



College of Engineering and Technology Mechanical Engineering
Department Graduation Project

Design and Building of a Driver for Atmospheric Water Vapor Harvesting Utilizing Solar Energy

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Graduation Project Evaluation

According to the project supervisor and according to the agreement of the Testing Committee Members, this project is submitted to the Department of Mechanical Engineering at College of Engineering and Technology in partial fulfillments of the requirements of (B.SC) degree.

Supervisor Signature

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Department Headmaster Signature

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Dedication

To The Souls Of Our Ancestors
Who Left With Rivers Of Benevolence
To Our Parents Those Who Were Mentors, Teachers and Friends
Who Were Guides With Their Endless Giving
To Our Teachers
Who Were Candles, Lighting Our Path to Excellence
To Our Beloved University
Where Our Hearts Will Remain
To The Entire Islamic Nation
To The Reading Public in General

To All Those
We Promise...

The Promise of the Blood of Martyrs
That We Will Forever Be Loyal Servants To Our Glorious Islam

Acknowledgment

All praise and thanks is due to Allah, the lord of mankind and all that exists for his blessings, benevolence, and guidance at every stage of our life

Our thanks go first to our advisors Eng. Kazem Osaili, Dr.Imad Al-Khateeb. Their guidance and support made this work possible. Their constant encouragement, intuitive wisdom, and resolute leadership were instrumental in completing this work.

And finally, ultimate thanks go to our families and to all the lecturers. Their efforts and their nice dealing with us improved our characters to become successful engineers in the future.

Abstract

يهدف هذا المشروع لتصميم وبناء جهاز لحصد الماء من الهواء الرطب المحيط, كي يصبح مصدراً جديداً للمياه. يعمل الجهاز على تبريد الهواء باستخدام دورة تبريد لإجبار بخار الماء للتكثف ومن ثم تجميعه بداخل خزان. هذا الجهاز يعمل على الطاقة الشمسية. لقد تم تصميم هذا الجهاز كي ينتج ما يزيد عن نصف لتر من الماء في الساعة, وقد تم بنا وتجربة الجهاز في مدينة الخليل وقد تم الحصول على النتائج المتوقعة.

This project aims to design and build a device that harvests water from humid surrounding air, to be as a new source of water. This device uses the vapor compression refrigeration cycle to cool the air first and to force water vapor in the air to condense and to be collected later in a tank. This project operates on the solar energy. This device was designed to produce more than 0.5 liters of water per hour, and it was built and tested in Hebron and it gave the expected results.

Recommendations

We recommend for this project to be developed in many forms. It could be developed as a small hand bag so that small-sized pieces can be used which facilitates the mobility of the users. Also it could be developed by utilizing the exiting cold air from the device by adding another cabin and entering this air to the new cabin to harvest more water, so higher efficiency will be achieved. Or to utilize the exiting air as a refrigerant to cool the new cabin which can be used for keeping foods.

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CHAPTER ONE

INTRODUCTION

CHAPTER ONE

INTRODUCTION

1.1 General Outlook

The water issue is a very controversial topic in our century, and several countries in the world are trying to get the highest possible amount of water, where in some cases it caused some political conflicts with the neighboring countries about the fresh water source that they might share. All of that is happening because of the fresh water scarcity in the world.

In spite of the fact that more than 70% of our planet is covered with water, but only less than 1% of this water is drinkable. Based on that, new ways of purifying water must be found to make it drinkable, and find new sources of fresh water to cover up the incredibly increasing number of people around the world.

Although many countries in the world suffer from the fresh water scarcity, but some of them actually face worse situations in that aspect especially remote areas that have no services and it's difficult to support these regions with fresh water, such as the desert in Namibia that receives less than 20 mm of rainfall per year to be of the driest areas in the world.

Fresh water used to be considered inexhaustible, and that is why people drained it badly. The incredibly increasing number of people around the world is demanding more sources of fresh water, while factories, sewages, and other polluting sources are damaging lakes and rivers by dumping their wastes in them. All these reasons and more, started to push the humanity for reconsidering their behavior, and looking for new sources of water, find new ways for purifying water.

Nevertheless, and even with the count of 1 billion people who do not have an access to clean water, and the UN estimation that is expecting 66 percent of the people in the world to suffer from water shortage by the year 2025.

However, though the future seems to suffer from water shortage, but the humanity will not give up and wait for that to happen. So we think about new ways of getting water which can be considered as a new source of drinkable water and irrigation of crops, which is sustainable and renewable.

In addition for water scarcity, there is another problem which is electricity that is difficult to deliver to far regions especially remote areas as it is the case in water. Electricity's problem is not only to deliver it, but also its production that mostly depend on petrol which will deplete soon in the world.

As a solution for this problem we are in need for a device that produces water from a new source which is humidity in the atmosphere, that operates by one of the renewable sources of energy such as the solar energy.

The amount of water that can be harvested from the humidity in thin air is reliable in most of the conditions even if it depends on the temperature and humidity at the place. Although the relative humidity in a certain place might be relatively low, it is still possible to get some fresh water out of it by using the right humidity harvesting method. The method of harvesting humidity can cover people basic needs from clean water in the dry areas, and especially is the coastal areas where there is a high humidity, but no source of fresh water.

1.2 Previous Studies

1.2.1 Harvesting Fog

Harvesting fog is an old and most popular way of extracting water from thin air. It has been used in Chile for almost 30 years, where local people used to use their nets to “hunt” fog in some areas where there is a heavy fog. This way provided locals with a reliable source of pure water and saved them from walking several kilometers to get fresh water. [1]



Figure 1.1 Fog Harvesting Meshes in a Dry Area. (Lummerich, 2010)

1.2.2 Tower Shaped Collectors

This innovation uses the renewable energy only (ScienceDaily, 2009), which makes it eligible to be used in the desert areas or far away rural areas where there is no access to public electricity. The main principle of this idea is more like a development of Ebrahim Al-Sharqawi master thesis of the glass box and the tortuous fabric saturated with a salt solution. It uses brine strings with a great interface with the surrounding air and the longest time possible to ensure that the brine was able to absorb the maximum amount of humidity. After that, the brine will flow slowly downward with the power of gravity and pass in multiple-stage vacuum evaporation (Fraunhofer Institute for Interfacial Engineering and Biotechnology. [1]



Figure 1.2 Tower shaped collector uses renewable energy. (ScienceDaily,2009)



Figure 1.3 The working process of the tower shaped water collector. (Fraunhofer Institute for Interfacial Engineering and Biotechnology (IGB), 2012)

1.2.3 Water Building Resort

In De Urrutia design, the southern side of the building will consist of massive number of photovoltaic glass that will cover the building totally from that side in order to gain the maximum benefit from the sun rays to produce solar power (Schwartz, 2010), and at the same time it will also allow the sunlight to inter the building from that side (Garvey, 2009). The northern side will have lattices in order to provide good aeration for the building, and also the needed air for producing water (Garvey, 2009). The air coming from the northern side will be handled by Teex Micron Atmospheric Water Generators, which will work on transferring the humidity in air into pure water for drinking, and those generators can produce from 35,000 to 109,000 gallons of water every day at their maximum (Schwartz, 2010). At the bottom of the building, there will be a water thermal facility where sea and rain water will be purified; also there will be a control center to ensure the water quality (Kain, 2011). [1]

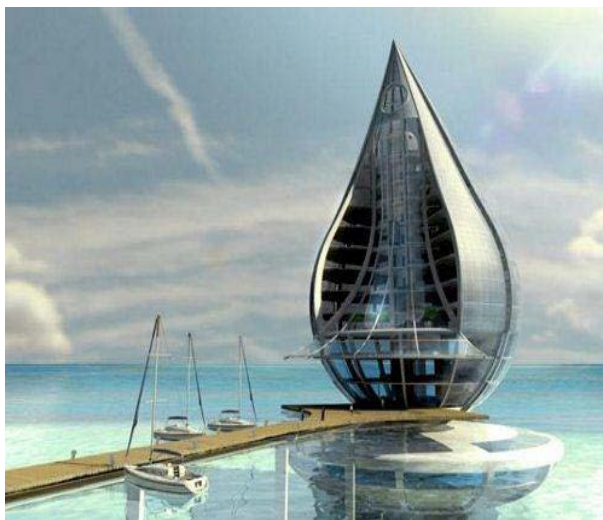


Figure 1.4 A Virtual picture of the Water Building Resort design. (Jolly, 2009)

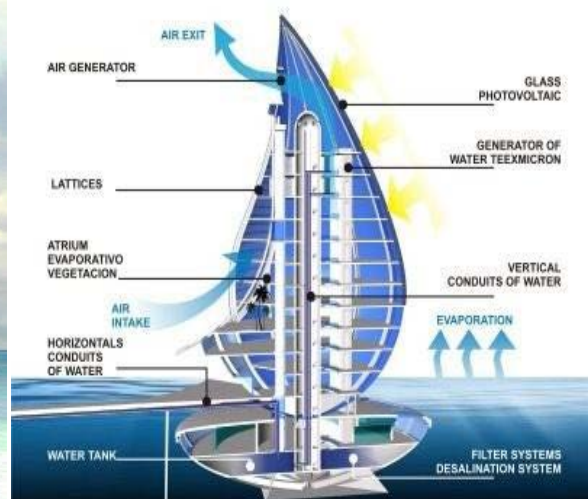


Figure 1.5 Working system of the Water Building Resort design. (Garvey, 2009)

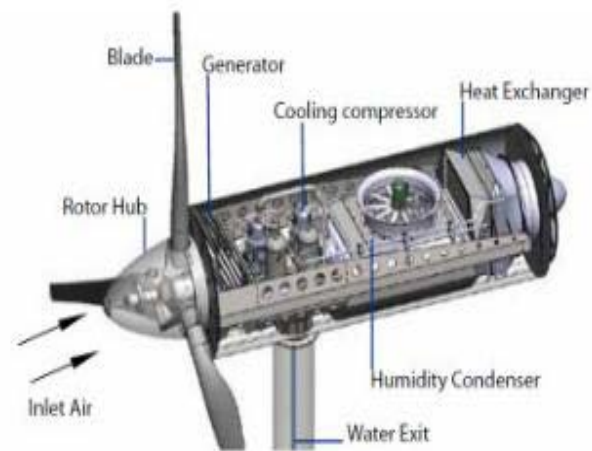
1.2.4 Wind Turbine

One of the newest humidity harvesting inventions, and the producing company of these wind turbines WM1000 “Eole” is supposed to start selling them sometime in this year 2012 as they just started working on advertising for the product.[1]

In this method of harvesting humidity, the company adjusted the normal wind power turbine in order to be capable of collecting water from thin air (Macguire, 2012). With 24 meter high turbine, it was possible to collect about 62 liters in an hour in the desert near Abu-Dhabi in UAE, but depending on the relative humidity, the temperature, and the speed of wind, it can produce water up to 1000 liters everyday which is enough to cover the drinking water need for a small community of 2000 to 3000 people (Mac-guire, 2012). [1]



Figure 1.6 Wind turbine WM1000 which can harvest humidity from air (Macguire, 2012).



COURTESY EOLE WATER

Figure 1.7 The inside machinery of the wind turbine WM1000 (Macguire, 2012).

1.2.5 Water from Air Conditioning (AC)

Air conditioning devices are able to produce water with different amounts depending on the relative humidity and the temperature of the air. This water resulting from air conditioning process is usually considered to be unwanted, hence; users try to find ways to dispose this water, and finally the water ends in the sink.

Nevertheless, if users start to collect the water coming out from their ACs, they actually would be able to save a huge amount of water that they are paying for.

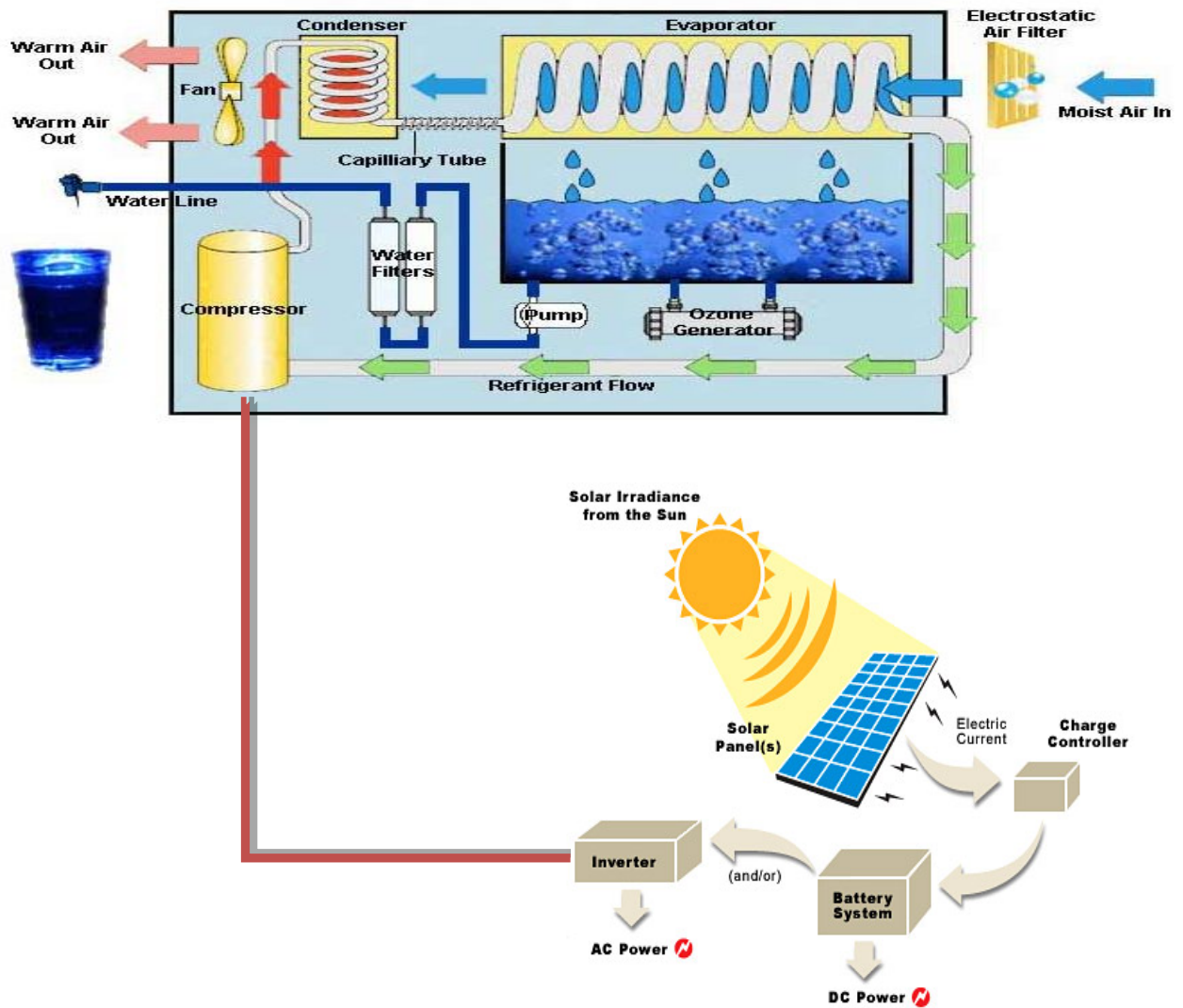


Figure 1.8 Air conditioning devices with solar energy.

1.2.6 The Water Quality

The quality of water coming from the ACs can be good if some adjustments were added to the device. For example, if the metal used in the pipes was replaced with another one usable for drinking purposes like Stainless steel, but more studies are needed to be done about that before it can be implemented for drinking purposes.

In comparison with the quality of harvesting fog water, the quality of water collected by ACs should be good after taking care of hygienic matters for the device inner parts. However, the AC

water quality should be analyzed in the target area first, and even if the water was good in some countries, we still have to analyze it every time we start a project in a new place.

It might good to use some water filtrations to remove different partials and sediments, but the type of filtration needed for water depends on the area conditions. Ultra-filters to clean the water for example, we need to know that even the best public water companies do not purify the water with ultra-filters.

In the other hand, it might be needed to add some minerals to the water to make it drinkable in case it was soft as it is condensed water, like appropriate amount of calcium and magnesium, but so far, there was no water device that needed to add any minerals to the produced water. However, analyzing the condensed water will give the needed answer about this point.

1.3 Project Objectives

The main aim of this project is to provide a clean and renewable source of fresh water for people in remote areas, with consideration to the cost of this water and the sustainability of the method used to get it. Hence, it will be an easy and cheap way for people in areas with low income. Using harvesting humidity method can support or replace some traditional ways of getting fresh water in the arid areas where there is a relatively high humidity (more than 40%), and low rainfall rate.

Another aim or objective of this project is to decrease the undesirable humidity in some cases such as high humidity in factories that harms the machines by the effect of corrosion by time. So by such a device harmful humidity will be converted to useful water.

1.4 Importance of the Project

The importance of this project is:

- It is mainly used to produce water for remote areas from a renewable source.
- To protect the environment from pollution caused by energy conversion when use conventional sources of energy.
- It can be used to lower local humidity.

1.5 Budget

The estimated budget for building the device is as shown in table (1.1).

Table 1.1 Budget

Task	Cost (JD)
DC Compressor	240
Evaporator	130
Condenser	120
Expansion Valve	30
Solar Plants	320
Charge Controller	56
Battery	100
Pipes	50
Fan	20
Filters	50
Wires	15
Miscellaneous	120
Total	1251

1.6 Time Planning

The project plan follows the following time schedule, which includes the related tasks of study and system analysis.

1.6.1 Semesters Time Table

The tasks for the first semester were as shown in table (1.2). All tasks were accomplished successfully.

Table 1.2 First Semester Time Table

Task/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Literature Review	■	■	■													
Project Concept			■	■												
Cooling Load				■	■	■	■									
Main Design of the Cycle and its Components							■	■	■	■	■	■	■			
Electrical Design and Selection													■	■		
Project Documentation				■	■	■	■	■	■	■	■	■	■	■	■	■

The tasks for the second semester were as shown in table (1.3). All tasks were accomplished successfully.

Table 1.3 Second Semester Time Table

Task/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Device Design Using Catia Program	■	■	■													
Buying and Installing Devices and Equipment			■	■	■	■	■									
Running the Device by the Solar Energy							■	■	■							
Testing the Performance of the System									■	■						
Recommendations											■					
Conclusions												■				
Project Documentation				■	■	■	■	■	■	■	■	■	■	■	■	■

CHAPTER TWO

COOLING LOAD

CHAPTER TWO

COOLING LOAD

2.1 Introduction

The total heat required to be removed from refrigerated space in order to bring it at the desired temperature and maintain it by the refrigeration equipment is known as cooling load. The purpose of load estimation is to determine the size of the refrigeration equipment that is required to maintain inside design conditions during periods of maximum outside temperatures. The design load is based on inside and outside design conditions and its refrigeration equipment capacity to produce and maintain satisfactory inside conditions.

