 Phase Three – Testing Design/Data Collection

Phase three involved testing the reverse cycle box air conditioner utilising the hot and cold packs to simulate the temperature of the heat source/sink. The initial testing involved a trial and error approach to find at which points throughout the vapour-compression cycle of the box air-conditioner would see power reductions depending on whether hot or cold packs were used and in what mode the a/c was operated. The packs were directly applied to the external surface of the refrigerant copper pipe in various locations with the aim of seeing an instantaneous power reduction reading on the power meter. Once an appropriate location was found for both heating and cooling mode using either the hot or cold pack, measurements were taken for refrigerant pipe temperatures, sink/source temperatures, ambient air temperatures, and power consumption before and after applying the sink/source. These measurements and data collection were conducted each day after the design was setup to provide a reasonable dataset to ensure effective and viable results. Further information on the measurements and data collection can be found in Section 3.2.

3.1.5 Phase Four – Calculations/Extrapolation

With the data collected from both the household air-conditioner baseline measurements and the box air-conditioner design measurements using the hot/cold packs to simulate the heat source/sink, several calculations were required to find refrigerant temperatures, mass flow rates and the heat transfer at the required points in the systems. Phase Four of the project utilised Microsoft Excel initially for these calculations, but it was soon established that due to the large datasets, the calculations needed to be performed in MATLAB using a program to iterate the solutions for the results to be acquired in a reasonable timeframe. Refer to Section 3.4 for further information on the calculations.

3.1.6 Phase Five – Comparisons

Comparing the results of the calculations of the household a/c system to the findings and calculations on the box a/c system was the basis of Phase Five. After the household a/c system calculations were extrapolated using MATLAB with the heat sink and source temperatures included, a comparison with the extrapolated data/results from the household a/c system and the results from the box a/c system was performed. These comparisons would be graphed and analysed in detail to see how close the extrapolated measurements/calculations on the household a/c system were to the measurements taken by directly using the heat sink/source on the box a/c system. The hypothesis was that if the calculations were similar to the measurements, it would show a good indication that the extrapolation is valid. If the calculations differed substantially from the measurements, the extrapolation may not be valid, however, the trends of the behaviour of the system (by changing a controlling parameter) would have been established. Results of the calculations and comparisons can be found in Section 4.

3.1.7 Phase Six – Feasibility Assessment

Assessing the feasibility of using such a design on a household system and determining any additional energy or capital costs with the design was the purpose of Phase Six. Based off the outcome of the Phase Five results and comparisons, a feasibility assessment was carried out to determine if the chosen design would be viable for residential use. Further work will be discussed in Sections 4 and 5 to ascertain if using alternate methods or wider ranging testing conditions could increase efficiencies of such a design and provide maximum energy and capital savings in the future.

3.2 Data Analysis and Measurements

3.2.1 Baseline Data Collection – Household a/c, Hot Water System, Pool Temperature

During the initial phase, the data collection and measurements taken were the most important part of the project. Ensuring accuracy and consistency with these measurements was paramount. The household solar hot water system and air-conditioner model was researched thoroughly to ensure the data captured could be used effectively (refer table 1 showing household air-conditioner specifications). Furthermore, the smart meter on the house was researched as accessing the appropriate menu to find instantaneous voltage, amps, power factor and net usage was not known initially. The smart meter was an EDMI Atlas Series Mk10D as discussed in 4.1 and the user manual was consulted on how to access the menu to cycle through the required data.

Table 1: Household Daikin ducted air-conditioner specifications from where data is being collected

20Kw Standard Ducted Inverter Reverse Cycle Air Conditioner		cooling	heating
Model: outdoor unit - RZQ200LY1	power input (rated)	6.44kW	7.00kW
Model: indoor unit - FDYQN200LBV1			
R410A			
3 Phase, 415v, 50Hz	high press side	4.0MPa	
Mass of refrigerant (outdoor unit)	low press side	2.7MPa	
8.3kg	**E.E.R/C.O.P**	3.11/3.20	
	Airflow Rate (Rated)	1200	
Improved Energy Efficiency			
Achieved through the use of a DC Fan motor, cross-pass heat exchanger and increased outdoor coil passes.			
	Compressor Type	Hermetically Sealed Scroll Type	
	Piping sizes	Refer to link	
Rated Capacity	Piping length	50m	
Cool - 20.0 kW			
Heat - 22.4 kW			
Capacity Range			
Cool - 12.0-20.0 kW			
Heat - 13.4-22.4 kW			
The Rated Capacity, Power Input and Running Current are measured in accordance with AS/NZS 3823.1.2			

3.2.2 Baseline System Measurements – Household Air Conditioner

Measurements were taken systematically from the 27th of January 2020. These measurements included pool water temperature, maximum local ambient air temperature, minimum local ambient air temperature, hot water system outlet temperature (out of hot water system and into house), hot water system inlet temperature, air-conditioning refrigerant exit temperature (external temperature of copper pipe after compressor), air-conditioning refrigerant inlet temperature (external temperature of copper pipe before compressor) and power consumption (calculated from Mk10D meter data). Power consumption data collection was delayed until the 6th of March due to challenges surrounding the Mk10D interface and 3-phase power as discussed in 5.1.1. This was tracked on two spreadsheets with the power measurements spreadsheet utilising the applicable formulas to convert and calculate the correct power for a 3-phase system. See tables 2 and 3 for a small excerpt of these measurements (refer to 7.4 Appendix C – System Measurements).

Table 2: Excerpt of the baseline temperature measurements taken

Temperature Measurements of Pool, Air, Hot Water System and Air-Conditioning System									
Date	Water Temp	Max Air Temp	Min Air Temp	HW out	HW in	AC out	AC in	Power Use (kW	Comments
09-May-20	22.5	27	11	37	25	27.5	3.5	5.59	
10-May-20	22.5	29	10	43	24.5	25	4.5	6.35	
11-May-20	22	24	8	38	23	21.5	6.5	3.88	
12-May-20	21	25	8	48	24	18	5	4.46	elec. HW on (manual)
13-May-20	21.5	25	9	43	23	22	4.5	5.78	
14-May-20	21.5	25	9	35.5	21.5	17.5	15.5	3.91	
15-May-20	21	24	9	48	24	21	11.5	3.7	elec. HW on (manual)
16-May-20	21	25	9	49	32	19	13.5	3.84	
17-May-20	21.5	25	10	40	26	24	10	3.75	
18-May-20	21.5	25	9	40	21.5	18	12	2.46	

Table 3: Excerpt of the baseline power measurements taken

Daily Power Measurement and Calculations						
Prior to turning a/c ON - No other household items ON (except in standby) - NOT including 1 x fridge, 1 x bar fridge and computers x 2.						
	Voltage (V)	Current (I)	Power Factor (PF)	Instantaneous Net Usage (kWh)	Power per Phase (W)	Total Power - (kW) (sum of phases)
L1	239	1.18	0.23	0.14	112.35	0.25
L2	240	0.49			46.85	
L3	238	0.91			86.28	
AC ON - 5mins into operation - No other household items ON (except in standby) - NOT including 1 x fridge, 1 x bar fridge and computers x 2.						
	Voltage (V)	Current (I)	Power Factor (PF)	Instantaneous Net Usage (kWh)	Power per Phase (W)	Total Power - (kW) (sum of phases)
L1	238	2.74	0.79	1.46	892.31	2.71
L2	239	2.84			928.76	
L3	237	2.73			885.32	
Difference between Total Power PRIOR to turning AC ON and Total Power WHILE AC is ON = AC Power Draw/Usage/Consumption						
			AC Power (kW)			
			2.46			

Measurements were not taken during the shoulder seasons: 19/05/20 – 14/06/20 and 26/08/20 – 30/09/20 as the efficiency of the a/c system was naturally high as the system did not have to do much to reach the set temperature. For example, the target conditioned temperature (inside the house) was no more than a few degrees different to the sink/source temperature (outside the house), depending on heating or cooling mode. Therefore, the proposed modifications of the designed air conditioning system are likely to consume more power than they will save if implemented on a residential household system during the shoulder seasons. The measurements resumed on the 15th of June when the household reverse cycle air conditioner was set to heating mode, however, did not resume after the 26th of August due to time constraints with residential schools, full-time work and other unit course work.

3.2.3 Box Air-Conditioner Data Collection

During the design and testing phase of the project the reverse cycle box a/c was researched to ascertain the specifications of the unit required for testing and further calculations (refer table 4 showing Fujitsu box a/c specifications). The temperature of the heat source (solar hot water system) and heat sink (pool) on the day was used for the hot pack and cold pack temperature respectively by heating the pack up in the microwave or cooling it down in the fridge to the required sink temperature (refer figure 16 showing the hot/cold packs).

Table 4: Fujitsu box air-conditioner specifications

FUJITSU Reverse Cycle Box Air-Conditioner		Cooling	Heating
Model: AKT9RGS-W	power input (rated)	1kW	0.98kW
Serial: T001871	Amps	4.4	4.4
R22			
240V 50Hz Plug	**E.E.R/C.O.P**	2.55/2.91	
Mass of refrigerant (outdoor unit)			
680 grams			
Rated Capacity			
Cool - 2.5 kW			
Heat - 2.8 kW			
	Compressor Type	Reciprocating	
Noise			
69dBA			
The Rated Capacity, Power Input and Running Current are measured in accordance with AS/NZS 3823.1.1 Table 1			

Figure 16. Hot/cold pack used to simulate the water source/sink temperature (eBay, 2020)

The power consumption data was collected prior to applying the heat sink or source and 30-60 seconds after applying the heat sink/source to the required refrigerant pipe. The power consumption of the reverse cycle box air-conditioner was measured using a 240V power meter (refer figure 17 displaying the power meter used).

Figure 17. Power meter used to measure the power consumption on the box air-conditioner

3.2.4 Box Air-Conditioner System Measurements

Measurements were taken on the box air-conditioning system from the 12th of August on several of the refrigerant lines within the unit, as well as the temperature of the sink or source, the maximum and minimum ambient air temperatures of the day and the power use of the unit before and after application of the hot or cold pack, simulating the sink or source temperature. One spreadsheet with two sheets was used for tracking this data, one for the heating mode of the reverse-cycle air-conditioner and one for the cooling mode of the reverse-cycle air-conditioner (refer table 5 and 6 for an excerpt of the temperature and power data collected for both cooling and heating mode).

Table 5: Excerpt of the box a/c temperature and power measurements in cooling mode

Date	Temp of Sink	Max Air Temp	Min Air Temp	Comp out temp	Comp in temp	Condenser (outdoor coil) in temp	Condenser (outdoor coil) out temp	evaporator (indoor coil) in temp	evaporator (indoor coil) out temp	Power Use before adding sink (kW)	Power Use after adding sink (kW)	Comments
12-Aug-20	43	24	8	46		40	26	1.5	9.5	0.834	0.811	Heat pack applied to compressor in
17-Aug-20	45	23	5	46	16	45	23	33	2	0.815	0.768	Heat pack applied to compressor in
18-Aug-20	46	25	10	45	14	37	24.5	28	2.5	0.864	0.802	Heat pack applied to compressor in
19-Aug-20	52	27	8	46	11.5	34	25.5	25	3.5	0.887	0.832	Heat pack applied to compressor in
20-Aug-20	44	23	7	44.5	14	32.5	24	21.5	1.5	0.827	0.796	Heat pack applied to compressor in
21-Aug-20	46	23	8	45	16	29	22	20	-1	0.81	0.793	Heat pack applied to compressor in
22-Aug-20	51	22	5	46	15	27	23	32	8.5	0.831	0.798	Heat pack applied to compressor in
23-Aug-20	46	21	5	46.5	17.5	26	21.5	29	10	0.845	0.809	Heat pack applied to compressor in
24-Aug-20	43	22	3	48	20	25	19.5	27	11.5	0.852	0.805	Heat pack applied to compressor in
25-Aug-20	41	22	4	56	11	43	29	32	8.5	0.945	0.849	Heat pack applied to compressor in

Table 6: Excerpt of the box a/c temperature and power measurements in heating mode

Date	Temp of Sink	Max Air Temp	Min Air Temp	Comp out temp	Indoor Coil in temp (condenser)	Indoor Coil out temp (condenser)	Outdoor Coil in temp (evaporator)	Outdoor Coil out temp (evaporator)	Power Use before adding sink (kW)	Power Use after adding sink (kW)	Comments
12-Aug-20	18.5	24	8	61.5	37	50	16.5	4	1.601	1.561	Cold pack applied to compressor outlet
13-Aug-20	19.5	27	8	65	35	58	18	6.5	1.716	1.682	Cold pack applied to compressor outlet
14-Aug-20	19	27	12	63	32.5	62	22	13.5	1.81	1.714	Cold pack applied to compressor outlet
15-Aug-20	20.5	23	8	62	35	63.5	24	14.5	1.674	1.606	Cold pack applied to compressor outlet
17-Aug-20	19.5	23	5	63	33	55.5	33	16	1.61	1.514	Cold pack applied to compressor outlet
18-Aug-20	20	25	10	71	36	61	32	18	1.703	1.645	Cold pack applied to compressor outlet
19-Aug-20	19.5	27	8	89	42	74	31	21	1.597	1.581	Cold pack applied to compressor outlet
20-Aug-20	19.5	23	7	86	38	68	29	16	1.622	1.574	Cold pack applied to compressor outlet
21-Aug-20	20.5	23	8	85	40	70	28.5	17.5	1.789	1.644	Cold pack applied to compressor outlet
22-Aug-20	17	22	5	87	39.5	71	30	16.5	1.602	1.514	Cold pack applied to compressor outlet
23-Aug-20	17	21	5	86.5	42	68.5	29	17	1.541	1.44	Cold pack applied to compressor outlet
24-Aug-20	17.5	22	3	86.5	32.5	42.5	53	25.5	1.751	1.718	Cold pack applied to compressor outlet
25-Aug-20	20.5	22	4	90	42	70	69	21.5	1.602	1.569	Cold pack applied to compressor outlet

Once again, measurements were not taken during the shoulder season of: 26/08/20 – 14/09/20 as the efficiency of the a/c system was naturally high. As measurements were performed in both heating and cooling mode during the winter August period in Ipswich, QLD, the results could be vastly different if performed in the warmer months of the year or in a different location. This limitation with the testing is discussed further in Section 3.3.

3.2.5 Errors/Assumptions – Data Collection and Measurements

The measurements had several assumptions made to account for small errors in the data collected. The main error laid with the power measurements. For this to be accurate, all other power sources in the household would need to be off prior to taking the measurements. The was not practical, as during the data collection period, full-time work was being conducted from home using lights, fans and computers. In addition, the household utilised one large kitchen fridge and a small bar fridge, which could not be turned off during the measurements. Therefore, a baseline power load was identified. To ensure the measurements were as accurate as possible, a data set for the power was taken before turning the air-conditioner on, with minimum household power on and then another data set taken 5 minutes after the air conditioner was turned on. The baseline power load was then subtracted from the power load with the a/c on, which provided an instantaneous power consumption for the air conditioner. The measurements were also taken at the same time each day for continuity at approximately 4pm (refer table 3).

Further issue with the power measurements was the instantaneous nature of the readings. These readings were constantly fluctuating, although minor in most cases. It was noted that during start-up the a/c would draw much more power and be constantly increasing/fluctuating as opposed to 5 minutes into operation when the readings had stabilised. Thus, this was chosen as an appropriate time to perform the measurements. A similar issue was found when taking temperature measurements of the refrigerant lines. If a delay of approximately 5 minutes was adhered to, the refrigerant temperatures had almost stabilised to achieve an accurate reading to ±0.2°C. The hot water system temperature measurements were erratic due to the solar aspect. The solar irradiance on the day, meant this temperature reading could fluctuate significantly and if it dropped below a certain temperature, the electric booster was required to supplement the solar energy source and provide sufficient hot water for the household.

3.3 Limitations/Constraints with Testing the Design

It was known from the start of the design testing that there were going to be limitations and constraints with the chosen design, which could be alleviated given more time, money and resources. However, due to the undergraduate nature of the project, the limitations and constraints were known and accounted for as well as possible.

The hot/cold packs used to simulate the solar hot water system water temperature and residential pool water temperature were approximately 270mm in length by 125mm in width. This proved to be a problem when applying the packs to the refrigerant copper pipe of the box air-conditioning system as the full length of the copper pipe could not be covered with the pack and therefore the heat transfer would not be as effective.

The packs had a slightly varied temperature at different points on the pack. This may have been due to the chemical properties of the fluid inside the pack, the pack exterior material or simply due to the size of the pack not heating and cooling with an even distribution of temperature. This was significantly more noticeable when heating the packs up. During this activity every effort was made to move the fluid around to evenly distribute the heat within the pack. Related to this limitation on the packs was that the packs would rapidly change temperature once applied to the copper pipe and therefore the effectiveness and validity of using the heat sink or source temperature was reduced very fast. Due to this constraint, measurements were taken for power consumption within 30-60 seconds of applying the pack.

Using a box air-conditioner presented some further difficulties in that the unit was small and compact. Approximate size in length by width by height was 530mm x 420mm x 340mm. The access area to the refrigerant pipes was even smaller and the condenser fan rotating at high rpm in the middle of the unit increased the difficulty of the task. The location of box a/c was outside on the patio and the unit was not fitted in a wall or window due to access issues for taking measurements, applying the hot/cold packs and nowhere in the premises to fit the unit. This was a substantial issue because the heating and cooling of the unit would not be as effective as the indoor coil space was the same as the outdoor coil space. Finally, environmental issues included only being able to perform the measurements in heating and cooling mode between July and September in the winter months, where significantly different results could occur in the hotter and/or more humid months of the year.

3.4 Mass Flow Rate of Refrigerant and Temperature Losses through Pipes

3.4.1 Mass Flow Rate

As defined by Cengel & Ghajar (2015, p. 12) mass flow rate is ‘the amount of mass flowing through a cross section of a flow device per unit time and is denoted by $\dot{m}$’. For mass flow rate of a fluid through a pipe in a controlled volume such as an air conditioning system, the system is said to be steady state. This meaning that the total change in energy for the control volume is equal to zero, or the energy entering the control volume is equal to the energy leaving the system. Furthermore, the flow of fluid in a pipe is approximated to be one-dimensional, as all properties vary in one direction only; this being the flow direction. Therefore, at any cross section normal to the flow direction, these properties are uniform. From this information, it can be said that the mass flow rate of a fluid in a pipe is proportional to the density of the fluid, the average velocity of the fluid and the cross-sectional area of the pipe ($\dot{m} = \rho VA_c$) in kg/s.

It is also worth noting that the rate of net heat transfer into or out of the controlled volume is proportional to the mass flow rate, the specific heat and the change in temperature of the fluid ($\dot{Q} = \dot{m}c_p\Delta T$). Determining the mass flow rate of the system will ultimately assist in finding the net heat transfer of the reverse cycle air conditioning system and the coefficient of performance for both air conditioning mode and heat pump mode ($COP_R = \frac{Desired\ Output}{Required\ Output} = \frac{Q_L}{W_{net,in}}$ or $COP_{HP} = \frac{Desired\ Output}{Required\ Output} = \frac{Q_H}{W_{net,in}}$) if required.

3.4.2 Temperature Losses through Pipes

In an a/c system, heat is always lost through the wall of the pipe. This was important to account for as temperature measurements for the experiment had been taken using an infrared thermometer directed at the outside of the copper pipe. The temperature loss needs to be found in order to find the temperature of the refrigerant inside the copper pipe. This heat loss/heat transfer through the pipe wall is said to be steady state and one dimensional as stated in Section 3.4.1. In addition to the heat transfer being steady and one dimensional, other assumptions during these calculations are that the thermal conductivity remains constant and there is no heat generation. Therefore, as the heat transfer occurs in the normal direction or radial direction to the pipe only, the wall thickness of the pipe must be accounted for by calculating the thermal resistance of the cylindrical layer against heat conduction (refer to R_{cyl} below). Furthermore, in calculating the thermal resistance of the entire system, both forced and natural convection must also be considered as shown below.

The following information was used on calculating pipe losses shown below from Cengel & Ghajar (2015):

- $Q_{cond,\ cyl} = Q_{conv,1} = Q_{cyl} = Q_{conv,2}$

Where:

- $Q_{cond,\,cyl} = \frac{T_{\infty1} - T_{\infty2}}{R_{total}}$
- $Q_{conv,1} = \frac{T_{\infty1} - T_1}{R_{conv,1}}$
- $Q_{cyl} = \frac{T_1 - T_2}{R_{cyl}}$
- $Q_{conv,2} = \frac{T_2 - T_{\infty2}}{R_{conv,2}}$

$R_{total} = R_{conv,\,1} + R_{cyl} + R_{conv,\,2} = \frac{1}{(2\pi r_1 L)h_1} + \frac{(\ln(r_2/r_1)}{2\pi Lk} + \frac{1}{(2\pi r_2 L)h_2}$; and,

$R_{cyl} = \frac{\ln(r_2/r_1)}{2\pi Lk}$; r_2 = outer radius ; r_1 = inner radius ; k = thermal conductivity ; L = length ;

area = A = 2πrL [where there is a specified inner and outer surface temp]

- Therefore, change in pipe temperature from the inside to the outside of the pipe is: $\Delta T_{pipe} = QR_{cyl}$

In practice, the natural convection heat loss needs to be accounted for with the pipes of an air conditioning system. This form of heat loss to the surroundings is equal to the conductive heat loss through the pipe, which is equal to the forced convection heat loss from the refrigerant to the inner surface of the pipe. Having the measurement of the pipe outer temperature and the ambient temperature will enable a calculation of the pipe inner temperature, and following this, iteration through MATLAB will be required to find refrigerant fluid temperature, mass flow rate and final heat transfer.

For these calculations, further research was conducted on natural and internal forced convection, once again utilising Heat and Mass Transfer: Fundamentals and Applications (Cengel & Ghajar, 2015), Chapters 8 and 9. *For natural convection calculations, in addition to the above formulae, the following was used:*

- Properties of air (table A-15) were interpolated to find thermal conductivity (k), kinematic viscosity (v) and the Prandtl number (Pr).
- The Rayleigh number could then be calculated using $R_a = \frac{g\beta(T_s - T_\infty)D^3}{v^2}\,Pr$; followed by,

- The Nusselt number using $Nu = \left\{0.825 + \frac{0.387 Ra_L^{\frac{1}{6}}}{\left[1+\left(\frac{0.492}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right\}^2$
- The heat transfer coefficient using: $h = \frac{k}{D} Nu$
- Using the above formulas for thermal resistance and natural convection for: $R_{conv,\,2}$, $Q_{conv,2}$, R_{cyl}, Q_{cyl} and T_1 (temp of inside wall of pipe).

For calculations relating to the internal refrigerant temperature and heat transfer, internal forced convection formulae were required from chapter 8 as follows:

- The Reynolds number using $Re = \frac{V_{avg}\,D}{D} = \frac{\rho\, V_{avg}\, D}{\mu} = \frac{\rho D}{\mu}\left(\frac{\dot{m}}{\frac{[\rho\,\pi\,D^2]}{4}}\right) = \frac{4\dot{m}}{\mu\,\pi\,D}$
- Determination of Laminar or Turbulent flow based of Reynolds number
- Using the applicable Nusselt number formula and finding h, $R_{conv,\,1}$ and $Q_{conv,1}$.

For calculations with copper pipes, the following information was sourced from Heat and Mass Transfer: Fundamentals and Applications: 5th edition in SI Units, (Cengel & Ghajar, 2015)(refer table 7 for details on the pure copper properties).

Table 7: Pure Copper properties from Heat and Mass Transfer textbook

Copper [Pure] Pipe Properties (Table A-3) - @ 300K (27°C)					
	ρ (kg/m$^{3)}$	**c_p (J/kg.K)**	**k (W/m.K)**	**α x10^6 (m^2/s)**	**ε**
Copper (Cu) Pure	8933	385	401	117	0.02 – highly polished

Upon further investigation, a more accurate source of copper properties was found for household air-conditioning, refrigerant and hot water system copper pipes. The source was The Plumbers Handbook, ninth edition, produced by the Australian Copper Tube Industry (Copper, T. A. and T. Industry, 2016). This included copper chemical, mechanical and physical properties and was primarily used to establish copper tube sizing such as outside diameter, wall thickness and inside diameter for use in the calculations. Copper tube for air conditioning, refrigeration and mechanical services complies with Australian and New Zealand Standard AS/NZS 1571 and is an alloy of phosphorus deoxidised copper high residual phosphorus called Alloy C12200, with a chemical composition of 99.90% minimum copper and 0.015%-0.040% phosphorus. From the information within this document (Copper, T. A. and T. Industry, 2016), the following properties were used in calculation in lieu of the above properties from Cengel & Ghajar (2015) (refer table 8 for Copper Alloy properties from The Plumbers Handbook).

Table 8: Copper Alloy properties from The Plumbers Handbook

Alloy C12200 properties @ 293K (20°C) – Page 10 - (Copper, T. A. and T. Industry, 2016):					
	ρ x 10^3 (kg/m^3)	**c_p (J/kg.K)**	**k (W/m.K)**	**α x10^6 (m^2/s)**	**ε**
Copper (Cu) Pure	8.94	385	355	117	0.02 – highly polished

3.5 Calculations from Data

As the project continued, further research was required to calculate the temperature losses of the refrigerant through the copper lines and thus the temperature of the refrigerant. In addition, the mass flow rate of the refrigerant and overall heat transfer for the system needed to be established in order to make accurate comparisons between the extrapolated data and results from the household a/c system and the results from the box a/c system, utilising the heat sink/source.

For natural convection calculations, the following process was used:

1. Properties of air (table A-15 - Cengel & Ghajar, 2015) at the film temperature were interpolated to find thermal conductivity (k), kinematic viscosity (v) and the Prandtl number (Pr);
2. The Rayleigh number could then be calculated using $R_a = \frac{g\beta(T_s - T_\infty)D^3}{v^2} Pr$;
3. The Nusselt number calculated using $Nu = \left\{0.825 + \frac{0.387 Ra_L^{\frac{1}{6}}}{\left[1+\left(\frac{0.492}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right\}^2$;
4. The heat transfer coefficient calculated using $h = \frac{k}{D} Nu$;
5. Using the formulas for thermal resistance as discussed in Section 3.4.2 and finding the natural convection heat transfer and T_1: $R_{conv,2}$, $Q_{conv,2}$, R_{cyl}, Q_{cyl} and T_1 (temp of inside wall of pipe).

For calculations relating to the internal refrigerant temperature and heat transfer, internal forced convection formulae were required from chapter 8 (Cengel & Ghajar, 2015) and the process was as follows:

1. The Reynolds number using $Re = \frac{V_{avg}\, D}{D} = \frac{\rho\, V_{avg}\, D}{\mu} = \frac{\rho D}{\mu}\left(\frac{\dot{m}}{\frac{[\rho\, \pi\, D^2]}{4}}\right) = \frac{4\dot{m}}{\mu\, \pi\, D}$;
2. Determination of Laminar or Turbulent flow based of Reynolds number ;
3. Using the applicable Nusselt number formula and finding h, $R_{conv,\,1}$ and $Q_{conv,1}$.

Due to the large amount of data collected over the year and certain key values not being available, a MATLAB program with nested while loops had to be created to perform two levels of iteration. The first being the value of mass flow rate, which was used to calculate the Reynolds number on the refrigerant. The second was the value of refrigerant temperature, used in calculating conductivity.

This proved challenging, however, the general methodology for this process in creating the MATLAB program was as follows:

1. Guess a value of mass flow rate
   a. Calculate Reynolds number
2. Guess a value of refrigerant temperature
   a. Calculate conductivity
   b. Use the forced convection heat transfer formula to calculate refrigerant temperature. If it differs to the value used in 2., adjust the refrigerant temperature value and repeat steps (a) and (b)
   c. Use the power consumption data collected to calculate mass flow rate. If it differs to the value used in 1., adjust the value of mass flow rate and repeat all steps.

For the extrapolation calculations of the household a/c, the heat sink and source temperatures were used for $T_{inf,2}$ and therefore forced convection formulae was used for both internal and external pipe calculations. In addition, the mass flow rate is the same in the unmodified and modified system so this value did not require iteration (refer Section 4.3). Only refrigerant temperature from the forced convection inside and outside the pipe required iteration and from there enthalpy 1 or 2 (either side of the compressor) depending on whether performing calculations for heating or cooling mode could be found, followed by power use/consumption of the household a/c system with the flowing water loop.

3.6 Quality Assurance Plan

Quality assurance is a crucial factor in determining the success of the research, both in the initial phases and towards the final stages of the project. As part of this process a series of reviews and supervisor meetings was conducted to ensure compliance with any relevant standards and regulations and to ensure the aim and objectives of the research project were being adequately met in a safe and effective manner. This limited erroneous results as the project continued, ensured correct and accurate data was captured and the appropriate task planning conducted. Furthermore, the process ensured the completion of the project in a timely manner and the viability and suitability of the information captured.

The following is a basic list of parameters and measures utilised as the project progressed to endeavour to maintain quality assurance:

- Any experimental design information is vetted by the supervisor during the review stage to ensure viability of parameters, assumptions, variables and definitions.
- Documentation and tracking of all data and research conducted in the excel worksheet created and stored in Microsoft OneDrive.
- Any safety or ethical related tasks involving companies, organisations and site visits will be assessed with high importance and discussed with the supervisors and/or other USQ representative with expertise on such matters.
- Methodology specifics such as quantitative or qualitative values, such as torques, pressures, clearances, power settings, parameters, assumptions, variables, definitions, statistical analysis, interviews and observations will be stated and baselined for various tasks. This will enable comparison against existing baselines, if required.
- Calibration of any testing equipment is carried out at the necessary interval and to the required level and level of accuracy defined.
- Fortnightly updates with the supervisor via email to address progress and any issues identified that require feedback or guidance.

