A six-chamber medium-to-high temperature refrigeration system for laboratory purposes

By

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DECLARATION

I, **BELAID ASHUR BELAID**, declare that the contents of this dissertation represent my own unaided work, and that the dissertation has not previously been submitted for academic examination towards any qualification. Furthermore, it represents my own opinions and not necessarily those of the Cape Peninsula University of Technology.

Signed **Date**

ABSTRACT

Food and water fuel and sustain life on earth. Since ancient times, their sourcing and preservation have been very important issues to humanity. In modern times, experimentation is a major step in analysing how cold storage problems in the medical and food science technology fields can be addressed. For investigating spoilage of new products and/or growth of pathogens in such cases, it is necessary to do experiments at different low temperatures for prolonged periods and check the effects. While this can be undertaken in a conventional two chambers fridge, it takes a long time to investigate a whole range of feasible storage temperatures. This dissertation describes work intended to treble the samples in such investigations and, therefore, significantly reduce the times. Six well-insulated chambers were constructed from plastic and wood and set at different temperatures. An ordinary deep freezer was used as the main heat sink for the chambers. Experiments were done with different methods to get the best result for the cold air flow into the chambers over period of eight months. The chambers temperatures were set between 0 and 15°C. It was finally established that a properly-designed six-chamber system could successfully be added to an ordinary freezer to provide different medium-to-high refrigeration temperatures without the use of a multi-pressure refrigeration cycle.

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DEDICATION

I would like to dedicate this thesis to my family and friends.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

GLOSSARY

Symbols	Description	Unit
A	Area	m^2
COP	Coefficient of performance	-
VARS	Vapour absorption refrigeration system	-
VCRS	Vapour compression refrigeration system	-
VCARS	Vapour compression – absorption refrigeration system	-
Ch1, Ch2, Ch3	Chamber1, ………., Chamber6	-
T_S	The temperature of the surroundings	℃
$T\text{-}_{MCC}$	The temperature of the main cooling chamber	℃
$T\text{-}_{BS}$	The temperature of the back space	℃
k	Thermal conductivity of wall material	$\mathrm{W/(m.K)}$
h_i	Inside surface conductance	$\mathrm{W/(m^2{\cdot}K)}$
h_o	Outside surface conductance	$\mathrm{W/(m^2{\cdot}K)}$
U	Overall heat transfer coefficient	$\mathrm{W/(m^2{\cdot}K)}$
m	Mass, mass of air	kg
$LMTD$	Log mean temperature difference	℃
P	Pressure, vapour pressure	Pa
h_{fg}	Latent heat of evaporation	kJ/kg
t	time	$hours$
c_p	Specific heat	$\frac{\mathrm{kJ}}{\mathrm{kg.K}}$
Q	Heat energy	kJ
T	Temperature	℃
φ	relative humidity, percent or fraction	%

CHAPTER ONE
INTRODUCTION TO THE FOOD STORAGE CONCEPT

1.1 BACKGROUND

Before the invention of the mechanical refrigeration system, various methods of preserving food were used by people of different cultures. Some preferred to use snow or ice that was available in and around the areas where they lived, brought down from the mountains and stored in cellars in the cold seasons to be used in the warmer or the hotter ones. The first cellars were holes dug into the ground, lined with wood or straw and then filled with ice and snow (PEAK Mechanical Partnership, nd). This refrigeration principle goes back to the year 1000 BC when ice was cut into pieces in winter and stored for the summer by the Chinese as well as the ancient Hebrews, Romans and the Greeks. This seasonal harvesting of ice or snow was very common throughout the ancient cultures. Pickling, salting, spicing, drying and smoking were other methods used for preservation. These techniques were used mainly for cheese, salted meats and bread. Rapid spoilage of food could not be prevented by the use of window boxes or cellar storage for products such as milk and cheeses. These products were considered to be the most difficult to keep fresh. Pasteurisation was not understood at the time, therefore, infestation of bacteria was widespread and in the colonial days, during the warm weather. The death was common in summer, because of spoiled food. Therefore, a form of improved and a better food preservation method was needed (PEAK Mechanical Partnership, nd).

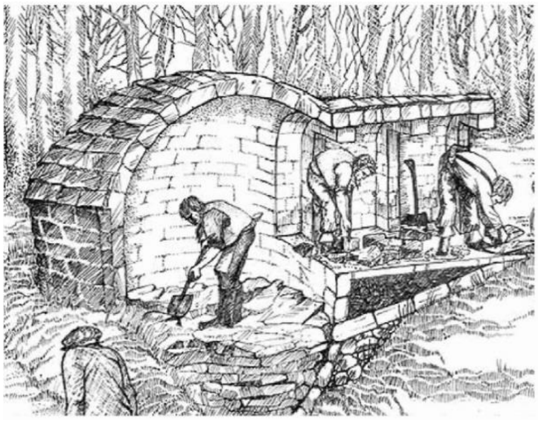


Figure 1.1: The ancient culture of ice storage 1000 BC(Ice house).

The recognition of this need was observed by the creative thinkers in India, who created and used the first evaporative cooling systems. These worked on the principle that when a liquid vaporises quickly, it rapidly expands and the kinetic energy increases due to the rising of the vapour molecules. This increases the energy drawn from the surrounding vapour immediately and causes it to cool (PEAK Mechanical Partnership, nd)..

Later on, reducing temperature with the addition of chemicals such as potassium nitrate or sodium nitrate was discovered. In 1550, the first cooling of wine using this technique was recorded and the term ‘refrigeration’ came into being. In 1600, cooled drinks became very popular in Europe, specifically in Spain, France and Italy. Long-necked bottles were rotated in water containing dissolved saltpetre. This was a new technique for cooling instead of cooling the water at night and using it during the day. This method was also used to create low temperatures in addition to the usage of ice. Frozen juice, liquors and iced drinks were popular in France by the end of the 17th century (PEAK Mechanical Partnership, nd).

A strong demand for ice was prompted at the end of the 17th century. In 1799, the first commercial shipment of ice was shipped out from Canal Street in New York to Charlestown. However, this attempt was a failure as there very little ice remained in the shipment when it arrived. The potential for the existence of ice as a business was seen by Frederick Tudor and Nathaniel Wyeth of New England. In the first half of the 1800s, the industry was revolutionised by their efforts. Shipping ice to the tropical climates was the concern of the Frederick Tudor 'Ice King', an American businessman. His project and experiments led to a decrease in the losses of ice shipments from 66% to 8% caused by ice melting (PEAK Mechanical Partnership, nd)..

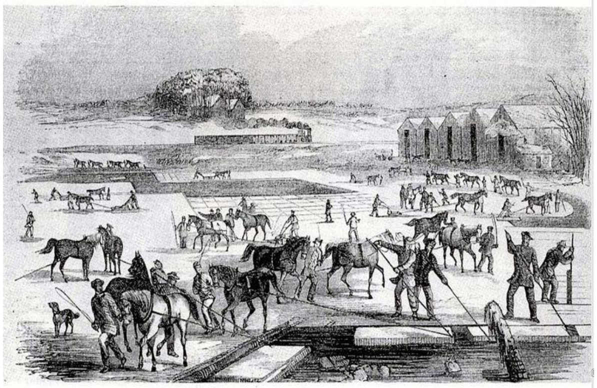


Figure 1.2: Ice Harvesting in Massachusetts 1850s (Refrigeration).

He experimented and tested different insulation materials, with the result that icehouses were built to decrease the loss caused by melting, thus ensuring the shipment's safety. A cheap and quick method for cutting ice into uniform blocks that the industry required was developed by Wyeth (PEAK Mechanical Partnership, nd). The handling technique of ice in storage improved and became easy, as well as the possibility of transporting and distributing ice with less waste. Therefore, more and more companies became involved in the ice business. Prices of ice fell and ice refrigeration became more realistic and available. In America, there were 35 commercial plants for ice production in 1879. A decade later, there were 200 and by 1909, this number had increased to 2000. In 1907, the consumption of ice was triple that of 1880 and it reached a peak of 15 million tons. Ponds were scraped everywhere for ice production. In 1847, approximately 1000 tons of ice was extracted on a daily basis from Thoreau's Walden Pond (PEAK Mechanical Partnership, nd).

Health problems occurred due to the fact that the ice scraped was not clean. (PEAK Mechanical Partnership, nd). Eventually, this problem became evident and finding clean natural ice sources became a difficult task. By the 1890s, sewage dumping and pollution had made it even more impossible. In the brewing industry, the first signs of health problems were noticed, followed by the dairy and meat-packing industry, which were severely affected. There was a desperate need for a new preservation method of food (PEAK Mechanical Partnership, nd).

After the cooling revolution through the use of chemical salts in Europe, scientists and inventers began to look at other ways of solving this issue. In 1720, the process of evaporating liquid in a vacuum was studied for the first time by a Scotsman, Dr William Cullen, and the first known artificial refrigeration was demonstrated in 1748 at The University of Glasgow, where he revealed the process of allowing ethyl ether to boil into a partial vacuum (PEAK Mechanical Partnership, nd).

Figure 1.3: William Cullen, 1720 (Refrigeration).

At the beginning of the ice industry revolution, the first refrigeration machine was designed by an American inventor, Oliver Events (PEAK Mechanical Partnership, nd). Vapour was used instead of liquid in 1805. Although the system was not built by him, a similar machine was used in 1842 by John Gorrie, an American physician, at a Florida hospital for cooling patients suffering from Yellow Fever (PEAK Mechanical Partnership, nd).. To date the refrigeration design most frequently used is based on the basic principle of John Gorrie's machine (PEAK Mechanical Partnership, nd).. This successful method involved compressing the gas which was then sent through radiating coils that lowered its temperature, and then the gas expanded to the lower temperature of the surroundings. He stopped practising medicine to focus on his experiments to make ice after the first United States patent for the production of mechanical refrigeration was granted to him in 1851(PEAK Mechanical Partnership, nd).

Back in the year 1820, the first cooling process produced by liquid ammonia was done by Michael Faraday in London (Figure 1.4). In 1859, the first refrigeration machine using ammonia and water was developed by Ferdinand Carré of France. Carl von Linde from Munich designed the first practical and portable refrigeration compressor in 1873. He substituted ammonia for methyl ether in the process in 1876. In 1894, large quantities of air

Figure 1.4: Michael Faraday, 1820 (Michael_Faraday).

were liquefied and this method was named the ‘Linde technique’ (PEAK Mechanical Partnership, nd).

The first recognition of the major benefits of using the refrigeration process system was in the brewing industry. In the 1840s, with the arrival of the German immigrants, a German lager beer was introduced in America. The beer proved to be superior to the American ale. Throughout the entire year, the breweries were able to make uniform products. In 1870, S. Liebmann’s Sons Brewing Company in Brooklyn, New York was the first to adopt the use of an absorption refrigeration system and by 1891, most of the breweries were using refrigerating machines (PEAK Mechanical Partnership, nd).

Approximately a decade later, the mechanical refrigeration system was adopted in Chicago for use by the meat-packing industry. The compression refrigeration system using ammonia had a capacity of over 90,000 tons per day in use by most of the meat-packing industry in America by 1914. Cars, cold storage and brunch house facilities such as restaurants, hotels, offices used this refrigeration system technology. The refrigeration system greatly improved the industry and meat could be transported to markets throughout the year (PEAK Mechanical Partnership, nd). Figure 1.5 demonstrates the timeline of refrigerant history.

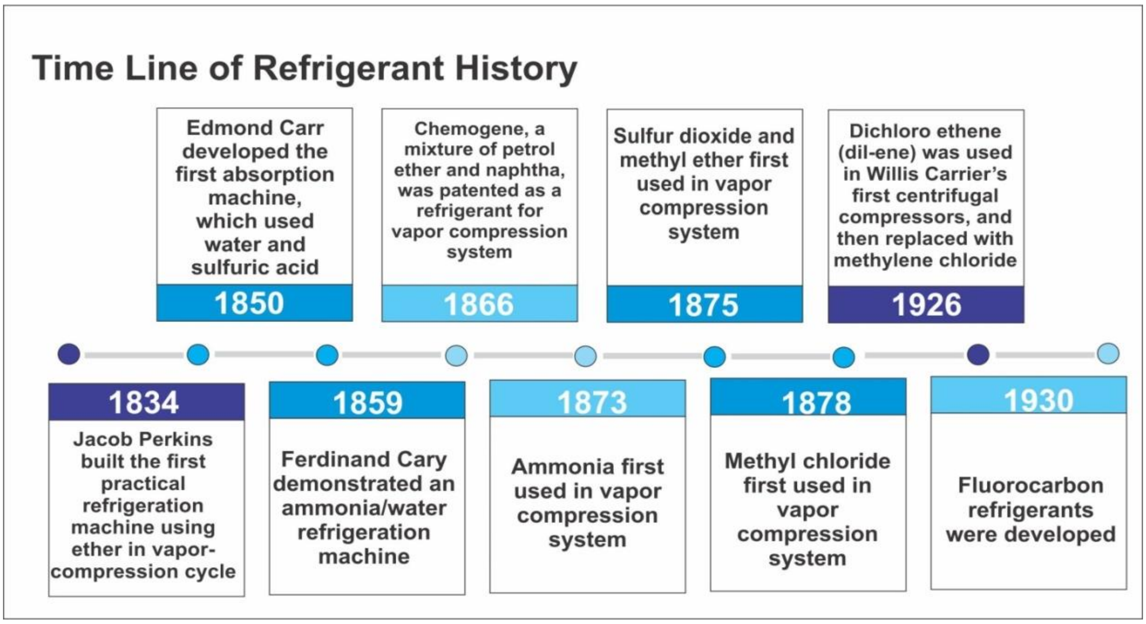


Figure 1.5: Timeline of refrigerant history

(Refrigerant-safety).

1.2 THE PHENOMENON

Heat is transferred in the direction of decreasing temperature, that is, from the higher temperatures to lower temperatures. Reversing this phenomenon is possible and is not in violation of the thermodynamic second law if additional work has been done on the system (James & James, 2010). A system called 'refrigerator' is required as an intermediate device to transfer heat from low temperature regions to high ones. The dividing line at -70 °C or 100 °F marks the temperatures within the refrigeration system where physical philosophies and methods are applied to achieve low temperatures. Below this, cryogenic devices take over.

Figure 1.6:1927 witnesses the first widespread refrigerator being used (Flik).

In industrial technology and science, experimentation is an important issue, where constant monitoring and observation of systems are done. When the desired result is not achieved, further research has to be conducted until a final and successful result is achieved. Such experiments may run for years to achieve the desired outcome if only one or two items are monitored at a time. Most refrigeration systems have one or two chambers and precise temperature control in these chambers is limited.

