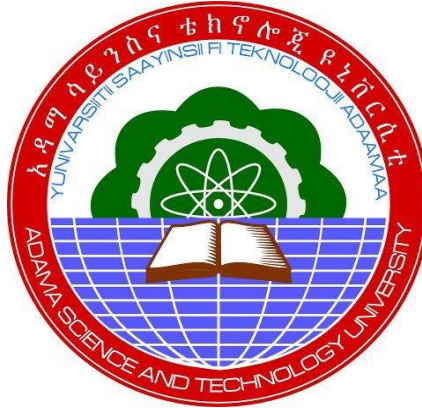


EXPERIMENTAL INVESTIGATION OF PHASE CHANGE
MATERIAL INTEGRATED VAPOR COMPRESSION
REFRIGERATION SYSTEM



By

Fikresendek Fikadu Birru

A Thesis Submitted to the Department of Thermal and Aerospace Engineering

School of Mechanical, Chemical & Materials Engineering

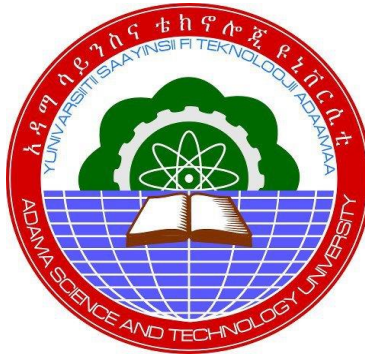
Office of Graduate Studies

Adama Science and Technology University

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July, 2020

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Date

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ABSTRACT

Usage of vapor compression refrigeration system imported from abroad has become very common in household and small to large scale food and beverage serving businesses in Ethiopia. But, the frequent and also sudden electric power interruption and cut for about eight hours in Ethiopia is causing spoilage of food items & products stored in the refrigerators and also shortage of cold drinks in commercial places. This problem can be solved by integrating (PCM) with the condenser and evaporator of any vapor compression refrigerator system.

The major objectives of this thesis are to analytically design and experimentally investigate the performance of a locally constructed vapor compression refrigeration system with and without integration of three different PCM on the evaporator and condenser of the refrigerator.

The main components of vapor compression refrigeration system were selected and analytically designed. PCM, Filter drier, and cold thermostat is selected based on the requirement. Finally, the refrigerator parts are assembled and leakage & pressure test were conducted. The refrigerator has a cooling capacity of 0.433 kW with compressor power input of 0.186 kW. Then, performance of the refrigerator was analyzed in terms of coefficient of performance, exergetic efficiency, on-off ratio, on-off frequency and measurement of temperatures of condenser and evaporator.

Experimental investigation done by integrating paraffin wax PCM (in the thickness range of 2 up to 5 mm) on the condensers shows a decrease in on-off ratio and 11-34% increase in COP. From the thickness-based investigation, it was observed that increasing the PCM thickness more than 2 mm causes decrease in the COP of the refrigerator. Integrating sodium chloride PCM on the evaporator gives a 2-9% COP increase, a decrease on-off ratio, and an increase on-off frequency. From the experiment 18 °C temperature fluctuation of the refrigerated cabinet was observed over three hours of running the refrigerator without PCM. But this fluctuation was reduced to 1.6 °C when integrating sodium chloride PCM on the evaporator. The experimental test done by integrating PCM both on the condenser and evaporator give: less temperature fluctuation, a 7-27% COP increase, increase on-off ratio, and decrease on-off frequency. The effect of integration of paraffin wax PCM on the condenser was also done on a standard household refrigerator (cooling capacity 1 kW; compressor power 0.25 kW) on freezing water of 30 kg which results in increase of COP by 26%. was obtained.

Key word: PCM; COP; VCRS; on-off ratio; on-off frequency.

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LIST OF SYMBOLS

Capital letters

<i>A</i>	Area (m^2)
<i>C</i>	Heat capacity ($\text{kJ} \cdot K^{-1}$)
<i>E</i>	Energy (kJ)
<i>F</i>	Correction factor
<i>G</i>	Mass velocity ($\text{kg} \cdot s^{-1} m^{-2}$)
<i>H</i>	Enthalpy (kJ)
<i>I</i>	Irreversibility (kW)
<i>K</i>	Thermal conductivity ($\text{W} m^{-1} K^{-1}$)
<i>L</i>	Length (m)
<i>P</i>	Power (kW)
<i>Q</i>	Heat (kJ)
<i>S</i>	Entropy ($\text{kJ} \cdot K^{-1}$)
<i>T</i>	Temperature (K)
<i>U</i>	Aver all heat transfer coefficient ($\text{W} \cdot m^{-2} K^{-2}$)
<i>V</i>	Volume (m^3)

Small letters

<i>a</i>	Diffusivity ($m^2 s^{-1}$)
<i>c</i>	Specific heat ($\text{kJ} \cdot \text{kg}^{-1} K^{-1}$)
<i>d</i>	Diameter (m)
<i>h</i>	Specific enthalpy ($\text{kJ} \cdot \text{kg}^{-1} K^{-1}$)
<i>l</i>	Latent heat (kJ)
<i>p</i>	Pressure (kPa)
<i>q</i>	Specific heat ($\text{kJ} \cdot \text{kg}^{-1}$)
<i>r</i>	Radius (m)
<i>s</i>	Specific Entropy ($\text{kJ} \cdot K^{-1} \text{kg}^{-1}$)
<i>t</i>	Time (s)
<i>v</i>	Velocity ($m \cdot s^{-2}$)
<i>x</i>	Dryness factor

Subscripts

<i>a</i>	Air	<i>o</i>	Outlet
<i>ac</i>	Actual	<i>s</i>	Solid
<i>ca</i>	Calculated	<i>fg</i>	Vaporization
<i>con</i>	Condenser	<i>v</i>	Vapor
<i>com</i>	Compressor	1	Compressor inlet
<i>e</i>	Evaporator	2	Condenser inlet
<i>f</i>	Liquid	3	Condenser outlet
<i>g</i>	Gaseous	4	Evaporator inlet
<i>i</i>	Inlet		

Greek letters

Λ	Specific enthalpy of fusion $kJ.kg^{-1}$
ε	Emissivity
μ	Viscosity ($kg.m^{-1}s^{-1}$)
N	Kinetic viscosity (m^2s^{-1})
π	Pi
ρ	Density ($kg.m^{-3}$)
σ	Entropy generated ($kW.K^{-1}$)
τ	Time (s)

Dimensionless number

Bi	Biot number
Re	Reynolds number
Nu	Nusselt number
Pr	Prandtl number

Abbreviation

ASTU	Adama science and technology university	R134a	Tetrafluoro ethane
CFC	Chlorofluorocarbon	R125	Pentafluoro ethane
COP	Coefficient of performance	R152a	Difluoro ethane
GWP	Global warming potentials	R600	Butane
HCFC	Hydro Chlorofluorocarbon	R290	Propane
HFC	Hydrofluorocarbon	R1270	Methylene Fluoride
ODP	Ozone depletion potential	R600a	Isobutane
PCM	Phase change material	R717	Ammonia
PW	Paraffin wax	CaCl ₂	Calcium chloride
PCT	Phase change temperature	NaCl ₂	Sodium chloride
R134a	Tetrafluoro ethane	Water	H ₂ O
SS	Shape stabilized		

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CHAPTER ONE

INTRODUCTION

1.1 Background

Refrigeration is needed in preserving and storing drinks, foods, medicines, and chemicals. It is important in food preservation, as low temperatures assist in preventing or delaying microbial, physiological, and chemical changes in food (Donhowe and Hartel, 1996). Maintaining stable temperatures in the storage and transportation of frozen foods is essential for products such as ice cream, frozen meat medicine, etc. Frozen foods must be kept at least below 18°C, and this temperature should be maintained from small domestic freezers, to larger cold stores and warehouses, as well as during moving in refrigerated trucks. However, during storage and transportation, frozen foods may experience temperature fluctuations due to heat loads imposed on the system.

The effect of temperature storage conditions has been studied for a number of food products such as vegetables (Cruz *et al.*, 2009), frozen dough (Phimolsiripol *et al.*, 2008), ice cream (Donhowe and Hartel, 1996a, b; Flores and Goff, 1999), and meat (Duun *et al.*, 2008; Yanniotis *et al.*, 2008). Duun *et al.* (2008) and Yanniotis *et al.* (2008) studied the drip loss of pork roast stored and bovine muscle (*Musculus semimembranosus*), respectively under different conditions. They found that there was greater drip loss on the foods when they stored in the refrigerator with temperature fluctuation than in the refrigerator with less temperature fluctuation. Ice cream has also been the subject of investigations into the effects of storage conditions. The investigation shows temperature fluctuation in ice cream storage will make the ice cream crystal larger than desirable. (Flores and Goff, 1999)

In Ethiopia, there is continues electric power interruption and fluctuation throughout the year. This electric power interruption and fluctuation is not only creating problems on foods stored in the refrigerator but also cause a shortage of cold drinks in hotels, restaurants and other places where cold drinks are in need. To decrease such effects refrigerator can be modified by integrating the compartment walls and the condenser with phase changing material (PCM) which are commonly known as PCM integrated cold storage. PCM integrated cold storage is a refrigeration device that uses PCM as cold thermal energy storage. Cold thermal energy storage is a process that involves

adding cold thermal energy to a medium and extracting it whenever it is needed. During the charging process, the available cold thermal energy can be accumulated into the storage medium. During the discharging process, the stored cold thermal energy is retrieved and supplied for the end-user. The cold thermal energy can be stored under a change in internal energy or phase transformation of the storage medium. (He and Setterwall, 2002 and Qureshi *et al.*, 2011).

Domestic refrigerators are among the high energy-consuming appliances. In many countries, efforts have been undertaken to enhance the thermal efficiency of domestic refrigerators, and such improvements have been focused mainly on compressor operation and electronic control. The use of phase change materials (PCMs) in domestic refrigerators decreases the cycling period of the compressor and reduces internal temperature fluctuations (Zarajabad and Ahmadi, 2018). Integrating phase change materials (PCMs) with domestic refrigerators is also one of the effective ways to reduce electric energy consumption and thermal environmental pollutions by using it as damper of the thermal energy rejected in the condenser; this may also help in sub cooling the refrigerant leaving the condenser.

Some of the common problem encountered in using standard refrigeration systems such as high electric power consumption, thermal environmental pollution and loss of cold item and foods due to thermal load fluctuation and interruption of electricity can be controlled through integration of phase change materials (PCMs) with domestic refrigeration systems.

This thesis focuses on the design, prototype construction and experimental performance evaluation of a vapor-compression refrigeration systems integrated with a PCM thermal storage system around the evaporator and condenser.

Figure 1.1 shows the SOLIDWORK model of the vapor compression refrigeration system refrigerator and Figure 1.2 shows its fabricated prototype. Vapor compression refrigeration system have four main components: compressor, condenser, thermal expansion valve, and evaporator. PCM can be integrated into evaporator, condenser, and refrigerator compartment.

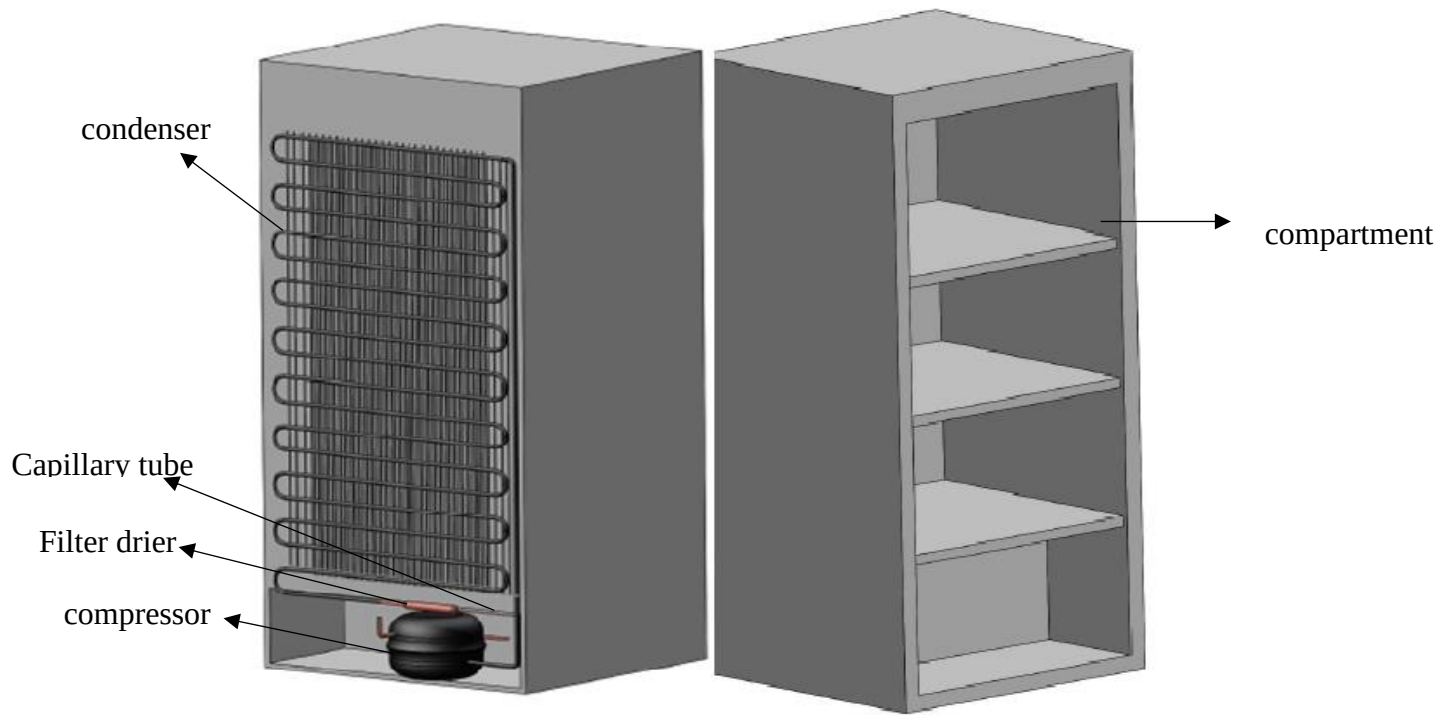


Figure 1.1: SOLIDWORK model of VCRS refrigerator body



Figure 1.2: The front and side view of assembled refrigerator prototype

Figure 1.3 shows the main components of vapor compression refrigeration system components used to fabricate the refrigerator. A compressor is the most important part of the refrigerator. Its function is continuously drawing the refrigerant vapor from the evaporator. So that the low pressure and low temperature can be maintained at the evaporator. The compressor has to raise the pressure and the temperature of the refrigerant so that it can condense by rejecting heat to the cooling medium. The condenser is a device used to condense the refrigerant from its gaseous to its liquid state, by cooling it by extracting its latent heat and transferred to the surrounding environment. A capillary tube is a fixed length of small diameter tubing lying between the outlet of the condenser and the inlet of the evaporator. It reduces the pressure of the refrigerant from the high-pressure side to the low-pressure side of the system and controls the flow of refrigerant to the evaporator. Evaporator is a device which cooling takes place. The refrigerant pass through evaporator extract heat from the refrigerated compartment. A filter drier is used to remove any particle such as dirt, metal, or chips from entering the refrigerant flow control. The cold thermostat is a switching device used to maintain the refrigerator temperature at the required level. Refrigerant is the fluid that circulates in the refrigeration system to provide cooling for the refrigerator at the evaporator portion of the refrigerator.



Figure.1.3: Refrigerator components

Figure 1. 4 shows the ideal p-h diagram of vapor compression refrigeration system. In ideal vapor compression refrigeration system the circulating refrigerant enters the compressor in the thermodynamic state known as a saturated vapor and it compressed in to a higher pressure, resulting in a higher temperature as well. The superheated refrigerant leaves the compressor and enters in to the condenser at state 2. At the condenser the refrigerant release its sensible and latent heat to the surrounding and become saturated liquid and enters in to the capillary tube in state 3.

In the capillary tube the temperature and pressure of the refrigerant dropped because of the pressure dropped caused by the friction and change in kinetic energy of the refrigerant. The vapour-liquid mixture refrigerant leaves the capillary tube at low pressure and temperature and enters into the evaporator at state four. In the evaporator the refrigerant liquid evaporates by extracting the heat from the refrigerator compartment. To complete the refrigeration cycle, the refrigerant vapor from the evaporator is again a saturated vapor and is routed back into the compressor.

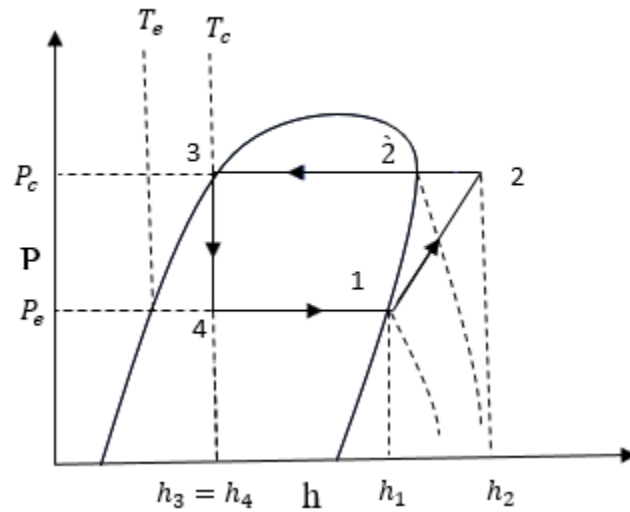


Figure 1.4: p-h diagram of VCRS (Arora, 2010)

The Condenser is placed at the backside of the refrigerator by screw the condenser in to the body of the refrigerator for support system and in order to gain enough space for PCM insertion and avoid vibration during working of compressor it tightly warped to the body by inserting bench with the required gap. Finally, it brazed one end with a compressor and other with filter drier by acetylene and oxygen gas welding. Evaporator warped in the cabinet of the refrigerator evaporator installation gives approximately uniform temperature in the compartment which was very essential for the experimentation. It also suitable to the evaporator PCM placement, and it brazed with the capillary tube in one end and with the compressor on the other end. Capillary tube brazed with evaporator inlet and filter drier.

Figure 1.5 shows the PCM integrated refrigerator at the condenser and evaporator. Two kinds of PCM (paraffin wax and calcium chloride-based salt hydrant) used in the condenser to increase the performance of the refrigerator. In the evaporator sodium chloride-based salt hydrant are used to increase the performance of the evaporator as well as reducing temperature fluctuation of

refrigerator cabinet during the time of power loss. Temperature of condenser, evaporator, compartment, ambient, and refrigerated products are measured to study the performance of the refrigerator. The performance of the ordinary refrigerator at no-load and with load are tested and compared the result with the performance of the PCM integrated refrigerator.



Figure 1.5: PCM integrated refrigerator: a: - back side view b: - frontal view,

1.2 Statement of the Problem

Nowadays in Ethiopia refrigerators are widely used at home, restaurants, hospitals etc. to store beverages, meats, medicine and many other perishable items using refrigerators procured by hard currencies from abroad. The design, fabrication and performance test of locally fabricated

refrigerator that work on vapor compression refrigeration system is limited in Ethiopia where there exists continuous electric power supply interruption. Power fluctuation and power cut lead to drip loss in meat and spoilage of other foods and it also creates a shortage of cold drinks in hotels, bars restaurants, etc. these problems are broadly seen in Ethiopia as there is continuous power fluctuation and losses throughout the year. Power fluctuation causes several problems in food preservation and storing of things that should be kept in refrigerator. Refrigerators should use phase change material (PCM) for preventing temperature fluctuation in cold storages and to increase cooling time after a power cut. Current refrigerators worldwide use wire and tube condenser for heat rejection to the surrounding. As heat rejection on the condenser has main effect on the performance and power consumption of the refrigerator, researches should be done to increase the heat rejection performance of the condenser and one of the ways to increase heat rejection on the condenser is that using PCM as heat sink in the condenser.

1.3 Objectives of the Study

1.3.1 General Objective

The general objective is analytically and experimentally investigating locally constructed vapor compression refrigerator system integrated with phase changing materials (PCM).

1.3.2 Specific Objectives

The specific objectives of the thesis are

- Assessing parameters that affect the performance of PCM integrated refrigerator.
- Designing the major components of the refrigerator that operates on the vapor compression refrigeration cycle.
- Constructing prototype of the refrigerator on the basis of the design results using materials purchased from local market.
- Evaluating the performance of the refrigerator.

1.4 Significance of the Study

The study is mainly intended to fabricate and evaluate the performance of PCM integrated refrigerator that can reduce the power consumption of refrigeration by increasing heat rejection at the condenser and at the evaporator at peak load time. It also reduces the temperature fluctuation of refrigerator during power loss and fluctuation that leads to the good preservation of foods and

vegetables in refrigerator and longer cooling cold drinks after a power cut. It also reduces the power consumption of the compressor by reducing condenser pressure.

1.5 Scope and Limitation of the Study

The study will conduct a design of PCM integrated refrigerator based on the vapor compression refrigeration cycle, and experimental investigation of PCM integrated refrigerator. This thesis includes Validation of experimental results with the numerical result, comparison of experimental results with the literature.

1.6 Organization of the study

This thesis is organized with 6 chapters. The first chapter introduces the concept of vapor absorption and refrigeration system and its component, literature survey on the same is presented in chapter two with its summary and gap. The third chapter presents the selection and sizing of system components and different PCMs for the experiment. The next chapter describes the construction process of the prototype and experimental procedure for the setup. The fifth chapter is composed of result and discussion of major experimental findings and their implication. The six chapter includes the conclusion of the whole work and recommendation with future scopes for further study.

1.7 Methodology

Analytical and experimental approaches was used as a methodology in this study. The analytical (theoretical) approach uses the governing equation in each analysis of components. For different analyses and modeling SOLIDWORK, Microsoft Excel, Microsoft word, Cool selector software was used. The experimental approach was used in this thesis to test and validate with an analytically developed model results at the same working condition. Therefore, to achieve the required objective, this study will use both analytical and experimental methods. The temperature of the evaporator, PCM, condenser, ambient, and products are measured to determine the performance of the refrigerator.

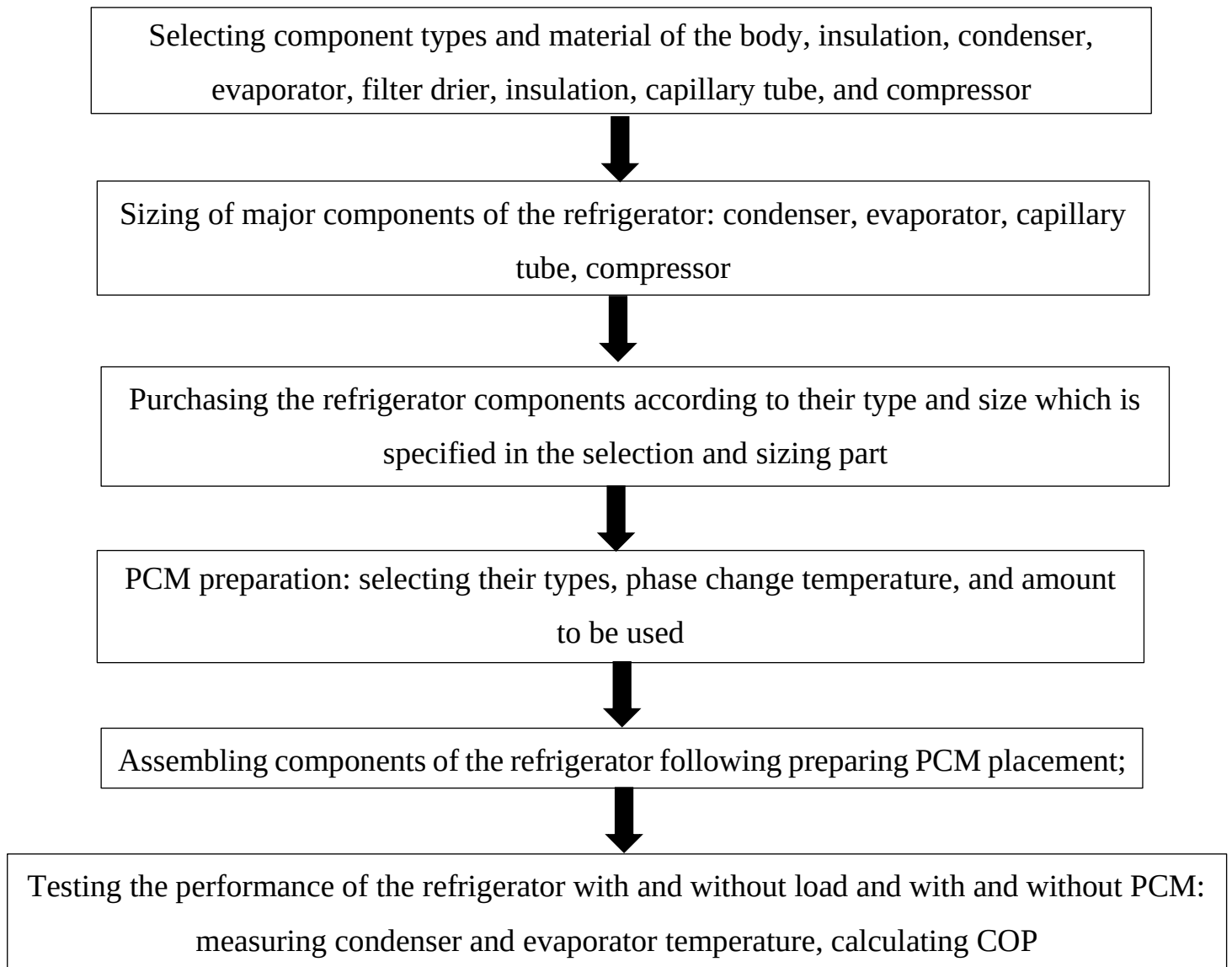


Figure 1.6: Flow chart of methodology

CHAPTER TWO

LITERATURE REVIEW

2.1 Introduction

In this chapter, available literature related to this study are reviewed within the purview of the present investigation. Review of brief history of VCRS, effects of temperature fluctuation on products, cold thermal storage PCMs and PCM integrated refrigerators, research studies from the literature are described in the following section.

2.2 Brief History of Vapor Compression Systems

The method of production of cold by mechanical process is quite recent. Back in 1778, William Colton of Glasgow University produced refrigeration by creating partial vacuum over ethyl ether. But he could not implement his experience in practice. The first development takes place in 1834 when Perkins proposed a hand-operated compressor machine working on ether. Then in 1851 came Gorrie's air refrigeration machine, and in 1856 Lunde developed a machine working on ammonia. The pace of development was slow at the beginning when steam engines are the only prime movers known to run the compressor. With the advent of electric motors consequent higher speed of compressor, the scope of application is widened. The pace of the development was considerably quickened in the 1920 decade when du Pont put a family of new working substances, the chloro-fluoro-chloro derivative of methane, ethane, etc. popularly known as chloro fluorocarbons CFCs under the name of Freons (**Arora, 2010**).

This day's domestic refrigerators working on the vapor compression refrigeration system are widely used in the whole world for preserving and storing drinks, foods, medicines, and chemicals. This makes domestic refrigerators are among the high energy-consuming appliances. In many countries, efforts have been undertaken to enhance the thermal efficiency of domestic refrigerators, and such improvements have been focused mainly on compressor operation, electronic control, and heat transfer enhancement of condenser and evaporator. Recent improvement done on the domestic refrigerators condenser and evaporator use phase change materials (PCMs) to decrease the cycling period of the compressor and reduce internal temperature fluctuations (**Zarajabad and Ahmadi, 2018**). Consequently, it reduces electric energy consumption and thermal environmental pollutions. Recent studies show that some of the common problems encountered in using a standard refrigerator system such as high electric power consumption, thermal

environmental pollution and loss of cold item and foods due to thermal load fluctuation and interruption of electricity can be also controlled through integration of phase change materials (PCMs) with domestic cold storage systems (**He and Setterwall, 2002; Qureshi et al., 2011**).

2.3 Effects of Temperature Fluctuation on Products in the Refrigerator

In this section previous works are reviewed by the intention of solidifying or giving scientific background for the problem of the study.

Gin et al. (2015) investigate the effect of phase change material panels placed against the internal walls of a freezer during repeated power loss every 24 h over 2 weeks was conducted. Comparisons of the freezer air temperature and product temperature in a freezer containing PCM panels showed lower temperature fluctuations than in a freezer without PCM. Product analysis was also conducted by measuring the drip loss in 11 cm cubes of bovine muscle (*Musculus semimembranosus*) and the ice crystal size in 1 L blocks of vanilla ice cream. Drip loss was measured by centrifuging the samples and weighing the drip collected. Analysis of ice crystal diameters was conducted by microscopy and an image analysis tool. Drip loss in meat was found to be lower and ice crystal sizes in the ice cream were found to be smaller in the freezer with PCM compared to without PCM.

Duun et al. (2008) have examined the drip loss from pork roast stored under different conditions. The result shows that there was greater drip loss from meat that had been subjected to temperature fluctuation than meat stored at constant temperatures. Constant temperature storage at 2°C was conducted over 90 days, whereas temperature abuse was also conducted over the same period and conditions apart from temperature abuse for four continuous days, where the maximum temperature reached was 3.4°C.

Yanniotis et al. (2008) studied the drip loss of bovine muscle (*Musculus semimembranosus*) stored in a freezer operating with defrost cycles every 8 h over 31 days. With defrosting, the drip loss in the meat samples (16% after 10 days rising to 25% after 31 days) was higher when compared to meat stored in a freezer without defrosting (8–10% after 10 and 31 days).

Flores and Goff (1999) found that the size of ice crystals was larger in samples stored under fluctuating conditions (15 ± 5) °C compared to storage at a constant 16°C.

2.4 PCM Integrated Cold Storages

The existing literature of PCM integrated cold storages experimental and numerical investigations are viewed with the intention of further improvement of the present study.

Oró *et al.* (2012) investigated improving the thermal performance of commercial freezers using phase change materials under door openings and electrical power failure. PCM was (Climsel-18) with a melting temperature of $-18\text{ }^{\circ}\text{C}$, w. During 3 h of electrical power failure, the use of PCM maintained the freezer temperature $4\text{--}6\text{ }^{\circ}\text{C}$ lower and that of the frozen products remains at acceptable levels for a much longer time. The results of the door opening test demonstrated the importance of the selection of the PCM with regards to its melting temperature. Operating with a storage temperature of $-19\text{ }^{\circ}\text{C}$ the benefit of the utilization of PCM was evident, while when the storage temperature was moved to $-22\text{ }^{\circ}\text{C}$, PCM did not show significant improvement due to the difference between the storage temperature and the phase change temperature of the PCM ($-18\text{ }^{\circ}\text{C}$). Therefore, it is important to select a PCM that has a phase change temperature near the storage temperature of the freezer regards of door openings.

Leducq *et al.* (2015) studied a PCM packaging and compares its performance to a polystyrene packaging configuration. The impact on temperature fluctuations and ice crystal size distribution was characterized experimentally during long term storage and temperature abuse. The results show that the use of an additional PCM packaging has a significant impact on the final quality of the product by keeping ice cream temperature stable and close to the phase change temperature of the PCM material. These results were compared with the insulation material results and discussed, showing that a material with a buffering heat capacity can be more efficient to reduce temperature fluctuations than a low conductivity material and that the same results can be usually obtained with a much thinner layer of material.

Bista *et al.* (2018) presented a review of various research investigations on the application of PCM in refrigeration systems and the result shows. Application of PCMs in vapor compression refrigeration systems has illustrated significant effects on the performance of the system, compressor on-off cycle, and electricity consumption reduction. The advantages and drawbacks of using PCM in the evaporator, condenser, compartment section, and compressor are evaluated. PCMs placed at the evaporator section gives slower fluctuation of compartment temperature and more stable conditions against thermal load variations. However, the incorporation of PCM at the evaporator section increases the compressor running time initially and raises the condensation temperature. Considering these problems, some investigations were performed to incorporate

PCM at the condenser section. The results also showed a higher COP, lower energy consumption lowers condensation temperature, and lower condensation pressure.

Oró *et al.* (2013) investigated on design and test of a phase change material package for commercial ice cream containers. A mathematical model was validated with experimental data and following a parametrical study of the design process was done. The results demonstrated that the use of the PCM package is beneficial for ice cream quality when it is placed out of the freezer.

Ismail *et al.* (2009) performed an experimental and numerical study on the solidification of different phase change materials encapsulated in spherical and cylindrical shells of different materials and diameters subject to constant surface temperature. The results show the increase of the glycol content in the mixture of water and glycol solution led to reducing the phase change temperature, reaching -15°C when the Glycol content is 50%. It is also found that the increase of Glycol content in the PCM led to increasing the time for complete solidification for all working temperatures. The effect of increasing the diameter of the spherical shell is found to increase the time for complete solidification.

2.5 Literature Review on PCM Integrated at Compartment Area.

In this section literature about PCM integration on the compartment, the area is viewed for understanding and further improvement.

Marques *et al.* (2013) investigated on PCM integrated refrigerator. In investigation numerical simulations using the computational fluid dynamics (CFD) software, ANSYS Fluent was undertaken to characterize the airflow and temperature distribution in a natural convection thermal energy storage refrigerator. The model compared the household refrigerator temperature stability with different phase change materials incorporated into the storage compartment. Scenarios investigated included the PCM orientation (vertical or horizontal), PCM temperature (use of water or eutectics), and compartment designs (conventional or drawer type appliance). The results suggested that a horizontal PCM configuration produces lower compartment temperatures than a vertical configuration. The temperature distribution with a horizontal PCM was tested experimentally and the results were in agreement with the CFD predictions. Both the simulation

and the experimental results suggest that a eutectic with a phase change temperature below 0 °C must be employed to maintain the compartment temperature within acceptable limits.

Barzin *et al.* (2015) presents an analysis of a price-based control system in conjunction with energy storage using phase change materials for two applications: space heating in buildings and domestic freezers. The freezer used a eutectic solution of ammonium chloride (melting point of −15 °C). In the building application, DuPont wallboards were used to provide thermal storage. Experimental results showed that using thermal storage material in conjunction with the proposed price-based control method can improve the performance of these systems and lead to a successful peak load shifting. Depending on electricity price trends, cost savings using the proposed strategy can vary. Savings of up to 16.5% and 62.64% per day were achieved for the freezer and building applications respectively, based on New Zealand electricity rates.

2.6 PCM Integrated at Evaporator and Condenser.

Azzouz *et al.* (2009) investigated the performance of a household refrigerator with latent heat storage and the result shows that, the PCM located on the backside of the evaporator to improve its efficiency and to provide storage capacity allowing several hours of refrigeration without power supply. The system has been tested with water and with a eutectic mixture (freezing point −3 °C) and for a range of operating conditions (PCM thickness, ambient temperature, thermal load). The analysis of the results shows a significant improvement in the performance compared to a conventional system.

Cheng and Yuan (2013) investigated on the dynamic model of a novel household refrigerator with SSPCM (shape-stabilized phase change material) heat storage condenser following the experiment. By simulation, the coefficient of performance is increased about 19% by a continuous heat transfer of condenser due to the latent heat storage of SSPCM, however, the energy saving is 12% and offset about 7% by the heat leakage increase because of the SSPCM inside the insulation layer. The effects of ambient temperature, freezer temperature, and phase change the temperature on the energy-saving are analyzed to provide a theoretical basis for the optimization design of the refrigerator with SSPCM. It can be concluded that the ambient temperature increasing and the freezer temperature decreasing can increase the energy-saving effect, the second phase change temperature region with the peak temperature of about 49 °C of SSPCM results in the minimum energy consumption of the novel refrigerator.

Azzouz et al. (2008) studied the effect of adding a phase change material slab on the outside face of a refrigerator evaporator. A dynamic model of the vapour compression cycle including the presence of the phase change material and its experimental validation is presented. The simulation results of the system with PCM show that the addition of thermal inertia globally enhances heat transfer from the evaporator and allows a higher evaporating temperature, which increases the energy efficiency of the system. The energy stored in the PCM is yielded to the refrigerator cell during the off-cycle and allows for several hours of continuous operation without power supply.

Azzouz et al. (2009) experimentally investigated the performance of a household refrigerator using a phase change material. The PCM is located on the backside of the evaporator to improve its efficiency and to provide storage capacity allowing several hours of refrigeration without power supply. The system has been tested with water and with a eutectic mixture (freezing point -3°C) and for a range of operating conditions (PCM thickness, ambient temperature, thermal load). The results indicate that the response of the refrigerator to the addition of PCM and its efficiency is strongly dependent on the thermal load. The integration of latent heat storage allows 5–9 h of continuous operation without electrical supply (to be compared to 1–3 h without PCM) and a 10–30% increase of the coefficient of performance, depending on the thermal load.