2.2 Cooling and Dehumidification

The cooling and dehumidifying process is shown in figure (2.1). It begins at state 1 and ends at state 2. Condensation of the water vapor in the air takes place, depending on the dew-point temperature of the air and on the final temperature required. A schematic device for cooling and dehumidification process is shown in figure (2.2) shows the process on the psychometric chart.

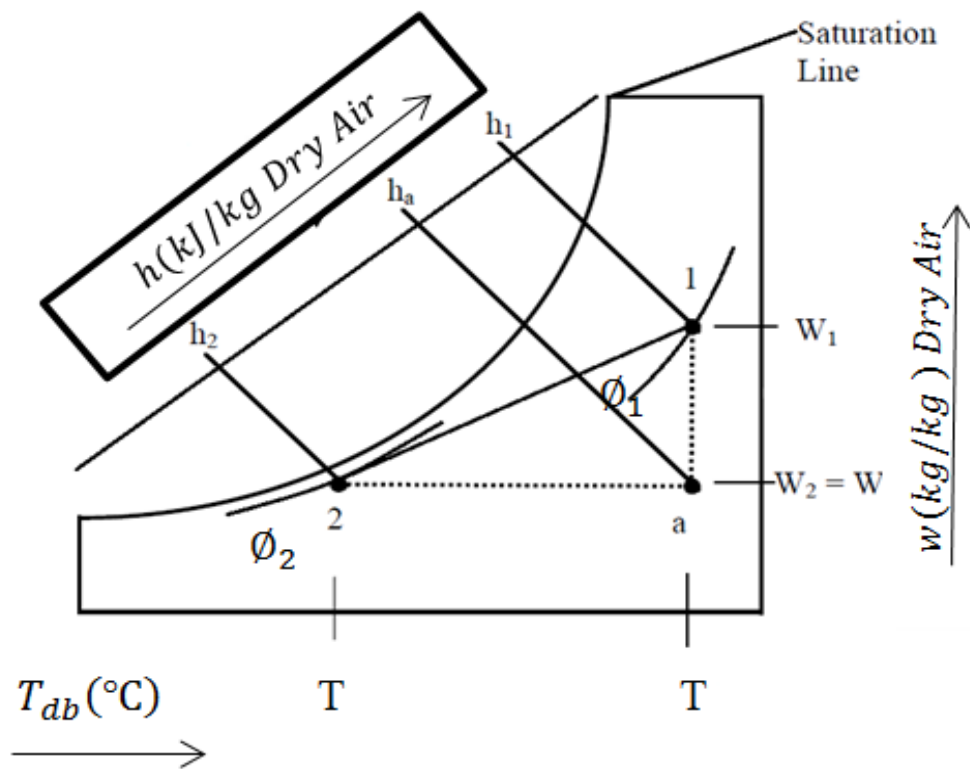


Figure 2.1 Cooling and dehumidification process.

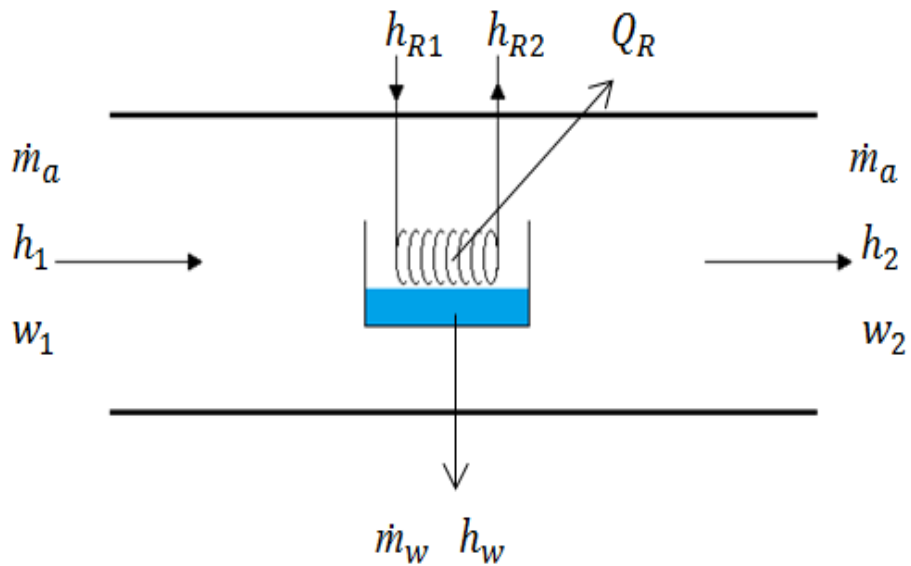


Figure 2.2 Schematic device of cooling and dehumidification device.

Application of the energy conservation principle for this process gives:

$$\dot{m}_a h_1 + \dot{m}_R h_{R1} = \dot{m}_a h_2 + \dot{m}_R h_{R2} + \dot{m}_w h_w \quad (2.1a)$$

Water vapor in:

$$\dot{m}_a h_1 - \dot{Q}_R = \dot{m}_a h_2 + \dot{m}_w h_w \quad (2.1b)$$

Application of the mass conservation principle for this process gives:

$$\dot{m}_a w_1 = \dot{m}_a w_2 + \dot{m}_w \quad (2.2.a)$$

Equation (2.2.a) can be expressed as:

$$\dot{m}_a = \frac{\dot{m}_w}{w_1 - w_2} \quad (2.2.b)$$

The refrigeration capacity required to accomplish this, is obtained from the energy balance:

$$\begin{aligned} \dot{Q}_R &= \dot{m}_a (h_1 - h_2) - \dot{m}_w h_w \quad (2.3) \\ \dot{m}_R (h_{R2} - h_{R1}) &= \dot{m}_a (h_1 - h_2) - \dot{m}_w h_w \end{aligned}$$

Where:

\dot{m}_a : The mass flow rate of air [kg/s].

\dot{m}_R : The mass flow rate of refrigerant [kg/s].

\dot{m}_w : The mass flow rate of water [kg/s].

h_1 & h_2 : The specific enthalpies of the air [kJ/kg].

h_{R1} & h_{R2} : The specific enthalpies of the refrigerant [kJ/kg].

h_w : The specific enthalpy of the saturated liquid [kJ/kg].

w_1 & w_2 : The humidity ratio of the air [kg/kg].

\dot{Q}_R : The refrigeration capacity [Watt].

The sensible heat ratio (SHR) for the process is then:

$$SHR = \frac{Q_s}{Q_s + Q_L} \quad (2.4)$$

$$Q_s = \dot{m}_a (h_1 - h_a) \quad (2.5)$$

$$Q_L = \dot{m}_a (h_a - h_1) \quad (2.6)$$

$$Q_T = Q_s + Q_L \quad (2.7)$$

Where:

SHR : The sensible heat ratio.

Q_s : The sensible heat [W].

Q_L : The latent heat [W].

Q_T : The total heat [W].

The following methods are used to cool and dehumidify the air:

- 1) By passing the air over a direct expansion –cooling (DX) coil in which refrigerant or chilled water is circulated. The coil surface temperature must be less than the dew-point temperature of the passing air for water vapor condensation to occur.
- 2) By forcing the air through a chilled water spray whose temperature is below the dew-point of the entering air. [2]

The first method is widely used for residential, commercial and industrial air conditioning applications. The cooling coil used in this method sized to match the total sensible cooling load, as calculated from Equation (2.3). The second method finds limited application due to the high cost of dehydrating agents and the cost of reactivating them.[2]

2.3 Coil Bypass Factor (BPF)

Consider a stream of air that passes through a cooling coil as shown in Figure(2.3). The ratio:

$$BPF = \frac{T_{a,out} - T_{App}}{T_{a,in} - T_{App}} \quad (2.8)$$

Where:

BPF : Is called the coil bypass factor.

$T_{a,in}$: Is the temperature of air stream entering the coil[°C].

$T_{a,out}$: Is the temperature of air stream leaving the coil[°C].

T_{App} : Is the apparatus dew point temperature[°C].

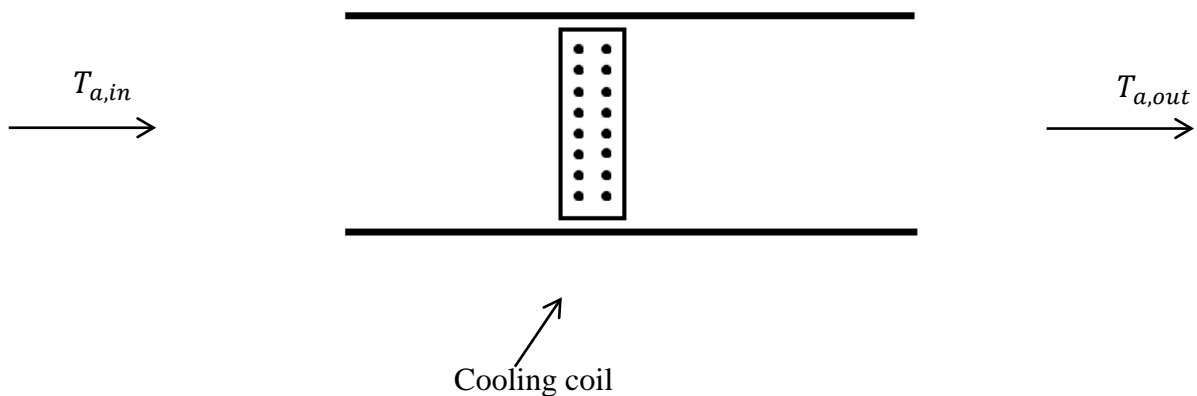


Figure 2.3 Nomenclatures for Coil Bypass Factor.

Practical values of BPF range from 0.1 to 0.2. The value of BPF depends on the number of the cooling coil rows (N) and its fin spacing. [2]

The value of BPF can be determined by the following equation: [2]

$$BPF = (0.56)^N \quad (2.9)$$

Where:

N: is the number of cooling coil rows.

On the other hand, a coil contact factor (CCF) is defined as: [2]

$$CCF = \frac{T_{a,in} - T_{a,out}}{T_{a,in} - T_{App}} = 1 - BPF \quad (2.10)$$

Practical value of CCF range from 0.8 to 0.9. From the above definition of CCF, it can be seen that the CCF approaches the value of 1.0 when temperature of the air leaving the coil approaches the value of the apparatus dew point temperature. [2]

2.4 Cooling Calculations

The device was designed at the worst surrounding temperature and relative humidity in Hebron which are 18°C and 30% respectively. [15]

Thus the dew point temperature was found as $T_{dp} = 0.2^\circ\text{C}$ from the psychrometric chart. It was assumed that the water harvesting is 0.5 L/hr.

2.4.1 The Temperature of Air Stream Leaving the Coil

The temperature of air stream leaving the coil can be calculated according to equation (2.8):

$$BPF = \frac{T_{a,out} - T_{App}}{T_{a,in} - T_{App}}$$

And the value of BPF can be calculated according to equation (2.9):

$$BPF = (0.56)^N$$

$$N = 4$$

$$BPF = (0.56)^4 = 0.1$$

Let the apparatus dew point temperature $T_{App} = 0^\circ\text{C}$. (To assure that harvested water not be iced).

$$0.1 = \frac{T_{a,out} - 0}{18 - 0}$$

$$T_{a,out} = 1.8^\circ\text{C}$$

And the value of CCF can be calculated according to equation (2.10):

$$CCF = \frac{18 - 1.8}{18 - 0} = 0.9$$

Atmospheric air ($p = 101.3\text{kPa}$) at $T_{db} = 18^\circ\text{C}$ and 30% relative humidity is cooled to dew point in device. The supply air temperature is 1.8°C and 15% relative humidity.

The process is represented on the psychometric chart as shown Figure(2.4) and shown Table(2.1).

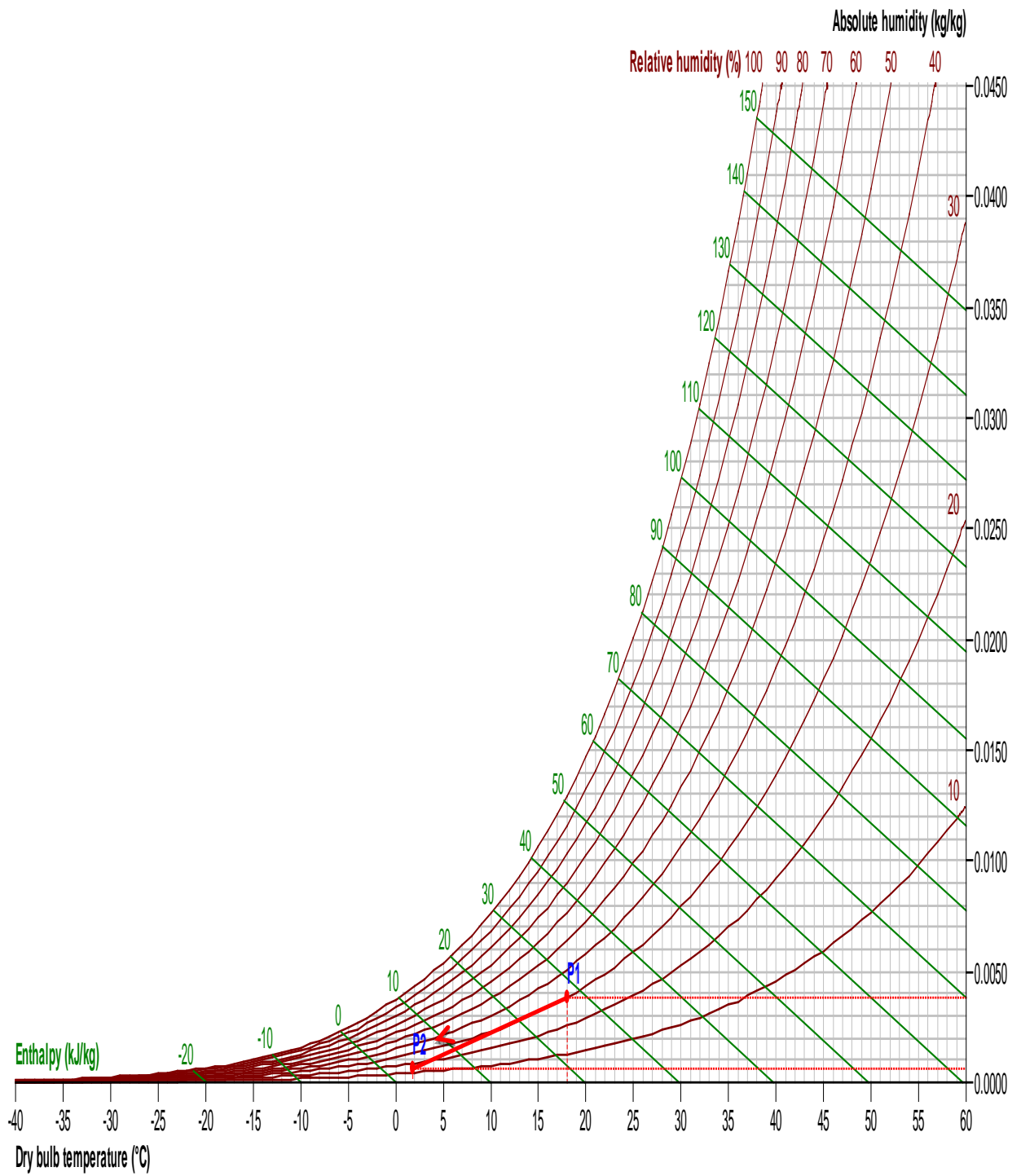


Figure 2.4 The process on the psychrometric chart.

Table 2.1 Properties of the Process

P1		P2	
T (dry bulb)	18.0°C	T (dry bulb)	1.8°C
T(wet bulb)	9.4°C	T (wet bulb)	-3.6°C
T (dew point)	0.2°C	T (dew point)	-19.9°C
Pressure	101325.0Pa	Pressure	101325.0Pa
RH	30.0%	RH	15.0%
w	0.0038kg/kg	w	0.0006kg/kg
Enthalpy	27.8kJ/kg	Enthalpy	3.4kJ/kg
Density	1.210kg/m ³	Density	1.283kg/m ³
Airflow	110.0l/s	Airflow	110.0l/s

2.4.2 The Mass Flow Rate of Air

The mass of air added can be calculated according to equation(2.2. b):

$$\dot{m}_a = \frac{\dot{m}_w}{w_1 - w_2}$$

$$\dot{m}_w = \frac{0.5L}{hr} = \frac{0.5L}{hr} * \frac{hr}{3600sec} = 1.389 * 10^{-4} L/s \text{ or } kg/s$$

$$\dot{m}_a = \frac{1.389 * 10^{-4}}{0.0038 - 0.0006} = 0.0434 \text{ kg/s}$$

2.4.3 The Sensible Heat Ratio (SHR)

The sensible heat can be calculated according to equation (2.5):

$$Q_s = \dot{m}_a(h_1 - h_a)$$

$$Q_s = 0.0434(27.8 - 19.9)$$

$$Q_s = 342.86 \text{ W}$$

The latent heat can be calculated according to equation (2.6):

$$Q_s = \dot{m}_a(h_a - h_2)$$

$$Q_s = 0.0434(19.9 - 3.4)$$

$$Q_s = 716.1 \text{ W}$$

The sensible heat ratio for the process is then:

$$SHR = \frac{Q_s}{Q_s + Q_l}$$

$$SHR = \frac{342.86}{342.86 + 716.1} = 0.323$$

2.4.4 The Refrigeration Capacity

The refrigeration capacity can be calculated according to equation (2.3):

$$Q_R = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$

Using table A-1 at temperature 0.2°C

$$h_w = 0.7633 \text{ kJ/kg}$$

$$Q_R = 0.0434(27.8 - 3.4) - 1.389 * 10^{-4} * 0.7633$$

$$Q_R = 1.0588 \text{ KW}$$

$$= 1058.8 \text{ W.}$$

2.4.5 Total Cooling Load

$$Q_T = 1058.8 \text{ W.}$$

Add 15% upon Q_T as safety factor.

$$\text{Total cooling load} = Q_T * 1.15$$

$$Q_{T,(fs)} = 1058.8 * 1.15$$

$$Q_{T,(fs)} = 1217.62 \text{ W.}$$

CHAPTER THREE

REFRIGERANT SELECTION AND CYCLE ANALYSIS

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REFRIGERANT SELECTION AND CYCLE ANALYSIS

3.1 Introduction

A refrigerant is the primary working fluid used for absorbing and transmitting heat in a refrigeration system or heat pump. Refrigerants absorb heat at a low temperature and low pressure and release heat at a higher temperature and pressure. A two stage systems consists of two compression stages connected in series.

3.2 Refrigerant Selection

In the selection of an appropriate refrigerant for use in refrigeration or heat pump system, there are many criteria to be considered. Briefly, the refrigerants are expected to meet the following conditions:

Thermodynamic properties:

- Low boiling point.
- High critical point temperature.
- High latent heat of vaporization.
- Low specific heat of liquid.
- Positive pressure at evaporator's temperature.
- Low specific volume at suction pressure and temperature.
- Mixes well with oil.

Thermo-physical properties:

- High thermal conductivity.
- High convection of heat transfer coefficient.
- Low dynamic viscosity.