- Regular (6 weekly) zoom meetings/phase completion email correspondence to occur with the supervisor to track progress and assist with any advice or problems in the project.
- Maintain the Gantt chart project timeline to ensure tasks are performed at the correct phase of the study and completed in a timely manner.

3.7 Resource Requirements

A variety of resources were required over the duration of the project to achieve the objectives and aim appropriately. As the project was being undertaken as an undergraduate, the lack of funding determined certain constraints and limitations for resources. The project was undertaken as an online student working full-time, thus, majority of project work will occur outside of the normal work hours and take up approximately 300 hours over the course of the two semesters. Consultation and communication with the required stakeholders took place within the working hours of the week if time was granted from the current employer. Furthermore, face-to-face, zoom meetings or emails with the project supervisor was ideally performed within working hours and adapted to suit availability of both student and supervisor as required.

A list of software, hardware and miscellaneous resource requirements are listed in table 9 below.

Table 9: Resources required for the project to be carried out successfully

Resource Item	Cost	Source	Quantity	Comment (If Required)
Laptop and PC	Nil – Unless broken and requires replacement	Student	1 of each	If updating is required
Current Microsoft Office Suite	Nil	USQ	1	N/A
Creo Parametric/Simulate	Nil	USQ	1	Modelling may be required.
MATLAB	Nil	USQ	1	Current version to assist with any calculations
Adobe Reader	Nil	USQ	1	N/A
USQ Library	Nil	USQ	N/A	N/A
Scopus	Nil	USQ	N/A	N/A

Science Direct	Nil	USQ	N/A	N/A
Up-to-date Standards and Regulations	Nil	USQ	Multiple	N/A
Household Pool	$$$ - offset by solar power	Student	1	Heater starts to run towards winter – further costs
Household Solar Hot Water system	$$	Student	1	On cloudy/rainy days electric booster must be turned on – cost supplemented by solar system
Household Air-Conditioning System	$$$	Student	1	Must be run in all seasons – even when not required to get measurements – Very costly when not required and only turned on for project.
Air-Conditioning parts for design build	$100-$200	Various - Student / USQ assistance	Multiple	Unsure currently on costing of this
Temperature measurement device	$15.00	Student	1	N/A
Tools - various	Nil – unless specialist tooling requires	Student	Multiple	Tools to build design
Vehicle	Fuel costs	Student	1	Transport to locations to pick up parts or USQ Springfield/Toowoomba
Sundry items (pens, pencils, printing, paper,	Approx. $60.00	Student	As Required	N/A

staples, rulers, calculator etc.)				

3.8 Risk Assessment/Management Plans

In designing the air-conditioning system utilising water storage sources such as the residential pool and hot water system to increase energy efficiencies there was several hazards that will require risk assessment and a risk management plan using the hierarchy of control measures (refer figure 18) to ensure health and safety, financial and environmental impacts are minimised for the variety of tasks throughout the project. As part of identifying these risks, it was important to understand and comply with Australian Standard (AS) ISO 31000:2018 – Risk Management – Guidelines (ISO, 2018), which identifies the principles, framework and risk management process to adhere to when performing any project. Australian Standard (AS/NZS) IEC 31010:2020 – Risk Management – Risk Assessment Techniques (IEC, 2020) was an important document to view as it provided guidance on selection and application of a variety of risk assessment techniques that may be used depending on the context of the task.

For the purposes of this research and design task, the Consequences/Likelihood Matrix (Risk Matrix) was used to identify, evaluate and control a variety of hazards that could occur throughout the duration of the project and specifically with the box a/c design testing (refer figure 19). The Risk Matrix was crucial in assessing the level of risk by using a likelihood versus consequences matrix, which identified if the current control measure was adequate to continue with the task or if further control measures or engineering approval was required. A variety of factors may be assessed here, including but not limited to health and safety, financial and environmental/community implications.

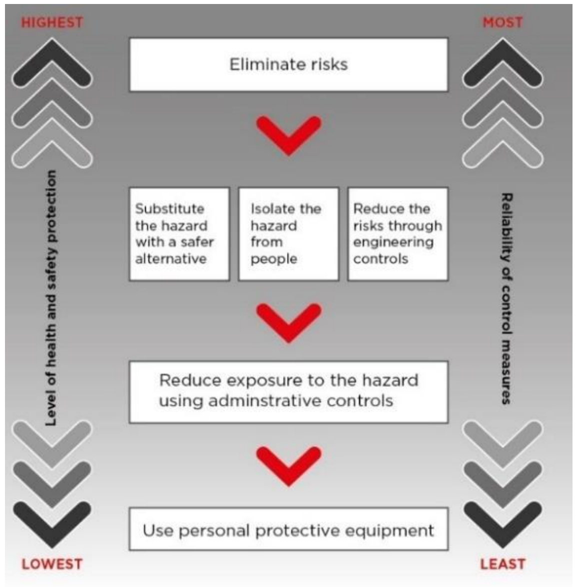

Figure 18. Hierarchy of control measures utilised for any risks identified during the research project (Australia, SW, 2020).

RISK MATRIX - ADAPTED FROM ISO 31000:2009

E - Extreme Risk - Detailed action plan required to manage risk before progressing
H - High Risk - Needs immediate senior management attention
M - Medium Risk - Specify management responsibility
L - Low Risk - Manage by routine procedures

					CONSEQUENCE				
				People	Injuries or ailments not requiring medical treatment	Minor injuries or First Aid treatment	Serious injury causing hospitilisation or multiple medical treatment cases	life threatening injury or multiple serious injuries causing hospitilisation	Death or multiple life threatening injuries
	Probability	Historical			insignificant 1	Minor 2	Moderate 3	Major 4	Catastrophic 5
LIKELIHOOD	>1 in 10	Is expected to occur in most circumstances	5	Almost Certain	M	H	H	E	E
	1 in 10 - 100	Will probably occur	4	Likely	M	M	H	H	E
	1 in 100 - 1000	Might occur at some time in the future	3	Possible	L	M	M	H	E
	1 in 1000 - 10 000	Could occur but doubtful	2	Unlikely	L	M	M	H	H
	1 in 10 000 - 100 000	May occur but only in exceptional circumstances	1	Rare	L	L	M	M	H

Figure 19. Risk matrix utilised when assessing risks for the research project (HSEMA, 2020).

Utilisation of both the Hierarchy of Control Measures and the Risk Matrix for hazards and risks identified for the project will appropriately risk manage and reduce the risk to As Low As Reasonably Practicable (ALARP).

The goal of any hazard/risk identified within this research project is primarily to eliminate the hazard. If this cannot be achieved, minimising the risk of the hazard could be carried out utilising the hierarchy of controls and by maintaining or reducing the likelihood of the risk exposure (hazard) occurring to rare, unlikely or possible; whilst the consequence should be maintained or reduced to insignificant or minor. If the likelihood or consequence was greater than any of the above and outside the Low field (green – bottom left corner of the matrix) than the task should be changed, or the risk needs to be addressed and authorised at a higher level of management. For the purposes of the research project, a level of Low was required to proceed with the task, as it was primarily conducted individually and no supervision of day to day activities occurred.

Table 10 displays the risk management plan for the risks and hazards identified for the overall project and the associated risk control measures used to manage the risk. The final two columns of the table display the risk matrix level before and after the risk control measure has been implemented to show that the risk is of an acceptable level. Table 11 displays a specific risk management plan for the identified risks and hazards when performing testing and data collection on the box air-conditioner.

Table 10: The risks/hazards associated with the research project, risk control technique utilised and the final risk matrix level

Risk/Hazard Description	Hierarchy of Control Measure	Risk Control	Risk Matrix level before Risk Control	Risk Matrix level after Risk Control
Poor working ergonomics, including bad posture, excessive screen time and minimal breaks.	Engineering and Administrative controls utilised	- Schedule breaks appropriately; - Set alarms to avoid excessive screen time. - Set up computer workspace ergonomically and be cognisant of posture and working environment. - Incorporate stretching, natural light in room and fresh air regularly etc.	Likely/ Insignificant **Medium**	Rare/ Insignificant **Low**
Loss of data and project information	Administrative controls	- Back up all data and project information to a	Likely/Minor **Medium**	Rare/Minor **Low**

due to technological issues. Faulty thumb drives, data corruption, faulty PC or laptop.		hard drive and online storage such as OneDrive as soon as practicable and on a regular basis.		
Insufficient/inadequate data when modelling in case study.	Administrative controls used	- Perform appropriate experimental design to ensure the amount of data required is defined prior to collection.	Possible/ Minor **Medium**	Rare / Insignificant **Low**
Tasks falling behind due to unforeseen circumstances.	Administrative controls used	- Assess issue promptly. - - If tasks can be caught up, schedule/plan for this. - If the tasks are too big, consult supervisor and request other options or workarounds.	Unlikely / Major **High**	Unlikely / Insignificant **Low**
Access to water storage systems to collect data and design system.	Engineering, Administrative and PPE controls utilised	- If accessing A/C & HW system for data collection use appropriate temp measuring device. - Do not tamper with operating system components - Wear appropriate PPE. E.g. Long shirt/pants, protective footwear, sun protection, gloves if required.	Likely/ Moderate **High**	Rare /Minor **Low**
Using hand and some power tools to assist in data collection and building design.	Substitution, Engineering, Administrative and PPE controls utilised	- Use improving tooling to prevent injury. - When building ensure working on flat sturdy surface; use clamps. - Do not work on any electrical components without turning system off and isolating power and understanding	Likely/ Moderate **High**	Rare /Minor **Low**

		residual current that may be present. - Wear appropriate PPE when using tools, building and collecting data.		
Interpreting data and analysing efficiencies of systems	Administration controls Utilised	- Research this area in detail. - Request feedback and advice from supervisor if required.	Possible/ Minor **Medium**	Rare / Insignificant **Low**

Table 11: Risk Management Plan for the Box Air-Conditioner Design during Set-Up, Testing and Data Collection

Risk/Hazard Description	**Hierarchy of Control Measure**	**Risk Control**	**Risk Matrix level before Risk Control**	**Risk Matrix level after Risk Control**
As cover is removed during testing, the fan is partially exposed creating a cutting hazard.	Isolation, Administrative and PPE controls utilised	- Don't put hands in or near a/c internals during operation. - When setting up heat sink/source on refrigerant lines, ensure a/c is OFF and unplugged. - Wear Safety Gloves and glasses during set-up, testing and data collection.	Unlikely / Moderate **Medium**	Rare / Minor **Low**
Burn hazard from refrigerant lines or other internal components of box a/c	Isolation, Administrative and PPE controls utilised	- Don't put hands in or near a/c internals during operation; - Allow a suitable amount of time after turning off system before working on the internal components to allow some of the heat to dissipate;	Unlikely / Moderate **Medium**	Rare / Minor **Low**

		- When setting up heat sink/source, ensure a/c is OFF and unplugged. - Wear Safety Gloves during testing/data collection.		
Electrical hazard from control box being slightly exposed when cover is removed	Isolation, Engineering, Administrative controls used	- Don't put hands in or near a/c internals during operation; - When setting up heat sink/source, ensure a/c is OFF and unplugged. - Ensure all wires are protected by insulation/conduit and wrap in electrical tape if required.	Rare / Major **Medium**	Rare / Insignificant **Low**
Injuries as a result of using hand and power tools to assist in data collection and building/testing design.	Substitution, Engineering, Administrative and PPE controls utilised	- Use improved tooling to prevent injury (drill driver, impact driver). - When data collecting/testing ensure working on flat sturdy surface; use clamps/tie-down straps if necessary. - Do not work on any electrical components without turning system OFF and isolating power; understanding residual current may be present. - Wear appropriate PPE when using tools, testing and collecting data such as safety gloves and glasses.	Likely/ Moderate **High**	Rare /Minor **Low**
Achieving accurate results/data will be difficult with outside location of	Administrative controls used	- Conduct data collection and testing at the same time each day and use	Possible / Moderate **Medium**	Unlikely / Insignificant **Low**

box a/c (not fitted in wall)		same methodology each time. - Use same temperature measuring device. - Request feedback and advice from supervisor if required.		
Interpreting data and analysing efficiencies of systems	Administration controls Utilised	- Research this area in detail. - Request feedback and advice from supervisor if required.	Possible / Moderate **Medium**	Unlikely / Insignificant **Low**

4. Results and Discussion

4.1 Interpretation and Analysis of Results

In achieved the outcomes of the project to an appropriate standard, the way in which the data and results were utilised was of the utmost importance. Using a variety of approaches throughout the research and documenting it consistently, correctly and concisely ensured the information could be found easily and adapted for each phase of the project. Microsoft Excel, Microsoft OneDrive and MATLAB R2018b were the main software for tracking this data for easy collation and managing of tasks and results. The project supervisor had further advice that has assisted in other methods of recording and tracking information, however, throughout the duration of the project these were the primary tools used for this purpose. For smaller day-to-day monitoring of tasks within the project, a diary was used in collaboration with the project plan/Gannt chart, especially when carrying out field tasks that required information to be taken down on the job. Using a OneDrive folder for the project ensured the information was kept in online storage in case of PC or Laptop data corruption and backed up on an external hard drive as often as possible. The OneDrive folder included sub folders dedicated to each phase of the research and was setup in an easy to use format.

Objective 4 of the project: design, build and test an air-conditioning system using pool water as a heat sink and solar hot water as a heat source required defining parameters, assumptions and variables for the design to ensure the task was performed as accurately as possible within the limitations, constraints and assumptions stated and to ensure the information gained would be viable and effective for further studies if required.

Evaluating all research findings and results was be kept consistent and accurate over the course of the project to ensure that replication of the data and findings could be carried out to further any research and studies in the area. The project supervisor assisted with regular reviews to ensure the objectives were accurate/on-track and maintaining viable and accurate results.

4.2 Box Air-Conditioning System

Initially, during the testing of the box air-conditioning system the hot/cold packs were applied to various refrigeration copper pipes throughout the system in both heating and cooling mode to see how the system would perform to a temperature change in each part of the vapour compression cycle. The trial and error stage included educated guesses, knowing the state and approximate temperature range of the refrigerant in the system at any one point.

It was found in cooling mode that by applying a hot pack to the compressor inlet simulating the heat source water temperature (solar hot water system) saw the most effective reduction in power consumption (refer figure 20 showing the location of the compressor inlet pipe and hot pack applied). Due to the shape of the pipe and proximity to the condenser fan it was difficult to apply the hot pack to the length of pipe desired or to encase the circumference of the pipe.

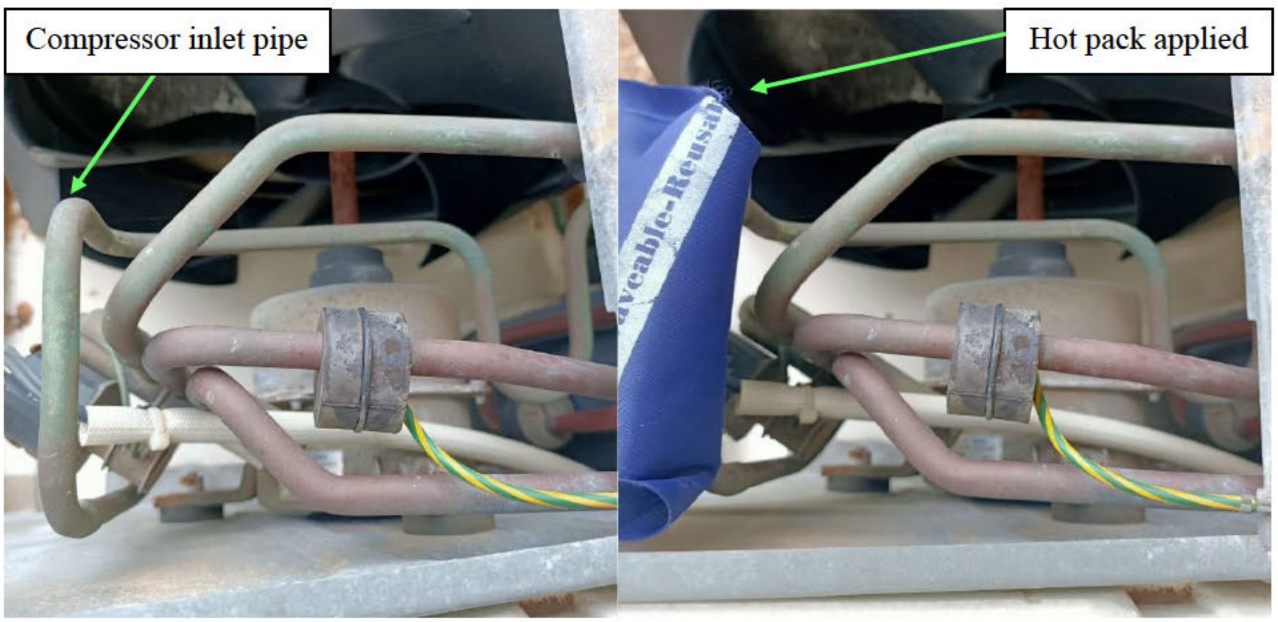

Figure 20. Compressor inlet pipe without and with hot pack applied during Cooling Mode

Figure 21 and 22 show graphs of power consumption versus temperature. The before temperature is the maximum ambient air temperature of the day (when the measurements were taken) and the after temperature is the temperature of the hot pack (heat source) applied to the refrigerant copper pipe at the compressor inlet pipe in cooling mode. It is important to note that this data for the box a/c was collected during August (winter) in Ipswich, QLD, where the ambient air temperatures are cool, so the effectiveness and viability of the results of the box a/c in cooling mode was not optimal.

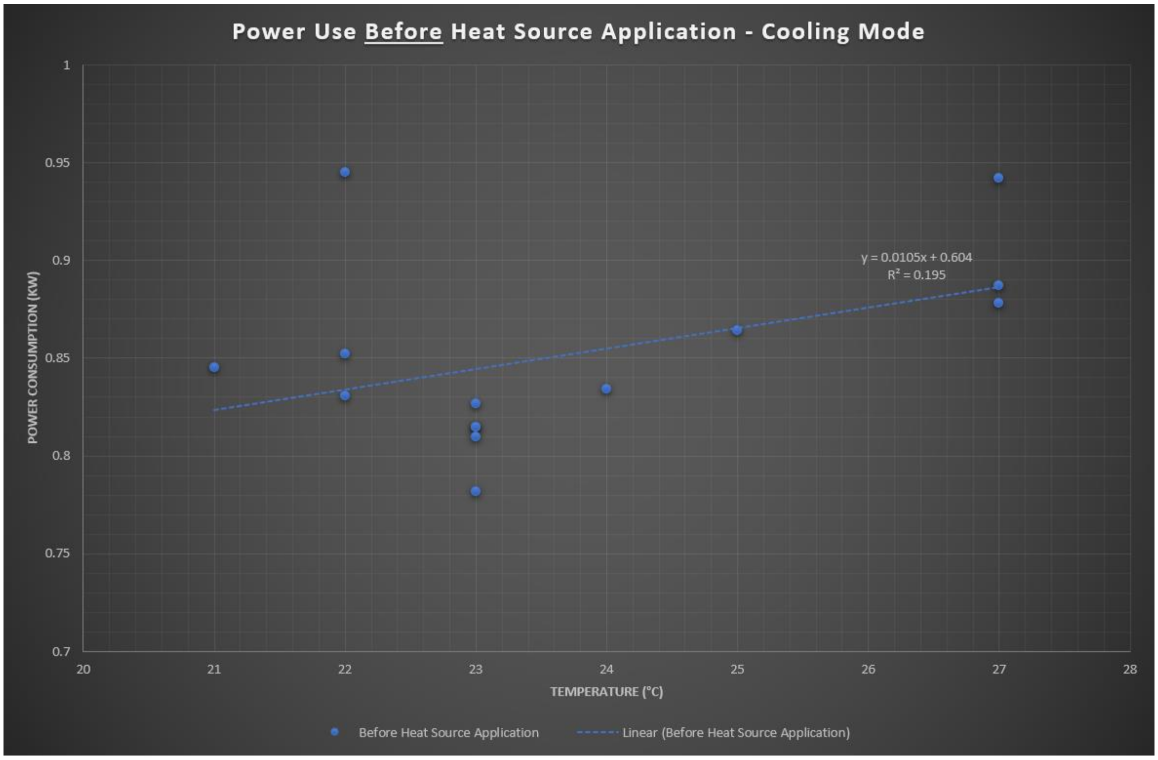

Figure 21. Power use of the box a/c before the hot pack was applied - Cooling Mode

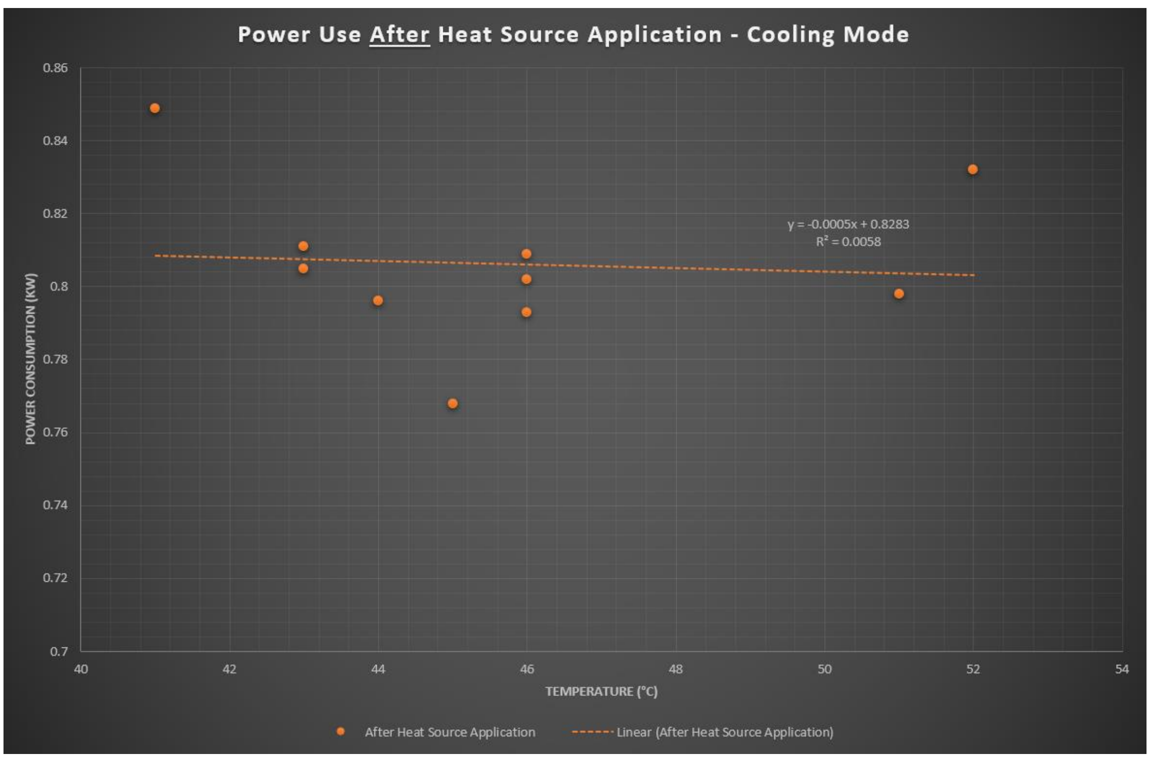

Figure 22. Power use of the box a/c after the hot pack was applied - Cooling Mode

Due to time constraints for the project, data for the box a/c was only collected for 14 consecutive days, however, a noticeable trend and reduction in power consumption was evident (refer table 12 for results of power consumption before and after heat source application to the compressor inlet pipe of the box a/c system). The findings show a power consumption average of 0.851929kW prior to the hot pack being applied to the compressor inlet pipe and an average power consumption of 0.812kW after application of the hot pack. This is a 4.687% reduction in power consumption for the box a/c system in cooling mode using a low-grade heat source applied to the compressor inlet copper pipe.

Table 12: Power consumption before and after heat source application to the compressor inlet pipe of the box a/c system in Cooling Mode.

Date	**Power use before adding source (kW)**	**Power use after adding source (kW)**
12-Aug 20	0.834	0.811
13-Aug 20	0.878	0.845
14-Aug 20	0.942	0.906
15-Aug 20	0.815	0.791
16-Aug 20	0.782	0.763
17-Aug 20	0.815	0.768
18-Aug 20	0.864	0.802
19-Aug 20	0.887	0.832
20-Aug 20	0.827	0.796
21-Aug 20	0.81	0.793
22-Aug 20	0.831	0.798
23-Aug 20	0.845	0.809
24-Aug 20	0.852	0.805
25-Aug 20	0.945	0.849
Average	**0.851929**	**0.812**
% Power Reduction	**4.687%**	

In heating mode, it was found by applying a cold pack to the compressor outlet simulating the heat sink water temperature (residential pool) saw the most effective reduction in power consumption (refer figure 23 showing the location of the compressor outlet pipe and cold pack applied). The compressor outlet pipe was easier to access and the cold pack could be applied to a greater length of the pipe when compared to the hot pack for cooling mode. For an optimum result however, the entire circumference of the pipe would need to be in contact with the cold pack and a greater length of the compressor outlet copper pipe cooled.

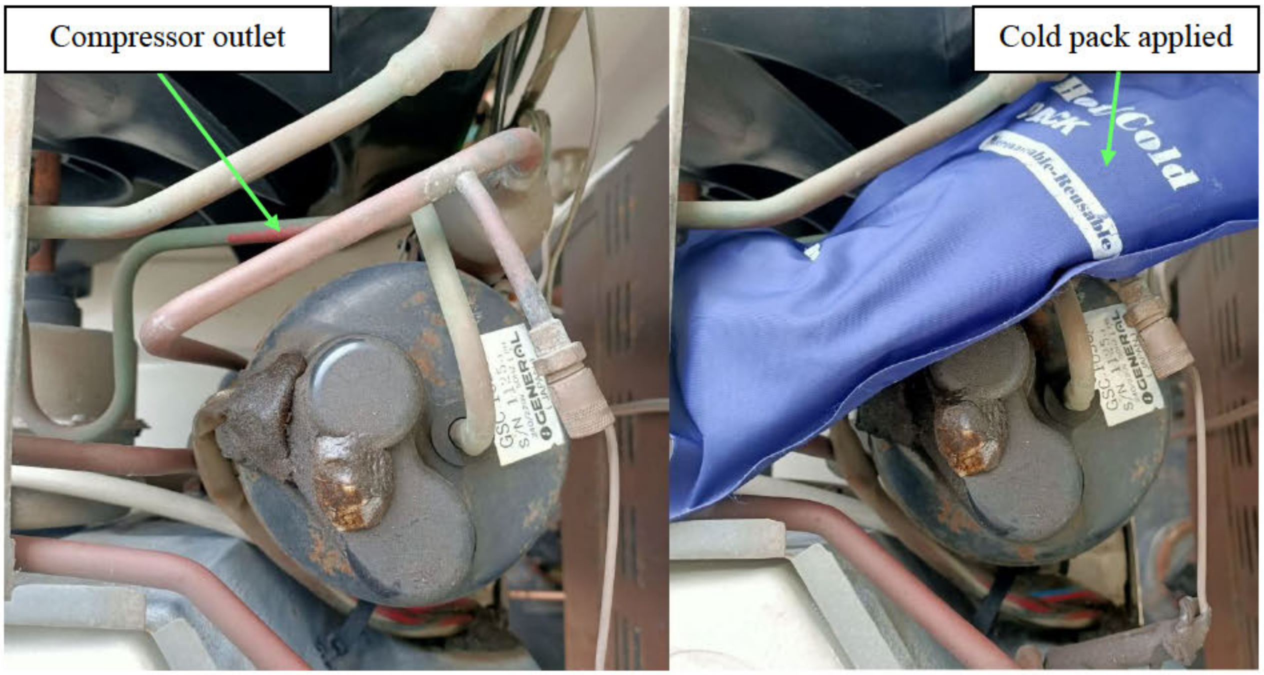

Figure 23. Compressor outlet pipe without and with cold pack applied during Heating Mode

Figure 24 and 25 show graphs of power consumption versus temperature. The before temperature is the maximum ambient air temperature of the day (when the measurements were taken) and the after temperature is the temperature of the cold pack (heat sink) applied to the refrigerant copper pipe at the compressor outlet pipe. As shown, the results before and after application of the cold pack are much more linear for heating mode due to the ambient air conditions where the testing was performed. These showing greater performance/effectiveness of heating in the cooler ambient air conditions. Further testing could be achieved in the warmer months to assess how the design would perform in these conditions, but as heating is generally not required during these months it may not be feasible.