Azzouz et al. (2007) studied the effect of adding a phase change material slab on the outside face of a refrigerator evaporator. A dynamic model of the vapor compression cycle including the presence of the phase change material and its experimental validation is presented. The simulation results of the system with PCM show that the addition of thermal inertia globally enhances heat transfer from the evaporator and allows a higher evaporating temperature, which increases the energy efficiency of the system. The energy stored in the PCM is yielded to the refrigerator cell during the off-cycle and allows for several hours of continuous operation without power supply.

Visek et al. (2014) investigated experimental results of the SDE prototype appliance with visible RC roll-bond evaporator in direct contact with phase change material to increase RC evaporation temperature. Melting PCM is continuously absorbing cabinet heat thus we could keep natural convection as a preferred mode of heat transfer in RC. Controls including a pump out phase and particular components were implemented to deal with refrigerant charge migration between refrigerator and freezer evaporators. RC evaporation temperature was increased by a maximum of

8.4 K. Energy consumption during RC operation was reduced by 19.9% and overall energy consumption dropped by 5.6% after the condenser fan was installed. At least 8.1% of energy-saving was predicted if condenser with a 30% higher UA value was implemented instead of the condenser fan

Khan and Afrozn (2013). Investigated on an experimental investigation of the performance improvement of a household refrigerator using PCM. The PCM is located behind the five sides of the evaporator cabinet in which the evaporator coil is immersed. The refrigeration system has been tested with two different PCMs (Water and Eutectic solution of melting point 0 and - 5°C, respectively) at different thermal loads. Experimental results show that the Coefficient of Performance of the refrigeration cycle with PCM is considerably higher than that of without PCM. Depending on the types of PCM and thermal load, around 20-27% COP improvement of the refrigeration cycle has been observed with PCM concerning without PCM. With the increase in the quantity of PCM (0.003-0.00425 m³) COP increases by about 6%. Between two different PCMs, the COP improvement for the Eutectic solution is higher than Water. The experimental results with PCM confirm that depending on the thermal load and the types of PCM average compressor running time per cycle is reduced significantly and it is found about 2-36% as compared to without PCM

Sonnenrein et al. (2015) Evaluated the influence of latent heat storage elements on the condenser temperature of a commercial household refrigerator. To determine the power consumption and the temperature distribution, a standard wire-and-tube condenser is equipped with different heat storage elements (containing water, paraffin, or copolymer compound). The results indicate that particularly the application of phase change materials lowers the condenser temperature, which leads to a significantly reduced power consumption

Yusufoglu et al. (2015) investigated the performance of household refrigerators by incorporating phase change materials. In the study, four different PCMs tested in two different refrigerator models. The compressor on/off time was optimized and better energy efficiency was achieved. Increasing the condenser surface area by 20% enhanced the PCM effect. The use of only 0.95 kg of PCM has resulted in a 9.4% energy saving. Economic analyses show that using PCMs in household refrigerators is a cost-effective method that saves energy and reduces harmful emissions

Zarajabad and Ahmadi (2018) investigated a household refrigerator with CTES system at the evaporator. Results show that adding three fins to PCM container, as the best choice, can keep the refrigerator's temperature in the standard temperature range for 68 min. In consequence, the operation of the refrigerator equipped with a PCM container with three fins is examined, experimentally. The experimental results of the charging and discharging process show good consistency with the numerical results. Furthermore, it is obtained that using the finned CTES system, would reduce the compressor's energy consumption by about 17.4% per day. Due to lower electrical demand for refrigerators, it would diminish the fossil fuel consumption and carbon dioxide emission for 12 L and 28 kg, respectively.

Maiorino et al. (2019) made an experiment on the thermal performances of a refrigerator incorporating a phase change material. Experimental results show that the introduction of the PCM has led to a noticeable reduction of the temperature gradient within the cabinet and the fluctuations of product temperature ensuring better conservation conditions, and extending the OFF time of the compressor. The tests also show that with PCM, hysteresis has a lower impact on the fluctuation of the product temperature, ON-time ratio, and energy consumption of the refrigerator than without it. On the contrary, the product temperature distribution is more affected by the hysteresis with PCM than without, leading to a better uniformity with higher hysteresis.

Cheng et al. (2011) investigated, a kind of shape-stabilized phase change material was adopted for constructing heat storage condensers. And a novel household refrigerator equipped with the heat storage condensers was set up based on an ordinary double-door three-star compartment refrigerator. The experimental investigation on the characteristics of the novel refrigerator and an ordinary refrigerator was carried out. For the novel refrigerator, part of the condensation heat was stored in the shape-stabilized PCM during the on-time and discharged to the environment while the compressor was off. Therefore, the heat dissipation of the novel refrigerator was continuous during a complete cycle (including a successive on-time and off-time period), different from the intermittent heat dissipation of the ordinary setup. Thus, the overall heat-transfer performances of the condensers could be significantly improved, which resulted in a lower condensation temperature, a higher evaporation temperature, and a much larger subcooling degree at the condenser outlet. Compared to the ordinary refrigerator, the total cycle time and the ratio of on-time to the total cycle time of the novel refrigerator were much smaller, which led to more frequent

starts of the compressor but lower energy consumption. Experiments demonstrated that the novel refrigerator could increase energy efficiency by about 12% with only a little increase in the cost.

2.7 PCM

CTES is a system of adding cold thermal energy to use it whenever needed. Cold thermal energy can be stored in three main ways as shown in Figure 2.1.

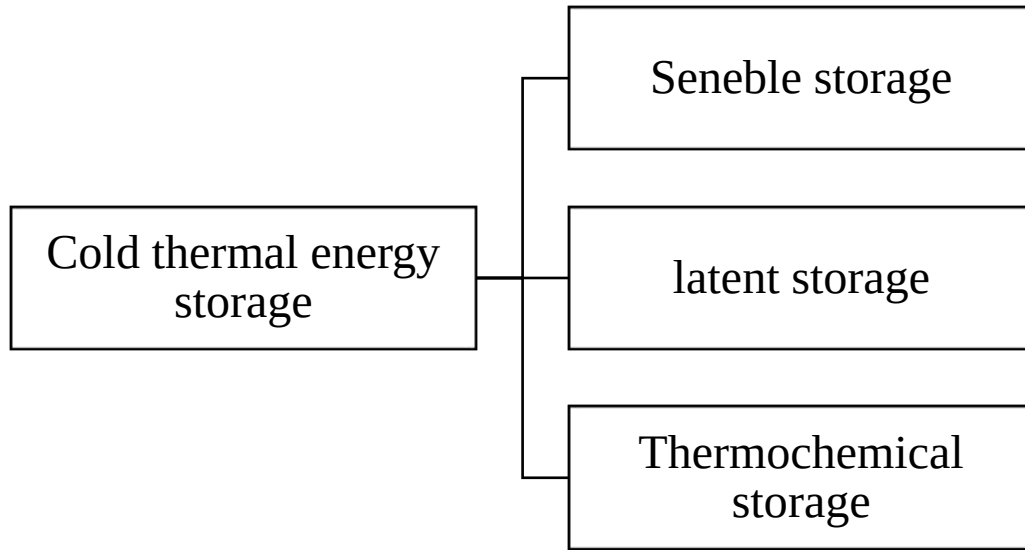


Figure 2.1: Classification of energy storage type

In cold thermal energy storage, sensible and latent storage methods are widely used. The energy storage capacity of the sensible heat storage system is based on specific heat capacity and the temperature difference. There is no phase change during energy storage and retrieval. The equation below gives the amount of energy stored in a sensible storage system. **(He and setter wall, 2002)**

$$Q = \int_{T_i}^{T_f} m C_p \Delta T \quad (3.81)$$

The latent heat storage which stores energy by phase change material (PCM) gives more storing ability because of its high energy storage density 5–14 times higher than sensible storage (Sharma *et al.*, 2009). In this technique, a phase change occurs during energy storage and retrieval. The amount of energy stored is based on the latent heat of fusion of the material. PCM is also used to increase the energy storage capacity of a system (Farid *et al.*, 2004). Eqn. 3.82 gives the amount of energy stored in a latent heat storage system.

$$Q = \int_{T_i}^{T_m} m C_p \Delta T + m a_m H + Q = \int_{T_m}^{T_f} m C_p \Delta T \quad (3.82)$$

Thermochemical energy storage depends on the energy released and absorbed during the reversible chemical process. The amount of energy stored during this chemical process depends on the number of chemicals reacting.

In this thesis, latent heat storage (PCM) is chosen as cold thermal energy storage because of its high energy density. There are some desirable thermo-physical, kinetic and chemical properties (Abhat, 1983; Cabeza *et al.*, 2001; Castell *et al.*, 2010; Farid and Khalaf, 1994) Classification of Phase Change Material (PCM)

2.7.1 Classification of PCM

Figure 2.2 classification of phase change material (PCM) used for cold thermal energy storage.

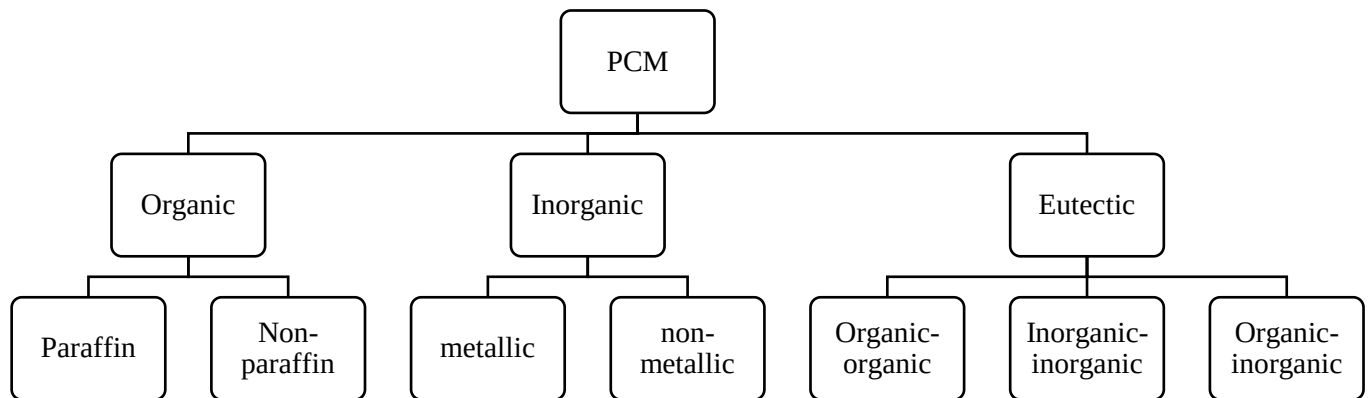


Figure 2.2: Classification of PCM

- **Organic PCM:** Organic PCMs are carbon-based compounds. These are generally classified as paraffin and non-paraffin. The latent heat of fusion of the PCM increases with an increase in molecular weight or number of carbon atoms. The advantages of organic PCMs are its chemical stability and high latent heat of fusion, but the main disadvantage is its low thermal conductivity.
- **Inorganic PCM:** Inorganic PCMs are generally metallic and hydrated salts. The main advantages of inorganic PCMs are its low cost and good thermal conductivity, but it is corrosive with the construction materials.
- **Eutectic PCM:** Eutectic PCM is a mixture of two or more PCMs to achieve the desired melting point. The melting point of the PCM is one of the major criteria for selecting PCM for cold storage applications (Sari, 2005). The eutectic PCM can be further divided into eutectic PCMs for the low-temperature cooling system and eutectic PCMs for high-

temperature cooling systems. Also, it can be categorized as organic eutectic PCMs and inorganic eutectic PCMs (Arkar and Medved, 2007).

2.7.2 PCM Selection Criteria

Many researchers have presented useful classifications of materials that may be used for thermal energy storage. According to the literature, the PCM used in the design of TES systems should have desirable thermophysical, kinetic, and chemical properties. It can be concluded from the information compiled over different studies that the main characteristics required for a good PCM are listed in Table 2.1.

Table 2.1: PCM selecting criteria (Mastani *et al.*, 2015)

Thermophysical properties.	Melting temperature in the desired operating temperature range
	High latent heat of fusion per unit volume
	High specific heat to provide additional significant sensible heat storage
	Small volume change on phase transformation and small vapor pressure at operating temperature
	Reproducible phase change
Chemical properties	Complete reversible freeze/melt cycle
	No degradation after a large number of freeze/melt cycles
	No corrosiveness to the construction/encapsulation materials
	Non-toxic, non-flammable, and non-explosive
Economics	Abundant
	Available
	Cost-effective
	Easy recycling and treatment
	Good environmental performance based on Live Cycle

Salt hydrants are the most suitable PCM types because of their high latent energy high specific heat capacity and they're abundant at low cost.

2.7.3 Effects of PCM Thickness

The PCM thickness greatly affects the performance of the refrigeration system as it is reported that an increase in the amount of PCM (approximately 40%) resulted in a 6% rise in the COP (Khan and Afroz, 2015). By increasing PCM thickness, its impact is on decreasing the on-off time ratio increases due to an extension of the compressor's off time (Cerri *et al.*, 2003). Application of thicker PCM incurs higher costs and initially requires higher compressor work for solidifying the PCM; hence, the thickness of PCM should be considered based on the load. Once the PCM has been selected, it is essential to determine the amount of PCM to be used. A minimum volume of PCM is required, which can be calculated accordingly (Onyejekwe, 1989). The PCM's stored energy (E) in a compartment is given by:

$$E = \rho v \lambda \quad (3.83)$$

In this equation, the sensible heat variations are neglected. The constants λ and ρ are the PCM's enthalpy of fusion and density, respectively. The unavoidable heat gain from the surrounding condition by the compartment is:

$$Q = (UA)_{Cold}(T_{amb} - T_{cold}) \quad (3.84)$$

Where UA is the overall heat coefficient thermal conductance. The stored energy in PCM is equal to the rate of passing energy from the compartment's walls during OFF time (t_{off}) periods of the compressor. Therefore, the minimum volume for PCM is given by (Onyejekwe, 1998):

$$V_{min} = \frac{t_{off}[(UA)_{Cold}(T_{amb} - T_{cold})]}{\rho v \lambda} \quad (3.85)$$

The amount of PCM is usually more than the computed value from Eq above because heat transfer via the compartment's walls during the

compressor off time is compensated by PCM. However, the system has to work longer to charge higher amounts of PCM. Besides, the ON time ratio is reduced as the compressor OFF duration is much longer. Besides, since all the thickness does not have the chance to undergo a phase change process, the PCM cannot be thicker than a certain amount. In such cases, partial melting/freezing of PCM occurs, which decreases its effectiveness. Thicker PCM not only needs longer compressor work for cold storage but also burdens higher costs and hence, PCM thickness is usually selected according to the load (Marques *et al.*, 2013).

Marques *et al.* 2013 used a refrigerator with a heat load of 23.8W to determine the PCM freezing and melting times for different thicknesses. This assumption corresponded to an ambient

temperature of 25 °C. The compressor was assumed to be off during the melting process and running continuously with a cooling capacity of 246W during the freezing process that corresponded to a −10 °C of evaporating temperature. In Figure 2.3, it is shown that the freezing time was observed as 13 mins and 34 min for 2mm and 5mm slabs respectively. As such, it was claimed that the melting and freezing times increased as the PCM got thicker i.e. a thicker PCM can be used for more time. Likewise,

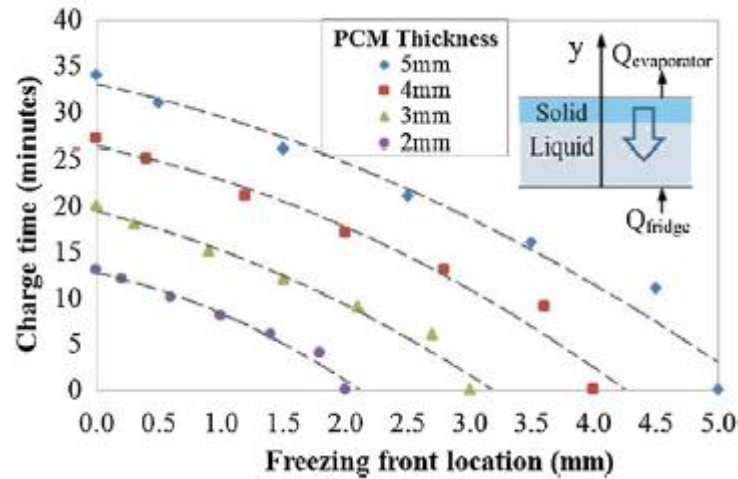


Figure 2.3: PCM freezing time with respect to the thickness of PCM (Marques *et al.* 2013)

Azzouz *et al*, 2013 evaluated the significant effect of PCM thickness on the running period and the autonomy of the refrigeration system. The autonomy of the refrigeration system for various scenarios is demonstrated in Figure 2.4. The figure indicates that 28.5–32.6% of running time was saved and autonomy of 4.7–8.5 h was achieved. The running time saving is defined as the ratio of the difference between the percentages of running time without/with PCM. As the PCM thickness increases, the capability of storing energy increases significantly and relatively more heat is absorbed. Henceforth, a greater amount of heat is released when the PCM solidifies increasing the autonomy of the system.

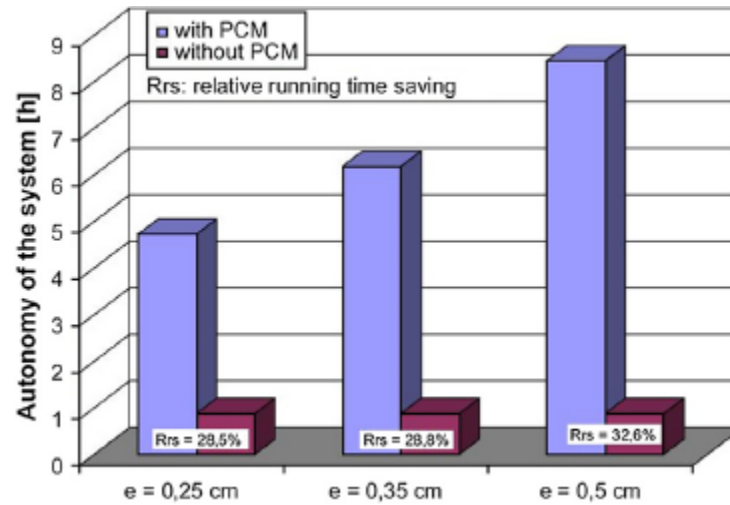


Figure 2.4: Autonomy and running time reduction ($T_{amb}=20\text{ }^{\circ}\text{C}$, zero door openings, $T_{pcm}=-3^{\circ}\text{C}$) (Azzouz *et al.*, 2009)

2.7.4 Effects of Phase Change Temperature of PCM

The principal purpose of refrigerators is to preserve food and maintain a refrigerated space within a required low temperature. Therefore, when a PCM is used, the refrigeration system performance and the quality of food are directly proportional to the PCT of that PCM. PCT plays an important role since the melting point of the selected PCM must be in the range of the thermostat temperature. Due to its lower energy consumption, PCM with high PCT enhances the COP of the system. Nevertheless, it harms the food, as it increases the temperature inside the compartment which results in deteriorating food quality (Cerri *et al.*, 2003). On the other hand, a very low PCT maintains a very low temperature inside the storage cabinet resulting in a better quality of food. To prevent freezing of food, it should be ensured that the cabinet's temperature is not below zero (Khan *et al.*, 2016) Therefore, it is necessary to select an allowable PCT that maintains these two upper and lower extremes. PCT varies accordingly with the selection of PCM (Mastani *et al.*, 2015). Figure 2.5 is an observation of the experiment conducted by (Azzouz *et al.*, 2005) which shows the effect of PCT on the COP of the refrigerator.

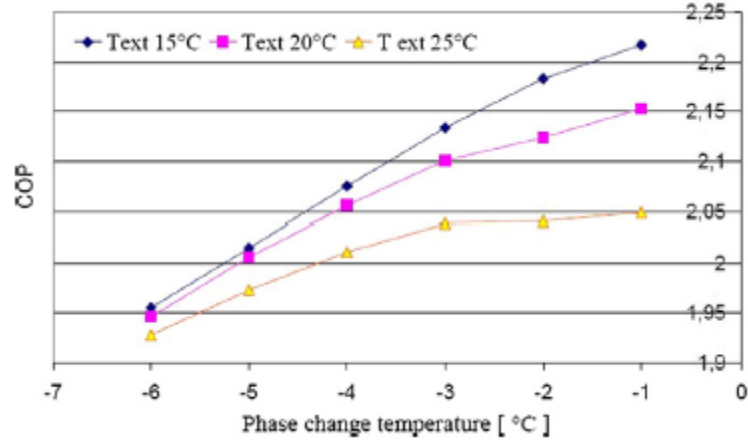


Figure 2.5: Change in COP with respect to change in phase change temperature

(Azzouz *et al.*, 2005)

According to this figure, as PCT increases, a significant increase of COP can be observed. With the increase in COP, lower power consumption was deduced by the authors because the pressure compression ratio reduced. However, it should be made sure that the chosen PCT is not too high. A higher solidification temperature will result in excessive air temperature inside the compartment during melting as shown in Figure 2.6. This in turn lowers the COP. Therefore, a compromise must be reached between the two extremes of PCT.

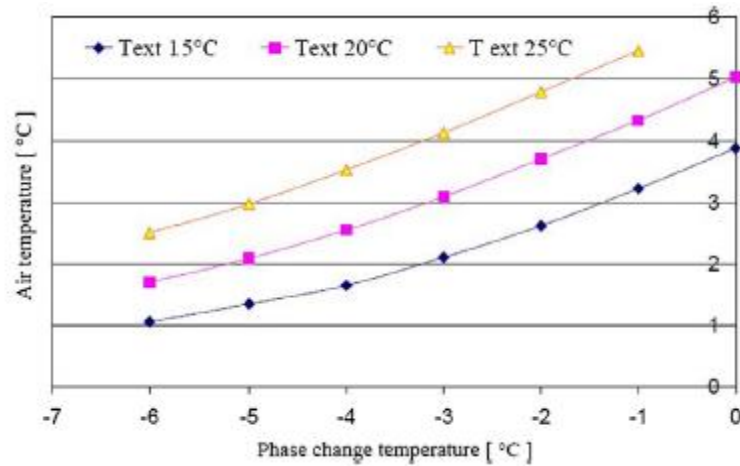


Figure 2.6: Behavior of the average air temperature for different phase change temperatures

(Azzouz *et al.*, 2005)

2.7.5 Effects of PCM on Evaporation Temperature and Pressure

Improving the efficiency of the evaporator is the most emphasized area in the performance enhancement of a refrigeration system. Depending upon the thermal load and type of PCM used,

nearly 2–5 °C higher evaporating temperature was achieved. as discussed in Figure 2.7 and Figure 2.8. The increase in evaporating temperature raises the COP of the system and decreases power consumption (Khan *et al.*, 2013).

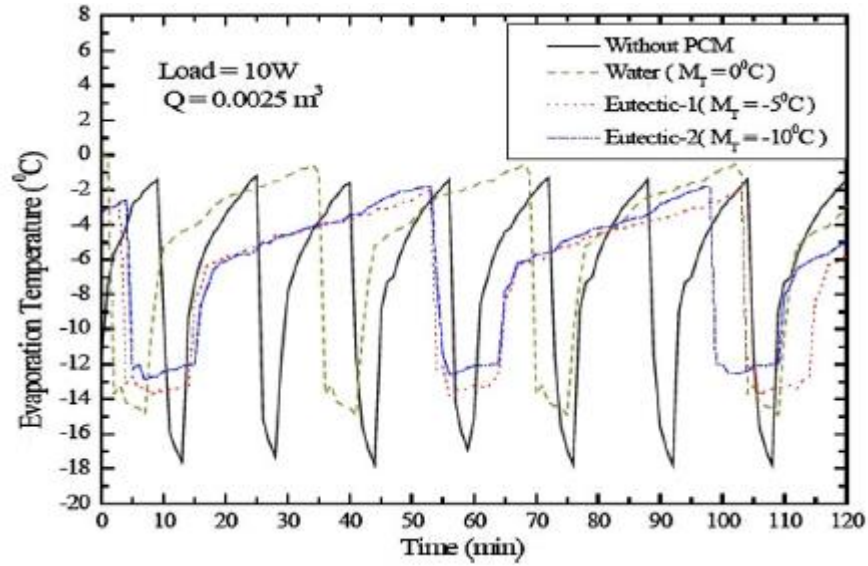


Figure 2.7: Effect of PCM on the evaporating temperature of the system with load 10W (Khan *et al.*, 2013).

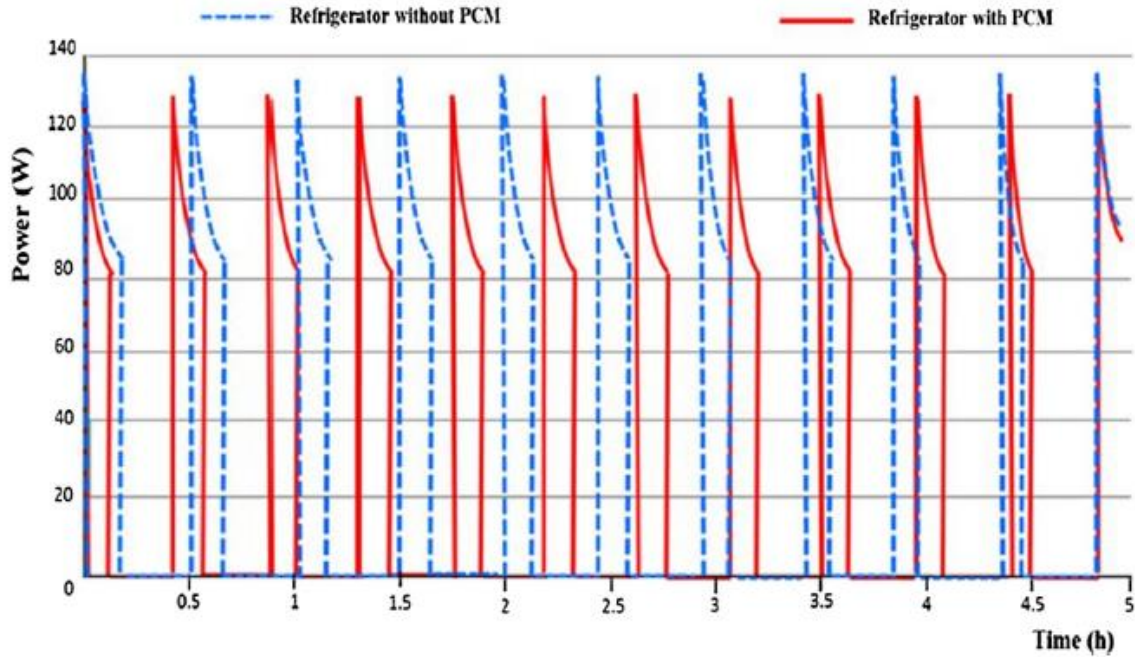


Figure 2.8: Effect of PCM on the power consumption of the refrigerator (Khan *et al.*, 2013).

2.7.6 Effect of PCM on Thermal Loads

Thermal loads have an inverse relationship concerning the performance of refrigeration systems i.e. the increase in thermal load decreases the performance of a refrigerator. While the use of PCM in a refrigeration system can improve system performance, it is essential to analyze the behavior of PCM at different loading conditions (Azzouz *et al.*, 2005). Cerri *et al.* 2003 stipulated that the COP decreases by raising the thermal load, despite the presence of a PCM. This is because escalating the thermal load causes the PCM to melt partially as there is not sufficient time for phase change (Azzouz *et al.*, 2008). The evolution of the COP is displayed in Figure 3.19 for different thermal loads. Generally, the COP values of configurations with PCM are higher than the one without PCM.

Nonetheless, it was observed that the COP decreased with higher loadings too. In Figure 2.9, it can be seen that the increase in COP ranges from 5 to 15 %. This is also due to others including factors such as the number of door openings, external temperature, and PCT. Khan and Afroz , 2015 claimed that a significant amount of COP decreases as the thermal load is increased due to the attenuation of the sub-cooling effect of the condenser which occurs at the higher operating pressure and temperature of condenser. An experimental investigation was set up for a 0.00425m^3 to compare the effects of PCM at various loads for a eutectic solution and water. The Eutectic solution, having a melting point of -5°C stored more enthalpy of fusion than water during the off mode of the compressor. This amount of heat, ultimately increased the evaporation temperature and pressure of the evaporator when transferred to the refrigerant by faster conduction method during the ON mode. Furthermore, a higher condensation temperature and pressure means more running time of the compressor, which results in higher on/off cycling.

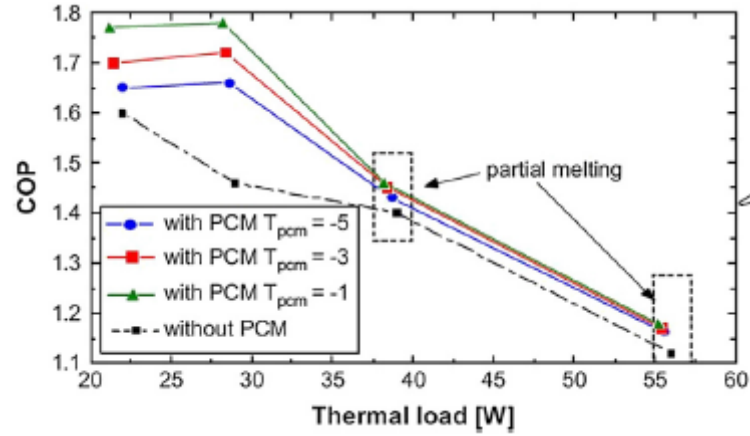


Figure 2.9: Effect of thermal loads on COP of the refrigeration system (Cerri *et al.*, 2003)

2.7.7 Effect of Ambient Temperature on PCM Performance

The ambient temperature has significant effects on the performance of PCM in the refrigeration system (Mastani *et al.*, 2013). System COP reduces in higher ambient temperatures because of the higher condensation pressure and temperature and higher cabinet air temperature. PCM's charging/ discharging period is affected by the high thermal load, which shortens the melting time and delays the freezing time. This is because the compressor should overcome both charging PCM and the thermal load (Marques *et al.*, 2013). The performance of PCM decreases in very low ambient temperatures.

The compartmental temperature drops faster if working in a low thermal load. As a consequence, there is not sufficient time for the PCM to completely solidify, before the compressor stops (Gerri *et al.*, 2003) (Azzoz *et al.*, 2009) evaluated the influences of ambient temperature on the charge and discharge rate for a 5mm PCM slab. As it is illustrated in Figure 2.10, when the ambient temperature rose from 20 °C to 30 °C the PCM melting time reduced from 320 min to 208 min which corresponded to the mitigation of 65% in refrigerator autonomy. Figure 2.11 shows that the freezing time increased slightly concerning an increase in ambient temperature, along with an additional 7 min being needed to freeze the PCM at 30 °C as compared to 20 °C.

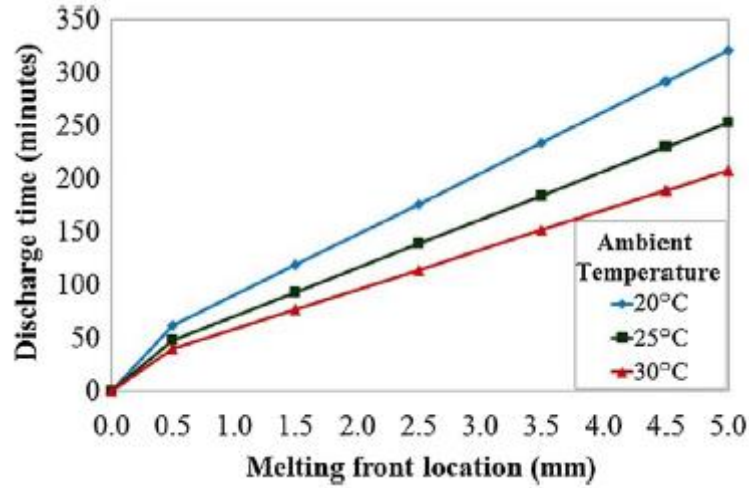


Figure 2.10: Melting time of a 5mm PCM with respect to the different ambient temperatures
(Azzoz *et al.*, 2009)

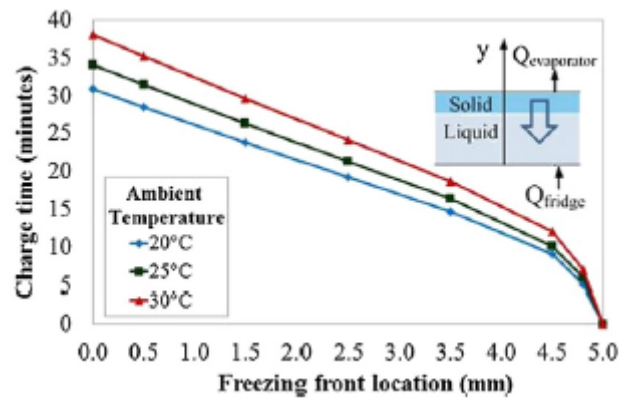


Figure 2.11: Freezing time of a 5mm PCM with respect to the various ambient temperatures
(Marques *et al.*, 2013)

Based on the above literature and selection criteria PCM to be used on the condenser and the evaporator are selected. The selected PCMs are 5% NaCl₂, 16% KCl, CaCl₂.6H₂O, and Paraffin wax. Table 2.2 shows the type and property of the PCM selected for the study.

Table 2.2: Selected PCMs for experimental testing

Property	5% NaCl ₂ Kumano <i>et al.</i> , (2009)	10% NaCl ₂ Kumano <i>et al.</i> , (2009)	CaCl ₂ .6H ₂ O Cabeza <i>et al.</i> , (2011) and Pielichowska <i>et al.</i> , (2014)	Paraffin wax Lingayat and supple (2.13)
Melting temperature (°C)	-3.5	-8	28	58/60
Freezing temperature (°C)	-3.5	-8	28	58/60
Latent heat of fusion ($kJ.kg^{-1}$)	231	179	174	189
Solid density ($kg.m^{-3}$)	980/1000	990	1.8	918
Base material	NaCl ₂	NaCl ₂	CaCl ₂	
C _p ($kJ.kg^{-1}°C$)	1.88			2.384

2.8 Literature Summary

The above literature shows that the use of PCM material increases COP of refrigerator enhance, cooling of a refrigerator after a power cut, and reduce electric conception of the refrigerator at peak load time by storing cold thermal energy. The important factors for performance improvement of PCM integrated cold storage are.

- 1) Position of PCM in the refrigerator: PCMs placed at the evaporator section gives slower fluctuation of compartment temperature and more stable conditions against thermal load variations. However, the incorporation of PCM at the evaporator section increases the compressor running time initially and raises the condensation temperature. Considering these problems, some investigations were performed to incorporate PCM at the condenser section. The results showed a higher COP, lower energy consumption lowers condensation temperature, and lower condensation pressure. Nevertheless, compressor on-off problems are more frequent, and more refrigerant displacement losses were observed. PCM inside the food

storage compartment gives Stable temperatures inside the compartment were observed. However, this system experienced a lower COP as compared to the other two positions.

- 2) The freezing point of PCM: literature research results suggest that a eutectic with a phase change temperature below 0 °C must be employed to maintain the compartment temperature within acceptable limits. it is also important to select a PCM that has a phase change temperature near the storage temperature of the freezer regards of door openings.
- 3) The quality of PCM: PCM with high latent heat capacity is more perforable. PCM high specific heat capacity can store sensible heat as well as latent heat.

2.9 Research Gap

As there is a different advantage and disadvantage of PCM placement their combined effect should be studied. Experimental study of effects of combined PCM position on the compartment, condenser, and the evaporator is not well studied. The effect of condenser PCM thickness is not well investigated as evaporator PCM. PCM that made from a salt hydrant is a high ability of cold energy storage so that more investigations should be made on salt hydrant on the evaporator. In this study thickness-based investigation of the condenser PCM, salt hydrant PCM, and the combination of the condenser and evaporator PCMs are done.

CHAPTER THREE

SELECTION AND DESIGN OF THE REFRIGERATOR SYSTEM MAIN COMPONENTS

3.1 Introduction

In this section selection of components and materials of vapour compression refrigeration system are done based on the component selection criteria found in the literature. After selecting the components of the refrigerator sizing of the refrigerator capacity and determining the important parameters of the selected refrigerator components are made. The theoretical analysis of the system COP is also done.