Environment friendly:

- Having zero ozone depletion potential (ODP).
- Having low global warming potential (GWP).
- Non contribution of the atmosphere's heat retention.
- Non corrosive to metal.
- Non-acidic in case of a mixture with water or air.
- Chemically stable.
- Easily detectable in case of leakage.
- Nonreactive with the lubricating oils of the compressor.
- Non-flammable and non-explosive.
- Non-toxic.

Economics:

- High operating efficiency.
- Low cost.
- Easy availability.

R134a was selected since this refrigerant has the needed properties as shown in table (3.1).

Table 3.1 R134-a Properties

Property	Value
Molecular Formula	C ₂ H ₂ F ₄
Molar Mass	102.03 g/mole
Appearance	Colorless gas
Solubility in Water	0.15 wt%
Boiling Point at 1 Atm (101.3 kPa or 1.013 bar)	-15.34 °F -26.3 °C
Freezing Point	-153.9 °F -103.3 °C
Critical Temperature	213.9 °F 101.1 °C
Property	Value
Critical Pressure	4060 kPa 588.9 lb/in ² abs
Critical Volume	1.94 × 10 ⁻³ m ³ /kg 0.031 ft ³ /lb
Critical Density	515.3 kg/m ³ 32.17 lb/ft ³
Density (Liquid) at 25 °C (77 °F)	1206 kg/m ³ 75.28lb/ft ³

Density (Saturated Vapor) at Boiling Point	5.25 kg/m ³ 0.328 lb/ft ³
Heat Capacity (Liquid) at 25 °C (77 °F)	0.339 kcal/kg·K 0.339 Btu/(lb) (°F)
Heat Capacity (Vapor at Constant Pressure) at 25 °C (77 °F) and 1 Atm (101.3 kPa or 1.013 bar)	0.204 kcal/kg·K 0.204 Btu/(lb) (°F)
Vapor Pressure at 25 °C (77 °F)	666.1 kPa 96.61 psia
Thermal Conductivity at 25 °C (77 °F) Liquid	0.0824 W/m·K 0.0478 Btu/hr·ft°F
Thermal Conductivity at 1 Atm (101.3 kPa or 1.013 bar) Vapor	0.0145 W/m·K, 0.00836 Btu/hr·ft°F
Viscosity at 25 °C (77 °F) Liquid	0.202 mPa·S (cP)
Viscosity at 1 Atm (101.3 kPa or 1.013 bar) vapor	0.012 mPa·S (cP)
Flammability Limits in Air at 1 Atm (101.3 kPa or 1.013 bar)	None
Ozone Depletion Potential	0

3.3 Cycle Analysis

3.3.1 Simple Refrigeration Cycle

The term "refrigeration" may be defined as the process of removing heat from a substance under controlled conditions. It also includes the process of reducing and maintaining the temperature of a body below the general temperature of its surroundings. In other words, the refrigeration means a continued extraction of heat from a body whose temperature is already below temperature of its surroundings.

Vapor compression cycle is an improved type of air refrigeration cycle in which a suitable working substance, termed as refrigerant, is used. In evaporating, the refrigerant absorbs its latent heat from the solution which is used for circulating it around the cold chamber and in condensing; it gives out its latent heat to the circulating water of the cooler.

Figure 3.3 shows the schematic of a standard, saturated, single stage vapor compression refrigeration system and the operating cycle on a T-s diagram. As shown in the figure the standard single stage, saturated vapor compression refrigeration system.

It Consists of the Following Four Processes:

Process 1-2: Isentropic compression of saturated vapor in compressor.

Process 2-3: Isobaric heat rejection in condenser.

Process 3-4: Isenthalpic expansion of saturated liquid in expansion device.

Process 4-1: Isobaric heat extraction in the evaporator.

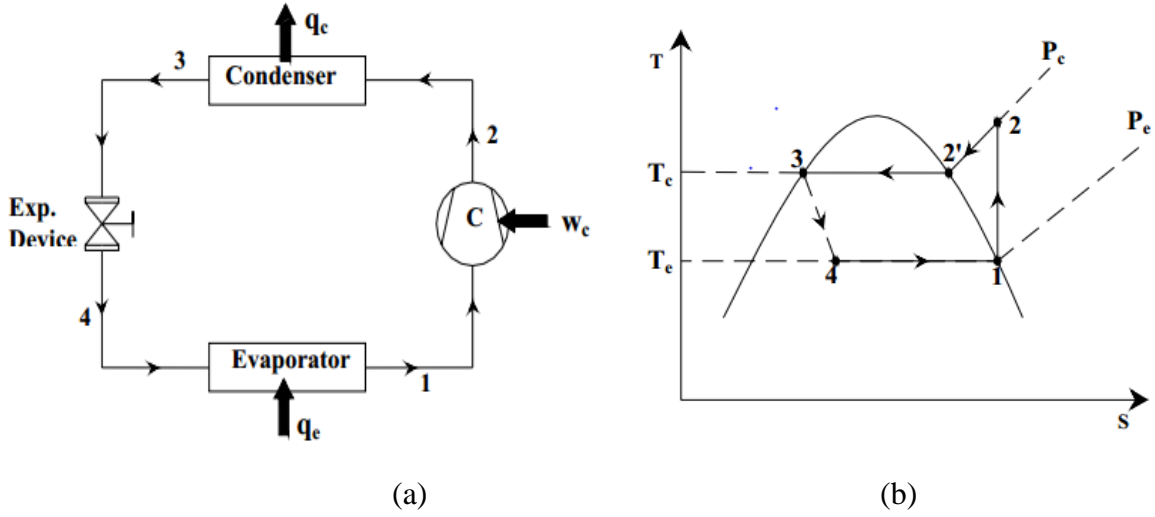


Figure 3.1 Single stage vapor compression refrigeration system: (a) Schematic diagram (b) Refrigeration cycle.

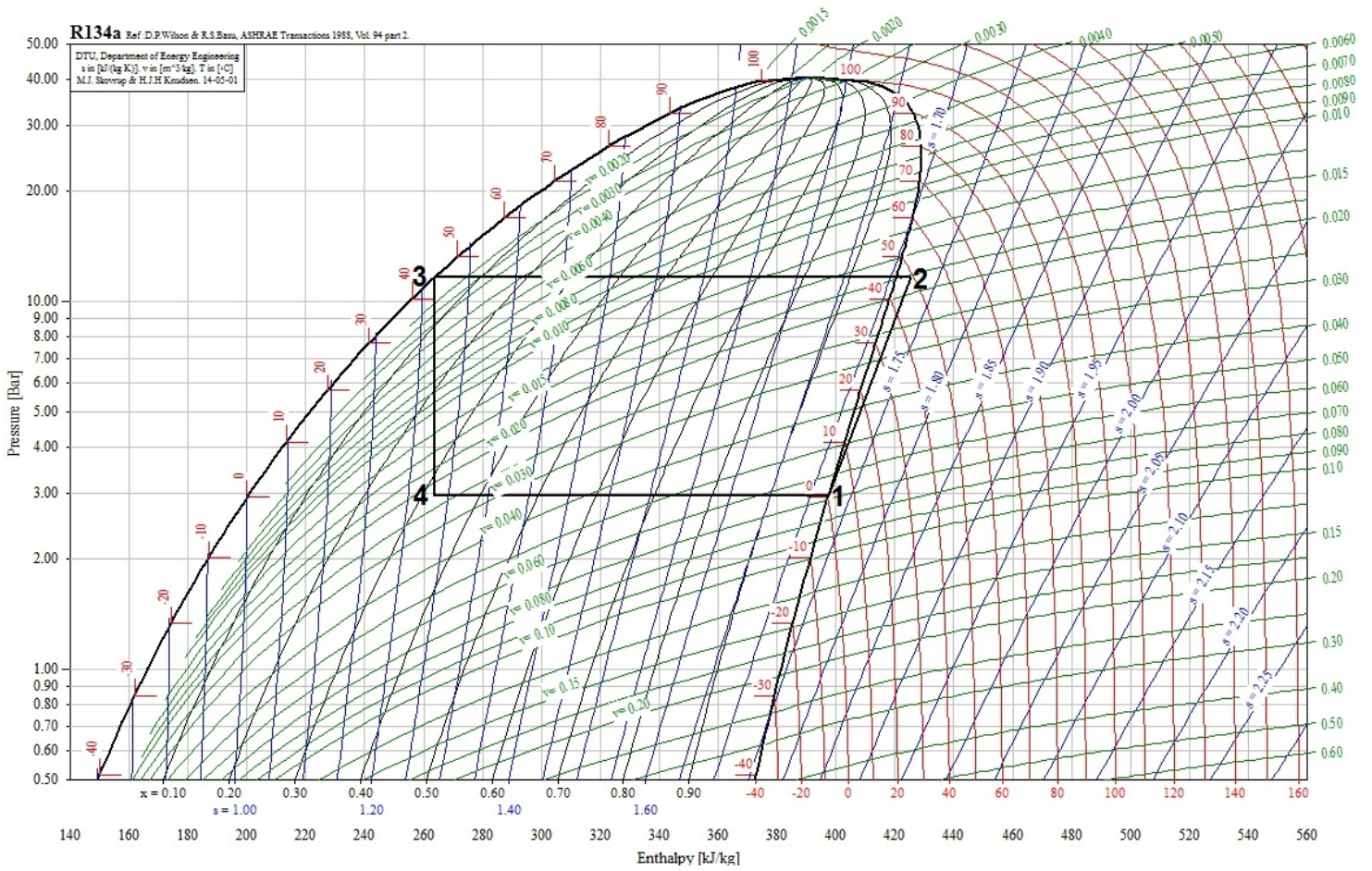


Figure 3.2 P-h Diagrams for R-134a.

The compression ratio for this cycle can be calculated as:

$$P_r = \frac{p_c}{p_e} \tag{3.1}$$

Where:

P_r : Compression ratio.

p_c : Condenser pressure.

p_e : Evaporator pressure $P_r = \frac{11.597}{2.949} = 3.93$

Table 3.2 Properties of Each Point on the Cycle

Point	T (°C)	P(bar)	h (KJ/K)	s (KJ/Kg.K)	V(m ³ /Kg)
1	0.2	2.949	397.319	1.7219	0.068411
2	49.267	11.597	425.6	1.7219	0.017832
3	45	11.597	263.172	1.214	0.00358
4	0.2	2.949	263.712	1.234	0.0125

3.3.2 Mass Flow Rate

The mass flow rate of refrigeration added can be determined by the following equation:

$$\dot{m}_R = \frac{Q_e}{q_e} \quad (3.2)$$

Where:

\dot{m}_R : Refrigerant flow rate [kg/s].

Q_e : Refrigeration capacity [KW].

q_e : Refrigeration effect [kJ/kg].

The refrigeration effect can be determined by the following equation:

$$q_e = (h_1 - h_4) \quad (3.3)$$

Where:

h_1 : Enthalpy of saturated vapor leaving the evaporator [kJ/kg].

h_4 : Enthalpy of the refrigerant entering the evaporator [kJ/kg].

$$q_e = 397.319 - 263.712 = 133.607 \text{ kJ/kg}$$

$$\dot{m}_R = \frac{1.2162}{133.607} = 0.00911 \text{ kg/s}$$

3.3.3 Condenser Load

The condenser load can be determined by the following equation:

$$Q_c = \dot{m}_R(h_2 - h_3) \quad (3.4)$$

Where:

Q_c : Condenser load [KW].

h_2 : Enthalpy of the gas discharged from the compressor [kJ/kg].

h_3 : Enthalpy of condensed liquid out of the condenser [kJ/kg].

$$Q_c = 0.00911 \times (425.6 - 263.172) = 1.48 \text{ KW}$$

3.3.4 Work of Compressor

The work of the compressor can be calculated by the following equation:

$$W = \dot{m}_R(h_2 - h_1) \quad (3.5)$$

Where:

W : Is the work of the compressor [KW].

$$W = 0.00911 \times (425.6 - 397.319) = 0.258 \text{ KW}$$

3.3.5 Coefficient of Performance

COP: Coefficient of performance of the cycle.

COP can be calculated by the following equation:

$$COP = \frac{Q_e}{W_{Total}} \quad (3.6)$$

$$COP = \frac{1.2162}{0.258} = 4.71$$

CHAPTER FOUR

DESIGN AND SELECTION OF THE SYSTEM'S COMPONENTS

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DESIGN AND SELECTION OF THE SYSTEM'S COMPONENTS

4.1 Introduction

In this chapter we will talk briefly about our device and its components. This device operates in two cycles, one is a closed cycle and the other is an open one.

The first cycle is the simple vapor compression refrigeration cycle, which is the closed cycle. The working fluid for this cycle is R134-a refrigerant. The aim of this cycle is to make the evaporator's temperature lower than the dew point temperature of air. The compressor in this cycle works by solar energy.

The second cycle is the open cycle, the working fluid for this cycle is the moist air. Here we use the evaporator of the first cycle to be the cooler of the moist air.

We enter the humid air to the device, humid air is sucked by a fan, air is forced to pass through the evaporator (air cooler), where water vapor condenses, also filters help us to catch water. We collect this liquid water and treat it depending on its use.

4.2 System Components Models

4.2.1 Compressor

In a refrigeration cycle, the compressor has two main functions within the refrigeration cycle. One function is to pump the refrigerant vapor from the evaporator so that the desired temperature

and pressure can be maintained in the evaporator. The second function is to increase the pressure of the refrigerant vapor through the process of compression, and simultaneously increase the temperature of the refrigerant vapor. By this change in pressure the superheated refrigerant flows through the system. [3]

Compressor selection criteria:

In the selection of a proper refrigerant compressor, the following criteria are considered:

- Refrigeration capacity.
- Volumetric flow rate.
- Compression ratio.
- Thermal and physical properties of the refrigerant.

In this project the hermetic compressor was selected as a result of the low capacity.

These compressors, are available for small capacities, motor and drive are sealed in compact welded housing. The refrigerant and lubricating oil are contained in this housing. Almost all small motor-compressor pairs used in domestic refrigerators, freezers, and air conditioners are of the hermetic type. Their revolutions per minute are either 1450 or 3000 rpm. Hermetic compressors can work for a long time in small-capacity refrigeration systems without any maintenance requirement and without any gas leakage, but they are sensitive to electric voltage fluctuations, which may make the copper coils of the motor burn. The cost of these compressors is very low. (Figure(4.1) shows hermetic compressor). [3]

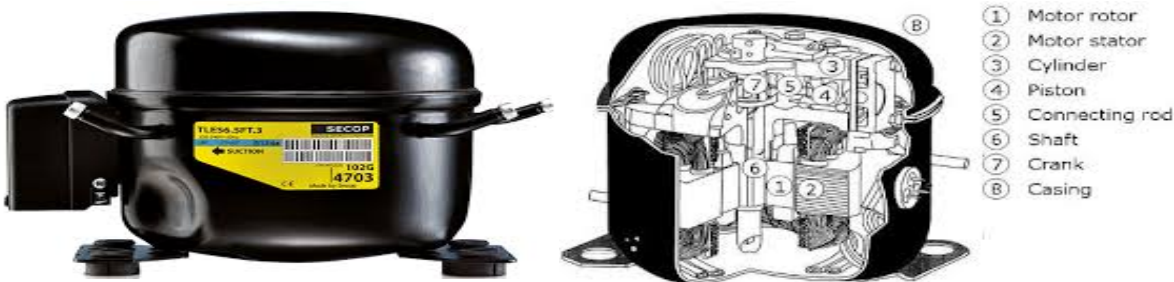


Figure 4.1 Reciprocating Hermetic Compressor.

4.2.1.1 Compressor Calculations

To determine the volumetric efficiency for the compressor equation (4.1) can be used:

$$\eta_v = \eta_c * \eta_h \quad (4.1)$$

Where:

η_v : Volumetric efficiency.

η_c : Volumetric efficiency due to clearance volume in compressor.

η_h : Volumetric efficiency due to heating occurs in compressor.

To determine the volumetric efficiency due to clearance volume in the compressor: [6]

$$\eta_c = 1 - c \left[\left(\frac{P_H}{P_L} \right)^{\frac{1}{n}} - 1 \right] \quad (4.2)$$

Where:

c : Clearance volume ratio between volumetric clearance and volume of cylinder of the compressor, $c = 0.04$ for low pressure different, $c = 0.02$ for high pressure different. [6]

n : Exponential coefficient of expansion for refrigerant, $n = 1$. [6]

P_H : High pressure of the cycle [bar].

P_L : Low pressure of the cycle [bar].

From table (3.2):

$P_H = 11.597$ bar.

$P_L = 2.949$ bar.

$$\eta_c = 1 - 0.02 \left[\left(\frac{11.597}{2.949} \right)^{\frac{1}{1}} - 1 \right] = 0.94 = 94\%$$

The volumetric efficiency due to heating occurs in compressor is: [7]

$$\eta_h = \frac{T_{evap}}{T_{cond}} \quad (4.3)$$

Where:

T_{evap} : Evaporator temperature [°K].

T_{cond} : Condenser temperature [°K].

$$\eta_h = \frac{273.2}{323} = 0.845 = 84.5\%$$

$$\eta_v = 0.94 * 0.845 = 0.8 = 80\%$$

The theoretical volume flow rate (V) of the compressor can be calculated in equation: [6]

$$V_{theo} = \dot{m} * v \quad (4.4)$$

Where:

V_{theo} : The theoretical volume flow rate of the compressor [m³/s].

\dot{m} : Mass flow rate of refrigerant [kg/s].

v : Specific volume at the inlet of compressor [m³/s].

$$V_{theo} = 0.00911 * 0.068411 = 6.23 * 10^{-4}.$$

To determine the actual volume flow rate by the equation: [7]

$$V_{act} = \frac{V_{theo}}{\eta_v} \quad (4.5)$$

Where:

V_{act} : Actual volumetric flow rate [m³/s].

$$V_{act} = \frac{6.23 * 10^{-4}}{0.8} = 7.8 * 10^{-4}$$

The main consider to select the compressor is the actual volumetric flow rate, so we choose a compressor that satisfy it.

From Tecumseh Company catalog we chose the compressor with code number AE4440Y which have displacement 10.33cm³ per revolution and 3000 RPM So:

The actual flow rate for the compressor can be calculated as the following:

$$V_{act} = V_{theo} * \eta_V \tag{4.6}$$

Where:

V_{act} : Actual volumetric flow rate for the compressor [m³/s].

V_{theo} : Theoretical volumetric flow rate for the compressor [m³/s].

η_V : Volumetric efficiency.

$$V_{theo} = 10.33 * 10^{-6} * (3000/60) = 5.165 * 10^{-4}$$

$$V_{act} = 5.165 * 10^{-4} * 0.8 = 4.132 * 10^{-4}$$

4.2.2 Evaporator

An evaporator, like condenser is also a heat exchanger. In an evaporator, the refrigerant boils or evaporates and in doing so absorbs heat from the substance being refrigerated. The name evaporator refers to the evaporation process occurring in the heat exchanger.[12]

The evaporator may be classified as natural convection type or forced convection type. In forced convection type, a fan or a pump is used to circulate the fluid being refrigerated and make it flow over the heat transfer surface, which is cooled by evaporation of refrigerant. In natural

convection type, the fluid being cooled flows due to natural convection currents arising out of density difference caused by temperature difference. The refrigerant boils inside tubes and evaporator is located at the top. The temperature of fluid, which is cooled by it, decreases and its density increases. It moves downwards due to its higher density and the warm fluid rises up to replace it. [12]

Air-forced convection evaporator was selected.