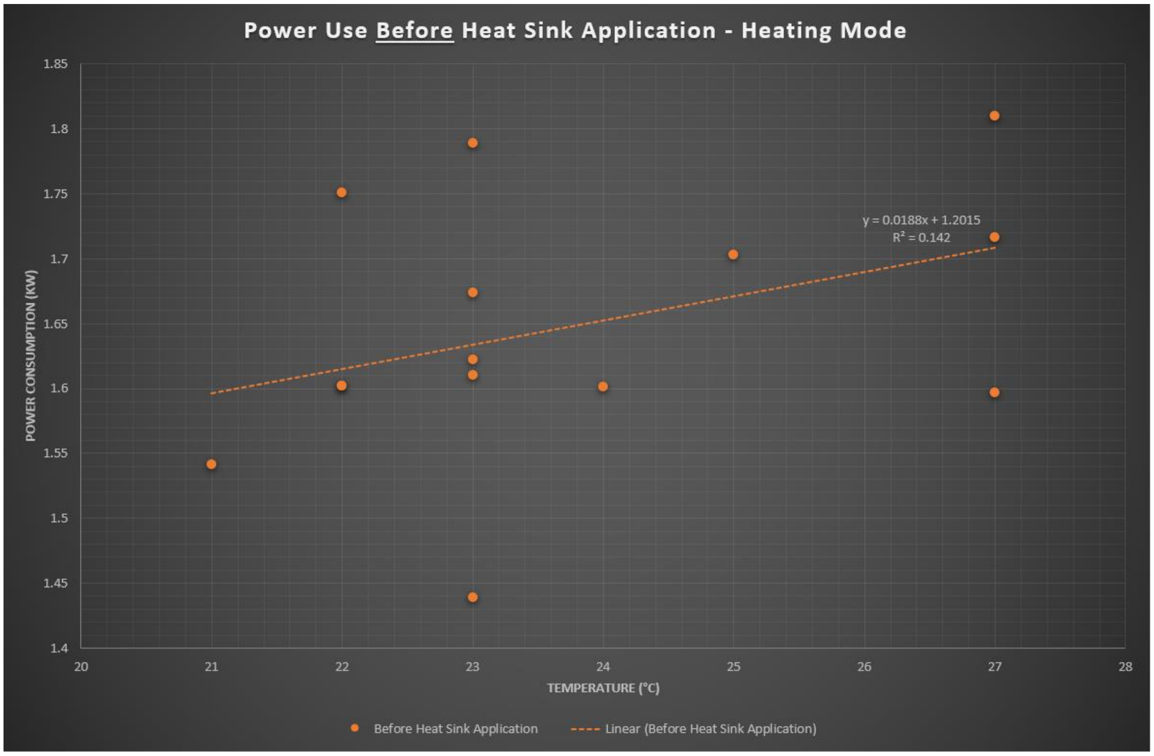

Figure 24. Power use of the box a/c before the cold pack was applied - Heating Mode

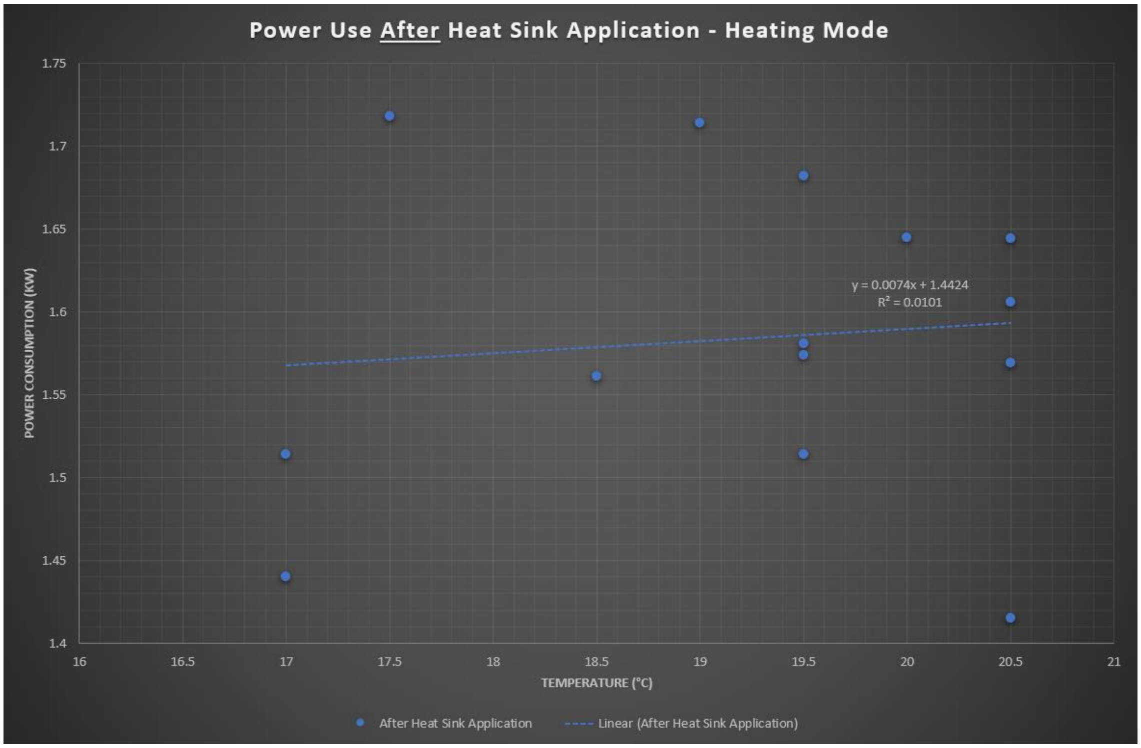

Figure 25. Power use of the box a/c after the cold pack was applied - Heating Mode

As with cooling mode, the data collected was for a 14-day period in August with results showing a slightly less reduction in power consumption for heating than for cooling, however, a trend and definite reduction was evident (refer table 13 for results of power consumption before and after heat source application to the compressor inlet pipe of the box a/c system). The findings show a power consumption average of 1.646929kW prior to the cold pack being applied to the compressor outlet pipe and an average power consumption of 1.584071kW after application of the cold pack. This is a 3.817% reduction in power consumption for the box a/c system in heating mode using a low-grade heat sink applied to the compressor outlet copper pipe.

Table 13: Power consumption before and after heat sink application to the compressor outlet pipe of the box a/c system in Heating Mode.

Date	Power Use before adding sink (kW)	Power Use after adding sink (kW)
12-Aug 20	1.601	1.561
13-Aug 20	1.716	1.682
14-Aug 20	1.81	1.714
15-Aug 20	1.674	1.606
16-Aug 20	1.439	1.415
17-Aug 20	1.61	1.514
18-Aug 20	1.703	1.645
19-Aug 20	1.597	1.581
20-Aug 20	1.622	1.574
21-Aug 20	1.789	1.644
22-Aug 20	1.602	1.514
23-Aug 20	1.541	1.44
24-Aug 20	1.751	1.718
25-Aug 20	1.602	1.569
Average	**1.646929**	**1.584071**
% Power Reduction	**3.817%**	

The results displayed above were using the temperature of the outside of the copper pipe as the refrigerant temperature using an infrared thermometer, however, it was found through calculation using MATLAB that the temperature of the refrigerant inside the copper pipe was only 0.065°C higher on average for heating mode and 0.014°C higher on average for cooling mode. The refrigerant temperatures, heat transfer rates, and mass flow rates of the refrigerant were determined for each day in MATLAB for this dataset with refrigerant temperature ($T_{inf,1}$) and mass flow rate being iterated through several loops until the correct temperature and mass flow rate was found to within $1x10^{-6}$ error tolerance (refer Section 3.4.2 and 3.5 for the methodology followed to find these values). The average mass flow rate for heating mode of the box a/c system was 0.042kg/s and for cooling mode was 0.026kg/s, with an overall mass of R22 refrigerant in the system of 0.68kg.

For the complete workings and MATLAB calculations performed for the box a/c system refer to Appendix J for the Box Air-Conditioning System MATLAB Calculations before the heat pack was applied in Cooling Mode; refer Appendix K for the Box Air-Conditioning System MATLAB Calculations after the heat pack was applied in Cooling Mode; refer Appendix L for the Box Air-Conditioning System MATLAB Calculations before the cold pack was applied in Heating Mode; refer Appendix M for the Box Air-Conditioning System MATLAB Calculations after the cold pack was applied in Heating Mode).

4.3 Household Air-Conditioning System

The data collected over the duration of the project year on the household a/c system was used as a baseline for ascertaining a reasonable energy consumption of the system without modification. The baseline data was used in MATLAB calculations similar to the box a/c calculations to find temperature of the refrigerant ($T_{inf,1}$), heat transfer rate, mass flow rate and enthalpy 1 and 2 (before and after the compressor respectively) on each day of the dataset (refer Section 3.4.2 and 3.5 for the methodology followed to find these values). The extrapolation of the data using MATLAB determined through calculation what the energy consumption would be if a heat sink or source flowing water loop was utilised on the a/c system.

The data that was crucial to this extrapolation included:

- Heat sink/source temperatures and associated properties
- Refrigerant pipe temperatures on either side of the compressor
- Refrigerant properties (R410A)
- Mass flow rate of the system prior to applying the sink/source
- Enthalpy 1 (before compressor), as this remained unchanged for heating mode when applying the cold water sink flowing water loop to the compressor outlet refrigerant pipe section but required calculation for cooling mode.
- Enthalpy 2 (after compressor), as this remained unchanged for cooling mode when applying the hot water source flowing water loop to the compressor inlet refrigerant pipe but once again required calculation for heating mode.
- Power consumption after heat sink/source application.

As applying the flowing water loop to the a/c system is essentially adding a heat exchanger to the required pipes, the modification was occurring on the outside of the system and the operating conditions of the system are therefore maintained on the inside of the building. Therefore, it was assumed that the mass flow rate was the same in the system before and after the modification and similarly the enthalpy was the same at the start and finish of the inside coil. From this information, in heating mode enthalpy 1 (compressor inlet) stayed the same before and after the modification and therefore used in the MATLAB calculations for the extrapolation, as the heat sink flowing water loop would be applied to the compressor outlet, changing enthalpy 2 only. For cooling mode, enthalpy 2 (compressor outlet) would remain the same before and after the modification and therefore used in the MATLAB calculations for the extrapolation, as the heat source flowing water loop would be applied to the compressor inlet, changing enthalpy 1. (refer Appendix F for the Household Air-Conditioning System MATLAB Calculations before extrapolation using flowing water loop design in Cooling Mode with a H_20 Mass Flow Rate of 1.33kg/s; refer Appendix G for the Household Air-Conditioning System MATLAB Calculations after extrapolation using flowing water loop design in Cooling Mode with a H_20 Mass Flow Rate of 1.33kg/s; refer Appendix H for the Household Air-Conditioning System MATLAB Calculations before extrapolation using flowing water loop design in Heating Mode with a H_20 Mass Flow Rate of 1.33kg/s; refer Appendix I for the Household Air-Conditioning System MATLAB Calculations after extrapolation using flowing water loop design in Heating Mode with a H_20 Mass Flow Rate of 1.33kg/s).

Three different pumps were chosen to create flow for the water loop, each with a different flow rate, to assess which pump would be most beneficial/effective at reducing power consumption for the design. The chosen pumps were two small jet pumps and a pond pump with flow rates 80LPM (1.33kg/s), 50LPM (0.83kg/s) and 25LPM (0.42kg/s) respectively. Through interchanging the water mass flow rate values in the MATLAB calculations, it was found that the highest mass flow rate pump (1.33kg/s) had the most effective heat transfer rate and greatest power reduction in the household a/c system for both heating and cooling mode (refer table 14 showing an excerpt comparing the refrigerant temperature and power consumption before and after modification with the 1.33kg/s water pump, 0.83kg/s water pump and 0.42kg/s water pump for the heat source flowing water loop applied to the compressor inlet in cooling mode and refer table 15 showing an excerpt of the comparison of refrigerant temperature and power consumption before and after modification with the 1.33kg/s water pump, in heating mode (refer Appendix O for the complete table showing this comparision of data between the three pumps for heating mode and Appendix N for the complete table showing this comparision of data between the three pumps for cooling mode). This could be trialled/tested quite easily with other pumps of larger or smaller mass flow rates or different diameter tubing to ascertain an optimal pump/flow rate selection.

Table 14: Excerpt comparing the power use/consumption and refrigerant temperature before modification and after modification with the 1.33kg/s water pump, 0.83kg/s water pump and 0.42kg/s water pump for the heat source flowing water loop applied to the compressor inlet – Cooling Mode.

BEFORE MODIFICATION				AFTER MOD WITH 1.33kg/s pump		AFTER MOD WITH 0.83kg/s pump		AFTER MOD WITH 0.42kg/s pump	
MFR	enthalpy2	T_refrig	poweruse	T_refrig	poweruse	T_refrig	poweruse	T_refrig	poweruse
0.158	480.620	33.0	5.030	37.3	4.254	35.7	4.535	34.4	4.766
0.126	479.303	32.0	3.990	37.4	3.217	35.3	3.498	33.7	3.730
0.193	477.311	30.5	6.060	34.0	5.269	32.7	5.553	31.6	5.787
0.182	473.922	28.0	5.590	31.7	4.792	30.3	5.079	29.2	5.316
0.209	471.839	26.5	6.350	29.8	5.540	28.5	5.830	27.5	6.068
0.126	473.918	28.0	3.880	33.4	3.093	31.4	3.380	29.7	3.615
0.143	475.967	29.5	4.460	34.2	3.675	32.5	3.960	31.0	4.194
0.188	473.919	28.0	5.780	31.6	4.980	30.3	5.268	29.1	5.505
0.128	472.534	27.0	3.910	32.3	3.119	30.3	3.406	28.7	3.643
0.117	479.967	32.5	3.700	38.3	2.931	36.1	3.211	34.3	3.442
0.156	479.307	32.0	4.940	36.3	4.161	34.7	4.443	33.4	4.675
0.180	479.306	32.0	5.700	35.8	4.916	34.3	5.199	33.2	5.431
0.175	480.622	33.0	5.560	36.9	4.780	35.4	5.062	34.2	5.293
0.193	479.305	32.0	6.100	35.5	5.314	34.2	5.597	33.1	5.829
0.138	480.619	33.0	4.400	37.9	3.628	36.0	3.908	34.5	4.139
0.112	487.761	38.5	3.670	44.5	2.940	42.2	3.204	40.4	3.423

Table 15: Excerpt comparing the power use/consumption and refrigerant temperature before modification and after modification with the 1.33kg/s water pump, 0.83kg/s water pump and 0.42kg/s water pump for the heat sink flowing water loop applied to the compressor outlet – Heating Mode.

BEFORE MODIFICATION				AFTER MOD WITH 1.33kg/s pump		AFTER MOD WITH 0.83kg/s pump		AFTER MOD WITH 0.42kg/s pump	
MFR	enthalpy1	T_refrig	poweruse	T_refrig	poweruse	T_refrig	poweruse	T_refrig	poweruse
0.1198	333.75	38.5	4.940	38.0	4.702	38.2	4.792	38.3	4.865
0.1562	363.51	44.0	4.650	43.6	4.443	43.8	4.521	43.9	4.585
0.1667	361.25	43.5	5.080	43.1	4.871	43.3	4.950	43.4	5.014
0.1749	379.07	48.0	4.560	47.7	4.374	47.8	4.444	47.9	4.502
0.1302	354.07	42.0	4.280	41.5	4.062	41.7	4.144	41.9	4.212
0.1530	373.74	46.5	4.150	46.1	3.957	46.3	4.030	46.4	4.089
0.1817	375.54	47.0	4.860	46.7	4.669	46.8	4.741	46.9	4.800
0.1540	351.45	41.5	5.210	41.1	4.990	41.3	5.073	41.4	5.141
0.1143	333.82	38.5	4.710	38.0	4.472	38.2	4.562	38.4	4.635
0.1219	356.59	42.5	3.900	42.0	3.685	42.2	3.766	42.4	3.832
0.2199	375.53	47.0	5.880	46.7	5.689	46.8	5.761	46.9	5.820
0.1809	354.01	42.0	5.950	41.7	5.733	41.8	5.814	41.9	5.882
0.2085	363.48	44.0	6.210	43.7	6.004	43.8	6.081	43.9	6.145
0.3406	385.46	50.0	8.560	49.8	8.385	49.9	8.451	50.0	8.505
0.2072	375.53	47.0	5.540	46.7	5.349	46.8	5.421	46.9	5.480
0.2788	377.30	47.5	7.360	47.3	7.172	47.4	7.243	47.4	7.301

Figure 26 shows the power consumption of the household a/c system prior to the heat source flowing water loop being applied to the compressor inlet pipe in cooling mode. Figure 27 shows the power consumption of the household a/c system after the heat source flowing water loop was applied to the compressor inlet pipe in cooling mode. Figure 28 shows the power consumption of the household a/c system prior to the heat sink flowing water loop being applied to the compressor outlet pipe in heating mode. Figure 29 shows the power consumption of the household a/c system after to the heat sink flowing water loop was applied to the compressor outlet pipe in heating mode.

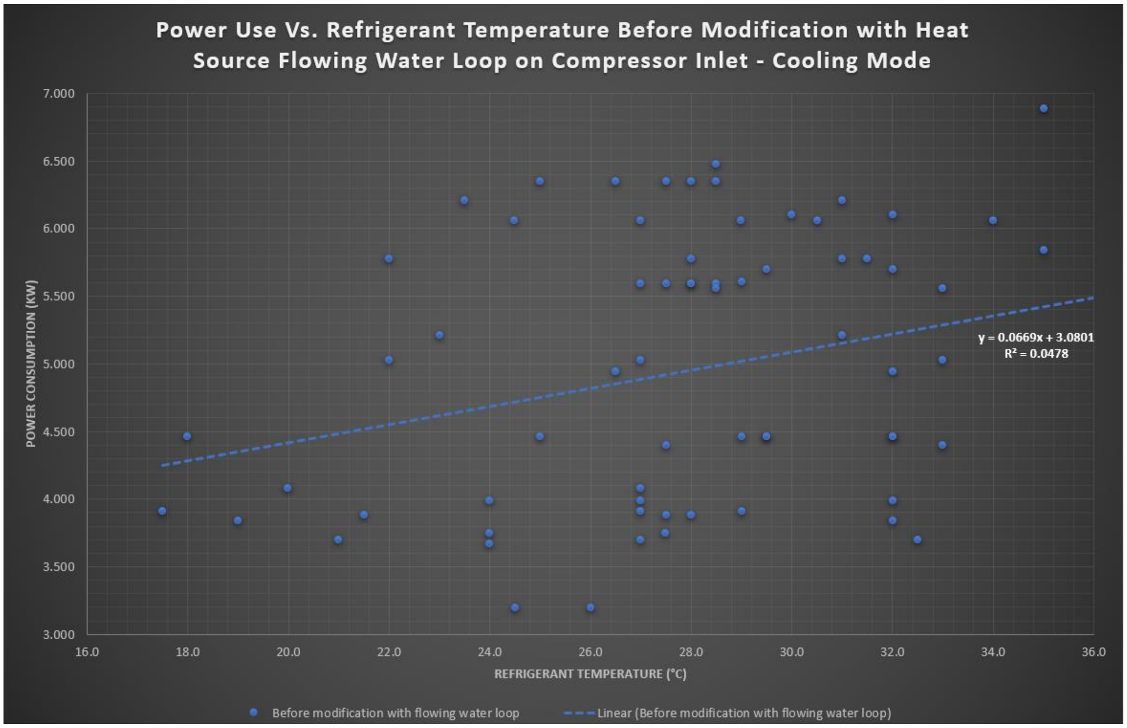

Figure 26. Power consumption of household a/c system before applying the heat source flowing water loop modification to the compressor inlet - Cooling Mode

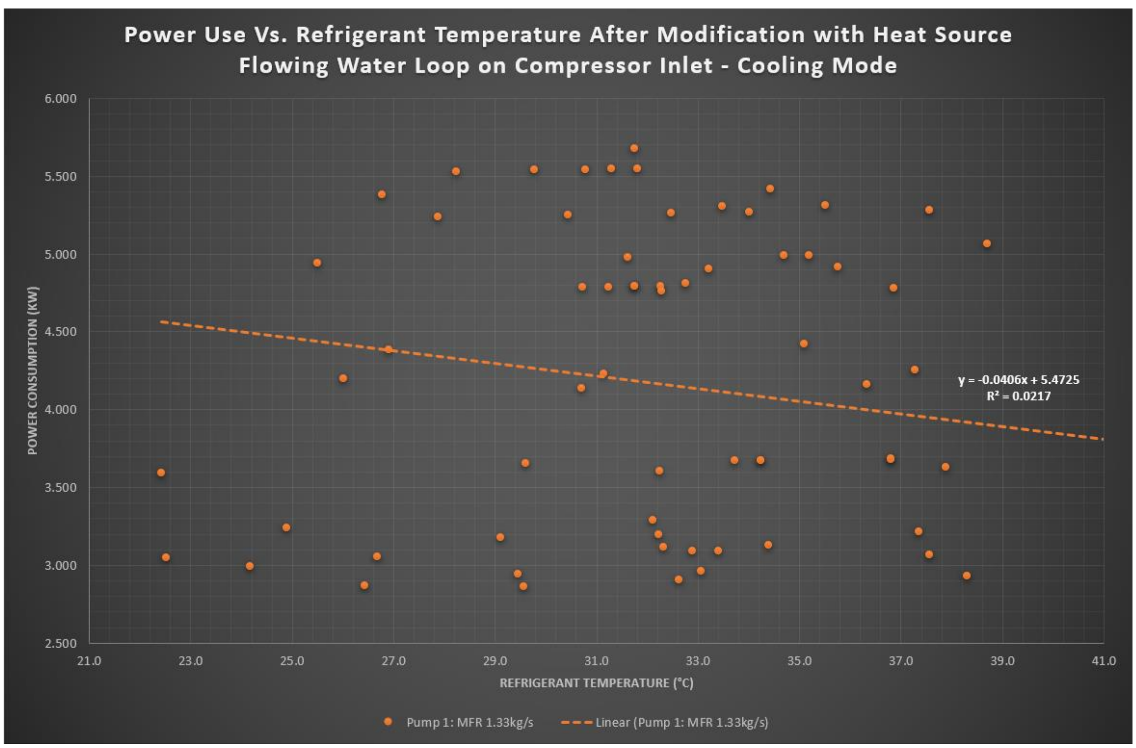

Figure 27. Power consumption of household a/c system after applying the heat source flowing water loop modification (water pump of mass flow rate 1.33kg/s) to the compressor inlet - Cooling Mode

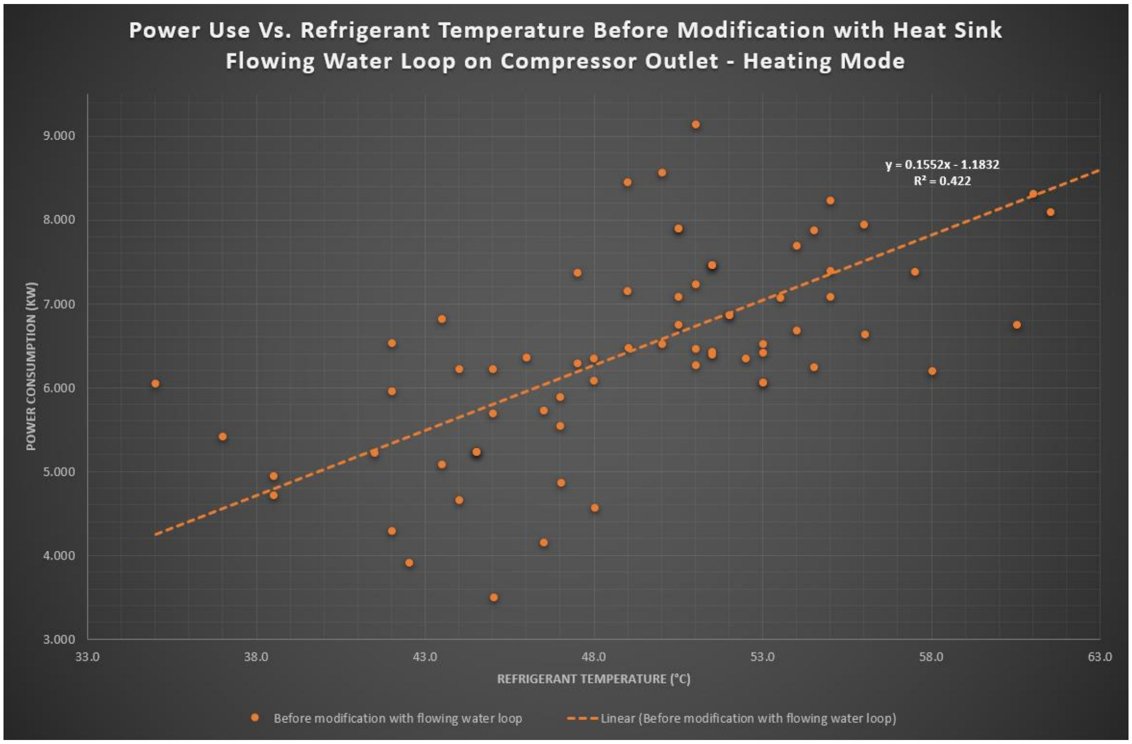

Figure 28. Power consumption of household a/c system before applying the heat sink flowing water loop modification to the compressor outlet - Heating Mode

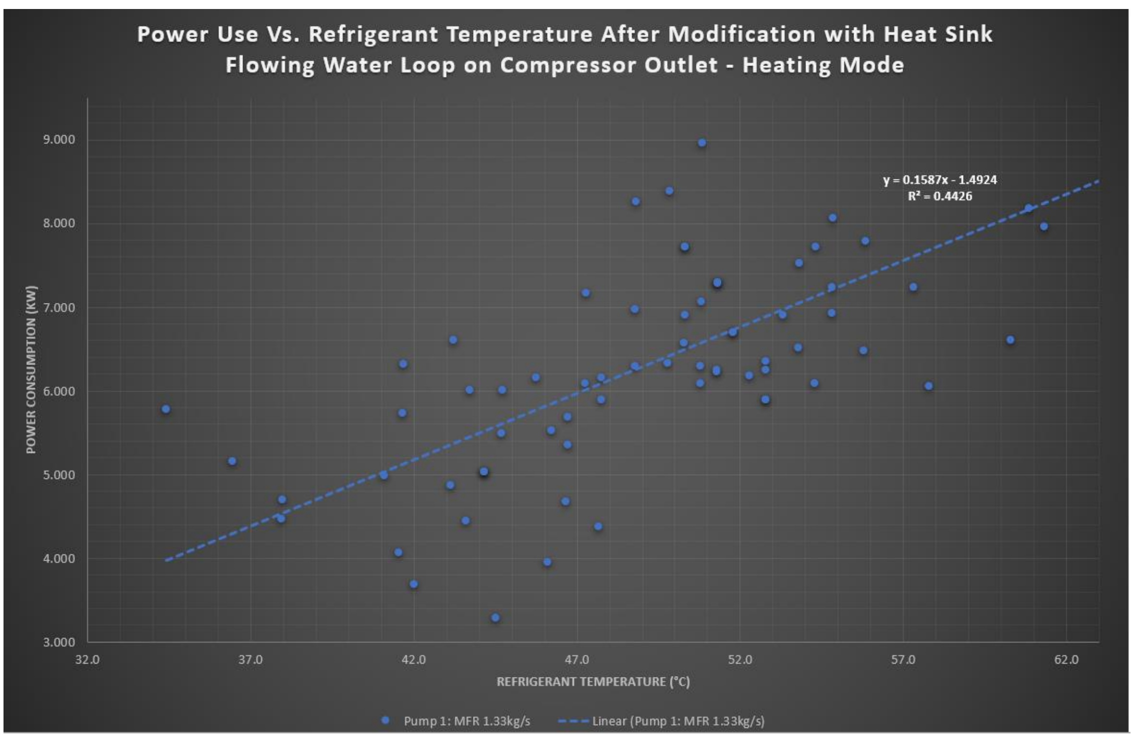

Figure 29. Power consumption of household a/c system after applying the heat sink flowing water loop modification to the compressor outlet - Heating Mode

From table 14 and figures 26 and 27, it is evident that the results after the modification with the 1.33kg/s water pump in cooling mode have shown an average increase in refrigerant temperature at the compressor inlet of 4.67°C and an average decrease in power consumption of 0.797kW per hour. This is a 16.11% reduction in power use for the household a/c system with a flowing water loop from a heat source (solar hot water system) in cooling mode using extrapolated data from MATLAB, which is a significant power saving for the unit. Figure 30 below shows the power consumption before modification and the power consumption after modification versus the refrigerant temperature at the compressor inlet using the heat source in cooling mode. The figure shows that before modification, as refrigerant temperature increases, power consumption increases. However, after the heat source is applied to the compressor inlet, as refrigerant temperature increases, power consumption decreases. This is due to the compressor performing work to increase the pressure and temperature of the refrigerant to the required level, but as the heat source increases the temperature of the refrigerant prior to entering the compressor, the compressor does less work to get the refrigerant temperature/pressure to the required level.