1.3 PROBLEM STATEMENT

Due to the growth of the world's population and the necessity to ensure adequate food supply, high-level experiments have skilfully been performed globally. To date, the issue of malnutrition remains a major challenge globally (Rolle, 2006). Sciences and technology have pushed the agricultural output to maximum beyond the market demand or supply in the most developed countries; this has led to degradation of the ecosystem of the agricultural environment in some countries. The industrial revolution in the 20th century in the most developed counties, and its effect on the ozone layer and the pollution has made the world live under the neutral crises umbrella. Climate change is the direct result of the behaviour of the developed countries. Flooding, dryness, wars due to the political conflict and bad management of the agricultural resources, are the major undernourishment factors in the less developed countries. Therefore, from my point view there is no shortage of food worldwide, at least in the agricultural product, but what is there is food waste in worldwide production especially the fruits and vegetables which is the most favourable to human and more favourable when they are fresh. The focus now shouldn't be on increasing the production, as it should be on reducing the percentage losses of the post-harvest as much as possible, but how can this be possible? The answer is refrigeration. It can be an attractive option for the reduction of post-harvest losses in fruits and vegetable, and in foodstuffs in general. This is because it preserves most of the initial food quality, and maintains the initial chemical, physical and nutritional properties with extent shelf life. Hence, consumers can be provided with wholesome and safe foodstuffs (Peck, 1997). The lack of a refrigeration infrastructure in many countries, and specifically in less developed ones where undernourishment along with the population growth is greatest, means it is still neither sufficiently nor equally used to ensure safety and security of food (Peck, 1997). Contributing to the reduction of post-harvest losses by quickly establishing suitable storage conditions under laboratory conditions is the broader aim of this study.

Other methods which can be used for reducing post-harvest losses of foodstuffs include high-pressure processing, salting, electrically pulsed food processing and canning (Peck, 1997). But none of those methods named above at the level of quality for food preservation as the refrigeration system for extending the life shelf and maintaining the initial sensory, chemical, physics, and nutritional properties for the extent of the consumer desired. It is reported that refrigeration technology would allow less-developed countries to increase their food supply of perishable foodstuffs extensively and reliably by about 15%. This would allow them to approach food security in a similar way to that in the more industrialised countries (Gustavsson, Cederberg & Sonesson, 2011). To cool and store food at the site of production is vital to ensure the effectiveness of the cold chain. Accordingly, adequate training of local engineers and technicians is essential for the cold chain plants.

According to Gustavsson *et al*., (2011), there is a growing demand for mainly processed food that are of a high quality, nutritious and easy to prepare. These demands have been met where the refrigerated foods were developed with an extant shelf life (Peck, 1997). Such types of food have been precooked or slightly processed where their shelf-life is limited and refrigeration is their preservation. They include cured meats, luncheon meats and the new generations of processed, refrigerated foods, like seafood, vegetable salads, pasta sauce, soups, complete meals and so on (Leistner, 2000). They require less heat to prepare than the heat required for commercial sterility.

One of the major causes of food loss in low-income countries is the poor environmental conditions while it is on display. According to Gustavsson *et al*. (2011), poor temperature management on shelves where food is displayed causes various problems like chilling or heat sensitivity.

According to Sonnino, Faus and Maggio (2014), refrigeration along the different stages of the food chain has an important contribution to make for sustainable food security, by applying the role of food-loss prevention for a period of time. However, refrigeration, especially in developed countries, also constitutes a major source of various types of emissions (Pelletier, Audsley, Brodt, Garnett, Henriksson, Kendall, Kramer, Murphy, Nemecek & Troell, 2011). Coulomb (2008) estimates, that 15% of electricity consumed worldwide is used for refrigeration. With current changes in ambient temperature, this value is likely to increase worldwide (James & James, 2010).

Transportation and refrigeration of food in the post-production stage, have received the most scientific and media attention and are closely connected to food security (Sonnino *et al*., 2014). Yilmaz and Karadeniz, (2014) point out that it is important to understand that every refrigerator has an area (or a surface) where the temperature is lower than that of ambient. For the cooling of an object, one needs to establish thermal contact between this object, the cold area of the refrigerator and the heat flowing from the object to the cold surface, until the temperature of the object is balanced with the cold area of the refrigerator.

For example, Yilmaz and Karadeniz (2014) carried out research on quince fruit, which is produced by the deciduous tree of the Rosaceae family, namely Cydonia Oblonga Miller, which is widely produced in Turkey. Samples were stored using four cooling rooms at 5, 20, 30 and 40 °C for nine months and analysed each month. The findings from this study show that antioxidant activity of quince nectar decreased significantly after storage at all four varying storage temperatures of the experiment. The ascorbic acid and phenolic compounds were the most important contributors to the antioxidant activity in the quince nectar. Kinetics of L-ascorbic acid degradation reaction showed that the first-order kinetic model was most suitable for quince nectar storage at 20–40 °C while non-enzymatic browning followed a zero-order reaction (Yilmaz & Karadeniz, 2014: 7).

Following the usefulness of a refrigerator for experiments and the industry, there is a need for new technology to facilitate researchers in the course of their studies. Furthermore, the industry needs refrigeration technology that will help to keep food at proper temperatures to avoid food spoilage.

1.4 FINDINGS

Gustavsson, Cederberg and Sonesson (2011), assert that billions of dollars are lost each year in both industrialised and less-industrialised countries, due to food lost as a result of the lack of proper cold storage. For example, the estimation of total losses and waste in Europe was 40 - 45 million tons, with a total cost of 57 billion US dollars (USD); in North America and Oceania, it was 23 - 37 million tons with a total cost of 33 billion USD; and in industrialised Asia, it was 90 - 95 million tons with a total cost of 29 billion USD. Other losses and the costs thereof are as follows: Sub-Saharan Africa loses 15 - 20 million tons with a total cost of 7 billion USD; North Africa, West Asia and Central Asia lose 25- 30 million tons with a total cost of 12 billion USD; South and Southeast Asia lose 85 – 90 million tons with a total cost of 24 billion USD and Latin America loses 20 - 25 million tons with total cost of 11 billion USD. Figures 1.7 and 1.8 demonstrate the losses in industrialised and in less-developed countries respectively.

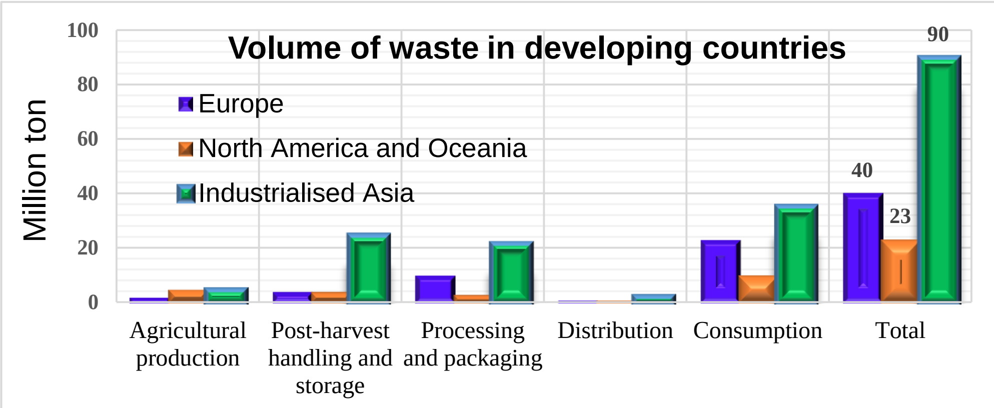


Figure 1.7: Yearly losses in Europe, North America & industrial. Asia (Gustavsson *et al.*, 2011).

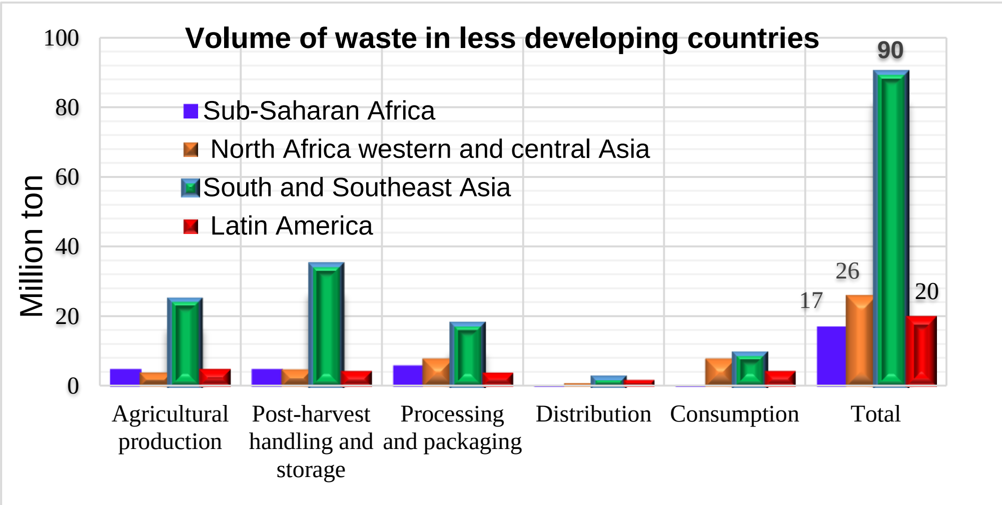


Figure 1.8: Yearly losses in Sub-Saharan Africa, North Africa, West and Central Asia, South and Southeast Asia and Latin America (Gustavsson *et al.*, 2013).

According to Sonnino, *et al*. (2014), refrigeration is the solution to food loss throughout the different stages of the food chain. Refrigeration makes an important contribution towards sustainable food security by fulfilling its role in preventing food loss. The aim of this project is to provide six temperature-controllable, thermally isolated chambers so that semi-processed food storage temperatures can be determined. The study was done in three phases, namely, the design, construction and testing phases. To ensure the quality of the output, each phase was tested before the researcher proceeded to the next phase.

1.5 JUSTIFICATION OF THE RESEARCH

Most refrigeration systems have one or two chambers and it is time-consuming for scientists who want to do experiments on storage temperatures. Therefore, there is a need for multiple, temperature-controllable, thermally-isolated chambers so that research times can be reduced.

1.6 RESEARCH OBJECTIVES

The objective of this study was to provide six chambers for a medium to high refrigeration system that can be used by scientists to do experiments on food products.

1.7 METHODOLOGY

This study was conducted in three phases:

- The design phase: a controllable-temperature, six chambers refrigerator for laboratory purposes was designed.
- The construction phase: a prototype was constructed in line with the design.
- The testing phase: four tests were done on the prototype, namely without load at fans speed 0, without load at fans speed maximum, without load at the chambers desired temperature, and testing with a load of 200 grams of apples in each chamber. Data was collected every half hour.

1.8 ASSUMPTIONS

The main cooling chamber below the experimental chambers runs at -30°C, which is usually the lowest temperature of the deep freezer. The assumption is that the main cooling chamber would be capable of bringing down the temperature of the experimental chambers down to 5° C, where the desired temperature for agriculture products lies between 5° C and 15° C.

CHAPTER TWO
LITERATURE REVIEW

2.1 OUTLINE

This chapter discusses the effects of food spoilage as well as the factors responsible for the spoilage of food. Food spoilage factors include temperature, desiccation, incompatibility and deterioration. The preservation of food to reduce losses will be demonstrated as well as the methods used, with the emphasis on refrigeration.

2.2 FOOD SPOILAGE FACTORS

Food spoilage is considered to have occurred when food undergoes undesired changes of state or contains objectionable material.

2.2.1 TEMPERATURE

According to Gould (1996), with a few exceptions, all foods lose quality and potential shelf life at some point. This is dependent on food type, composition, formulation, packaging and storage conditions. Spoilage or other changes that lead to loss of shelf-life can occur at any of the many stages between the acquisition of raw materials and eventual consumption of the finished product (Gould, 1996: 52).

Figure2.1: Chill Damage (Article-view).

Fruits and vegetables have a ‘critical temperature’ below which undesirable and irreversible reactions or ‘chill damage’ takes place (Gould, 1996: 52). For example, carrots blacken and become soft and the cell structure of potatoes is destroyed. The storage temperature always has to be above this critical temperature. Even when the thermostat is set at a temperature above the critical temperature, care must be taken that the thermostatic oscillation in temperature does not result in the storage temperature falling below the critical temperature. Even 0.5 °C below the critical temperature can result in ‘chill damage’ as can be seen in Table 2.1, which details the key temperatures for various fruits and vegetables.

Table 2.1: Storage conditions for fruits and vegetable (NPCS, 2016)

Item	Temperature ºC	Relative Humidity %	Maximum storage time recommended	Storage time in cold stores for vegetables in tropical countries
Apple	0-4	90-95	2-6 m*	
Beetroot	0	95-99		
Cabbage	0	95-99	5-6 m	2 m
Carrots	0	98-99	5-9 m	2 m
Cauliflower	0	95	2-4 w	1+ w
Cucumber	10-13	90-95		
Eggplant	8-10	90-95		
Lettuce	1	95-99		
Leeks	0	95	1-3 m	1 m
Oranges	0-4	85-90	3-4 m	
Pears	0	90-95	2-5 m	
Pumpkin	10-13	70-75		
Spinach	0	95	1-2 m	1 w
Tomatoes	13-21	85-90		

*m = months; +w = weeks

2.2.2 DESICCATION

Desiccation is a process in which the spoilage of unpacked fresh foods, such as meat, fish, fruits and vegetables, occurs due to the loss of moisture from the surface of the product by evaporation into the surrounding air (ASHRAE, 2002). In other words, desiccation is the state or the process of extreme dryness. In fruits and vegetables, for example, desiccation is accompanied by a considerable loss of weight and vitamin content. Water loss also causes a product to shrivel. This is the only physical factor related to the evaporative potential of air and can be expressed as illustrated in the formula below (ASHRAE, 2002:10.1):

$$P_D = \frac{P(100-\emptyset)}{100} \qquad (1.1)$$

Where:

P_D= *vapour pressure deficit, indicating combined influence of temperature and relative humidity on evaporative potential of air*

p = vapour pressure of water at given temperature

φ = relative humidity, percentage

The comparison of the air potential evaporating in two different storage rooms, one at 0 ºC and the other at 10 ºC dry bulb temperature (DBT), and for each storage room with 90% relative humidity (RH), the deficit of the air vapour pressure at 0 ºC is 60 Pa, whereas at 10 ºC, it is 120 Pa. Assuming all the other factors are equal, there is a tendency to lose water at 0 ºC and the DBT is lower than 10 ºC. (ASHRAE, 2002).

Figure 2.2: Desiccated peppers (Products-12).

2.2.3 INCOMPATIBILITY

According to Dinçer and Kanoĝlu (2011), if different products are stored in the same room there is a risk of the transfer of odours. Table 2.2 below shows the incompatibilities of different products.

Table 2.2: Incompatibilities between products: Source:(NPCS, 2016)

Item	Apple	Banana	Cabbage	Grapes	Oranges	Potatoes	Vegetable
Apples	-	N	SR	Y	Y	SR	Y
Bananas	N	-	N	Y	N	N	Y2
Cabbage	SR	Y	-	SR	N	SR	SR
Grapes	Y	Y	SR	-	Y	Y	Y
Oranges	Y	N	N	Y	-	Y	Y
Potatoes	SR	N	SR	Y	Y	-	Y
Vegetables	Y	Y	SR	Y	Y	Y	-

Y= No cross action SR =Slight danger BR=Danger N=Cross-action will take place

2.2.4 DETERIORATION

The environment in which harvested produce is placed may greatly influence not only the respiration rate but also other changes and products formed in related chemical reactions. Changes in fruit are called ripening. Fruits such as pears and bananas require the ripening process to make them edible. The continuation of the ripening process leads to deterioration which softens the fruit, causing it to lose flavour and the tissues undergo a breakdown (NPCS, 2016).