4.2.2.1 Evaporator Calculations and Design

Figure 4.2 description of the available evaporator and the following is geometrical data:

Evaporator length $L_e = 55$ cm.

Evaporator height $H_e = 13$ cm.

Evaporator width $W_e = 15$ cm.

Outer diameter of the pipe = 10 mm.

Number of rows = 4.

Number of columns = 6.

Number of fins in a row = 97 fins.

S_T : Transverse pitch of a tube bank [m].

S_L : Longitudinal pitch of a tube bank [m].

S_D : Diagonal pitch of a tube bank [m].

The transverse pitch of a tube bank (S_T) can be determined by the following equation:

$$S_T = \frac{\text{evaporator height}}{\text{number of rows}} \quad (4.7)$$

$$S_T = \frac{0.13}{4} = 0.0325 \text{ m}$$

The longitudinal pitch of a tube bank (S_L) can be determined by the following equation:

$$S_L = \frac{\text{evaporator width}}{\text{number of column}} \quad (4.8)$$

$$S_L = \frac{0.15}{6} = 0.025 \text{ m}$$

The diagonal pitch of a tube bank (S_D) can be determined by the following equation:[8]

$$S_D = \sqrt{(S_L)^2 + (S_T/2)^2} \quad (4.9)$$

$$S_D = \sqrt{(0.025)^2 + (0.0325/2)^2}$$

$$S_D = 0.0298 \text{ m}$$

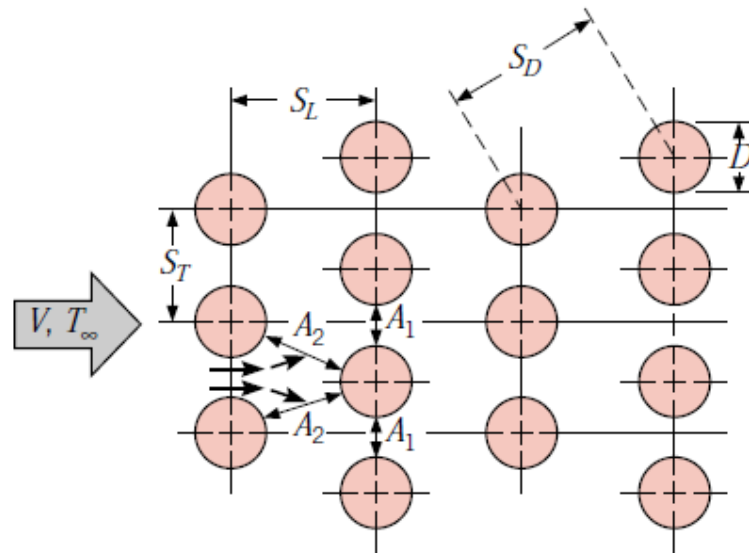


Figure 4.2 Construction of the evaporator.

Fin length (L_f) can be determined by the following equation:

$$L_f = S_T - D_o \quad (4.10)$$

$$L_f = 0.0325 - 0.01 = 0.0225 \text{ m}$$

Fin width (W_f) can be determined by the following equation:

$$W_f = S_L - D_o \quad (4.11)$$

$$W_f = 0.025 - 0.01 = 0.015 \text{ m}$$

Figure 4.3 show the fin elements number of fins in row.

To calculate the fin pitch the following equation can be used:

$$P_f = \frac{L_e}{\text{Number of fins}} \quad (4.12)$$

$$P_f = \frac{55}{97} = 0.567 \text{ cm}$$

Fine thickness:

$$t_f = 0.03 \text{ cm}$$

Then the bare tube thickness is:

$$t_b = P_f - t_f \quad (4.13)$$

$$t_b = 0.567 - 0.03 = 0.537 \text{ cm}$$

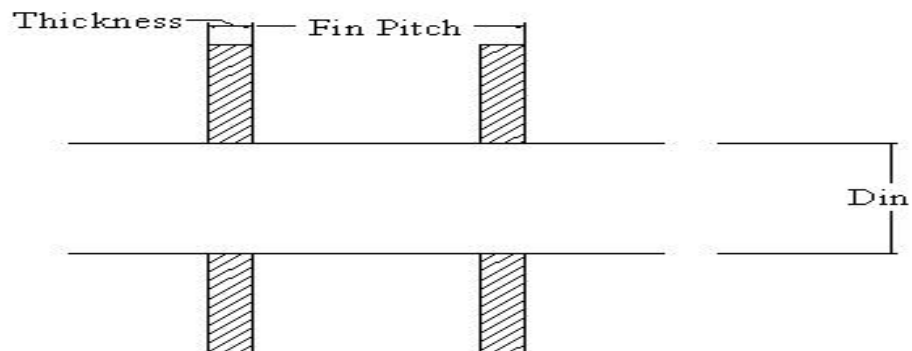


Figure 4.3 An element in the evaporator.

The design of the evaporator required many calculations such as fluid mechanical calculation, thermal calculation and area calculation, the sequence of design start with fluid mechanical calculation, in thermal calculation will be used the convection heat transfer equations for outer surface neglected the small thermal radiation from the wall until reaching area calculation.

$$Q = h_o * A * (T_w - T_\infty) \quad (4.14)$$

Where:

h_o : External convection heat transfer coefficient [$W/m^2\text{°C}$].

A : Surface area of heat transfer [m^2].

T_w : Outer evaporator wall temperature [°C].

T_{∞} : Free air temperature [°C].

To determine the external convection heat transfer coefficient (h_o) the following equation can be used: [4]

$$h_o = \frac{Nu * k}{D} \quad (4.15)$$

Where:

Nu : Nusselt's number.

k : Thermal conductivity of air at entrance of evaporator [$W/m\text{°C}$].

D : Outer diameter of condenser [m].

The Nusselt's number can be calculated by the equation:[4]

$$Nu_D = C Re^N Pr^{1/3} \quad (4.16)$$

$$0.4 < Re < 400000$$

Where:

Re : Reynolds number.

Pr : Prandtl number of air at film temperature.

C, N : Constants can be obtained from table (A-7) according to the following Reynolds number.

Reynolds number can be calculated from equation:

$$Re = \frac{\rho * V_{max} * D}{\mu} \quad (4.17)$$

Where:

ρ : Density of air at film temperature [kg/m^3].

V_{max} : Maximum velocity of air between the evaporator tubes [m/s].

D: Outer diameter of the evaporator tubes [m].

μ : Dynamic viscosity of air at film temperature [kg/m.s].

For flows normal to staggered arrangement the maximum flow velocity can be calculated as the following: [8]

$$V_{max} = \frac{S_T}{2(S_D - D)} V_{\infty} \quad (4.18)$$

$$\text{if } S_D = \sqrt{(S_L)^2 + (S_T/2)^2} < S_T + D/2$$

or

$$V_{max} = \frac{S_T}{S_T - D} V_{\infty} \quad (4.19)$$

Where:

V_{∞} : Is the free air velocity entering the evaporator [m/s].

The value of air velocity can be calculated by the following equation:

$$V_{\infty} = \frac{\dot{m}_a}{A\rho} \quad (4.20)$$

Where:

A: Cross-sectional area of fan [m²].

ρ : Density of air at film temperature [kg/m³].

The properties of air are evaluated at the film temperature.

$$T_f = \frac{T_w + T_{\infty}}{2} \quad (4.21)$$

Where:

T_f : Film temperature [°C].

T_{∞} : Free air temperature [°C].

T_w : Wall surface temperature [°C], assume that it equals the refrigerant's temperature.

$$T_f = \frac{0.2 + 18}{2} = 9.1^\circ\text{C} = 282.1\text{K}$$

Then from table (A-2):

$$\rho = 1.2197 \text{ kg/m}^3$$

$$\mu = 1.7577 * 10^{-5} \text{ kg/m.s}$$

$$K = 0.02482 \text{ W /m.}^\circ\text{C}$$

$$Pr = 0.713012$$

The value of air velocity can be calculated according to equation (4.20):

$$V_\infty = \frac{\dot{m}_a}{A\rho}$$

$$A = 0.245 * 0.035 = 8.575 * 10^{-3} \text{ m}^2$$

$$\dot{m}_a = 0.0434 \text{ kg/s}$$

$$V_\infty = \frac{0.0434}{8.575 * 10^{-3} * 1.2197} = 4.17 \text{ m/s}$$

The maximum flow velocity:

$$S_D < S_T + D/2$$

$$S_D = 0.0298 \text{ m}$$

$$\frac{S_T + D}{2} = \frac{0.0325 + 0.01}{2} = 0.02125 \text{ m}$$

The maximum flow velocity can be calculated according to equation (4.19):

$$V_{max} = \frac{S_T}{S_T - D} V_\infty$$

$$V_{max} = \frac{0.0325}{0.0325 - 0.01} * 4.17 = 6 \text{ m/s}$$

Then Reynolds number can be calculated from equation (4.17):

$$Re = \frac{\rho * V_{max} * D}{\mu}$$

$$Re = \frac{1.2197 * 6 * 0.01}{1.7577 * 10^{-5}}$$

$$Re = 4163.5$$

From staggered arrangement tube banks table (A-7):

$$C = 0.193$$

$$N = 0.618$$

By substitution in equation (4.16), Nussult's number is:

$$Nu_D = C Re^N Pr^{1/3}$$

$$Nu_D = 0.193(4163.5)^{0.618}(0.713012)^{1/3}$$

$$Nu_D = 29.7$$

Then, substitute in equation (4.15):

$$h_o = \frac{Nu * k}{D}$$

$$h_o = \frac{29.7 * 0.02482}{0.01}$$

$$h_o = 73.83 \text{ W/m}^2 \cdot \text{°C}$$

In order to calculate the total heat transfer from one element (one fin and on bare tube) the following equation is used:[9]

$$q_{total} = q_{fin \text{ act}} + q_{original} \tag{4.22}$$

Where:

q_{total} : The total heat transfer from the element [W].

$q_{fin \text{ act}}$: Actual heat transfer rate per fin [W].

$q_{original}$: Heat transfer rate from tube without fin [W].

To calculate the heat transfer rate from tube without fin, the following equation can be used:

$$q_{original} = h_o * A_{original} * (T_W - T_\infty) \tag{4.23}$$

Where:

$A_{original}$: The outer surface area of bare tube [m²].

T_w : Outer condenser wall temperature [°C].

Following equation used to calculate the outer surface area of the tube:

$$A_{original} = \pi DL$$

$$A_{original} = \pi * 10 * 10^{-3} * 5.37 * 10^{-3}$$

$$A_{original} = 1.68 * 10^{-4} m^2$$

Then, substitution in equation (4.23):

$$q_{original} = h_o * A_{original} * (T_w - T_\infty)$$

$$q_{original} = 73.83 * 1.68 * 10^{-4} * (18 - 0.2)$$

$$q_{original} = 0.22 W$$

q_{fin} can be calculated by the equation :

$$q_{fin} = h_o * A_{fin} * (T_w - T_\infty) \tag{4.24}$$

Where:

q_{fin} : Theoretical heat transfer rate per fin [W].

A_{fin} : The outer surface area for fin [m²].

To calculate the outer surface area for fin:

$$A_{fin} = 2(S_T S_L - A_{pip}) \tag{4.25}$$

$$A_{fin} = 2[0.0325 * 0.025 - \frac{\pi}{4}(0.01)^2]$$

$$A_{fin} = 1.468 * 10^{-3} m^2$$

Then substitute in equation (4.24) to calculate q_{fin} :

$$q_{fin} = h_o * A_{fin} * (T_w - T_\infty)$$

$$q_{fin} = 73.83 * 1.468 * 10^{-3} * (18 - 0.2)$$

$$q_{fin} = 1.93 W$$

Fin efficiency calculation:

To calculate the efficiency of the fin the following equation must be used: [10]

$$\eta_f = \frac{\tanh(mL'_f)}{mL'_f} \quad (4.26)$$

To calculate (L'_f) the following equation is used: [10]

$$L'_f = \left(\frac{L_f}{2}\right) \left[1 + 0.35 \ln \frac{\frac{d_o}{2} + \frac{L_f}{2}}{\frac{d_o}{2}}\right] \quad (4.27)$$

$$L'_f = \left(\frac{0.015}{2}\right) \left[1 + 0.35 \ln \frac{\frac{0.01}{2} + \frac{0.015}{2}}{\frac{0.01}{2}}\right]$$

$$L'_f = 0.01m$$

Then, to calculate (m) the following equation is used: [10]

$$m = \sqrt{\frac{2Ph_o}{kA_f}} \quad (4.28)$$

Where:

k: Thermal conductivity of aluminum fin [W/m°C].

$$P = 2 * t_f + 2 * L_f P = 2 * 0.3 * 10^{-3} + 2 * 0.0225$$

$$P = 0.0456m$$

$$A_f = t_f L_f$$

$$A_f = 0.3 * 10^{-3} * 0.0225$$

$$A_f = 6.75 * 10^{-6} m^2$$

$$m = \sqrt{\frac{73.83 * 0.0456}{202 * 6.75 * 10^{-6}}}$$

$$m = 49.7$$

Then substitute (m & L_f') in equation (4.26):

$$\eta_f = \frac{\tanh(49.7 * 0.01)}{49.7 * 0.01}$$

$$\eta_f = 0.925$$

So;

The heat transfer flow from the fin is:

$$q_{fin\ act} = q_{fin} \eta_f$$

$$q_{fin\ act} = 1.93 * 0.925$$

$$q_{fin\ act} = 1.78\ W$$

Now the total heat transfer from the element is:

$$q_{total} = q_{fin\ act} + q_{original}$$

$$q_{total} = 1.78 + 0.22$$

$$q_{total} = 2\ W$$

Now the number of elements (n) that needed to perform the evaporator load can be determined by the following equation:

$$n = \frac{Q_e}{q_{total}} \tag{4.29}$$

$$n = \frac{1217.62}{2}$$

$$n = 608\ \text{element.}$$

Number of element in available evaporator = N*R.

Where:

N: Number of element in rows.

R: Number of rows.

Number of element in available evaporator = $97*26=2522$ elements.

4.2.3 Condenser

There are several condensers to be considered when making a selection for installation. They are air-cooled, water-cooled, shell and tube, shell and coil, tube within a tube, and evaporative condensers. Each type of condenser has its own unique application. Some determining factors include the size and the weight of the unit, weather conditions, location (city or rural), availability of electricity, and availability of water. A wide variety of condenser configurations are employed in the process industry. Selection of condenser type is not easy and depends on the following criteria: [3]

- Condenser heat capacity.
- Condensing temperature and pressure.
- The flow rates of refrigerant and coolant.
- Design temperature for water and/or air.
- Operation period.
- Climatic conditions.

4.2.3.1 Air-cooled Condenser

As the name implies, in air-cooled condensers air is the external fluid, therefore; the refrigerant rejects heat to air flowing over the condenser. Air-cooled condensers can be further classified into natural convection type or forced convection type. [12]

Forced convection type was selected.

4.2.3.2 Forced Convection Condenser

In forced convection type condensers, the circulation of air over the condenser surface is maintained by using a fan or a blower. These condensers normally use fins on air-side for good heat transfer. The fins can be either plate type or annular type. [12]

The air-cooled condensers find applications in domestic, commercial, and industrial refrigerating, chilling, freezing, and air conditioning systems. The centrifugal fan air-cooled condensers are particularly used for heat recovery and auxiliary ventilation applications. In fact, they employ outside air as the cooling medium. Fans draw air past the refrigerant coil and the latent heat of the refrigerant is removed as sensible heat by the air stream. Figure(4.4) shows air-cooled condensers. [3]



Figure 4.4 Air-Cooled Condensers.

4.2.3.3 Condenser Calculations and Design

Figure(4.5) description of the available evaporator and the following is geometrical data:

Evaporator length $L_c = 30$ cm.

Evaporator height $H_c = 26$ cm.

Evaporator width $W_c = 5.7$ cm.

Outer diameter of the pipe = 10 mm.

Number of rows = 10.

Number of column = 3.

Number of fins in a row = 94 fins.

S_T : Transverse pitch of a tube bank [m].

S_L : Longitudinal pitch of a tube bank [m].

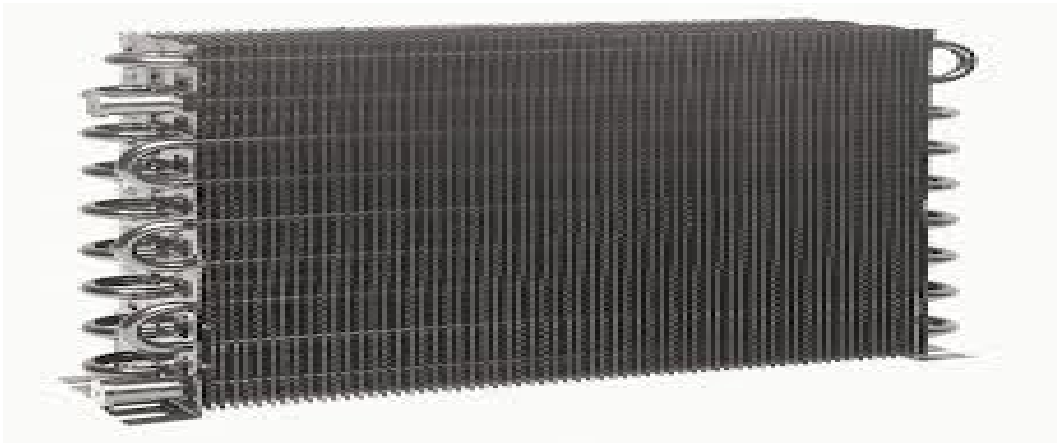


Figure 4.5 Construction of Condenser.

The transverse pitch of a tube bank (S_T) can be determined by the following equation:

$$S_T = \frac{\text{Condenser height}}{\text{number of rows}} \quad (4.30)$$

$$S_T = \frac{0.26}{10} = 0.026 \text{ m}$$

The longitudinal pitch of a tube bank (S_L) can be determined by the following equation:

$$S_L = \frac{\text{Condenser width}}{\text{number of column}} \quad (4.31)$$

$$S_L = \frac{0.057}{3} = 0.019 \text{ m}$$

Fin length (L_f) can be calculated according to equation (4.10):

$$L_f = S_T - D_o$$

$$L_f = 0.026 - 0.01 = 0.016 \text{ m}$$

Fin width (W_f) can be calculated according to equation (4.11):

$$W_f = S_L - D_o$$

$$W_f = 0.019 - 0.01 = 0.009 \text{ m}$$

Figure(4.6) show the fin elements number of fins in row.