3.2 Selection of Main Components of the Vapor Compression Refrigeration System

The refrigerator type selected for the study is a novel household refrigerator with a single door that works in the principle of vapor compression refrigeration cycle. The components of the refrigerator are selected after considering deferent types of the component based on the selection criteria.

3.2.1 Compressor Selection

A compressor is the most important part of the refrigerator. Its function is to continuously draw the refrigerant vapor from the evaporator. So that the low pressure and low temperature can be maintained. The compressor has to raise the pressure and the temperature of the refrigerant so that it can condense by rejecting heat to the cooling medium.

The selection of a compressor is made based on system cooling characteristics and where it will be installed (Kharagpur, 2008).

- I. Minimum Evaporator Temperature
- II. Cooling Capacity or Thermal Load
- III. Refrigerant Type
- IV. Ambient Temperature
- V. Electrical Power Available
- VI. Compressor and Drive Cooling

Compressors are classified depend on their working principles and their arrangement. Figure 3.1 present the classification of compressor.

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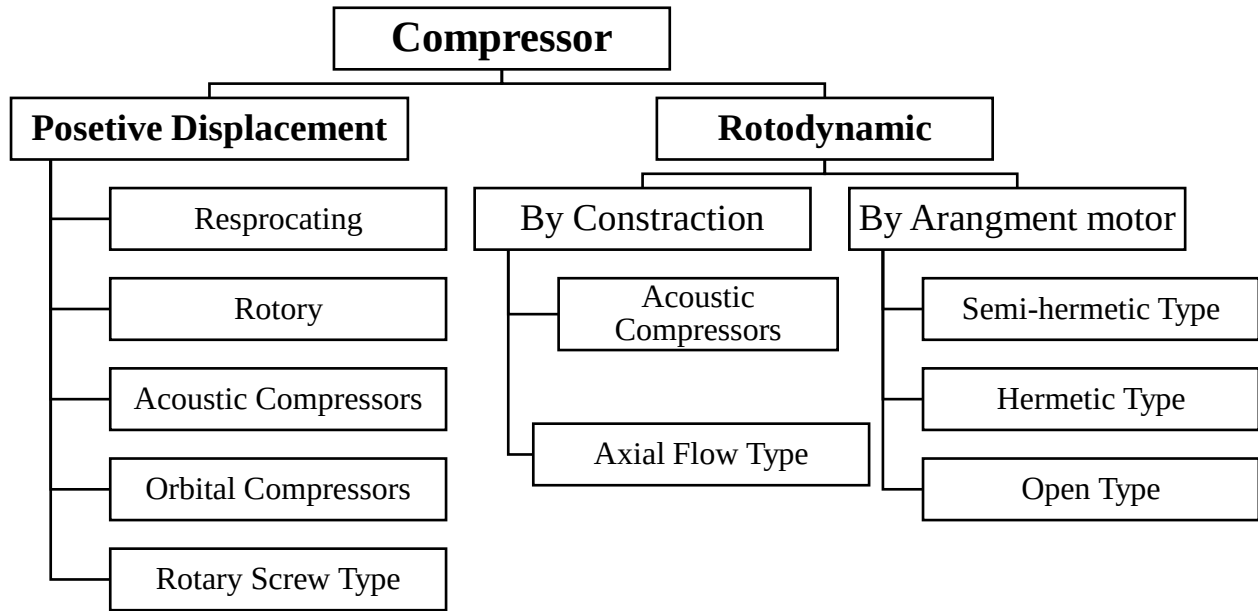


Figure 3.1: Compressor classification (Chaudhary and Gupta, 2000).

In positive displacement compressors, compression is achieved by trapping a refrigerant vapor into an enclosed space and then reducing its volume. As volume is reduced then it gets pressurized.

In roto-dynamic compressors, the pressure rise of the refrigerant is achieved by imparting kinetic energy to a steady flowing stream of refrigerant by rotating mechanical element and then converting it into pressure as the refrigerant flows through a diverging passage.

In open-type compressors, the rotating shaft of the compressor extends through a seal in the crankcase for an external drive. The external drive may be an electrical motor or an engine (e.g. diesel engine). The compressor may be belt-driven or gear driven. Open type compressors are normally used in medium to a large capacity refrigeration system for all refrigerants and ammonia (due to its incompatibility with hermetic motor materials). In hermetic compressors, the motor and the compressor are enclosed in the same housing to prevent refrigerant leakage. The housing has welded connections for refrigerant inlet and outlet and power input socket. As a result of this, there is virtually no possibility of refrigerant leakage from the compressor (Chaudhary and Gupta, 2000).

For the prototype hermetic centrifugal compressor is selected because the hermetic compressor prevents refrigerant leakage. The housing has welded connections for refrigerant inlet and outlet and power input socket. As a result of this, there is virtually no possibility of refrigerant leakage from the compressor. It has also low levels of noise and energy consumption

3.2.2 Condenser Selection

The condenser is a device used to condense a substance from its gaseous to its liquid state, by cooling it by extracting its latent heat and transferred to the surrounding environment. Condensers can be made according to various designs, and come in many sizes ranging from small (hand-held) to very large (industrial-scale units used in plant processes). The refrigerator uses a condenser to get rid of heat extracted from the interior of the unit to the outside air (Arora, 2009).

There are three types of condensers used in HVAC systems as shown in Figure 3.2:

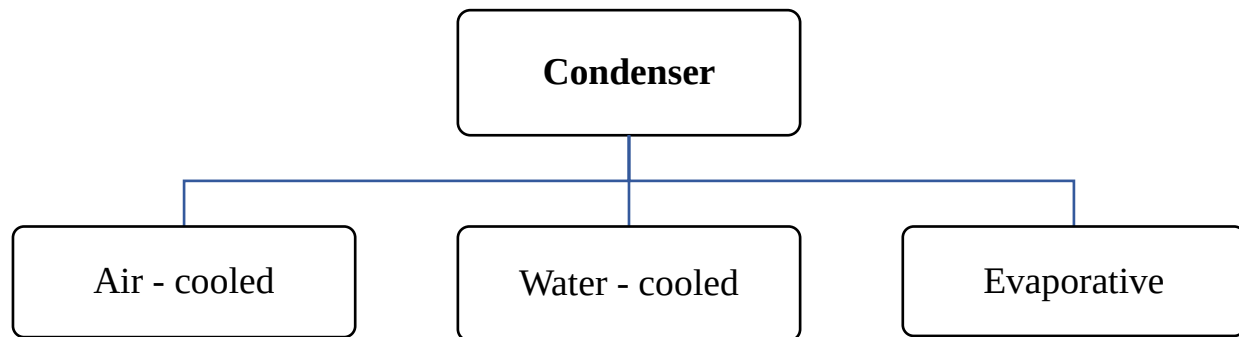


Figure 3.2: Condenser classification

Air-cooled – If the condenser is located on the outside of the unit, the air-cooled condenser can provide the easiest arrangement. These types of condensers reject heat to the outdoors and are simple to install. The most common uses for this condenser are domestic refrigerators, upright freezers, and in residential packaged air conditioning units. A great feature of the air-cooled condenser is that they are very easy to clean. Since dirt can cause serious issues with the performance of the condenser, it is highly recommended that these be kept clear of dirt.

Water-cooled – Although a little pricier to install, these condensers are the more efficient type. Commonly used for swimming pools and condensers piped for city water flow, these condensers require regular service and maintenance. They also require a cooling tower to conserve water. To prevent corrosion and the forming of algae, water-cooled condensers require a constant supply of makeup water along with water treatment. Depending on the application you can choose from a tube in tube, shell, and coil or shell and tube condensers. All are essentially made to produce the same outcome, but each in a different way.

Evaporative – While these remain the least popular choice, they are used when either water supply is inadequate to operate water-cooled condenser or condensation temperature is lower than can be achieved by the air-cooled condenser. Evaporative condensers can be used inside or outside of a building and under typical conditions, operate at a low condensing temperature. Typically, these are used in large commercial air-conditioning units. Although effective, they are not necessarily the most efficient.

In this study, air-cooled wire-and-tube type condenser was selected, because as the removal of heat is done by air it is less power consuming, do not use extra moving parts, no noise, It consists of steel or copper tubing through which the refrigerant flows. The size of the tube usually ranges from 6 mm to 12 mm outside diameter, depending upon the size of condenser. Generally, copper tubes are used because of their excellent heat transferability. The tubes are usually provided with plate-type fins to increase the surface area of heat transfer. The fins are usually made from aluminum because of its lightweight. The flow of the liquefied refrigerant will be assisted by gravity, so the inlet will be at the top of the condenser and an outlet at the bottom. Rising pipes should be avoided in the design, and care is needed in installation to get the level of the pipes. The fins spacing is quite wide to reduce dust clogging (Kharagpar. 2008).

3.2.3 Selection of the Expansion Device of the Refrigeration System

The refrigerator thermos-expansion valve is selected based on the following design condition:

- Type of refrigerant
- Evaporator capacity, Q_e
- Evaporating temperature/pressure, $\frac{T_e}{P_e}$
- Lowest possible condensing temperature/pressure, $\frac{T_c}{P_c}$
- Liquid refrigerant temperature, T_l
- Pressure drop in the liquid line, distributor and evaporator, ΔP

Expansion device are classified based on their working principle as shown in the Figure 3.3

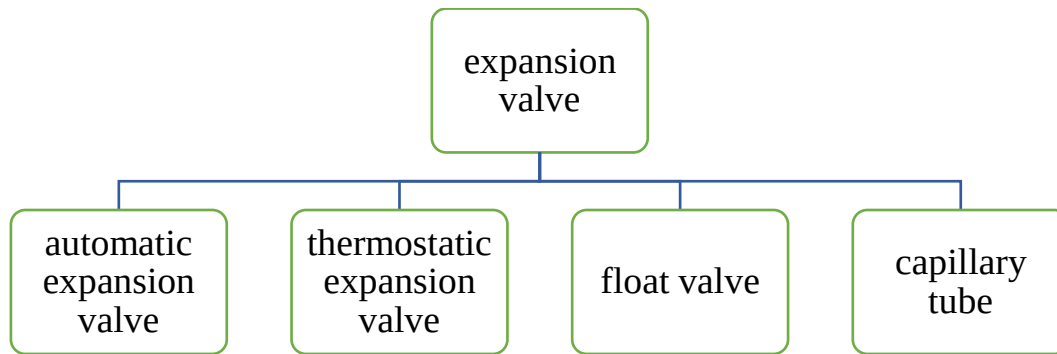


Figure 3.3: Expansion valve classification

1. **Automatic Expansion Valve** regulates the flow of refrigerant from the liquid line to the evaporator by using a pressure-actuated diaphragm. It maintains a constant pressure in the evaporator. The setback is that it is not efficient if the load fluctuates hence this type is not suitable for use in air conditioning as the load fluctuates a lot during its operation.
2. **Thermostatic Expansion Valve** uses a valve mechanism to control the flow of liquid refrigerant into the evaporator coil. The flow is controlled by the pressure in the evaporator. The setback is that the system should install very precisely so that the sensor works accurately any kind of miss fit will reduce its function
3. **Float Valve** is actuated by a float that is immersed in the liquid refrigerant. Both low-side float and high side-float are used to control the flow of liquid refrigerant. The low-side float helps to maintain a constant level of liquid refrigerant in the evaporator. It opens when there is no liquid in the evaporation. and closes when there is liquid in the evaporation.
4. **Capillary Tube:** A capillary tube is a fixed length of small diameter tubing lying between the outlet of the condenser and the inlet of the evaporator. It reduces the pressure of the refrigerant from the high-pressure side to the low-pressure side of the system and controls the flow of refrigerant to the evaporator. Capillary tube expansion devices are widely used in small refrigeration equipment such as household refrigerators, freezers, and air conditioners because of their simplicity, reliability, and low cost. moving part and lasts longer. In order to use this device, the amount of

refrigerant in the system must be properly calibrated at the factory level (Chaudhary and Gupta, 2000).

By considering the above conditions, for this study Coiled type copper capillary tube was selected because of its durability, simplicity, reliability, low cost, and no moving part.

3.2.4 Evaporator Selection

The evaporator consists of a coil of copper, aluminum, or aluminum alloy tubing installed in the space to be refrigerated. Aluminum tubing with copper fins is used in ammonia systems. The liquid refrigerant enters the tubing at reduced pressure and, therefore, with a lower boiling point. As the refrigerant passes through the evaporator, the heat flowing to the coil from the surrounding air causes the rest of the liquid refrigerant to boil and vaporize. The refrigerant temperature in an evaporator is directly related to refrigerant pressure. After the refrigerant has absorbed its latent heat of vaporization (that is, after it is entirely vaporized), the refrigerant continues to absorb heat until it becomes superheated by approximately -26.4°C . The amount of superheat is determined by the amount of liquid refrigerant admitted to the evaporator. This, in turn, is controlled by the spring adjustment of the throttling valve. A temperature range of -29.77° to -25.33°C of superheat is considered desirable. It increases the efficiency of the plant and evaporates all of the liquid. This prevents liquid carry-over into the compressor (flooding back). Excessive circulation of the lubricating oil will cause the evaporator temperature to increase. The main cause of slugging is improperly adjusted thermal expansion valve (Kharagpar. 2008). Defrosting of evaporator coils of a multi-box, direct expansion type refrigeration systems, and ice machines can be accomplished by passing hot vapors from the compression cycle through the coils. A re-evaporator is one way to overcome the possibility of a large slug of liquid refrigerant entering the compressor suction when hot gas defrosting a refrigeration system. The evaporators used for refrigeration and air conditioning applications have different types of construction depending on the application. Based on their construction the various types of evaporators are:

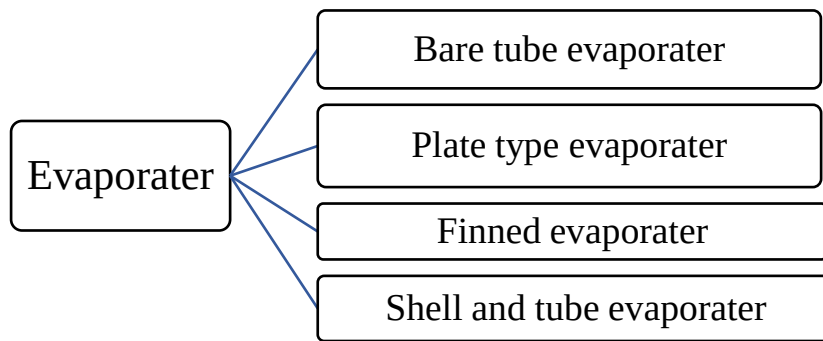


Figure 3.4: Evaporator classification

- **Bare Tube Evaporators** are made up of copper tubing or steel pipes. The copper tubing is used for small evaporators where the refrigerant other than ammonia is used, while the steel pipes are used with the large evaporators where ammonia is used as the refrigerant. The bare tube evaporator comprises of several turns of the tubing, though most commonly flat zigzag and oval trombone is the most common shapes. The bare tube evaporators are usually used for liquid chilling. In the blast cooling and the freezing operations, the atmospheric air flows over the bare tube evaporator and the chilled air leaving it used for cooling purposes. The bare tube evaporators are used in very few applications, however, the bare tube evaporators fitted with the fins, called as finned evaporators are used very commonly.
- **Plate Type of Evaporators:** In the plate type of evaporators the coil usually made up of copper or aluminum is embedded in the plate so as to form a flat looking surface. Externally the plate type of evaporator looks like a single plate, but inside it, there are several turns of the metal tubing through which the refrigerant flows. The advantage of the plate type of evaporators is that they are more rigid as the external plate provides lots of safety. The external plate also helps to increase the heat transfer from the metal tubing to the substance to be chilled. Further, the plate type of evaporators is easy to clean and can be manufactured cheaply.
- **Finned Evaporators:** The finned evaporators are the bare tube type of evaporators covered with the fins. When the fluid (air or water) to be chilled flows over the bare tube evaporator lots of cooling effect from the refrigerant goes wasted since there is less surface for the transfer of heat from the fluid to the refrigerant. The fluid tends to move between the open spaces of the tubing and does not come in contact with the surface of the coil, thus the bare tube

evaporators are less effective. The fins on the external surface of the bare tube evaporators increase the contact surface of the metallic tubing with the fluid and increase the heat transfer rate, thus the finned evaporators are more effective than the bare tube evaporators.

- **Shell and Tube types of Evaporators:** The shell and tube types of evaporators are used in the large refrigeration and central air conditioning systems. The evaporators in these systems are commonly known as the chillers. The chillers comprise of a large number of the tubes that are inserted inside the drum or the shell. Depending on the direction of the flow of the refrigerant in the shell and tube type of chillers, they are classified into two types: dry expansion type and flooded type of chillers. In dry expansion chillers, the refrigerant flows along the tube side and the fluid to be chilled flows along the shell side. The flow of the refrigerant to these chillers is controlled by the expansion valve. In the case of the flooded type of evaporators, the refrigerant flows along the shell side and fluid to be chilled flows along the tube. In these chillers, the level of the refrigerant is kept constant by the float valve that acts as the expansion valve also. Depending on the direction of the flow of the refrigerant in the shell and tube type of chillers, they are classified into two types: dry expansion type and flooded type of chillers. In dry expansion chillers, the refrigerant flows along the tube side and the fluid to be chilled flows along the shell side. The flow of the refrigerant to these chillers is controlled by the expansion valve. In the case of the flooded type of evaporators, the refrigerant flows along the shell side and fluid to be chilled flows along the tube. In these chillers, the level of the refrigerant is kept constant by the float valve that acts as the expansion valve also (Kharagpar. 2008).

For this study, a bare copper tube evaporator was selected because of its simplicity for construction and cleaning from the outside surface, high heat transfer ability, non-corrosiveness, and market availability.

3.2.5 Filter Driers Selection

Filter driers are devices used in HVAC systems that are a combination of filter and dryer (or drier)? A filter is used to remove any particle such as dirt, metal, or chips from entering the refrigerant flow control. The refrigerant flow control device could be a thermostatic expansion valve or simply a capillary tube. The filter is sometimes also referred to as a strainer. These particles must be filtered out and prevented from going into the metering device. It can cause blockage to the passage

flow of the refrigerant in the expansion valve and cause the improper operation to the system. A drier as its name implies is used to remove the moisture from the refrigerant. Sometimes it is also referred to as dehydrator or dryer. It is necessary to remove moisture from the system otherwise it may freeze inside the tube causing the flow of refrigerant to be restricted. The moisture will also cause the formation of acids and sludge when in contact with oil in the system. These devices are usually used in a bigger system where the refrigerant circuit is designed for field service. Residential units that are charged with refrigerant at the factory level and hermetically sealed do not use these devices as the possibility of moisture and particles entering the system are minimal.

Filter-dryer comes in two different types: throw away and refillable type. The former is thrown away if it is not functioning appropriately and it is replaced with a new one. In case of refillable type, the desiccant granules are replaced by a fresh charge after removing the flange, thereafter the same is filled in the line. The desiccant for common use is aluminum sulfate, silica gel, zeolite, etc. for this study throw away type filter drier is selected based on the market availability cost and easily replaceable during maintenance (Maurice, 2011).

3.2.6 Thermostat

Thermostat is a is the switching device used to control the temperature of the device. There are two kinds of the thermostat cold thermostat and hot thermostat. For this study cold thermostat is selected to control the refrigerator temperature of the refrigerator. The cold thermostat used to maintain the refrigerator temperature at the required level. The thermostat operates by using a temperature-sensitive gas sealed inside the thermos bulb which is attached to bellows assembly inside the thermostat housing. As the compartment temperature increases, the gas expands which forces the bellows to expand causing points inside the housing to start the compressor. Once the compartment temperature begins to decrease, the sensitive gas will contract until the bellows shrink and separate the two contacts, cutting off the power to the compressor, thus shutting off the refrigeration systems. If for any reason the thermos-bulb would become broken, allowing the sensitive gas to escape, the thermostat would fail in the open position. This would result in no refrigeration and a new thermostat would have to be installed. The cold thermostat has an external adjustment which may increase or decrease the temperature of the compartment approximately

2°C. Never turn the thermostat to the off position (completely-counter clockwise) as this will turn off the refrigeration system altogether (John, 2017).

3.2.7 Refrigerant Selection

Refrigerant is the fluid that circulates in the refrigeration system to provide cooling for the refrigerator at the evaporator portion of the refrigerator. Refrigerant would have favorable thermodynamic properties, be noncorrosive to mechanical components, and be safe, including freedom from toxicity and flammability. It would not cause ozone depletion or climate change. Since different fluids have the desired traits to different degrees, the choice is a matter of trade-offs. The desired thermodynamic properties are a boiling point somewhat below the target temperature, the high heat of vaporization, a moderate density in liquid form, a relatively high density in gaseous form, and a high critical temperature. Since boiling point and gas density are affected by pressure, refrigerants may be made more suitable for a particular application by appropriate choice of operating pressures (James, 2008).

For this study R134a is selected because it is, available, working in the desired temperature nontoxic and non-corrosive

The materials and types of main components of the refrigerator are selected as summarized in Table 3.1.

Table 3.1: Revision of selected refrigerator components.

No	Component/system	Type of selected component/system
1	Refrigeration system	Vapor compression refrigeration system
2	Refrigerator type	Vertical single door
3	Condenser	Air-cooled copper wire and tube type
4	Compressor	Hermetic
5	Evaporator	Copper Bare tube type
6	Expansion device	Cooper Coiled tube capillary tube
7	Filter drier	Throw away type
8	Thermostat	Cold thermostat
9	Refrigerant fluid	R134a
10	Insulation	Thermo foam

3.3 Sizing of the Refrigerator Components

In this chapter thermodynamics and thermal analysis of the refrigeration system components are made based on the specified capacity of the refrigerator for sizing the refrigerator components. The literature selected for the validation of the study is. Azzouz *et al.* (2009), Cheng *et al.* (2016) and Berdja *et al.* (2018) are around the capacity of 100 and 200 liter. This study has taken the volume of the refrigerator to be 100 L.

3.3.1 Cooling Load Calculation

To specify the cooling load based on the volume of the refrigerator the following load should consider: The quantity of heat to be removed from the product placed in the refrigerator is calculated by using its mass of the product, specific heat capacity above and below freezing temperature, and latent heat of the product. The quantity of heat of the product to be stored above its freezing point temperature is given by (Arora, 2010);

$$Q_{pa} = m * c * \Delta T \quad (3.1)$$

the quantity of latent heat stored in the melting is given as

$$Q_f = m * l_f \quad (3.2)$$

Total load to cool and freeze is given as

$$Q_{tp} = Q_{pa} + Q_f \quad (3.3)$$

Total load to be refrigerated by 24 hours ($\dot{Q}_{Tp}$)

$$\dot{Q}_{Tp} = \frac{Q_{tp}}{24 \text{ hour}} \quad (3.4)$$

Where: Q_{pa} = the quantity of heat of the product above freezing point temperature (kJ)

Q_f = the quantity of heat in freezing (kJ)

l_f = latent heat of fusion ($\text{kJ} \cdot \text{kg}^{-1}$)

m = mass of product (kg)

c = the specific heat capacity at that state ($\text{kJ} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$)

ΔT = the change in the product temperature (K)

The calculation is done by assuming water as a load on the refrigerator. the assumption is taken by two reasons the first reason is that water found in most foods and drinks in high weight percentage.

The other reason is that since water have high specific and latent heat capacity it helps to predict the maximum load on the refrigerator. The calculation is begun by considering the water enter into the refrigerators are at standard temperature 25°C and frozen up to 0°C. Multiplying the cooling load by 24 hours and dividing the outcome by the desired running time in hours to determine the average load alongside putting into consideration the factor of safety, gives the required equipment capacity $\dot{Q}_e$. This average load is referred to as required equipment capacity which is used as a basis for equipment selection. To size, the refrigerator, the loss that occurred on the refrigerator should be considered the cooling load on refrigerating equipment occasionally results from any one source of heat but is the summation of the heat evolving from several different sources heat loss due to walls, because of imperfect insulation and door openings (Andrew *et al*, 2005).

$$\dot{Q}_e = \frac{(\text{total colling load})(24 \text{ hour})}{(\text{desired running time in an hour})} * f \quad (3.5)$$

The amount of refrigeration effect required to cool 100 kg of water from 20 °C to 0°C and 75% of it is changed to water in 24 hours is 0.483 kW. By considering a factor of safety of 10% on the total load the required refrigeration effect becomes (Radha *et al.*, 2012).

$$Q_{pa} = 100 \text{ kg} * 4.18 \text{ kJ.kg}^{-1}\text{K}^{-1} * 20 \text{ K} = 8360 \text{ kJ}$$

$$Q_f = 75 \text{ kg} * 334 \text{ kJ.kg}^{-1} = 25050 \text{ kJ}$$

$$Q_{tp} = 8360 \text{ kJ} + 25050 \text{ kJ} = 33410 \text{ kJ}$$

$$\dot{Q}_{Tp} = \frac{33410 \text{ kJ}}{(24*60*60)s} = 0.387 \text{ kW}$$

$$\dot{Q}_e = \frac{0.387 \text{ kW}/(24 * 60 * 60) \text{ s}}{((24 * 60 * 60) \text{ s})} * 1.1 = 0.43 \text{ kW}$$

3.3.2 Sizing of the Compressor

Compressor is the main component of a vapor compression refrigeration system. Its function is to raise the pressure of the refrigerant and provide the primary force to circulate the refrigerant. The refrigerant thus produces the refrigeration effect in the evaporator, condenses into liquid form in the condenser, and throttles to a lower pressure through the throttling device. The function of the compressor in the vapor compression refrigeration cycle is to compress the low-pressure superheated vapor leaving the evaporator to the high-pressure condenser inlet state. In this study, a hermetically sealed type, the isentropic compressor is selected and sized. In a hermetic

compressor, the motor and the compressor are sealed in the same housing. Hermetic compressors have two advantages: They minimize leakage of refrigerant, and the motor can be cooled by the suction vapor flowing through the motor windings, which results in a small and cheaper compressor-motor assembly. Motor windings in hermetic compressors must be compatible with the refrigerant and lubrication oil, resist the abrasive effect of the suction vapor, and have high dielectric strength (Wang and Shan, 2018).

The following steps are followed in the design of the compressor.

- Select discharge and suction pressure of the compressor
- Take the inlet state as the exit state of the evaporator.
- Take the entropy of exit state equal to inlet entropy.
- Designate the exit state.
- Read the thermos-physical properties of the refrigerant at the inlet and exit states.
- Calculate the suction pressure.
- Calculate the compressor power requirement.
- Determine the volumetric efficiency.

Operating pressure and temperature of the compressor: -Referring to standard vapor compression refrigeration cycle. Figure 3.5 shows the T-s diagram of vapor compression refrigeration cycle.

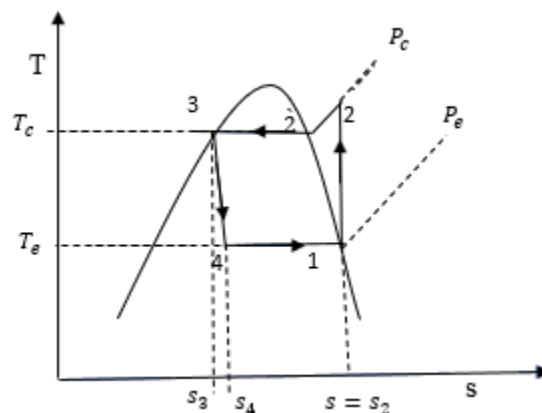


Figure 3.5: Temperatures vs. entropy diagram of an ideal VCRC (Cengel and Boles, 2008).

- Process.1-2 isentropic compression: - refrigerant vapor received from the evaporator is compressed isentropically in a compressor by an external source of energy (work in-out), pressure and temperature increase

- Process 2-3 condensation: - compressor discharges vapor into the condenser where it condensed completely i.e. turns in to liquid. Heat is rejected from the refrigerant to the cooling medium, usually air or water.
- Process 3-4 expansion: - from condenser liquid refrigerant passes through the expansion valve where it is throttled resulting in a drop of temperature and pressure. However, enthalpy remains constant (throttling expansion).
- Process 4-1 evaporation: - liquid refrigerant at a low temperature passes into the evaporator where it extracts heat from the product to be cooled. Due to the absorption of extract heat, the liquid refrigerant turns into vapor and enters into the compressor.

A simple analysis of VCRS can be carried out using assumptions

- I. Steady flow
- II. Negligible kinetic and potential energy changes across each component
- III. No heat transfers in connecting pipelines.
- IV. The steady flow energy equation is applied to each of the four components.

Heat balance for the open system of closed cycle as given by (Cengel and Boles, 2008).

$$E_{in} - E_{out} = \Delta E_{system} \quad (3.6)$$

$$Q_{in} - W_{in} - (Q_{out} + W_{out}) = \Delta E_{system} \quad (3.7)$$

$$Q_e + W_c - Q_c = 0 \quad (3.8)$$

Where: E_{in} = energy interred to the system (kJ)

E_{in} = energy leaves the system (kJ)

Q_e = heat gain by an evaporator (kJ)

Q_c = heat released by a condenser (kJ)

3.3.2.1 Determination of discharge and suction pressure of the compressor

Suction pressure (P_e) and evaporator temperature (T_e)

The refrigerator compartment in this case 0°C. The evaporator temperature should be on the working range of the refrigerant. Consider additional heat transfer from the environment to the refrigerator due to lack of perfect insulation to compensate this temperature difference between the refrigerant and the compartment of the refrigerator must be higher. If a larger temperature difference between inside the compartment and evaporator the T_e value will be lower and compressor power input will be higher. A compromise between energy cost and initial investment

should be made through life-cycle cost analysis. So, it is not recommended much lower evaporator temperature.

In most refrigerator designs the value of temperature difference between the refrigerator compartment and the evaporator is preferred to be between 6 and 10°F (3.3 and 15.6°C). For energy-saving models, the temperature difference between the compartment of the refrigerator and the evaporator should be as low as 4 to 7°F (2.2 to 3.9°C) (Wang and Shon, 2000). Depend on the above consideration assume evaporator temperature ($T_e = -10^\circ\text{C}$) and the Suction pressure the saturation vapor pressure of the refrigerant R134a (evaporator coil working pressure) = 200.6 kpa (Wang and Shan, 2000).

3.3.2.2 Discharge pressure P_c (condenser coil working pressure) and saturated temperature of the condenser (T_c).

The temperature of the refrigerant at the exit of the condenser must be higher than the outside ambient air temperature because heat always flows in the direction of decreasing temperature and to have heat transfer there must be a temperature difference.

Discharge pressure must be greater than atmospheric pressure (1 atm = 101.3 kpa). The saturated temperature of the condenser (T_c) must be higher than the evaporator temperature. The total heat of rejection is equal to net refrigeration at the evaporator (cooling capacity) plus the energy input into the refrigerant by the compressor. If the compressor discharge temperature is very high then it may result in the breakdown of the lubricating oil, causing excessive wear and reduced life of the compressor valves (mainly the discharge valve). In hermetic compressors, the high discharge temperature adversely affects the motor insulation (unless the insulation is designed for high temperatures). When the temperature is high, undesirable chemical reactions may take place inside the compressor, especially in the presence of water. This may ultimately damage the compressor (Kharagpar, 2008).

At a constant evaporator temperature as the condensing temperature increases, then the enthalpy of a refrigerant at the inlet to the evaporator increases. Since the evaporator enthalpy remains constant at a constant evaporator temperature, the refrigeration effect decreases with an increase in condensing temperature. The refrigeration capacity also reduces with an increase in condensing temperature. A high condensing temperature T_c means a higher condensing pressure P_c a higher discharge temperature. High T_c and P_c result in a higher compressor power input and more often unsafe operating conditions so that both are undesirable.

The condenser temperature difference (CTD) for an air-cooled condenser is defined as the difference in saturated condensing temperature corresponding to the refrigerant pressure at the inlet and the air intake dry-bulb temperature $T_2 - T_{air}$ the total heat rejection Q_r of an air-cooled condenser is directly proportional to its CTD. The Q_r of an air-cooled condenser operated at a CTD of 30°F (16.7°C) is approximately 50 percent greater than if the same condenser is operated at a CTD of 15°F (8.3°C). On the other hand, for a specific value of Q_r an air-cooled condenser selected with a CTD of 15°F (8.3°C) is larger than a condenser elected with a CTD of 30°F (16.7°C) (Wang and Shon, 2000). Depend on the above consideration the condensing temperature is assumed to be 60°C .Therefore, the saturated pressure of the condenser (at $T_c= 60^\circ\text{C}$) by interpolation from the saturated table of refrigerant R134a $p_a = 1656 \text{ kPa}$. The saturation properties of R134a at the evaporation and condenser are presented in Table 3.2.

Table 3.2: R134a saturation properties

Temp T °C	Pressure P kpa	Specific volume $\frac{\text{m}^3}{\text{kg}}$	Specific enthalpy $\frac{\text{kJ}}{\text{kg}}$		Specific entropy $\frac{\text{kJ}}{\text{kg} \cdot \text{K}}$	
		v_f	h_f	h_g	s_f	s_g
-10	200.7	0.09963	186.78	392.75	0.9509	1.7337
60	16815	0.01146	287.49	426.86	1.2847	1.703

Figure 3.6: p-h diagram of R134a describes the ideal cycle of the vapor compression refrigeration system. Table 3. presents the properties of R134a in the selected condensation and evaporation pressure.

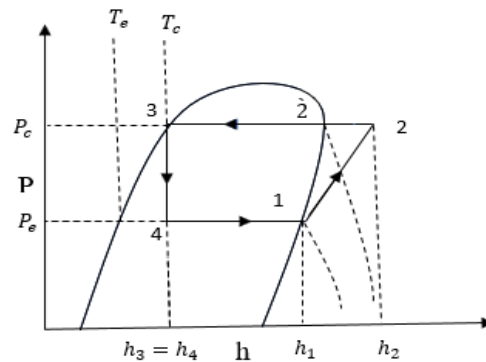


Figure 3.6: p-h diagram of vapor compression refrigeration cycle (Kharagpar, 2008).

Table 3.3: Refrigeration system design data

	Temp. °C	Pressure <i>kpa</i>	Specific volume $\frac{10^3 m^3}{kg}$	Specific enthalpy $\frac{kJ}{kg}$	Specific entropy $\frac{kJ}{kg.K}$	Dryness fraction
@ point 1	-10	200.7	0.09960	392.4	1.733	1
@ point 2	66	1656	0.0124	436..2	1.732	
@point 2'	60	1656	0.01162	427	1.705	1
@ point 3	60	1656	0.000968	287.7	1.286	0
@ point 4	-10	200.7	0.04951	287.7	1.335	0.491

Coefficient of performance of the refrigerator:

The coefficient of performance is an index of the performance of a thermodynamic cycle or a thermal system. Because the COP can be greater than 1, COP is used instead of thermal efficiency.

The COP can be determined using the expression given by (Kharagpar, 2008);

$$COP_i = \frac{\text{input}}{\text{desired output}} = (h_1 - h_4)/(h_2 - h_1) \quad (3.8)$$

Where: h_1 = specific enthalpy at evaporator outlet ($kJ.kg^{-1}$)

h_2 = specific enthalpy at condenser inlet ($kJ.kg^{-1}$)

h_4 = specific enthalpy at evaporator inlet ($kJ.kg^{-1}$)

COP_i = ideal coefficient of performance

The cooling effect (q_e) and the cooling capacity of the evaporator (Q_c): The cooling effect refers to the enthalpy difference when the refrigerant passes through the evaporator coil.

$$q_e = h_1 - h_4 \quad (3.9)$$

$$(\dot{m}_r) = \frac{\dot{Q}_e}{q_e} \quad (3.10)$$

Where: q_e = cooling effect ($kJ.kg^{-1}$)

$\dot{m}_r$ = mass flow rate of refrigerant $kg.s^{-1}$

Power of compressor (P) Cooling capacity is given by

$$P = \dot{m}_r * (h_2 - h_1) \quad (3.11)$$

By using the above equations, the following results are achieved.

$$q_e = (392.4 - 287.7) \frac{kJ}{kg} = 104.7 \frac{kJ}{kg}$$

$$\dot{m}_r = \left(\frac{0.433}{104.7} \right) \frac{kg}{s} = 0.00414 \frac{kg}{s}$$

$$P = 0.00414 \frac{kg}{s} * 44.3 \text{ kJ} \cdot kg^{-1} = 0.183 \text{ kW}$$

$$COP = 2.38$$

3.3.3 Sizing of Evaporator Coil

The main parameters to consider for sizing the evaporator are:

To design evaporator, coil the following parameters must be consider:

- a) It is based on the time taken to extract the required heat from the refrigerator compartment by vaporizing R134a.
- b) It depends on the property of the materials for which the evaporator coil is made because of its thermal conductivity. If the thermal conductivity is high rapid heat exchange occurs from the high-temperature area to low temperature area unless a longer time is required to vaporize the refrigerant. The best material with high thermal conductivity is copper.
- c) The performance of the compressor or the working pressure of both condenser and evaporator will affect. If the pressure is high enough it will require longer if not, it needs smaller coils to exchange the required heat in both a condenser and evaporator.
- d) Mainly it depends on the overall heat transfer coefficient of the hot and cold fluid. Heat transfer analysis on condenser and evaporator coil

There are three modes of heat transfers in evaporator are given by (Theodore *et al.*, 2008):

$$\text{Convection heat transfer } Q_{cov} = h_i A (T_{air} + T_e) \quad (3.12)$$

$$\text{Conduction heat transfer } Q_{con} = \frac{KA}{t} (T_{sur} + T_e) \quad (3.13)$$

$$\text{Radiation heat transfer } Q_{con} = A \epsilon \sigma (T_{sur}^4 + T_e^4) \quad (3.14)$$

$$\text{Overall heat transfer } Q_e = UA \Delta T \quad (3.15)$$

Where: h_i = inside convective heat transfer coefficient

K = Conduction heat transfer coefficient ($W \cdot m^{-1} K^{-1}$)

t = Time (s)

T_{air} = Aire temperature (K)

T_e = Evaporator temperature (K)

T_{sur} = Surface temperature (K)

A = Area (m^2)

U = Overall heat transfer coefficient ($W.m^{-2}K^{-1}$)

Thermal analysis for evaporator coils an evaporator should transfer enough heat from as small size as possible. It should be light, compact, safe, and durable. The pressure loss should be as low as possible. Design factors are: Evaporator temperature, refrigerant properties, refrigerant effect, and tube selection.