The rate of deterioration is directly proportional to the temperature. The specific relationships between temperature and deterioration rate vary considerably among commodities. A generalisation, assuming that there is a deterioration rate of 1 for a fruit at −1 °C, is shown in Table 2.3 (NPCS, 2016).

Table 2.3: Approximate deterioration rate of fresh product. Source:(NPCS, 2016)

Temperature, ºC	Deterioration factor rate
20	8 to 10
10	4 to 5
5	3
3	2
0	1.25
-1	1

For example, a commodity that is marketable for 12 days at -1 ºC, might lose only four days (12/3=4) at 5 ºC. The proper temperature for slowing down the deterioration is where the fruit can be safely maintained without freezing or preserving the commodity, which is about 0.5 - 1 K above the freezing point of the commodity (NPCS, 2016).

Figure 2.3: Deterioration of peppers (Docrep)

2.3 PRESERVATION

In order to extend the life span or storage time of food to reduce the food loss, different methods have been proposed and these include, among others, hurdle technology, dehydration and refrigeration.

2.3.1 HURDLE TECHNOLOGY

Hurdle technology is a crucial concept for the mild preservation of food, as the hurdles in a stable product concertedly control microbial spoilage and food poisoning, leaving the designed fermentation processes unaffected (Leistner & Gorris, 1995: 41). Hurdle technology aims to apply an intelligent mix of hurdles to deterioration so that the total quality of foods is improved. This technology was motivated by the current consumer demand for more natural and fresh foods, which places pressure on food manufacturers to use only mild preservation techniques, for example, refrigeration, modified-atmosphere packaging and bio conservation (Leistner & Gorris, 1995: 41). The concept of food preservation is to use a hostile environment in which to put the microorganisms to shorten their survival and, therefore, lead to their death or inhibit their growth. Their feasible response to the hostile environment would determine, whether they grow or die. Recent advances have been made by considering the homeostasis, metabolic exhaustion and stress reactions of microorganisms in relation to hurdle technology, as well as by introducing the novel concept of multi-target preservation for a gentle but most effective preservation of hurdle-technology foods to avoid dehydration (Leistner, 1995a, 1995b).

2.3.2 DEHYDRATION

Dehydration is a common technique for the preservation of agricultural and other products, including fruits and vegetables. In developing countries, for example, the traditional method of dehydration is by open air drying, which often results in food contamination and nutritional deterioration (Ratti & Mujumdar, 1997). Pineapples, for instance, are an important crop in tropical countries and are eaten fresh or mostly canned, as juice or other products (Tanaka, Hilary & Ishzaki, 1999). However, technologies for canning or storing fresh fruits are very limited in developing countries. In addition, farmers face problems in shipping fresh fruits from rural areas to potential markets in urban areas of these countries, resulting in the spoilage of the fresh produce and economic loss because of limited or a complete lack of refrigeration facilities or systems (Madhlopa & Ngwalo, 2007: 455).

2.3.3 REFRIGERATION

The principle of refrigeration is the transferral of heat from a space to be cooled by an electrical motor. Boiling fluids with low temperatures are called refrigerants, where the thermal energy is absorbed and vaporised in the evaporator in the region where it is being cooled (Neuburger, 2003). Moving heat from one location to another is the process of refrigeration (Madhlopa & Ngwalo, 2007). Mechanical work is the traditional way of transferring heat, but there are other methods such as, among others, the use of laser, electricity and magnetism. Refrigeration applications are unlimited and can include air-conditioning, cryogenics, industrial freezers and household refrigerators (Neuburger, 2003). Neuburger (2003) points out that the heat pumps are similar to refrigeration units, where the heat pumps use the heat output of the refrigeration process and could be designed to be reversible. The literature review on refrigeration classifies refrigeration systems under the following themes: vapour-absorption refrigeration systems, vapour-compression refrigeration systems and vapour compression-absorption refrigeration systems.

2.3.3.1 VAPOUR-ABSORPTION REFRIGERATION SYSTEM

The attraction of the vapour-absorption refrigeration systems (VARS) method is that it has a very small liquid pump for the pressure difference as a moving part and utilises low-grade energy directly for cooling. This is a significant advantage compared to the high-grade energy conventional vapour-compression system. According to Harman, Webster and Farsad (2013), a vapour-absorption system comprises an evacuation chamber adapted to receive a secondary liquid and a vacuum pump which operates on the evacuation chamber reducing gas pressure within the chamber.

This operation promotes vaporisation of the secondary liquid. The vacuum pump is operated by means of a primary liquid passing through the vacuum pump. The vacuum pump is configured to enable the primary liquid to receive the vapour from the secondary liquid and to cause the vapour to condense within the primary liquid, providing condensed liquid mixed with the primary liquid. The absorption of vapour within the system is effective enough to cause the production of more vapour (Harman *et al*., 2013).

Figure 2.4: A VARS of 2700 kW at −30 ◦C installed in a refinery in Germany (Dinçer & Kanaĝlu, 2011,183-b)

However, the refrigerant flows from the condenser to the evaporator and then to the generator via the absorber and back to the condenser, while the absorbent flows from the absorber to the generator and back to the absorber. For maximum efficiency, the pressure difference between the low and high pressures should be as small as possible (Dinçer and Kanoĝlu, 2011).

Researchers such as Ataer and Gögüs (1991), Alizadeh, Bahar and Geoola (1997) and Morosuk and Tsatsaronis (2008) have carried out studies in the field of vapour-absorption refrigeration, using different working pairs with the most common being LiBr-H2O and NH3-H_2O.

The water-lithium bromide refrigeration cycle was an optimisation and theoretical study by Alizadeh *et al.* (1997). The conclusion was that a high cooling ratio is directly proportional to the high generator temperature, with a small surface for heat exchange and low cost (Alizadeh *et al.*, 1997).

Tyagi (1988) carried out a detailed study on the aqua-ammonia vapour-absorption refrigeration system (VARS) and plotted the coefficient of performance and mass flow rates as a function of operating parameters (for example, the temperatures of the generator, evaporator and absorber). Tyagi established that the temperature of the evaporator, absorber, condenser and generator are the base functions of the coefficient of performance (COP) and the work carried out added to their properties of binary solution.

Ataer and Gögüs (1991) indicate that the second law of thermodynamic analysis is the basis of irreversibility in components of the aqua-ammonia absorption refrigeration system. Hence, they calculate the dimensionless exergy loss for each element, energetic coefficients of performance and circulation ratios for different generators, absorber evaporators and condenser temperature. A high ammonia concentration needs a rectifier for the aqua-ammonia system in cases where it would lead to extra losses of exergy in the system. Their observation is that absorber loss is the direct result of the highest exergy loss in the evaporator (Ataer & Gögüs, 1991).

Bell, Al-Daini, Al-Ali, Habib, Abdel-Gayed and Duckers (1996) experimented with a solar energy-driven absorption cooling-system called a LiBr-H20. To observe all the processes, evacuated glass cylinders housed the components of the system to determine the thermodynamic performance by applying mass and energy balance for all the elements.

However, their work was based on the assumption that working fluids are in equilibrium and the temperature of the working fluid leaving the generator and absorber is equal to the temperature of the generator and absorber respectively. The conclusion was that the system COP depends on the temperature of the generator and the optimum value of the generator temperature is where COP is maximised. Additionally, COP is obtained at the low temperature of 68 ºC, when the condenser and the absorber of the system operate at the same temperature.

Morosuk and Tsatsaronis (2008) used an absorption refrigeration machine to represent splitting the exergy destruction into endogenous/exogenous and unavoidable/avoidable parts. This was a new development in the exergy analysis of energy conversion systems. Their conclusion was that additional useful information would be supplied by an absorption refrigeration machine of advanced energetic evaluation, where exergy analysis cannot be undertaken or is not possible.

2.3.3.2 VAPOUR-COMPRESSION REFRIGERATION SYSTEM

Wightman (2002) points out that, in a closed-loop vapour-compression cycle, the heat transfer fluid changes state from a vapour to a liquid in the condenser, giving off heat and then changes state from a liquid to a vapour in the evaporator, absorbing the heat during vaporisation. A typical vapour-compression refrigeration system (VCRS) includes a compressor for pumping the heat transfer fluid, such as Freon, to a condenser, where heat is produced as the vapour condenses into a liquid. The liquid flows through a line to a thermostatic expansion valve, while the heat transfer fluid undergoes a volumetric expansion. The heat transfer fluid, which then exits from the thermostatic expansion valve, is a low-quality liquid-vapour mixture (Wightman, 2002).

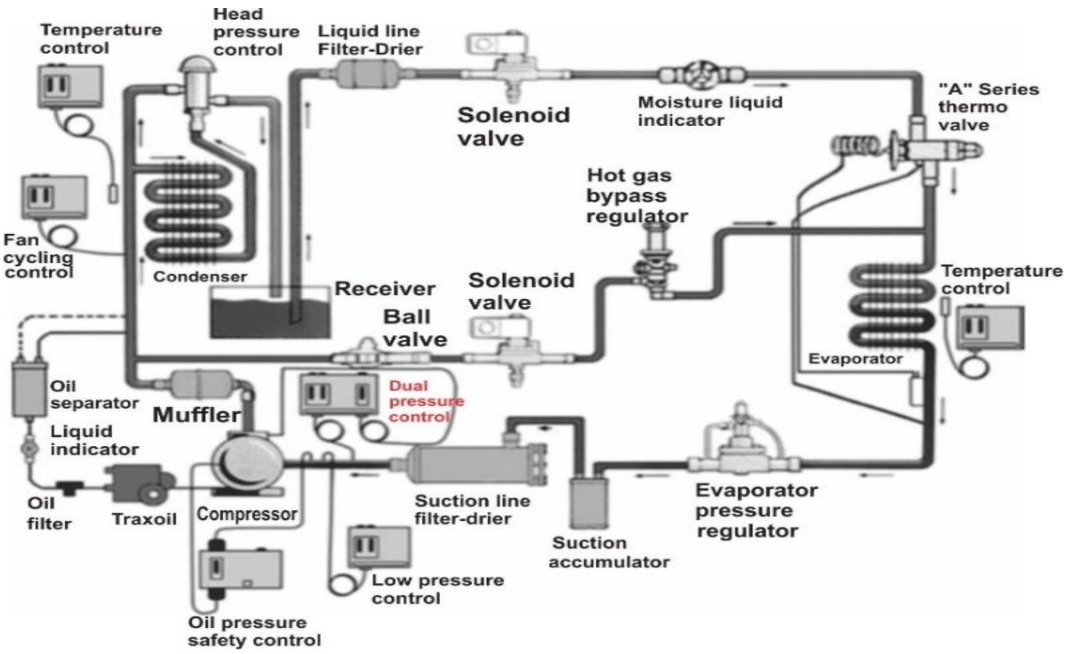


Figure 2.5: A (VCRS) with the control devices (Dinçer & Kanaĝlu, 2011, 142).

The CFC12 refrigerant in a two-stage vapour-compression refrigeration system's (VCRS) optimum inter-stage pressure was determined by Keshwani and Rastogi (1968). Their research concentrated on the minimisation of the work done by the compressor between the stages, with or without intercooling. The principle was discussed by Katz (1962) and was used by Arora and Dhar (1971) for solving R-12 compression systems for the multi-stages of an optimum inter-stage pressure problem. They concluded that the optimum inter-stage pressure approximately equals the geometric means of the condensation and evaporation pressure but when the flash intercooler was incorporated, they found a considerable difference between the geometric means and the optimal pressure values.

R-11 and R-12, used as a working fluid for a system of vapour-compression refrigeration and its method of carrying out exergy analysis, was demonstrated by Kumar, Prevost and Bugarel (1989). The exergy-enthalpy diagram was presented to facilitate the analysis. They found that the system demonstrated that different components showed various losses during the calculation procedure.

Zubair and Khan (1995) indicated that the optimum inter-stage pressure for a two-stage refrigeration system can be approximated by the saturation pressure corresponding to the arithmetic mean of the condensation and evaporation temperatures.

R-502 is the proper and a suitable refrigerant for a refrigeration system of vapour-compression where widely used for supermarket and other cold storage. The Zeotropic mixture of R-22 and R-115 makes up the components for R-502.

2.3.3.3 VAPOUR COMPRESSION-ABSORPTION REFRIGERATION SYSTEM

The principle of heat pump or vapour compression-absorption refrigeration (VCARS) is the mechanical compression of the vapour. This vapour is then desorbed after being absorbed, where a cycle of liquid solution is used. The compression and the absorption of vapour by those systems would be considered as being hybrid systems. These two types of systems combine the concepts of a heat pump, the compression and the absorption. The refrigerants mixture is the working fluid serving as absorbent and desorbent for both of them. The mixture availability of the range extent of the temperature is the advantage of the hybrid heat pump over the pure refrigerant. Another advantage is the irreversibility factor reduced during the process of heat exchanges between the working fluids, leading to the improvement of the performance of the system due to temperature gliding in both the absorber and desorber.

Pourreza-Djourshari and Radermacher (1996) obtained the performance characteristics for two vapour-compression heat pump cycles with a single-stage and two-stage circuit. R-22-DEGDME was the working fluid, significantly increasing the COP compared to R-22. The results indicate that there is a potential to control the capacity by a ratio of 7:1, energy saving of up to 50% and significant reduction in pressure ratio compared to a conventional R-22 cycle. Radermacher (1987) examined the performance of a vapour-compression heat pump cycle with desorber/absorber heat exchange working on an R-22-R-113 mixture, using the successive-substitution method. The results showed an improvement in the cooling COP by 57% with a 69% pressure ratio reduction in comparison to a conventional R-22 cycle.

The process of overall heat-transfer resistance was included in the first simulation model presented by Stokar and Trepp (1987). The calculation of heat-transfer resistance was introduced as a function of the mass-flow rate for the working pair NH_3-H_2O from the experimental data. The achievement of 23% of energy saving was the result from testing the plant by heating water from 40 C° - 70 ºC and from 40 ºC - 15 ºC as the water cools.

A study on a compression-absorption heat bump with R-22-Dimethy1 form during performance was undertaken by George, Marx and Srinivasa Murthy (1989) using a thermodynamic analysis. The heating COP, concentration difference and the circulation ratio were calculated by varying the compression ratio and operating temperatures during absorption and desorbtion. The assumptions were that the absorbent does not evaporate sufficiently in the considered temperature range to necessitate rectification.

At the desorber/absorber exit point, the condition of equilibrium exists, which means there is 100% heat exchanger effectiveness in the compressor. There is isotropic compression in the pressure-reducing valve as well as isenthalpic expansion and less heat loss, causing a drop in pressure in the various components. George, J.M., Marx, W. & Srinivasa Murthy's (1989) conclusion was that a temperature of 60 ºC with a high CPU of six conditions was achieved.