The fin pitch can be calculated according to equation (4.12):

$$P_f = \frac{L_e}{\text{Number of fins}}$$

$$P_f = \frac{40}{94} = 0.425 \text{ cm}$$

Fin thickness:

$$t_f = 0.02 \text{ cm}$$

Then the bare tube thickness can be calculated according to equation (4.13):

$$t_b = P_f - t_f$$

$$t_b = 0.425 - 0.02 = 0.405 \text{ cm.}$$

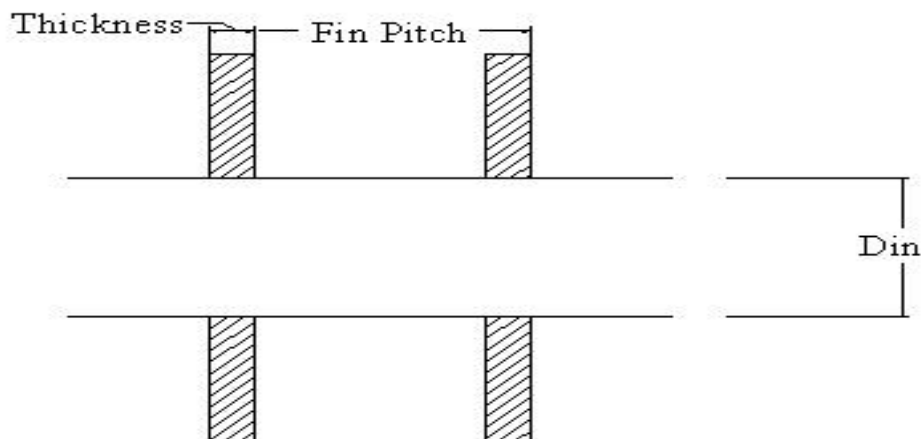


Figure 4.6 An element in the condenser.

The design the condenser required many calculations such as fluid mechanical calculation, thermal calculation and area calculation, the sequence of design start with fluid mechanical calculation, in thermal calculation will be used the convection heat transfer equations for outer surface neglected the small thermal radiation from the wall until reaching area calculation.

$$Q = h_o * A * (T_w - T_\infty) \quad (4.32)$$

Where:

h_o : External convection heat transfer coefficient [$W/m^2\text{°C}$].

A : Surface area of heat transfer [m^2].

T_w : Outer condenser wall temperature [°C].

T_∞ : Free air temperature [°C].

To determine the external convection heat transfer coefficient (h_o) can be calculated according to equation (4.15):

$$h_o = \frac{Nu * k}{D}$$

The Nusselt's number can be calculated according to equation (4.16):

$$Nu_D = C Re^N Pr^{1/3}$$

C, N : Constants can be obtained from table (A-5) according to the following considerations:

$$\frac{S_T}{D} = \frac{0.026}{0.01} = 2.6$$

$$\frac{S_L}{D} = \frac{0.019}{0.01} = 1.9$$

From staggered arrangement tube banks table A-5:

$$C = 0.521$$

$$N = 0.565$$

For flows normal to staggered arrangement the maximum flow velocity can be calculated as the following:[4]

$$V_{max} = \frac{V_{\infty} \left(\frac{S_T}{2}\right)}{\sqrt{\left[\left(\frac{S_T}{2}\right)^2 + S_L^2\right]} - D} \quad (4.33)$$

V_{∞} : Is the free air velocity entering the condenser (m/s).

The value of air velocity can be calculated by the following equation:

$$V_{\infty} = \frac{V \cdot}{A} \quad (4.34)$$

Where:

A : Cross sectional area of condenser [m^2].

$V \cdot$: Flow rate of air in the condenser [m^3/s].

Selected fan is (Perma-cool 19127 from SUMMIT Company).

$$V \cdot = 412 \text{ cfm}$$

$$V \cdot = 0.1944423 \text{ m}^3/s$$

$$A = 0.3 * 0.26 = 0.078 \text{ m}^2$$

$$V_{\infty} = \frac{0.1944423}{0.078} = 2.49 \text{ m/s}$$

$$V_{max} = \frac{2.49 \left(\frac{0.026}{2}\right)}{\sqrt{\left[\left(\frac{0.026}{2}\right)^2 + 0.019^2\right]} - 0.01} = 2.488 \text{ m/s}$$

The properties of air are evaluated at the film temperature can be calculated according to equation (4.21):

$$T_f = \frac{T_w + T_{\infty}}{2}$$

$$T_f = \frac{50 + 35}{2} = 42.5^\circ\text{C} = 315.5\text{K}$$

Then from table (A-2):

$$\rho = 1.12358 \text{ kg/m}^3$$

$$\mu = 1.93064 * 10^{-5} \text{ kg/m.s}$$

$$K = 0.027537 \text{ W /m.}^\circ\text{C}$$

$$Pr = 0.7047$$

Then Reynolds number can be calculated from equation (4.17):

$$Re = \frac{\rho * V_{max} * D}{\mu}$$

$$Re = \frac{1.12358 * 2.88 * 0.01}{1.93064 * 10^{-5}}$$

$$Re = 1676.1$$

By substitution in equation (4.16), Nussult's numbers is:

$$Nu_D = 0.521(1676.1)^{0.565}(0.7047)^{1/3}$$

$$Nu_D = 30.75$$

Then, substitute in equation (4.15):

$$h_o = \frac{Nu * k}{D}$$

$$h_o = \frac{30.75 * 0.027537}{0.01}$$

$$h_o = 84.67 \text{ W/m}^2 \cdot ^\circ\text{C}$$

This is the heat transfer coefficient that would be obtained if there are ten rows deep of tubes in the direction of the flow. Because there are only three rows, this value must be multiplied by the factor 0.83, as determined from table A-6. [4]

$$h_o = 84.67 * 0.83 = 70.27 \text{ W/m}^2 \cdot ^\circ\text{C}$$

The total heat transfer from one element (one fin and on bare tube) can be calculated according to equation (4.22):

$$q_{total} = q_{fin\ act} + q_{original}$$

The heat transfer rate from tube without fin can be calculated according to equation (4.23):

$$q_{original} = h_o * A_{original} * (T_w - T_\infty)$$

Following equation used to calculate the outer surface area of the tube:

$$A_{original} = \pi DL$$

$$A_{original} = \pi * 10 * 10^{-3} * 3 * 10^{-3}$$

$$A_{original} = 9.42 * 10^{-5} m^2$$

Then, substitution in equation (4.23):

$$q_{original} = 70.27 * 9.42 * 10^{-5} * (50 - 35)$$

$$q_{original} = 0.099 [W]$$

q_{fin} can be calculated according to equation (4.24):

$$q_{fin} = h_o * A_{fin} * (T_w - T_\infty)$$

To calculate the outer surface area for fin using (4.25):

$$A_{fin} = 2(S_T S_L - A_{pip})$$

$$A_{fin} = 2[0.026 * 0.019 - \frac{\pi}{4} (0.01)^2]$$

$$A_{fin} = 8.31 * 10^{-4} m^2$$

Then substitute in equation (4.24) to calculate q_{fin} :

$$q_{fin} = 70.27 * 8.31 * 10^{-4} (50 - 35)$$

$$q_{fin} = 0.8 W$$

Fin efficiency calculation:

The efficiency of the fin can be calculated according to equation (4.26):

$$\eta_f = \frac{\tanh(mL'_f)}{mL'_f}$$

(L'_f) can be calculated according to equation (4.27):

$$L'_f = \left(\frac{L_f}{2}\right) \left[1 + 0.35 \ln \frac{\frac{d_o}{2} + \frac{L_f}{2}}{\frac{d_o}{2}} \right]$$

$$L'_f = \left(\frac{0.016}{2}\right) \left[1 + 0.35 \ln \frac{\frac{0.01}{2} + \frac{0.016}{2}}{\frac{0.01}{2}} \right]$$

$$L'_f = 0.0106m$$

Then, to calculate (m) can be calculated according to equation (4.28):

$$m = \sqrt{\frac{2Ph_o}{kA_f}}$$

$$P = 2 * t_f + 2 * L_f P = 2 * 0.2 * 10^{-3} + 2 * 0.026$$

$$P = 0.0524m$$

$$A_f = t_f L_f$$

$$A_f = 0.2 * 10^{-3} * 0.026$$

$$A_f = 5.2 * 10^{-6} m^2$$

$$m = \sqrt{\frac{70.27 * 0.0524}{202 * 5.2 * 10^{-6}}}$$

$$m = 59.21$$

Then substitute $(m \& L'_f)$ in equation (4.26):

$$\eta_f = \frac{\tanh(59.21 * 0.01)}{59.21 * 0.01}$$

$$\eta_f = 0.897$$

So;

The heat transfer flow from the fin is:

$$q_{fin\ act} = q_{fin}\eta_f$$

$$q_{fin\ act} = 0.8 * 0.897$$

$$q_{fin\ act} = 0.717\ W$$

Now the total heat transfer from the element is:

$$q_{total} = q_{fin\ act} + q_{original}$$

$$q_{total} = 0.717 + 0.099$$

$$q_{total} = 0.816\ W$$

Now the number of elements (n) needed to perform the condenser's load can be calculated according to equation (4.29):

$$n = \frac{Q_c}{q_{total}}$$

$$n = \frac{1480.3}{0.816}$$

$$n = 1814\ \text{elements.}$$

Number of elements in available condenser = N*R

Where:

N: Number of element in rows.

R: Number of rows.

Number of elements in available condenser = 94*30= 2820 elements.

4.2.4 Connecting Pipes

Refrigerant pipes transport refrigerant through the compressor, condenser, expansion valve, and evaporator to provide the required refrigeration effect. For halocarbon refrigerants, the refrigerant pipes are usually made of copper. However, if ammonia is used as the refrigerant, the pipes are always made of steel. [5]

Three types of copper tubing are used for refrigerant piping. Type K is heavy-duty, and type M is lightweight. Type L is the standard copper tubing most widely used in refrigeration systems. Copper tubing installed in refrigeration systems should be entirely free of dirt, scale, and oxides. The open ends of clean new tubes should be capped to keep contaminants out. [5]

4.2.4.1 Pipes Design

Pipes carrying fluids at high pressure develop both radial and tangential stresses with values that depend on the radius of the element under consideration. In determining the radial stress (σ_r) and the tangential stress (σ_t), by using the assumption that the longitudinal elongation is constant around the circumference of the pipe. By referring to figure 4.7 it can be shown that tangential and radial stresses exist whose magnitudes represented in equation (4.35) and (4.36).

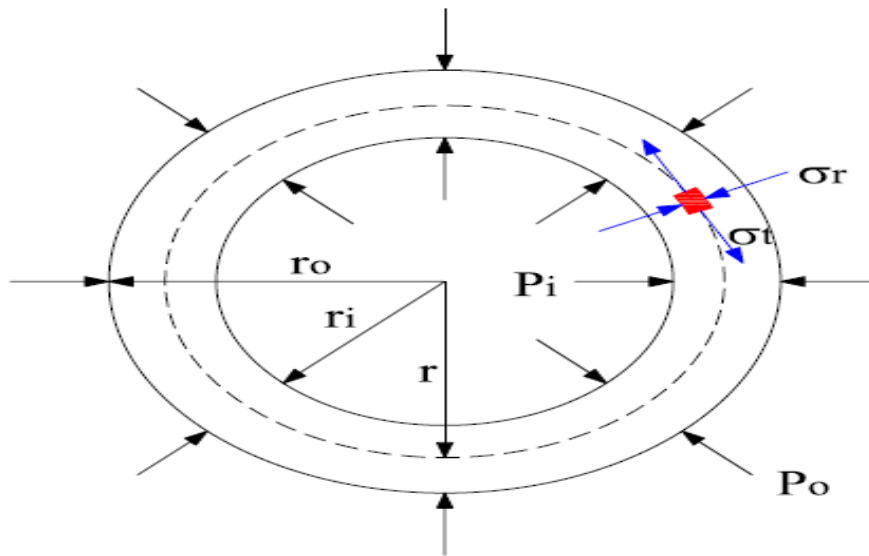


Figure 4.7 Stresses, internal and external pressures.

The design should be taken at inner diameter ($r = r_i$), because it is the critical point (dangerous point). And at ($P_o = 0$) as usual positive indicate tension and negative compression. The equation becomes as the following:

$$\sigma_t = \frac{P_i r_i^2 - P_o r_o^2 - \frac{r_i^2 r_o^2 (P_o - P_i)}{r^2}}{r_o^2 - r_i^2} \quad (4.35)$$

$$\sigma_r = \frac{P_i r_i^2 - P_o r_o^2 + r_i^2 r_o^2 (P_o - P_i) / r^2}{r_o^2 - r_i^2} \quad (4.36)$$

Where:

σ_t : Tangential stress [MPa].

σ_r : Radial stress [MPa].

P_i : Inner pressure [MPa].

P_o : Outer pressure [MPa].

r_o : Inner diameter [m].

r_i : Outer diameter [m].

r : Mean diameter [m].

The design should be taken at inner diameter ($r = r_i$), because it is the critical point (dangerous point). And at ($P_o=0$) as usual positive indicate tension and negative compression. The equation becomes as the following: [11]

At $r=r_i$, $P_o=0$, then:

$$\sigma_t = \left[\frac{r_i^2 + r_o^2}{r_o^2 - r_i^2} \right] \times P_i \quad (4.37)$$

$$\sigma_r = \left[\frac{r_i^2 - r_o^2}{r_o^2 - r_i^2} \right] \times P_i \quad (4.38)$$

$$\sigma_r = -P_i(4.39)$$

σ_t can be calculated from the following equation: [11]

$$\sigma' = \sqrt{\sigma_t^2 - \sigma_t \sigma_r + \sigma_r^2} \quad (4.40)$$

$$n = \frac{S_y}{\sigma'} \quad (4.41)$$

Where:

σ' : Von Misses stress (equivalent stress) [MPa]. [11]

N: Factor of safety.

If $n > 1$ the system is safe, if $n \leq 1$ the system is failure.

S_y : Yield strength (for copper 70 MPa).

To calculate the thickness of the pipe according to the previous equations, the internal diameter of the pipe should be calculated, by knowledge the velocity of the refrigerant inside pipes.

4.2.4.2 Pipe Design in the Suction Line

After the compressor the pressure is 2.949 bar and pipes in this side must bear this pressure and the internal diameter can be determined by the following:

$$Q = VA \tag{4.42}$$

Where:

Q : Flow rate [m^3/s].

V : Velocity of the refrigerant [m/s].

A : Cross sectional area [m^2].

So

$$Q = \dot{m} * v \tag{4.43}$$

Where:

\dot{m} : Mass flow rate of refrigerant (0.00908 kg/s).

v : Specific volume at compressor outlet (0.017832 m^3/kg).

So by equation (4.42):

$$Q = 0.00792 \times 0.017532 = 1.412 \times 10^{-4} \text{ m}^3/\text{s}.$$

By substitution in equation (4.43)

From table A-8 $V = 4 \text{ m}/\text{s}$

$$A = \frac{Q}{V} = \frac{1.412 \times 10^{-4}}{4} = 3.53 \times 10^{-5} m^2$$

$$A = \frac{\pi d^2}{4} \rightarrow d = \sqrt{\frac{4 \times 3.53 \times 10^{-5}}{\pi}} = 6.7 \times 10^{-3} m$$

So the internal diameter of the pipe in high pressure side is $d = 3.871 \times 10^{-3} m$.

By substitution in equation (4.41)

$$n = \frac{S_y}{\sigma'}$$

Let $n=2$

$$\sigma' = \frac{70}{2} = 35 MPa = 350 bar$$

By substitution in equation (4.40)

$$\sigma' = \sqrt{\sigma_t^2 - \sigma_t \sigma_r + \sigma_r^2}$$

$$\sigma_r = -P_i = -2.949 bar$$

$$350 = \sqrt{\sigma_t^2 - 2.949\sigma_t + 8.7}$$

$$\sigma_t^2 + 2.949\sigma_t = 122491.3$$

$$\sigma_t = 354.3 bar$$

From equation (4.37):

$$\sigma_t = \left[\frac{r_i^2 + r_o^2}{r_o^2 - r_i^2} \right] \times P_i$$

$$r_i = \frac{6.7}{2} = 3.35 mm$$

$$354.3 = \frac{(3.35)^2 + r_o^2}{r_o^2 - (3.35)^2} \times 2.949$$

$$r_o = 3.38 \text{ mm}$$

The thickness (t) of the pipe in high pressure side is

$$t = r_o - r_i = 3.38 - 3.35 = 0.03 \text{ mm}$$

The internal and outer diameter in inch:

$$d_{i,inch} = \frac{d_{i,mm}}{25.4} = \frac{3.38}{25.4} = 0.133 \text{ in}$$

$$d_{o,inch} = \frac{d_{o,mm}}{25.4} = \frac{3.35}{25.4} = 0.132 \text{ in}$$

4.2.4.3 Pipe Design in the Discharge Line

After the compressor the pressure is 11.597 bar and pipes in this side must bear this pressure and the internal diameter can be determined by the following:

By equation (4.42):

$$Q = 0.00792 \times 0.017532 = 1.412 \times 10^{-4} \text{ m}^3/\text{s}.$$

By substitution in equation (4.43)

From table (A-8) $V = 10 \text{ m/s}$

$$A = \frac{Q}{V} = \frac{1.412 \times 10^{-4}}{10} = 1.412 \times 10^{-5} \text{ m}^2$$

$$A = \frac{\pi d^2}{4} \rightarrow d = \sqrt{\frac{4 \times 1.412 \times 10^{-5}}{\pi}} = 4.24 \times 10^{-3} \text{ m}$$

So the internal diameter of the pipe in high pressure side is $d = 4.24 \times 10^{-3} \text{ m}$.

By substitution in equation (4.41)

$$n = \frac{S_y}{\sigma'}$$

Let $n=2$

$$\sigma' = \frac{70}{2} = 35 \text{ MPa} = 350 \text{ bar}$$

By substitution in equation (4.40):

$$\sigma' = \sqrt{\sigma_t^2 - \sigma_t \sigma_r + \sigma_r^2}$$

$$\sigma_r = -P_i = -11.597 \text{ bar.}$$

$$350 = \sqrt{\sigma_t^2 - 11.597\sigma_t + 134.5}$$

$$\sigma_t^2 + 11.597\sigma_t = 122365$$

$$\sigma_t = 344.1 \text{ bar}$$

From equation (4.37):

$$\sigma_t = \left[\frac{r_i^2 + r_o^2}{r_o^2 - r_i^2} \right] \times P_i$$

$$r_i = \frac{4.24}{2} = 2.12 \text{ mm}$$

$$344.1 = \frac{(2.12)^2 + r_o^2}{r_o^2 - (2.12)^2} \times 11.597$$

$$r_o = 2.19 \text{ mm}$$

The thickness (t) of the pipe in high pressure side is

$$t = r_o - r_i = 2.19 - 2.12 = 0.07 \text{ mm}$$

The internal and outer diameter in inch:

$$d_{o,inch} = \frac{d_{o,mm}}{25.4} = \frac{2.19}{25.4} = 0.086 \text{ in}$$

$$d_{i,inch} = \frac{d_{i,mm}}{25.4} = \frac{2.12}{25.4} = 0.083 \text{ in}$$

4.2.4.4 Pipe Design in Liquid Line after the Condenser

After the compressor the pressure is 11.597 bar and pipes in this side must bear this pressure and the internal diameter can be determined by the following:

By equation (4.42) :

$$Q = 0.00792 \times 0.017532 = 1.412 \times 10^{-4} \text{ m}^3/\text{s}.$$

From table (A-8) $V = 1.5 \text{ m/s}$

By substitution in equation (4.43)

$$A = \frac{Q}{V} = \frac{1.412 \times 10^{-4}}{1.5} = 9.41 \times 10^{-5} \text{ m}^2$$

$$A = \frac{\pi d^2}{4} \rightarrow d = \sqrt{\frac{4 \times 9.41 \times 10^{-5}}{\pi}} = 10.9 \times 10^{-3} \text{ m}$$

So the internal diameter of the pipe in high pressure side is $d = 10.9 \times 10^{-3} \text{ m}$.