The selected evaporator for the design is natural convection bare tube Evaporators. Heat reaches the Evaporator by all three methods of heat transfer. To determine the length of the evaporator the total heat transfer from the evaporator can be written as given by (Cengel and Boles, 2008).

$$Q_c = Q_e = U_o * A_e = F * LMTD \quad (3.16)$$

Where: Q_e = total heat transfer from the evaporator (kJ)

Q_c = Cooling capacity (kJ)

U_o = overall heat transfer based on the outside ($W.m^{-2}K^{-1}$)

A_e = Total area of the evaporator (m^2)

F = Correction factor

LMTD = Long mean temperature difference

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln \Delta T_1 / \Delta T_2} \quad (3.17)$$

Where:

a) $\Delta T_1 = T_{compart} - T_{evap}$ (K)

b) $\Delta T_2 = T_{pro} - T_e$ (K)

Table 3.4: Standard pipe size for evaporator and condenser

Tube size	Outside diameter		BWG	Wall thickness	
	Inch	Mm		Inch	mm
¼	0.250	6.350	22	0.028	0.711
			24	0.022	0.559
3/8	0.375	9.525	18	0.049	1.245
			20	0.035	0.889
			22	0.028	0.711
½	0.500	12.700	18	0.049	1.245
			20	0.035	0.889
5/8	0.625	15.875	16	0.065	1.651
			18	0.049	1.245
			20	0.035	0.889
¾	0.750	19.050	12	0.109	2.769
			14	0.083	2.108
			16	0.065	1.651
			18	0.049	1.245
			20	0.035	0.889

[<http://WWW.engineeringpage.com/technology/thermal/tubesize.html>]

From Table 3.4 the diameter of the evaporator and condenser is selected to be outside diameter 6.35 mm and 0.711 wall thickness based on market availability and literature.

Estimation of overall heat transfer coefficient (U_o)

$$\frac{1}{U_o} = \frac{A_o}{A_i h_i} + \frac{A_o \ln(d_o/d_i)}{2\pi K L} + \frac{1}{h_o} \quad (3.18)$$

As

$$A_o = \pi d_o L \quad (3.19)$$

And

$$A_i = \pi d_i L \quad (3.20)$$

By Substituting Eqn. 3.21 and 3.22 on Eqn. 3.20 the overall heat transfer becomes

$$1/U_o = \frac{d_o}{d_i h_i} + \frac{d_o \ln(d_o/d_i)}{2K} + \frac{1}{h_o} \quad (3.21)$$

To find U_o first, h_i and h_o should be found

h_i estimation

$$h_i = Nu * \frac{K}{d_i} \quad (3.22)$$

To find the Nusselt number, the flow property should be specified depending on the Reynolds number as shown.

$$Re = \frac{\dot{m}_{rif}}{\pi d_i \mu} \quad (3.23)$$

If the flow is turbulence flow inside a tube it considered as fully developed turbulent flow inside a tube and governed by the recommended correlation by Dittes and Boelter for cooling of the fluid is given as.

$$Nu = 0.023 * Re^{\frac{4}{5}} * Pr^{0.3} \quad (3.24)$$

The inside heat transfer coefficient analysis of the evaporator is based on the property of the refrigerant fluid at the evaporator temperature and pressure. To find the property of the fluid after throttling the flash gas percentage after throttling should be calculated by.

$$x = \frac{h_4 - h_f}{h_g - h_f} \quad (3.25)$$

This helps to calculate the viscosity and thermal conductivity of the fluid as shown in Eqn. 3.28 and 3.29.

$$\mu = \mu_l + x(\mu_v - \mu_l) \quad (3.26)$$

$$k = k_l + x(k_v - k_l) \quad (3.27)$$

At T inside the tube is saturation vapor R134a.

From table R134a saturation vapor table @ $T_c = -10^\circ\text{C}$

- The flash gas percentage after throttling: $x = \frac{h_4 - h_f}{h_g - h_f} = 0.491$
- $\mu = 171.38 * 10^{-6} \text{ kg.m}^{-1}\text{s}^{-1}$
- $\dot{m}_r = 0.00414 \text{ kg.s}^{-1}$
- $k = 55.2 * 10^{-3} \text{ W.m}^{-1}\text{K}^{-1}$
- $k_{copper} = 384 \text{ W.m}^{-1}\text{K}^{-1}$

- $Cp = 1.078 \text{ kJ.kg}^{-1}\text{K}^{-1}$
- $Pr = \frac{\mu cp}{k} = 3.39$
- for $Re = \frac{4\dot{m}_r}{\pi d_i \mu} = 4846$ turbulence flow inside a tube fully developed turbulent flow inside a tube the recommended correlation by Dittes and Boelter for cooling of the fluid is given by based on the given data and equations internal convective heat transfer coefficient becomes.
- $LMTD = 92.4 \text{ }^\circ\text{C}$

$$h_i = 255.92 \text{ W.m}^{-2}\text{K}^{-1}$$

h_o estimation

h_o Can be estimated by the following formula given by (Cengel and Boles, 2008);

$$h_o = Nu * \frac{K}{d_o} \quad (3.28)$$

As

$$Nu = c * Re^m Pr^n \left(\frac{Pr}{Pr_s} \right)^{0.25} \quad (3.29)$$

$$Re = \frac{V * d_o}{\nu} \quad (3.30)$$

$$\text{and } Pr = \frac{\nu}{\alpha} = \frac{\mu cp}{k} \quad (3.31)$$

h_o Becomes

$$h_o = c * \left(\frac{V * d_o}{\nu} \right)^m \left(\frac{\mu cp}{k} \right)^n \left(\frac{\frac{\mu cp}{k}}{\frac{\mu cp}{k}} \right)^{0.25} * \frac{K}{d_o} \quad (3.32)$$

Hence, from Table 3.5 for $Re = 258.36$ $4.6 \text{ C} = 0.51$ and $m = 0.5$. Also, since $Pr < 10$, $n = 0.37$.

It follows the Property of air at an assumed mean temperature of the air at 273 k

- $\nu = 13.665 * 10^{-6} \text{ m}^2\text{s}^{-1}$
- $\dot{m}_r = 0.00414 \text{ kg.s}^{-1}$
- $k = 24.3 * 10^{-3} \text{ W.m}^{-1}\text{K}^{-1}$
- $Cp = 1.007 \text{ kJ.kg}^{-1}\text{K}^{-1}$
- $Pr = 0.713$
- $\alpha = 19.2 * 10^{-6}$
- $u_\infty = 0.5 \text{ m.s}^{-1}$
- $p_{rs} = 0.808 \text{ (@ Tsat -10 }^\circ\text{C)}$

Table 3.5: C and M selection table for different Reynolds Table (Theodore, 2011)

Re_D	C	M
1-40	0.75	0.4
40-1000	0.51	0.5
$10^3 - 2 * 10^5$	0.26	0.6
$2 * 10^5 - 10^6$	0.076	0.7

$$h_o = 24.2 \text{ W.m}^{-2}\text{K}^{-1}$$

Finally, U_o becomes:

$$U_o = 21.9 \text{ W.m}^{-2}\text{K}^{-1}$$

$$(Q_e) = U_o * A_e * F * LMTD = 0.433 \text{ kW}$$

$$A_e = 0.225 \text{ m}^2$$

$$\text{As } A_e = 2\pi rL$$

then L becomes= 11.325 m

3.3.4 Sizing of the Condenser

In this section dimensions of a condenser the diameter and thickness of the tube are decided, length of the tube is calculated based on thermal analysis.

The condenser is sized by analyzing the following parameters and conditions:

Thermal property of the condenser material and the refrigerant R134a the higher conductivity it has the higher heat transfer rate will be. In the study, copper is selected because of its high thermal conductivity (Theodore *et al.*, 2008).

Surface area to volume ratio has its effect

$$\text{surface area } (A_s) = 2\pi rL \quad (3.33)$$

Where r = radius of the coil

L= length of the coil

$$\text{Volume } (V_{eva}) = \pi r^2 L \quad (3.34)$$

$$\text{Surface area to volume ratio} = \frac{A_s}{V_{eva}} = \frac{2}{r} \quad (3.35)$$

The temperature difference between ambient air and condensing fluid. High-temperature difference required a smaller length compared to low-temperature difference. Merely it depends on the overall heat transfer coefficient of the hot and cold fluid. Heat transfer analysis on the condenser. There are three modes of heat transfers in the condenser are given by (Cengel and Boles, 2008).

$$\text{Convection heat transfer } Q_{cov} = h_i A (T_{air} + T_e) \quad (3.36)$$

$$\text{Conduction heat transfer } Q_{con} = \frac{KA}{t} (T_{AL} + T_e) \quad (3.37)$$

$$\text{Radiation heat transfer } Q_{rad} = A \varepsilon \sigma (T_{AL}^4 + T_e^4) \quad (3.38)$$

$$\text{Overall heat transfer } Q_e = U A \Delta T \quad (3.39)$$

The diameter and the thickness of the condenser are decided from literature then the length of the condenser calculated based on the diameter thickness of the condenser.

Since inside the condenser tubes, there are two different flow regimes, namely, the superheated vapor and the two-phase mixture, the lengths of these regions are calculated separately, and some of these lengths gives the total length of the condenser. The superheated vapor, which leaves the compressor, enters the condenser and leaves as a saturated liquid. The total heat that should be removed from the condenser is the heat gain from the evaporator and heat added during compression (Cengel and Boles, 2008).

$$\dot{Q}_{cond} = \dot{w}_{com} + \dot{Q}_{evap} \quad (3.40)$$

The heat rejected from the condenser is in two regions first from the superheated region (single-phase region) then from the condensation region (double phase region).

$$\dot{Q}_{cond} = \dot{Q}_{super} + \dot{Q}_{double} = U * F * A_c * LMTD \quad (3.41)$$

LMTD of the condenser is calculated by inlet and outlet temperature and F can be get from the graph by using inlet and outlet temperature as shown below.

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln \Delta T_1 / \Delta T_2} \quad (3.42)$$

Where

The temperature of refrigerant-R134a at the inlet of the condenser $T_2 = 66^\circ\text{C}$

The temperature of refrigerant-R1034a at the outlet of the condenser $T_3 = 60^\circ\text{C}$

The inlet air temperature in the condenser is $T_{air\ inlet} = 25^\circ\text{C}$

The outlet air temperature from the condenser is $T_{air\ outlet} = 30\ ^\circ\text{C}$ (assume value)

For the superheat region $\Delta T_1 = T_3 - T_{air\ inlet} = 41^\circ\text{C}$

$$\Delta T_2 = T_3 - T_{air\ outlet} = 36^\circ\text{C}$$

For the condensation region $\Delta T_1 = T_2 - T_{air\ inlet} = 35^\circ\text{C}$

$$\Delta T_2 = T_3 - T_{air\ inlet} = 30^\circ\text{C}$$

LMTD for superheat region

$$LMTD = 38.9$$

LMTD for condensation region

$$LMTD = 42.2$$

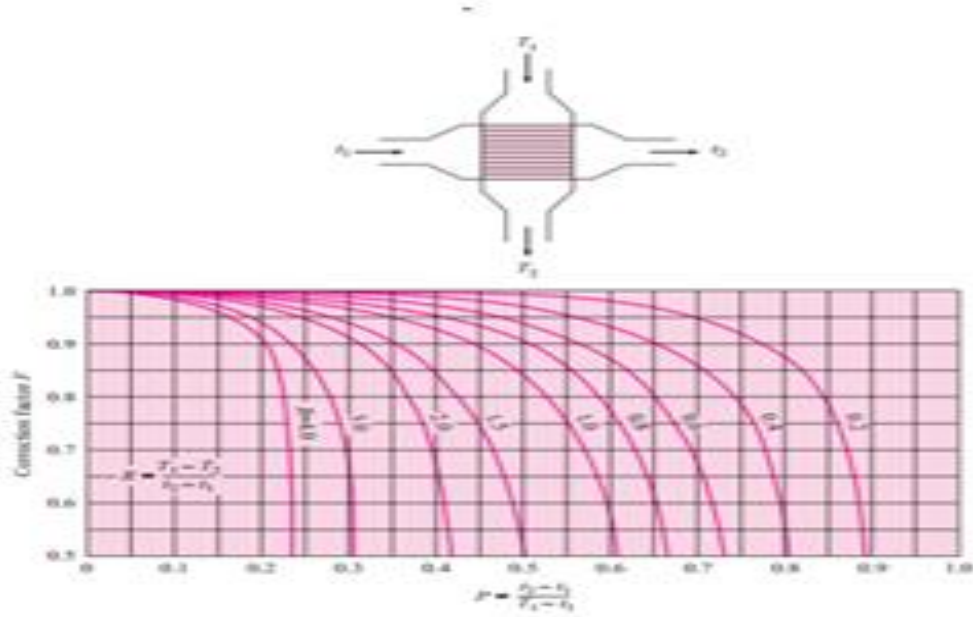


Figure 3.7: Correction-factor plot for single-pass cross-flow exchanger (Chaudhary and Gupta, 2000).

From Figure 3.7 Correction-factor for super-heated region is $f = 0.95$

0.635 mm internal diameter and 0.000711 m thickness of condenser are taken based on literature and market availability. The length of the condenser is calculated below. As heat rejected at two regions the length of the condenser is total of the length required for the superheat region and the length required for the condensing region.

First Estimation of overall heat transfer coefficient (U_o)

$$1/U_o = \frac{A_o}{A_i h_i} + \frac{A_o \ln(d_o/d_i)}{2\pi K L} + \frac{1}{h_o} \quad (3.43)$$

$$\text{As } A_o = \pi d_o L \quad (3.44)$$

$$\text{and } A_i = \pi d_i L \quad (3.45)$$

by substitution Eqn. 3.42 and 3.43 in Eqn. 3.41.

$$1/U_o = \frac{d_o}{d_i h_i} + \frac{d_o \ln(d_o/d_i)}{2K} + \frac{1}{h_o} \quad (3.46)$$

To find U_o first, we have to find h_i and h_o

$$h_i = Nu * \frac{K}{d_i} \quad (3.47)$$

Property of R134a from the table at 60 °C

- $\mu = 14.67 * 10^{-6} \text{ kg.m}^{-1}\text{s}^{-1}$
- $\dot{m}_r = 0.00414 \text{ kg s}^{-1}$
- $k = k_v = 18.19 * 10^{-3} \text{ W.m}^{-1}\text{K}^{-1}$
- $Cp = 1.357 \text{ kJ.kg}^{-1}\text{K}^{-1}$
- $p_r = 1.094$
- $\text{for } Re = \frac{4\dot{m}_{rif}}{\pi d_i \mu} = 56614$ turbulence flow inside a tube fully developed turbulent flow inside a tube the recommended correlation by Dittes and Boelter for cooling of the fluid is given by.
- $Nu = 0.023 * Re^{\frac{4}{5}} * Pr^{0.3}$
- $h_i = Nu * \frac{K}{d_i}$
- $h_i = 0.023 * Re^{\frac{4}{5}} * Pr^{0.3} * \frac{K}{d_i}$
- $h_i = 29.3 \text{ W.m}^{-2}\text{K}^{-1}$

Property of air at an assumed mean temperature of the air at 300 k

- $v = 15.89 * 10^{-6} \text{ m}^2\text{s}^{-1}$
- $\dot{m}_r = 0.00414 \text{ kg s}^{-1}$
- $k = 26 * 10^{-3} \text{ W.m}^{-1}\text{K}^{-1}$
- $Cp = 1.007 \text{ kJ.kg}^{-1}\text{K}^{-1}$
- $p_r = 0.707$

- $\alpha = 22.5 * 10^{-6}$

By the above calculations the convective heat transfer and over all heat transfer became:

$$h_o = 26.13 \text{ W.m}^{-2}\text{K}^{-1}$$

$$U_o = 6.742 \text{ W.m}^{-2}\text{K}^{-1}$$

The length required at the superheat region (L_{sup})

For the superheat (single-phase) region the condenser heat rejection is equal to

$$\dot{Q}_{sup} = \dot{m}_{rif}(h_2 - h_2) = U_o * F * A_c * LMTD = 0.0372 \text{ kW}$$

$$A_c = \pi d_o L_{sup} = \frac{\dot{Q}_{sup}}{U_o * F * LMTD} = 0.151 \text{ m}^2$$

$$L_{sup} = 0.76 \text{ m}$$

Length of the condenser at the double phase region (L_{double})

Heat to be removed from double phase region

$$\dot{Q}_{doubl} = \dot{m}_{rif}(h_2 - h_3) = U_o * A_o * F * LMTD = 0.577 \text{ kw}$$

$$A_o = \pi d_o L_{doubl} = \frac{\dot{Q}_{douple}}{U_o * F * LMTD} = 2.12 \text{ m}^2$$

$$L_{doubl} = 10.7 \text{ m}$$

$$L_{total} = 10.7 \text{ m} + 0.76 \text{ m} = 11.465 \text{ m}$$

3.3.5 Sizing of the Capillary Tube

The capillary tube is the fixed restriction type device. It is a long and narrow tube that connects the condenser directly to the evaporator. The pressure drop in the capillary tube is due to the following factors (Arora, 2010).

- Friction, due to fluid viscosity, resulting in frictional pressure drop.
- Acceleration, due to the flashing of liquid refrigerant into vapor, resulting in a momentum pressure drop.

The cumulative pressure drop must be equal to the pressure difference between condenser and evaporator. the smaller the diameter and the longer its length of the capillary tube increases pressure drop. Sizing of the capillary tube is in effect selecting the length and diameter of the tube. Any different combinations of the length and diameter may give the same pressure drop in the system. There are actually two basic methods to size the tube. The first method is commonly used

in mass production. After selecting the diameter and length of the hose, with greater length is installed in the system. The hose is cut until the desired pressure drop is achieved. This method is called the cut-and-try method. The second method is used commonly for unique system designs. The length and diameter of the hose are selected and a hose which is a little shorter than the desired length is installed in the system. The desired pressure drop is then obtained by pinching the hose at some points (Kharagpar, 2008).

Either using the cut-and-try method or the pinching method, the length and the tube diameter should be decided first. Hence, in the present study, the design of the capillary tube is performed by determining these two parameters. In this study, an isenthalpic capillary tube is considered. The flow in a capillary tube is three-dimensional, compressible, and in two-phase. However, the analytical studies with incompressible, one-dimensional, and single-phase flow as shown in Figure 3.8 assumptions yield highly accurate results.

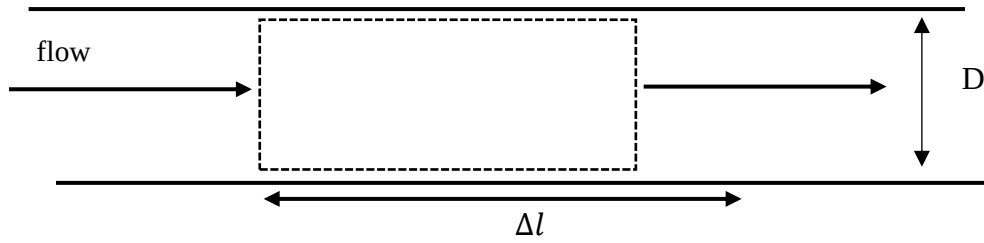


Figure 3.8: Capillary tube

There are two approaches to determine the length of the capillary tube

- a) Isenthalpic expansion.
- b) Adiabatic or Fanno-line expansion.

The isenthalpic assumption is the common thermodynamic assumption. In actual practice, however, expansion takes place adiabatically. But there is no much difference in the pressure drop between the two approaches.

To find the pressure drop in the section, conservation of mass and momentum equations should be solved for this control volume (CV). Conservation of Mass as given by (Kharagpar, 2008);

Conservation of mass:

$$\frac{\partial}{\partial t} \int_{cv} \rho \, dv + \frac{\partial}{\partial t} \int_{cs} \rho v \cdot \hat{n} \, dA = 0 \quad (3.48)$$

$$\frac{\partial(\rho v)}{\partial t} + \rho V A_c - \rho V A_c = 0 \quad (3.49)$$

$$\frac{\partial(\rho v)}{\partial t} = 0 \quad (3.50)$$

Where;

$$A_c = \pi \frac{D^2}{4} \quad (3.51)$$

Hence, $\rho V = \text{constant}$

Conservation of Momentum:

$$\sum M_{out} - \sum M_{in} = \sum F_{cv} \quad (3.52)$$

In other words, the difference between the momentums entering and leaving the CV is equal to the sum of the forces on the control volume as shown in equation 3.51 given by (Kharagpar, 2008).

Including the gravitational force and shear forces in the equation;

$$A_c \left(\rho V^2 + \rho V \frac{\partial V}{\partial L} \Delta L \right) - A_c \rho V^2 = -A_c \frac{\partial P}{\partial L} \Delta L - \bar{\rho} g A_c \Delta L - \pi D \Delta L \tau_w \quad (3.53)$$

Since ΔL is an infinitesimal length, the gravitational force is neglected and higher-order terms are diminished. Dividing both sides by $A_c \Delta L$ as ΔL converges to zero gives.

$$\rho V \frac{\partial V}{\partial L} = -\frac{\partial P}{\partial L} - 4 \frac{\tau_w}{D} \quad (3.54)$$

Shear stress causes a frictional pressure drop in the pipe and therefore, it is written in terms of the friction factor.

$$\tau_w = D \frac{\Delta p_{fric}}{4 \Delta L} \quad (3.55)$$

Where frictional pressure drop is calculated as

$$\Delta p_{fric} = \rho f \frac{\Delta L}{D} \frac{V^2}{2} \quad (3.56)$$

The friction factor is defined for laminar flow ($Re < 2300$) as

$$f = 64/Re \quad (3.57)$$

For turbulent flow, the friction factor is defined by Blasius as

$$f = 0.33 Re^{-0.25} \quad (3.58)$$

For $2300 < Re < 10^5$ the shear stress is written as

$$\tau_w = \rho f \frac{V^2}{4D} \quad (3.59)$$

Substitution of G in the above formula yields

$$G \frac{\partial V}{\partial L} = -\frac{\partial P}{\partial L} - f \frac{VG}{2D} \quad (3.60)$$

Integration of the above formula over the infinitesimal length ΔL gives (Arora, 2010)

$$G\Delta V = -\Delta P - \left(\left(f \frac{VG}{2D} \right)_2 - \left(f \frac{VG}{2D} \right)_1 \right) \Delta L \quad (3.61)$$

Taking the friction factor and the velocity as the average values for states 1 and 2

$$G\Delta V = -\Delta P - G \frac{\Delta L}{2D} f V_{mean} \quad (3.62)$$

The total pressure drop may be found as

$$\Delta P = G\Delta V + G \frac{\Delta L}{2D} f V_{mean} \quad (3.63)$$

To reduce the margin of errors in calculations, the total length of the hose is divided into segments and the total length of the hose is found as the sum of the lengths of these segments. The calculation procedure is explained. as shown in Figure 3.9.

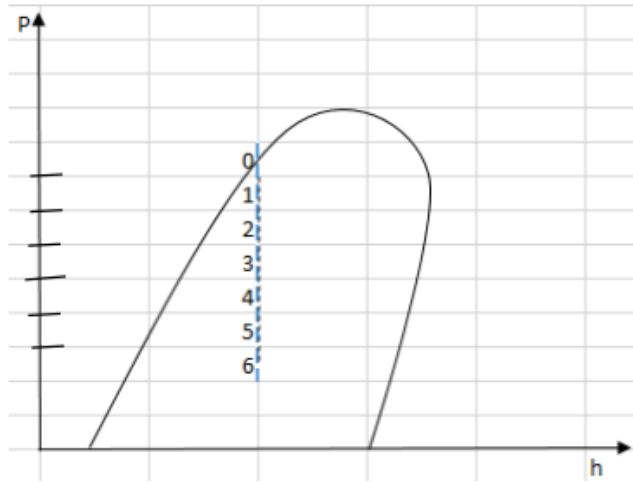


Figure 3.9: Segmentation in capillary tube (Chaudhary and Gupta, 2000).

First, a differential temperature value should be assumed for all segments as seen in Figure 3.9. Thus, $T_1 - T_0 = \Delta T$ The whole temperature and pressure drops from the capillary tube inlet to the outlet is divided into suitable elementary steps of intermediate nodes, such as 60°C, 50°C, 40°C, 30°C, 20°C, 10°C, 0°C and -10°C at their corresponding saturation pressures.

Depending on market availability and literature 1.88 mm internal diameter of capillary tube is selected. By following the Adiabatic or Isenthalpic expansion length determination system the following results attain are presented in the Table 3.6 up to 3.8.

As $\dot{m}_r = 0.00414 \text{ kg}$

➤ $A = 2.55 * 10^{-6} \text{ m}^2$

➤ $G = \frac{\dot{m}}{A} = 1623 \text{ kg.s}^{-1}\text{m}^{-2}$

➤ $Y = \frac{G}{2D} = 461874 \text{ kg.s}^{-1}\text{m}^{-3}$

➤ $Z = DG = 3$

Table 3.6: Calculated enthalpy due to change in speed

Section	T (K)	P (kpa)	$v (m^3/kg)$	$u (m/s)$	$\Delta h (kJ/kg)$
K	333	1686	0.001017	1.691271	0
1	323	1317	0.0025	4.1575	0.072122043
2	313	1011	0.004715	7.841045	0.220985902
3	303	762.7	0.008017	13.332271	0.581337317
4	293	565.3	0.01294	21.51922	1.426636897
5	283	411.6	0.02032	33.79216	3.39416624
6	273	294.5	0.03144	52.28472	7.95890934
7	263	199.8	0.041	68.183	9.576147718

Table 3.7: Friction factor determination

POINT	T (K)	X	$\mu_f \left(\frac{kg}{m.s} \right)$	$\mu_g \left(\frac{kg}{m.s} \right)$	$\mu \left(\frac{kg}{m.s} \right)$	Re	F
K	333	0	0.138	0.014.71	0.138	21686	0.02637
1	323	0.112	0.158	0.0138	0.1418496	21078.6636	0.026557624
2	313	0.202	0.17	0.013	0.138286	21621.85615	0.026389231
3	303	0.275	0.2	0.012	0.1483	20161.83412	0.026854525
4	293	0.341	0.225	0.0119	0.1523329	19628.06459	0.027035264
5	283	0.398	0.254	0.011	0.157286	19009.95639	0.027252396
6	273	0.456	0.287	0.01094	0.16111664	18557.98383	0.027416833
7	263	0.498	0.326	0.01049	0.16887602	17705.29647	0.027741132

Table 3.8: Length calculation Table

Section	Δl (m)
k-1	2.385096921
1-2	1.045908626
2-3	0.482099705
3-4	0.227986118
4-5	0.10445578
5-6	0.043472016

Total change length of the capillary tube become:

$$L = 4.3 \text{ m}$$

3.3.6 Insulation

Thermal conductivity of thermo foam outside area of the refrigerator 2.64 m^2

Properties of air average temperature 298 k

- $\nu = 15.89 * 10^{-6} \text{ m}^2 \text{ s}^{-1}$
- $L = 0.05 \text{ m}$
- $k = 26 * 10^{-3} \text{ W.m}^{-1} \text{ K}^{-1}$
- $Cp = 1.007 \text{ kJ.kg}^{-1} \text{ K}^{-1}$
- $p_r = 0.707$
- $\alpha = 22.5 * 10^{-6}$
- $U_\infty = 0.5 \text{ m}^3$

$$\text{Re} = \frac{U_\infty L}{\nu} \quad (3.64)$$

Then h_o can be calculated by Eqn. 3.31.

$$h_o = 26.4 \text{ W.m}^{-2} \text{ K}^{-1}$$

$$h_i = 26.5 \text{ W.m}^{-2} \text{ K}^{-1}$$

U_o can be calculated by Eqn. 3.63

$$\frac{1}{U_o} = \frac{1}{Ah_i} + \frac{L}{KA} + \frac{1}{h_o A} \quad (3.65)$$

$$U_o = 0.3045 \text{ W.m}^{-2} \text{ K}^{-1}$$

$$Q_{lost} = AU_o(T_{\infty} - T_{COLD}) \quad (3.66)$$

$$Q_{lost} = 20 \text{ W}$$

3.3.7 Exegetic Analysis

The expansion valve was assumed to work adiabatically (isenthalpic), where enthalpy of the refrigerant entering and exiting the expansion valve is equal ($h_3 - h_4$).

$$COP = \frac{q_{re}}{w_k} = \frac{h_1 - h_4}{h_2 - h_1} \quad (3.67)$$

Irreversibility at the compressor, condenser, expansion valve, evaporator, and receiver is given in the Equation as derived by Tambunan and Abdullah, (2017). The total irreversibility of the system is the sum of the irreversibility of each component. Total irreversibility is regarded as a loss of work. Reversible work can be calculated. Accordingly, the energetic efficiency of the system can be calculated by using the Equation stated by Yumrutas *et al.* (2000). Energy balance on a closed system work is generally defined in terms of the rate change, like the following Eqn. 3.66.

$$\frac{dE}{dt} = Q - W + \sum \dot{m}_o h_o - \sum h_1 \dot{m}_1 \quad (3.68)$$

The compressor's real work generally generates heat (Q_{comp}) and releases it around, so the energy the balance on a compressor is as in the following Eqn. 3.67:

$$0 = \dot{Q} - W + \dot{m}_2 h_2 - \dot{m}_1 h_1 \quad (3.69)$$

Subscripts

1: at compressor inlet 2: at compressor outlet 3: at condenser outlet 4: at evaporator inlet. The compressor works is modeled as an open system, so the entropy generation process (σ) as shown in the Eqn. 3.68:

$$0 = \frac{\dot{Q}_{COMP}}{T_{COMP}} - W + \dot{m}(s_2 - s_1) + \sigma_{comp} \quad (3.70)$$

Ambient temperature is the dead-state temperature T_o ; then, the entropy generation process at the compressor is based on the dead-state temperature ($T_o \sigma_{comp}$) same as exergy destroyed during the compression process and stated as irreversibility process (I_{comp}), or

$$I_{comp} = T_o \sigma_{comp} \quad (3.71)$$

The isentropic process takes place and is assumed to have no heat transfer (adiabatic),

so that $Q = 0$.

Then, the exergy balance of compressor is calculated using the following equation:

$$0 = \dot{Q}_{comp}(1 - \frac{T_o}{T_{comp}}) - w + \dot{m}(h_2 - h_1) - T_o(s_2 - s_1) + T_o\sigma_{comp} \quad (3.72)$$

So that the exergy balance of each component can be obtained by the Eqn. 3.71:

$$\text{Exergy at any point exergy} = (h - h_o) - T_o(s - s_o) \quad (3.73)$$

Compressor entropy generation

$$T_o\sigma_{comp} = I_{comp} = \dot{Q}_{comp}(1 - \frac{T_o}{T_{comp}}) - w + \dot{m}(h_2 - h_1) - T_o(s_2 - s_1) \quad (3.74)$$

Condenser entropy generation

$$T_o\sigma_{cond} = I_{cond} = \dot{Q}_{cond}(1 - \frac{T_o}{T_{cond}}) - w + \dot{m}(h_3 - h_2) - T_o(s_3 - s_2) \quad (3.75)$$

Expansion valve entropy generation

$$T_o\sigma_{expa} = I_{expa} = \dot{m}(h_4 - h_3) - T_o(s_4 - s_3) \quad (3.76)$$

Evaporator entropy generation

$$T_o\sigma_{evap} = I_{evap} = \dot{Q}_{evap}(1 - \frac{T_o}{T_{evap}}) - w + \dot{m}(h_1 - h_4) - T_o(s_1 - s_4) \quad (3.77)$$

Total irreversibility process

$$I_{total} = I_{comp} + I_{cond} + I_{expa} + I_{eva} \quad (3.78)$$

Compressor reversible work

$$w_{rev} = \dot{Q}_{eve}(\frac{T_{cond}}{T_{evap}} - 1) \quad (3.79)$$

Where

- T_{comp} the refrigerant temperature at compressor outlet, (K)
- T_{cond} is the refrigerant temperature at condenser inlet, (K)
- T_{evap} is the refrigerant temperature in the evaporator plate (K)
- T_o is the temperature of the surrounding air, (K)

Exergy efficiency

The efficiency of thermodynamic law known as exergy efficiency or effectiveness can be defined as the ratio of minimum/reversible work (w_{rev}) against real work (w_{rc}) and the amount of

compressor work loss equal to the amount of irreversibility ($w_l = I_{total}$) so, exergy efficiency can be calculated as in Eqn. 3.78.

$$\mu_{exe} = \frac{w_{rev}}{w_{rev} + w_l} \quad (3.80)$$

The total irreversibility can be accounted for as loss of work, while the sum of the loss of work and reversible work can be regarded as the measured work consumption by the compressor. Accordingly, the exergy efficiency can be calculated using the Eqn. 3.81.

$$\mu_{exe} = 1 - \frac{w_l}{w} \quad (3.81)$$

CHAPTER FOUR

CONSTRUCTION OF THE REFRIGERATION SYSTEM

4.1 Introduction to Fabrication Process

Performance analysis of PCM integrated refrigerator require suitable PCM placement for frequent changing of the PCM at the condenser and evaporator for the sake of that the prototype is fabricated locally. The vapor compression refrigeration system prototype was fabricated locally on the basis of the design decisions and calculated values of the main components discussed in Chapter 3. In this chapter the fabrication of evaporator, expansion valve, condenser, and other components with specifications indicated in Table 4.1 are discussed. After prototype fabricated by the specification of the components found in the sizing portion and assembly of parts is made in a way suitable for the experiment. The body of the refrigerator with the designed component and size with the required volume is purchased. The wire and tube condenser which is greater than the designed length is purchased and removed the extra part of the condenser to the line between the compressor and the condenser. A copper tube evaporator, capillary tube, and filter drier are purchased as specified in chapter 3. Table 4.1 reviews the overall component type and size of the refrigerator. The packing frame for the PCM materials were also prepared. Materials required for the experimentation such as PCM materials ($\text{CaCl}_2 \cdot 6\text{H}_2\text{O}$, NaCl_2 , and paraffin wax), wrappings for the PCMs, water, container, measuring jars, balance, glue, melting pan, thermometer are prepared. The instruments needed for the experiment, such as thermocouple wire, thermocouple thermometers, mercury thermometers, digital thermocouples are calibrated. Finally, the temperature measuring points and PCM integration places are shown.