An optimisation study was done by Ǎhlby, Hodgett and Rademacher (1993) on the cycle of compression-absorption, with NH_3-H2O as a working fluid mixture. The performance of the cycle was improved by the optimisation of the absorber temperature gradient where consideration was given to small external temperature gradient situations in particular. The analysis of the assumptions was that at the outlets of both the absorber and the desorber; adiabatic absorption and desorption takes place respectively until reaching the state of equilibrium. For the heat exchange UA (overall heat transfer coefficient ($W/(m^2 \cdot K)$* heat transfer surface area (m^2)) value constant, they found that there were neither pressure drops nor heat losses. Therefore, the operation's optimum point was found by studying the changes in the compressor, the pump and observing the heat losses in the solution heat exchanger with the changes taking place when operational. Their conclusion was that an optimum point could be found for the working conditions of each external situation.

A working fluid, NH_3-H_2O-LiBr, was used to study the compression-absorption heat pump performance. The 10% cycle performance performed better than the binary fluid NH_3-H_2O, as shown by the ternary mixture with 60% of mass salt concentration. The estimation from NH_3-H_2O-60% LiBr and NH_3-H_2O properties have not been validated by experimentation, therefore, the calculation cannot be considered reliable. A salty solution of 40- 50% by mass would be reliable and the best mixture, as indicated by the results.

Solutions of two different versions of working fluid were compared in performance. They were circuited in the two-stage vapour compression heat pump. This experiment was carried out by Rane, Amrane and Radermacher (1993). A cascade system was presented. The mass and heat balance for each component were the basis for the computer simulation of the model developed. The input parameters were *UA* values for the calculations of heat exchange. The cycle performance was compared and it was established that the cycle of the bleed line and super heater provided 40-50% *COP,* higher than the cycle with the rectifier. Various weak parameters of solution-concentration were studied for the parameters of cooling COP, effectiveness of heat-exchange solution, ratio of pressure, glides of temperature in absorber and desorber and low-temperature desorber load. The results indicate the system above could operate at 100 ºC and more than 100K lift in temperature was achieved.

Itard and Machielsen (1994) investigated the problems encountered when modelling compression/resorption heat pumps. For large temperature glides, the Log Mean Temperature Difference (LMTD) method was used less for heat-exchange modelling and the calculations of COP. Their conclusion was that pure refrigerant was less advantageous than a mixture, and, for certain external conditions, an optimum overall concentration existed that was the determinant factor for the COP of the system.

A nitrate absorption/compression refrigeration cycle of ammonia/lithium study was carried out by Ayala, Heard and Holland (1997). With a range from 0 to 100, the mechanical vapour compressions were modelled and different generations of power and efficiencies of distribution were applied to drive the primary ratio of energy. The heat loss of zero was the main consideration for the hybrid model assumption, except that, in the generator, the work of the pump was excluded. It was found that an achievement of 10% in overall efficiency increase is possible by using the absorption/compression refrigeration system.

Kairouani and Nahdi (2006) developed a novel combined refrigeration system with the purpose of discussing the thermodynamic analysis of the cycle. A study of the possibility of the use of geothermal energy for the hybrid system, was carried out, where R-22, and R-134 and R-717 were selected as working fluids for the conventional ammonia-water pair absorption system. A range of 343 - 349K was the geothermal temperature source and the result was that the COP for the combined system was significantly high compared to the COP of a single-stage refrigeration system. This would lead to a reduction of the consumption of electrical energy.

A thermodynamic investigation comparison was carried out by Satapathy, Gopal and Arora (2007) on R-22-E181 and R-134a-E181 as working pairs for the vapour compression-absorption system for cooling and heating applications. With the low-concentration solution and low-system capacity, the R-134a-181 provides a better performance, while with the high-solution concentration and high-system capacity, R-2-E181 is better than R-134a-E181. Concluded that for higher volumetric capacities, R-22 was considered to be the proper refrigerant to be used because it could be used for many years, but, due to the use of non-optimisational components, the actual performance would be poor.

2.4 REFRIGERATION SUMMARY

A refrigeration system is a combination of components and equipment connected in a sequential order to produce the desired refrigeration effect for cooling or heating (Klein, 1992).

The refrigeration systems could be classified into three themes:

- Vapour absorption refrigeration systems (VARS)
- Vapours compression refrigeration system (VCRS)
- Vapour compression-absorption refrigeration system (VCARS)

As literature has indicated, refrigerators are designed in stages or chambers.

Refrigeration applications touch our daily lives in deferent forms such as: household refrigerator, commercial refrigerator as in the supermarket display, restaurants, and cafeterias, food processing like refrigerated warehouse, industrial tools for liquefaction of gases, chemical process cooling, crystallization and transport refrigeration for transporting the product from place to another.

CHAPTER THREE
DESIGN CONCEPT

3.1 PROLOGUE

In this study, six-chamber refrigeration has been selected with the understanding that if each chamber has its own temperature, cold air supplies, thermostat and its own door, this would reduce the testing time to reach the proper temperature for storage of the products six times less than that in the experiments.

3.2 DESIGN CONSIDERATIONS

Refrigeration can be described as the process of cooling materials where the heat is transferred from a lower to a higher temperature by a process such as mechanical work. The temperature may span the range of -157 ºC to +4 ºC. The study of the production of ultra-low temperature cooling from absolute zero, -273 ºC to -157 ºC, is commonly referred to as cryogenics. This study will focus on how to extract the heat from the products in each cooling chamber and keep them at the required temperature, where each thermostat of each chamber is set.

Having the main cooling chamber below the experimental chambers will lead to a violation of the air properties. Therefore the cooled air has to be forced to flow into the experimental chambers. Fans are the key, but there is also the issue of the heat that is produced by them. The calculation of the heat loss is another issue to be taken into account. If the chambers are arranged so that they are separated, this will allow each to be surrounded with the same air temperature on all sides and the calculation of one of the four sides of the chamber will cover the other three sides. Chapter Four will describe this in detail.

The ejected air from the experimental chambers has to be directed to the bottom of the deep freezer for larger amount of heat exchange between cold and hot air.

3.3 THE PROTOTYPE COMPONENTS

The prototype is made up of six similar chambers, base and back cover.

3.3.1 THE TESTING CHAMBER

The testing chamber is made up as follows: -

1. The outer container: This is a box measuring 300×300 mm with a depth of 350 mm and is made of 6 mm-thick wood material which serves to contain the rest of the components of each chamber. There is a hole 100 mm in diameter in the top corner of the back of the chamber and another hole 50 mm in diameter in the opposite bottom corner of the back (Figure 3.1).

Figure 3.1 The outer container

2. The plastic box: This is a box measuring 200×200×300 mm in width, height and depth respectively. The sides were cut with a laser-cutting machine at the CPUT Project Lab as in Figure 3.2(a), and assembled with the use of super glue to join the sides together.

3. The air tray: Two plastic strips with a height of 25 mm and a rectangular piece measures 200×330 mm. The two strips will be lifting the rectangular piece from the bottom by 25 mm to allow the ejected air to pass out of the experimental chamber. All the components are glued together with super glue. This is illustrated in Figure 3.2(b).

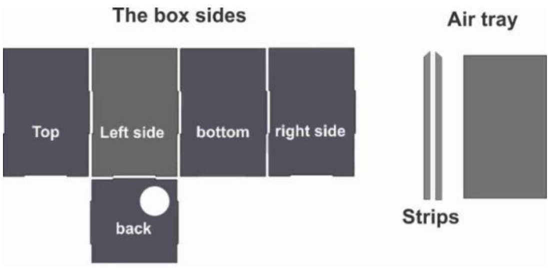


Figure 3.2(a): The construction of the chamber's plastic box

Figure 3.2(b): The chamber's plastic box after assemblage

4. The sheet metal rim: A piece of square sheet metal with dimensions of 256×256 mm, and square cut at the medal with dimensions of 206×206 mm with left-over of 150 mm in length and 20 mm in width for fitting the rim to the body of the chamber . The inner and outer edges are bent in the same direction. Figure 3.3 illustrates the layout of the design. It needs to be taken into consideration that the material of the rim is magnetic ‘ferrite’.

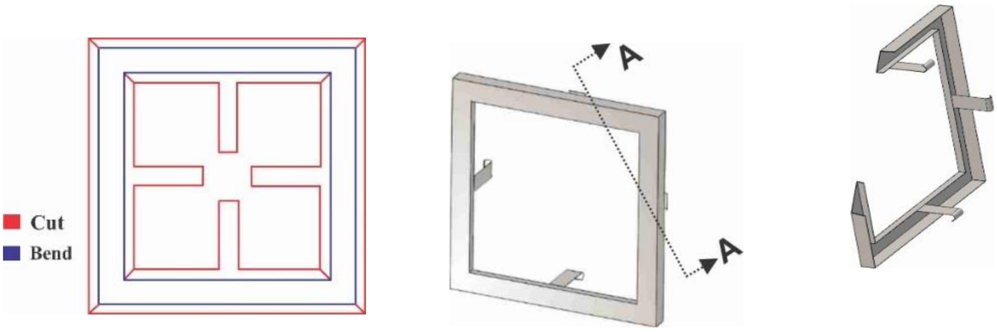


Figure 3.3: The layout of the cut and bend of the sheet metal plate.

6. The digital thermometer-holder: The digital thermometer holder is made up of two pieces of plastic with the dimensions of 80×50 mm cut with the laser-cutter machine. The inner piece is 5 mm thick from which 60×30 windows are cut out. The second piece is 3 mm in thickness with 44×24 mm windows cut out. These pieces are stuck together with super glue as demonstrated in figure 3.4 below.

Figure 3.4: The digital thermometer-holder components.

7. The door of the cooling chamber: Two layers of transparent plastic material, 50 mm in thickness and with the dimensions of 270×270 mm, are fitted with a rubber gasket to the inner portion to seal the chamber tightly to prevent air from leaking. Both of these layers are connected by a double-sided self-adhesive foam strip, 16 mm thick and 15 mm wide, so that the two layers are separated by the thickness of the foam, making the door heat resistant (Figure 3.5).

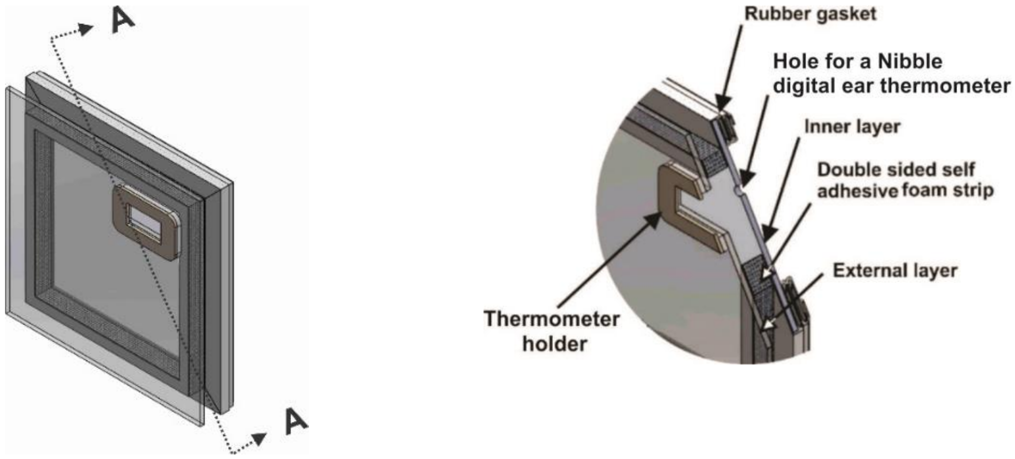


Figure 3.5: The door of the cooling chamber

8. Inlet and outlet: the outlet is a 100 mm plastic tube, 150 mm in length, and is connected to the edge of the plastic box through the wood shield. The inlet is 50 mm in diameter with a 90º elbow connected to the outer container at the inlet hole and the 50 mm diameter flexible tube is linked to the other end of the elbow and left hanging inside the main cooling chamber, which is referred to as the deep freezer.

9. The electrical unit: A small fan measuring 100 mm×100 mm is connected at the end of the inlet tube inside the plastic box. The fan speed is controlled by a sensitive thermostat. There is a Nibble digital ear thermometer installed in the thermometer holder at the chamber door. Figure 3.6 illustrates the testing chamber after it has been assembled.

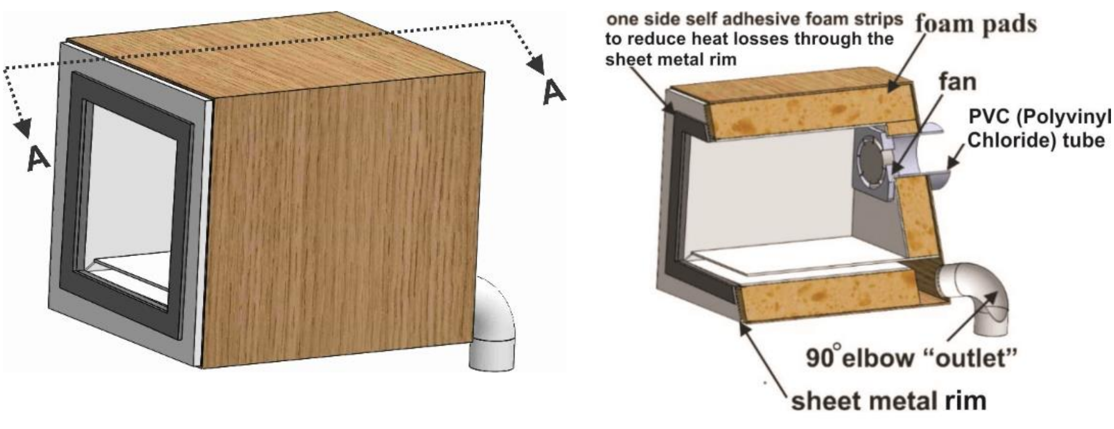
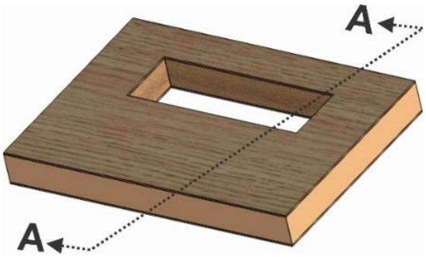


Figure 3.6: The testing chamber body

3.3.2 THE BASE

The base has a top and bottom made from two layers of 6 mm thick plywood to provide enough strength and its dimensions are 1000×700 mm. The window dimension is 200×600 mm in both layers, separated by two frames, outer frame and the inner frame by the size of the windows on the plywood layers. Both frames are made from pin bar wood, measuring 25 ×100 mm. The rest of the shallow filled with a block of foam, 100 mm thick, cut to shape and placed between the layers for insulation (Figure 3.7).