By substitution in equation (4.41):

$$n = \frac{S_y}{\sigma'}$$

Let $n=2$

$$\sigma' = \frac{70}{2} = 35 \text{ MPa} = 350 \text{ bar}$$

By substitution in equation (4.40):

$$\sigma' = \sqrt{\sigma_t^2 - \sigma_t \sigma_r + \sigma_r^2}$$

$$\sigma_r = -P_i = -11.597 \text{ bar.}$$

$$350 = \sqrt{\sigma_t^2 - 11.597\sigma_t + 134.5}$$

$$\sigma_t^2 + 11.597\sigma_t = 122365$$

$$\sigma_t = 344.1 \text{ bar}$$

From equation (4.37):

$$\sigma_t = \left[\frac{r_i^2 + r_o^2}{r_o^2 - r_i^2} \right] \times P_i$$

$$r_i = \frac{10.9}{2} = 5.45 \text{ mm}$$

$$344.1 = \frac{(5.45)^2 + r_o^2}{r_o^2 - (5.45)^2} \times 11.597$$

$$r_o = 5.64 \text{ mm}$$

The thickness (t) of the pipe in high pressure side is

$$t = r_o - r_i = 5.64 - 5.45 = 0.19 \text{ mm}$$

The internal and outer diameter in inch:

$$d_{i,inch} = \frac{d_{i,mm}}{25.4} = \frac{5.64}{25.4} = 0.222 \text{ in}$$

$$d_{o,inch} = \frac{d_{o,mm}}{25.4} = \frac{5.45}{25.4} = 0.215 \text{ in}$$

4.2.4.5 Pipe Design in Liquid Line after the Expansion Valve

After the compressor the pressure is 2.949 bar and pipes in this side must bear this pressure and the internal diameter can be determined by the following:

By equation (4.42):

$$Q = 0.00792 \times 0.017532 = 1.412 \times 10^{-4} \text{ m}^3/\text{s}.$$

By substitution in equation (4.43)

From table (A-8) $V = 1.5 \text{ m/s}$

$$A = \frac{Q}{V} = \frac{1.412 \times 10^{-4}}{1.5} = 9.41 \times 10^{-5} \text{ m}^2$$

$$A = \frac{\pi d^2}{4} \rightarrow d = \sqrt{\frac{4 \times 9.41 \times 10^{-5}}{\pi}} = 10.9 \times 10^{-3} \text{ m}$$

So the internal diameter of the pipe in high pressure side is $d = 3.871 \times 10^{-3} \text{ m}$.

By substitution in equation (4.41)

$$n = \frac{S_y}{\sigma'}$$

Let $n=2$

$$\sigma' = \frac{70}{2} = 35 \text{ MPa} = 350 \text{ bar}$$

By substitution in equation (4.40)

$$\sigma' = \sqrt{\sigma_t^2 - \sigma_t \sigma_r + \sigma_r^2}$$

$$\sigma_r = -P_i = -2.949 \text{ bar}$$

$$350 = \sqrt{\sigma_t^2 - 2.949\sigma_t + 8.7}$$

$$\sigma_t^2 + 2.949\sigma_t = 122491.3$$

$$\sigma_t = 354.3 \text{ bar}$$

From equation (4.37)

$$\sigma_t = \left[\frac{r_i^2 + r_o^2}{r_o^2 - r_i^2} \right] \times P_i$$

$$r_i = \frac{10.9}{2} = 5.45 \text{ mm}$$

$$354.3 = \frac{(3.35)^2 + r_o^2}{r_o^2 - (3.35)^2} \times 2.949$$

$$r_o = 5.54 \text{ mm}$$

The thickness (t) of the pipe in high pressure side is

$$t = r_o - r_i = 5.54 - 5.45 = 0.09 \text{ mm}$$

The internal and outer diameter in inch:

$$d_{o,inch} = \frac{d_{o,mm}}{25.4} = \frac{5.54}{25.4} = 0.218 \text{ inch}$$

$$d_{i,inch} = \frac{d_{i,mm}}{25.4} = \frac{5.45}{25.4} = 0.215 \text{ inch}$$

4.2.4.6 Pipes Selection

By referring to copper hand book (appendix A, table A-3), the suitable type selected is ACR type (Air-conditioning and Refrigeration Field Service), and according to pervious calculations, the following result is for pipe selection using table (A-4) The following table represents the pipes dimensions (calculated and standards):

Table 4.1 Pipes Dimensions

Pipe Position	Calculated Diameters (mm)		Calculated Diameters (inch)		Standard Size (inch)	Nominal Dimensions (inch)		
	d _i	d _o	d _i	d _o		d _i	d _o	Thickness
Between Evaporator & Compressor	6.70	6.76	0.264	0.266	3/8	0.311	0.375	0.032
Between Compressor & Condenser	4.30	4.38	0.166	0.172	1/4	0.19	0.25	0.03
Between Condenser & Expansion Valve	10.90	11.28	0.430	0.444	1/2	0.436	0.50	0.032
Between Expansion Valve & Evaporator	10.90	11.08	0.430	0.436	1/2	0.436	0.50	0.032

4.2.5 Throttling Devices

4.2.5.1 Thermostatic Expansion Valve (TEV)

Thermostatic expansion valve is the most versatile expansion valve and is most commonly used in refrigeration systems. A thermostatic expansion valve maintains a constant degree of superheat at the exit of evaporator; hence it is most effective for dry evaporators in preventing the slugging of the compressors since it does not allow the liquid refrigerant to enter the compressor. [12]

For this reason the thermostatic expansion valve was selected in this project.

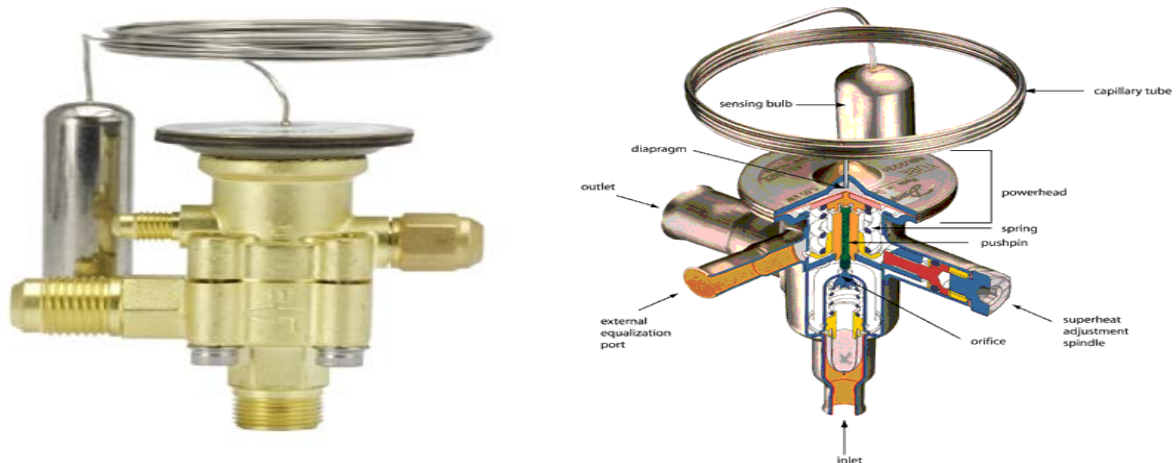


Figure 4.8 Thermostatic Expansion Valve.

4.2.6 Auxiliary Components

The auxiliary components is very important in the refrigeration system its working together with main components allowing system works very well, and we will discuss some auxiliaries in the following:

4.2.6.1 Filter Dryer

Moisture may freeze in the expansion especially in low temperature system, and impair expansion valve operation. Foreign matter in the refrigeration system may also damage the expansion valve, control valves, and compressor. Moisture and foreign matter must be removed from the refrigeration system by means of a filter dryer. Figure 4.9 shows a filter dryer. It contains a molded, porous core made of material with high moisture affinity, such as activated alumina, silica gel, and acid-neutralizing agents to remove moisture and foreign matter from the refrigerant. The core should be replaced after a certain period of operation. The filter dryer is usually installed at the liquid line immediately before the expansion valve. [5]



Figure 4.9 Filter dryer. .

4.2.6.2 Sight Glass

A sight glass is a small glass port used to observe the condition of refrigerant in the liquid line. The sight glass is located just before the expansion valve. Bubbles seen through the sight glass indicate the presence of the flash gas instead of the refrigerant. The presence of flash gas always

indicates that the evaporators capacity is reduced because of a shortage of refrigerant or insufficient sub-cooling. [5]



Figure 4.10 Sight glass.

CHAPTER FIVE

ELECTRICAL DESIGN AND SELECTION

CHAPTER FIVE

ELECTRICAL DESIGN AND SELECTION

5.1 Introduction

This chapter presents the protection of the system. Protection is one of the most important issue in any system. Any electrical failure occurs may destroy some expensive components especially the compressor.

5.2 Types of Electrical Circuits

There are two types of electrical circuits in general. There are power circuits and control circuits. Capacities for small units are usually the control and power of one; either for units with high capacities controlling circuit is controlled separately from the power circuit. [14]

5.2.1 Control Circuit

This circuit is working to influence the controls to follow up the implementation of required control program as defined by introducing elements operating according to the requirements of control thermostat and unequivocal pressure and break convection. Also working to introduce elements of the capacity as the exact timing advance. Often control circuit is working with single

phase. And potential voltages in control circuit are less or equal in power circuit. The energy consumed to control much less of the energy power circuit. [14]

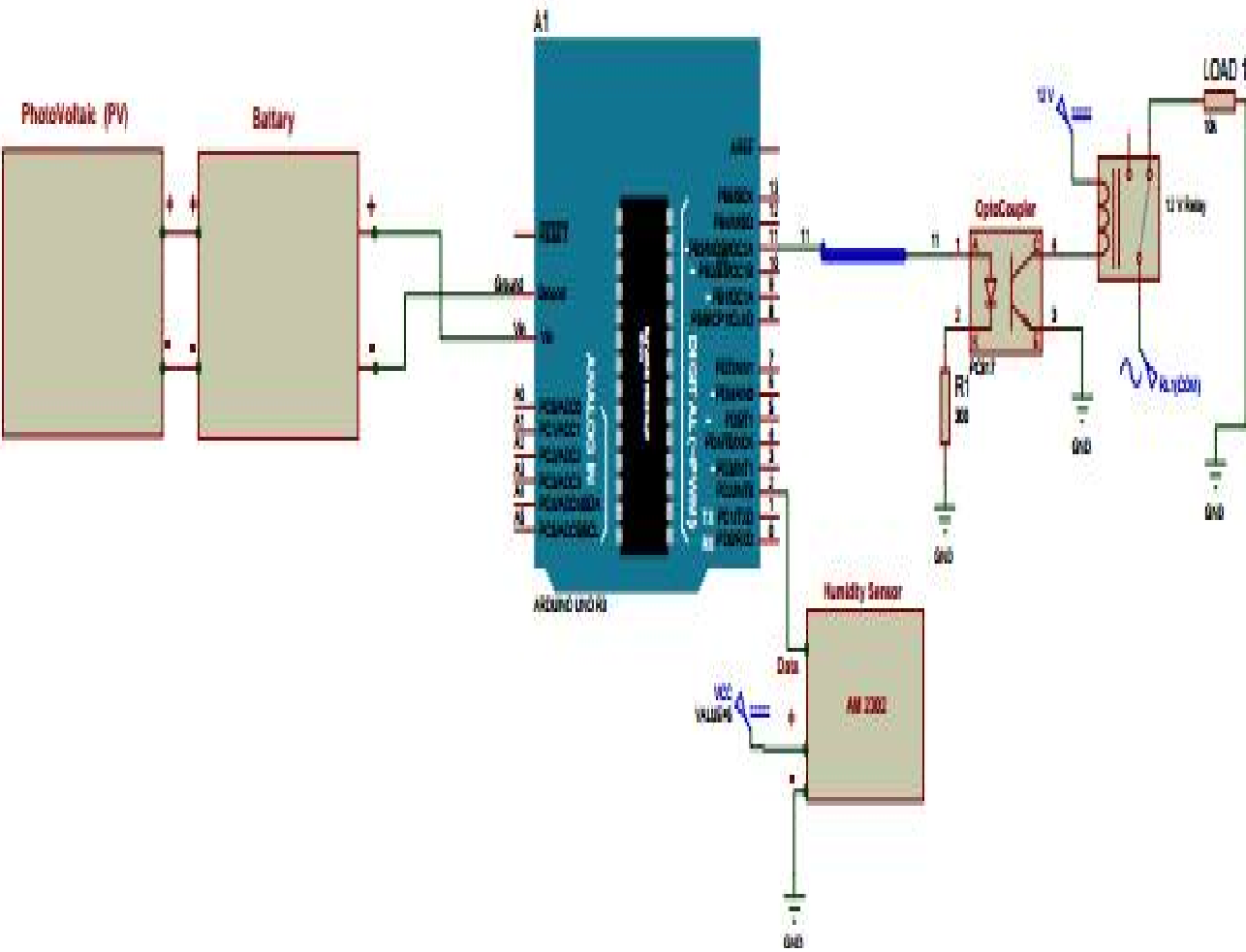


Figure 5.1 Control unit.

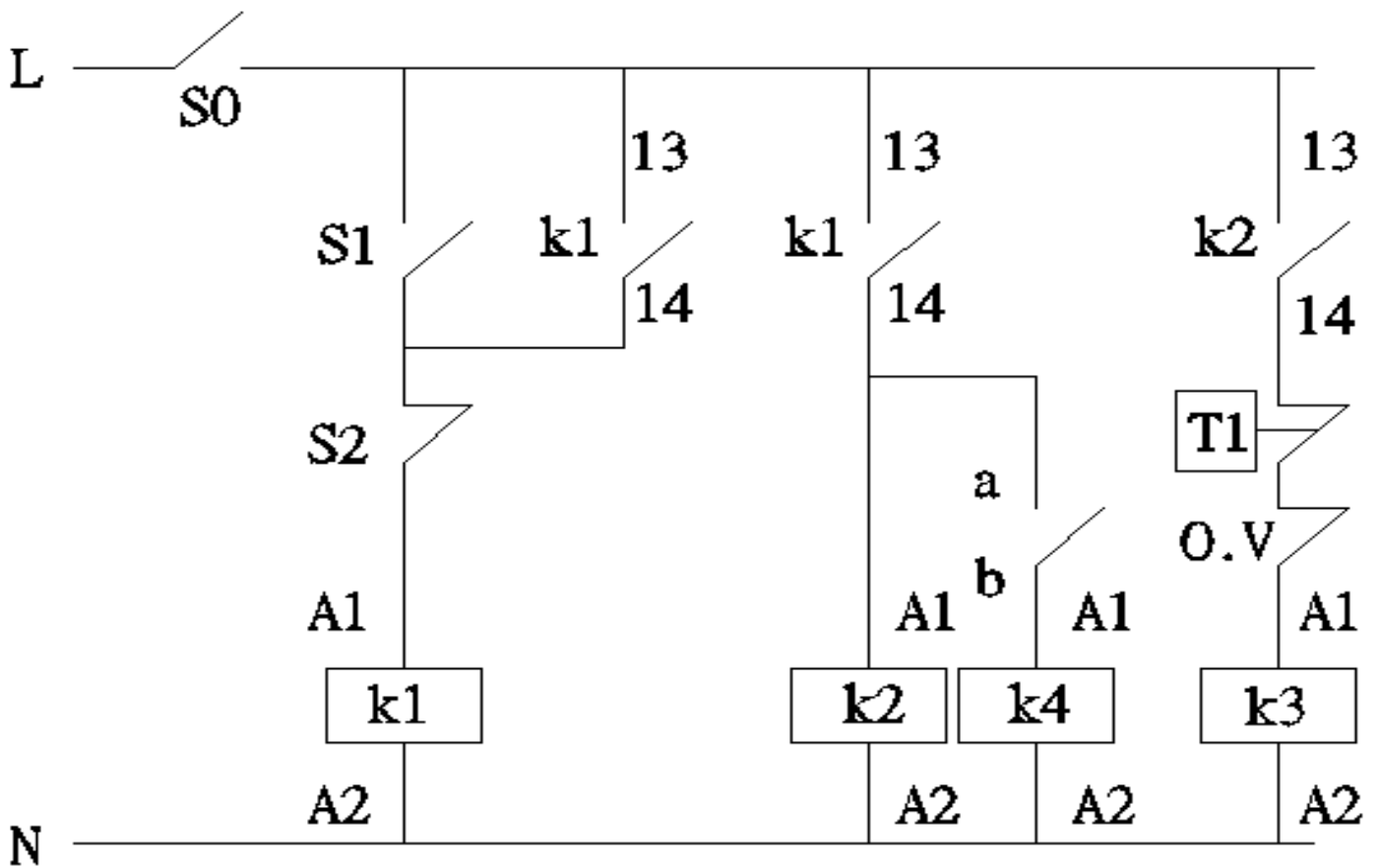


Figure 5.2 Control circuit.