Table 4.1: Refrigerator system components description

No	Component/system	Type of selected component/system	Dimension
1	Refrigeration system	Vapor compression refrigeration system	
2	Refrigerator type	Vertical single door	Body size: Width 480 mm, height 122 mm, 52 mm. Insulation 5 mm
3	Condenser	Air-cooled wire and tube type	Condenser size: width 420 mm, height 900 mm, tube dimension: length 11.465 mm, diameter 6.35 mm, Wire dimension: diameter 1.5 mm, height 900 mm distance between wire 10 mm, the total number of wires on each side 46
4	Compressor	Centrifugal	$\frac{1}{4}$ hp compressor (L72CZ) Danfoss hermetic compressor
5	Evaporator	Bare tube, free convection, and roll-bond	Evaporator length 11.352 m, outer diameter 6.35 mm, and 0.711 mm thickness
6	Expansion device	Coiled tube capillary tube	1.88 mm internal diameter 0.22 mm thickness, and 4.3 meter long.
7	Filter drier	Throw away type	
8	Thermostat	Cold thermostat	

4.2 Fabrication of Evaporator

As selected and sized in chapter three, a roll-bound copper tube outer diameter of 6.35 mm, 0.711 mm thick and 15 m long is purchased. Usually the evaporator is placed at the back side of the refrigerator behind the internal wall of the refrigerator cabinet and separated by insulator usually by thermo-foam the outside wall of the refrigerator. in ordinary house hold refrigerator the cabinet have three parts. At the top there will be a freezer mostly used for making ice. The next part is a

moderately cold part which is the middle of the cabinet it mainly used to store processed food and the lower part of the cabinet have relatively higher temperature and it mostly chose to place fruits. This kind of temperature distribution is gained by warping the evaporator at the upper part of the evaporator and letting the lower parts of the cabinet cooled by convectional heat transfer from the top to the bottom. But in this thesis a different approach of evaporator installation has been used for three reasons. The first reason is that during experimental testing the PCM is frequently placed and removed from the evaporator. Placing the evaporator in the back of the inside wall require frequent infiltration of the wall. This dictate the evaporator to be placed in the conferrable way to do so. The second reason is lack of uniform temperature distribution in the cabinet affect the measurement of inside temperature. The third reason is making the temperature measurement easier as the evaporator temperature should be measure in different parts of the evaporator. In this thesis the evaporator is placed in the inside of the cabinmate this helps to easily measure the temperature of the cabinet, achieving uniform temperature, and conferrable PCM placement. As shown in the Figure 4.1 the evaporator is folded in the frames in three layers and 8 rows in each layer to place it in the cabinet.



Figure 4.1: Fabrication of evaporator

4.3 Fabrication of Capillary Tube

In section a capillary tube expansion valve is designed for throttling process. It designed by choosing the diameter of the capillary tube based on literatures and market availability and second calculating the required length for the required pressure drop. As shown in Figure 4.2 copper tube of 5 m long and 1.88 mm diameter of capillary tube is purchased from the market. In the capillary tube installation, the capillary tube is first cut slightly smaller than the designed value and the required temperature is attain by pinching the capillary tube until the required temperature is gain at the outlet. The same procedure is followed in this thesis.

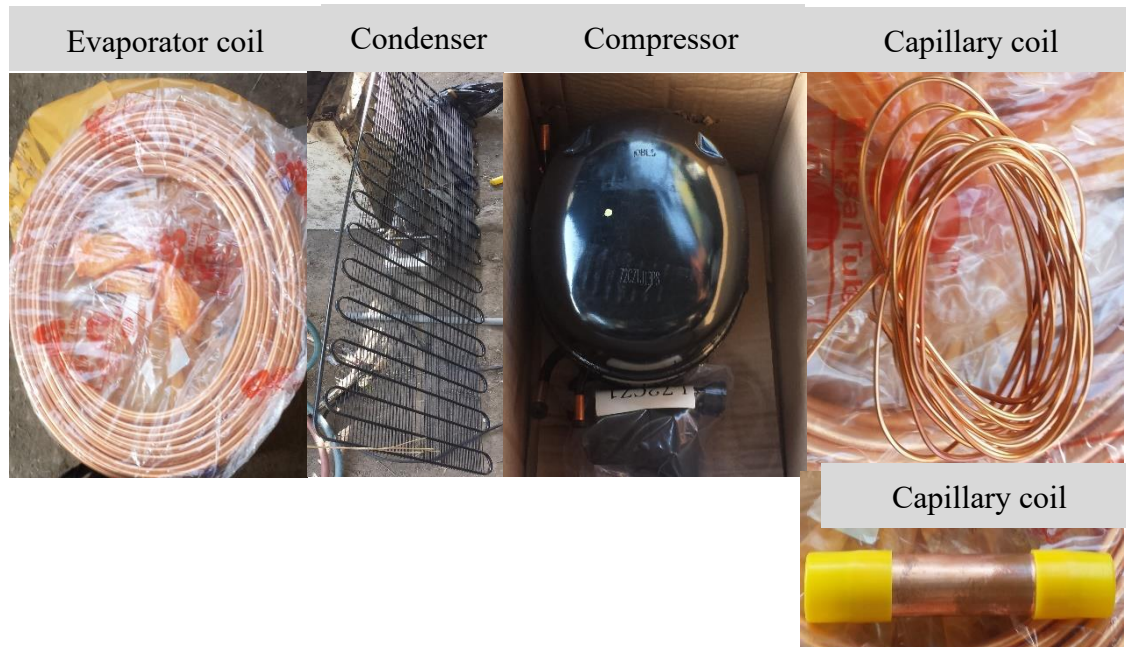


Figure 4.2: Materials for fabrication of evaporator, capillary tube, and filter drier

4.4 Fabrication of Condenser

The air-cooled wire and tube copper condenser are selected in the chapter three and in the sizing section of chapter three the length is designed to be 10 m long, 6.35 mm diameter and 0.77 mm thick. As shown in Figure 4.4, wire and tube condenser with the required diameter, thickness, and length is purchased. with 46 fins in each side (at the front and the back of condenser) the gap between the fins is 10 mm as listed in Table 4.1.

4.5 Other Components

Compressor, filter drier and cold thermostat are purchased due to the specification in the selection and designing section. A ¼ hp compressor (L72CZ) Danfoss hermetic compressor is purchased.

4.6 Assembly of Refrigerator Parts

The condenser is placed at the backside of the refrigerator by screwing the condenser in to the body of the refrigerator for support system and in order to gain enough space for PCM insertion and avoid vibration during working of compressor it tightly warped to the body by inserting bench with the required gap. Finally, it brazed one end with a compressor and other with filter drier by acetylene and oxygen gas welding.

Evaporator wrapped in the cabinet of the refrigerator as shown in Figure 4.3; this kind of evaporator installation gives approximately uniform temperature in the compartment which is very essential for the experimentation. It also makes the placement on the evaporator easy which means the PCM can easily be placed in the evaporator without affecting the insulation of the refrigerator. And it brazed with the capillary tube in one end and with the compressor on the other end. Capillary tube brazed with evaporator inlet and filter drier.



Figure 4.3: Evaporator installation on the refrigerator

Finally, the compressor is assembled by brazing it with the evaporator outlet and condenser inlet. The electrical system is installed with the connection of the thermostat which sensed the outlet temperature of the evaporator to auto-start and stops the compressor as shown in figure 4. 4.



Figure 4.4: Placement of compressor on the refrigerator

4.7 Leakage Test

Leakage tests are done before and after charging refrigeration gas into the system. After the system fully assembled it evacuated from the air and tested for leakage by plugging pressure gage and measuring vacuum pressure of an hour, it maintains the same pressure which means no linkage in the system. The R134a gas has charged in to the system with the required pressure the second leakage test is done after working for 14 hours and the pressure reads shows that there is no leakage in the system. Figure 4.5 shows the pressure measurement during evacuation.



Figure 4.5: Measuring the pressure of the refrigerator

After the refrigerator is secured from leakage the R134a refrigerant is charged to the compressor. The property of the refrigerant R134a major properties are listed in Table 4.2.

Table 4.2: Property of R134a

Property	Value
Boiling point at 1 atm (101.3kpa or 1.013bar)	-26.3 °C
Freezing point	-103.3 °C
Critical temperature	101.1 °C
Critical pressure	4060 KPa
Critical volume	$1.94 * 10^{-3} m^3/kg$
Density (liquid at 25 °C)	$1206 kg/m^3$

4.8 Experimental Setup

Figure 4.6 shows the refrigerator proposed to the experiment. Conventionally, an air-cooled wire and tube type condenser is attached to the back of the refrigerator. The refrigerant flowing inside the tube rejects heat to the ambient due to the natural convection. It is a known fact that with the higher ambient temperature, the condenser temperature and condenser pressure settle on the higher side.



Figure 4.6: Assembled refrigerator

. Due to this, during the noontime, the refrigerator compressor power consumption would be the highest. To minimize the energy consumption at the peak duration as well as increasing heat transfer on other times a PCM-based condenser is used.

4.8.1 PCM Preparation

To study which phase change temperature is suitable for better COP Two types of PCM are used in the condenser one with phase change temperature 28°C the other with phase change temperature $58/60^{\circ}\text{C}$ and their effect on COP considered with without PCM. Two types of Hydrated salt PCMs are placed in the evaporator portion with phase change temperature below zero (-4°C) and below zero (-8°C) and their COP considered with refrigerator COP without PCM. After separately considered the effect of PCM placed in the condenser and in the evaporator the dual effect of PCM placed in both condenser and evaporator at the same time is measured. The property of the PCM used in the performance study of the refrigerator are reviewed in Table 2.2.

Basically, there are two types of PCM used in this thesis, hydrated salt and paraffin wax. The hydrated salt has properly measured and mixed with water to form PCM with the desired phase change temperature. Figure 4.7 illustrates the measuring device used for preparing PCM:

measuring jars, analytical balance and mixing jars are the main instruments used in the preparation process.



Figure 4.7: Measuring instrument used in PCM preparation

Figure 4.8 shows the preparation of aluminum package for the PCM. The package for the PCM is made from aluminum. aluminum is selected because of its good heat conducting ability, ductility, and its non toxicity for the foods and other products in the refrigerator. the other reason aluminum has used for condenser PCM packaging is its high reflecting ability of the radiation of the condenser heat to the surrounding.



Figure 4.8: Aluminum foil for PCM packaging

After the PCM poured in to the aluminum package its openings are closed by silica-gel and it takes 24 hours until the silica-gel dries and ready for use. The paraffin wax is first measured by the analytical weight balance and melted in the stove so that it can be poured to the aluminum foil as shown in the Figure 4.9 after the aluminum foil warped on the PCM its edges are attached each other by silica-gel.



Figure 4.9: Preparation of aluminum foil

a

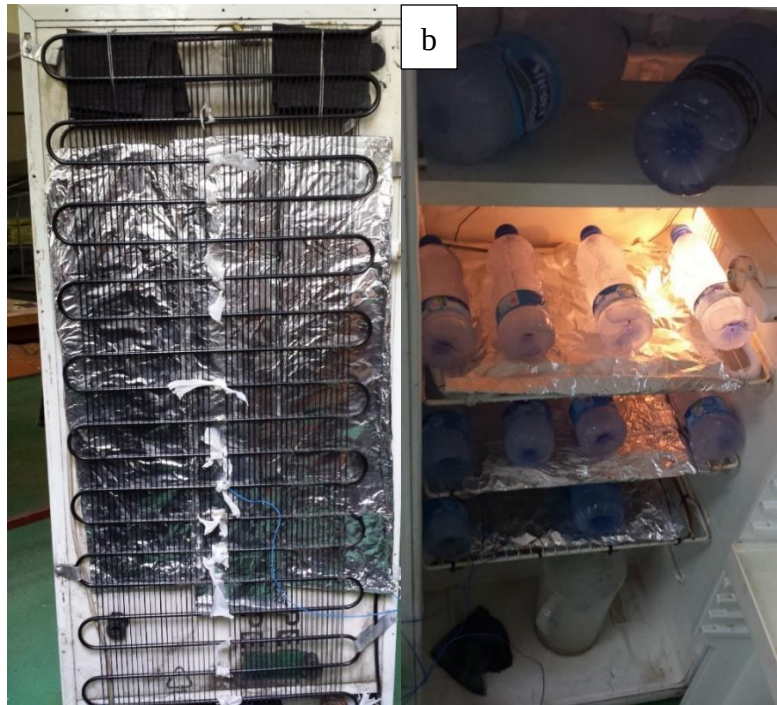


Figure 4.10: PCM placement on evaporator and condenser (a) Backside view (b) Front side view After the PCM poured in to the aluminum foil first prepared and wrapped it warped again in the other aluminum foil to strengthen the aluminum foil further. The Figure 4.10 shows the PCM in the refrigerator for the condenser area. After the preparation is done the PCM placed in the condenser back as planned in this thesis.

4.9 Instrumentation

As shown in Figure 4.11, K-type Thermocouple thermometers (HT-9815 and WL-362) are used for the measurement of temperature. Mercury thermometer is used for calibrating thermocouples. Digital balance and volume measuring jar are used for PCM preparation as shown in Figure 4.11.

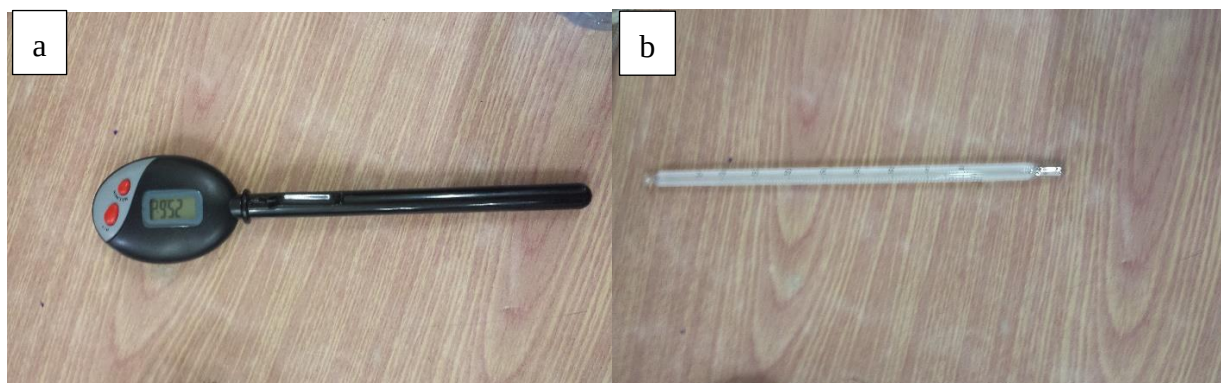




Figure 4.11: Temperature measuring instruments: (a) Digital thermocouple thermometer, (b) mercury thermometer (c) k type thermocouple (HT-9815) (d) k type thermocouple thermometer (WL-362),.

4.10 Calibration of Thermocouples

As shown in Figure 4.12 thermocouples are calibrated with a mercury thermometer. The calibration has two steps first the thermocouples have inserted in the water ice bath at 0°C along with mercury thermometers and readings are taken. For better results, this process done by boiling the water the readings are recorded. After the recording completed the deviation of thermocouple thermometers is calculated. Calibration of thermocouples are explained in appendix D



Figure 4.12: Calibration of temperature measuring instrument with ice bath

4.11 Temperature Measurement

Usually, the refrigerant fluid at the condenser is at three states superheat, phase changing, and subcooling. According to the state of the refrigerant, the temperature gradient changes over the length of the condenser. To determine how the temperature changes over the length of the condenser many readings are taken on the condenser. Different points of temperature measurement taken are shown in Figure 4.13.

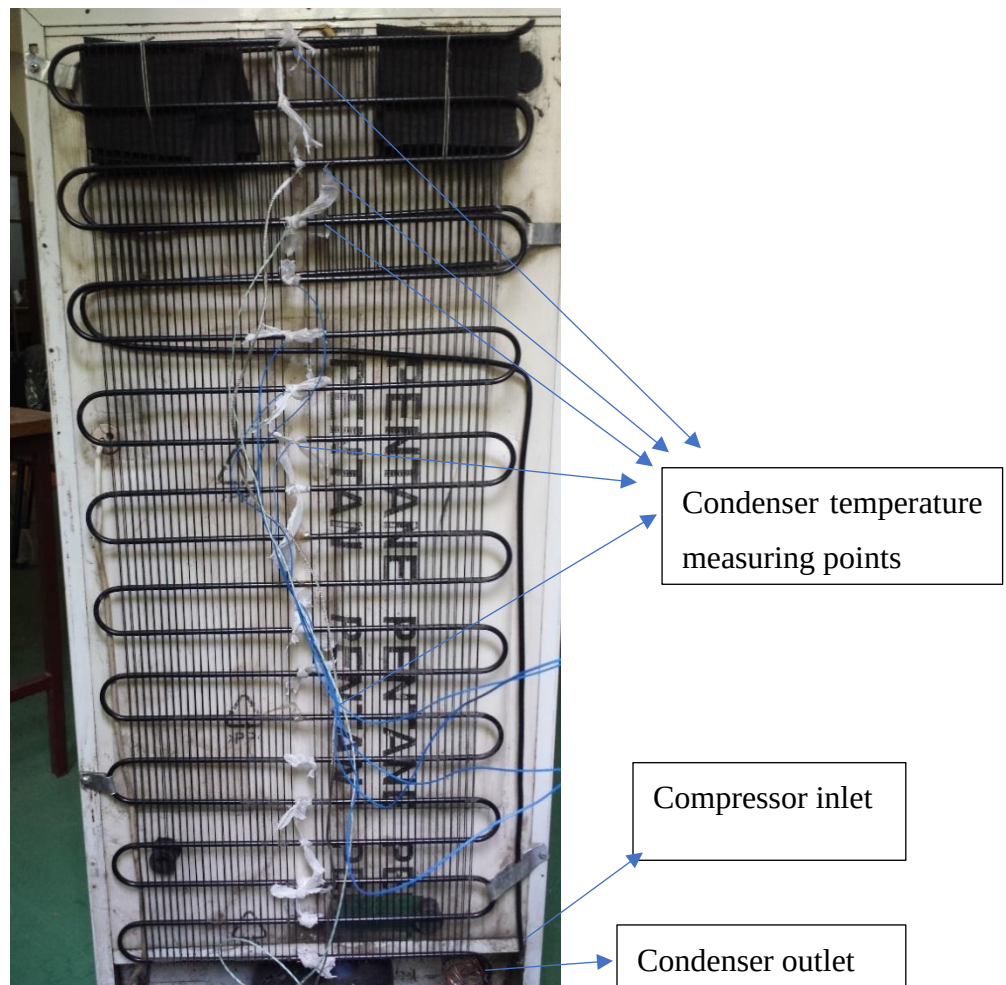


Figure 4.13 Condenser temperature measuring points

The evaporator temperature is measured at three points at the inlet, middle, and outlet of the evaporator. The temperature of the compartment and water which took as the load on the refrigerator are taken at deferent points as shown in the figure below.

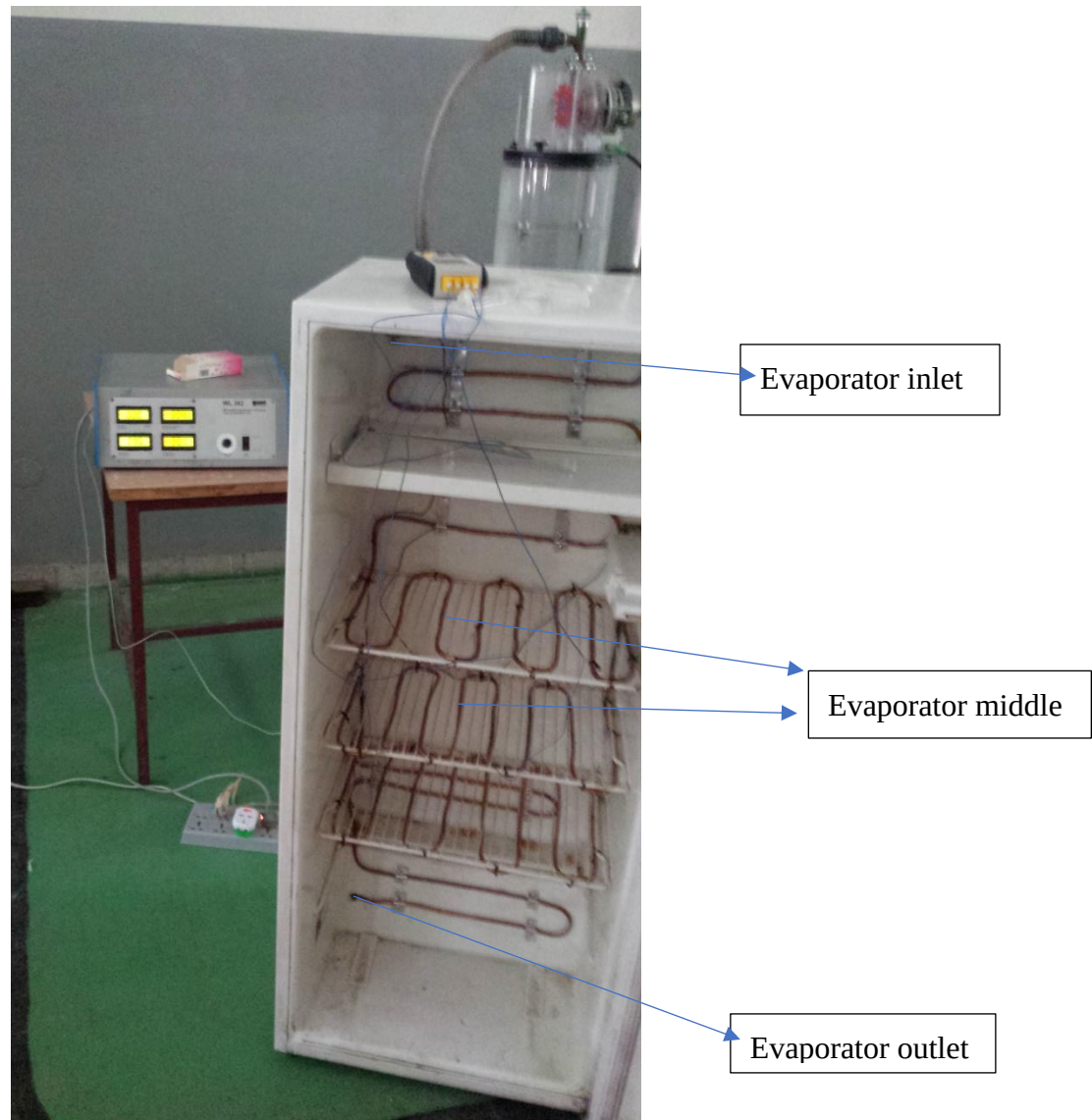


Figure 4.14: Evaporator temperature and compartment temperature measuring points

PCM temperature is measured at its surface in the middle and at the backside of different length and width of PCM. This test helps to examine the melting and solidification of PCM. Measuring the temperature of the PCM were specialty done on the temperature fluctuation test because the temperature of the PCM should seated at the sub cooling temperature -9°C to see the full effect of the evaporator placed PCM on the temperature fluctuation of the refrigeration cabinet.

CHAPTER FIVE

EXPERIMENTAL PERFORMANCE OF THE REFRIGERATION SYSTEM

5.1 Introduction

In this chapter, the result and its implication obtained from investigating the experimental performance of the fabricated refrigerator with and without PCM integration are presented and discussed. Experimentation was carried out to investigate the performance of the fabricated refrigerator with and without load by measuring surface temperature of the condenser, evaporator, compartment, product, and ambient temperature. By using four different amount of liquid water and three PCM materials as loads the experimental performance of the refrigerator was investigated. Since the performance of the PCM integrated refrigerator mainly depend on the place where PCM integrated, the phase change temperature of the PCM, the amount of PCM, and their combination, these effects are experimentally investigated and discussed in the section below. Before starting PCM based tests, the performance test of the refrigerator as designed is done. After the refrigerator fabricated and conducted leakage test by running the refrigerator for 24 hours the no-load test and the load test are done for 195 minutes, to compare the performance of the refrigerator. The performance of the refrigerator was evaluated in terms of coefficient of performance (COP) forms: ideal coefficient of performance (COP_i), actual coefficient of performance (COP_{ac}), and calculated coefficient of performance (COP_{ca}). The ideal coefficient of performance of the refrigerator is the coefficient performance of the refrigerator working with the analytically designed cycle between -10 and 60 °C evaporating and condensing temperature, respectively as shown in Eqn. 5.1. The calculated coefficient of the refrigerator is calculated by the enthalpy which is gained by using temperature measured values at condenser and evaporator inlet and outlet points as shown in Eqn. 5.2. The calculated coefficient of performance of the refrigerator is calculated by taking the ration of the amount of heat extracted from the water inside the refrigerator to the actual compressor power as shown by (Arora, 2010) Eqn. 5.3.

$$COP_i = \frac{h_1 - h_4}{h_2 - h_1} \quad (5.1)$$

$$COP_{ca} = \frac{h_{1ca} - h_{4ca}}{h_{2ca} - h_{1ca}} \quad (5.2)$$

$$COP_{ac} = \frac{m(c_p\Delta T + h_{fg})}{\dot{w}_{comp}t} \quad (5.3)$$

Where: h_4 , h_1 , and h_2 , are the designed value of evaporator outlet, evaporator inlet, and compressor outlet specific enthalpy, respectively.

h_{4ca} , h_{1ca} , and h_{2ca} , Are the evaporator outlet, evaporator inlet, and compressor outlet specific enthalpy respectively calculated from the measured points of temperature of the condenser and evaporator. ΔT is the difference of final and initial temperature of the water temperature. c_p the specific heat of the water. $\dot{w}_{comp}$ is the work done by the compressor. m, h_{fg} mass of water and enthalpy of solidification of water respectively. t the compressor running time which can be calculated by the total testing time multiplied by the on-off ratio of the compressor.

Along with the COP the performance of the refrigerator is also depend on the on-off ratio and exergy efficiency. The exergy efficiency of the refrigerator is calculated by Eqn. 3.79 after calculating the entropy generation on the system. On-off ratio of the compressor is the ratio of the compressor working time to the time which the compressor stops.

Energy balance

The energy balance of the refrigerator is calculated by the following equations (Arora, 2010)

$$\dot{Q}_{cond} = \dot{w}_{com} + \dot{Q}_{evap} \quad (5.4)$$

$$\dot{Q}_{evap} = \dot{Q}_{lost} + \dot{Q}_{abs} \quad (5.5)$$

$$\dot{Q}_{cond} = \dot{w}_{com} + \dot{Q}_{lost} + \dot{Q}_{abs} \quad (5.6)$$

$$COP_{ac} = \frac{\dot{Q}_{abs}}{\dot{w}_{com}} \quad (5.7)$$

$$COP_{ca} = \frac{\dot{Q}_{evap}}{\dot{w}_{com}} \quad (5.8)$$

5.2 No-Load Versus on Load Test

Figure 5.1 and 5.2 illustrates the comparison between no-load and on-load performance parameter of the condenser. The temperature measurement is taken in the inlet, middle, and outlet of the condenser and evaporator. For the both tests (no-load and load) temperature measurement taken at the start and end of the compressor and every five minutes between the start and the stop of the compressor. in the on-load test, the refrigerator is loaded with water at room temperature around 25 °C and the temperature change is measured by inserting thermocouple in the water bottles and the result is taken to calculate the refrigeration effect of the refrigerator. First, the refrigerator is

loaded with 15, 20, 25, and 30 kg of water respectively, and the performance of the refrigerator expressed in terms of temperature versus time graph and, COP, and exergy efficiency.

As shown in the Figure 5.2 the evaporator temperature at the inlet in the on-load test is higher than that of the no-load test this in turn increases the compressor outlet temperature (around 72 °C from 52 °C). This process decreases the COP of the refrigerator from 4.25 to 2.9 at the on-load time from the no-load time. In short, the COP of the refrigerator is proportionally decreased when the load of the refrigerator is increased.

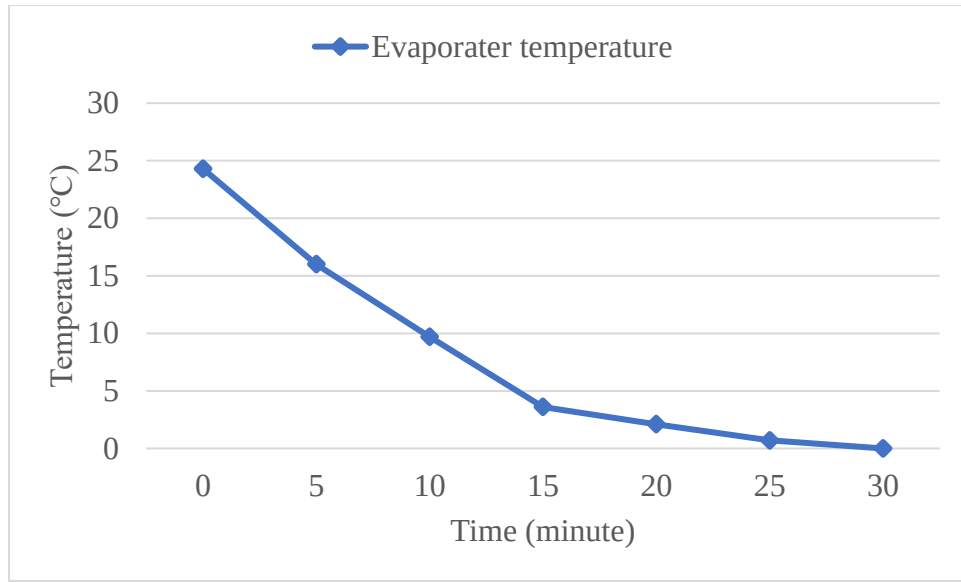


Figure 5.1: Evaporator middle temperature starting from atmospheric temperature

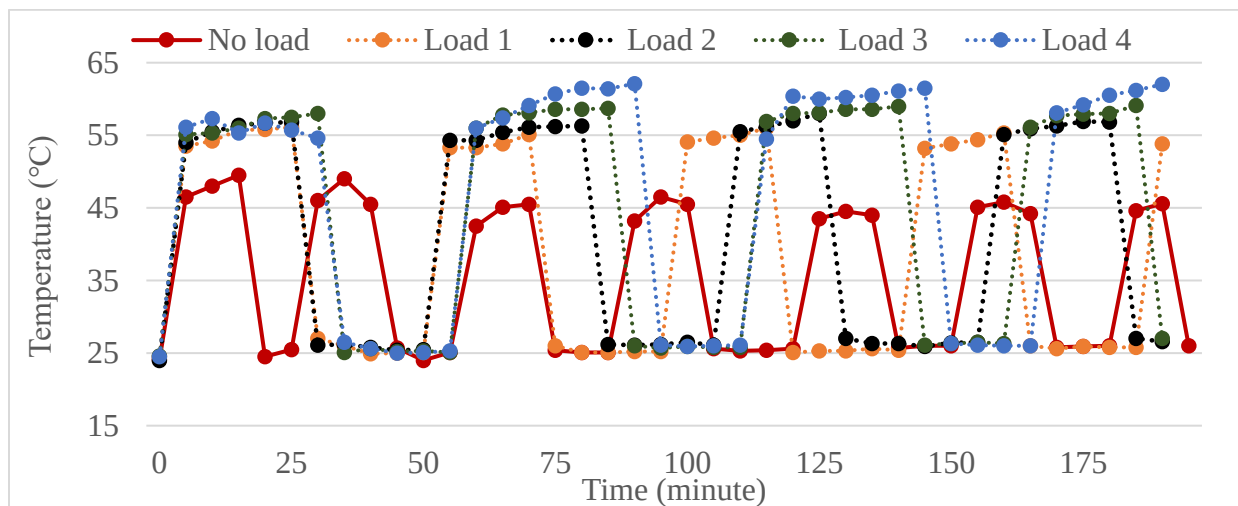


Figure 5.2: Middle temperature of condenser for on-load and no-load conditions.

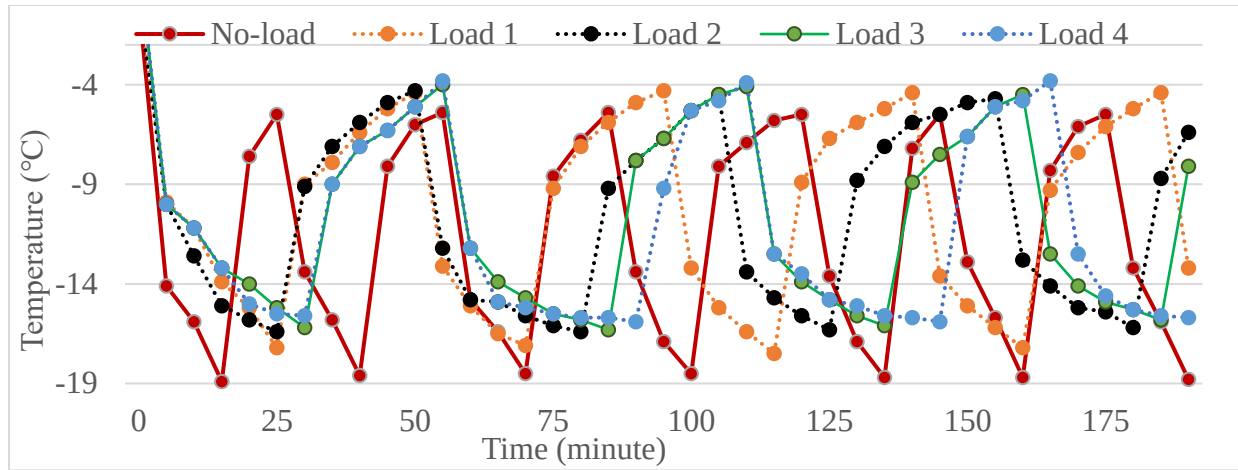


Figure 5.3: Inlet temperature of evaporator for on-load and no-load conditions

Increasing the load on the refrigerator make the refrigerant completely evaporate and become superheated at the compressor inlet this causes to increase the condensing temperature as well as pressure as illustrated in Figure 5.4.

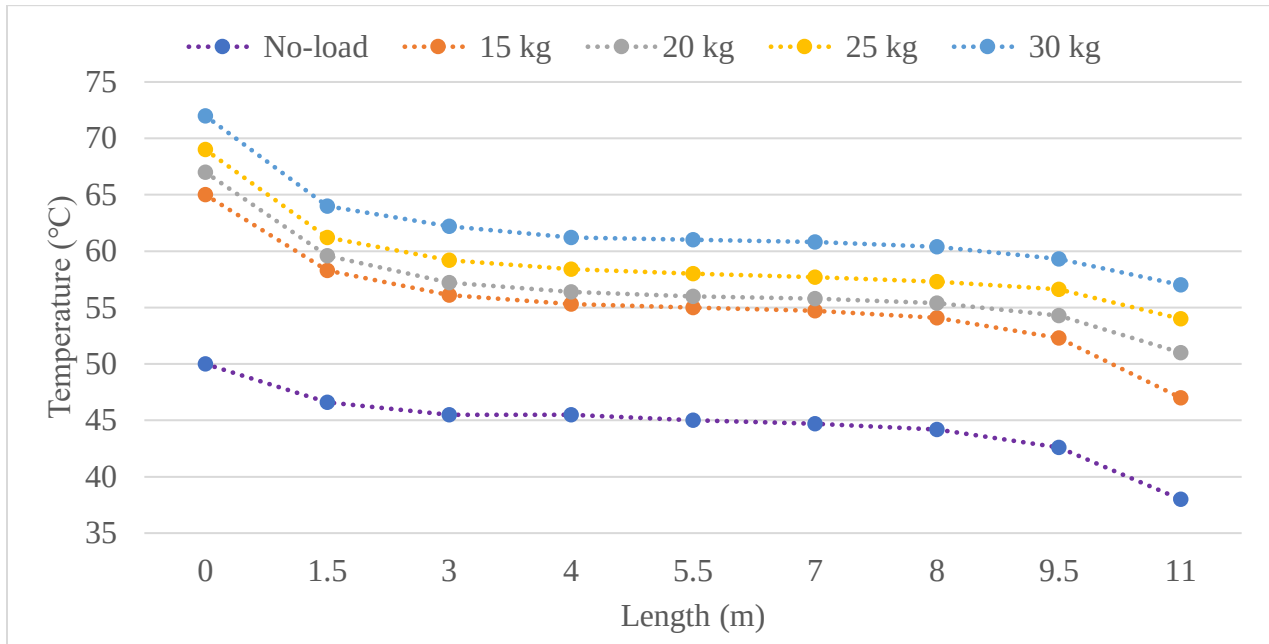


Figure 5.4: Condenser temperature distribution

As listed in Table 5.1 when on the no-load test frequent on-off of the compressor is observed but as the load increases the on-off frequency is decreased. The average on-off ratio of the refrigerator is tending to increase when the load of the refrigerator increases this case an increase power consumption because of longer compressor on time and lower COP. The COP of the refrigeration system when the refrigerator starts from the equilibrium position from the ambient is higher than

the refrigerator operating at the stable operating condition. When the evaporator temperature increases the COP of the refrigerator increase due to the decrease in the solid fraction and the suction pressure of the refrigerator.