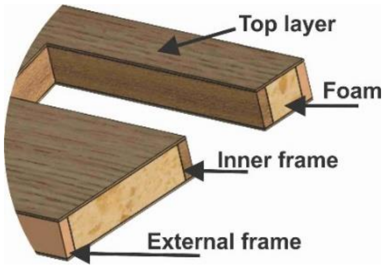


Figure 3.7: The base

3.3.3 THE BACK SPACE COVER

The cover is constructed from two right-angled triangular pieces of wood, connected at their hypotenuse with a rectangular piece of the same wood to form the shape of a pyramid. This shape is covered from the inside by a layer of foam, 50 mm thick, as insulation. This component of the prototype serves to cover the back of the chambers and the window on the base. Thus the cover, the base and the back of the six chambers will keep the air circulating in that space. Figure 3.8 illustrates the back space cover.

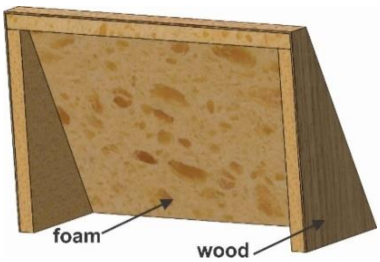


Figure 3.8: The back cover

Note: All the components are made of wood and varnished to be water and moisture-proof.

3.4 THE DESIGN EQUATION

The purpose of the construction is to suck up cold air from the bottom of the deep freezer, forcing it into the experimental chamber. This is called the forced-convection method and cools the experimental chamber space and brings it down to the desired temperature (Figure 3.9). An accurate thermometer is installed in each chamber to control the fan speed.

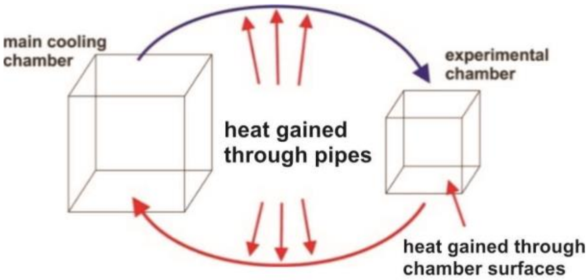


Figure 3.9: The cooling cycle diagram

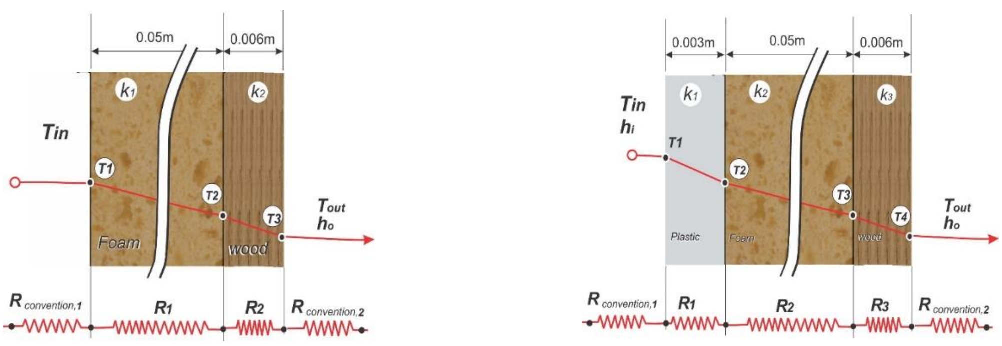
3.4.1 BASIC DATA CALCULATIONS FOR EFFECTIVE AREA FOR HEAT LOSS

The transfer of heat is determined by the contact surface area. Therefore, all the areas of the prototype, except the deep freezer, will have to be calculated accurately. See Appendix 1 which represents the basic calculations for the area of the prototype.

3.4.2 THE HEAT LOSS RESISTANCE CALCULATIONS

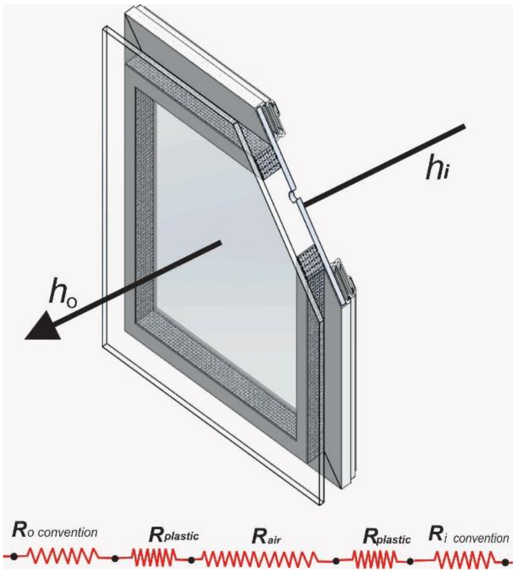
The resistance equations of the heat-loss calculations for the back cover and sides of the experimental chambers are demonstrated in Figures 3.10 and 3.11 below.

The back space cover | Experimental chambers



$R_{total} = R_{conv,1} + R_{foam} + R_{wood} + R_{conv,2}$ $\quad$ $R_{total} = R_{conv,1} + R_{Plastic} + R_{foam} + R_{wood} + R_{conv,2}$

Figure 3.10 The resistance equations for the sides of the chamber



$$R_{total} = R_{conv,i} + R_{plastic} + R_{air} + R_{plastic} + R_{conv,o}$$

Figure 3.11: The resistance equations for the chamber door

3.4.3 DEFINITIONS OF HEAT LOSS AND LOAD DETERMINANTS

According to ASHRAE (2002), there are five stages in the refrigeration of the total load as listed below:

- Heat transferred into the refrigeration space through its surface is called the transmission load.
- Heat removed from the product or produced by the product is called the product load.
- Heat produced by internal sources such a fan, lights, etc. is called the internal load.
- Heat gained by transferring air to the refrigeration space is called the infiltration air load.
- Equipment-related load such as fan motors, defrost heater, humidity control.

3.4.4.1 THE TRANSMISSION LOAD

According to the ASHRAE (2002) handbook, the calculation of transmission load equations is:

$$q = UA\Delta t \tag{3.1}$$

Where:

q = heat gain W

A = Average surface area for heat transfer, where the outer and inner surface area divided by 2. The unit is m^2.

Δt = Difference between outside air temperature and air temperature of refrigeration space $°C$.

U = overall thermal coefficient of heat transfer of wall, W/ (m^2.K)

$$U = \frac{1}{h_o \frac{A_O}{A} + {}^{x_1}/_{k_1} + {}^{x_2}/_{k_2} + {}^{x_3}/_{k_3} + h_i\, A_i} \qquad (3.2)$$

Where:

X = the layer thickness of the wall m.

k= thermal conductivity of material layers of the wall, W (m.k)

h_i = inside surface conductivity of the wall, W (m^2.K)

h_o = outside surface conductance, W (m^2.K)

A_o = outside surface area of the wall, m^2

A_i = inside surface area of the wall, m^2

Note: The effective area is determined by the average of the outside area and the inside area

$$\text{The average area} = \frac{A_{in} + A_{out}}{2} \qquad (3.3)$$

The inlet area where the fan is located is a circle, 100 mm in diameter, cut out of the back wall of the chamber, therefore.

$$\text{The area of the circle} = A = \pi r^2 \qquad (3.4)$$

3.4.4.2 THE INTERNAL LOAD

There is generally no other internal load beside the fan’s load, therefore,

$$Q = n \times P \times t \qquad (3.5)$$

Q= heat produced by the fan J (P [W] * t [s])

n= number of fans

t= time of operation in seconds

P = the fan power (watts)

3.4.4.3 THE PRODUCT LOAD

Certain foodstuffs in storage are living organisms and give off heat as their starch or sugar reserves are slowly being consumed. Oxygen is consumed for the process. This is called heat of respiration.

According to ASHRAE (2002), for the product load calculation, there are four phases of heat-removal during the cooling process of the products:

1- Cooling to some lower temperature from the initial temperature above freezing:

$$Q_1 = mc_1\ (T_1 - T_2) \qquad (3.6)$$

2- Cooling to the freezing point of the product from the initial temperature:

$$Q_2 = mc_1\ (T_1 - T_f) \qquad (3.7)$$

3- Cooling to freeze the product:

$$Q_3 = mh_{if} \qquad (3.8)$$

4- Cooling to the final temperature below the freezing point from the freezing point:

$$Q_4 = mc_2\ (T_f - T_3) \qquad (3.9)$$

Where:

Q_1, Q_2, Q_3, Q_4 = heat removed, kJ

m = mass of product, kg

c_1 = specific heat of product above freezing, kJ/ (kg·K)

T_1 = initial temperature of product above freezing, °C

T_2 = lower temperature of product above freezing, °C

T_f= freezing temperature of product, °C

h_{if}= latent heat of fusion of product, kJ/kg

c_2= specific heat of product below freezing, kJ/ (kg·K)

T_3= final temperature of product below freezing, °C

Thus, the equation for the refrigeration load required for the product is:

$$q = \frac{\sum Q}{3600\, n} \qquad (3.10)$$

q = average cooling load, kW

n = allotted time, h

Note: This is the basic calculation formula. For a sample of the calculations, see the appendices 1 and more equations for heat-transfer calculations.

CHAPTER FOUR
READINESS, TESTING AND OBSERVATIONS

4.1 READINESS

The prototype components have to be assembled to be ready for testing. The components are the base, six chambers, back space cover and chest deep freezer.

1. The Base: the base was assembled by removing the original cover of a deep freezer, then using 25 mm high and 50 mm width one-sided self-adhesive foam strips cut to the length and the width of the freezer. These strips were placed along the edges with the adhesive side facing up. Before placing the base on the top of the deep freezer, the underside of the base was covered with a self-adhesive aluminium sheet that is the size of the top of the deep freezer after taking the lid off. This will allow the heat to be reflected in the deep freeze and create resistance to the transfer of heat through the base, even though the insulation in the base is 100 mm thick foam as an additional precaution. The next step is to remove the protection tape of the one-sided self-adhesive 25×50 mm foam strips which have already been placed on the edge of the top of the deep freezer. The base must then be placed on top of the top of the deep freezer, onto the tape of the self-adhesive foam. It is essential that this is symmetrical, as now the base acts as the deep freezer door except that the window in the base allows for the circulation of air between the deep freezer, which acts as the main cooling chamber, through the window in the base up to the space created by the back cover into the chamber, then to the main cooling chamber through the out let (as mentioned in Chapter Three, Section 3.4.3). See Figure 4.1.

Note: one or two thermometers could be installed in the base for monitoring the temperature of the deep freezer (main cooling chamber).

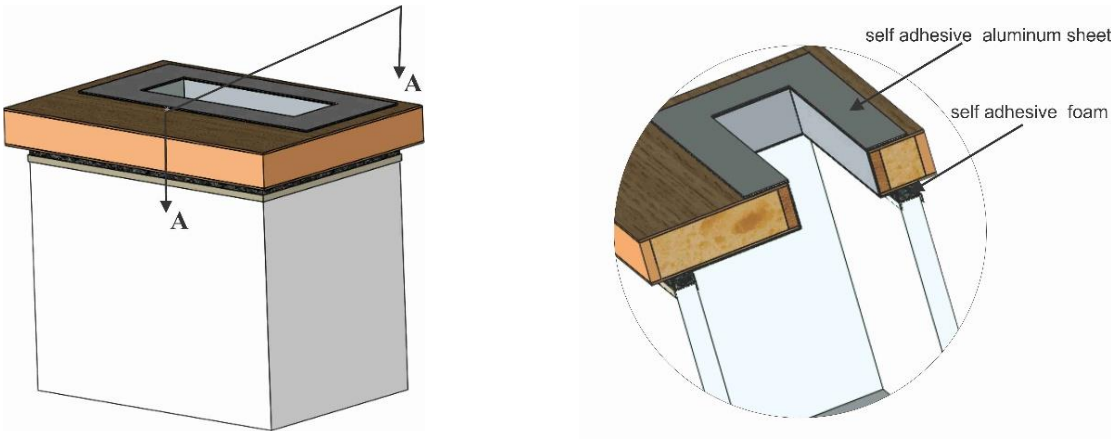


Figure 4.1: Attaching the base to the chest deep freezer.

2. The chambers: six chambers have to be arranged so that there are only two rows with three columns. This is the best arrangement for reducing the height of the chambers as much as possible. Double-sided self-adhesive foam measuring 20×25 mm is used to separate the chambers from one another (see Chapter 3, Section 3.2). The outlet tubes are connected in place and suspended in the deep freeze or main cooling chamber. Figures 4.2 and 4.3 illustrate how the experimental chambers are assembled.

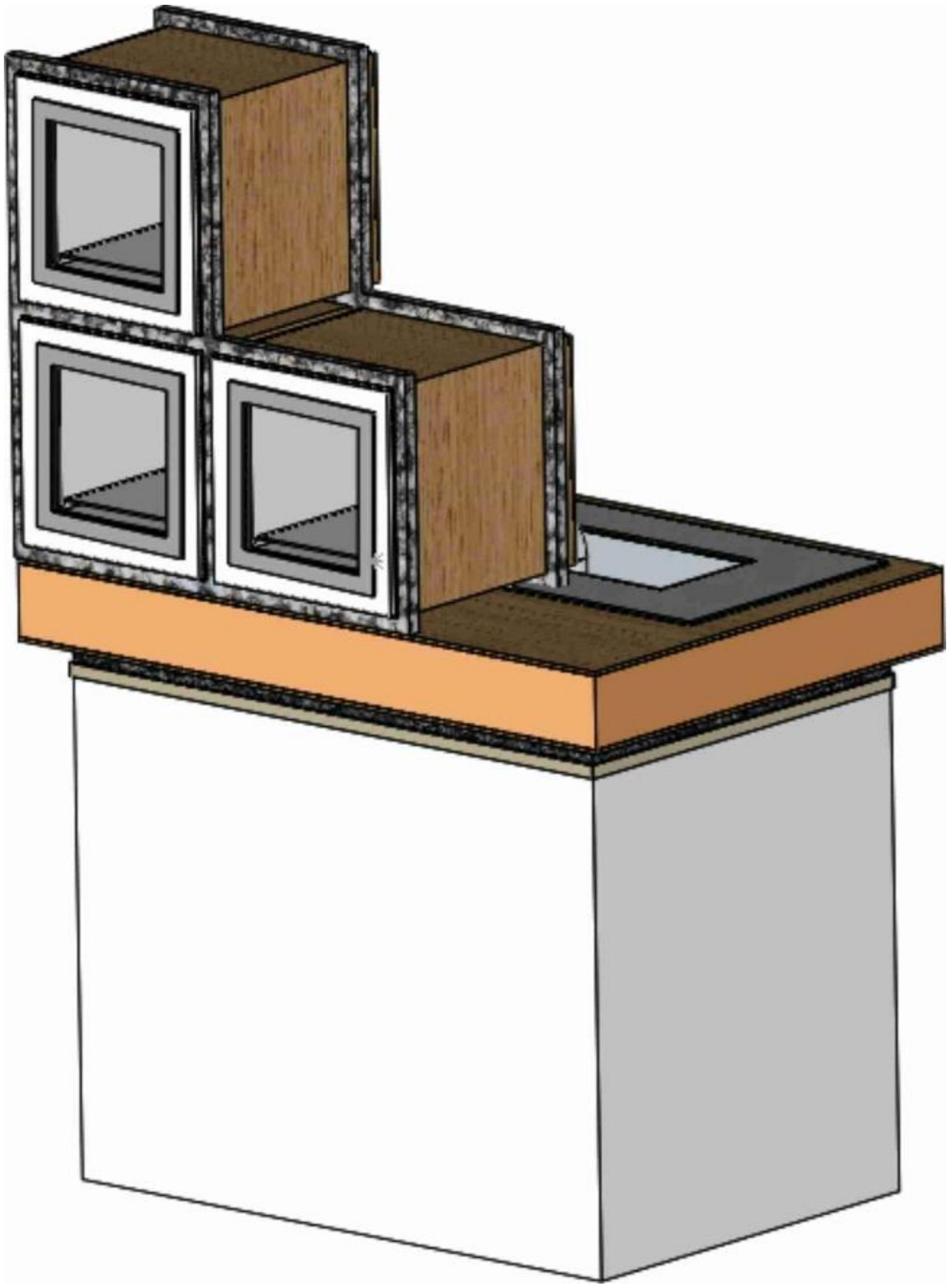


Figure 4.2: Attaching the experimental chambers to the base.