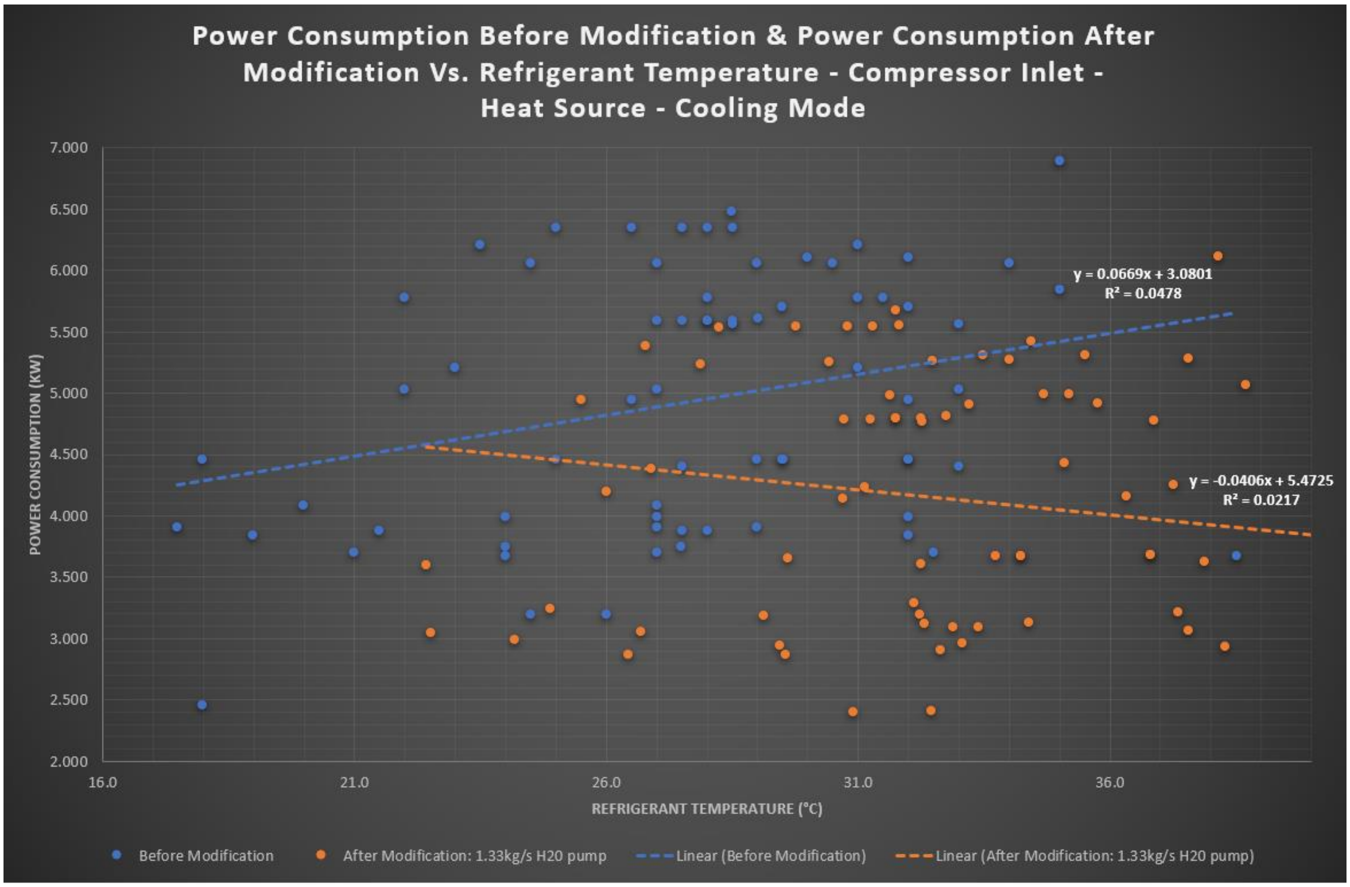

Figure 30. Power Consumption Before Modification & Power Consumption After Modification Vs. Refrigerant Temperature - Compressor Inlet - Heat Source - Cooling Mode

From table 15 and figures 28 and 29, it is evident that the results after the modification with the 1.33kg/s water pump in heating mode have shown an average decrease in refrigerant temperature at the compressor outlet of 0.3°C and an average decrease in power consumption of 0.179kW per hour. This is a 2.74% reduction in power use for the household a/c system with a flowing water loop from a heat sink (residential pool) in heating mode using extrapolated data from MATLAB, which does not represent a great power saving but could be substantial over the entire winter period. Figure 31 below shows the power consumption before modification and the power consumption after modification versus the refrigerant temperature at the compressor outlet using a heat sink in heating mode. The figure shows that as the refrigerant temperature at the compressor outlet increases, the power consumption increases, as the compressor has a higher work input to increase the pressure and temperature of the refrigerant to the required level. By applying the heat sink at this point in the cycle, the power reduction is slightly reduced by reducing the refrigerant temperature, effecting the cycle/system upstream and therefore reducing the amount of work input from the compressor.

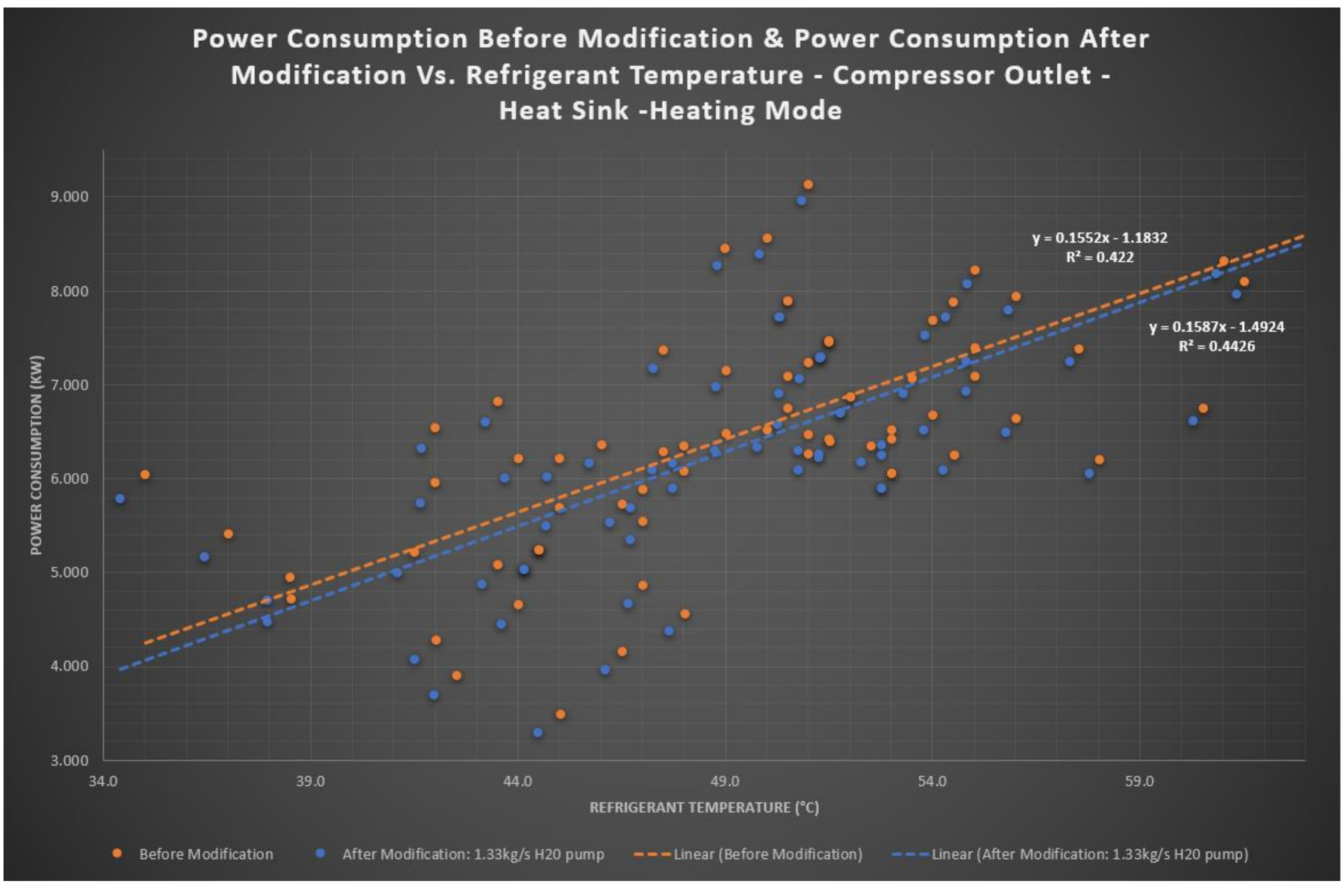

Figure 31. Power Consumption Before Modification & Power Consumption After Modification Vs. Refrigerant Temperature - Compressor Outlet - Heat Sink -Heating Mode

Figures 32 and 33 below show the power consumption with modification versus the power consumption without modification for both cooling and heating mode. These graphs show a very linear fit indicating the formula for this straight line could be used to model the outcome of this design. This does not account for flowing water loop pump power consumption (refer Feasibility Assessment - Section 4.5.1).

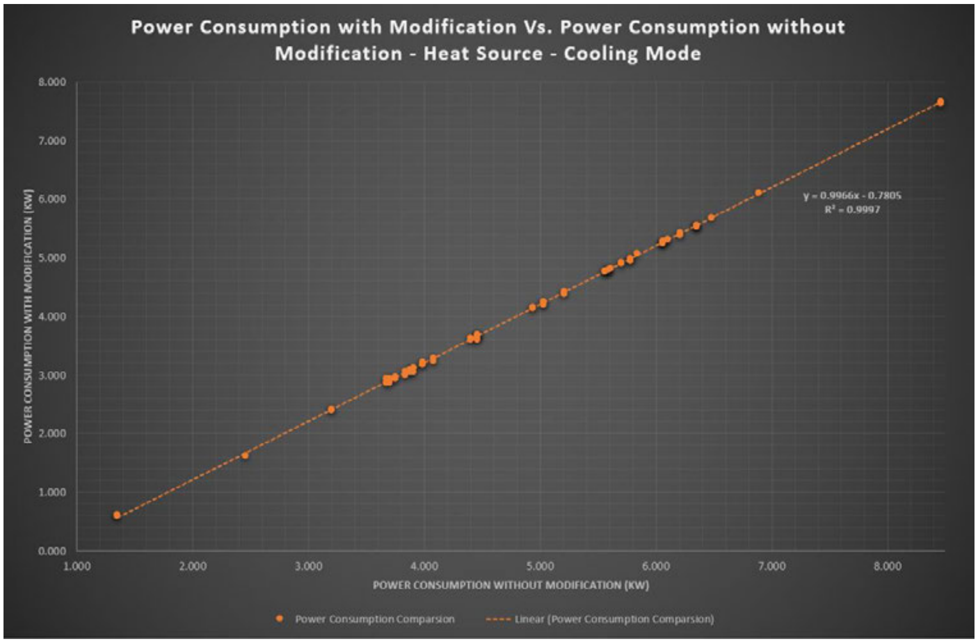

Figure 32. Power Consumption with Modification Vs. Power Consumption without Modification - Heat Source - Cooling Mode

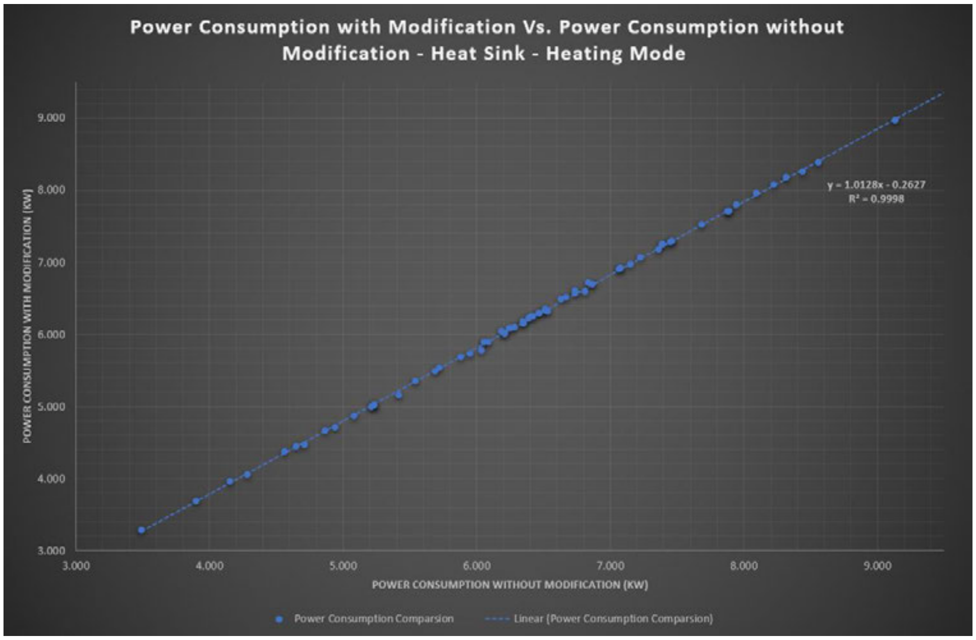

Figure 33. Power Consumption with Modification Vs. Power Consumption without Modification - Heat Sink - Heating Mode

4.4 Validation of Household a/c Extrapolation with Box a/c results

To appropriately validate the household a/c extrapolation results from MATLAB with the box a/c results, the boundary conditions must be stated. For the box a/c, the boundary condition is a fixed temperature (the hot and cold packs), but for the household a/c system, the boundary condition is a fixed heat transfer rate. Therefore, to validate these results effectively the change in refrigerant temperatures of the box a/c results before and after the hot/cold pack application must be compared with the change in refrigerant temperature of the household a/c system before and after the flowing water loop modification. If the change is refrigerant temperature is similar, it will give a good indication that the extrapolation is valid. For the purposes of this validation, the household a/c system flowing water loop refrigerant temperatures used were from the 1.33kg/s water pump, as this resulted in the greatest heat transfer and power reduction. Also, an assumed water temperature change of 0.01K through the flowing water loop was used.

4.4.1 Heating Mode Validation

For heating mode, the box a/c system had an average sink temperature of 19.214°C before the cold pack was applied and after the cold pack was applied to the compressor outlet pipe, the refrigerant temperature was 19.210°C. This is a difference of 0.004°C before and after the cold pack is applied. For the household a/c system, the average refrigerant temperature before the flowing water loop modification was 49.7°C and after the modification was 49.4°C. This is a difference of 0.3°C before and after modification. Therefore, the difference between the refrigerant temperatures of the two systems in heating mode was 0.296°C, which was consistent based off the assumptions made. The extrapolation of the household a/c system is therefore validated with the box a/c system results in heating mode.

4.4.2 Cooling Mode Validation

For cooling mode, the box a/c system had an average source temperature of 45.70°C before the hot pack was applied and after the hot pack was applied to the compressor inlet pipe, the refrigerant temperature was 45.749°C. This is a difference of 0.049°C before and after the hot pack is applied. For the household a/c system, the average refrigerant temperature before the flowing water loop modification was 27.892°C and after the modification was 28.359°C. This is a difference of 0.467°C before and after modification. Therefore, the difference between the refrigerant temperatures of the two systems in cooling mode was 0.418°C, which was once again consistent based off the assumptions made. The extrapolation of the household a/c system is therefore validated with the box a/c system results in cooling mode.

4.5 Feasibility Assessment

The feasibility of this design was primarily assessed from a cost/benefit perspective with capital and energy costs being presented and summed to produce an overall cost and breakeven time. As the research project was performed using hot and cold packs to simulate the heat sink/source, the feasibility assessment was carried out based off the household air conditioning system including a flowing water loop from each source/sink to the applicable refrigerant pipes. Further to this, any assumptions, limitations and issues with the proposed design was identified in this assessment.

4.5.1 Flowing Water Loop Design

Various options are possible for installing a flowing water loop to an existing household reverse cycle a/c system from a heat source (solar hot water system) and heat sink (residential pool or water tank). As the most effective power reduction in the results from the extrapolated data in both heating and cooling mode was using the 1.33kg/s (80LPM) water pump, this pump use will be assessed below. Furthermore, one pump could be used for both heating and cooling mode or two separate pumps, one for cooling and one for heating mode. This would increase the initial capital costs of the design but reduce the overall power consumption of the device and be more cost effective over time. Assumptions were made on labour prices and time to install the system; however, Australian retail prices were used for the pump, insulated copper piping and electricity rates.

Pump: 80LPM (1.33kg/s) Water Pump – Cooling Mode – Heat Source (Solar hot water system)

Capital Costs:

- Davey SJ60-08PC Silver Series Jet Pump - **$399.00** (Davey SJ60-08PC Silver Series Jet Pump (Flow 80 LPM), 2020)

- Insulated piping (10m) - **$93.18** (5M Air Conditioner Pair Coil Tube 1/4” 3/8” Insulated Copper Pipes R32/R410A, 2020)

- **Labour:** $73.00 per hour = 3 hours work = **$219.00**

Total Capital Cost: $711.18 for 1.33kg/s Water Pump installation

Energy Cost Before and After Modification (refer table 16):

Table 16: Energy cost analysis of the flowing water loop using 80LPM (1.33kg/s) water pump – Cooling Mode – Heat Source (Solar hot water system)

Water pump energy consumption	0.9kWh
Hot water system energy consumption without Solar (worst case scenario – raining, etc.)	2.3kWh
Electricity supply charge / usage charge	$1.282 per day / $0.2772 per kWh
a/c power consumption before modification (cooling)	4.95kWh
Energy cost before modification (cooling) – 8hr period	(4.95*8)+(0.2772*8)+1.282 = **$43.10**
a/c power consumption after modification (cooling)	4.15kWh
Energy cost after modification (cooling mode) – 8hr period	(4.15*8)+(0.2772*8)+(0.9*8)+(2.3*8)+1.282 = **$62.30 without Solar Hot Water** (4.15*8)+(0.2772*8)+(0.9*8)+1.282 **= $43.90 with Solar Hot Water**

Energy Saving:

$43.10 - $62.30 = -$19.20 per 8hr period (1 day) - Without use of Solar Hot Water

$43.10 - $43.90 = -$0.80 per 8hr period (1 day) - With use of Solar Hot Water

As this is not an energy saving using the 1.33kg/s water pump, it would not be financially feasible to use this option. Using the 0.83kg/s water pump, the results were as follows:

Pump: 50LPM (0.83kg/s) Water Pump – Cooling Mode – Heat Source (Solar hot water system)

Capital Costs:

- Davey SJ35-04PC Silver Series Jet Pump - **$310.00** (Davey SJ35-04PC Silver Series Jet Pump (Flow 50 LPM), 2020)

- Insulated piping (10m) - **$93.18** (5M Air Conditioner Pair Coil Tube 1/4" 3/8" Insulated Copper Pipes R32/R410A, 2020)

- **Labour:** $73.00 per hour = 3 hours work = **$219.00**

Total Capital Cost: $622.18 for 0.83kg/s Water Pump installation

Energy Cost Before and After Modification (refer table 17):

Table 17: Energy cost analysis of the flowing water loop using 50LPM (0.83kg/s) water pump – Cooling Mode – Heat Source (Solar hot water system)

Water pump energy consumption (kW)	0.37kWh
Hot water system energy consumption without Solar (worst case scenario – raining, etc.)	2.3kWh
Electricity supply charge / usage charge	$1.282 per day / $0.2772 per kWh
a/c power consumption before modification (cooling)	4.95kWh
Energy cost before modification (cooling) – 8hr period	(4.95*8)+(0.2772*8)+1.282 = **$43.10**
a/c power consumption after modification (cooling)	4.44kWh
Energy cost after modification (cooling mode) – 8hr period	(4.44*8)+(0.2772*8)+(0.37*8)+(2.3*8)+1.282 = **$60.38 without Solar Hot Water** (4.44*8)+(0.2772*8)+(0.37*8)+1.282 **= $41.98 with Solar Hot Water**

<u>Energy Saving:</u>

$43.10 - $60.38 = -$17.28 per 8hr period (1 day) - Without use of <u>Solar</u> Hot Water

$43.10 - $41.98 = $1.12 per 8hr period (1 day) - With use of <u>Solar</u> Hot Water

This is not an energy saving if no solar hot water is used with the 0.83kg/s water pump, however, it would save $1.12 per day if solar hot water is used.

Based off the total capital costs, it would take: $622.18 / $1.12 = <u>555.5 days</u> (1.52 years) to breakeven and start saving money using the design with the 0.83kg/s Water Pump using solar hot water (best case – minimum time) in cooling mode. These calculations have not considered the household having solar panels on the roof of the property. The additional cost of the water pump energy consumption could essentially be disregarded if this design was used throughout the daylight hours, further reducing overall costs and time to breakeven.

One of the main concerns with using the solar hot water system in a flowing water loop is the volume of the hot water system versus the flow rate of the water pump used. The current household hot water system is 300 litres; using an 80 litre per minute jet pump will cycle the entire contents of the hot water system through the loop in 3.75 minutes and 6 minutes for the 50 litre per minute jet pump. If the lower flow rate pump was used (25LPM), the entire contents of the hot water system would cycle through the loop in 12 minutes. This would not allow the hot water system to reach its required temperature to maintain the power reduction required for the flowing water loop. As well as not enough hot water for general household use. Other options could include using a dedicated solar hot water system of larger volume for the flowing water loop and/or having a closed loop system entailing an increase in solar collectors or using an evacuated tube solar hot water system. However, this would drastically increase the capital cost of the system by up to $5000.00 and increase the breakeven time under optimal solar conditions to 5020 days (13.75 years). Depending on location and climate conditions of the region where the unit is located, using such a design on the hot water system (particularly for cooling mode) may not be feasible, especially with the large initial capital costs. It may be most beneficial to have this design solely on the heating side of the a/c system.

Pump: 80LPM (1.33kg/s) Water Pump – Heating Mode – Heat Sink (Residential Pool)

Capital Costs:

- Davey SJ60-08PC Silver Series Jet Pump - **$399.00** (Davey SJ60-08PC Silver Series Jet Pump (Flow 80 LPM), 2020)

- Insulated piping (10m) - **$93.18** (5M Air Conditioner Pair Coil Tube 1/4" 3/8" Insulated Copper Pipes R32/R410A, 2020)

- **Labour:** $73.00 per hour = 2 hours work = **$146.00**

Total Capital Cost: $638.18 for 1.33kg/s Water Pump installation

Energy Cost Before and After Modification (refer table 18):

Table 18: Energy cost analysis of the flowing water loop using 80LPM (1.33kg/s) water pump – Heating Mode – Heat Sink (Residential Pool)

Water Pump Energy Consumption (kW)	0.9kWh
Electricity supply charge / usage charge	$1.282 per day / $0.2772 per kWh
a/c power consumption before modification (heating)	6.53kWh
Energy cost before modification (heating) – 8hr period	(6.53*8)+(0.2772*8)+1.282 = **$55.74**
a/c power consumption after modification (heating)	6.35kWh
Energy cost after modification (heating mode) – 8hr period	(6.35*8)+(0.2772*8)+(0.9*8)+1.282 = **$61.50**

Energy Saving:

$55.74 - $61.50 = -$5.76 per 8hr period (1 day)

As this is not an energy saving using the 1.33kg/s water pump, it would not be financially feasible to use this option. Using the 0.83kg/s water pump, the results were as follows:

Pump: 50LPM (0.83kg/s) Water Pump – Heating Mode – Heat Sink (Residential Pool)

Capital Costs:

- Davey SJ35-04PC Silver Series Jet Pump - **$310.00** (Davey SJ35-04PC Silver Series Jet Pump (Flow 50 LPM), 2020)
- Insulated piping (10m) - **$93.18** (5M Air Conditioner Pair Coil Tube 1/4" 3/8" Insulated Copper Pipes R32/R410A, 2020)
- **Labour:** $73.00 per hour = 2 hours work = **$146.00**

Total Capital Cost: $549.18 for 0.83kg/s Water Pump installation

Energy Cost Before and After Modification (refer table 19):

Table 19: Energy cost analysis of the flowing water loop using 50LPM (0.83kg/s) water pump – Heating Mode – Heat Sink (Residential Pool)

Water Pump Energy Consumption (kW)	0.37kWh
Electricity supply charge / usage charge	$1.282 per day / $0.2772 per kWh
a/c power consumption before modification (heating)	6.53kWh
Energy cost before modification (heating) – 8hr period	(6.53*8)+(0.2772*8)+1.282 = **$55.74**
a/c power consumption after modification (heating)	6.42kWh
Energy cost after modification (heating mode) – 8hr period	(6.42*8)+(0.2772*8)+(0.37*8)+1.282 = **$57.82**

Energy Saving:

\$55.74 - \$57.82 = -\$2.08 per 8hr period (1 day)

As this is not an energy saving using the 0.83kg/s water pump, it would not be financially feasible to use this option. Using the 0.42kg/s water pump, the results were as follows:

Pump: 25LPM (0.42kg/s) Water Pump – Heating Mode – Heat Sink (Residential Pool)

Capital Costs:

- Reefe RP1500 Pond & Water Feature Pump - **\$120.00** (Reefe RP1500 Pond & Water Feature Pump (Flow 25 LPM), 2020)

- Insulated piping (10m) - **\$93.18** (5M Air Conditioner Pair Coil Tube 1/4" 3/8" Insulated Copper Pipes R32/R410A, 2020)

- **Labour:** \$73.00 per hour = 2 hours work = **\$146.00**

Total Capital Cost: \$359.18 for 0.42kg/s Water Pump installation

Energy Cost Before and After Modification (refer table 20):

Table 20: Energy cost analysis of the flowing water loop using 25LPM (0.42kg/s) water pump – Heating Mode – Heat Sink (Residential Pool)

Water Pump Energy Consumption (kW)	0.02kWh
Electricity supply charge / usage charge	\$1.282 per day / \$0.2772 per kWh
a/c power consumption before modification (heating)	6.53kWh
Energy cost before modification (heating) – 8hr period	(6.53*8)+(0.2772*8)+1.282 = **\$55.74**
a/c power consumption after modification (heating)	6.48kWh
Energy cost after modification (heating mode) – 8hr period	(6.48*8)+(0.2772*8)+(0.02*8)+1.282 = **\$55.50**

Energy Saving:

$55.74 - $55.50 = $0.24 per 8hr period (1 day)

This is a very small energy saving using the 0.42kg/s water pump, with a total of $0.24 saved per day. Based off the total capital costs, it would take: $359.18 / $0.24 = 1497 days (4.1 years) to breakeven and start saving money using the design with the 0.42kg/s Water Pump in heating mode.

These calculations have not considered the household having solar panels on the roof of the property. The additional cost of the water pump energy consumption could essentially be disregarded if this design was used throughout the daylight hours, further reducing overall costs and time to breakeven by 599 days (1.7 years).

In heating mode, cycling water through a residential pool or large water tank would see a much more effective result for the design as these heat sinks are much larger thermal reservoirs (pool - 30,000L or water tank - 10,000L) that can supply or absorb finite amounts of heat without undergoing any change in temperature. Over an 8-hour day using the flowing water loop, the pool would change in temperature only a very small amount, maintaining the efficiency and energy savings of the design.

Overall, the 50LPM (0.83kg/s) water pump in cooling mode using the heat source (solar hot water system) saw the most savings financially, with $1.12 saved per day if solar hot water is used and 555.5 days (1.52 years) to breakeven and start saving money using the design. The 25LPM (0.42kg/s) water pump in heating mode using the heat sink (residential pool) saw the most savings financially (although small), with $0.24 saved per day and 1497 days (4.1 years) to breakeven and start saving money using the design. These two options could be used individually on the household a/c system or together depending on household requirements and the climate where the system in located.

5. Conclusions

5.1 Findings/Outcomes of Project

The aims and objectives of this research project were formed using an achievable project plan with defined goals and results to be attained. The design and methodology throughout the project year endeavoured to produce quality results that would have tangible outcomes for the dissertation and university. A review was performed of the current energy market in Australia relating to existing household reverse cycle air conditioning systems and the use of water as a source or sink of low-grade heat. It was found that several small-scale designs relating to these systems had been previously implemented, mostly overseas. A design was produced to simulate heat sink and source temperatures from water sources.

Using a solar hot water system as a source of low-grade heat and a residential pool as a sink of low-grade heat, these temperatures were simulated to improve the efficiency/energy consumption of a box air conditioning system. By applying the sink temperature to the compressor outlet refrigerant pipe in heating mode on the box air conditioner using a cold pack, a 3.8% power reduction was realised. By applying the source temperature to the compressor inlet refrigerant pipe in cooling mode on the box air conditioner, a 4.7% power reduction was realised. Through MATLAB calculations, the refrigerant temperature, mass flow rate and heat transfer rate were determined for the system in the required location (compressor inlet or outlet for cooling and heating respectively).

The data collected over the course of the project year was used in several MATLAB calculations to extrapolate the results to calculate what the refrigerant temperature and power consumption would be if the flowing water loop was used on the household system. The power consumption, mass flow rate, refrigerant temperature and heat transfer was established before and after the modification. The results were a 2.74% reduction in power use for the household a/c system with a flowing water loop from a heat sink (residential pool) in heating mode and a 16.11% reduction in power use for the household a/c system with a flowing water loop from a heat source (solar hot water system) in cooling mode. These results are from the highest performing water pump for the flowing water loop with a mass flow rate of 1.33kg/s. Unfortunately, this pump was not feasible for use on the current household setup but may be suitable on other systems.

The feasibility assessment included a broad overview of capital and energy costs for the design to be retrofitted to an existing household reverse cycle air conditioning system in cooling and heating mode with some assumptions made due to not physically installing the design. A cost saving and breakeven time was established and determined that for cooling mode using the solar hot water system as a source of low-grade heat with a 50LPM (0.83kg/s) water pump for the flowing water loop saved $1.12 per day with 8 hours use. This equated to a breakeven time of 555.5 days (1.52 years) to start saving money using the design. For heating mode, using the 25LPM (0.42kg/s) water pump for the flowing water loop to a residential pool as a sink of low-grade heat saw the most savings financially (although small), with $0.24 saved per day and 1497 days (4.1 years) to breakeven and start saving money using the design. These two options for heating and cooling could be used individually on the household air conditioning system or together depending on household requirements and the climate where the system is located.

The validation of the household air conditioning system's extrapolated data from MATLAB was found to be consistent with that of the box air conditioning system's results. In heating mode, there was a difference between the refrigerant temperatures of the two systems of 0.296°C, based off the assumptions presented. In cooling mode, there was a difference between the refrigerant temperatures of the two systems of 0.418°C, based off the assumptions presented. The extrapolation of the household air conditioning system was therefore validated and deemed consistent with the box air conditioning system's results in both heating and cooling mode.

The primary objectives of the research project were achieved in line with the project specification with the secondary objectives unable to be accomplished in the required time. The overall aim to conduct a feasibility study of improving the efficiency of reverse-cycle air-conditioning systems using water from a solar hot-water system and a residential pool as a source/sink of low-grade heat was met successfully with further work to continue the research if desired. This study could see benefits in the future on a residential and possibly commercial scale, as retrofitting existing household and HVAC systems with this design could see improved energy performance and large cost savings. With a breakeven time for the design of 1.5 years for cooling mode and 4.1 years for heating mode, it is highly recommended that research in this area continues and specifically the topic of using water as a low-grade heat sink/source to improve the efficiency of reverse cycle air conditioning systems.

5.2 Further Work and Recommendations

Following this dissertation, further research or projects could be undertaken to assess the feasibility of such a design more precisely. The project identified certain limitations and assumptions throughout the design and testing phase that were accounted for if possible. These limitations reduced the overall accuracy of the results due to time, resource and financial constraints. The following options could be designed and tested to further the current research and provide a higher level of accuracy and viability for application and implementation to an existing household system on a residential scale.