Table 5.1: On-off ratio of the refrigerator COP of the refrigerator

Load	Total number of cycles in three hours	On-off ratio	COP_{ca}	COP_{ac}
No-load	7	1	4.25	0
Lod 1 (15 kg of water)	4.25	0.8	3.4	1.967264586
Load 2 (20 kg of water)	3.95	1.25	3.27	2.06214864
Load 3 (25 kg of water)	3.75	1.2	3.051	2.2939155
Load 4 (30 kg of water)	3.5	1.2	2.926	2.31159

Figure 5.5 illustrates the relation between COP_{ca} and COP_{ac} with amount of load in the refrigerator. Since there is no load in the no-load test the COP_{ac} is zero. The cooling effect of the evaporator is wasted only by extracting heat from the refrigerator compartment well, air, and recovering the heat lost because of imperfect insulation. When the refrigerator is load increase the amount of heat removed from the water increased while the amount of heat wasted by the cabinet and insulation is remain nearly constant.

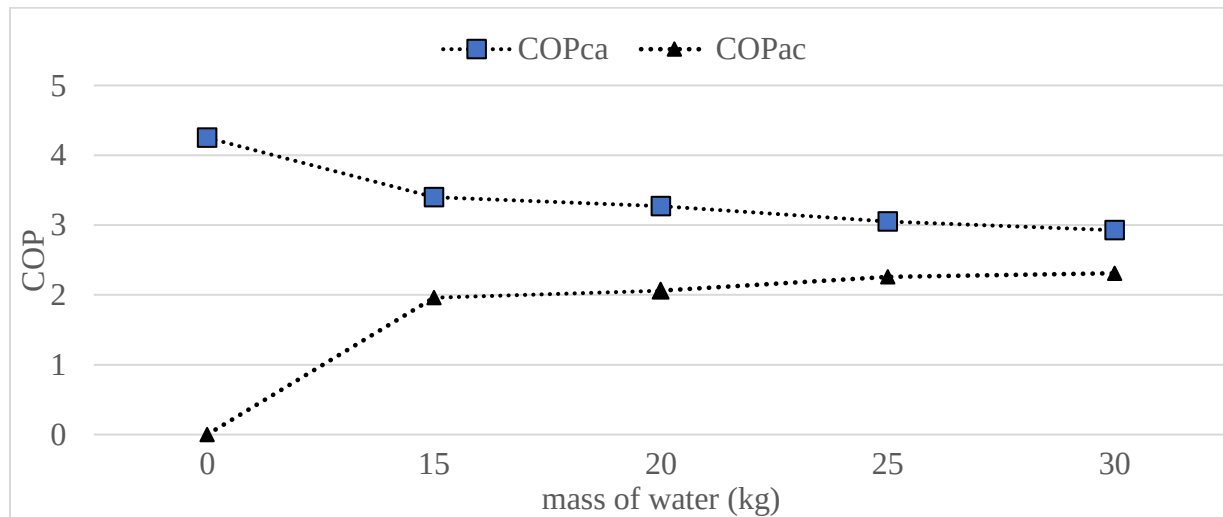


Figure 5.5: The relation between COP_{ca} and COP_{ac} with amount of load in the refrigerator

The test done on the increase in the load of the refrigerator shows similar result with Cerri *et al.*, 2003 which get 5-25% decrease with increasing load.

5.3 PCM on the Condenser

The main objective of using PCM based condenser is to increase the heat transfer rate and elongating the condenser cooling process in the compressor off time. In order to achieve this appropriate type of PCM with suitable amount should be used. In this chapter tests are done to investigate the suitable phase change temperature and thickness of PCM for better performance. All tests are done by loading the refrigerator with 30 kg water of 20 °C.

These tests are done in two categories:

- i. By comparing paraffin wax, hydrated salt PCMs, and non- PCM
- ii. By comparing 2 mm, 3 mm, 4 mm, and 5 mm thickness paraffin wax.

For the first test 0.5 kg of paraffin wax and hydrated salt is used. For the next tests the amount of PCM increases with the increase in the thickness of the PCM by 0.25kg associated with each thickness.

5.3.1 Performance Comparison of PCM Based on the Phase Change Temperature

Figure 5.6 and 5.7 shows the comparison between PCM integrated condenser and normal condenser evaporator inlet and outlet temperature respectively. For both PCMs integrated at the condenser refrigerator, the evaporating temperature reaches a minimum value (around - 15 °C) after the compressor starts, when the compressor stops the evaporating temperature increase sharply and approach to (-4 °C) the evaporating temperature of the refrigerator with PCM at the condenser varied in the smaller range but it has shorter cycle time which helps maintain the temperature stability of the refrigerator.

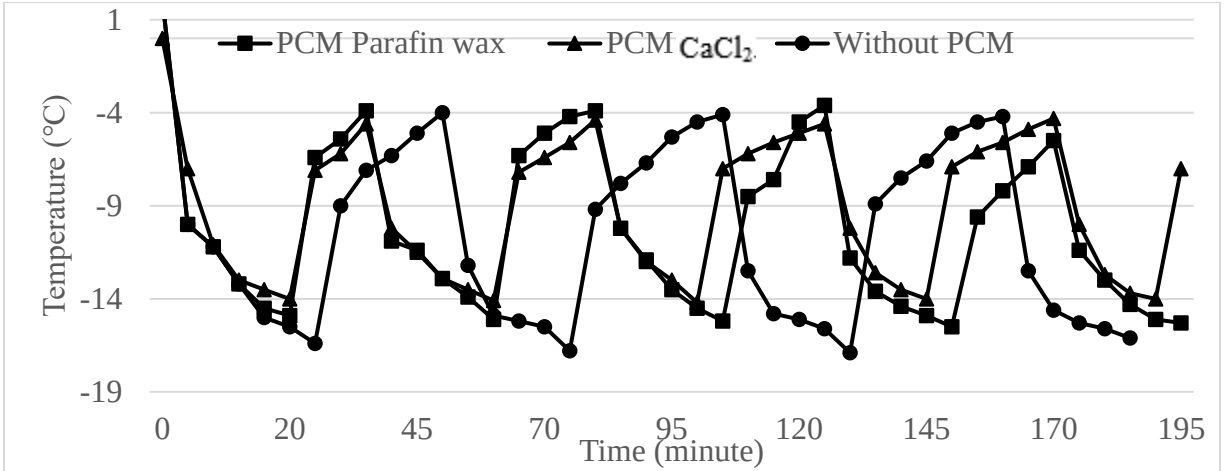


Figure 5.6: Evaporator inlet temperature comparison for different phase change temperature

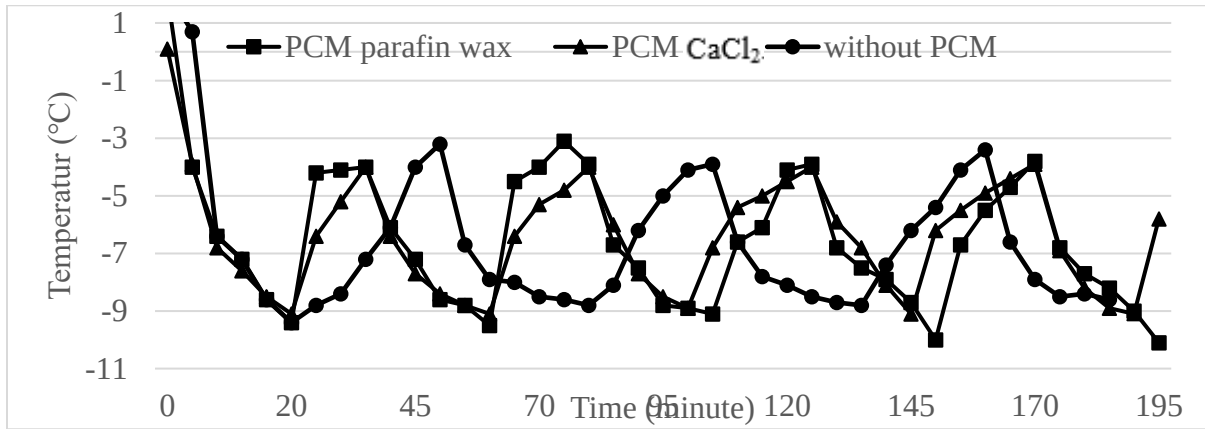


Figure 5.7: Evaporator outlet temperature comparison of the condenser with PCM $\text{CaCl}_2 \cdot 6\text{H}_2\text{O}$ and paraffin wax with and without PCM.

Figure 5.8 and 5.9 shows the comparison of condenser inlet and outlet temperature between PCM integrated condensers and normal condensers, respectively. For the normal condenser outlet and inlet temperature rise rapidly to the highest temperature (about 70 °C and 55 °C) at the start and maintain constant value around that after the compressor starts. When the compressor is off, the condenser inlet and outlet temperature decrease sharply (about 26 °C) which is near to the ambient temperature and then maintain it until the compressor starts again. CaCl_2 PCM shows a lesser compressor outlet than the normal one. The condenser inlet and outlet temperature of CaCl_2 rise slowly to the highest temperature (around 65 °C and 52 °C respectively) at the start and it maintains constant temperature around that. This gain 24% improvement in the COP from non PCM condenser. For paraffin wax PCM the condenser inlet and outlet temperature rise rapidly to its highest temperature (about 60 °C and 50 °C respectively) and maintain it until the compressor

stops. the temperature difference between ambient temperature and the phase change temperature of paraffin wax increase the heat transfer from the PCM during phase change process. This helps to rapidly decrease the PCM temperature during off cycle. Because of the above reason, the COP of paraffin wax integrated refrigerator increase up to 34%.

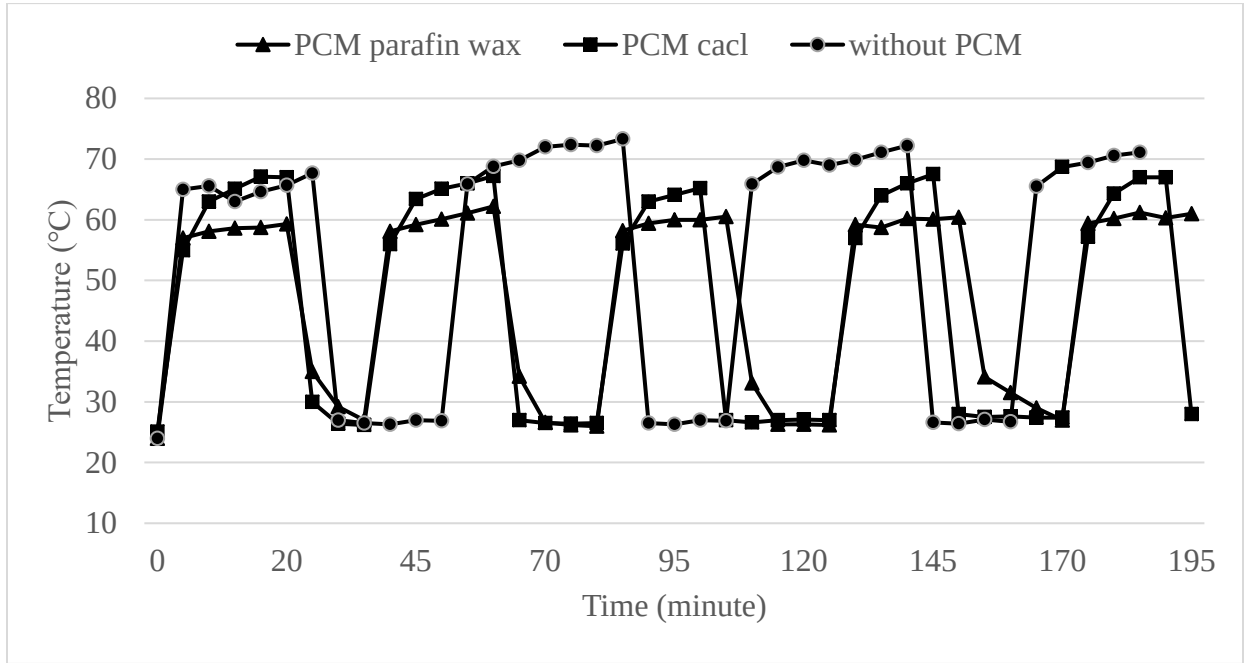


Figure 5.8: Condenser inlet temperature comparison of the condenser with PCM $\text{CaCl}_2 \cdot 6\text{H}_2\text{O}$ and paraffin wax with without PCM.

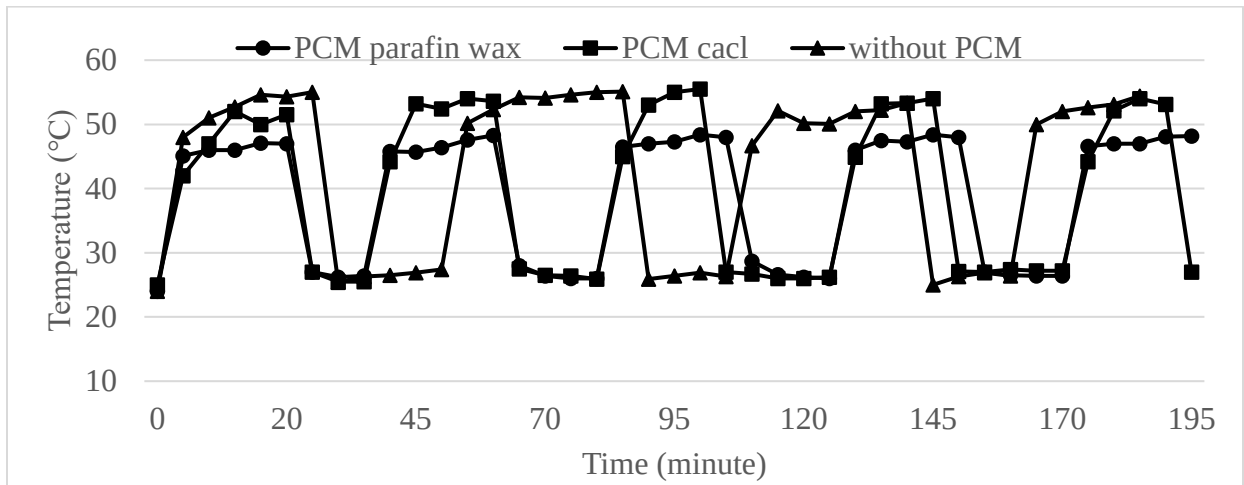


Figure 5.9: Condenser outlet temperature comparison of the condenser with PCM $\text{CaCl}_2 \cdot 6\text{H}_2\text{O}$ and paraffin wax with without PCM.

The other difference between the three cases is the running and stopping time of the compressor as well as the on-off ratio of the compressor. As listed in Table 5.2 the on-off ratio of the compressor lower during PCM integrated tests which leads to less power conception of the compressor.

Figure 5.10 describes the effect of the PCMs in the condensing pressure. The PCM causes the refrigerant to subcooled during condensation this reduce the condensing pressure. Because of the decrease in the condensation pressure and temperature compressor work decreases.

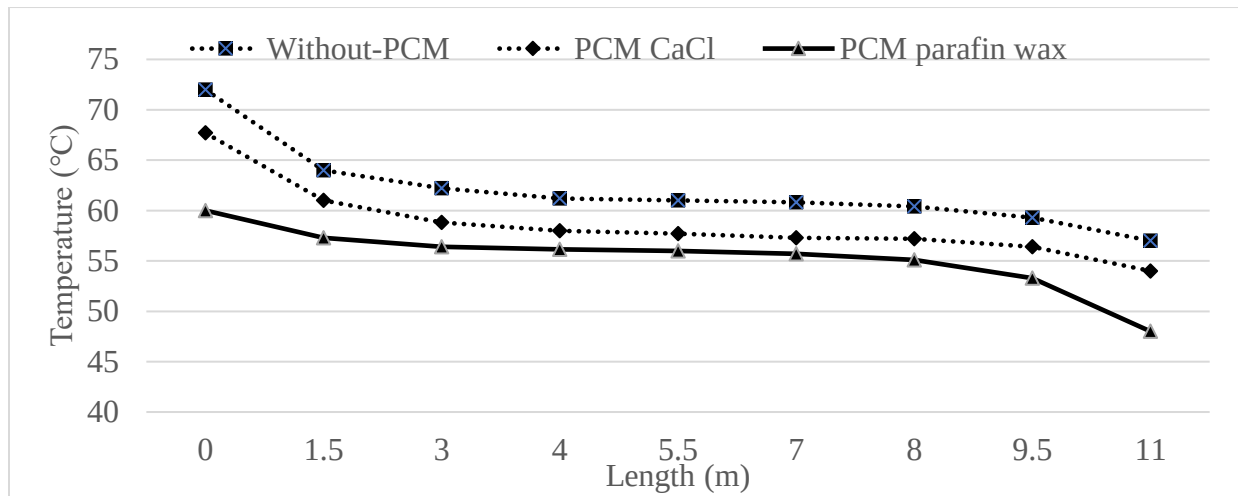


Figure 5.10: Condenser temperature distribution

Since all tests are done with the same loading condition of 30 kg of water their deferencee in COP_{ac} depend on the effect of PCM used in the condenser. As discussed in Table 5.2 the working time of the compressor decreases and as shown in Figure 5.8 and 5.9 the working temperature of condenser decrease relatively from the non-PCM condenser this reduce the compressor power consumption resulted the increase in the COP_{ac} of the refrigerator.

Table 5.2: Performance discription of different phase change temperature

Type	On-off ratio	No of cycle in three hours	COP_{ca}	COP_{ac}
PCM $CaCl_2$	0.667	4.5	3.65	2.96
PCM paraffin wax	0.71	5	3.907692	3.12
Without PCM	1.2	3.5	2.9	2.31

5.3.2 Comparisons of Condenser PCM With Different Thickness

The test done on the paraffin wax shows higher improvement in the COP and power consumption of the refrigerator. So further tests are done on the thickness-based performance of the refrigeration. The tests are done on the 2 mm, 3 mm, 4 mm, and 5 mm thickness of PCM. Figure 5.11 and 5.12 illustrates the effect of thickness of the PCM on the condenser outlet and inlet temperature respectively. the increase of the thickness of the PCM gives good result at the beginning because of the high thermal inertia gained due to the mass of the PCM but in the next cycles it causes decrease in the COP which caused by the increase in the condenser temperature. The main advantage of inserting PCM in the condenser is extending the cooling process to the compressor of time. When the thickness increases the time taken to release the heat stored in the PCM is increased as well. The compressor off time is not enough to release the heat of PCM to make its temperature close to the ambient temperature.

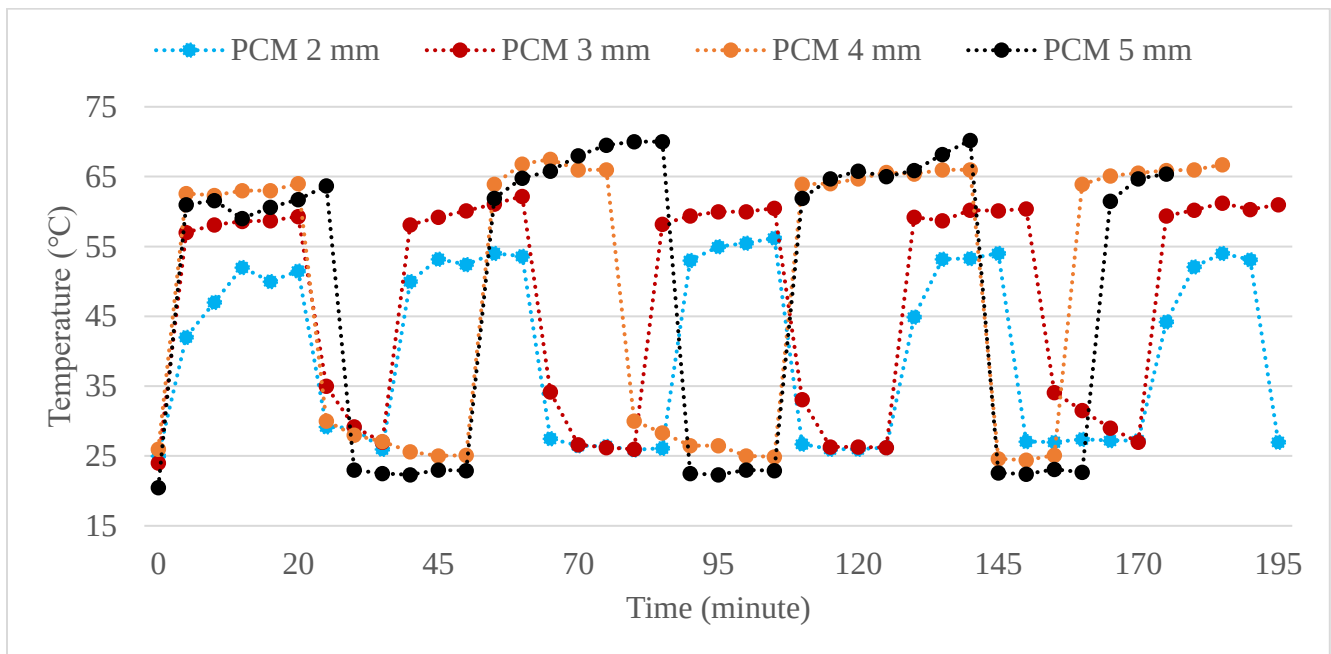


Figure 5.11: Condenser inlet temperature of different PCM thickness.

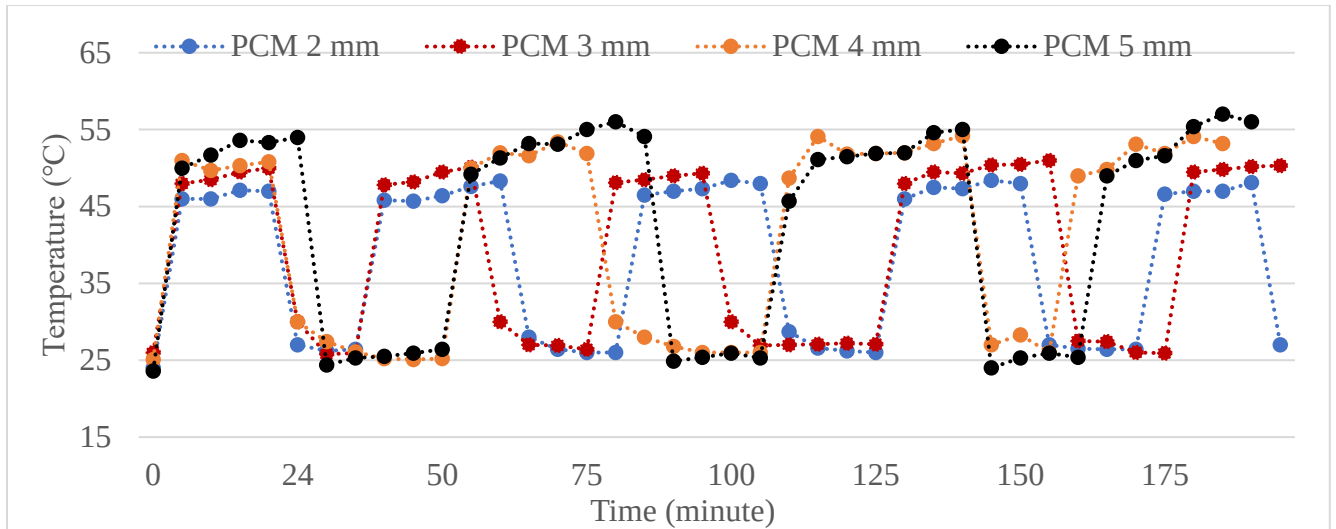


Figure 5.12: Condenser outlet temperature of condenser at different PCM thickness.

Figure 5.13 discuss the effect of thickness of the PCM on the condensing temperature. the increase in the thickness of the PCM allows the PCM to store more heat this decreases the condensing pressure at the start but since it stores much energy and the compressor on-off time is not enough to release the heat stored by the PCM the condensing temperature increases.

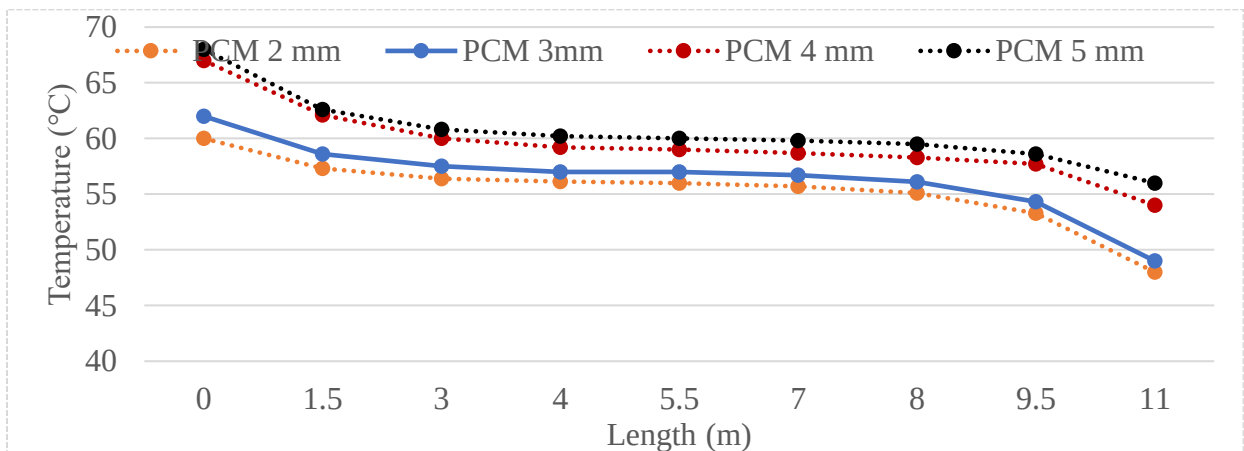


Figure 5.13: Condenser temperature distribution over the length

Figure 5.14 and 5.15 shows effect of condenser temperature on the evaporator outlet and inlet temperatures

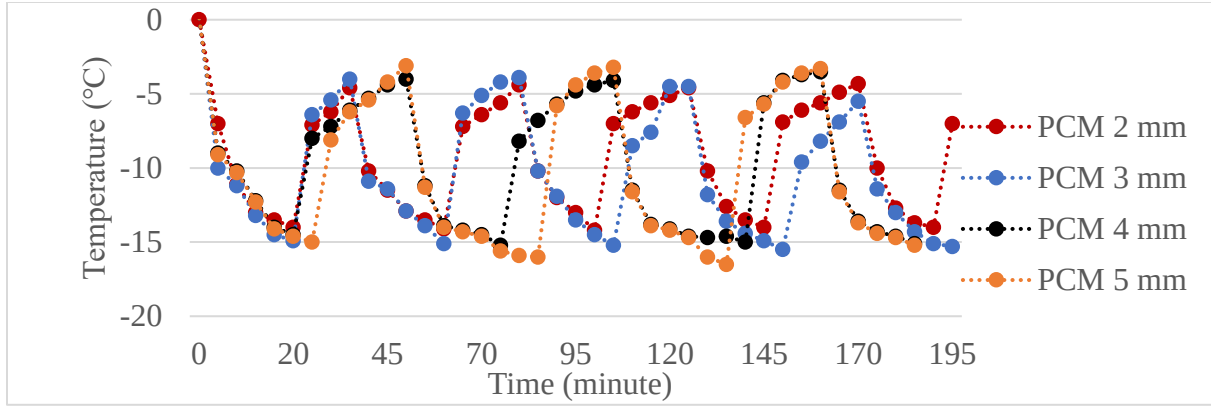


Figure 5.14: Outlet temperature of Evaporator at different PCM thickness integrated at the condenser.

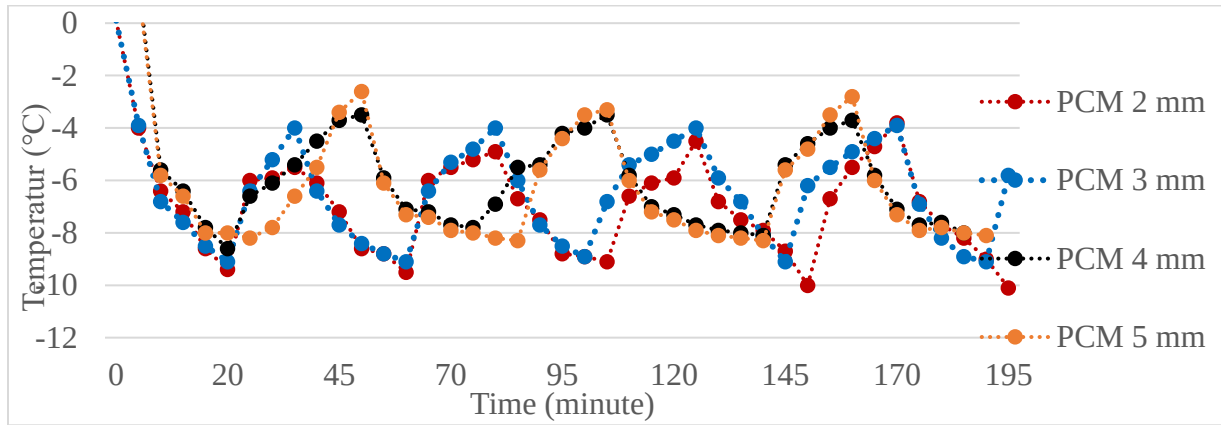


Figure 5.15: Evaporator inlet temperature for different PCM thickness at the condenser.

Table 5.3 discuss the on-off ratio, frequency, and COP comparisons. Increasing the thickness of the PCM gives low frequency of on-off cycle as advantage but it increases compressor on time and decreases COP as disadvantage.

Table 5.3: Performance comparison of different thickness of condenser PCM.

Type	Number of cycles in three hours	On-off ratio	COP	
PCM 2 mm	4.5	0.834	3.907692308	3.12
PCM 3 mm	4.75	0.8	3.68662614	3.101
PCM 4 mm	3.5	1	3.507954545	2.51
PCM 5 mm	3.5	1.2	3.232804233	2.35
Non-PCM	3.5	1.2	2.9	2.31

As shown in Figure 5.15 increasing the thickness of the PCM from 2 mm to 3 mm do not have the same COP_{ca} and COP_{ac} trend with increasing the thickness from 3 mm to 4mm and 5 mm this is because increasing the thickness from 2 mm to 3 mm reduce the compressor working time as shown in Table 5.3 on-off ratio unlike the others.

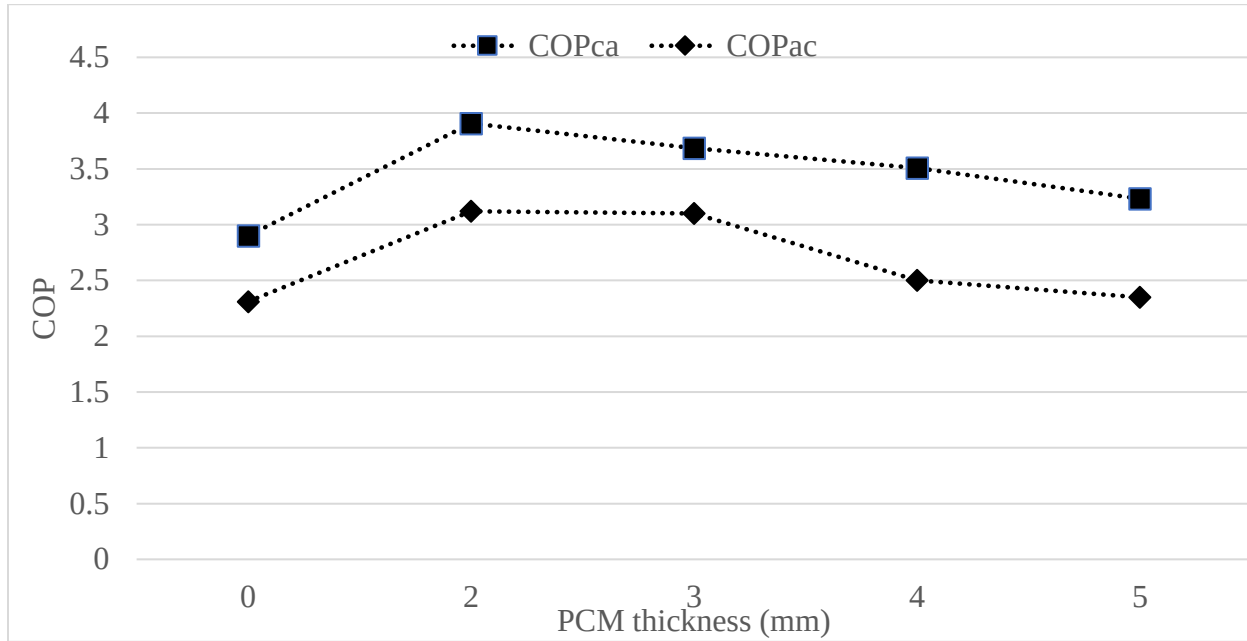


Figure 5.16: The comparison between COP_{ca} and COP_{ac} with the thickness of PCM in the condenser

5.4 PCM Integrated Evaporator

The main advantage of integrating PCM in the evaporator is to increase heat transfer from the evaporator to refrigerated cabinet and to decrease the temperature fluctuation of refrigerated cabinet in case of power loss. This phenomenon has affected by thickness, of PCM types of PCM, phase change temperature of PCM, amount of PCM heat of fusion of PCM, specific heat of PCM, and heat transfer property of PCM. So, in the thesis tests are done on the effect of phase change temperature, amount of PCM on the temperature fluctuation and performance of the refrigerator. All tests were done by loading the refrigerator with 30 kg of water. These testes are done in four categories these are:

- PCM with different phase change temperature on the performance of the refrigerator.
- PCM with different amount on the performance of the refrigerator.
- Effect of PCM phase change temperature on the temperature fluctuation of the refrigerator.
- Effect of different amount of PCM on the temperature fluctuation of the refrigeration.

5.4.1 Comparison of PCM With Different Phase Change Temperature

Figures 5.17 to 5.20, describes the comparison of evaporator and condenser temperature between PCM integrated condenser refrigerator and normal refrigerator. For the PCM 1 (5% NaCl₂) integrated evaporator, the inlet and outlet temperature of the evaporator ranges from -16 °C to -4 °C and the condenser inlet-outlet temperature range from 70°C to 55 °C the temperature increases in both evaporator and condenser section of the refrigerator. The test on PCM 2 (10% NaCl₂) gives approximately higher evaporator inlet and outlet temperature than PCM 1. This cause the enthalpy difference between the inlet and the outlet of the evaporator increases the compressor inlet specific volume increases thus the refrigeration capacity increases and the refrigeration efficiency are improved, which can be explained that a part of high-temperature refrigerant liquid from the condenser will flash to steam after throttling, while the rest of the liquid temperature decreases, thus the refrigerant which enters the evaporator is the gas-liquid mixture. evaporator heat transfer is mainly caused by the liquid portion Increasing the evaporating pressure can reduce the differential pressure after throttling so flash steam also reduce after the throttle the number of amount increases there for cooling capacity is increased under the condition of the constant load, the increases of the evaporating can lead to the high and low-pressure differential, consequentially the compressor power is reduced.

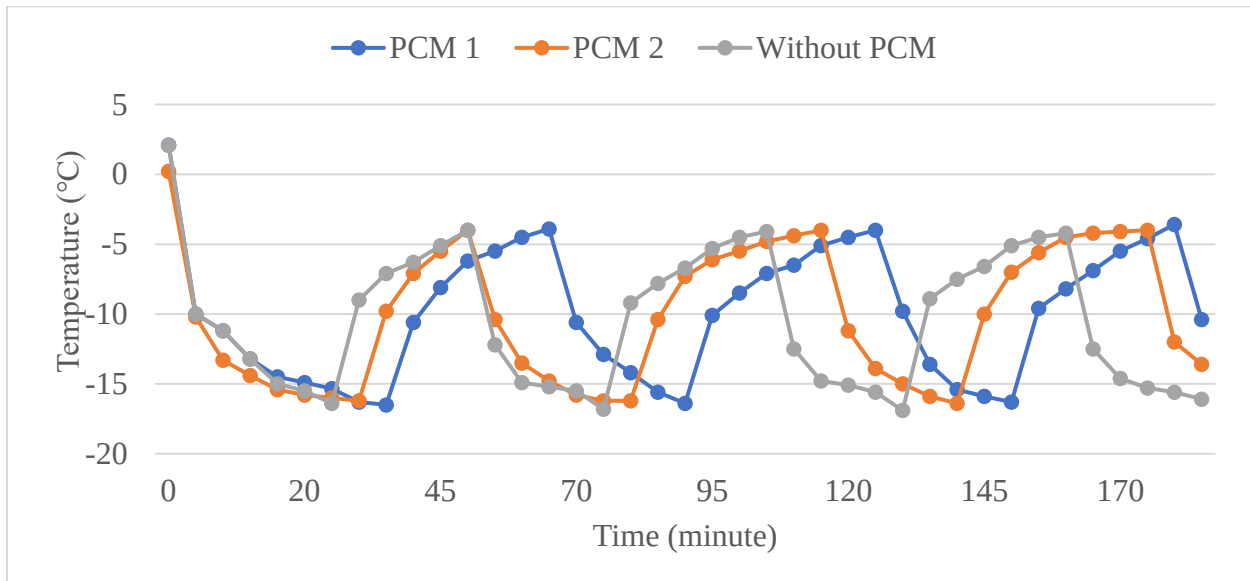


Figure 5.17: Evaporator inlet temperature comparison of the condenser with PCM 1, PCM 2, and without PCM.