Note: the strips of the double sided self-adhesive foam at the back should not only serve as separators between the chambers, but also, along with the back of the chambers, as a cover for the cold air at the back of the chambers.

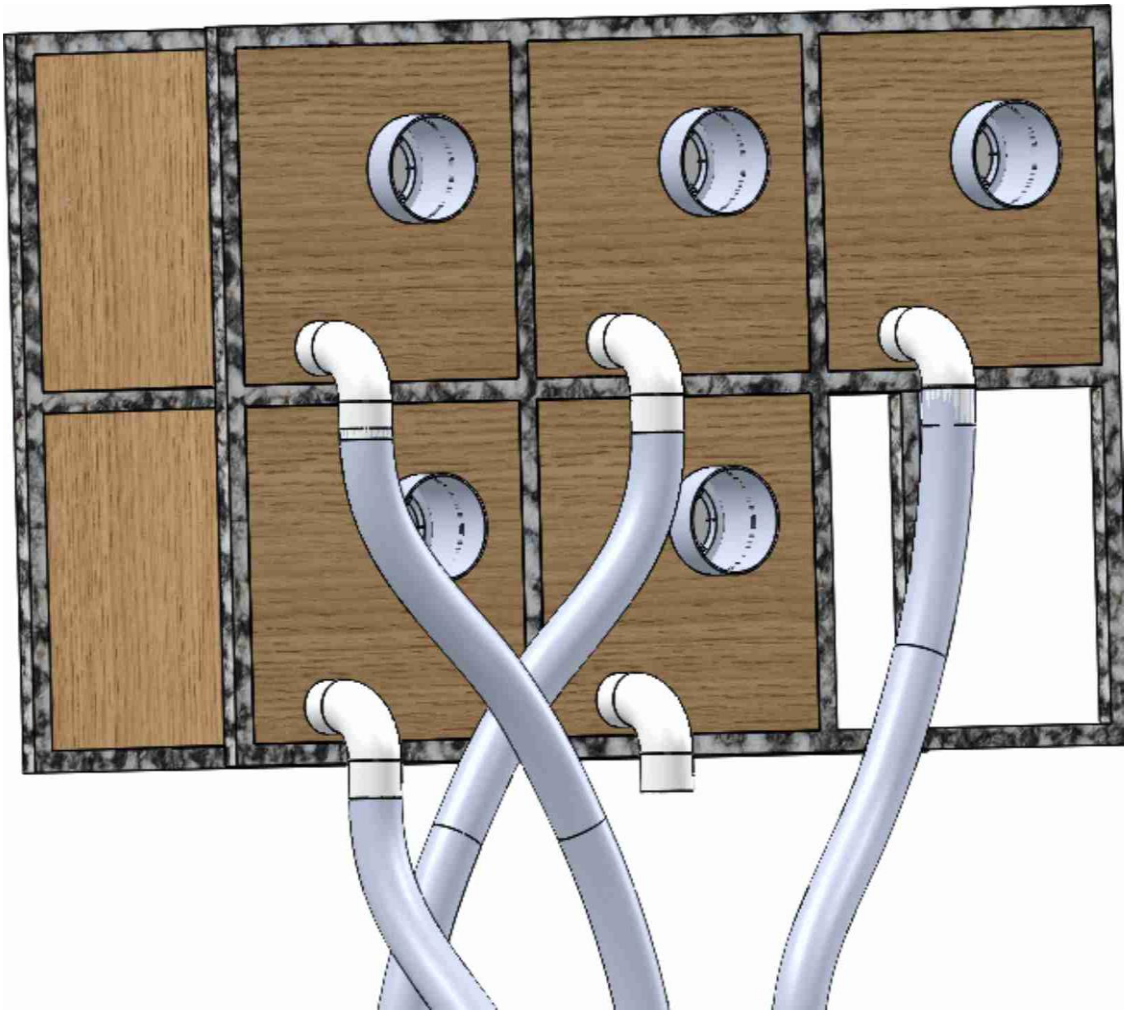


Figure 4.3: Back view of the chambers illustrating how some of the outlet tubes have been assembled.

3. The back cover: after the base and the chambers have been assembled, the cover is replaced in the space between the base and the chambers at their highest point where the air, which is delivered to the testing chamber, will be expelled back to the main chamber. Figure 4.4 demonstrates how the cover is assembled with the base and the testing chambers.

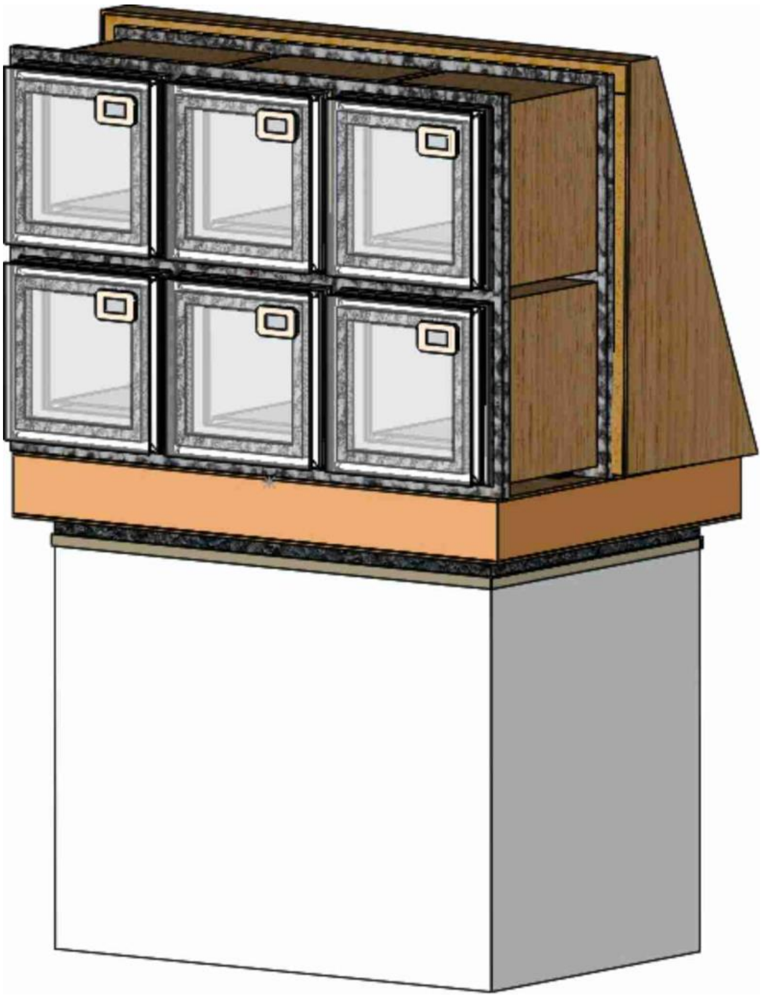


Figure 4.4: Attaching the back cover, the chambers and the base.

4.2 TESTING AND EXPERIMENTAL THE PROTOTYPE BEFORE LOADING

At this stage, the prototype is ready for testing and experimentation and the results are then collected and compared with one another.

Note: It would be wise to have a digital thermometer installed in the back cover to measure the temperature of the cold air at the back of the chambers.

Furthermore, the upper chambers are set to a higher temperature and the lower chambers to a lower temperature to comply with the principles of air movement, where hot air always rises compared to cold air which always falls. For further practice, some tests are run on the prototype and each one provides data. These tests must occur in the order as listed below:

1-Testing at 0 fan speed: the testing chambers' doors are placed on the sheet metal rim to close them. The temperature is recorded in each chamber. Initially, all chambers are at the same temperature. The deep freezer is then switched on and runs for 12 hours; to reach the lowest temperature and -30 ºC is recorded. This test demonstrates how much heat will be transferred to the chambers by comparing the first and last readings from the digital thermometers on the doors of the chambers, recording the temperatures of the deep freezer and that at the back cover space. If the temperature in the testing chamber standing still or has negligible change or slightly has dropped, this will be acceptable as, at this stage, the design of the prototype will be considered satisfactory. If there is a greater degree of change, the design for the control inlet will have to be modified.

2- Testing at maximum fan speed: this second test is done after the prototype has passed the previous test. The thermostat of each testing chamber is set to the same temperature for all the chambers and the fans switched on when the deep freezer temperature has reached its maximum. The minimum time for monitoring the drop in temperature for each of the testing chambers was 45 minutes to an hour. The test provided information about the cooling speeds for the testing chambers and whether the testing chambers would all cool down to the same temperature at the same time or not.

3- Testing at chamber temperature: this test follows if the previous test has been successful. The thermostat of each testing chamber is set to the desired temperature. The drop in temperature for the testing chambers is monitored every 30 minutes for four hours or longer, as this provides information regarding the question of how higher temperature chamber fan stoppages would affect the performance of lower ones.

These tests provide information for future research into the analysis of the design while the results of these experiments with this prototype and the design are regarded as the outputs of the current project. Figure 4.5 demonstrates the final prototype design.

Figure 4.5: The original built prototype.

4.3 OBSERVATIONS AND RESULTS OF TESTING

This section will demonstrate the results of the three tests applied to the prototype, bearing in mind that other designs have gone through the same tests, but this design is considered to be the final one.

1- **At 0 fan speed:** after 48 hours, the temperature of the main chamber was at its lowest. The result appeared to be satisfactory except for a negligible change in the air temperature of the back cover. Figure 4.6 demonstrates the first test result.

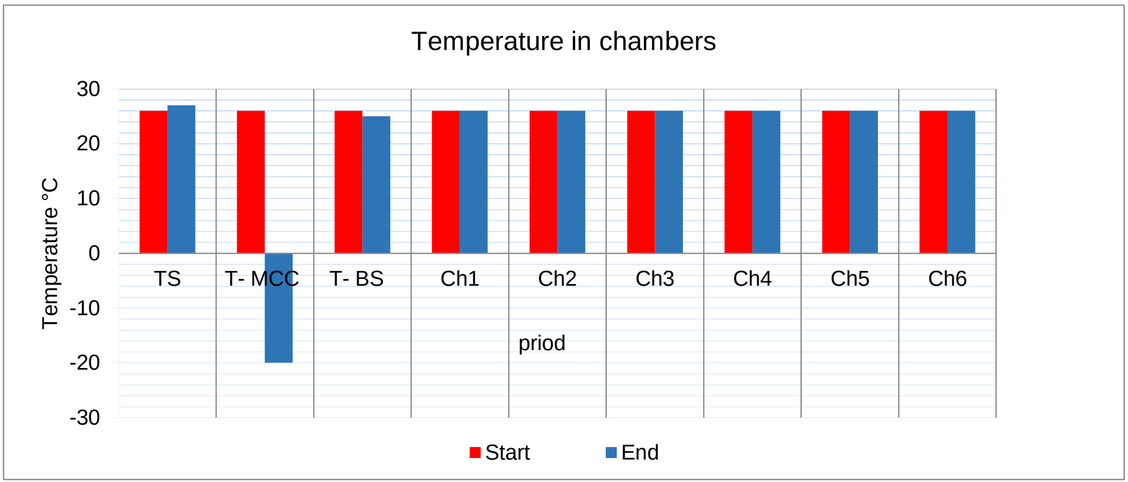


Figure 4.6: A graphic presentation of the first test results.

2- **At maximum fan speed:** when the temperature peaks in the main chamber, the fans of the chambers are turned on and the thermostat is set to 0 ºC, rapidly reducing the temperature of the chambers as well as of the back cover space. The second test was completed after 48 hours. The result of this test is demonstrated in Figure 4.7.

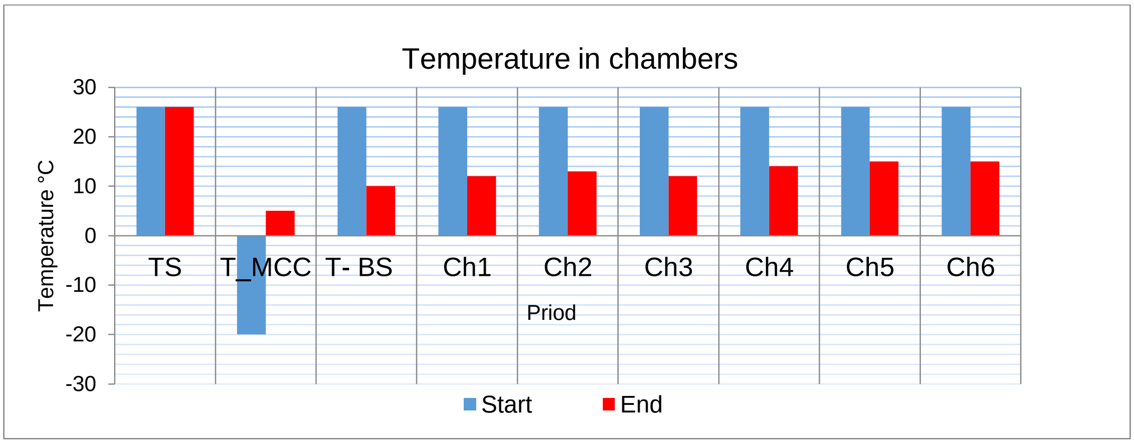


Figure 4.7: A graphic presentation of the second test results.

Exactly four more hours were given to this test, but the result never improved. Thus, if there is no heat loss, then the internal heat is produced by the fans in the chambers.

3- At chamber temperature: the temperature in the chamber is measured after the prototype has passed the two previous tests. Again, at the peak temperature of the main chamber, the ambient temperature on that day was 28 ºC. Each chamber was set at the desired temperature and the fans turned on. The temperatures of the chambers in the bottom row, namely chambers 1, 2 and 3, were 5 ºC, 7 ºC and 9 ºC respectively. The temperatures of the chambers in the top row, namely chambers 4, 5 and 6, were 11 ºC, 13 ºC and 15 ºC respectively. The chambers temperature was monitored every 30 minutes.

The chart in Figure 4.8 demonstrates the result of the test.