5.2.2 Power Circuit

Power circuit is working to operate or stop power elements such as motors depending on the signal of the control circuit. The potential voltage and the electric power consumed in the power circuit equal to or greater than what is used in the control circuit. The power circuit is working in one or three phases. [14]

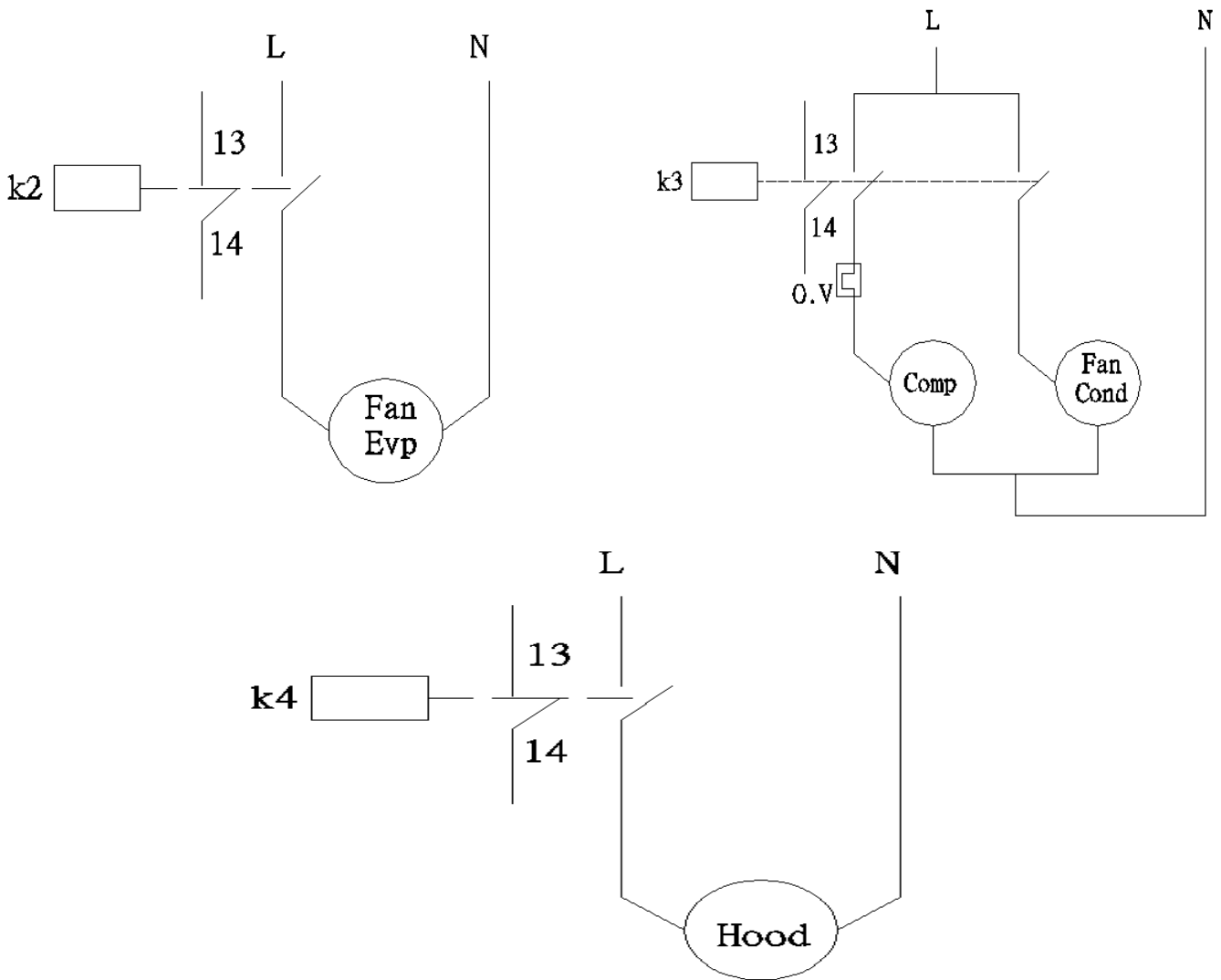


Figure 5.3 Power circuit.

5.3 Components of Electrical Circuits

The electrical circuit is built with more conditions; the first the requirement is economy and this is important in the installation of any equipment. Building cost is an important part of the total cost, repair and maintenance costs are also part of total cost. The third requirement for driving the equipment is simplicity, simple to be understood, simple to operate, maintaining and repair. There are many components which are used in this project to protect the refrigeration cycle.

5.3.1 Thermostat

A thermostat is a device for regulating the temperature of a system so that the system's temperature is maintained near a desired set point temperature. The thermostat does this by controlling the flow of heat energy into or out of the system. That is, the thermostat switches heating or cooling devices on or off as needed to maintain the correct temperature. Figure(5.1) shows thermostat. [14]

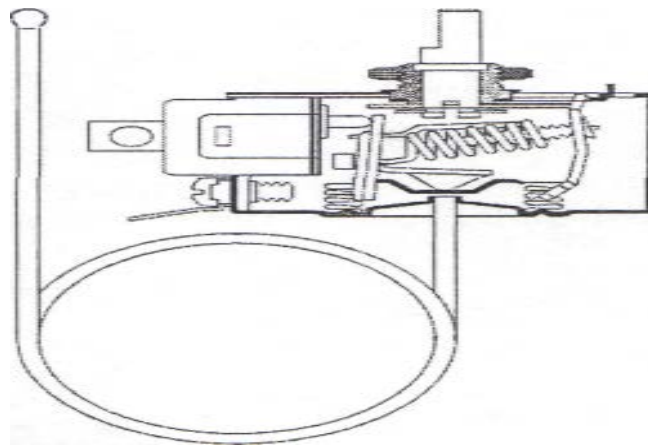


Figure 5.4 Thermostat.

5.3.2 Potential Relay

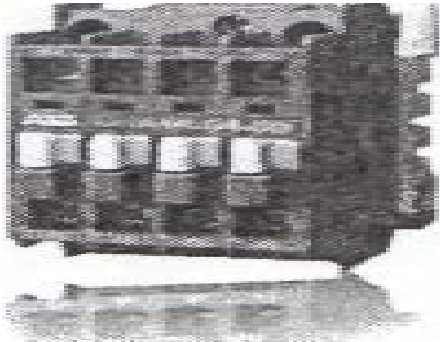
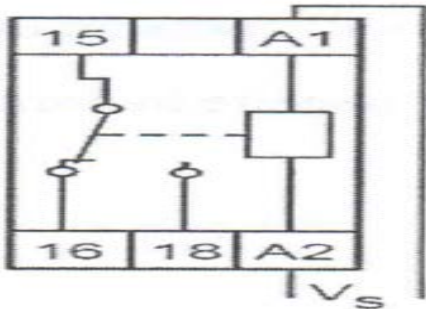
Potential relay, also known as voltage relay, is used on single phase motors that require large starting torque. During the off cycle, the contacts are closed to the start winding. This prevents arcing and possible burning of the contacts. When the thermostat contacts close, electricity is applied to the start and run winding in the motor. As the motor picks up speed, voltage is actually generated in the winding and causes a small amount of current to flow. As the motor approaches operating speed this counter electromotive force becomes strong enough to open the

start winding relay and power is no longer applied to the start winding. Power remains on the run winding and the motor continues to run. [14]

5.3.3 Contactor

Figure 5.3 shows the schematic diagram of contactor and contactor device, contactor is electric switch. It is similar to relay but it used at higher amperage. One of the contactor components is magnet, which is used to move switch open and close.

Motor starters are basically contactors with overload protection built in. This overload protection supplements the protection offered by breakers. Breakers protect the entire circuit, while the protection built into the stator protects a specific. [14]



(a)

(b)

Figure 5.5 (a) A schematic diagram of contactor, (b) Contactor device.

5.3.4 Overload

The most common cause of motor failure is overheating. The condition is created when a motor exceeds its normal operating current flow. The result can be either a breakdown of the motor

winding insulation and a short circuit, or a winding burn-out. For this reason overload protection is provided in the form of a current and temperature sensitive control which will open the circuit before any damage can occur. [14]

5.3.5 Optocoupler

Optocoupler is a component that transfers electrical signals between two isolated circuits by using light. Opto-isolators prevent high voltage from affecting the system receiving the signal.

5.3.6 The Arduino Uno

The Arduino Uno is a microcontroller board based on the ATmega328 (datasheet). It has 14 digital input/output pins (of which 6 can be used as PWM outputs), 6 analog inputs, a 16 MHz ceramic resonator, a USB connection, a power jack, an ICSP header, and a reset button. It contains everything needed to support the microcontroller; simply connect it to a computer with a USB cable or power it with a AC-to-DC adapter or battery to get started.

AM2302 capacitive humidity sensing digital temperature and humidity module is one that contains the compound has been calibrated digital signal output of the temperature and humidity sensors. Application of a dedicated digital modules collection technology and the temperature and humidity sensing technology, to ensure that the product has high reliability and excellent long-term stability. The sensor includes a capacitive sensor wet components and a high-precision temperature measurement devices, and connected with a high-performance 8-bit microcontroller. The product has excellent quality, fast response, strong anti-jamming capability, and high cost. Each sensor is extremely accurate humidity calibration chamber calibration. The form of

procedures, the calibration coefficients stored in the microcontroller, the sensor within the processing of the heartbeat to call these calibration coefficients. Standard single-bus interface, system integration quick and easy. Small size, low power consumption, signal transmission distance up to 20 meters. It is used in our device to detect humidity in the cabin.

5.4 Solar Panel

5.4.1 Solar Photovoltaic (PV) Power

Solar PV power is commercially available and Reliable technology with a significant potential for long-term growth in nearly all world regions. The estimated research showed that by 2050, PV will provide around 11% of global electricity production, and avoid 2.3 Giga-tones of CO₂ emissions per year. [14]

There is a pressing need to accelerate the development of advanced clean energy technologies in order to address the global challenges of energy security, climate change and sustainable development. [14]

Photovoltaic's (PV) comprises the technology to convert sunlight directly into electricity. The term "photo" means light and "voltaic" electricity. A photovoltaic (PV) cell, also known as "solar cell," is a semiconductor device that generates electricity when light falls on it. Although photovoltaic effect was observed in 1839 by the French scientist Edmund Becquerel, it was not fully comprehensible until the development of quantum theory of light and solid state physics in early to middle 1900s. Since its first commercial use in powering orbital satellites of the US space programs in the 1950s, PV has made significant progress with total U.S. photovoltaic module and cell shipments reaching \$131 million dollars in 1996. [14]

While most PV cells in use today are silicon-based, cells made of other semiconductor materials are expected to surpass silicon PV cells in performance and cost and become available competitors in the PV marketplace. [14]

5.4.2 Photovoltaic (PV) Analyses and Principal of Operation

When sunlight strikes a PV cell, the photons of the absorbed sunlight dislodge the electrons from the atoms of the cell. The free electrons then move through the cell, creating and filling in holes in the cell. This movement of electrons and holes that generate electricity. The physical process in which a PV cell converts sunlight into electricity is known as the " photovoltaic effect". One single PV cell produces up to 2 watts of power, too small even for powering pocket calculators or wristwatches. To increase power output, many PV cells are connected together to form modules.

Which are further assembled into larger units called arrays. This modular nature of PV enables designers to build PV systems with various power output for different types of applications. [14]

A complete PV system consists not only of PV modules, but also the "balance of system" or BOS - the support structures, wiring, storage, conversion devices, etc. i.e. everything else in a PV system except the PV modules. Two major types of PV systems are available in the marketplace today flat plate and concentrators. [14]

As the most prevalent type of PV systems, flat plate systems build the PV modules on a rigid and flat surface to capture sunlight. Concentrator systems use lenses to concentrate sunlight on the PV cells and increase the cell power output. Comparing the two systems, flat plate systems are typically less complicated but employ a larger number of cells while the concentrator systems use smaller areas of cells but require more sophisticated and expensive tracking systems. Unable to focus diffuse sunlight, concentrator systems do not work under cloudy conditions. [14]

5.4.3 Photovoltaic (PV) Technologies

Crystalline silicon: (c-Si) modules represent 85-90% of the global annual market today. C-Si modules are subdivided in two main categories:

- Single crystalline (sc-Si).
- Multi-crystalline (mc-Si).

Thin films: currently account for 10% to 15% of global PV module sales. They are subdivided into three main families:

- Amorphous (a-Si) and micro morph silicon (a-Si/ μ c-Si).
- Cadmium-Telluride (CdTe).
- Copper-Indium-Diselenide (CIS) and Copper-Indium-Gallium-Diselenide (CIGS).

Emerging technologies: encompass advanced thin films and organic cells. The latter are about to enter the market via niche applications.

Concentrator technologies (CPV): use an optical concentrator system which focuses solar radiation, onto a small high-efficiency cell. CPV technology is currently being tested in pilot applications.

Novel PV concepts: aim at achieving ultra-high efficiency solar cells via advanced materials and new conversion concepts and processes. They are currently the subject of basic research.

[14]

5.4.4 Types of Photovoltaic (PV) Cell Materials

PV cells are made of semiconductor materials. The major types of materials are crystalline and thin films, which vary from each other in terms of light absorption efficiency, energy conversion efficiency, manufacturing technology and cost of production. [14]

Table 5.7 Current Efficiencies of Different PV Technology Commercial Modules

Water-based c-Si		Thin Films		
Sc-Si	mc-Si	a-Si; a-Si/ μ c-Si	CdTe	CIS/CIGS
14-20%	13-15%	6-9%	9-11%	10-12%



Figure 5.6 Amorphous solar panel.

With good efficiency and availability in Palestine market amorphous PV cells are selected, with Efficiency 10-15%, Model VAF 1400 serial number 0000 10000551, made in France. 24 volt, 110 W, two panels, which shown in figure 5.3 below .

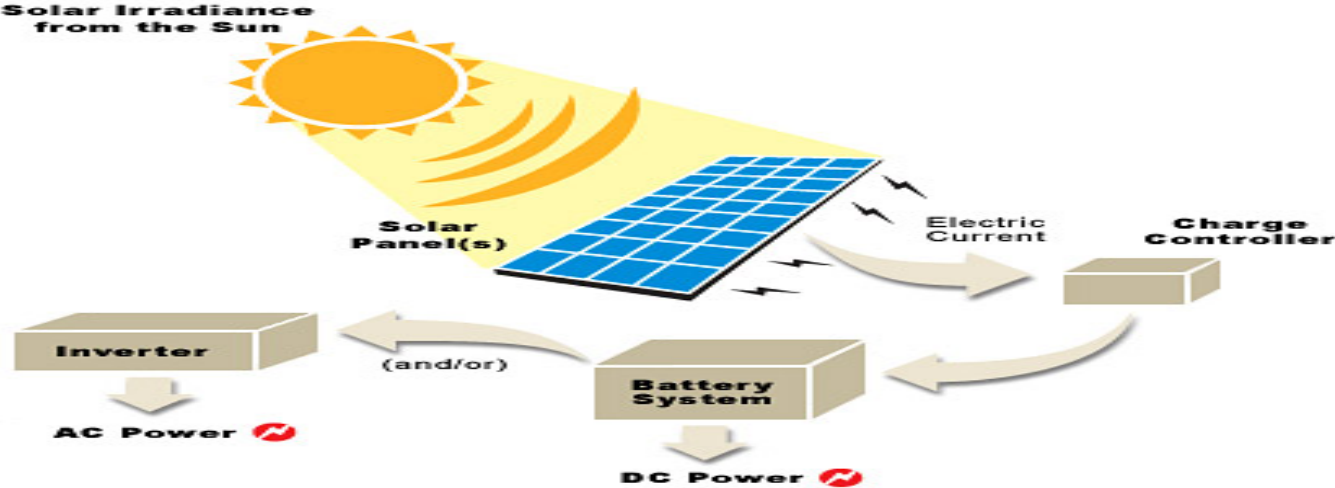


Figure 5.7 The conversion of solar energy to electrical energy.

5.4.5 Calculations of Solar Panel

Load of compressor is 258 w and calculate load of solar panel is :

- efficiency, η_{pv} of solar panel is 10-15%.
- efficiency, η_{ch} of charge controller 90-95%.
- efficiency, η_{inv} of inverter 85-88%.

$$E_{load} = (Q_{comp} + Q_f + Q_{fc} + Q_{fe}) * time\ operating \tag{5.1}$$

Applying equation (5.1):

$$E_{load} = (258 + 5 + 10 + 5) * 2\ h/day = 556\ W.h/day$$

The voltage of inverter (E_{inv}) can be determined by the following equation:

$$E_{inv} = \frac{E_{comp}}{\eta_{inv}} \quad (5.2)$$

$$E_{inv} = \frac{556}{0.85} = 322.5 \text{ W.h/day}$$

The Power of charge controller (E_{ch}) can be determined by the following equation:

$$E_{ch} = \frac{E_{inv}}{\eta_{ch}} \quad (5.3)$$

$$E_{ch} = \frac{322.5}{0.9} = 358 \text{ W.h/day}$$

The Power of solar panel (E_{pv}) can be determined by the following equation:

$$E_{pv} = \frac{E_{ch}}{\eta_{pv}} \quad (5.4)$$

$$E_{pv} = \frac{358}{0.1} = 3583 \text{ W.h/day}$$

To calculate the area of solar panel the following relation must be used:

$$1 \text{ m}^2 \text{ ---} \rightarrow 2750 \text{ W.h/day}$$

$$A \text{ ---} \rightarrow E_{pv}$$

Where:

2750 [W.h/day]: Annual rate of radiation.

A: area of solar panel [m^2].

$$1 \text{ m}^2 \text{ ---} \rightarrow 2750 \text{ W.h/day}$$

$$A \text{ ---} \rightarrow 3583 \text{ W.h/day}$$

$$A = \frac{3583}{2750} = 1.3 \text{ m}^2$$

5.4.6 PV Charge Controller

The main function of charge controller is to regulate the flow of electricity from the photovoltaic panels to the batteries. In PV systems with batteries, the batteries must be protected from overcharging and be maintained at fully charged state.

The PV Charge Controller uses the Micro-Processor and PWM (Pulse Width Modulation) to give optimal and safe charging. It makes varying On-Off pulses of electrical energy varying photovoltaic (PV) panel in charging the battery according to the battery state. It has 5 stages of charging as follows: [14]

A. BULK CHARGE:

At this mode a preset maximum constant amount of current (amps) is fed into the battery as the no PWM is present. As the battery is being charged up, the voltage of the battery increases gradually.

B. ABSORPTION CHARGE:

After the preset voltage is reached (approximately 4.3 volts for a 12 volt system) the voltage is then held constant. As the battery continues to be charged at constant voltage, the charging current decreases. The charging voltage is held at the Bulk Voltage Setting for one full hour with various rapid On-Off pulses (PWM). It then switches to Float Charge Mode. [14]

C. FLOAT CHARGE:

The controller will maintain the battery voltage at the float voltage setting by giving shorter On-pulse charge to make up for any detected self-discharge of the battery. When the battery voltage drops below the Float Voltage Setting for a total period of 10 minutes, a new charging cycle is activated in Bulk or Absorption Charge.

The three stages charging method works well with the chemical reaction that occurs as a battery is being charged. When a battery is more discharged, a regulated maximum current can be

applied, since there is a lot of material available for the reactions to occur. As the battery refills, less and less chemical material is available for the reaction. By using PWM to slowly reducing the charge current, while maintaining a preset high voltage, the battery is more closely refilled at the reaction rate of the chemicals. Finally, the Float voltage keeps the battery fully charged at all times taking care of the self-discharge. [14]

After electrical calculation and design. Solar charger Info24V, 10A ISSC-BASIC.100 VOLTRONICS, was selected.



Figure 5.8 PV Charge Controller.

5.4.7 Inverter

DC-AC inverters are electronic devices used to produce mains voltage AC Power from low voltage DC energy (from a battery or solar panel). This makes them very suitable for when you need to use AC power tools or application but the usual AC mains power is not available. Examples include operation application in caravans and mobile homes, and also running audio, video and computing equipment in remote areas.