The current research utilised hot and cold packs to simulate the heat sink and source temperature on the box air conditioner and this sink or source temperature was used in extrapolation calculations for the household air conditioning system. To move the research to the next phase would entail using a flowing water loop from the heat sink and source to the appropriate refrigerant lines in the air conditioning system. This would require a small pump to create flow for the loop with insulation wrapping the tubing from the source/sink location to the a/c unit (as discussed in Section 4.4 and 4.5). The tubing would then be wound/coiled around the refrigerant pipe in the required locations to provide maximum heat transfer. The locations being the length of the compressor inlet pipe for the hot water source (solar hot water system) in cooling mode and the length of the compressor outlet pipe for the cold-water sink (pool) in heating mode. By maximising the length of pipe covered by the tubing, the heat transfer between the refrigerant and source/sink would be increased further. The small pump would create a certain amount of power consumption (refer Section 4.5); however, it is hypothesised from the extrapolated results and feasibility assessment in the project that there would still be energy savings and much larger efficiency increases in the system.

Due to the time frames of the design testing and project completion, no data was able to be captured outside of the July, August and September period. It is recommended to conduct further testing over the course of a year using the above flowing water loop or similar. This will assess the use of the design in varying seasons, temperatures and environmental conditions. This could be further expanded to conduct the testing in different locations of the state/country under varying humidity's, barometric pressures and temperatures differentials. On wider scale implementation, this would be essential to assess whether the design would be suitable in different parts of the year and at what times of the year. As discussed in previous sections, the design may not be feasible in the shoulder seasons when inside ambient air temperatures are similar to outside ambient air temperatures.

The options for ways forward to ensure precise and effective results are presented above as only some of the possibilities available to one looking to further this research. The prospects of this innovation could have large impacts on air conditioning systems and energy markets in the future. Through further studies and feasibility assessments it is hoped that the suggested research could be performed over a longer duration and with greater resource allocation and financial support to assist in accuracy of the results. The future of innovative air conditioning technologies is bright and will see efficient and creative designs and systems like this brought to market on both a residential and commercial scale. It is hoped that the presented research and findings in this dissertation will form part of the body of information that will assist in creating a more sustainable and energy efficient air conditioning market in the Asia-Pacific region.

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7. Appendix A – Project Specification

For: Michael Osgood

Title: Feasibility assessment of using water storage sources to improve the efficiency of air-conditioning systems.

Major: Mechanical Engineering

Supervisor: Andrew Wandel

Enrolment: ENG4111 – ONL S1, 2020
ENG4112 – ONL S2, 2020

Project Aim: Feasibility study of improving the efficiency of reverse-cycle air-conditioning systems using water from a solar hot-water system and a pool as a source/sink of low-grade heat.

Programme: Version 1, 18th March 2020

1. Review existing household reverse-cycle air-conditioning systems and evaluate any current use of heat sinks/sources in this area.

2. Research existing heat exchanger systems relating to reverse cycle air-conditioning systems to evaluate the feasibility of integrating heat sinks/sources into existing systems.

3. Perform data collection over all seasons on a pool, solar hot water system and reverse-cycle air-conditioning system to assess how temperatures and refrigerant mass flow rate changes under a variety of environmental conditions.

4. Design, build and test an air-conditioning system using pool water as a heat sink and solar hot water as a heat source.

5. Compare improvements in efficiency between the designed and built air-conditioning system using a heat sink/source and a standard air-conditioning system.

6. Investigate any additional energy/capital costs associated with installing the designed heat-exchanger system to an existing air-conditioning system.

If time and resources permit:

7. Field testing of the designed heat exchanger system on an existing reverse-cycle air-conditioning system.

8. Compare and evaluate results from design testing with results from field testing.

9. Make recommendations on refinement of current heat exchanger design for further efficiency gains.

8. Appendix B – Timeline/Gantt Chart and Plan

8.1 Timeline/Gant Chart and Key Tasks

Creating a timeline and identifying key tasks within the project were vital to the success of the project. As such, a Gantt chart and Plan was created and is located on the following pages, displaying the timeline to achieve key tasks within the project. This was an important factor for this study as a body of knowledge must be ascertained and understood before moving to the next phase and maintaining a timeline ensured the project was completed by the due date.

See below for a list of key tasks identified for the project duration:

- Project reviewed on a regular basis to ensure the goals and objectives are being met as required. Occurred every 6 weeks as part of moving to the next phase of the project.
- A series of meetings/emails occurred with the project supervisor throughout the duration of the project. Meetings occurred approximately once or twice every 6 weeks, to coincide with the project review and phase change. These were in the form of email updates/guidance or zoom sessions as stated above.
- Phase changes occur every 6 weeks of the semester and with this, analysis and documentation of tasks undertaken occurred and were used in the next phase of the project.
- ENG4903 was used to highlight and ensure adequate preparation of the presentation and dissertation.
- Draft dissertation due date (09/09/2020)
- Final dissertation due date (15/10/2020)

8.2 Project Plan

Project planning was crucial to the success of the project and went hand in hand with the Gantt Chart and key tasks described above. A weekly plan was established to maintain the schedule and to ensure all tasks were completed by the due date and an outcome was achieved (refer Appendix B – 8.2). The project was scheduled over the duration of the year 2020, semesters 1 and 2, however, due to COVID-19 restrictions throughout the first semester and lack of face-to-face interaction with the project supervisor some delays did occur. These delays were able to be caught up during the recess periods of the semester and prior to semester 2.

Phase 1	Project Allocation / Specification / Planning
1A **Week 1-2**	- ***Project approval from USQ.*** ****DUE: 04th March 2020**** - Project initiation confirm supervisor (communication frequency, etc.). - Discuss scope and processes.
1B **Week 1-28**	- Data Collection – Daily temperatures for pool, air, hot water and air-conditioning refrigerant lines.
1C **Week 2-3**	- Discuss any major challenges/issues with supervisor and work on solutions and workarounds for project. - Define scope of project, including primary aims.
1D **Week 3-4**	- Begin Project Specification, Project Plan and Project Resource requirements. - Submit draft Project Specification to supervisor.
1E **Week 4-10**	- Literature review on existing household reverse-cycle air-conditioning systems and existing heat exchanger systems. - Research how to determine mass flow rate of refrigerant. - Research how to determine efficiency gains through energy reductions.
1F **Week 4**	- Finalise Project Specification, Project Plan, Project Resource requirements and supplementary information. - ***Submit Project Specification, Project Plan, Project Resource requirements and supplementary information to Study Desk.*** ****DUE: 18th March 2020****
1G **Week 5-6**	- Review risk assessment and compliance documentation to ensure safety and quality of research. - Amend/adjust project plan post supervisor feedback and setup weekly workload schedule.
1H **Week 6**	- Conduct project review, finalise phase requirements and consult supervisor for any further advice and progress update.
Phase 2	**Design, Build and Test**

2A **Week 7-9**	- Perform research in conjunction with literature review on how to build a car air-conditioning system that can incorporate the heat sink/source. - Design the air-conditioning system.
2B **Week 8-10**	- Acquire parts and begin building air-conditioning system
2C **Week 8-12**	- Begin Progress Report. - Draft Progress Report to supervisor. - Continue build of air-conditioning system.
2D **Week 10-12**	- Complete building a/c system incorporating heat sink/source. - Begin testing/recording results using heat sink/source in system. - Feedback from supervisor regarding Progress Report.
2E **Week 12-13**	- Conduct project review finalise phase requirements and consult supervisor for advice and progress update. - Finalise Progress Report.
Phase 3	**Progress Report / Draft Dissertation / Analysis**
3A **Week 14**	- ***Submit Progress Report to Study Desk.*** ****DUE: 27th May 2020****
3B **Week 15-16**	- Begin analysing results from a/c design and compare to standard system. - Identify any efficiencies or possible changes in design to increase efficiencies.
3C **Week 16-20**	- Begin drafting parts of dissertation.
3D **Week 17**	- Conduct project review, finalise phase requirements and consult supervisor for advice and progress update.
Phase 4	**Compare / Analysis Efficiencies / Investigate Costs / Continue Dissertation**
4A **Week 21-26**	- Begin comparison efficiency within the built air-conditioning system using heat sink/source with a standard air-conditioning system.
4B **Week 21-29**	- Work on Partial Draft Dissertation.

4C **Week 23-28**	- Analyse efficiencies of the built air-conditioning system using heat sink/source.
4D **Week 24-28**	- Investigate energy/capital costs associated with installing the designed heat-exchanger system to an existing air-conditioning system.
4E **Week 25-26**	- Continue drafting dissertation and preparing presentation for ENG4903 – Professional Practice 2. - Submit draft dissertation as it stands to supervisor for review.
4F **Week 26**	- Conduct project review, finalise phase requirements and consult supervisor for advice and progress update.
Phase 5	**Partial Draft Dissertation / ENG4903 – Professional Practice 2 Presentation**
5A **Week 27-29**	- Finalise Partial Draft Dissertation. - Address feedback areas of Partial Draft Dissertation from supervisor. - Work on results/findings/recommendations.
5B **Week 27-30**	- Continue project presentation for ENG4903 – Professional Practice 2. - Submit to supervisor for review. - Address areas of feedback in presentation. - Practice presentation for ENG4903.
5C **Week 29**	- ***Submit Partial Draft Dissertation to Study Desk.*** ****DUE: 09th September 2020****
5D **Week 31**	- **Complete and carry out Project Presentation at USQ Toowoomba / Springfield for ENG4903 – Professional Practice 2.** ****DUE: 21st-25th September 2020****
5E **Week 32**	- Conduct project review, finalise phase requirements and consult supervisor for advice and progress update.
Phase 6	**Partial Draft Dissertation Feedback / Findings / Recommendations / Final Dissertation submission**
6A **Week 33-34**	- Compile all data, documentation and results to begin final stages of project/dissertation.

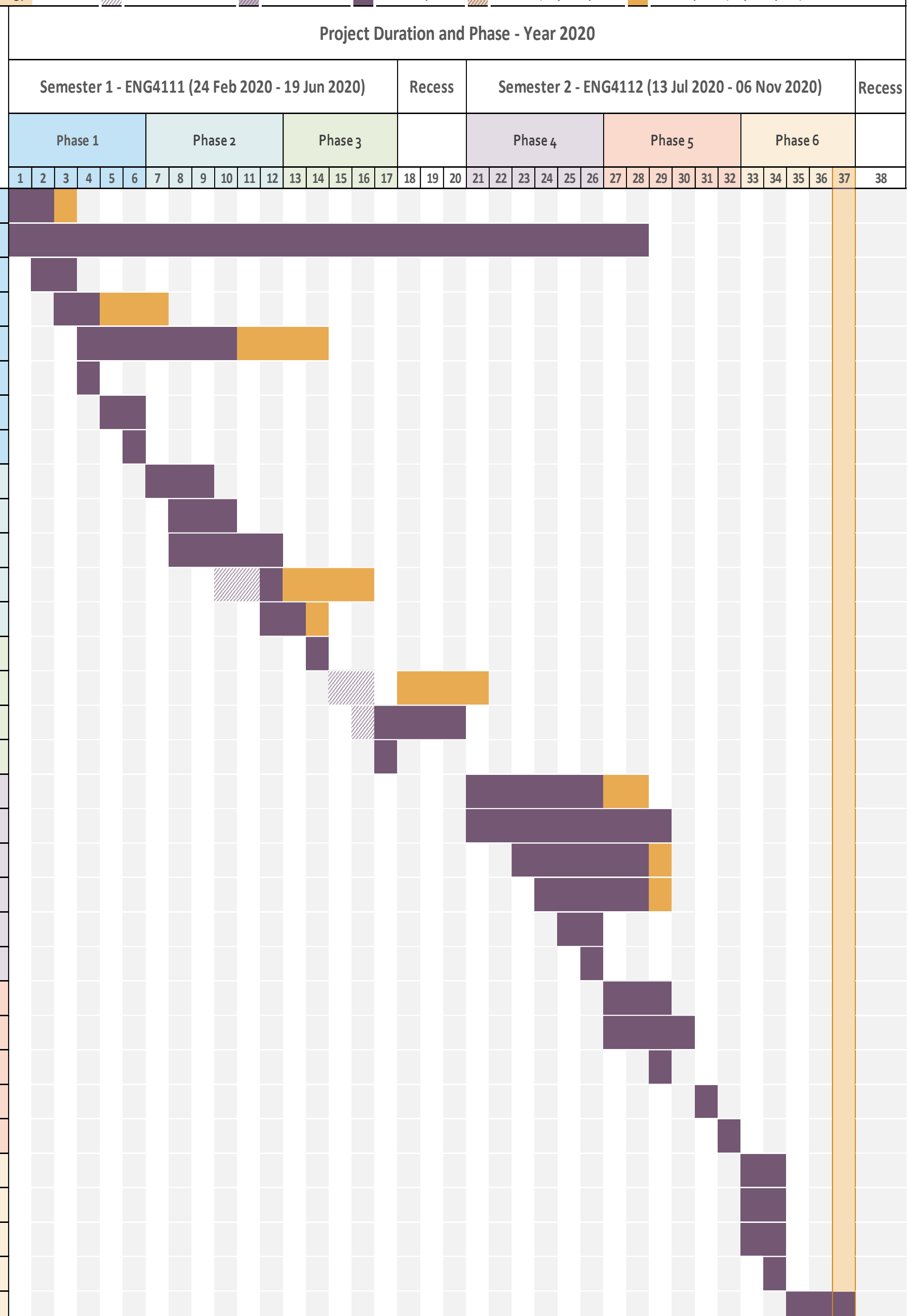
6B **Week 33-34**	- Attain findings and conduct discussion with supervisor for dissertation.
6C **Week 33-34**	- Review feedback from draft dissertation submission and amend as required. - Finalise Final Dissertation ready for submission. - Final feedback from supervisor.
6D **Week 34**	- ***Submit Final Dissertation with conclusions and possible future work to be carried out in the area to Study Desk.*** ****DUE: 15th October 2020****
6E **Week 35-37**	Discuss results and overall process with supervisor/examiner.

SEE GANTT CHART ON FOLLOWING PAGE

Project Plan - Michael Osgood - ENG4111/ENG4112 - Research Report

Period Highlight: 37 | Plan Duration | Actual Start | % Complete | Actual (beyond plan) | % Complete (beyond plan)

ACTIVITY	PLAN START (week)	PLAN DURATION (week)	ACTUAL START (week)	ACTUAL DURATION (week)	PERCENT COMPLETE
1A	1	2	1	3	100%
1B	1	28	1	28	100%
1C	2	2	2	2	100%
1D	3	2	3	5	100%
1E	4	7	4	11	100%
1F	4	1	4	1	100%
1G	5	2	5	2	100%
1H	6	1	6	1	100%
2A	7	3	7	3	100%
2B	8	3	8	3	100%
2C	8	5	8	5	100%
2D	10	3	12	5	100%
2E	12	2	12	3	100%
3A	14	1	14	1	100%
3B	15	2	18	4	100%
3C	16	5	17	4	100%
3D	17	1	17	1	100%
4A	21	6	21	8	100%
4B	21	9	21	9	100%
4C	23	6	23	7	100%
4D	24	5	24	6	100%
4E	25	2	25	2	100%
4F	26	1	26	1	100%
5A	27	3	27	3	100%
5B	27	4	27	4	100%
5C	29	1	29	1	100%
5D	31	1	31	1	100%
5E	32	1	32	1	100%
6A	33	2	33	2	100%
6B	33	2	33	2	100%
6C	33	2	33	2	100%
6D	34	1	34	1	100%
6E	35	3	35	3	100%
			Overall Completion		100%



9. Appendix C – Household a/c System Measurements

Temperature Measurements of Pool, Air, Hot Water System and Air-Conditioning System									
Date	**Water Temp**	**Max Air Temp**	**Min Air Temp**	**HW out**	**HW in**	**AC out**	**AC in**	**Power Use (kW)**	**Comments**
27-Jan-20	31.5	32	21						**All performed with a/c set to 24°c**
28-Jan-20	31	31	21						**All performed with Pool Heater auto on at 27°c**
29-Jan-20	32	35	22						
30-Jan-20	31.5	34	21						
31-Jan-20	31.5	34	22						
01-Feb-20	31.5	34	20						
02-Feb-20	30	34	20						
03-Feb-20	32.5	39	23						
04-Feb-20	30	30	20						
05-Feb-20	28	26	19						
06-Feb-20	27	25	20						
07-Feb-20	28.5	28	21						
08-Feb-20	29	30	22						
09-Feb-20	28.5	29	22						
10-Feb-20	29	29	22						
11-Feb-20	29	30	22						
12-Feb-20	29	29	22						
13-Feb-20	28	28	21						
14-Feb-20	29	31	21						
15-Feb-20	30	33	22						
16-Feb-20	31.5	33	22						
17-Feb-20	32	32	22						
18-Feb-20	32	33	23						
19-Feb-20	32.5	36	22						
20-Feb-20	32	35	21						

21-Feb-20	31.5	31	21						
22-Feb-20	30	29	20						
23-Feb-20	28.5	26	20						
24-Feb-20	28	27	19						
25-Feb-20	29	30	20						
26-Feb-20	29	32	21						
27-Feb-20	30	30	21						
28-Feb-20	30	32	21						
29-Feb-20	29	30	19						
01-Mar-20	31	32	20						
02-Mar-20	31	34	20						
03-Mar-20	31.5	33	22						
04-Mar-20	29	31	21						
05-Mar-20	30.5	30	22						
06-Mar-20	30	30	22	40	32	33	7	5.03	
07-Mar-20	29	30	20	40	32	32	6	3.99	elec. HW on (manual)
08-Mar-20	29.5	28	19	38	28	30.5	8	6.06	
09-Mar-20	27	24	17	30	20	28	10	5.59	elec. HW on (manual)
10-Mar-20	27.5	26	17	38	24.5	26.5	9	6.35	
11-Mar-20	28	28	17	40	27	28	6	3.88	elec. HW on (manual)
12-Mar-20	27.5	27	17	43	21	29.5	8	4.46	
13-Mar-20	27	27	17	48	32	28	17	5.78	elec. HW on (manual)
14-Mar-20	28	28	17	43	32	27	5	3.91	
15-Mar-20	28	28	17	43	29	32.5	6	3.7	
16-Mar-20	28	27	16	47	33.5	32	4.5	4.94	
17-Mar-20	27.5	27	16	44	30	32	6.5	5.7	
18-Mar-20	28	28	15	50	36	33	5	5.56	
19-Mar-20	28.5	28	15	45	29.5	32	4.5	6.1	
20-Mar-20	29	31	17	49	34	33	4	4.4	
21-Mar-20	29	34	20	58	39	38.5	6	3.67	
22-Mar-20	28.5	32	20	48	38.5	34	5	1.35	
23-Mar-20	29	28	19	47	31	32	4	8.46	

24-Mar-20	27.5	27	17	38	26.5	27	11.5	4.08	
25-Mar-20	28	27	17	55	27	32	7	4.46	elec. HW on (manual)
26-Mar-20	28.5	28	18	55.5	34	34	6	6.06	
27-Mar-20	27.5	27	16	50.5	31	27	4	5.59	
28-Mar-20	28	27	17	42	29.5	28	4.5	6.35	elec. HW on (manual)
29-Mar-20	28	28	17	56	30	27.5	9	3.88	
30-Mar-20	28.5	29	18	48.5	33	29	5	4.46	
31-Mar-20	29	31	19	49	29	31.5	5	5.78	
01-Apr-20	28.5	29	17	46	30	29	4.5	3.91	
02-Apr-20	28.5	29	18	51	32	27	20	3.7	
03-Apr-20	29	31	20	48	32	32	6	3.84	
04-Apr-20	29	31	18	42.5	28.5	27.5	6.5	3.75	
05-Apr-20	29	29	15	48.5	32.5	32	7	4.46	
06-Apr-20	28.5	29	16	50.5	34	31	6.5	5.21	
07-Apr-20	28.5	28	16	49.5	34	31	6	6.21	
08-Apr-20	27.5	27	16	42	26	26	7.5	3.2	
09-Apr-20	27.5	27	17	45	27.5	27	6.5	5.03	
10-Apr-20	27.5	27	17	42	27	27	6	3.99	
11-Apr-20	28.5	31	14	46	31	29	6.5	6.06	
12-Apr-20	28	27	12	50.5	29.5	28	6.5	5.59	
13-Apr-20	27.5	27	14	49	29	27.5	6	6.35	
14-Apr-20	27	28	14	55	32	29	5	5.61	
15-Apr-20	27	28	13	52	30.5	29.5	5.5	4.46	
16-Apr-20	27.5	31	16	53	32.5	31	5	5.78	
17-Apr-20	28	33	18	52	34	35	6.5	5.84	
18-Apr-20	27	30	15	50	24.5	28.5	5	6.48	
19-Apr-20	27	28	11	50	27.5	24.5	19	6.06	elec. HW on (manual)
20-Apr-20	27.5	28	13	52	28	28.5	3.5	5.59	
21-Apr-20	27.5	29	15	47.5	29.5	28.5	4	6.35	
22-Apr-20	29.5	31	14	52.5	35.5	35	7.5	6.89	
23-Apr-20	27.5	28	13	46.5	28	26.5	4	4.94	
24-Apr-20	27	29	13	47.5	28.5	29.5	6	5.7	

25-Apr-20	27.5	29	13	49	28	28.5	4.5	5.56	
26-Apr-20	28	29	12	49	30	30	5	6.1	
27-Apr-20	27	29	15	45.5	31	27.5	5.5	4.4	
28-Apr-20	27	27	14	35.5	24	24	10	3.67	
29-Apr-20	27	28	13	39.5	28.5	27.5	5	1.35	
30-Apr-20	27.5	30	10	43	28.5	27	5	8.46	
01-May-20	27	24	7	42	22	20	8.5	4.08	
02-May-20	25	24	7	46	27.5	25	13	4.46	
03-May-20	27	25	8	45	25	23	5	5.21	
04-May-20	25	25	8	38	23	23.5	3.5	6.21	
05-May-20	23.5	25	11	41	24	24.5	6	3.2	**Pool Heater turned off**
06-May-20	22.5	25	10	48	24	22	7.5	5.03	elec. HW on (manual)
07-May-20	22.5	26	12	39	25.5	24	6	3.99	
08-May-20	23	27	12	40	25	27	3.5	6.06	
09-May-20	22.5	27	11	37	25	27.5	3.5	5.59	
10-May-20	22.5	29	10	43	24.5	25	4.5	6.35	
11-May-20	22	24	8	38	23	21.5	6.5	3.88	
12-May-20	21	25	8	48	24	18	5	4.46	elec. HW on (manual)
13-May-20	21.5	25	9	43	23	22	4.5	5.78	
14-May-20	21.5	25	9	35.5	21.5	17.5	15.5	3.91	
15-May-20	21	24	9	48	24	21	11.5	3.7	elec. HW on (manual)
16-May-20	21	25	9	49	32	19	13.5	3.84	
17-May-20	21.5	25	10	40	26	24	10	3.75	
18-May-20	21.5	25	9	40	21.5	18	12	2.46	
19-May-20									
20-May-20									
21-May-20									
22-May-20									
23-May-20									
24-May-20									
25-May-20									
26-May-20									

27-May-20									
28-May-20									
29-May-20									
30-May-20		Shoulder period - no measurements taken							
31-May-20									
01-Jun-20									
02-Jun-20									
03-Jun-20									
04-Jun-20									
05-Jun-20									
06-Jun-20									
07-Jun-20									
08-Jun-20									
09-Jun-20									
10-Jun-20									
11-Jun-20									
12-Jun-20									
13-Jun-20									
14-Jun-20									
15-Jun-20	19.5	24	9	53	28	29	38.5	4.94	**Heating Mode - 24°c - elec. HW on (manual) used periodically over this time**
16-Jun-20	19.5	24	10	47	30	27	44	4.65	
17-Jun-20	19	24	8	43.5	28	26.5	43.5	5.08	
18-Jun-20	19	23	11	41	23.5	28	48	4.56	
19-Jun-20	18.5	21	11	48.5	26	25	42	4.28	
20-Jun-20	18	19	9	44	24	24	46.5	4.15	
21-Jun-20	18.5	23	12	42	22.5	24	47	4.86	
22-Jun-20	18	21	8	41.5	23	28.5	41.5	5.21	
23-Jun-20	18	19	4	45.5	21.5	29	38.5	4.71	
24-Jun-20	17.5	20	3	45.5	24.5	27	42.5	3.9	
25-Jun-20	17	21	3	41	17	23	47	5.88	
26-Jun-20	17	22	6	53	33	29	42	5.95	

27-Jun-20	16	24	6	45	25	26	44	6.21	
28-Jun-20	17	21	6	41	20	25	50	8.56	
29-Jun-20	17.5	22	6	53	27	24	47	5.54	
30-Jun-20	18	22	5	31	22.5	26	47.5	7.36	
01-Jul-20	17	22	6	53.5	23	25.5	46	6.35	
02-Jul-20	18	23	6	43	27.5	26	42	6.53	
03-Jul-20	18.5	25	7	42.5	30.5	28	45	6.21	
04-Jul-20	18	20	1	42.5	19.5	30	49	6.47	
05-Jul-20	17.5	20	3	40.5	20	30	65	6.83	
06-Jul-20	16.5	22	4	59.5	28	29	46.5	5.72	
07-Jul-20	17	22	6	41.5	22	20	48	6.08	
08-Jul-20	17	22	8	45.5	27	27	43.5	6.81	
09-Jul-20	17.5	19	8	50	19.5	32	61	8.31	
10-Jul-20	18	22	6	46	22.5	28	55	8.22	
11-Jul-20	17.5	24	8	52	24.5	29	51.5	7.45	
12-Jul-20	18	25	6	41	25	30	60.5	6.74	
13-Jul-20	18.5	21	4	49	25	22	45	5.69	
14-Jul-20	17.5	19	5	39.5	17.5	24	53.5	7.06	
15-Jul-20	17	22.5	2.5	41.5	26.5	29.5	56	6.63	
16-Jul-20	16.5	22	2	48	24	30.5	57.5	7.38	
17-Jul-20	17	23	3	46.5	25.5	31	50.5	7.89	
18-Jul-20	17	23	3	44	27	28	49	7.15	
19-Jul-20	17.5	24	3	41	24.5	26	61.5	8.09	
20-Jul-20	17.5	25	6	43.5	27.5	28.5	54	7.68	
21-Jul-20	17	21	7	46	24	29	56	7.94	
22-Jul-20	16.5	18	7	55	20	32	55	7.39	
23-Jul-20	16.5	18	8	41	19	24	66.5	14	
24-Jul-20	17	19	10	51.5	22	23.5	50.5	7.08	
25-Jul-20	17.5	21	10	44	21	26.5	54.5	7.87	
26-Jul-20	17.5	24	8	41	22.5	28	51.5	7.46	
27-Jul-20	18	20	6	34	18.5	24	51	6.26	
28-Jul-20	18.5	21	5	42	20.5	22	44.5	5.23	

29-Jul-20	20	24	6	48	21	22	37	5.41	
30-Jul-20	19.5	23	7	44	23.5	21	35	6.04	
31-Jul-20	18	22	7	53	27.5	24	53	6.51	
01-Aug-20	18	22	6	56.5	22	23	53	6.05	
02-Aug-20	18.5	24	6	48	22.5	23.5	52.5	6.34	
03-Aug-20	18	23	7	54	21.5	24.5	58	6.19	
04-Aug-20	20.5	24	6	47.5	23.5	21	45	3.49	
05-Aug-20	20	25	7	44.5	24	22.5	51	6.46	
06-Aug-20	19.5	26	9	48.5	23	24	54	6.67	
07-Aug-20	18	24	11	57	18	25	51	9.13	
08-Aug-20	18.5	25	6	37	22	24	49	8.44	
09-Aug-20	19.5	20	5	37	20	22	55	7.08	
10-Aug-20	17	19	6	44	17.5	25	51	7.23	
11-Aug-20	19.5	23	7	46	23.5	23.5	52	6.86	
12-Aug-20	18.5	24	8	43	29	22	51.5	6.42	
13-Aug-20	22	27	8	48	24	22.5	50.5	6.74	
14-Aug-20	18.5	27	12	45	27	23	53	6.41	
15-Aug-20	20.5	23	8	53	22.5	25	54.5	6.24	
16-Aug-20	20.5	23	4	42	25	24.5	51.5	6.39	
17-Aug-20	19.5	23	5	45	22	22.5	50	6.51	
18-Aug-20	20	25	10	46	24.5	26	47.5	6.28	
19-Aug-20	19.5	27	8	52	26.5	27	48	6.34	
20-Aug-20	19	23	7	44	23	23.5	52	6.86	
21-Aug-20	20.5	23	8	46	22.5	28	51.5	7.46	
22-Aug-20	17	22	5	51	21	23	53	6.05	
23-Aug-20	17	21	5	46	23.5	22	44.5	5.23	
24-Aug-20	17.5	22	3	43	24	31	50.5	7.89	
25-Aug-20	20.5	22	4	41	23	22	44.5	5.23	
26-Aug-20									
27-Aug-20									
28-Aug-20									
29-Aug-20									

30-Aug-20									
31-Aug-20									
01-Sep-20									
02-Sep-20									
03-Sep-20									
04-Sep-20		Shoulder period - no measurements taken							
05-Sep-20									
06-Sep-20									
07-Sep-20									
08-Sep-20									
09-Sep-20									
10-Sep-20									
11-Sep-20									
12-Sep-20									
13-Sep-20									
14-Sep-20									
15-Sep-20									**Cooling Mode**
16-Sep-20									
17-Sep-20									
18-Sep-20									
19-Sep-20									
20-Sep-20									
21-Sep-20									
22-Sep-20									
23-Sep-20									
24-Sep-20									
25-Sep-20									
26-Sep-20									
27-Sep-20									
28-Sep-20									
29-Sep-20									
30-Sep-20									

10. Appendix D – Household a/c System Power Measurements

Daily Power Measurement and Calculations

Prior to turning a/c ON - No other household items ON (except in standby) - NOT including 1 x fridge, 1 x bar fridge and computers x 2.