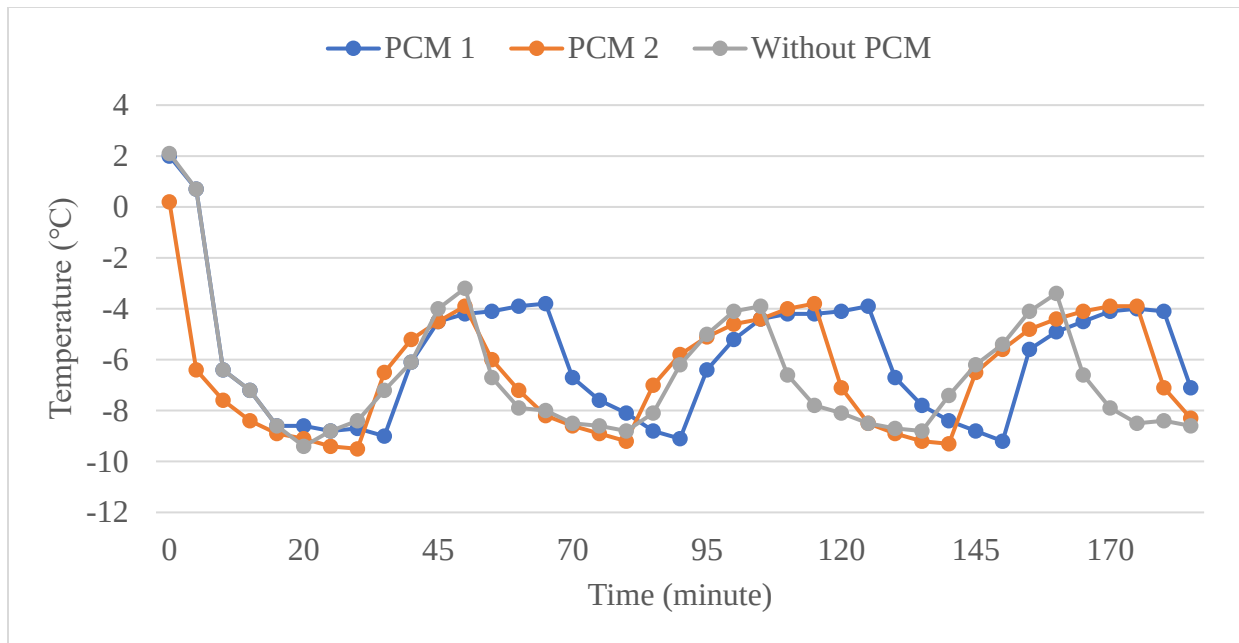


Figure 5.18: Evaporator outlet temperature comparison of the condenser with PCM $\text{NaCl}_2 \cdot 6\text{H}_2\text{O}$, $\text{NaCl}_2 \cdot 6\text{H}_2\text{O}$, and without PCM.

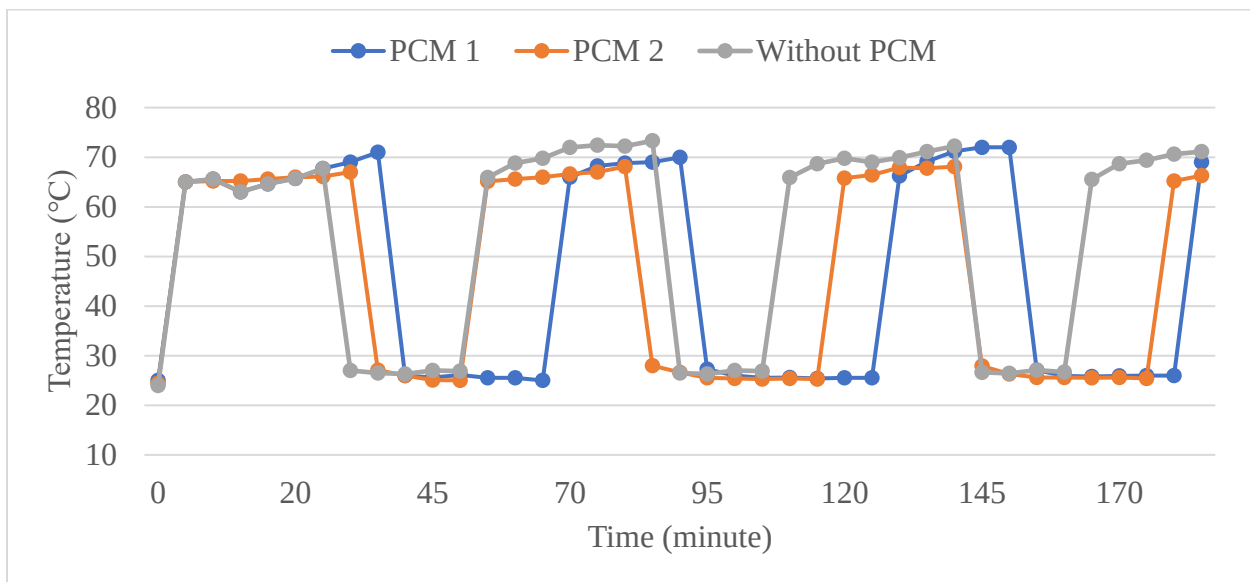


Figure 5.19: Condenser inlet temperature comparison of the condenser with PCM 1, PCM 2, and without PCM.

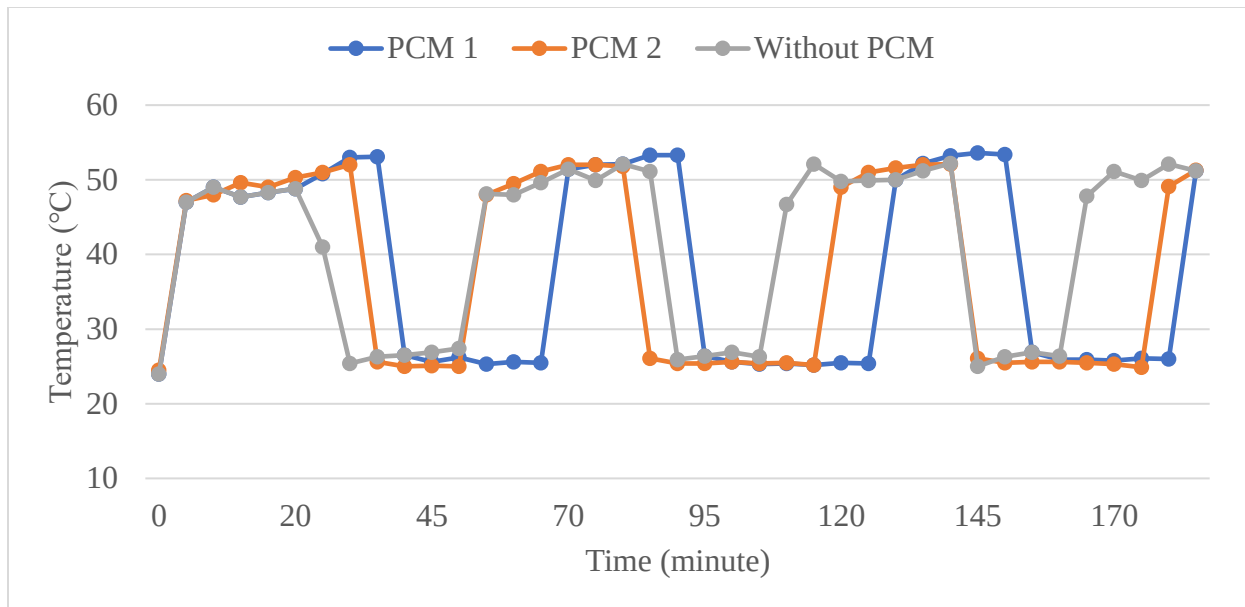


Figure 5.20: Condenser outlet temperature comparison of the condenser with PCM 1, PCM 2, and without PCM.

As shown in Figure 5.21, the condensing pressure of the evaporator with-PCM is lower. Initially the PCM act as load on the refrigerator but once it cools it helps to maintain the lower compartment temperature. Consequently, the condenser pressure and temperature are decrease.

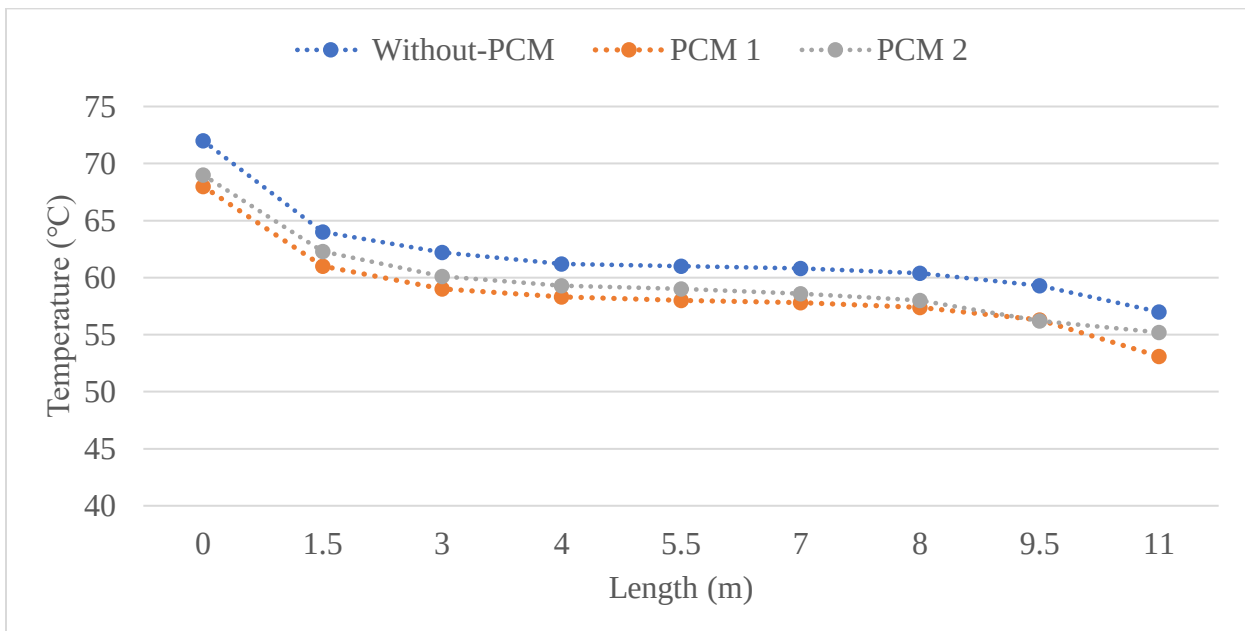


Figure 5.21: Condenser temperature distribution over length with and without PCM

Table 5.4: Performance comparison of different phase change temperature PCM in the evaporator

Type	On-off ratio	Number of cycles in three hours	COP_{ca}	COP_{ac}
PCM 1	0.667	3.25	3.166667	2.57
PCM 2	0.833	3.25	3.089430894	2.45
Without PCM	1.2	3.5	2.9	2.31

In these tests the main contributor to the increase of the COP_{ac} is the decrease in the compressor running time caused by the PCMs.

5.4.2 Effect of Evaporator PCM in the Temperature Fluctuation of the Refrigerated Space.

In this section the effect of PCM on the temperature fluctuation of the refrigerated cabinet have been tested. This test is done in two categories.

- I. By changing the amount of PCM.
- II. By changing the phase change temperature of the PCM.

For each case, the test is started after the PCM is completely solidified and subcooled to -9 °C.

5.4.2.1 Temperature Fluctuation Comparison at Different PCM Amount

Figure 5.22 describes the temperature fluctuation of the compartment of the refrigerator with different PCM amounts of 5% NaCl₂ solution prepared with 2kg, 4kg, and 6kg as pre-sized in chapter three. for each case, the test is started after the PCM is completely solidified and reach the subcooling temperature of -9 °C. As the result shows in the Figure 5.22 the temperature of the compartment remains stable around 0 °C for all PCMs unlike in the non-PCM test in which the temperature rises to 18 °C. For 2kg, 4 kg, and 6kg the maximum temperature rises during three-hour power loss was 1.8°C, 1.1°C, and 0.6 °C respectively. Increasing the amount of PCM gains a decrease in temperature rise and temperature fluctuation. This means that the amount of PCM can determine the rate of temperature change in the refrigerator. To maintain the compartment temperature for a longer time increasing the amount of PCM is necessary.

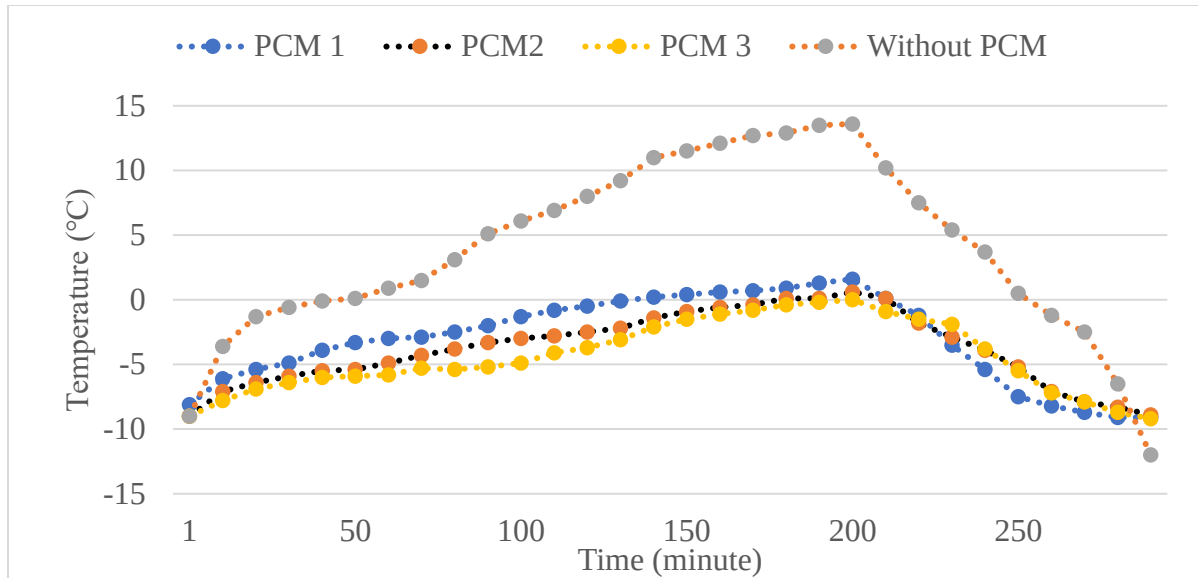


Figure 5.22: Comparisons of compartment temperature fluctuation for 5% NaCl₂ & 95%water solution of total mass 2 kg, 4 kg, 6 kg, and without PCM.

5.4.2.2 Effect of PCM Phase Change Temperature on the Temperature Fluctuation

The test is done with -3 °C (PCM 1), -5°C (PCM 2) and -7 °C (PCM 3) phase change temperature which results from 2kg NaCl₂ of 5, 7.5, and 10% concentration, respectively. As discussed in the Figure 5.22 the inside temperature is maintained around 0 °C for -3 °C PCT but when the PCT decreases to the negative the inside temperature becomes more stable around -2 °C and -3 °C for -5 °C and -7 °C of PCT respectively. This shows that the phase change temperature of the PCM is selected based on the desired temperature for the products in case of cold drinks and foods the temperature of the is desired to be 5 up to 10 °C which is necessary to choose a phase change temperature around that. Similar results are obtained with researches done by **Oró *et al.* (2012)** and **Azzouz *et al.* (20012)** which also achieved similar result 4-5 hours and 6-5 hours longer cooling by using PCM on the evaporator.

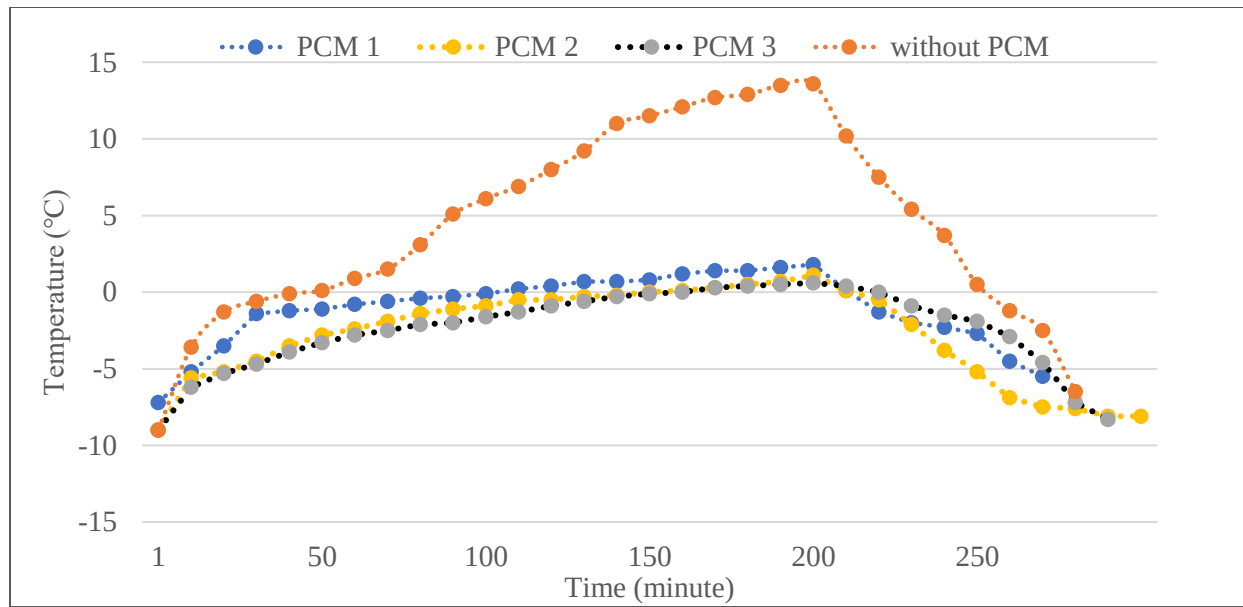


Figure 5.23: Comparisons of compartment temperature fluctuation for different PCM amount.

5.5 PCM Placed in Both Condenser and Evaporator

As discussed in the above section the PCM placed at the condenser gives less power consumption because of a significant increase of COP and lower on-off ratio. But it also causes a higher frequency of on-off which will case compressor damage in the long run. On the other side, the PCM placed at the evaporator gives the stable temperature of the compartment, lower on-off frequency, and higher COP as an advantage and increasing the temperature of the condenser which subsequently case less subcooling in the condensation process as a disadvantage.

In this section the combined effect of PCM placed in the evaporator and condenser at the same time are discussed. The combined effect of PCM placed in the evaporator and condenser are done with four combinations of PCM. In the above section the condenser PCM test shows that paraffin wax gives higher COP and lower on-off time so four combinations of PCM is used by changing the PCM in the evaporator while maintaining paraffin wax at the condenser.

- I. PCM 1 paraffin wax with 2 kg of 5% NaCl_2
- II. PCM 2 paraffin wax with 4 kg of 5% NaCl_2
- III. PCM3 paraffin wax with 2 kg of 10% NaCl_2
- IV. PCM 4 paraffin wax with 4 kg of 10% NaCl_2

Figure 5.24 and 5.25 discuss the inlet-outlet evaporator temperature comparison respectively. The evaporating temperature for different PCMs has nearly the same inlet and outlet temperature

(around -17°C and -4°C) which is slightly lower inlet and outlet evaporator temperature than the non- PCM test. temperature and non PCM and each other.

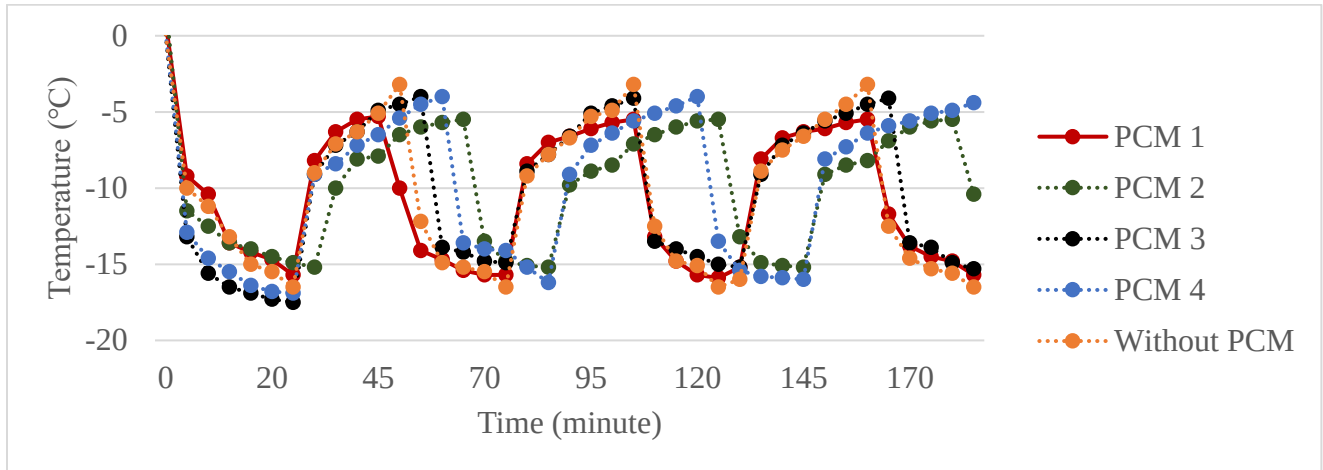


Figure 5.24: Evaporator inlet temperature comparison of PCM integrated with both condenser and evaporator.

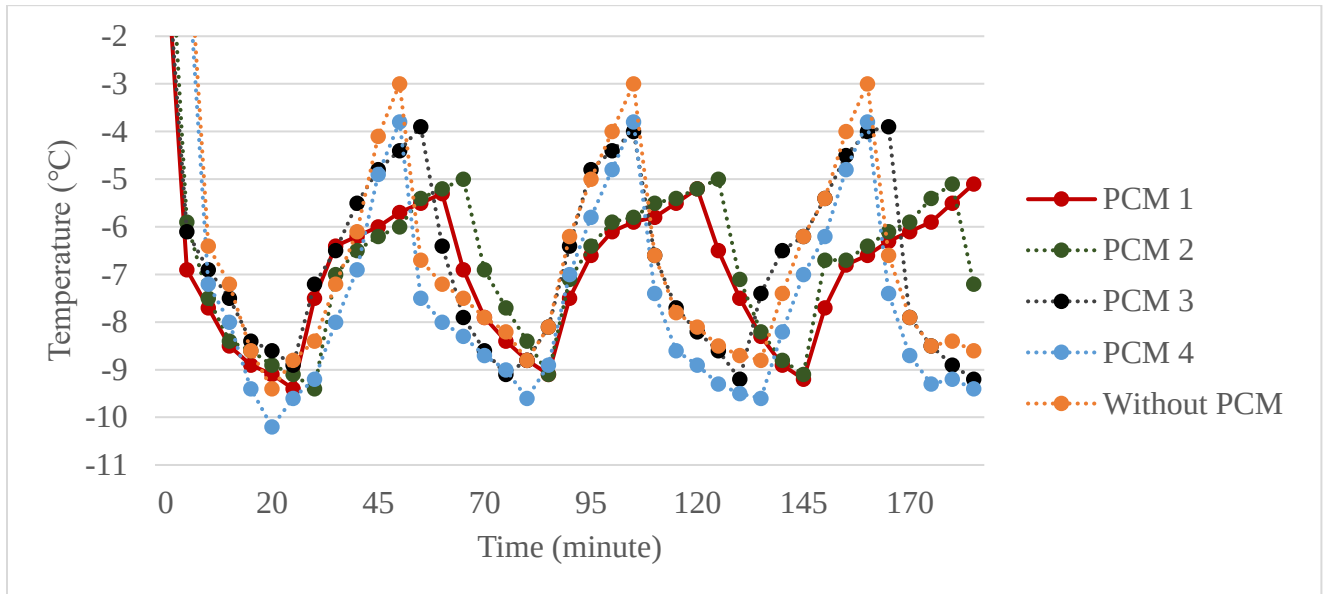


Figure 5.25: Evaporator outlet comparison of PCM integrated with both in the condenser and the evaporator of the refrigerator.

Figure 5.26 and 5.27 illustrates the inlet-outlet temperature comparison. Condenser inlet temperature increase compared to the test of PCM integrated condenser but lower than the non-PCM condenser and PCM integrated at the evaporator. This in turn changes the outlet temperature of the condenser. This is caused by the higher temperature transfer in the evaporator which leads to a rise in the inlet temperature of the compressor.

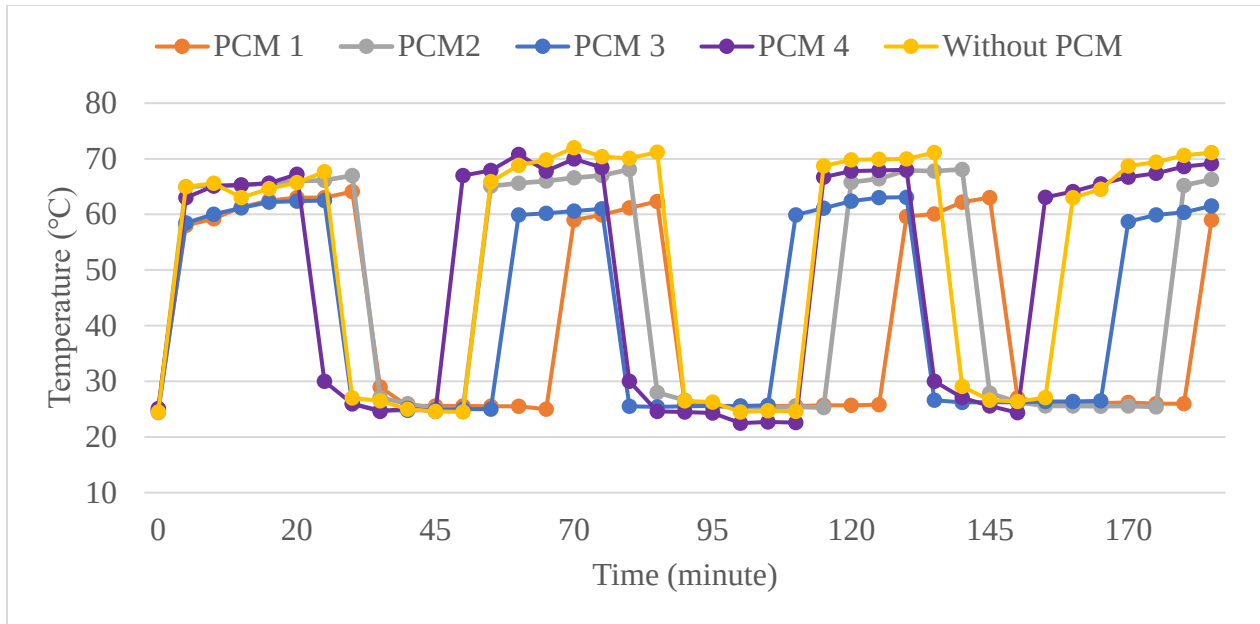


Figure 5.26: Comparison of condenser inlet temperature for PCM integrated both in the condenser and evaporator.

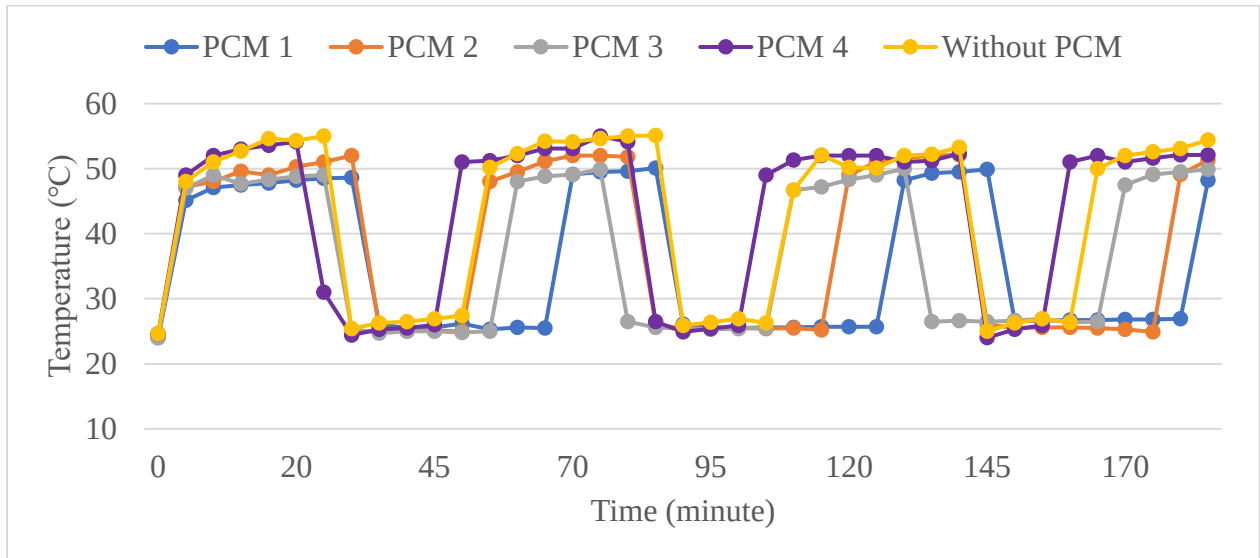


Figure 5.27: Condenser outlet temperature comparison of PCM integrated both evaporator and condenser.

In the PCM integrated condenser test, the main problem was the repeated on-off frequency of the compressor. And the main problem in the PCM integrated evaporator test was the higher condensation pressure. The result in the PCM integrated both in the condenser and evaporator test gives relief for both problems the on-off frequency decreased and the condenser outlet temperature rise decreased. The decrease of the outlet temperature and the increase in the evaporator subcooling

gives a higher COP than both the PCM integrated evaporator and the non-PCM test. The power consumption of the refrigerator is reduced during this test not only because of the higher COP but also by the decrease of the on-off ratio as discussed in the Table 5.5.

Table 5.5: On-off time and ratio comparison.

Type	Number of cycles in three hours	On-off ratio	COP_{ca}	COP_{ac}
PCM 1	3.25	0.67	3.70	3.01
PCM 2	3.25	1	3.41	2.57
PCM 3	3.5	0.84	3.57	2.93
PCM 4	3.5	1.2	3.12	2.41
Non-PCM	3.5	1.2	2.9	2.31

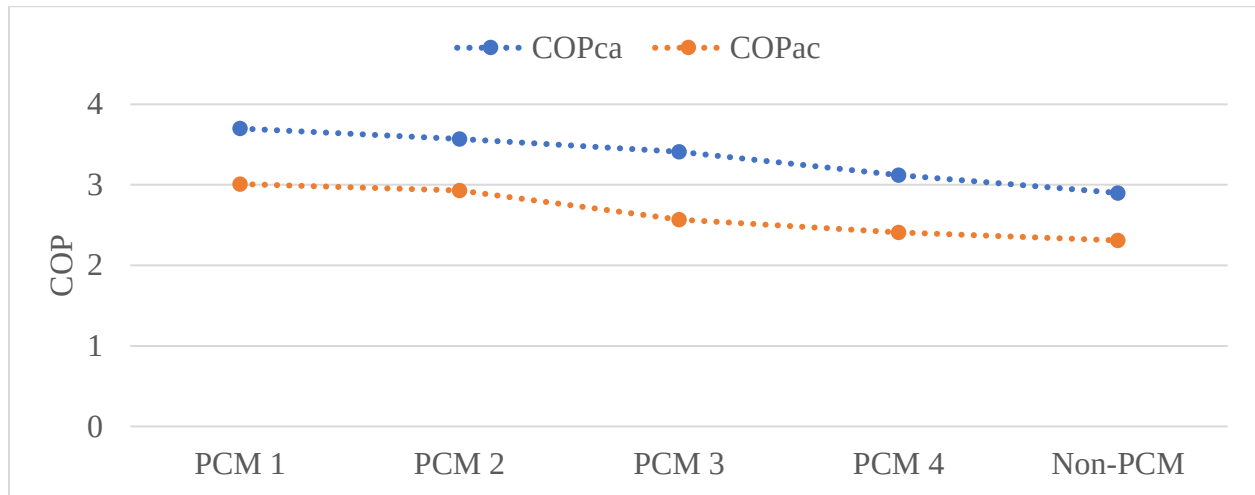


Figure 5.28: COP_{ac} and COP_{ca} effect of PCM placed at the evaporator and condenser
As Figure 5.29 shows the condensing temperature is decreased by the combination of the condenser and evaporator PCM.

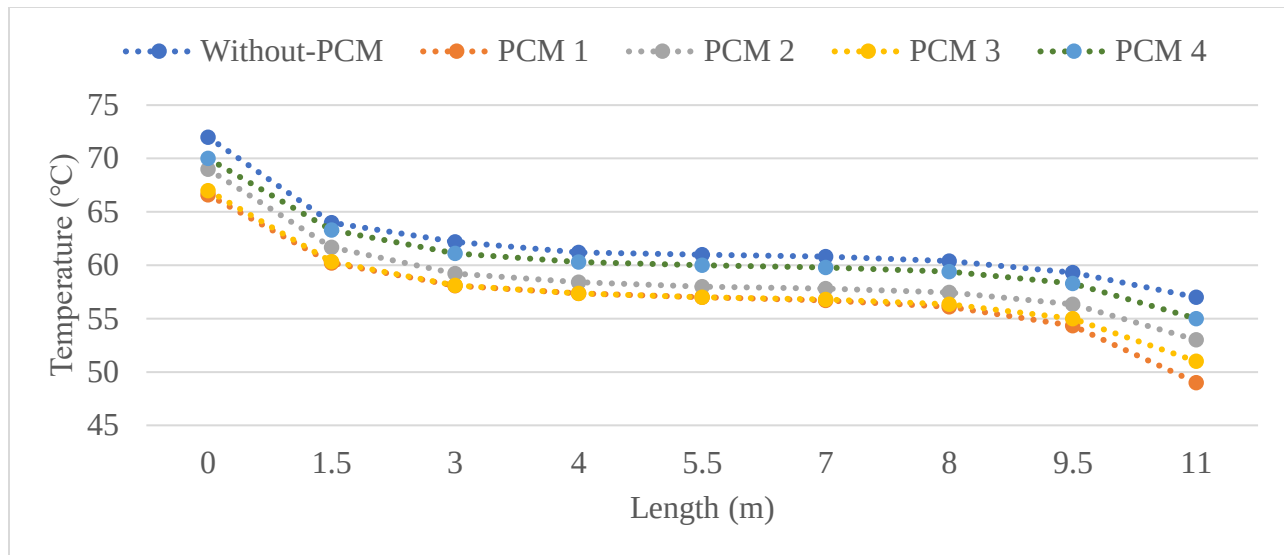


Figure 5.29: Condenser temperature distribution.

Figure 5.30 and 5.31 illustrates the COP and exergy of all tests done on this thesis respectively. The comparison COP increase and decrease are already presented on the above section of this chapters on the figures and tables individually. In this section the total comparisons of COP and exergy-efficiency are presented in figurative form.