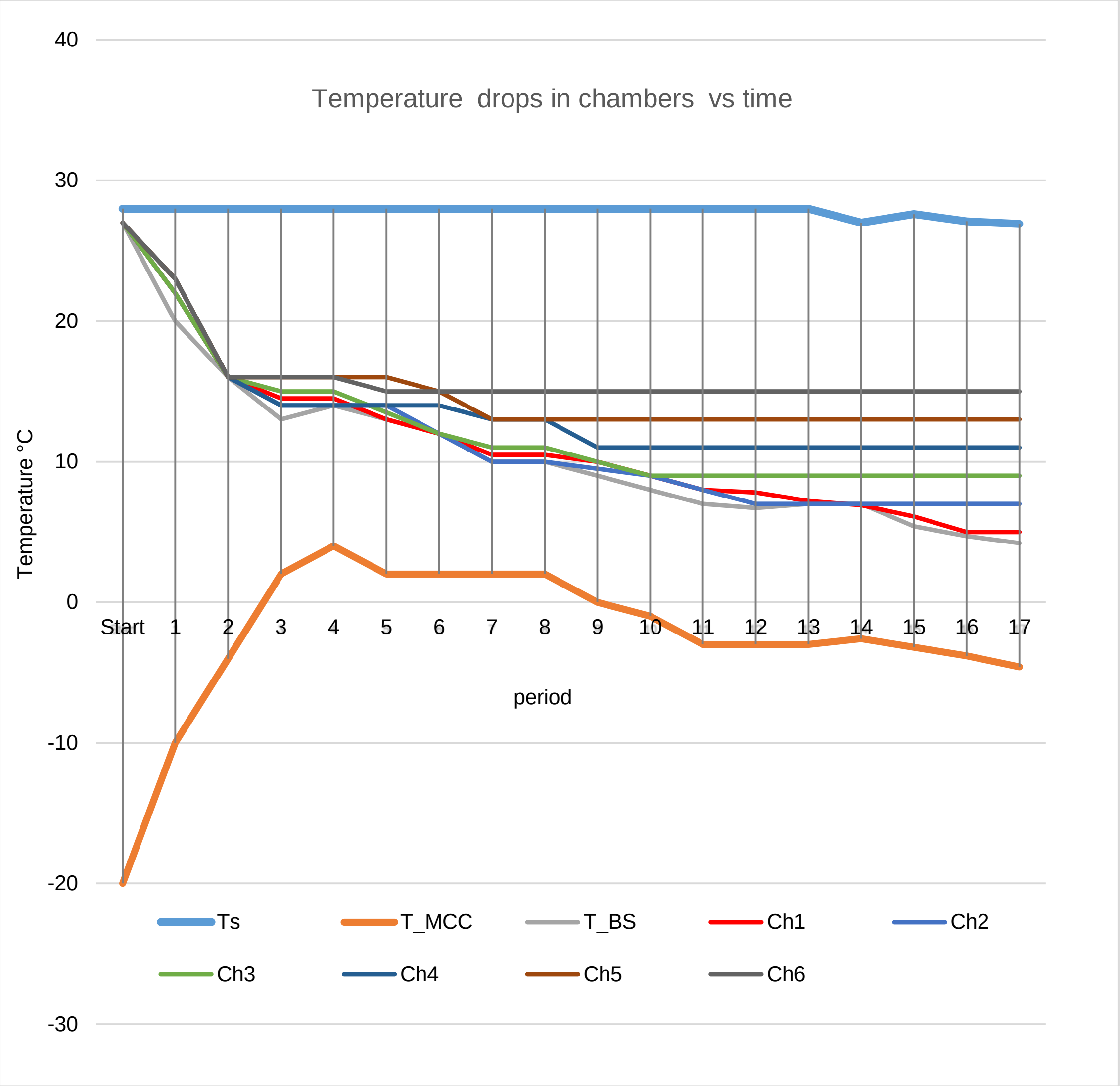


Figure 4.8: A graphic presentation of the third test results.

4.4 TESTING AFTER LOADING

The fans were switched off and the each chamber was loaded with one apple. Each apple weighs approximately 200 grams. The ambient temperature of the apples before they were loaded into the chambers had been 28 ºC for six hours. The thermostat of each chamber was kept the same as the previous test which represented the final temperature that the chambers had reached when they were still empty. Data were collected every half hour. The chart in Figure 4.9 demonstrates the results of the test.

After the data were collected according to the equation shown in Chapter 3.1 in the previous chapter, an analysis was conducted to measure the heat gain of the prototype components.

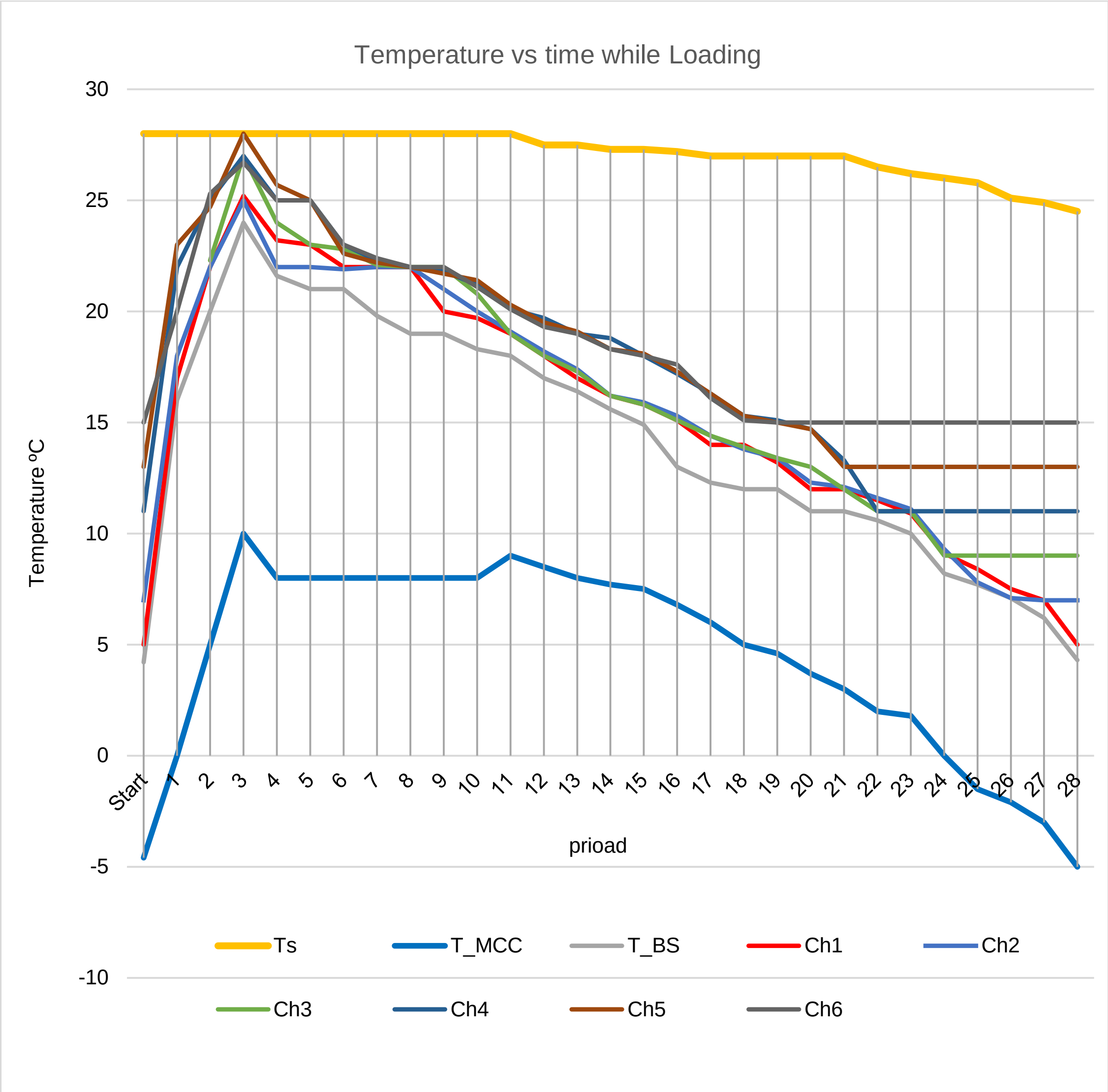


Figure 4.9: A graphic presentation of the test results of the loaded chambers

4.5 HEAT GAINED BY THE CH'S, T_BS AND T_MCC

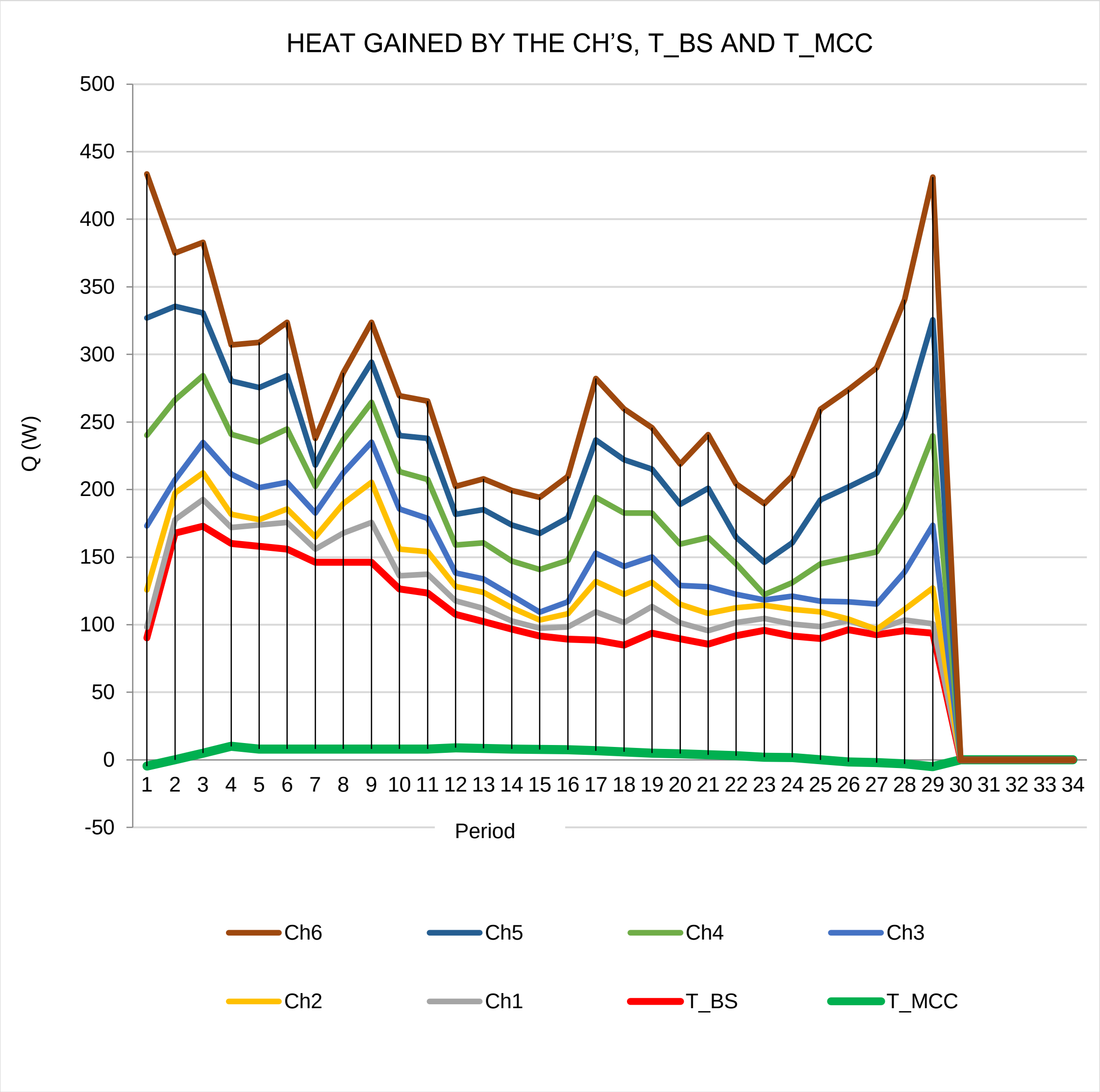


Figure 4.10: Heat gained by THE CH'S, T_BS AND T_MCC

CHAPTER FIVE
CONCLUSION AND RECOMMENDATIONS

5.1 CONCLUSION

On the basis of the results obtained in this study, the following conclusions can be drawn:

- The first test proved that the air is very poor at self-cooling by convection if the desired space to be cooled is directly above the source of the cold (evaporator). The temperature in the back space decreased by only one degree from 26º C to 25º C after 48 hours. The temperatures of the chambers did not change, which provides self-regulation of the air flowing to the chambers.
- The second test shows that after 48 hours, the temperature of the main cooling chamber rose. The temperatures in the lower chambers were lower than those in the upper chambers.
- The third test ran without a load in the chambers. The result shows that the different desired temperatures in the chambers had been reached periodically after seven and a half hours. They remained constant in each chamber at the levels at which they had been set. The phenomenon is that when the thermostat of the sixth chamber sets the fan speed to zero, the heat generated by this fan is eliminated. This is evident in the temperature decreasing in the temperature of the main cooling chamber and when the thermostat set the fan speed to zero in the different chambers, the temperature in the main cooling chamber decreased every time.
- The fourth test ran with a load in the testing chambers after the third test. The result shows that the temperature of the chambers increased rapidly, especially the upper chambers where the temperatures were higher. The temperature of the main cooling chamber increased too. The temperature of the main cooling chamber dropped slightly within half an hour and then stabilised for three hours, after which, for some unknown reason, it rose up for half an hour, before it started to decline again. The temperatures of chambers one, two and three were constant at first. Later, they dropped at different rates, with chamber 2 showing the slowest rate. A constant temperature in the lowest chamber was achieved at 5°C. Hence, the objective of having different chambers at different set temperatures between 5 and 15 °C was achieved. The system can therefore be used to test food storage at these ‘high’ refrigeration temperatures.

5.2 RECOMMENDATIONS

For this project, it was ascertained that the design would be better off with an evaporator above the area needing to be cooled because the results of the experiments show that the time to reach constant temperatures is too long. This could be because of the deficit caused by dragging or pushing the cold air from the main cooling chamber up to the experimental chambers, which is in violation of the principles governing the movement of cold air. Another device, such as a fan, is needed. This will make it possible for more loads to be added. Finding another way for controlling the air flow in the experimental chambers (for example, such as using a hydraulic thermostat) would also help.