	Voltage (V)	Current (I)	Power Factor (PF)	Instantaneous Net Usage (kWh)	Power per Phase (W)	Total Power - (kW) (sum of phases)
L1	239	1.18	0.23	0.14	112.35	**0.25**
L2	240	0.49			46.85	
L3	238	0.91			86.28	

AC ON - 5mins into operation - No other household items ON (except in standby) - NOT including 1 x fridge, 1 x bar fridge and computers x 2.

	Voltage (V)	Current (I)	Power Factor (PF)	Instantaneous Net Usage (kWh)	Power per Phase (W)	Total Power - (kW) (sum of phases)
L1	238	2.74	0.79	1.46	892.31	**2.71**
L2	239	2.84			928.76	
L3	237	2.73			885.32	

Difference between Total Power PRIOR to turning AC ON and Total Power WHILE AC is ON = AC Power Draw/Usage/Consumption

AC Power (kW)
2.46

Power per phase calculated using 3-phase power formula - P = I x (sqrt(3) x PF x V)

L = Phase
AC = air-conditioning
PF = Power Factor
I = Current
V = Voltage
W = Watts
kW = Kilowatts

11. Appendix E – Box a/c System Measurements

Temperature and Power Measurements Box Air-Conditioning System using Heat Source - Cooling Mode												
Date	Temp of Sink	Max Air Temp	Min Air Temp	Comp out temp	Comp in temp	Condenser (outdoor coil) in temp	Condenser (outdoor coil) out temp	evaporator (indoor coil) in temp	evaporator (indoor coil) out temp	Power Use before adding sink (kW)	Power Use after adding sink (kW)	Comments
12-Aug-20	43	24	8	46		40	26	1.5	9.5	0.834	0.811	Heat pack applied to compressor in
13-Aug-20	18.5	27	8	51		38.5	26.5	3	8	0.878	0.845	Cold pack applied to expander (throttling valve) outlet
14-Aug-20	18.5	27	12	58		36	28	4.5	9	0.942	0.906	Cold pack applied to expander (throttling valve) outlet
15-Aug-20	20.5	23	8	44		31.5	25.5	24	2	0.815	0.791	Cold pack applied to expander (throttling valve) outlet
16-Aug-20	20.5	23	4	37		27	23	21.5	-1	0.782	0.763	Cold pack applied to expander (throttling valve) outlet
17-Aug-20	45	23	5	46	16	45	23	33	2	0.815	0.768	Heat pack applied to compressor in
18-Aug-20	46	25	10	45	14	37	24.5	28	2.5	0.864	0.802	Heat pack applied to compressor in
19-Aug-20	52	27	8	46	11.5	34	25.5	25	3.5	0.887	0.832	Heat pack applied to compressor in
20-Aug-20	44	23	7	44.5	14	32.5	24	21.5	1.5	0.827	0.796	Heat pack applied to compressor in
21-Aug-20	46	23	8	45	16	29	22	20	-1	0.81	0.793	Heat pack applied to compressor in
22-Aug-20	51	22	5	46	15	27	23	32	8.5	0.831	0.798	Heat pack applied to compressor in
23-Aug-20	46	21	5	46.5	17.5	26	21.5	29	10	0.845	0.809	Heat pack applied to compressor in
24-Aug-20	43	22	3	48	20	25	19.5	27	11.5	0.852	0.805	Heat pack applied to compressor in
25-Aug-20	41	22	4	56	11	43	29	32	8.5	0.945	0.849	Heat pack applied to compressor in

Temperature and Power Measurements Box Air-Conditioning System using Heat Sink - Heating Mode											
Date	Temp of Sink	Max Air Temp	Min Air Temp	Comp out temp	Indoor Coil in temp (condenser)	Indoor Coil out temp (condenser)	Outdoor Coil in temp (evaporator)	Outdoor Coil out temp (evaporator)	Power Use before adding sink (kW)	Power Use after adding sink (kW)	Comments
12-Aug-20	18.5	24	8	61.5	37	50	16.5	4	1.601	1.561	Cold pack applied to compressor outlet
13-Aug-20	19.5	27	8	65	35	58	18	6.5	1.716	1.682	Cold pack applied to compressor outlet
14-Aug-20	19	27	12	63	32.5	62	22	13.5	1.81	1.714	Cold pack applied to compressor outlet
15-Aug-20	20.5	23	8	62	35	63.5	24	14.5	1.674	1.606	Cold pack applied to compressor outlet
16-Aug-20	20.5	23	4	63	37	61	25	16	1.439	1.415	Cold pack applied to outdoor coil inlet
17-Aug-20	19.5	23	5	63	33	55.5	33	16	1.61	1.514	Cold pack applied to compressor outlet
18-Aug-20	20	25	10	71	36	61	32	18	1.703	1.645	Cold pack applied to compressor outlet
19-Aug-20	19.5	27	8	89	42	74	31	21	1.597	1.581	Cold pack applied to compressor outlet
20-Aug-20	19.5	23	7	86	38	68	29	16	1.622	1.574	Cold pack applied to compressor outlet
21-Aug-20	20.5	23	8	85	40	70	28.5	17.5	1.789	1.644	Cold pack applied to compressor outlet
22-Aug-20	17	22	5	87	39.5	71	30	16.5	1.602	1.514	Cold pack applied to compressor outlet
23-Aug-20	17	21	5	86.5	42	68.5	29	17	1.541	1.44	Cold pack applied to compressor outlet
24-Aug-20	17.5	22	3	86.5	32.5	42.5	53	25.5	1.751	1.718	Cold pack applied to compressor outlet
25-Aug-20	20.5	22	4	90	42	70	69	21.5	1.602	1.569	Cold pack applied to compressor outlet

12. Appendix F – Household a/c System MATLAB Calculations – Before Extrapolation Using Flowing Water Loop Design - Cooling Mode

```
% Load the data
data = readtable('Household ac Temperature Measurements - Research Project - Michael Osgood - COOLING.xlsx', 'Range','A:J');
% This is the temperature before the compressor
T2 = data.ACOut;
Tinf2 = data.MaxAirTemp;
poweruse = data.PowerUse_kW_;

% temperature after compressor for calculation of enthalpy 2 (assumed pipe
% temp is refrigerant temp
T_h2 = data.ACIn;

% Convert to Kelvin
CtoK = 273.15;
Tinf2 = Tinf2 + CtoK;
T2 = T2 + CtoK;
T_h2 = T_h2 + CtoK;

% Total number of days
ndays = length(T2);

% Error tolerances for values
reltol = 1e-6;

% Initialise the arrays
% Mass flow rate
MFR_refrig = nan(ndays,1);
% Temperature of the refrigerant (K)
T_refrig = MFR_refrig;
% enthalpy 2 of the refrigerant
enthalpy2_refrig = T_refrig;
```

```
% Setup vectors for interpolation of properties
% Data for properties of air
T_air = [0 5 10 15 20 25 30 35] + CtoK;
k_air = [0.02364 0.02401 0.02439 0.02476 0.02514 0.02551 0.02588 0.02625];
v_air = [1.338e-5 1.382e-5 1.426e-5 1.47e-5 1.516e-5 1.562e-5 1.608e-5 1.655e-5];
Pr_air = [0.7362 0.7350 0.7336 0.7323 0.7309 0.7296 0.7282 0.7268];
Tf = ((T2+Tinf2)/2);

% Data for thermal conductivity of R410a
T_k_R410A = [255.04 260.32 274.02 294.26 314.17 331.97 346.17];
k_R410A = [9.98 10.43 11.43 12.92 14.40 15.73 16.85] * 1e-3;

% Data for Prandtl Number of R410a from https://irc.wisc.edu/properties/
Pr_R410A = [0.832 0.829 0.822 0.814 0.807 0.799 0.793];

% Data for dynamic viscosity of R410a
T_R410A = [253.13 263.28 272.14 282.83 293.39 303.02 314.29 323.44 331.95];
u_R410A = [10.75 11.41 11.86 12.53 13.32 14.11 15.40 16.91 19.03] * 1e-6;

% Calculate for all the days
for day = 1:ndays
%for day = 1:ndays
   T22 = T2(day);
   Tinf22 = Tinf2(day);
   % Only do analysis if there is data for that day!
   if ~isnan(T22)  &&  ~isnan(Tinf22)  &&  ~isnan(poweruse(day))
      % Natural convection calculations for outside the pipe
      % Thermal conductivity of the air
      k = spline(T_air,k_air,Tinf22);
      % Kinematic viscosity of the air
      v = spline(T_air,v_air,Tinf22);
      % Prandtl number of the air
      Pr = spline(T_air,Pr_air,Tinf22);

      % Heat transfer rate
```

```
% Data for Rayleigh number
g = 9.81;
B = 1./Tf(day);
%Area of outer pipe (L = 1) in m^2
r2 = 0.009525;
r1 = 0.008385;
A_r2 = 2*pi*r2;
A_r1 = 2*pi*r1;
D = 2*r2;
D1 = 2*r1;

%Rayleigh number calculation
Ra = (((g).*(B).*abs(T22-Tinf22).*(D).^3)/(v).^2).*Pr;

%Nusselt number calculation
Nu =(0.825+((0.387*(Ra).^(1/6))/((1+(0.492/Pr).^(9/16)).^(8/27)))).^2;

%Natural Convection heat transfer coefficient (W.m^2.K)
h = (k/D)*Nu;

%Thermal resistance of outer pipe (Rconv,2)
Rconv2 = 1/(h.*A_r2);

%Heat transfer rate (Qconv,2) in Watts (W)
Qconv2 = (T22-Tinf22)./Rconv2;

%Qcyl = Qconv,2 therefore:
%Thermal conductivity of copper pipe (W/(m.K)
k_cop = 355;

%Thermal resistance of pipe (Rcyl)
Rcyl = (log(r2/r1))/(2*pi*k_cop);

% Calculate the inner pipe temperature (T1)
T1 = (Qconv2.*Rcyl)+T22;
```

```
% Initial guess for mass flow rate
if day == 1  ||  isnan(MFR_refrig(day-1))
   MFR=0.02;
else
   MFR = MFR_refrig(day-1);
end
MFR_refrig(day) = MFR;
% Initial error for mass flow rate
MFR_err = 1e6*reltol*(MFR+1);
MFR_count = 0;
% Iterate guess of mass flow rate
while abs(MFR_err) > reltol*MFR  &&  MFR_count < 100
   % Initial guess of refrigerant temperature
   if day == 1  ||  isnan(T_refrig(day-1))
      T = 273;
   else
      T = T_refrig(day-1);
   end
   T_refrig(day) = T;
   % Initial error for temperature
   T_err = 1e6*reltol*(T+1);
   T_count = 0;

   % Iterate guess of temperature
   while abs(T_err) > reltol*T  &&  T_count < 100
      % Calculation of Reynolds number (using current guess of
      % mass flow rate and temperature)
      Re = 4*MFR / (pi*spline(T_R410A,u_R410A,T)*D1);
      % Calculation of convective heat transfer coefficient
      if Re < 2300
         % Laminar pipe flow Nusselt number
         % Circular tube, laminar (Ts = constant):
         % Nu = hD/k = 3.66
         Nu_R410A = 3.66;
      else
         % Surface roughness: Drawn Tubing (table 8.1 - Fox and
```

```
        % McDonalds - Fluid Mechanics
        e = 0.0015;
        % Friction factor: Haaland formula, after Eq. (8.37)
        f = (1/(-1.8*log10(((e/D1)/3.7).^1.11+(6.9/Re)))).^2;

        % Turbulent pipe flow Nusselt number: Eq. (8-71)
        Pr_Nu = spline(T_k_R410A,Pr_R410A,T);

        Nu_R410A = ((f/8)*(Re-1000)*(Pr_Nu)) / (1+(((12.7*(f/8)).^0.5)*(((Pr_Nu).^(2/3))-1)));
    end

    % Forced Convective heat transfer coefficient
    k_h = spline(T_k_R410A,k_R410A,T);

    h_R410A = ((k_h)/D1)*Nu_R410A;

    % Thermal resistance of inner pipe (Rconv,1)
    Rconv1 = 1/(h_R410A.*A_r1);

    %Heat transfer rate (Qconv,1) in Watts (W)
    Qconv1 = Qconv2;

    % Calculate new refrigerant
    T = (Qconv1*Rconv1)+T1;

    % Recalculate error
    T_err = abs(T - T_refrig(day));
    % Latest guess for this day
    T_refrig(day) = T;
    T_count = T_count + 1;
end

% Calculate new value of MFR
% MFR = electrical power consumption divided by the enthalpy
% change across the compressor
```

```
    % Enthalpy before compressor is enthalpy from Tinf1
    % Using Superheated table (1200kPa) for R410A from DuPont:
    h1_T = [15 20 25 30 35 40 45 50 55 60 65 70 75 80 85 90 95 100] + 273.15;
    h1 = [427.6 433.8 439.7 445.4 451.0 456.5 461.8 467.1 472.4 477.6 482.7 487.8 492.9 498.0 503.1 508.1 513.2 518.2];

    enthalpy1 = spline(h1_T,h1,T);

    % Entropy after compressor assumes isentropic flow through
    % compressor: entropy is the same after compressor

    % Entropy before compressor (low pressure):
    % Using Superheated table (1200kPa) for R410A from DuPont:
    s1_T = [15 20 25 30 35 40 45 50 55 60 65 70 75 80 85 90 95 100] + 273.15;
    s1 = [1.7948 1.8159 1.8359 1.8550 1.8733 1.8909 1.9079 1.9245 1.9406 1.9563 1.9716 1.9866 2.0014 2.0159 2.0301 2.0441
    2.0580 2.0716] ;

    entropy2 = spline(s1_T,s1,T);

    % Enthalpy after compressor (high pressure)
    % From superheated table at high pressure (3400kPa) for R410A from DuPont:
    h2 = [423.1 434.1 443.7 452.2 460.2 467.6 474.8 481.6 488.2 494.7 501.0 507.1 513.2 519.2 525.1 531.0 536.8 542.6];
    s2 = [1.6987 1.7321 1.7604 1.7856 1.8086 1.8299 1.8499 1.8688 1.8869 1.9043 1.9211 1.9373 1.9531 1.9684 1.9834 1.9981
    2.0124 2.0265];

    enthalpy2 = spline(s2,h2,entropy2);

    % Mass flow rate
    MFR = poweruse(day)/(enthalpy2-enthalpy1);

    % Recalculate error
    MFR_err = abs(MFR - MFR_refrig(day));
    % Latest guess for this day
    MFR_refrig(day) = MFR;
    MFR_count = MFR_count + 1;
    enthalpy2_refrig(day) = enthalpy2;
end
```

```
        disp(['For day ',num2str(day),  ...
            ' the refrigerant temperature = ',num2str(T-CtoK),'C and mass flow rate = ',num2str(MFR),'kg/s.',  ...
            ' The inner pipe temperature = ',num2str(T1-CtoK),'C'])
    end
end

% Convert back to Celsius
T_refrig = T_refrig - CtoK;
% Output data
data.T_refrig = T_refrig;
data.MFR = MFR_refrig;
data.enthalpy2 = enthalpy2_refrig;
writetable(data,'Household ac Temperature Calculations - Research Project - Michael Osgood - COOLING.xlsx');
```

13. Appendix G – Household a/c System MATLAB Calculations – After Extrapolation Using Flowing Water Loop Design - Cooling Mode

(H_20 Mass Flow Rate = 1.33kg/s)

```
% Load the data
data = readtable('Household ac Temperature Calculations - Research Project - Michael Osgood - COOLING.xlsx');
% This is the temperature before the compressor
Tinf2 = data.HWOut;
T2 = Tinf2 - 0.1;
MFR_refrig = data.MFR;
enthalpy2_refrig = data.enthalpy2;
T_refrig_ord = data.T_refrig;

% Convert to Kelvin
CtoK = 273.15;
Tinf2 = Tinf2 + CtoK;
T2 = T2 + CtoK;
T_refrig_ord = T_refrig_ord + CtoK;

% Total number of days
ndays = length(Tinf2);

% Error tolerances for values
reltol = 1e-6;

% Initialise the arrays
% Temperature of the refrigerant (K)
T_refrig = nan(ndays,1);
% poweruse of the system (kw)
poweruse_refrig = nan(ndays,1);

% https://justwaterpumps.com.au/pond-pumps/
```

```
% 80 LPM = 1.33 kg/s - Davey SJ60-08PC Silver Series Jet Pump
MFR_H20_1 = 1.33;

% Setup vectors for interpolation of properties
% Data for properties of water - Table A-9 - Heat and Mass Transfer
T_H20 = [10 15 20 25 30 35 40 45 50 55 60] + CtoK;
k_H20 = [0.580 0.589 0.598 0.607 0.615 0.623 0.631 0.637 0.644 0.649 0.654];
u_H20 = [1.307e-3 1.138e-3 1.002e-3 0.891e-3 0.798e-3 0.720e-3 0.653e-3 0.596e-3 0.547e-3 0.504e-3 0.467e-3];
Pr_H20 = [9.45 8.09 7.01 6.14 5.42 4.83 4.32 3.91 3.55 3.25 2.99];
Cp_H20 = [4194 4185 4182 4180 4178 4178 4179 4180 4181 4183 4185];

% Data for thermal conductivity of R410a
T_k_R410A = [255.04 260.32 274.02 294.26 314.17 331.97 346.17];
k_R410A = [9.98 10.43 11.43 12.92 14.40 15.73 16.85] * 1e-3;

% Data for Prandtl Number of R410a from https://irc.wisc.edu/properties/
Pr_R410A = [0.832 0.829 0.822 0.814 0.807 0.799 0.793];

% Data for dynamic viscosity of R410a
T_R410A = [253.13 263.28 272.14 282.83 293.39 303.02 314.29 323.44 331.95];
u_R410A = [10.75 11.41 11.86 12.53 13.32 14.11 15.40 16.91 19.03] * 1e-6;

% Formula for specific heat of R410a (J/kg.K) (Temperature in K)
cp_R410A = @(T) 2.676084e2 + 2.115353*T - 9.848184e-4*T.^2 + 6.493781e-8*T.^3;

% Calculate for all the days
for day = 1:ndays
%for day = 1:ndays
    T22 = T2(day);
    Tinf22 = Tinf2(day);
    % Only do analysis if there is data for that day!
    if ~isnan(T22)  &&  ~isnan(Tinf22)
        %Heat transfer rate (Qconv,2) in Watts (W)
        % Q = MFR*Cp*Delta T
        % Specific Heat of the water
        Cp = spline(T_H20,Cp_H20,Tinf22);
```

```
    Qconv2 = MFR_H20_1*Cp*0.1;

     % This day's mass flow rate
       MFR = MFR_refrig(day);

    % Refrigerant temperature in ordinary operation
    T1 = T_refrig_ord(day);
    % Calculate new refrigerant temperature (based on reduction due to
    % specific heat)
    Qconv1 = Qconv2;
    Tinf1 = (Qconv1/(MFR*cp_R410A(T1))) + T1;

       % MFR = electrical power consumption divided by the enthalpy
       % change across the compressor

       % Enthalpy before compressor is enthalpy from Tinf1
       % Using Superheated table (1200kPa) for R410A from DuPont:
       h1_T = [15 20 25 30 35 40 45 50 55 60 65 70 75 80 85 90 95 100] + 273;
       h1 = [427.6 433.8 439.7 445.4 451.0 456.5 461.8 467.1 472.4 477.6 482.7 487.8 492.9 498.0 503.1 508.1 513.2 518.2];

       enthalpy1 = spline(h1_T,h1,Tinf1);

       % Value is same as from household normal calculations
       enthalpy2 = enthalpy2_refrig(day);

       % Power use
       poweruse = MFR*(enthalpy2-enthalpy1);

       % Store this value
       poweruse_refrig(day) = poweruse;
       T_refrig(day) = Tinf1;
  end
     disp(['For day ',num2str(day),  ...
       ' the refrigerant temperature = ',num2str(Tinf1-CtoK),'C and mass flow rate = ',num2str(MFR),'kg/s.',  ...
       ' The inner pipe temperature = ',num2str(T1-CtoK),'C.',' The power use with heat source application = ',num2str(poweruse),'kW'])
end
```

```
% Convert back to Celsius
T_refrig = T_refrig - CtoK;

% Output data
data.T_refrig = T_refrig;
data.MFR = MFR_refrig;
data.poweruse = poweruse_refrig;
writetable(data,'Household ac Temperature Calculations - Research Project - Michael Osgood - COOLING - Extrapolation - MFR - 1.33.xlsx');
```

14. Appendix H – Household a/c System MATLAB Calculations – Before Extrapolation Using Flowing Water Loop Design - Heating Mode

```
% Load the data
data = readtable('Household ac Temperature Measurements - Research Project - Michael Osgood - HEATING.xlsx', 'Range','A:J');
% This is the temperature after the compressor
T2 = data.ACIn;
Tinf2 = data.MaxAirTemp;
poweruse = data.PowerUse_kW_;

% temperature before compressor for calculation of enthalpy 1 (assumed pipe
% temp is refrigerant temp
T_h1 = data.ACOut;

% Convert to Kelvin
CtoK = 273.15;
Tinf2 = Tinf2 + CtoK;
T2 = T2 + CtoK;
T_h1 = T_h1 + CtoK;

% Total number of days
ndays = length(T2);

% Error tolerances for values
reltol = 1e-6;

% Initialise the arrays
% Mass flow rate
MFR_refrig = nan(ndays,1);
% Temperature of the refrigerant (K)
T_refrig = MFR_refrig;
% enthalpy 1 of the refrigerant
enthalpy1_refrig = T_refrig;
```

```
% Setup vectors for interpolation of properties
% Data for properties of air
T_air = [0 5 10 15 20 25 30 35] + CtoK;
k_air = [0.02364 0.02401 0.02439 0.02476 0.02514 0.02551 0.02588 0.02625];
v_air = [1.338e-5 1.382e-5 1.426e-5 1.47e-5 1.516e-5 1.562e-5 1.608e-5 1.655e-5];
Pr_air = [0.7362 0.7350 0.7336 0.7323 0.7309 0.7296 0.7282 0.7268];
Tf = ((T2+Tinf2)/2);

% Data for thermal conductivity of R410a
T_k_R410A = [255.04 260.32 274.02 294.26 314.17 331.97 346.17];
k_R410A = [9.98 10.43 11.43 12.92 14.40 15.73 16.85] * 1e-3;

% Data for Prandtl Number of R410a from https://irc.wisc.edu/properties/
Pr_R410A = [0.832 0.829 0.822 0.814 0.807 0.799 0.793];

% Data for dynamic viscosity of R410a
T_R410A = [253.13 263.28 272.14 282.83 293.39 303.02 314.29 323.44 331.95];
u_R410A = [10.75 11.41 11.86 12.53 13.32 14.11 15.40 16.91 19.03] * 1e-6;

% Calculate for all the days
for day = 1:ndays
%for day = 1:ndays
    T22 = T2(day);
    Tinf22 = Tinf2(day);
    % Only do analysis if there is data for that day!
    if ~isnan(T22)  &&  ~isnan(Tinf22)  &&  ~isnan(poweruse(day))
        % Natural convection calculations for outside the pipe
        % Thermal conductivity of the air
        k = spline(T_air,k_air,Tinf22);
        % Kinematic viscosity of the air
        v = spline(T_air,v_air,Tinf22);
        % Prandtl number of the air
        Pr = spline(T_air,Pr_air,Tinf22);

        % Heat transfer rate
```

```
% Data for Rayleigh number
g = 9.81;
B = 1./Tf(day);
%Area of outer pipe (L = 1) in m^2
r2 = 0.009525;
r1 = 0.008385;
A_r2 = 2*pi*r2;
A_r1 = 2*pi*r1;
D = 2*r2;
D1 = 2*r1;

%Rayleigh number calculation
Ra = (((g).*(B).*abs(T22-Tinf22).*(D).^3)/(v).^2).*Pr;

%Nusselt number calculation
Nu =(0.825+((0.387*(Ra).^(1/6))/((1+(0.492/Pr).^(9/16)).^(8/27)))).^2;

%Natural Convection heat transfer coefficient (W.m^2.K)
h = (k/D)*Nu;

%Thermal resistance of outer pipe (Rconv,2)
Rconv2 = 1/(h.*A_r2);

%Heat transfer rate (Qconv,2) in Watts (W)
Qconv2 = (T22-Tinf22)./Rconv2;

%Qcyl = Qconv,2 therefore:
%Thermal conductivity of copper pipe (W/(m.K)
k_cop = 355;

%Thermal resistance of pipe (Rcyl)
Rcyl = (log(r2/r1))/(2*pi*k_cop);

% Calculate the inner pipe temperature (T1)
T1 = (Qconv2.*Rcyl)+T22;
```

```
% Initial guess for mass flow rate
if day == 1  ||  isnan(MFR_refrig(day-1))
   MFR=0.002;
else
   MFR = MFR_refrig(day-1);
end
MFR_refrig(day) = MFR;
% Initial error for mass flow rate
MFR_err = 1e6*reltol*(MFR+1);
MFR_count = 0;
% Iterate guess of mass flow rate
while abs(MFR_err) > reltol*MFR  &&  MFR_count < 100
   % Initial guess of refrigerant temperature
   if day == 1  ||  isnan(T_refrig(day-1))
      T = 273;
   else
      T = T_refrig(day-1);
   end
   T_refrig(day) = T;
   % Initial error for temperature
   T_err = 1e6*reltol*(T+1);
   T_count = 0;

   % Iterate guess of temperature
   while abs(T_err) > reltol*T  &&  T_count < 100
      % Calculation of Reynolds number (using current guess of
      % mass flow rate and temperature)
      Re = 4*MFR / (pi*spline(T_R410A,u_R410A,T)*D1);
      % Calculation of convective heat transfer coefficient
      if Re < 2300
         % Laminar pipe flow Nusselt number
         % Circular tube, laminar (Ts = constant):
         % Nu = hD/k = 3.66
         Nu_R410A = 3.66;
      else
         % Surface roughness: Drawn Tubing (table 8.1 - Fox and
```

```
        % McDonalds - Fluid Mechanics
        e = 0.0015;
        % Friction factor: Haaland formula, after Eq. (8.37)
        f = (1/(-1.8*log10(((e/D1)/3.7).^1.11+(6.9/Re)))).^2;

        % Turbulent pipe flow Nusselt number: Eq. (8-71)
        Pr_Nu = spline(T_k_R410A,Pr_R410A,T);

        Nu_R410A = ((f/8)*(Re-1000)*(Pr_Nu)) / (1+(((12.7*(f/8)).^0.5)*(((Pr_Nu).^(2/3))-1)));
    end

    % Forced Convective heat transfer coefficient
    k_h = spline(T_k_R410A,k_R410A,T);

    h_R410A = ((k_h)/D1)*Nu_R410A;

    % Thermal resistance of outer pipe (Rconv,1)
    Rconv1 = 1/(h_R410A.*A_r1);

    %Heat transfer rate (Qconv,1) in Watts (W)
    Qconv1 = Qconv2;

    % Calculate new refrigerant
    T = (Qconv1*Rconv1)+T1;

    % Recalculate error
    T_err = abs(T - T_refrig(day));
    % Latest guess for this day
    T_refrig(day) = T;
    T_count = T_count + 1;
end

% Calculate new value of MFR
% MFR = electrical power consumption divided by the enthalpy
% change across the compressor
```

```
    % Enthalpy before compressor is enthalpy from Tinf1
    % Using Superheated table (1200kPa) for R410A from DuPont:
    h1_T = [15 20 25 30 35 40 45 50 55 60 65 70 75 80 85 90 95 100] + CtoK;
    h1 = [427.6 433.8 439.7 445.4 451.0 456.5 461.8 467.1 472.4 477.6 482.7 487.8 492.9 498.0 503.1 508.1 513.2 518.2];

    enthalpy1 = spline(h1_T,h1,T_h1(day));

    % Enthalpy after compressor (high pressure)
    % From superheated table at high pressure (3400kPa) for R410A from DuPont:
    h2 = [423.1 434.1 443.7 452.2 460.2 467.6 474.8 481.6 488.2 494.7 501.0 507.1 513.2 519.2 525.1 531.0 536.8 542.6];
    h2_T = [55 60 65 70 75 80 85 90 95 100 105 110 115 120 125 130 135 140] + CtoK;
    s2 = [1.6987 1.7321 1.7604 1.7856 1.8086 1.8299 1.8499 1.8688 1.8869 1.9043 1.9211 1.9373 1.9531 1.9684 1.9834 1.9981
    2.0124 2.0265];

    enthalpy2 = spline(h2_T,h2,T);

    entropy2 = spline(h2_T,s2,T);

    % Enthalpy before compressor assuming isentropic compressor
    % Using Superheated table (1200kPa) for R410A from DuPont:
    h1 = [427.6 433.8 439.7 445.4 451.0 456.5 461.8 467.1 472.4 477.6 482.7 487.8 492.9 498.0 503.1 508.1 513.2 518.2];
    s1 = [1.7948 1.8159 1.8359 1.8550 1.8733 1.8909 1.9079 1.9245 1.9406 1.9563 1.9716 1.9866 2.0014 2.0159 2.0301 2.0441
    2.0580 2.0716];

    enthalpy1 = spline(s1,h1,entropy2);

    % Mass flow rate
    MFR = poweruse(day)/(enthalpy2-enthalpy1);

    % Recalculate error
    MFR_err = abs(MFR - MFR_refrig(day));
    % Latest guess for this day
    MFR_refrig(day) = MFR;
    MFR_count = MFR_count + 1;
    enthalpy1_refrig(day) = enthalpy1;
end
```

```
        disp(['For day ',num2str(day),  ...
           ' the refrigerant temperature = ',num2str(T-CtoK),'C and mass flow rate = ',num2str(MFR),'kg/s.',  ...
           ' The inner pipe temperature = ',num2str(T1-CtoK),'C'])
    end
end

% Convert back to Celsius
T_refrig = T_refrig - CtoK;
% Output data
data.T_refrig = T_refrig;
data.MFR = MFR_refrig;
data.enthalpy1 = enthalpy1_refrig;
writetable(data,'Household ac Temperature Calculations - Research Project - Michael Osgood - HEATING.xlsx');
```