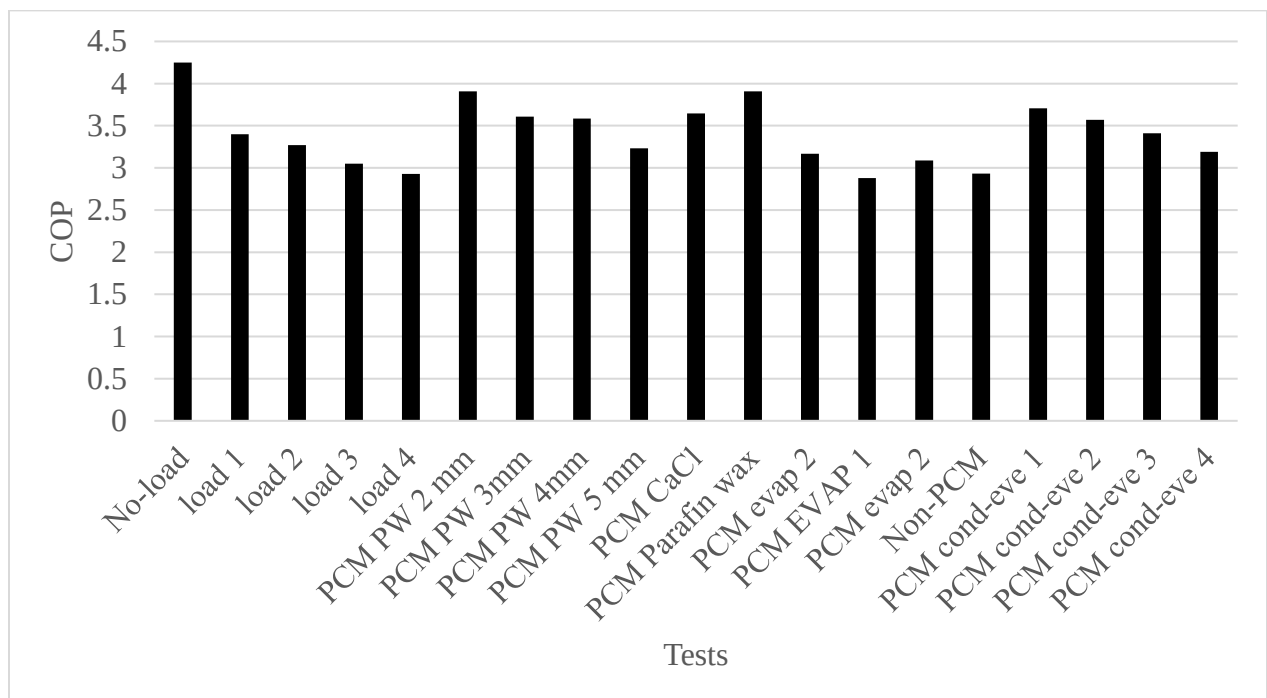


Figure 5.30: Overall revision performance test COP

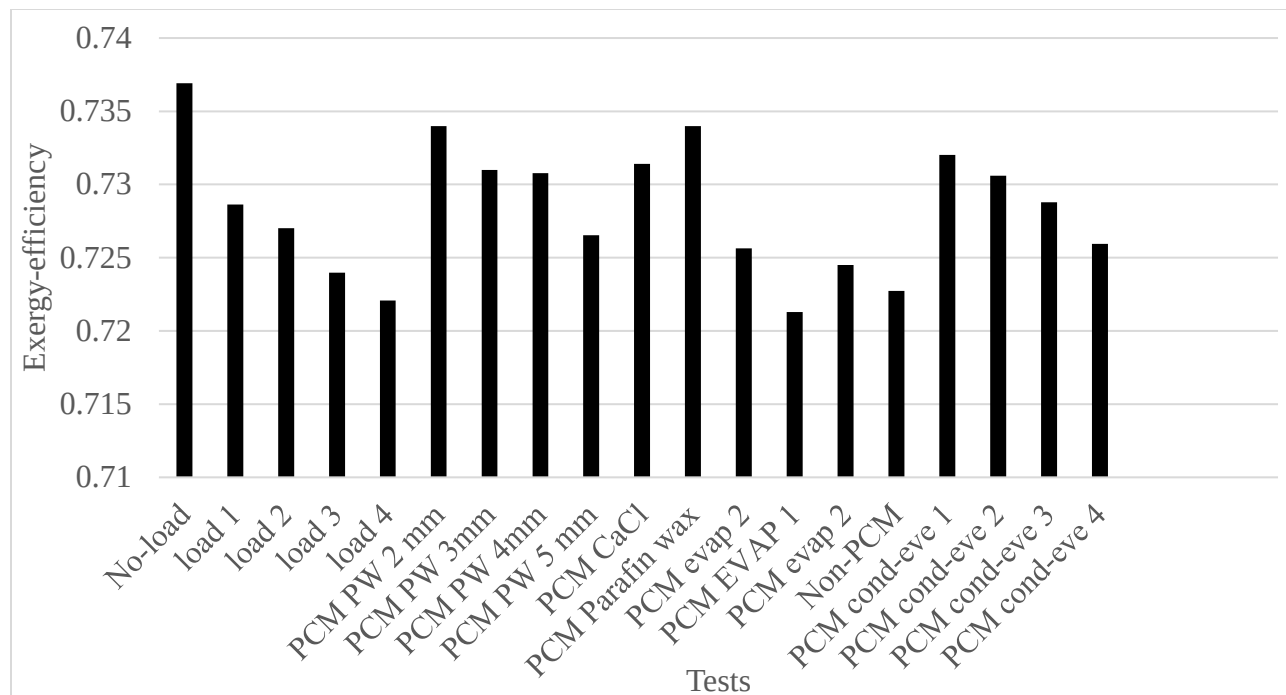


Figure 5.31: Overall revision of performance test exergy-efficiency

5.6 Validation of Experimental result of the constructed prototype with the house hold refrigerator

In order to validate the experimental result, obtain from PCM integrated at the condenser of the prototype the same experiment was conducted on the normal household refrigerator with the capacity of 1 kW and 1/3 hp compressor power. The house hold refrigerator loaded with 30 kg of water and its temperature is measured at the condenser evaporator and the water. The house hold refrigerator integrated its condenser with the 2 mm thickness paraffin wax PCM results the similar result as the experiment conducted on the prototype which is 26 % increase in the COP and on-off ratio and on-off time condenser with.

5.7 Validation of Experimental Result with Literature

Validation of Experimental result of the constructed prototype with the house hold refrigerator

As shown Table 5.2 The PCM integration study of this thesis gives similar result with the study done by Berdja *et al.*, (2018), Azzouz *et al.* (2009), Khan and Afrozn (2013), and Cheng *et al.*, (2016).

Table 5.6: Validation of results with preceding researches

	COP improvement of integrated PCM at evaporator	COP improvement of PCM integrated at condenser	COP improvement of PCM integrated at both condenser and evaporator
Experimental result	2-9%	11-34%	7-27%
Azzouz <i>et al.</i> (2009)	10-30%		
Cheng <i>et al.</i> (2016)	24.7%	18.6%	38.8%
Berdja <i>et al.</i> (2018)	5.5%		
Khan and Afrozn (2013)	2-30%		

CHAPTER SIX

CONCLUSIONS AND RECOMMENDATION

6.1 Conclusions

In this thesis, a vapor compression refrigeration system was designed & constructed which has a cooling capacity of 0.433 kW, 0.186 kW compressor power input and cyclically operating R134a refrigerant working with -10 and 60°C temperature of the condenser and evaporator, respectively in the pipe circuit. After constructing the refrigeration system, refrigerant leakage and pressure tests were done to ensure that the refrigerator is ready for the experiment. Experimental performance investigation of the system was done as designed condition without load and with load of 15, 20, 25, 30 kg of water.

In addition, experimentation on the performance of the refrigerator for 30 kg load was done by integrating PCM on three different locations: on the evaporator, on the condenser, and on the condenser & the evaporator at the same time. The test on integrating the PCM on the condenser was done by varying the thickness of the PCM (from 2 mm, 3 mm, 4 mm, 5 mm) and by integrating two different PCMs namely: $\text{CaCl}_2 \cdot 6\text{H}_2\text{O}$ and paraffin wax which have different phase change temperature. Three different tests were done by integrating a total solution mass of sodium chloride & water solution of 2, 4, and 6 kg on the evaporators. Keeping these three total mass constants the experiment was done two times by dissolving 5 and 10% of NaCl_2 solution in water for identifying the effect of COP, on-off frequency and on off ratio. In addition, for the three total solution mass of sodium chloride & water, one experimentation each, was done by dissolving 5, 7, and 10% NaCl_2 to identify the effect of temperature fluctuation of the evaporator. The test performances on the refrigerators are presented in terms of the inlet-outlet temperature of condenser and evaporator, on-off frequency, COP, and exergy efficiency of the system. The result gained from the loading and unloading tests are used to compare the effect of the PCM on the performance of the refrigerator. From the result of the experimental test, the following points are concluded:

- 1) The experimental test done on the refrigerator as designed without load and with 15, 20, 25, 30 kg load decrease in the calculated COP because of the increase in the load of the compressor but it shows the increase in the actual coefficient of performance of the refrigerator. On-off ratio increases and the decrease in on-off frequency.

- 2) The experimental test done on the PCM integrated, paraffin wax of 2 mm gives 34% of COP increase and 5/6 on-off ratio while the same amount of calcium chloride-based salt hydrant gives a 27% increase in COP 4/5 on-off ratio. The result implicates that increasing the phase change temperature of the PCM increase the COP and on-off ratio of the refrigerator. Both PCMs resulted low on-off ratio and high on-off frequency of the compressor.
- 3) The test performed by changing the thickness of the paraffin wax PCM at 2 mm, 3 mm, 4 mm, and 5 mm shows that increasing the thickness of the PCM more than 2 mm will decrease the COP because of the heat stored in the refrigerator do not have enough time to be released during the compressor off time. The result also shows that increasing the thickness causes an increase in the on-off ratio and decrease in the on-off frequency which can cause increase in power consumption of the compressor.
- 4) 34% and 21% increase in COP was obtained using paraffin wax and $\text{CaCl}_2 \cdot 6\text{H}_2\text{O}$ with 2 mm thickness in the condenser respectively. On-off frequency increases and on-off ratio decreases for both PCMs relative to normal condenser.
- 5) Increasing the thickness of the paraffin wax PCM further from 2 mm causes decrease in the COP, exergy-efficiency and on-off frequency but decrease in the on-off ratio of the compressor.
- 6) Integration of $\text{NaCl}_2 \cdot 10\text{H}_2\text{O}$, $\text{NaCl}_2 \cdot 5\text{H}_2\text{O}$ in the evaporator resulted increase in the COP, on-off ratio, and on-off frequency and stable temperature in the refrigerator cabinet during power loss. Increasing their amount in evaporator increase the stability of the temperature in the time of power loss. The temperature of the refrigerator cabinet air depends on the evaporator PCM phase change temperature.
- 7) PCM placed both in the evaporator and condenser gives higher COP stable cabinet temperature higher on-off frequency and lower on-off ratio.

6.2 Recommendation

Integration of a maximum 2 mm thick paraffin wax on the condenser is recommended since the COP increase, on-off time reduces and on-off frequency increased compared to integration of the paraffin wax of thickness 3 mm, 4 mm and 5 mm.

6.3 Future works

In this thesis performance evaluation of PCM with different phase change temperature and thickness are done to decrease the power conception of the compressor by decreasing the on-off ratio and increasing COP as well as gaining the stable temperature of refrigerator cabinet during power loss. For increasing the performance of the refrigerator further studies based on the enthalpy of fusion of PCM should be done. Further investigation also should be done on the other types of PCM like organic and inorganic eutectic mixture. Heat transfer enhancement of the PCM is one of the main performance parameters of the PCM further studies should be done to increase the heat transfer of the PCM.

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APPENDICES

Appendix A

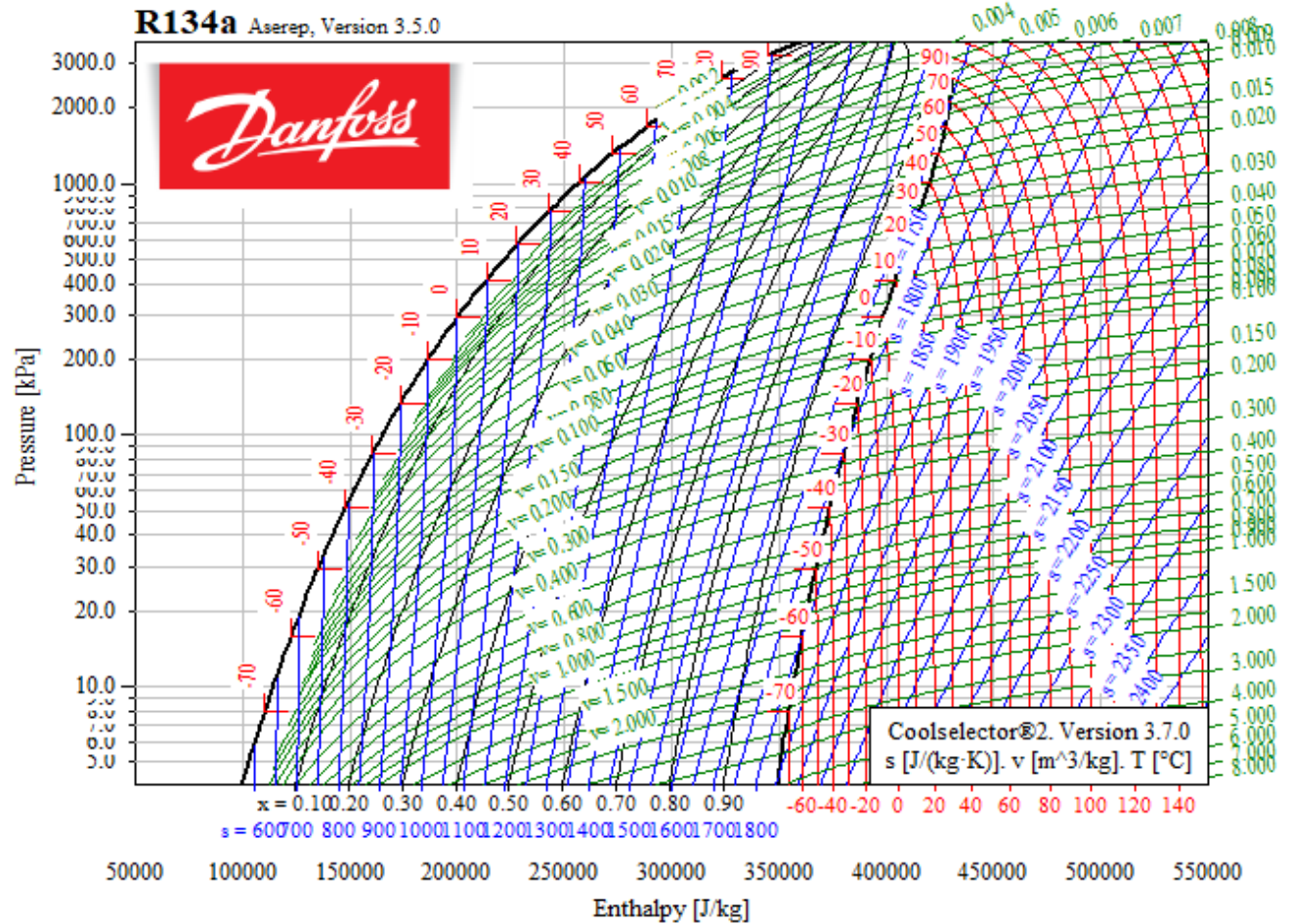


Figure A.6.1: p-h diagram of R134a refrigerant (coolselector, 2016)

Appendix B

Table B.1: Properties of common metals (Arora, 2010)

Composition	Melting Point (K)	Properties at 300 K			
		ρ (kg/m ³)	c_p (J/kg · K)	k (W/m · K)	$\alpha \cdot 10^6$ (m ² /s)
Aluminum					
Pure	933	2702	903	237	97.1
Alloy 2024-T6 (4.5% Cu, 1.5% Mg, 0.6% Mn)	775	2770	875	177	73.0
Alloy 195, Cast (4.5% Cu)		2790	883	168	68.2
Beryllium	1550	1850	1825	200	59.2
Bismuth	545	9780	122	7.86	6.59
Boron	2573	2500	1107	27.0	9.76
Cadmium	594	8650	231	96.8	48.4
Chromium	2118	7160	449	93.7	29.1
Cobalt	1769	8862	421	99.2	26.6
Copper					
Pure	1358	8933	385	401	117
Commercial bronze (90% Cu, 10% Al)	1293	8800	420	52	14
Phosphor gear bronze (89% Cu, 11% Sn)	1104	8780	355	54	17
Cartridge brass (70% Cu, 30% Zn)	1188	8530	380	110	33.9
Constantan (55% Cu, 45% Ni)	1493	8920	384	23	6.71
Germanium	1211	5360	322	59.9	34.7

Appendix C

Table C.2 Properties of R134a (Arora, 2010)

Temp., T °C	Sat. press., P_{sat} kPa	Specific volume, m ³ /kg		Internal energy, kJ/kg			Enthalpy, kJ/kg			Entropy, kJ/kg·K		
		Sat. liquid, v_f	Sat. vapor, v_g	Sat. liquid, u_f	Evap., u_{fg}	Sat. vapor, u_g	Sat. liquid, h_f	Evap., h_{fg}	Sat. vapor, h_g	Sat. liquid, s_f	Evap., s_{fg}	Sat. vapor, s_g
-40	51.25	0.0007054	0.36081	-0.036	207.40	207.37	0.000	225.86	225.86	0.00000	0.96866	0.96866
-38	56.86	0.0007083	0.32732	2.475	206.04	208.51	2.515	224.61	227.12	0.01072	0.95511	0.96584
-36	62.95	0.0007112	0.29751	4.992	204.67	209.66	5.037	223.35	228.39	0.02138	0.94176	0.96315
-34	69.56	0.0007142	0.27090	7.517	203.29	210.81	7.566	222.09	229.65	0.03199	0.92859	0.96058
-32	76.71	0.0007172	0.24711	10.05	201.91	211.96	10.10	220.81	230.91	0.04253	0.91560	0.95813
-30	84.43	0.0007203	0.22580	12.59	200.52	213.11	12.65	219.52	232.17	0.05301	0.90278	0.95579
-28	92.76	0.0007234	0.20666	15.13	199.12	214.25	15.20	218.22	233.43	0.06344	0.89012	0.95356
-26	101.73	0.0007265	0.18946	17.69	197.72	215.40	17.76	216.92	234.68	0.07382	0.87762	0.95144
-24	111.37	0.0007297	0.17395	20.25	196.30	216.55	20.33	215.59	235.92	0.08414	0.86527	0.94941
-22	121.72	0.0007329	0.15995	22.82	194.88	217.70	22.91	214.26	237.17	0.09441	0.85307	0.94748
-20	132.82	0.0007362	0.14729	25.39	193.45	218.84	25.49	212.91	238.41	0.10463	0.84101	0.94564
-18	144.69	0.0007396	0.13583	27.98	192.01	219.98	28.09	211.55	239.64	0.11481	0.82908	0.94389
-16	157.38	0.0007430	0.12542	30.57	190.56	221.13	30.69	210.18	240.87	0.12493	0.81729	0.94222
-14	170.93	0.0007464	0.11597	33.17	189.09	222.27	33.30	208.79	242.09	0.13501	0.80561	0.94063
-12	185.37	0.0007499	0.10736	35.78	187.62	223.40	35.92	207.38	243.30	0.14504	0.79406	0.93911
-10	200.74	0.0007535	0.099516	38.40	186.14	224.54	38.55	205.96	244.51	0.15504	0.78263	0.93766
-8	217.08	0.0007571	0.092352	41.03	184.64	225.67	41.19	204.52	245.72	0.16498	0.77130	0.93629
-6	234.44	0.0007608	0.085802	43.66	183.13	226.80	43.84	203.07	246.91	0.17489	0.76008	0.93497
-4	252.85	0.0007646	0.079804	46.31	181.61	227.92	46.50	201.60	248.10	0.18476	0.74896	0.93372
-2	272.36	0.0007684	0.074304	48.96	180.08	229.04	49.17	200.11	249.28	0.19459	0.73794	0.93253
0	293.01	0.0007723	0.069255	51.63	178.53	230.16	51.86	198.60	250.45	0.20439	0.72701	0.93139
2	314.84	0.0007763	0.064612	54.30	176.97	231.27	54.55	197.07	251.61	0.21415	0.71616	0.93031
4	337.90	0.0007804	0.060338	56.99	175.39	232.38	57.25	195.51	252.77	0.22387	0.70540	0.92927
6	362.23	0.0007845	0.056398	59.68	173.80	233.48	59.97	193.94	253.91	0.23356	0.69471	0.92828
8	387.88	0.0007887	0.052762	62.39	172.19	234.58	62.69	192.35	255.04	0.24323	0.68410	0.92733
10	414.89	0.0007930	0.049403	65.10	170.56	235.67	65.43	190.73	256.16	0.25286	0.67356	0.92641
12	443.31	0.0007975	0.046295	67.83	168.92	236.75	68.18	189.09	257.27	0.26246	0.66308	0.92554
14	473.19	0.0008020	0.043417	70.57	167.26	237.83	70.95	187.42	258.37	0.27204	0.65266	0.92470
16	504.58	0.0008066	0.040748	73.32	165.58	238.90	73.73	185.73	259.46	0.28159	0.64230	0.92389
18	537.52	0.0008113	0.038271	76.08	163.88	239.96	76.52	184.01	260.53	0.29112	0.63198	0.92310

Table C.2: Properties of R134a (concluded)

Temp., T °C	Sat. press., P_{sat} kPa	Specific volume, m^3/kg		Internal energy, kJ/kg			Enthalpy, kJ/kg			Entropy, $\text{kJ/kg}\cdot\text{K}$		
		Sat. liquid, v_f	Sat. vapor, v_g	Sat. liquid, u_f	Evap., u_{fg}	Sat. vapor, u_g	Sat. liquid, h_f	Evap., h_{fg}	Sat. vapor, h_g	Sat. liquid, s_f	Evap., s_{fg}	Sat. vapor, s_g
20	572.07	0.0008161	0.035969	78.86	162.16	241.02	79.32	182.27	261.59	0.30063	0.62172	0.92234
22	608.27	0.0008210	0.033828	81.64	160.42	242.06	82.14	180.49	262.64	0.31011	0.61149	0.92160
24	646.18	0.0008261	0.031834	84.44	158.65	243.10	84.98	178.69	263.67	0.31958	0.60130	0.92088
26	685.84	0.0008313	0.029976	87.26	156.87	244.12	87.83	176.85	264.68	0.32903	0.59115	0.92018
28	727.31	0.0008366	0.028242	90.09	155.05	245.14	90.69	174.99	265.68	0.33846	0.58102	0.91948
30	770.64	0.0008421	0.026622	92.93	153.22	246.14	93.58	173.08	266.66	0.34789	0.57091	0.91879
32	815.89	0.0008478	0.025108	95.79	151.35	247.14	96.48	171.14	267.62	0.35730	0.56082	0.91811
34	863.11	0.0008536	0.023691	98.66	149.46	248.12	99.40	169.17	268.57	0.36670	0.55074	0.91743
36	912.35	0.0008595	0.022364	101.55	147.54	249.08	102.33	167.16	269.49	0.37609	0.54066	0.91675
38	963.68	0.0008657	0.021119	104.45	145.58	250.04	105.29	165.10	270.39	0.38548	0.53058	0.91606
40	1017.1	0.0008720	0.019952	107.38	143.60	250.97	108.26	163.00	271.27	0.39486	0.52049	0.91536
42	1072.8	0.0008786	0.018855	110.32	141.58	251.89	111.26	160.86	272.12	0.40425	0.51039	0.91464
44	1130.7	0.0008854	0.017824	113.28	139.52	252.80	114.28	158.67	272.95	0.41363	0.50027	0.91391
46	1191.0	0.0008924	0.016853	116.26	137.42	253.68	117.32	156.43	273.75	0.42302	0.49012	0.91315
48	1253.6	0.0008996	0.015939	119.26	135.29	254.55	120.39	154.14	274.53	0.43242	0.47993	0.91236
52	1386.2	0.0009150	0.014265	125.33	130.88	256.21	126.59	149.39	275.98	0.45126	0.45941	0.91067
56	1529.1	0.0009317	0.012771	131.49	126.28	257.77	132.91	144.38	277.30	0.47018	0.43863	0.90880
60	1682.8	0.0009498	0.011434	137.76	121.46	259.22	139.36	139.10	278.46	0.48920	0.41749	0.90669
65	1891.0	0.0009750	0.009950	145.77	115.05	260.82	147.62	132.02	279.64	0.51320	0.39039	0.90359
70	2118.2	0.0010037	0.008642	154.01	108.14	262.15	156.13	124.32	280.46	0.53755	0.36227	0.89982
75	2365.8	0.0010372	0.007480	162.53	100.60	263.13	164.98	115.85	280.82	0.56241	0.33272	0.89512
80	2635.3	0.0010772	0.006436	171.40	92.23	263.63	174.24	106.35	280.59	0.58800	0.30111	0.88912
85	2928.2	0.0011270	0.005486	180.77	82.67	263.44	184.07	95.44	279.51	0.61473	0.26644	0.88117
90	3246.9	0.0011932	0.004599	190.89	71.29	262.18	194.76	82.35	277.11	0.64336	0.22674	0.87010
95	3594.1	0.0012933	0.003726	202.40	56.47	258.87	207.05	65.21	272.26	0.67578	0.17711	0.85289
100	3975.1	0.0015269	0.002630	218.72	29.19	247.91	224.79	33.58	258.37	0.72217	0.08999	0.81215

Table C.3: Superheated property of R134a (Arora, 2010)

T °C	v m ³ /kg	u kJ/kg	h kJ/kg	s kJ/kg-K	v m ³ /kg	u kJ/kg	h kJ/kg	s kJ/kg-K	v m ³ /kg	u kJ/kg	h kJ/kg	s kJ/kg-K
$P = 0.50 \text{ MPa } (T_{\text{sat}} = 15.71^\circ\text{C})$					$P = 0.60 \text{ MPa } (T_{\text{sat}} = 21.55^\circ\text{C})$				$P = 0.70 \text{ MPa } (T_{\text{sat}} = 26.69^\circ\text{C})$			
Sat.	0.041118	238.75	259.30	0.9240	0.034295	241.83	262.40	0.9218	0.029381	244.48	265.03	0.9199
20	0.042115	242.40	263.46	0.9383								
30	0.043338	250.84	273.01	0.9703	0.035584	249.32	270.81	0.9499	0.029966	247.48	268.45	0.9313
40	0.044856	259.26	282.68	1.0011	0.037865	257.86	280.58	0.9816	0.031696	256.39	278.57	0.9641
50	0.046499	267.72	291.96	1.0309	0.039659	266.48	290.28	1.0121	0.033322	265.20	288.53	0.9954
60	0.050485	276.25	301.50	1.0599	0.041589	275.15	299.98	1.0417	0.034875	274.01	298.42	1.0256
70	0.052427	284.89	311.10	1.0883	0.043069	283.89	309.73	1.0705	0.036373	282.87	308.33	1.0549
80	0.055331	293.64	320.80	1.1162	0.044710	292.78	319.55	1.0987	0.037829	291.80	318.28	1.0835
90	0.058205	302.51	330.61	1.1436	0.046318	301.67	329.46	1.1264	0.039259	300.82	328.29	1.1114
100	0.059953	311.50	340.53	1.1705	0.047900	310.79	339.47	1.1536	0.040642	309.95	338.40	1.1389
110	0.059880	320.63	350.57	1.1971	0.049458	319.91	349.59	1.1803	0.042010	319.19	348.60	1.1658
120	0.061687	329.89	360.73	1.2233	0.050997	329.29	359.82	1.2067	0.043358	328.55	358.90	1.1928
130	0.063479	339.29	371.03	1.2491	0.052519	338.87	370.18	1.2327	0.044688	338.04	369.32	1.2188
140	0.065256	348.83	381.46	1.2747	0.054027	348.25	380.66	1.2584	0.046004	347.66	379.85	1.2444
150	0.067021	358.51	392.02	1.2999	0.055522	357.96	391.27	1.2838	0.047306	357.41	390.52	1.2699
160	0.068775	368.33	402.72	1.3249	0.057006	367.81	402.01	1.3088	0.048597	367.29	401.31	1.2951
$P = 0.80 \text{ MPa } (T_{\text{sat}} = 31.31^\circ\text{C})$					$P = 0.90 \text{ MPa } (T_{\text{sat}} = 35.51^\circ\text{C})$				$P = 1.00 \text{ MPa } (T_{\text{sat}} = 39.37^\circ\text{C})$			
Sat.	0.025621	246.79	267.29	0.9183	0.023683	248.85	265.26	0.9169	0.020313	250.68	270.99	0.9155
40	0.027035	254.83	276.45	0.9480	0.023375	253.13	274.17	0.9327	0.020406	251.30	271.71	0.9179
50	0.028547	263.86	286.69	0.9802	0.024899	262.44	284.77	0.9660	0.021796	260.34	282.74	0.9523
60	0.029973	272.83	296.81	1.0110	0.026146	271.60	295.13	0.9976	0.023068	270.32	293.38	0.9850
70	0.031340	281.81	306.88	1.0408	0.027613	280.72	305.39	1.0300	0.024261	279.59	303.85	1.0160
80	0.032659	290.84	316.97	1.0698	0.028630	289.86	315.63	1.0578	0.025398	288.86	314.25	1.0458
90	0.033941	299.98	327.10	1.0981	0.029806	299.06	325.89	1.0860	0.026492	298.15	324.64	1.0748
100	0.035193	309.15	337.30	1.1258	0.030951	308.34	336.19	1.1140	0.027562	307.51	335.06	1.1031
110	0.036420	318.45	347.59	1.1530	0.032068	317.70	346.56	1.1414	0.028684	316.94	345.53	1.1308
120	0.037625	327.87	357.97	1.1798	0.033164	327.18	357.02	1.1684	0.029792	326.47	356.06	1.1580
130	0.038813	337.40	368.45	1.2061	0.034241	336.76	367.58	1.1949	0.030881	336.11	366.69	1.1846
140	0.039985	347.06	379.05	1.2321	0.035302	346.46	378.23	1.2210	0.031954	345.85	377.40	1.2109
150	0.041143	356.88	389.76	1.2577	0.036349	356.28	389.00	1.2467	0.032912	355.71	388.22	1.2368
160	0.042290	366.78	400.59	1.2830	0.037384	366.23	399.88	1.2721	0.033887	365.70	399.15	1.2623
170	0.043427	376.81	411.55	1.3080	0.038408	376.31	410.88	1.2972	0.034832	375.81	410.20	1.2875
180	0.044554	386.99	422.64	1.3327	0.039423	386.52	422.00	1.3221	0.035817	386.04	421.36	1.3128
$P = 1.20 \text{ MPa } (T_{\text{sat}} = 46.29^\circ\text{C})$					$P = 1.40 \text{ MPa } (T_{\text{sat}} = 52.40^\circ\text{C})$				$P = 1.60 \text{ MPa } (T_{\text{sat}} = 57.88^\circ\text{C})$			
Sat.	0.016715	253.81	273.87	0.9130	0.014107	256.37	276.12	0.9105	0.012123	258.47	277.86	0.9078
50	0.017201	257.63	278.27	0.9267								
60	0.018404	267.56	289.64	0.9614	0.015005	264.46	285.47	0.9383	0.012372	260.89	280.63	0.9163
70	0.019802	277.21	300.81	0.9938	0.016060	274.62	297.10	0.9733	0.013490	271.76	293.25	0.9535
80	0.020529	286.75	311.39	1.0248	0.017023	284.51	308.34	1.0056	0.014362	282.09	305.07	0.9875
90	0.021536	296.26	322.07	1.0556	0.017923	294.28	319.37	1.0364	0.015215	292.17	316.52	1.0194
100	0.022442	306.80	332.73	1.0866	0.018778	304.01	330.30	1.0661	0.016014	302.14	327.76	1.0500
110	0.023348	316.38	343.40	1.1118	0.019597	313.76	341.19	1.0949	0.016773	312.07	338.91	1.0795
120	0.024228	325.69	354.11	1.1394	0.020388	323.55	352.69	1.1230	0.017500	322.02	350.02	1.1081
130	0.025086	334.77	364.88	1.1664	0.021155	333.41	363.02	1.1504	0.018201	332.00	361.12	1.1360
140	0.025927	344.61	375.72	1.1930	0.021904	343.34	374.01	1.1773	0.018882	342.05	372.26	1.1632
150	0.026753	354.56	386.66	1.2192	0.022636	353.37	385.07	1.2038	0.019545	352.17	383.44	1.1900
160	0.027566	364.61	397.69	1.2449	0.023355	363.51	396.20	1.2298	0.020194	362.38	394.69	1.2163
170	0.028367	374.78	408.82	1.2703	0.024061	373.75	407.43	1.2564	0.020830	372.69	406.02	1.2421
180	0.029158	385.08	420.07	1.2954	0.024757	384.10	418.76	1.2807	0.021456	383.11	417.44	1.2676

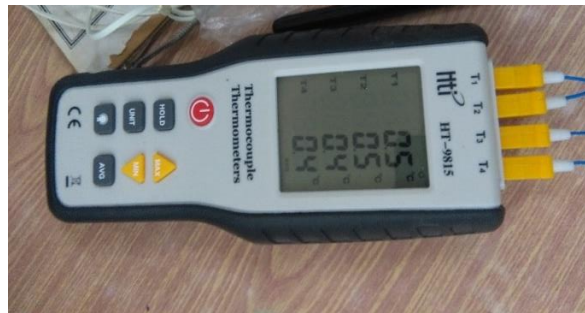
Appendix D

Thermocouple calibration

Thermocouples are sensors used to measure temperature by forming a junction between two wires of separate metals. One end of the junction is called the measurement “hot” junction and the other end is called the reference “cold” junction. When there is a temperature variation in the joint, voltage is created and this voltage can be converted into temperature with a thermocouple reference table or data logger. Thermocouples can be calibrated using a reference thermometer in a temperature calibration test by comparing both measurement values of a reference thermometer and other thermometers. The thermocouple used in this study is a K-type four channel thermocouple and a constant high temperature bath was used for the calibration. A laboratory standard thermocouple was used as a reference for the calibration and the results of the temperature measurement of both the reference and the K-type thermocouple is shown in Table D.1.



a.



b.

Figure D.1: Thermocouples a) reference thermocouple (WL-362) and b) K-type thermocouple.

K-type thermocouple specifications

Serial number: HT-9815

Temperature range: 0-800°C

Power supply: 9V

Weight: approx. 250g

Table D.1: Data from thermocouples during calibration.

Time	T _{w1} (WL-362) (°C)	T _{w2} (K-type thermocouple) (°C)	Uncertainty (%) $\frac{(T_{w1}-T_{w2})}{T_{w1}} * 100$
1	30.1	29.8	0.99668
3	35.6	35.4	0.5618
5	47.3	47.1	0.42283
7	57.1	56.8	0.52539
9	66.3	65.9	0.60332
11	72.6	72.1	0.68871
13	76.8	76.4	0.52083
15	79.8	79.5	0.37594
17	82.6	82.1	0.60533
19	84.3	83.9	0.4745
20	86.6	86.2	0.46189
22	88.1	87.8	0.34052
24	90.5	90.1	0.44199
26	91.3	90.9	0.43812
Average uncertainty			0.53

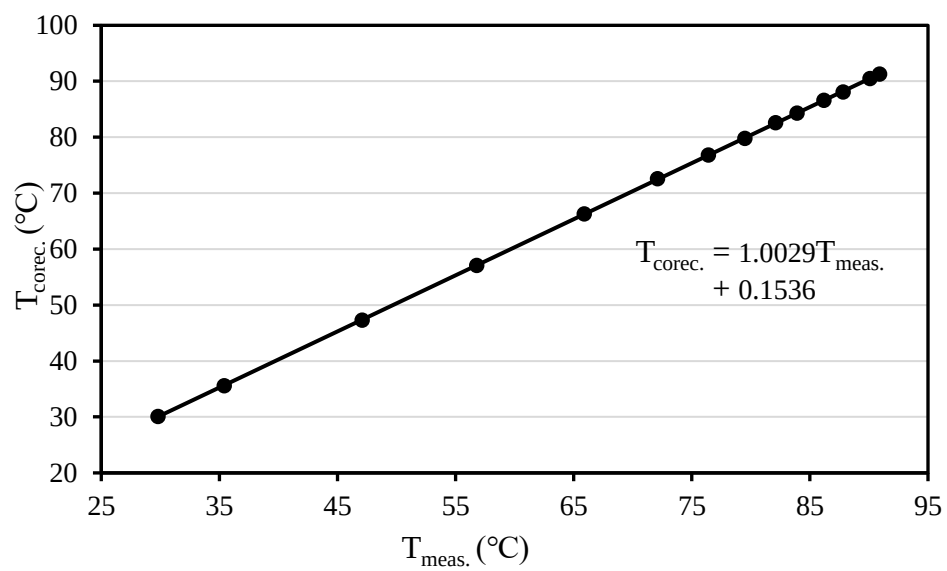


Figure D.2: Measured vs corrected temperature.