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APPENDICES

APPENDIX A: THE CALCULATION OF THE EFFECTIVE ARE FOR HEAT TRANSFER

A 1 THE EFFECTIVE AREA OF THE BACK COVER

The back cover is made of four pieces, two sides, as in Figure A2, and a rectangular shape at the back and a rectangular shape at the top. Omitting the edges, therefore:

1- The area of the sides

The area of the angles

= ½ base ×height = ½ (25× 10^{-2} m) × (0.675 m) = 8.437× $10^{-2}\mathrm{m}^2$

The area of the rectangular shape

=length ×width =(50× 10^{-2} m) × (0.675 m) = 3.375 × $10^{-2}\mathrm{m}^2$

Therefore, the total area of the sides

= (8.437× $10^{-2}\mathrm{m}^2$ + 3.375 × $10^{-2}\mathrm{m}^2$) × 2 = 23.624 × $10^{-2}\mathrm{m}^2$

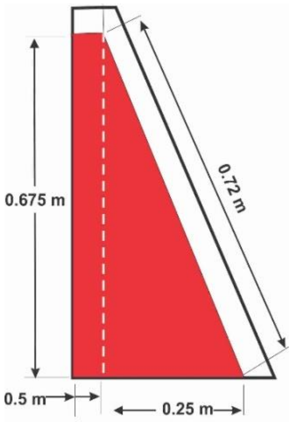

Figure A1.1: The back cover side (Foam)

2- The area of the back rectangle

The back rectangle measures (76× 10^{-2} m)× 1.00 m

= (72 × $10^{-2}\mathrm{m}$)× (1.00)= 72 × $10^{-2}\mathrm{m}^2$

3- The area of the top rectangle

= (0.5)× (1.00)= 5.0 × $10^{-2}\mathrm{m}^2$

From 1 and 2 and 3 the total area of the back cover is equal to the sum of the three areas

= 23.624× $10^{-2}\mathrm{m}^2$ + 72 × $10^{-2}\mathrm{m}^2$ + 5 × $10^{-2}\mathrm{m}^2$ = 100.624 × $10^{-2}\mathrm{m}^2$

A 2 THE FRONT AREA OF THE BACK COVER

The area of the front of the back cover is the area of double sides' self-adhesive foam and the back of the experimental chambers.

The area of the double sides' self-adhesive foam

= ((0.675 m × 1.00 m)) – 6 × (0.30 m ×0.30 m) =

= 54 × $10^{-2}\mathrm{m}^2$

Figure A1.2: The back cover side with double sides self-adhesive net

A 3 THE EFFECTIVE HEAT TRANSFER AREA OF THE CHAMBER

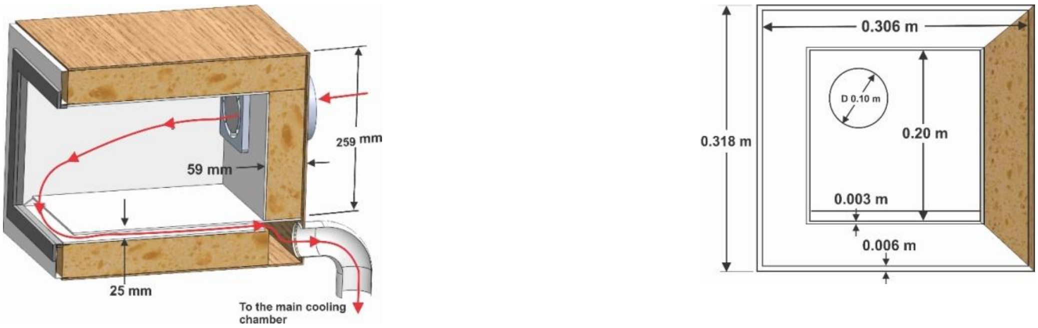

Figure A1.3: The chamber dimensions for the effective area

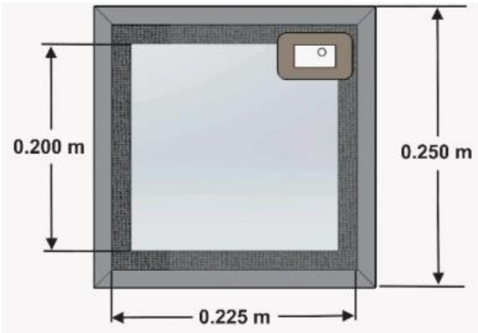

Figure A1.4: The chamber door dimensions diagram for the effective area

Thus: the chamber door has two plastic layers separated by layer of air and foam at the edges, therefore,

The area of the foam separator = (0.25 m× 0.25m) - (0.20 m×0.20 m) = 1.06 × 10^{-2}m^2

By using the Microsoft Excel software for calculations, the results are shown in Table A1.

Table A1: Calculations of the chamber effective area for heat transfer

No;	Part name	The inner area	The external area	#	Average Area m^2×#
1	The chamber sides	0.200 m×0.250 m	0.318 m×0.309 m	4	29.16× 10^{-2}m^2
2	The chamber back side	0.200 m×0.175 m	0.318 m×0.259 m	1	6.31× 10^{-2}m^2
3	The plastic doors (air separator)	0.20 m×0.20 m	0.20 m×0.20 m	1	4.0× 10^{-2}m^2
4	The plastic doors (foam separator)	(0.25 m× 0.25m) -(0.20 m×0.20 m)		1	1.06 × 10^{-2}m^2
5	Convection area of the chamber door	0.2250 m×0.2250 m		1	5.06 × 10^{-2}m^2

Notes:

The back area of the plastic box, foam and the wood box should be subtracted from the area of inlet and outlet as demonstrated below in the equation (Chapter3, Section 3.4)

The inlet area: A = 3.1415 × $(0.100m)^2$ = 0.031415 m^2 = 3.1415 ×$10^{-2}m^2$

Thus: The total area for the back components of the chamber: -

Back side area of the chamber = (6.310 – 3.1415) ×$10^{-2}m^2$ = 3.168 ×$10^{-2}m^2$

The inside area for convection = (3.168 + 5.06+ 29.16) ×$10^{-2}m^2$ = 37.388× $10^{-2}m^2$

A 4 THE HEAT LOSS CALCULATIONS

Given:

$k_{plywood}$ =*0.13* W/ (m.K)

$k_{plastic}$ =*0.03* W/ (m.K)

K (foam) = 0.03 W/ (m.K)

K (air) = 0.0257 W/ (m.k)

The main cooling chamber temperature = -20 ºC

The surrounding outside temperature = 27 ºC

The calculation for the heat losses would be measured by the constant state of the chambers, therefore

h_o = 1.6 W/ (m^2.K), due to the still air at the steady state. According to (ASHRE, 2000), hand book.

At working temperatures of 5 -15°C, the air properties were taken as approximately constant:

c_{p-air} =1.005kJ/kg.K

ρ_{air} = 1.222 kg/m^3

Fan air flow = 0.00802 m^3/sec

By inducting equation no. 3.1 in Chapter 3, where the first T is the chamber temperature and the second T is the back space temperature or inlet temperature, just before the chamber is loaded. Neither the density nor the specific heat of the air would have significant changes within the temperature range reserved, therefore, from appendices B, table B4 and the first row of data for the chamber 1, the calculation of the heat gained:

$\dot{Q} = \dot{m}\ c_p \Delta T$ = (1.222 kg/m^3 × 0.00802 m^3/sec) × 1.005 kJ/kg. °C × (5- 4.2) °C

$\dot{Q}$ = 0.007879 kW

When the chamber is loaded with an apple weights 200 grams, the temperature of the chamber rises due to the ambient air which flowed in while loading the chamber with the apple, and the heat produced by the apple. The thermostat switches the fan on.

Note:

T_s: surrounding temperature

T_{MCC}: main cooling chamber temperature

T_{BS}: back space cover temperature

Ch1… Ch6: chamber1 …, chamber6

All the temperatures were measured in degrees Celsius.

APPENDIX B: THE RESULTS OF THE FOUR TESTS CONDUCTED ON THE PROTOTYPE

Table B1.1: The first test result:

Components	T_s	T_{MCC}	T_{BS}	**Ch1**	**Ch2**	**Ch3**	**Ch4**	**Ch5**	**Ch6**
Start (ºC)	26	26	26	26	26	26	26	26	26
End (ºC)	27	-20	25	26	26	26	26	26	26

Table B1.2: The second test result:

Components	T_s	T_{MCC}	T_{BS}	**Ch1**	**Ch2**	**Ch3**	**Ch4**	**Ch5**	**Ch6**
Start (ºC)	26	-20	26	26	26	26	26	26	26
End (ºC)	26	5	10	12	13	12	14	15	15

Table B1.3: The third test result before loading the chambers.

Compone	T_s	T_{MCC}	T_{BS}	Ch1	Ch2	Ch3	Ch4	Ch5	Ch6
Start (ºC)	28	-20	27	27	27	27	27	27	27
1	28	-10	20	22	22	22	23	23	23
2	28	-4	16	16	16	16	16	16	16
3	28	2	13	14.5	14	15	14	16	16
4	28	4	14	14.5	14	15	14	16	16
5	28	2	13	13	14	13.5	14	16	**15**
6	28	2	12	12	12	12	14	15	**15**
7	28	2	10	10.5	10	11	13	**13**	**15**
8	28	2	10	10.5	10	11	13	**13**	**15**
9	28	0	9	10	9.5	10	**11**	**13**	**15**
10	28	-1	8	9	9	**9**	**11**	**13**	**15**
11	28	-3	7	8	8	**9**	**11**	**13**	**15**
12	28	-3	6.7	7.8	**7**	**9**	**11**	**13**	**15**
13	28	-3	7	7.2	**7**	**9**	**11**	**13**	**15**
14	27	-2.6	7	6.9	**7**	**9**	**11**	**13**	**15**
15	27.6	-3.2	5.4	6.1	**7**	**9**	**11**	**13**	**15**
16	27.1	-3.8	4.7	**5**	**7**	**9**	**11**	**13**	**15**
17	26.9	-4.6	4.2	**5**	**7**	**9**	**11**	**13**	**15**

Table B1.4: The test result after loading the chambers

Components	T_s	T_{MCC}	T_{BS}	Ch1	Ch2	Ch3	Ch4	Ch5	Ch6
Start (ºC)	28	-4.6	4.2	**5**	**7**	**9**	**11**	**13**	**15**
1	28	0	16	17	18	17	22	23	20
2	28	5	20	22	22	22.3	25	24.7	25.3
3	28	10	24	25.2	25	27	27	28	26.7
4	28	8	21.6	23.2	22	24	25	25.7	25
5	28	8	21	23	22	23	25	25	25
6	28	8	21	22	21.9	22.8	23	22.6	23
7	28	8	19.8	22	22	22.1	22.3	22.2	22.4
8	28	8	19	22	22	22	22	22	22
9	28	8	19	20	21	22	21.8	21.7	22
10	28	8	18.3	19.7	20	20.8	21.2	21.4	21.1
11	28	9	18	19	19.1	19	20.1	20.3	20.1
12	27.5	8.5	17	18	18.2	18	19.7	19.5	19.3
13	27.5	8	16.4	17.	17.4	17.3	19	19.1	19
14	27.3	7.7	15.6	16.2	16.2	16.2	18.8	18.3	18.3
15	27.3	7.5	14.9	15.8	15.9	15.8	18	18.1	18
16	27.2	6.8	13	15.1	15.3	15.1	17.2	17.3	17.6
17	27	6	12.3	14	14.4	14.4	16.3	16.3	16.1
18	27	5	12	14	13.8	13.9	15.3	15.3	15.1
19	27	4.6	12	13.2	13.4	13.4	15.1	15	**15**
20	27	3.7	11	12	12.3	13	14.7	14.7	**15**
21	27	3	11	12	12.1	12	13.3	**13**	**15**
22	26.5	2	10.6	11.5	11.6	11	**11**	**13**	**15**
23	26.2	1.8	10	10,9	11.1	11	**11**	**13**	**15**
24	26	0	8.2	9.1	9.3	**9**	**11**	**13**	**15**
25	25.8	-1.5	7.7	8.4	7.8	**9**	**11**	**13**	**15**
26	25.1	-2.1	7.1	7.5	7.1	**9**	**11**	**13**	**15**
27	24.9	-3	6.2	7	**7**	**9**	**11**	**13**	**15**
28	24.5	-5	4.3	**5**	**7**	**9**	**11**	**13**	**15**

APPENDIX C: THE AIR PROPERTIES TABLE

Table C1.1: Air properties

Temperature ***- t -*** ***(ºC)***	**Density** ***- ρ -*** ***(kg/m^3)***	**Specific Heat** ***- c_p -*** ***(kJ/(kg K))***	**Thermal Conductivity** ***- k -*** ***(W/(m K))***	**Kinematic Viscosity** ***- ν -*** ***x 10^{-6} (m^2/s)***	**Expansion Coefficient** ***- b -*** ***x 10^{-3} (1/K)***	**Prandtl's Number** ***- P_r -***
-150	2.793	1.026	0.0116	3.08	8.21	0.76
-100	1.980	1.009	0.0160	5.95	5.82	0.74
-50	1.534	1.005	0.0204	9.55	4.51	0.725
0	1.293	1.005	0.0243	13.30	3.67	0.715
20	1.205	1.005	0.0257	15.11	3.43	0.713
40	1.127	1.005	0.0271	16.97	3.20	0.711
60	1.067	1.009	0.0285	18.90	3.00	0.709
80	1.000	1.009	0.0299	20.94	2.83	0.708
100	0.946	1.009	0.0314	23.06	2.68	0.703
120	0.898	1.013	0.0328	25.23	2.55	0.70
140	0.854	1.013	0.0343	27.55	2.43	0.695
160	0.815	1.017	0.0358	29.85	2.32	0.69
180	0.779	1.022	0.0372	32.29	2.21	0.69
200	0.746	1.026	0.0386	34.63	2.11	0.685
250	0.675	1.034	0.0421	41.17	1.91	0.68
300	0.616	1.047	0.0454	47.85	1.75	0.68

Temperature ***- t -*** ***(^oC)***	**Density** ***- ρ -*** ***(kg/m^3)***	**Specific Heat** ***- c_p -*** ***(kJ/(kg K))***	**Thermal Conductivity** ***- k -*** ***(W/(m K))***	**Kinematic Viscosity** ***- ν -*** ***x 10^{-6} (m^2/s)***	**Expansion Coefficient** ***- b -*** ***x 10^{-3} (1/K)***	**Prandtl's Number** ***- P_r -***
350	0.566	1.055	0.0485	55.05	1.61	0.68
400	0.524	1.068	0.0515	62.53	1.49	

APPENDIX F: THE MATERIAL COST FOR CONSTRUCTION

No	Item	Quantity	Price	Labour	Total
1	SCI Impulse P/B 10 mm Round	6	R25	-	R150
2	Ambiant T/Stat -35/+35 DEG C RAMB	6	R75	-	R450
3	Digital Thermometer white RT-LTD10	6	R45	-	R270
4	Fan Motor 80×80×25 mm	6	R80	-	R480
5	3/8×3/4 wall Insulation Gulf o FL RFINS - 11838	6	R25	-	R150
6	Special Gasket Screw-On Grey: 601152/017-Grey	6	R75	-	R450
7	GALVF-White Sheet metal 0.50×2450×1225 mm	1	R250	R500	R750
8	PLYWOOD COMM EXT 9×2440×1220 mm	3	R386.40	R150	R1309.30
9	ALCOLIN Cold Glue, Wood Glue	1	R120	-	R120
10	Spray fume cans	6	R120	-	R720
11	50 mm in diameter plastic Tubes	10 m	R10/m	-	R100
12	Other	-	-	-	R600
13	Chest deep freezer	1	R 1900	-	R1900
total					**R7449.30**