15. Appendix I – Household a/c System MATLAB Calculations – After Extrapolation Using Flowing Water Loop Design - Heating Mode

(H_20 Mass Flow Rate = 1.33kg/s)

```
% Load the data
data = readtable('Household ac Temperature Calculations - Research Project - Michael Osgood - HEATING.xlsx');
% This is the temperature after the compressor
Tinf2 = data.WaterTemp;
T2 = Tinf2 - 0.1;
MFR_refrig = data.MFR;
enthalpy1_refrig = data.enthalpy1;
T_refrig_ord = data.T_refrig;

% Convert to Kelvin
CtoK = 273.15;
Tinf2 = Tinf2 + CtoK;
T2 = T2 + CtoK;
T_refrig_ord = T_refrig_ord + CtoK;

% Total number of days
ndays = length(Tinf2);

% Error tolerances for values
reltol = 1e-6;

% Initialise the arrays
% Temperature of the refrigerant (K)
T_refrig = nan(ndays,1);
% poweruse of the system (kw)
poweruse_refrig = nan(ndays,1);

% https://justwaterpumps.com.au/pond-pumps/
```

```
% 80 LPM = 1.33 kg/s - Davey SJ60-08PC Silver Series Jet Pump
MFR_H20_1 = 1.33;

% Setup vectors for interpolation of properties
% Data for properties of water - Table A-9 - Heat and Mass Transfer
T_H20 = [10 15 20 25 30 35 40 45 50 55 60] + CtoK;
k_H20 = [0.580 0.589 0.598 0.607 0.615 0.623 0.631 0.637 0.644 0.649 0.654];
u_H20 = [1.307e-3 1.138e-3 1.002e-3 0.891e-3 0.798e-3 0.720e-3 0.653e-3 0.596e-3 0.547e-3 0.504e-3 0.467e-3];
Pr_H20 = [9.45 8.09 7.01 6.14 5.42 4.83 4.32 3.91 3.55 3.25 2.99];
Cp_H20 = [4194 4185 4182 4180 4178 4178 4179 4180 4181 4183 4185];

% Data for thermal conductivity of R410a
T_k_R410A = [255.04 260.32 274.02 294.26 314.17 331.97 346.17];
k_R410A = [9.98 10.43 11.43 12.92 14.40 15.73 16.85] * 1e-3;

% Data for Prandtl Number of R410a from https://irc.wisc.edu/properties/
Pr_R410A = [0.832 0.829 0.822 0.814 0.807 0.799 0.793];

% Data for dynamic viscosity of R410a
T_R410A = [253.13 263.28 272.14 282.83 293.39 303.02 314.29 323.44 331.95];
u_R410A = [10.75 11.41 11.86 12.53 13.32 14.11 15.40 16.91 19.03] * 1e-6;

% Formula for specific heat of R410a (J/kg.K) (Temperature in K)
cp_R410A = @(T) 2.676084e2 + 2.115353*T - 9.848184e-4*T.^2 + 6.493781e-8*T.^3;

% Calculate for all the days
for day = 1:ndays
%for day = 1:ndays
    T22 = T2(day);
    Tinf22 = Tinf2(day);
    % Only do analysis if there is data for that day!
    if ~isnan(T22)  &&  ~isnan(Tinf22)
        %Heat transfer rate (Qconv,2) in Watts (W)
        % Q = MFR*Cp*Delta T
        % Specific Heat of the water
        Cp = spline(T_H20,Cp_H20,Tinf22);
```

```
    Qconv2 = -MFR_H20_1*Cp*0.01;

     % This day's mass flow rate
       MFR = MFR_refrig(day);

    % Refrigerant temperature in ordinary operation
    T1 = T_refrig_ord(day);
    % Calculate new refrigerant temperature (based on reduction due to
    % specific heat)
    Qconv1 = Qconv2;
    Tinf1 = (Qconv1/(MFR*cp_R410A(T1))) + T1;

       % MFR = electrical power consumption divided by the enthalpy
       % change across the compressor

       % Value is same as from household normal calculations
       enthalpy1 = enthalpy1_refrig(day);

       % Enthalpy after compressor (high pressure) is enthalpy from Tinf1
       % From superheated table at high pressure (3400kPa) for R410A from DuPont:
       h2 = [423.1 434.1 443.7 452.2 460.2 467.6 474.8 481.6 488.2 494.7 501.0 507.1 513.2 519.2 525.1 531.0 536.8 542.6];
       h2_T = [55 60 65 70 75 80 85 90 95 100 105 110 115 120 125 130 135 140] + CtoK;

       enthalpy2 = spline(h2_T,h2,Tinf1);

       % Power use
       poweruse = MFR*(enthalpy2-enthalpy1);

       % Store this value
       poweruse_refrig(day) = poweruse;
       T_refrig(day) = Tinf1;
  end
    disp(['For day ',num2str(day),  ...
       ' the refrigerant temperature = ',num2str(Tinf1-CtoK),'C and mass flow rate = ',num2str(MFR),'kg/s.',  ...
       ' The inner pipe temperature = ',num2str(T1-CtoK),'C.',' The power use with heat sink application = ',num2str(poweruse),'kW'])
end
```

```
% Convert back to Celsius
T_refrig = T_refrig - CtoK;

% Output data
data.T_refrig = T_refrig;
data.MFR = MFR_refrig;
data.poweruse = poweruse_refrig;
writetable(data,'Household ac Temperature Calculations - Project - HEATING - Extrapolation - MFR - 1.33.xlsx');
```

16. Appendix J – Box a/c System MATLAB Calculations – Before Heat Pack - Cooling Mode

```
% Load the data
data = readtable('Box ac design - Temperature Measurements - Michael Osgood - COOLING.xlsx', 'Range','A:M');
% This is the temperature before the compressor
T2 = data.Comp_in_temp;
Tinf2 = data.Max_Air_Temp;
poweruse_prior = data.Power_Use_before_adding_sink_kW;

% Convert to Kelvin
CtoK = 273.15;
Tinf2 = Tinf2 + CtoK;
T2 = T2 + CtoK;

% Total number of days
ndays = length(T2);

% Error tolerances for values
reltol = 1e-6;

% Initialise the arrays
% Mass flow rate
MFR_refrig = nan(ndays,1);
% Temperature of the refrigerant (K)
T_refrig = MFR_refrig;

% Setup vectors for interpolation of properties
% Data for properties of air
T_air = [0 5 10 15 20 25 30 35] + CtoK;
k_air = [0.02364 0.02401 0.02439 0.02476 0.02514 0.02551 0.02588 0.02625];
v_air = [1.338e-5 1.382e-5 1.426e-5 1.47e-5 1.516e-5 1.562e-5 1.608e-5 1.655e-5];
Pr_air = [0.7362 0.7350 0.7336 0.7323 0.7309 0.7296 0.7282 0.7268];
Tf = ((T2+Tinf2)/2);
```

```
% Data for thermal conductivity of R22 from https://irc.wisc.edu/properties/
% Data used at atmospheric pressure: 101kPa
T_k_R22 = [-40 -30 -20 -10 0 10 20 30 40 50 60 70 80 90 100] + CtoK;
k_R22 = [7.09 7.58 8.08 8.61 9.15 9.71 10.3 10.9 11.5 12.1 12.8 13.4 14.1 14.8 15.5] * 1e-3;

% Data for Prandtl Number of R22 from https://irc.wisc.edu/properties/
% Data used at atmospheric pressure: 101kPa
Pr_R22 = [0.833 0.824 0.817 0.81 0.804 0.798 0.793 0.788 0.783 0.778 0.773 0.769 0.764 0.76 0.755];

% Data for dynamic viscosity of R22 from https://irc.wisc.edu/properties/
T_R22 = [-40 -30 -20 -10 0 10 20 30 40 50 60 70 80 90 100] + CtoK;
u_R22 = [9.73 10.2 10.6 11.1 11.5 12 12.4 12.8 13.3 13.7 14.1 14.6 15 15.4 15.8] * 1e-6;

% Calculate for all the days
for day = 1:ndays
%for day = 1:ndays
   T22 = T2(day);
   Tinf22 = Tinf2(day);
   % Only do analysis if there is data for that day!
   if ~isnan(T22)  &&  ~isnan(Tinf22)  &&  ~isnan(poweruse_prior(day))
      % Natural convection calculations for outside the pipe
      % Thermal conductivity of the air
      k = spline(T_air,k_air,Tinf22);
      % Kinematic viscosity of the air
      v = spline(T_air,v_air,Tinf22);
      % Prandtl number of the air
      Pr = spline(T_air,Pr_air,Tinf22);

      % Heat transfer rate
      % Data for Rayleigh number
      g = 9.81;
      B = 1./Tf(day);
      %Area of outer pipe (L = 1) in m^2
      r2 = 0.00476;
      r1 = 0.00405;
      A_r2 = 2*pi*r2;
```

```
A_r1 = 2*pi*r1;
D = 2*r2;
D1 = 2*r1;

%Rayleigh number calculation
Ra = (((g).*(B).*abs(T22-Tinf22).*(D).^3)/(v).^2).*Pr;

%Nusselt number calculation
Nu =(0.825+((0.387*(Ra).^(1/6))/((1+(0.492/Pr).^(9/16)).^(8/27)))).^2;

%Natural Convection heat transfer coefficient (W.m^2.K)
h = (k/D)*Nu;

%Thermal resistance of outer pipe (Rconv,2)
Rconv2 = 1/(h.*A_r2);

%Heat transfer rate (Qconv,2) in Watts (W)
Qconv2 = (T22-Tinf22)./Rconv2;

%Qcyl = Qconv,2 therefore:
%Thermal conductivity of copper pipe (W/(m.K)
k_cop = 355;

%Thermal resistance of pipe (Rcyl)
Rcyl = (log(r2/r1))/(2*pi*k_cop);

% Calculate the inner pipe temperature (T1)
T1 = (Qconv2.*Rcyl)+T22;

% Initial guess for mass flow rate
if day == 1  ||  isnan(MFR_refrig(day-1))
   MFR=0;
else
   MFR = MFR_refrig(day-1);
end
MFR_refrig(day) = MFR;
```

```
% Initial error for mass flow rate
MFR_err = 1e6*reltol*(MFR+1);
MFR_count = 0;
% Iterate guess of mass flow rate
while abs(MFR_err) > reltol*MFR  &&  MFR_count < 100
   % Initial guess of refrigerant temperature
   if day == 1  ||  isnan(T_refrig(day-1))
      T = 273;
   else
      T = T_refrig(day-1);
   end
   T_refrig(day) = T;
   % Initial error for temperature
   T_err = 1e6*reltol*(T+1);
   T_count = 0;

   % Iterate guess of temperature
   while abs(T_err) > reltol*T  &&  T_count < 100
      % Calculation of Reynolds number (using current guess of
      % mass flow rate and temperature)
      Re = 4*MFR / (pi*spline(T_R22,u_R22,T)*D1);
      % Calculation of convective heat transfer coefficient
      if Re < 2300
         % Laminar pipe flow Nusselt number
         % Circular tube, laminar (Ts = constant):
         % Nu = hD/k = 3.66
         Nu_R22 = 3.66;
      else
         % Surface roughness: Drawn Tubing (table 8.1 - Fox and
         % McDonalds - Fluid Mechanics
         e = 0.0015;
         % Friction factor: Haaland formula, after Eq. (8.37)
         f = (1/(-1.8*log10(((e/D1)/3.7).^1.11+(6.9/Re)))).^2;

         % Turbulent pipe flow Nusselt number: Eq. (8-71)
         Pr_Nu = spline(T_k_R22,Pr_R22,T);
```

```
        Nu_R22 = ((f/8)*(Re-1000)*(Pr_Nu)) / (1+(((12.7*(f/8)).^0.5)*(((Pr_Nu).^(2/3))-1)));
    end

    % Forced Convective heat transfer coefficient
    k_h = spline(T_k_R22,k_R22,T);

    h_R22 = ((k_h)/D1)*Nu_R22;

    % Thermal resistance of outer pipe (Rconv,1)
    Rconv1 = 1/(h_R22.*A_r1);

    %Heat transfer rate (Qconv,1) in Watts (W)
    Qconv1 = Qconv2;

    % Calculate new refrigerant
    T = (Qconv1*Rconv1)+T1;

    % Recalculate error
    T_err = abs(T - T_refrig(day));
    % Latest guess for this day
    T_refrig(day) = T;
    T_count = T_count + 1;
end

% Calculate new value of MFR
% MFR = electrical power consumption divided by the enthalpy
% change across the compressor

% Enthalpy before compressor is enthalpy from Tinf1
% Using Superheated table (500kPa) for R22 from DuPont:
h1_T = [5 10 15 20 25 30 35 40 45 50 55 60 65 70 75 80 85 90 95 100] + 273.15;
h1 = [408.7 412.3 416.0 419.6 423.2 426.8 430.4 434.1 437.7 441.3 445.0 448.6 452.3 456.0 459.7 463.4 467.2 470.9 474.7
478.5];

enthalpy1 = spline(h1_T,h1,T);
```

```
        % Entropy after compressor assumes isentropic flow through
        % compressor: entropy is the same after compressor

        % Entropy before compressor (low pressure):
        % Using Superheated table (500kPa) for R22 from DuPont:
        s1_T = [5 10 15 20 25 30 35 40 45 50 55 60 65 70 75 80 85 90 95 100] + 273.15;
        s1 = [1.764 1.777 1.789 1.802 1.814 1.826 1.838 1.849 1.861 1.872 1.884 1.895 1.906 1.916 1.927 1.938 1.948 1.959 1.969
        1.979];

        entropy2 = spline(s1_T,s1,T);

        % Enthalpy after compressor (high pressure)
        % From superheated table at high pressure (1700kPa) for R22 from DuPont:
        h2 = [417.6 422.6 427.5 432.2 436.8 441.2 445.7 450.0 454.4 458.7 463.0 467.2 471.5 475.7 479.9 484.1 488.3 492.6 496.8
        501.0];
        s2 = [1.695 1.711 1.726 1.740 1.754 1.767 1.780 1.792 1.804 1.816 1.828 1.839 1.851 1.862 1.873 1.884 1.894 1.905 1.915
        1.925];

        enthalpy2 = spline(s2,h2,entropy2);

        % Mass flow rate
        MFR= poweruse_prior(day)/(enthalpy2-enthalpy1);

        % Recalculate error
        MFR_err = abs(MFR - MFR_refrig(day));
        % Latest guess for this day
        MFR_refrig(day) = MFR;
        MFR_count = MFR_count + 1;
    end
    disp(['For day ',num2str(day),  ...
        ' the refrigerant temperature = ',num2str(T-CtoK),'C and mass flow rate = ',num2str(MFR),'kg/s.',  ...
        ' The inner pipe temperature = ',num2str(T1-CtoK),'C'])
  end
end
```

```
% Convert back to Celsius
T_refrig = T_refrig - CtoK;
% Output data
data.T_refrig = T_refrig;
data.MFR = MFR_refrig;
writetable(data,'Box ac design - Temperature Calculations - Michael Osgood - COOLING.xlsx');
```

17. Appendix K – Box a/c System MATLAB Calculations – After Heat Pack - Cooling Mode

```
% Load the data
data = readtable('Box ac design - Temperature Measurements - Michael Osgood - COOLING - HP Applied.xlsx', 'Range','A:M');
% This is the temperature before the compressor
T2 = data.TempOfSink;
Tinf2 = data.Max_Air_Temp;
poweruse_after = data.Power_Use_after_adding_sink_kW;

% Convert to Kelvin
CtoK = 273.15;
Tinf2 = Tinf2 + CtoK;
T2 = T2 + CtoK;

% Total number of days
ndays = length(T2);

% Error tolerances for values
reltol = 1e-6;

% Initialise the arrays
% Mass flow rate
MFR_refrig = nan(ndays,1);
% Temperature of the refrigerant (K)
T_refrig = MFR_refrig;

% Setup vectors for interpolation of properties
% Data for properties of air
T_air = [0 5 10 15 20 25 30 35] + CtoK;
k_air = [0.02364 0.02401 0.02439 0.02476 0.02514 0.02551 0.02588 0.02625];
v_air = [1.338e-5 1.382e-5 1.426e-5 1.47e-5 1.516e-5 1.562e-5 1.608e-5 1.655e-5];
Pr_air = [0.7362 0.7350 0.7336 0.7323 0.7309 0.7296 0.7282 0.7268];
Tf = ((T2+Tinf2)/2);
```

```
% Data for thermal conductivity of R22 from https://irc.wisc.edu/properties/
% Data used at atmospheric pressure: 101kPa
T_k_R22 = [-40 -30 -20 -10 0 10 20 30 40 50 60 70 80 90 100] + CtoK;
k_R22 = [7.09 7.58 8.08 8.61 9.15 9.71 10.3 10.9 11.5 12.1 12.8 13.4 14.1 14.8 15.5] * 1e-3;

% Data for Prandtl Number of R22 from https://irc.wisc.edu/properties/
% Data used at atmospheric pressure: 101kPa
Pr_R22 = [0.833 0.824 0.817 0.81 0.804 0.798 0.793 0.788 0.783 0.778 0.773 0.769 0.764 0.76 0.755];

% Data for dynamic viscosity of R22 from https://irc.wisc.edu/properties/
T_R22 = [-40 -30 -20 -10 0 10 20 30 40 50 60 70 80 90 100] + CtoK;
u_R22 = [9.73 10.2 10.6 11.1 11.5 12 12.4 12.8 13.3 13.7 14.1 14.6 15 15.4 15.8] * 1e-6;

% Calculate for all the days
for day = 1:ndays
%for day = 1:ndays
  T22 = T2(day);
  Tinf22 = Tinf2(day);
  % Only do analysis if there is data for that day!
  if ~isnan(T22)  &&  ~isnan(Tinf22)  &&  ~isnan(poweruse_after(day))
    % Natural convection calculations for outside the pipe
    % Thermal conductivity of the air
    k = spline(T_air,k_air,Tinf22);
    % Kinematic viscousity of the air
    v = spline(T_air,v_air,Tinf22);
    % Prandtl number of the air
    Pr = spline(T_air,Pr_air,Tinf22);

    % Heat transfer rate
    % Data for Rayleigh number
    g = 9.81;
    B = 1./Tf(day);
    %Area of outer pipe (L = 1) in m^2
    r2 = 0.00476;
    r1 = 0.00405;
    A_r2 = 2*pi*r2;
```

```
A_r1 = 2*pi*r1;
D = 2*r2;
D1 = 2*r1;

%Rayleigh number calculation
Ra = (((g).*(B).*abs(T22-Tinf22).*(D).^3)/(v).^2).*Pr;

%Nusselt number calculation
Nu =(0.825+((0.387*(Ra).^(1/6))/((1+(0.492/Pr).^(9/16)).^(8/27)))).^2;

%Natural Convection heat transfer coefficient (W.m^2.K)
h = (k/D)*Nu;

%Thermal resistance of outer pipe (Rconv,2)
Rconv2 = 1/(h.*A_r2);

%Heat transfer rate (Qconv,2) in Watts (W)
Qconv2 = (T22-Tinf22)./Rconv2;

%Qcyl = Qconv,2 therefore:
%Thermal conductivity of copper pipe (W/(m.K)
k_cop = 355;

%Thermal resistance of pipe (Rcyl)
Rcyl = (log(r2/r1))/(2*pi*k_cop);

% Calculate the inner pipe temperature (T1)
T1 = (Qconv2.*Rcyl)+T22;

% Initial guess for mass flow rate
if day == 1  ||  isnan(MFR_refrig(day-1))
   MFR=0;
else
   MFR = MFR_refrig(day-1);
end
MFR_refrig(day) = MFR;
```

```
% Initial error for mass flow rate
MFR_err = 1e6*reltol*(MFR+1);
MFR_count = 0;
% Iterate guess of mass flow rate
while abs(MFR_err) > reltol*MFR  &&  MFR_count < 100
   % Initial guess of refrigerant temperature
   if day == 1  ||  isnan(T_refrig(day-1))
      T = 273;
   else
      T = T_refrig(day-1);
   end
   T_refrig(day) = T;
   % Initial error for temperature
   T_err = 1e6*reltol*(T+1);
   T_count = 0;

   % Iterate guess of temperature
   while abs(T_err) > reltol*T  &&  T_count < 100
      % Calculation of Reynolds number (using current guess of
      % mass flow rate and temperature)
      Re = 4*MFR / (pi*spline(T_R22,u_R22,T)*D1);
      % Calculation of convective heat transfer coefficient
      if Re < 2300
         % Laminar pipe flow Nusselt number
         % Circular tube, laminar (Ts = constant):
         % Nu = hD/k = 3.66
         Nu_R22 = 3.66;
      else
         % Surface roughness: Drawn Tubing (table 8.1 - Fox and
         % McDonalds - Fluid Mechanics
         e = 0.0015;
         % Friction factor: Haaland formula, after Eq. (8.37)
         f = (1/(-1.8*log10(((e/D1)/3.7).^1.11+(6.9/Re)))).^2;

         % Turbulent pipe flow Nusselt number: Eq. (8-71)
         Pr_Nu = spline(T_k_R22,Pr_R22,T);
```

```
        Nu_R22 = ((f/8)*(Re-1000)*(Pr_Nu)) / (1+(((12.7*(f/8)).^0.5)*(((Pr_Nu).^(2/3))-1)));
    end

    % Forced Convective heat transfer coefficient
    k_h = spline(T_k_R22,k_R22,T);

    h_R22 = ((k_h)/D1)*Nu_R22;

    % Thermal resistance of outer pipe (Rconv,1)
    Rconv1 = 1/(h_R22.*A_r1);

    %Heat transfer rate (Qconv,1) in Watts (W)
    Qconv1 = Qconv2;

    % Calculate new refrigerant
    T = (Qconv1*Rconv1)+T1;

    % Recalculate error
    T_err = abs(T - T_refrig(day));
    % Latest guess for this day
    T_refrig(day) = T;
    T_count = T_count + 1;
end

% Calculate new value of MFR
% MFR = electrical power consumption divided by the enthalpy
% change across the compressor

% Enthalpy before compressor is enthalpy from Tinf1
% Using Superheated table (500kPa) for R22 from DuPont:
h1_T = [5 10 15 20 25 30 35 40 45 50 55 60 65 70 75 80 85 90 95 100] + 273.15;
h1 = [408.7 412.3 416.0 419.6 423.2 426.8 430.4 434.1 437.7 441.3 445.0 448.6 452.3 456.0 459.7 463.4 467.2 470.9 474.7
478.5];

enthalpy1 = spline(h1_T,h1,T);
```

```
        % Entropy after compressor assumes isentropic flow through
        % compressor: entropy is the same after compressor

        % Entropy before compressor (low pressure):
        % Using Superheated table (500kPa) for R22 from DuPont:
        s1_T = [5 10 15 20 25 30 35 40 45 50 55 60 65 70 75 80 85 90 95 100] + 273.15;
        s1 = [1.764 1.777 1.789 1.802 1.814 1.826 1.838 1.849 1.861 1.872 1.884 1.895 1.906 1.916 1.927 1.938 1.948 1.959 1.969
        1.979];

        entropy2 = spline(s1_T,s1,T);

        % Enthalpy after compressor (high pressure)
        % From superheated table at high pressure (1700kPa) for R22 from DuPont:
        h2 = [417.6 422.6 427.5 432.2 436.8 441.2 445.7 450.0 454.4 458.7 463.0 467.2 471.5 475.7 479.9 484.1 488.3 492.6 496.8
        501.0];
        s2 = [1.695 1.711 1.726 1.740 1.754 1.767 1.780 1.792 1.804 1.816 1.828 1.839 1.851 1.862 1.873 1.884 1.894 1.905 1.915
        1.925];

        enthalpy2 = spline(s2,h2,entropy2);

        % Mass flow rate
        MFR= poweruse_after(day)/(enthalpy2-enthalpy1);

        % Recalculate error
        MFR_err = abs(MFR - MFR_refrig(day));
        % Latest guess for this day
        MFR_refrig(day) = MFR;
        MFR_count = MFR_count + 1;
    end
    disp(['For day ',num2str(day),  ...
        ' the refrigerant temperature = ',num2str(T-CtoK),'C and mass flow rate = ',num2str(MFR),'kg/s.',  ...
        ' The inner pipe temperature = ',num2str(T1-CtoK),'C'])
  end
end
```

```
% Convert back to Celsius
T_refrig = T_refrig - CtoK;
% Output data
data.T_refrig = T_refrig;
data.MFR = MFR_refrig;
writetable(data,'Box ac design - Temperature Calculations - Michael Osgood - COOLING - HP Applied.xlsx');
```

18. Appendix L – Box a/c System MATLAB Calculations – Before Cold Pack - Heating Mode

```
% Load the data
data = readtable('Box ac design - Temperature Measurements - Michael Osgood - HEATING.xlsx', 'Range','A:L');
% This is the temperature before the compressor
T2 = data.Comp_out_temp;
Tinf2 = data.Max_Air_Temp;
poweruse_prior = data.Power_Use_before_adding_sink_kW;

% Convert to Kelvin
CtoK = 273.15;
Tinf2 = Tinf2 + CtoK;
T2 = T2 + CtoK;

% Total number of days
ndays = length(T2);

% Error tolerances for values
reltol = 1e-6;

% Initialise the arrays
% Mass flow rate
MFR_refrig = nan(ndays,1);
% Temperature of the refrigerant (K)
T_refrig = MFR_refrig;

% Setup vectors for interpolation of properties
% Data for properties of air
T_air = [0 5 10 15 20 25 30 35] + CtoK;
k_air = [0.02364 0.02401 0.02439 0.02476 0.02514 0.02551 0.02588 0.02625];
v_air = [1.338e-5 1.382e-5 1.426e-5 1.47e-5 1.516e-5 1.562e-5 1.608e-5 1.655e-5];
Pr_air = [0.7362 0.7350 0.7336 0.7323 0.7309 0.7296 0.7282 0.7268];
Tf = ((T2+Tinf2)/2);
```

```
% Data for thermal conductivity of R22 from https://irc.wisc.edu/properties/
% Data used at atmospheric pressure: 101kPa
T_k_R22 = [-40 -30 -20 -10 0 10 20 30 40 50 60 70 80 90 100] + CtoK;
k_R22 = [7.09 7.58 8.08 8.61 9.15 9.71 10.3 10.9 11.5 12.1 12.8 13.4 14.1 14.8 15.5] * 1e-3;

% Data for Prandtl Number of R22 from https://irc.wisc.edu/properties/
% Data used at atmospheric pressure: 101kPa
Pr_R22 = [0.833 0.824 0.817 0.81 0.804 0.798 0.793 0.788 0.783 0.778 0.773 0.769 0.764 0.76 0.755];

% Data for dynamic viscosity of R22 from https://irc.wisc.edu/properties/
T_R22 = [-40 -30 -20 -10 0 10 20 30 40 50 60 70 80 90 100] + CtoK;
u_R22 = [9.73 10.2 10.6 11.1 11.5 12 12.4 12.8 13.3 13.7 14.1 14.6 15 15.4 15.8] * 1e-6;

% Calculate for all the days
for day = 1:ndays
%for day = 1:ndays
  T22 = T2(day);
  Tinf22 = Tinf2(day);
  % Only do analysis if there is data for that day!
  if ~isnan(T22)  &&  ~isnan(Tinf22)  &&  ~isnan(poweruse_prior(day))
    % Natural convection calculations for outside the pipe
    % Thermal conductivity of the air
    k = spline(T_air,k_air,Tinf22);
    % Kinematic viscosity of the air
    v = spline(T_air,v_air,Tinf22);
    % Prandtl number of the air
    Pr = spline(T_air,Pr_air,Tinf22);

    % Heat transfer rate
    % Data for Rayleigh number
    g = 9.81;
    B = 1./Tf(day);
    %Area of outer pipe (L = 1) in m^2
    r2 = 0.00476;
    r1 = 0.00405;
    A_r2 = 2*pi*r2;
```

```
A_r1 = 2*pi*r1;
D = 2*r2;
D1 = 2*r1;

%Rayleigh number calculation
Ra = (((g).*(B).*abs(T22-Tinf22).*(D).^3)/(v).^2).*Pr;

%Nusselt number calculation
Nu =(0.825+((0.387*(Ra).^(1/6))/((1+(0.492/Pr).^(9/16)).^(8/27)))).^2;

%Natural Convection heat transfer coefficient (W.m^2.K)
h = (k/D)*Nu;

%Thermal resistance of outer pipe (Rconv,2)
Rconv2 = 1/(h.*A_r2);

%Heat transfer rate (Qconv,2) in Watts (W)
Qconv2 = (T22-Tinf22)./Rconv2;

%Qcyl = Qconv,2 therefore:
%Thermal conductivity of copper pipe (W/(m.K)
k_cop = 355;

%Thermal resistance of pipe (Rcyl)
Rcyl = (log(r2/r1))/(2*pi*k_cop);

% Calculate the inner pipe temperature (T1)
T1 = (Qconv2.*Rcyl)+T22;

% Initial guess for mass flow rate
if day == 1  ||  isnan(MFR_refrig(day-1))
   MFR = 0.02;
else
   MFR = MFR_refrig(day-1);
end
MFR_refrig(day) = MFR;
```

```
% Initial error for mass flow rate
MFR_err = 1e6*reltol*(MFR+1);
MFR_count = 0;
% Iterate guess of mass flow rate
while abs(MFR_err) > reltol*MFR  &&  MFR_count < 100
   % Initial guess of refrigerant temperature
   if day == 1  ||  isnan(T_refrig(day-1))
      T = 273;
   else
      T = T_refrig(day-1);
   end
   T_refrig(day) = T;
   % Initial error for temperature
   T_err = 1e6*reltol*(T+1);
   T_count = 0;

   % Iterate guess of temperature
   while abs(T_err) > reltol*T  &&  T_count < 100
      % Calculation of Reynolds number (using current guess of
      % mass flow rate and temperature)
      Re = 4*MFR / (pi*spline(T_R22,u_R22,T)*D1);
      % Calculation of convective heat transfer coefficient
      if Re < 2300
         % Laminar pipe flow Nusselt number
         % Circular tube, laminar (Ts = constant):
         % Nu = hD/k = 3.66
         Nu_R22 = 3.66;
      else
         % Surface roughness: Drawn Tubing (table 8.1 - Fox and
         % McDonalds - Fluid Mechanics
         e = 0.0015;
         % Friction factor: Haaland formula, after Eq. (8.37)
         f = (1/(-1.8*log10(((e/D1)/3.7).^1.11+(6.9/Re)))).^2;

         % Turbulent pipe flow Nusselt number: Eq. (8-71)
         Pr_Nu = spline(T_k_R22,Pr_R22,T);
```

```
        Nu_R22 = ((f/8)*(Re-1000)*(Pr_Nu)) / (1+(((12.7*(f/8)).^0.5)*(((Pr_Nu).^(2/3))-1)));
    end

    % Forced Convective heat transfer coefficient
    k_h = spline(T_k_R22,k_R22,T);

    h_R22 = ((k_h)/D1)*Nu_R22;

    % Thermal resistance of inner pipe (Rconv,1)
    Rconv1 = 1/(h_R22.*A_r1);

    %Heat transfer rate (Qconv,1) in Watts (W)
    Qconv1 = Qconv2;

    % Calculate new refrigerant
    T = (Qconv1*Rconv1)+T1;

    % Recalculate error
    T_err = abs(T - T_refrig(day));
    % Latest guess for this day
    T_refrig(day) = T;
    T_count = T_count + 1;
end

% Calculate new value of MFR
% MFR = electrical power consumption divided by the enthalpy
% change across the compressor

% Enthalpy before compressor is enthalpy from Tinf1
% Using Superheated table (500kPa) for R22 from DuPont:
h1_T = [5 10 15 20 25 30 35 40 45 50 55 60 65 70 75 80 85 90 95 100] + 273.15;
h1 = [408.7 412.3 416.0 419.6 423.2 426.8 430.4 434.1 437.7 441.3 445.0 448.6 452.3 456.0 459.7 463.4 467.2 470.9 474.7
478.5];

enthalpy1 = spline(h1_T,h1,T);
```

```
        % Entropy after compressor assumes isentropic flow through
        % compressor: entropy is the same after compressor

        % Entropy before compressor (low pressure):
        % Using Superheated table (500kPa) for R22 from DuPont:
        s1_T = [5 10 15 20 25 30 35 40 45 50 55 60 65 70 75 80 85 90 95 100] + 273.15;
        s1 = [1.764 1.777 1.789 1.802 1.814 1.826 1.838 1.849 1.861 1.872 1.884 1.895 1.906 1.916 1.927 1.938 1.948 1.959 1.969
        1.979];

        entropy2 = spline(s1_T,s1,T);

        % Enthalpy after compressor (high pressure)
        % From superheated table at high pressure (1700kPa) for R22 from DuPont:
        h2 = [417.6 422.6 427.5 432.2 436.8 441.2 445.7 450.0 454.4 458.7 463.0 467.2 471.5 475.7 479.9 484.1 488.3 492.6 496.8
        501.0];
        s2 = [1.695 1.711 1.726 1.740 1.754 1.767 1.780 1.792 1.804 1.816 1.828 1.839 1.851 1.862 1.873 1.884 1.894 1.905 1.915
        1.925];

        enthalpy2 = spline(s2,h2,entropy2);

        % Mass flow rate
        MFR= poweruse_prior(day)/(enthalpy2-enthalpy1);

        % Recalculate error
        MFR_err = abs(MFR - MFR_refrig(day));
        % Latest guess for this day
        MFR_refrig(day) = MFR;
        MFR_count = MFR_count + 1;
    end
    disp(['For day ',num2str(day),  ...
        ' the refrigerant temperature = ',num2str(T-CtoK),'C and mass flow rate = ',num2str(MFR),'kg/s.',  ...
        ' The inner pipe temperature = ',num2str(T1-CtoK),'C'])
  end
end
```

```
% Convert back to Celsius
T_refrig = T_refrig - CtoK;
% Output data
data.T_refrig = T_refrig;
data.MFR = MFR_refrig;
writetable(data,'Box ac design - Temperature Calculations - Michael Osgood - HEATING.xlsx');
```

19. Appendix M – Box a/c System MATLAB Calculations – After Cold Pack - Heating Mode

```
% Load the data
data = readtable('Box ac design - Temperature Measurements - Michael Osgood - HEATING - CP Applied.xlsx', 'Range','A:L');
% This is the temperature before the compressor
T2 = data.TempOfSink;
Tinf2 = data.Max_Air_Temp;
poweruse_after = data.Power_Use_after_adding_sink_kW;

% Convert to Kelvin
CtoK = 273.15;
Tinf2 = Tinf2 + CtoK;
T2 = T2 + CtoK;

% Total number of days
ndays = length(T2);

% Error tolerances for values
reltol = 1e-6;

% Initialise the arrays
% Mass flow rate
MFR_refrig = nan(ndays,1);
% Temperature of the refrigerant (K)
T_refrig = MFR_refrig;

% Setup vectors for interpolation of properties
% Data for properties of air
T_air = [0 5 10 15 20 25 30 35] + CtoK;
k_air = [0.02364 0.02401 0.02439 0.02476 0.02514 0.02551 0.02588 0.02625];
v_air = [1.338e-5 1.382e-5 1.426e-5 1.47e-5 1.516e-5 1.562e-5 1.608e-5 1.655e-5];
Pr_air = [0.7362 0.7350 0.7336 0.7323 0.7309 0.7296 0.7282 0.7268];
Tf = ((T2+Tinf2)/2);
```

```
% Data for thermal conductivity of R22 from https://irc.wisc.edu/properties/
% Data used at atmospheric pressure: 101kPa
T_k_R22 = [-40 -30 -20 -10 0 10 20 30 40 50 60 70 80 90 100] + CtoK;
k_R22 = [7.09 7.58 8.08 8.61 9.15 9.71 10.3 10.9 11.5 12.1 12.8 13.4 14.1 14.8 15.5] * 1e-3;

% Data for Prandtl Number of R22 from https://irc.wisc.edu/properties/
% Data used at atmospheric pressure: 101kPa
Pr_R22 = [0.833 0.824 0.817 0.81 0.804 0.798 0.793 0.788 0.783 0.778 0.773 0.769 0.764 0.76 0.755];

% Data for dynamic viscosity of R22 from https://irc.wisc.edu/properties/
T_R22 = [-40 -30 -20 -10 0 10 20 30 40 50 60 70 80 90 100] + CtoK;
u_R22 = [9.73 10.2 10.6 11.1 11.5 12 12.4 12.8 13.3 13.7 14.1 14.6 15 15.4 15.8] * 1e-6;

% Calculate for all the days
for day = 1:ndays
%for day = 1:ndays
   T22 = T2(day);
   Tinf22 = Tinf2(day);
   % Only do analysis if there is data for that day!
   if ~isnan(T22)  &&  ~isnan(Tinf22)  &&  ~isnan(poweruse_after(day))
      % Natural convection calculations for outside the pipe
      % Thermal conductivity of the air
      k = spline(T_air,k_air,Tinf22);
      % Kinematic viscousity of the air
      v = spline(T_air,v_air,Tinf22);
      % Prandtl number of the air
      Pr = spline(T_air,Pr_air,Tinf22);

      % Heat transfer rate
      % Data for Rayleigh number
      g = 9.81;
      B = 1./Tf(day);
      %Area of outer pipe (L = 1) in m^2
      r2 = 0.00476;
      r1 = 0.00405;
      A_r2 = 2*pi*r2;
```

```
A_r1 = 2*pi*r1;
D = 2*r2;
D1 = 2*r1;

%Rayleigh number calculation
Ra = (((g).*(B).*abs(T22-Tinf22).*(D).^3)/(v).^2).*Pr;

%Nusselt number calculation
Nu =(0.825+((0.387*(Ra).^(1/6))/((1+(0.492/Pr).^(9/16)).^(8/27)))).^2;

%Natural Convection heat transfer coefficient (W.m^2.K)
h = (k/D)*Nu;

%Thermal resistance of outer pipe (Rconv,2)
Rconv2 = 1/(h.*A_r2);

%Heat transfer rate (Qconv,2) in Watts (W)
Qconv2 = (T22-Tinf22)./Rconv2;

%Qcyl = Qconv,2 therefore:
%Thermal conductivity of copper pipe (W/(m.K)
k_cop = 355;

%Thermal resistance of pipe (Rcyl)
Rcyl = (log(r2/r1))/(2*pi*k_cop);

% Calculate the inner pipe temperature (T1)
T1 = (Qconv2.*Rcyl)+T22;

% Initial guess for mass flow rate
if day == 1  ||  isnan(MFR_refrig(day-1))
   MFR = 0.02;
else
   MFR = MFR_refrig(day-1);
end
MFR_refrig(day) = MFR;
```

```
% Initial error for mass flow rate
MFR_err = 1e6*reltol*(MFR+1);
MFR_count = 0;
% Iterate guess of mass flow rate
while abs(MFR_err) > reltol*MFR  &&  MFR_count < 100
   % Initial guess of refrigerant temperature
   if day == 1  ||  isnan(T_refrig(day-1))
      T = 273;
   else
      T = T_refrig(day-1);
   end
   T_refrig(day) = T;
   % Initial error for temperature
   T_err = 1e6*reltol*(T+1);
   T_count = 0;

   % Iterate guess of temperature
   while abs(T_err) > reltol*T  &&  T_count < 100
      % Calculation of Reynolds number (using current guess of
      % mass flow rate and temperature)
      Re = 4*MFR / (pi*spline(T_R22,u_R22,T)*D1);
      % Calculation of convective heat transfer coefficient
      if Re < 2300
         % Laminar pipe flow Nusselt number
         % Circular tube, laminar (Ts = constant):
         % Nu = hD/k = 3.66
         Nu_R22 = 3.66;
      else
         % Surface roughness: Drawn Tubing (table 8.1 - Fox and
         % McDonalds - Fluid Mechanics
         e = 0.0015;
         % Friction factor: Haaland formula, after Eq. (8.37)
         f = (1/(-1.8*log10(((e/D1)/3.7).^1.11+(6.9/Re)))).^2;

         % Turbulent pipe flow Nusselt number: Eq. (8-71)
         Pr_Nu = spline(T_k_R22,Pr_R22,T);
```

```
        Nu_R22 = ((f/8)*(Re-1000)*(Pr_Nu)) / (1+(((12.7*(f/8)).^0.5)*(((Pr_Nu).^(2/3))-1)));
    end

    % Forced Convective heat transfer coefficient
    k_h = spline(T_k_R22,k_R22,T);

    h_R22 = ((k_h)/D1)*Nu_R22;

    % Thermal resistance of inner pipe (Rconv,1)
    Rconv1 = 1/(h_R22.*A_r1);

    %Heat transfer rate (Qconv,1) in Watts (W)
    Qconv1 = Qconv2;

    % Calculate new refrigerant
    T = (Qconv1*Rconv1)+T1;

                % Recalculate error
    T_err = abs(T - T_refrig(day));
    % Latest guess for this day
    T_refrig(day) = T;
    T_count = T_count + 1;
end

% Calculate new value of MFR
% MFR = electrical power consumption divided by the enthalpy
% change across the compressor

% Enthalpy before compressor is enthalpy from Tinf1
% Using Superheated table (500kPa) for R22 from DuPont:
h1_T = [5 10 15 20 25 30 35 40 45 50 55 60 65 70 75 80 85 90 95 100] + 273.15;
h1 = [408.7 412.3 416.0 419.6 423.2 426.8 430.4 434.1 437.7 441.3 445.0 448.6 452.3 456.0 459.7 463.4 467.2 470.9 474.7
478.5];

enthalpy1 = spline(h1_T,h1,T);
```

```
        % Entropy after compressor assumes isentropic flow through
        % compressor: entropy is the same after compressor

        % Entropy before compressor (low pressure):
        % Using Superheated table (500kPa) for R22 from DuPont:
        s1_T = [5 10 15 20 25 30 35 40 45 50 55 60 65 70 75 80 85 90 95 100] + 273.15;
        s1 = [1.764 1.777 1.789 1.802 1.814 1.826 1.838 1.849 1.861 1.872 1.884 1.895 1.906 1.916 1.927 1.938 1.948 1.959 1.969
        1.979];

        entropy2 = spline(s1_T,s1,T);

        % Enthalpy after compressor (high pressure)
        % From superheated table at high pressure (1700kPa) for R22 from DuPont:
        h2 = [417.6 422.6 427.5 432.2 436.8 441.2 445.7 450.0 454.4 458.7 463.0 467.2 471.5 475.7 479.9 484.1 488.3 492.6 496.8
        501.0];
        s2 = [1.695 1.711 1.726 1.740 1.754 1.767 1.780 1.792 1.804 1.816 1.828 1.839 1.851 1.862 1.873 1.884 1.894 1.905 1.915
        1.925];

        enthalpy2 = spline(s2,h2,entropy2);

        % Mass flow rate
        MFR= poweruse_after(day)/(enthalpy2-enthalpy1);

        % Recalculate error
        MFR_err = abs(MFR - MFR_refrig(day));
        % Latest guess for this day
        MFR_refrig(day) = MFR;
        MFR_count = MFR_count + 1;
    end
    disp(['For day ',num2str(day),  ...
        ' the refrigerant temperature = ',num2str(T-CtoK),'C and mass flow rate = ',num2str(MFR),'kg/s.',  ...
        ' The inner pipe temperature = ',num2str(T1-CtoK),'C'])
  end
end
```

```
% Convert back to Celsius
T_refrig = T_refrig - CtoK;
% Output data
data.T_refrig = T_refrig;
data.MFR = MFR_refrig;
writetable(data,'Box ac design - Temperature Calculations - Michael Osgood - HEATING - CP Applied.xlsx');
```

20. Appendix N – Comparison of Refrigerant Temperature and Power Consumption Before Modification with Flowing Water Loop and After Modification using 3 Different Mass Flow Rate Water Pumps – Household a/c - Cooling Mode

BEFORE MODIFICATION				AFTER MOD WITH 1.33kg/s pump		AFTER MOD WITH 0.83kg/s pump		AFTER MOD WITH 0.42kg/s pump	
MFR (kg/s)	enthalpy2 (kJ/kg)	T refrig (°C)	poweruse (kW)	T refrig (°C)	Poweruse (kW)	T refrig (°C)	Poweruse (kW)	T refrig (°C)	Poweruse (kW)
0.158	480.620	33.0	5.030	37.3	4.254	35.7	4.535	34.4	4.766
0.126	479.303	32.0	3.990	37.4	3.217	35.3	3.498	33.7	3.730
0.193	477.311	30.5	6.060	34.0	5.269	32.7	5.553	31.6	5.787
0.182	473.922	28.0	5.590	31.7	4.792	30.3	5.079	29.2	5.316
0.209	471.839	26.5	6.350	29.8	5.540	28.5	5.830	27.5	6.068
0.126	473.918	28.0	3.880	33.4	3.093	31.4	3.380	29.7	3.615
0.143	475.967	29.5	4.460	34.2	3.675	32.5	3.960	31.0	4.194
0.188	473.919	28.0	5.780	31.6	4.980	30.3	5.268	29.1	5.505
0.128	472.534	27.0	3.910	32.3	3.119	30.3	3.406	28.7	3.643
0.117	479.967	32.5	3.700	38.3	2.931	36.1	3.211	34.3	3.442
0.156	479.307	32.0	4.940	36.3	4.161	34.7	4.443	33.4	4.675
0.180	479.306	32.0	5.700	35.8	4.916	34.3	5.199	33.2	5.431
0.175	480.622	33.0	5.560	36.9	4.780	35.4	5.062	34.2	5.293
0.193	479.305	32.0	6.100	35.5	5.314	34.2	5.597	33.1	5.829
0.138	480.619	33.0	4.400	37.9	3.628	36.0	3.908	34.5	4.139
0.112	487.761	38.5	3.670	44.5	2.940	42.2	3.204	40.4	3.423
0.042	481.932	34.0	1.350	50.0	0.619	44.0	0.887	39.0	1.108
0.267	479.304	32.0	8.460	34.5	7.660	33.6	7.944	32.8	8.176
0.134	472.535	27.0	4.080	32.1	3.288	30.2	3.575	28.6	3.813
0.141	479.308	32.0	4.460	36.8	3.683	35.0	3.965	33.5	4.198

0.189	481.931	34.0	6.060	37.6	5.281	36.2	5.561	35.1	5.791
0.183	472.535	27.0	5.590	30.7	4.787	29.3	5.076	28.2	5.314
0.206	473.919	28.0	6.350	31.3	5.547	30.1	5.835	29.0	6.071
0.127	473.227	27.5	3.880	32.9	3.090	30.9	3.378	29.2	3.614
0.144	475.286	29.0	4.460	33.7	3.673	31.9	3.959	30.5	4.194
0.183	478.640	31.5	5.780	35.2	4.994	33.8	5.277	32.7	5.510
0.126	475.286	29.0	3.910	34.4	3.127	32.4	3.412	30.7	3.647
0.121	472.532	27.0	3.700	32.6	2.910	30.5	3.197	28.8	3.435
0.121	479.302	32.0	3.840	37.6	3.068	35.5	3.349	33.8	3.581
0.122	473.222	27.5	3.750	33.1	2.962	31.0	3.249	29.3	3.485
0.141	479.305	32.0	4.460	36.8	3.684	35.0	3.966	33.5	4.198
0.166	477.977	31.0	5.210	35.1	4.425	33.6	4.709	32.3	4.942
0.197	477.978	31.0	6.210	34.4	5.420	33.1	5.704	32.1	5.937
0.106	471.139	26.0	3.200	32.5	2.409	30.0	2.698	28.0	2.936
0.165	472.535	27.0	5.030	31.1	4.231	29.6	4.519	28.3	4.757
0.131	472.535	27.0	3.990	32.2	3.198	30.3	3.486	28.6	3.723
0.195	475.284	29.0	6.060	32.5	5.263	31.2	5.549	30.1	5.785
0.182	473.919	28.0	5.590	31.7	4.792	30.3	5.079	29.2	5.316
0.207	473.228	27.5	6.350	30.8	5.545	29.6	5.833	28.5	6.070
0.181	475.287	29.0	5.610	32.8	4.815	31.3	5.102	30.2	5.337
0.143	475.966	29.5	4.460	34.2	3.675	32.5	3.960	31.0	4.194
0.184	477.975	31.0	5.780	34.7	4.992	33.3	5.276	32.2	5.509
0.181	483.228	35.0	5.840	38.7	5.068	37.3	5.345	36.2	5.574
0.210	474.603	28.5	6.480	31.7	5.678	30.5	5.965	29.5	6.202
0.203	469.035	24.5	6.060	27.9	5.239	26.6	5.533	25.6	5.775
0.181	474.605	28.5	5.590	32.3	4.794	30.8	5.080	29.7	5.317
0.206	474.604	28.5	6.350	31.8	5.549	30.6	5.836	29.5	6.072
0.214	483.230	35.0	6.890	38.1	6.111	37.0	6.389	36.0	6.618
0.162	471.837	26.5	4.940	30.7	4.139	29.1	4.428	27.8	4.666
0.183	475.964	29.5	5.700	33.2	4.907	31.8	5.193	30.7	5.428
0.180	474.604	28.5	5.560	32.3	4.764	30.9	5.051	29.7	5.287
0.195	476.639	30.0	6.100	33.5	5.307	32.2	5.592	31.1	5.826

0.144	473.226	27.5	4.400	32.2	3.607	30.5	3.895	29.0	4.132
0.123	468.332	24.0	3.670	29.6	2.865	27.5	3.157	25.8	3.398
0.044	473.226	27.5	1.350	43.0	0.591	37.1	0.866	32.4	1.100
0.277	472.533	27.0	8.460	29.5	7.639	28.5	7.929	27.8	8.167
0.141	462.719	20.0	4.080	24.9	3.240	23.1	3.542	21.5	3.794
0.148	469.741	25.0	4.460	29.6	3.654	27.9	3.945	26.5	4.186
0.176	466.934	23.0	5.210	26.9	4.385	25.4	4.681	24.2	4.926
0.209	467.635	23.5	6.210	26.8	5.381	25.5	5.677	24.5	5.921
0.107	469.038	24.5	3.200	30.9	2.402	28.5	2.692	26.5	2.932
0.171	465.534	22.0	5.030	26.0	4.198	24.5	4.497	23.3	4.744
0.134	468.334	24.0	3.990	29.1	3.182	27.2	3.474	25.6	3.716
0.198	472.535	27.0	6.060	30.4	5.254	29.1	5.543	28.1	5.781
0.182	473.228	27.5	5.590	31.2	4.790	29.8	5.078	28.7	5.315
0.211	469.736	25.0	6.350	28.2	5.531	27.0	5.823	26.0	6.065
0.133	464.834	21.5	3.880	26.7	3.055	24.7	3.353	23.1	3.600
0.157	459.844	18.0	4.460	22.4	3.597	20.8	3.906	19.4	4.164
0.197	465.534	22.0	5.780	25.5	4.942	24.2	5.242	23.1	5.489
0.138	459.109	17.5	3.910	22.5	3.048	20.6	3.357	19.1	3.616
0.127	464.131	21.0	3.700	26.4	2.872	24.4	3.171	22.7	3.419
0.134	461.290	19.0	3.840	24.2	2.993	22.2	3.298	20.6	3.552
0.126	468.336	24.0	3.750	29.4	2.944	27.4	3.236	25.7	3.478
0.086	459.835	18.0	2.460	26.0	1.623	23.0	1.925	20.5	2.178
Averages		**27.892**	**4.947**	**32.565**	**4.150**	**30.808**	**4.437**	**29.368**	**4.675**

21. Appendix O – Comparison of Refrigerant Temperature and Power Consumption Before Modification with Flowing Water Loop and After Modification using 3 Different Mass Flow Rate Water Pumps – Household a/c - Heating Mode

BEFORE MODIFICATION				AFTER MOD WITH 1.33kg/s pump		AFTER MOD WITH 0.83kg/s pump		AFTER MOD WITH 0.42kg/s pump	
MFR (kg/s)	enthalpy1 (kJ/kg)	T refrig (°C)	poweruse (kW)	T refrig (°C)	poweruse (kW)	T refrig (°C)	poweruse (kW)	T refrig (°C)	poweruse (kW)
0.1198	333.75	38.5	4.940	38.0	4.702	38.2	4.792	38.3	4.865
0.1562	363.51	44.0	4.650	43.6	4.443	43.8	4.521	43.9	4.585
0.1667	361.25	43.5	5.080	43.1	4.871	43.3	4.950	43.4	5.014
0.1749	379.07	48.0	4.560	47.7	4.374	47.8	4.444	47.9	4.502
0.1302	354.07	42.0	4.280	41.5	4.062	41.7	4.144	41.9	4.212
0.1530	373.74	46.5	4.150	46.1	3.957	46.3	4.030	46.4	4.089
0.1817	375.54	47.0	4.860	46.7	4.669	46.8	4.741	46.9	4.800
0.1540	351.45	41.5	5.210	41.1	4.990	41.3	5.073	41.4	5.141
0.1143	333.82	38.5	4.710	38.0	4.472	38.2	4.562	38.4	4.635
0.1219	356.59	42.5	3.900	42.0	3.685	42.2	3.766	42.4	3.832
0.2199	375.53	47.0	5.880	46.7	5.689	46.8	5.761	46.9	5.820
0.1809	354.01	42.0	5.950	41.7	5.733	41.8	5.814	41.9	5.882
0.2085	363.48	44.0	6.210	43.7	6.004	43.8	6.081	43.9	6.145
0.3406	385.46	50.0	8.560	49.8	8.385	49.9	8.451	50.0	8.505
0.2072	375.53	47.0	5.540	46.7	5.349	46.8	5.421	46.9	5.480
0.2788	377.30	47.5	7.360	47.3	7.172	47.4	7.243	47.4	7.301
0.2303	371.77	46.0	6.350	45.7	6.154	45.8	6.228	45.9	6.288
0.1985	354.00	42.0	6.530	41.7	6.313	41.8	6.395	41.9	6.462
0.2173	367.75	45.0	6.210	44.7	6.009	44.8	6.085	44.9	6.147
0.2533	382.37	49.0	6.470	48.8	6.290	48.9	6.358	48.9	6.413

0.2601	417.53	65.1	6.830	64.8	6.716	64.9	6.759	65.0	6.794
0.2108	373.68	46.5	5.720	46.2	5.527	46.3	5.600	46.4	5.659
0.2332	379.05	48.0	6.080	47.7	5.895	47.8	5.964	47.9	6.022
0.2235	361.23	43.5	6.810	43.2	6.601	43.3	6.680	43.4	6.744
0.3276	410.84	61.0	8.310	60.8	8.181	60.9	8.230	61.0	8.269
0.3360	398.70	55.0	8.220	54.8	8.067	54.9	8.125	55.0	8.172
0.3014	389.82	51.5	7.450	51.3	7.282	51.4	7.345	51.5	7.397
0.2669	409.94	60.5	6.740	60.3	6.610	60.4	6.659	60.5	6.699
0.1991	367.78	45.0	5.690	44.7	5.489	44.8	5.565	44.9	5.627
0.2885	395.11	53.5	7.060	53.3	6.901	53.4	6.961	53.5	7.010
0.2703	400.97	56.0	6.630	55.8	6.481	55.9	6.537	56.0	6.583
0.2988	404.15	57.5	7.380	57.3	7.238	57.4	7.291	57.5	7.335
0.3160	386.96	50.5	7.890	50.3	7.717	50.4	7.782	50.5	7.835
0.2799	382.35	49.0	7.150	48.8	6.970	48.9	7.038	48.9	7.093
0.3176	411.71	61.5	8.090	61.3	7.963	61.4	8.011	61.5	8.050
0.3141	396.32	54.0	7.680	53.8	7.523	53.9	7.582	54.0	7.630
0.3237	400.96	56.0	7.940	55.8	7.792	55.9	7.847	56.0	7.893
0.3021	398.71	55.0	7.390	54.8	7.237	54.9	7.295	55.0	7.342
0.5269	419.82	66.5	14.000	66.4	13.890	66.5	13.932	66.5	13.965
0.2835	386.98	50.5	7.080	50.3	6.907	50.4	6.972	50.5	7.025
0.3219	397.53	54.5	7.870	54.3	7.715	54.4	7.773	54.5	7.821
0.3018	389.82	51.5	7.460	51.3	7.292	51.4	7.355	51.5	7.407
0.2521	388.43	51.0	6.260	50.8	6.089	50.9	6.154	50.9	6.206
0.1794	365.69	44.5	5.230	44.2	5.026	44.3	5.103	44.4	5.166
0.1168	323.25	37.0	5.410	36.4	5.163	36.7	5.256	36.8	5.332
0.1100	307.17	35.0	6.040	34.4	5.780	34.6	5.878	34.8	5.958
0.2657	393.84	53.0	6.510	52.8	6.348	52.9	6.409	53.0	6.459
0.2469	393.85	53.0	6.050	52.8	5.888	52.9	5.949	52.9	5.999
0.2582	392.54	52.5	6.340	52.3	6.176	52.4	6.238	52.4	6.288
0.2498	405.18	58.0	6.190	57.8	6.050	57.9	6.103	58.0	6.146
0.1222	367.82	45.0	3.490	44.5	3.289	44.7	3.365	44.9	3.427
0.2601	388.42	51.0	6.460	50.8	6.289	50.9	6.354	50.9	6.406

0.2728	396.33	54.0	6.670	53.8	6.513	53.9	6.572	53.9	6.620
0.3676	388.40	51.0	9.130	50.8	8.959	50.9	9.024	51.0	9.076
0.3304	382.33	49.0	8.440	48.8	8.260	48.9	8.328	49.0	8.383
0.2894	398.71	55.0	7.080	54.8	6.927	54.9	6.985	55.0	7.032
0.2912	388.43	51.0	7.230	50.8	7.059	50.9	7.124	51.0	7.176
0.2786	391.20	52.0	6.860	51.8	6.694	51.9	6.756	52.0	6.808
0.2597	389.83	51.5	6.420	51.3	6.252	51.4	6.315	51.4	6.367
0.2699	386.95	50.5	6.740	50.3	6.567	50.4	6.632	50.4	6.685
0.2616	393.83	53.0	6.410	52.8	6.248	52.9	6.309	52.9	6.359
0.2552	397.54	54.5	6.240	54.3	6.085	54.4	6.143	54.5	6.191
0.2585	389.83	51.5	6.390	51.3	6.222	51.4	6.285	51.4	6.337
0.2590	385.48	50.0	6.510	49.8	6.334	49.9	6.401	49.9	6.455
0.2379	377.30	47.5	6.280	47.2	6.092	47.3	6.163	47.4	6.221
0.2431	379.03	48.0	6.340	47.7	6.155	47.9	6.224	47.9	6.282
0.2786	391.20	52.0	6.860	51.8	6.694	51.9	6.756	52.0	6.808
0.3018	389.82	51.5	7.460	51.3	7.292	51.4	7.355	51.5	7.407
0.2469	393.85	53.0	6.050	52.8	5.888	52.9	5.949	52.9	5.999
0.1794	365.69	44.5	5.230	44.2	5.026	44.3	5.103	44.4	5.166
0.3160	386.96	50.5	7.890	50.3	7.717	50.4	7.782	50.5	7.835
0.1794	365.68	44.5	5.230	44.2	5.026	44.3	5.103	44.4	5.166
Averages		**49.7**	**6.532**	**49.4**	**6.353**	**49.538**	**6.421**	**49.6**	**6.476**