As explained earlier, most DC-AC inverters deliver a modified sine wave. Output voltage, because they convert the incoming DC into AC by using MOSFET Transistor as electronic

switches. This gives very high conversion efficiency, but the alternating pulses output wave from is also relatively rich in harmonics.

The most common type of pure sine wave inverter operates by first converting the low voltage DC into high voltage DC, using high frequency DC-DC convertor. It then uses a high frequency POWM system to convert the high voltage DC into chopped AC, which is passed through an L-C low pass filter to produce the final clean 50 Hz sine wave output. This is like a high-voltage version of the single-bit digital to analog conversion used in many CD players.

After electronic calculation and design, inverter 240020, rating power 2400VA are selected.



Figure 5.9 PV inverter.

5.4.8 Storage Batteries

Batteries are needed to supply compressor and electrical elements (fan condenser, signal light) with suitable electric current in the absence of direct electricity from solar radiation.

Compressor power 258 W, power factor 0.85, voltage of batteries =12 V, supply needed time from batteries =2 hours, So the energy of inverter equal 322.5 *W. h/day*.

$$I = \frac{322.5}{12 * 0.85 * 0.8} = 39.5 \cong 40 \text{ A. h/day}$$

$$I = 40 * 2 = 80 \text{ A. h/day}$$

75 A.h/day

So, one (12V, 100A) battery type UBS 100A.h vision 6FM100-X are needed for this operating time (2hr).



Figure 5.10 PV storage batteries.

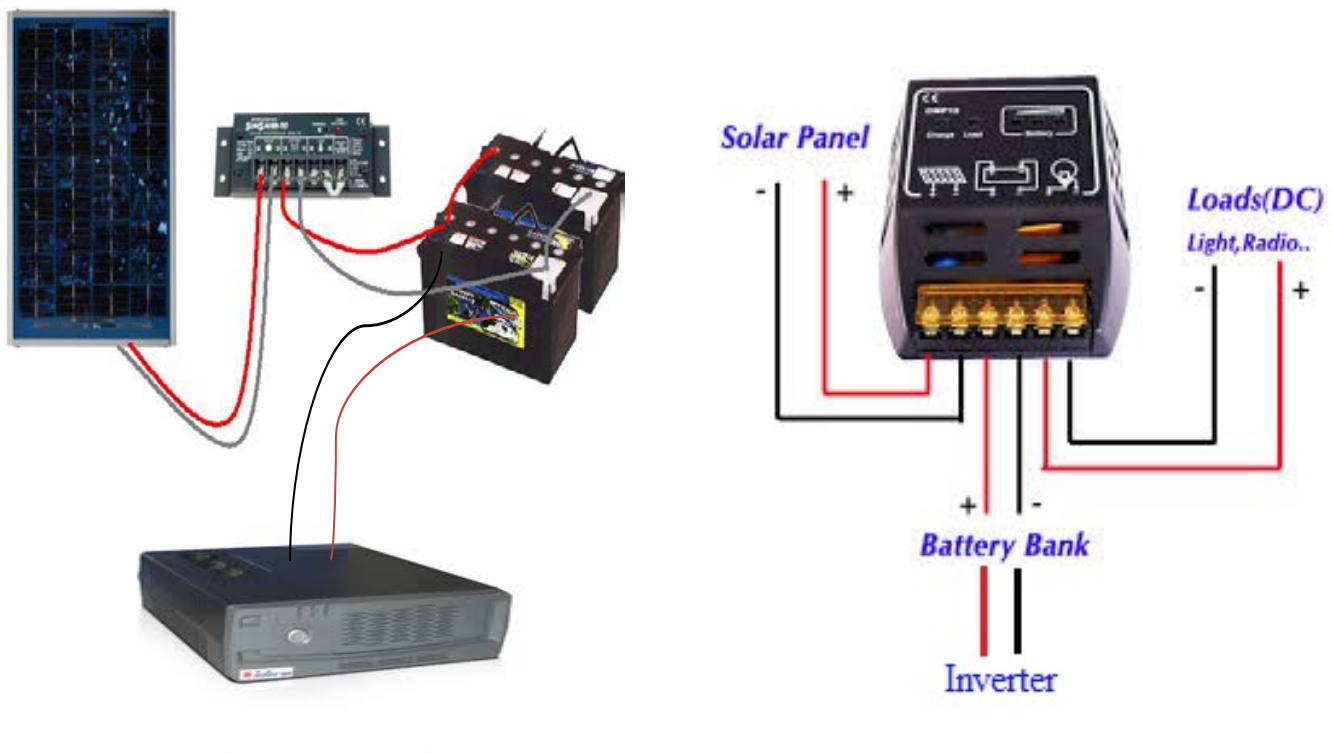


Figure 5.11 Connection of PV charge controller, inverter and battery.

CHAPTER SIX

DESIGN AND BUILDING OF THE DEVICE

CHAPTER SIX

DESIGN AND BUILDING OF THE DEVICE

6.1 Introduction

After all the calculations, the next step is the building of the device. This device was designed by using the solid work program.

6.2 Pictures of the Device

Here are some pictures of the device from all sides:

Figure (6.1) below, shows the structure of the device at the beginning of the building process. It consists of two bases; the lower base and the upper one. Its dimensions are as follows : height is 70 cm, length is 120 cm and width is 50 cm.



Figure 6.1 The structure of the device.

Figure (6.2) shows the lower side of the device; which contains some parts of the refrigeration cycle including the condenser , the compressor, the accumulator, the sight glass, the filter and the connecting pipes. Also this picture shows some of the electrical components which are the battery, the inverter and the charge controller. The right side of the picture shows the water tank where the harvested water is collected and stored.

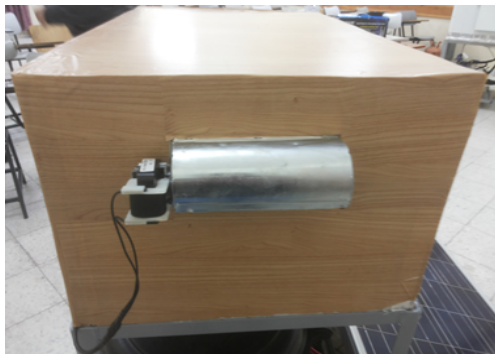


Figure 6.2 The lower part of the device.

Figure (6.3) contains three pictures (a), (b) and (c), it shows the upper part of the device from outside. Figure (6.3) (a) mainly shows of the cabin where air is cooled and water harvesting occurs. The thermostat is fixed on the cabin and behind the thermostat the arduino is fixed inside the white box, the right side of the picture shows a window at where air leaves the cabin. Figure (6.3) (b) shows the electric hood which enters air to the cabin, and Figure (6.3) (c) shows the control circuit.



(a)



(b)



(c)

Figure 6.3 The upper side of the device: (a) The cabin from outside (b) The hood (c) The control unit.

Figure (6.4) shows the cabin from inside at which air is cooled and water vapor is condensed. The figure shows the evaporator, the humidity sensor, the evaporators fan, the expansion valve, the sink and the connecting pipes.

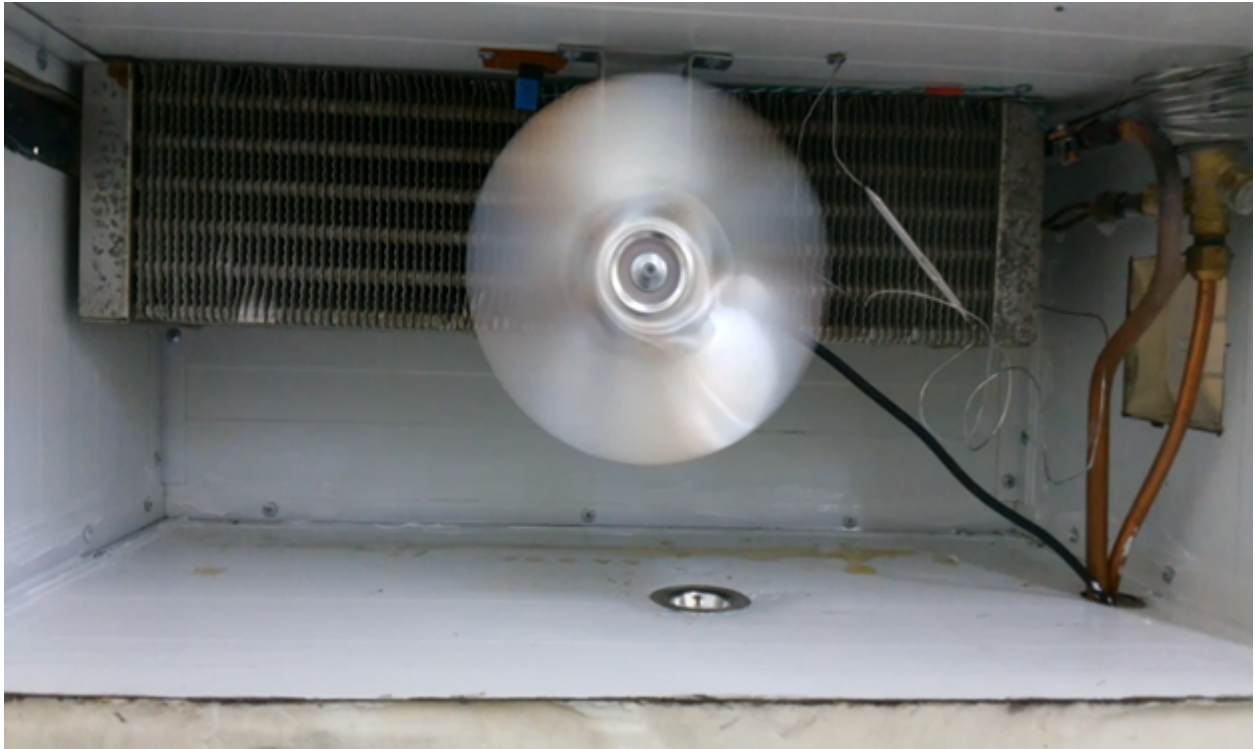


Figure 6.4 The cabin from inside.

Figure (6.5) shows the overview of the device; where the mechanical and electrical parts are obvious. The solar panel is on the left side of the figure.



Figure 6.5 The overview of the device.

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15- http://www.phg.org/data/files/publications/general_reports/Reports/Older/passive_humidity_harvest_03_05.pdf

16- Cool-pack software.

17- Psychrometric software.

APPENDIX A

Table A-1 Saturated Water – Temperature

Temp., T °C	Sat. press., P_{sat} kPa	Specific volume, m^3/kg		Internal energy, kJ/kg			Enthalpy, kJ/kg			Entropy, $kJ/kg \cdot K$		
		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
0.01	0.6117	0.001000	206.00	0.000	2374.9	2374.9	0.001	2500.9	2500.9	0.0000	9.1556	9.1556
5	0.8725	0.001000	147.03	21.019	2360.8	2381.8	21.020	2489.1	2510.1	0.0763	8.9487	9.0249
10	1.2281	0.001000	106.32	42.020	2346.6	2388.7	42.022	2477.2	2519.2	0.1511	8.7488	8.8999
15	1.7057	0.001001	77.885	62.980	2332.5	2395.5	62.982	2465.4	2528.3	0.2245	8.5559	8.7803
20	2.3392	0.001002	57.762	83.913	2318.4	2402.3	83.915	2453.5	2537.4	0.2965	8.3696	8.6661
25	3.1698	0.001003	43.340	104.83	2304.3	2409.1	104.83	2441.7	2546.5	0.3672	8.1895	8.5567
30	4.2469	0.001004	32.879	125.73	2290.2	2415.9	125.74	2429.8	2555.6	0.4368	8.0152	8.4520
35	5.6291	0.001006	25.205	146.63	2276.0	2422.7	146.64	2417.9	2564.6	0.5051	7.8466	8.3517
40	7.3851	0.001008	19.515	167.53	2261.9	2429.4	167.53	2406.0	2573.5	0.5724	7.6832	8.2556
45	9.5953	0.001010	15.251	188.43	2247.7	2436.1	188.44	2394.0	2582.4	0.6386	7.5247	8.1633
50	12.352	0.001012	12.026	209.33	2233.4	2442.7	209.34	2382.0	2591.3	0.7038	7.3710	8.0748
55	15.763	0.001015	9.5639	230.24	2219.1	2449.3	230.26	2369.8	2600.1	0.7680	7.2218	7.9898
60	19.947	0.001017	7.6670	251.16	2204.7	2455.9	251.18	2357.7	2608.8	0.8313	7.0769	7.9082
65	25.043	0.001020	6.1935	272.09	2190.3	2462.4	272.12	2345.4	2617.5	0.8937	6.9360	7.8296
70	31.202	0.001023	5.0396	293.04	2175.8	2468.9	293.07	2333.0	2626.1	0.9551	6.7989	7.7540
75	38.597	0.001026	4.1291	313.99	2161.3	2475.3	314.03	2320.6	2634.6	1.0158	6.6655	7.6812
80	47.416	0.001029	3.4053	334.97	2146.6	2481.6	335.02	2308.0	2643.0	1.0756	6.5355	7.6111
85	57.868	0.001032	2.8261	355.96	2131.9	2487.8	356.02	2295.3	2651.4	1.1346	6.4089	7.5435
90	70.183	0.001036	2.3593	376.97	2117.0	2494.0	377.04	2282.5	2659.6	1.1929	6.2853	7.4782
95	84.609	0.001040	1.9808	398.00	2102.0	2500.1	398.09	2269.6	2667.6	1.2504	6.1647	7.4151
100	101.42	0.001043	1.6720	419.06	2087.0	2506.0	419.17	2256.4	2675.6	1.3072	6.0470	7.3542
105	120.90	0.001047	1.4186	440.15	2071.8	2511.9	440.28	2243.1	2683.4	1.3634	5.9319	7.2952
110	143.38	0.001052	1.2094	461.27	2056.4	2517.7	461.42	2229.7	2691.1	1.4188	5.8193	7.2382
115	169.18	0.001056	1.0360	482.42	2040.9	2523.3	482.59	2216.0	2698.6	1.4737	5.7092	7.1829
120	198.67	0.001060	0.89133	503.60	2025.3	2528.9	503.81	2202.1	2706.0	1.5279	5.6013	7.1292
125	232.23	0.001065	0.77012	524.83	2009.5	2534.3	525.07	2188.1	2713.1	1.5816	5.4956	7.0771
130	270.28	0.001070	0.66808	546.10	1993.4	2539.5	546.38	2173.7	2720.1	1.6346	5.3919	7.0265
135	313.22	0.001075	0.58179	567.41	1977.3	2544.7	567.75	2159.1	2726.9	1.6872	5.2901	6.9773
140	361.53	0.001080	0.50850	588.77	1960.9	2549.6	589.16	2144.3	2733.5	1.7392	5.1901	6.9294
145	415.68	0.001085	0.44600	610.19	1944.2	2554.4	610.64	2129.2	2739.8	1.7908	5.0919	6.8827
150	476.16	0.001091	0.39248	631.66	1927.4	2559.1	632.18	2113.8	2745.9	1.8418	4.9953	6.8371
155	543.49	0.001096	0.34648	653.19	1910.3	2563.5	653.79	2098.0	2751.8	1.8924	4.9002	6.7927
160	618.23	0.001102	0.30680	674.79	1893.0	2567.8	675.47	2082.0	2757.5	1.9426	4.8066	6.7492
165	700.93	0.001108	0.27244	696.46	1875.4	2571.9	697.24	2065.6	2762.8	1.9923	4.7143	6.7067
170	792.18	0.001114	0.24260	718.20	1857.5	2575.7	719.08	2048.8	2767.9	2.0417	4.6233	6.6650
175	892.60	0.001121	0.21659	740.02	1839.4	2579.4	741.02	2031.7	2772.7	2.0906	4.5335	6.6242
180	1002.8	0.001127	0.19384	761.92	1820.9	2582.8	763.05	2014.2	2777.2	2.1392	4.4448	6.5841
185	1123.5	0.001134	0.17390	783.91	1802.1	2586.0	785.19	1996.2	2781.4	2.1875	4.3572	6.5447
190	1255.2	0.001141	0.15636	806.00	1783.0	2589.0	807.43	1977.9	2785.3	2.2355	4.2705	6.5059

Table A-2 Properties of Air Atmospheric Pressure

The values of μ , k , c_p , and Pr are not strongly pressure-dependent and may be used over a fairly wide range of pressures							
T, K	ρ kg/m ³	c_p kJ/kg·°C	$\mu \times 10^5$ kg/m·s	$\nu \times 10^6$ m ² /s	k W/m·°C	$\alpha \times 10^4$ m ² /s	Pr
100	3.6010	1.0266	0.6924	1.923	0.009246	0.02501	0.770
150	2.3675	1.0099	1.0283	4.343	0.013735	0.05745	0.753
200	1.7684	1.0061	1.3289	7.490	0.01809	0.10165	0.739
250	1.4128	1.0053	1.5990	11.31	0.02227	0.15675	0.722
300	1.1774	1.0057	1.8462	15.69	0.02624	0.22160	0.708
350	0.9980	1.0090	2.075	20.76	0.03003	0.2983	0.697
400	0.8826	1.0140	2.286	25.90	0.03365	0.3760	0.689
450	0.7833	1.0207	2.484	31.71	0.03707	0.4222	0.683
500	0.7048	1.0295	2.671	37.90	0.04038	0.5564	0.680
550	0.6423	1.0392	2.848	44.34	0.04360	0.6532	0.680
600	0.5879	1.0551	3.018	51.34	0.04659	0.7512	0.680
650	0.5430	1.0635	3.177	58.51	0.04953	0.8578	0.682
700	0.5030	1.0752	3.332	66.25	0.05230	0.9672	0.684
750	0.4709	1.0856	3.481	73.91	0.05509	1.0774	0.686
800	0.4405	1.0978	3.625	82.29	0.05779	1.1951	0.689
850	0.4149	1.1095	3.765	90.75	0.06028	1.3097	0.692
900	0.3925	1.1212	3.899	99.3	0.06279	1.4271	0.696
950	0.3716	1.1321	4.023	108.2	0.06525	1.5510	0.699
1000	0.3524	1.1417	4.152	117.8	0.06752	1.6779	0.702
1100	0.3204	1.160	4.44	138.6	0.0732	1.969	0.704
1200	0.2947	1.179	4.69	159.1	0.0782	2.251	0.707
1300	0.2707	1.197	4.93	182.1	0.0837	2.583	0.705
1400	0.2515	1.214	5.17	205.5	0.0891	2.920	0.705
1500	0.2355	1.230	5.40	229.1	0.0946	3.262	0.705
1600	0.2211	1.248	5.63	254.5	0.100	3.609	0.705
1700	0.2082	1.267	5.85	280.5	0.105	3.977	0.705
1800	0.1970	1.287	6.07	308.1	0.111	4.379	0.704
1900	0.1858	1.309	6.29	338.5	0.117	4.811	0.704
2000	0.1762	1.338	6.50	369.0	0.124	5.260	0.702
2100	0.1682	1.372	6.72	399.6	0.131	5.715	0.700
2200	0.1602	1.419	6.93	432.6	0.139	6.120	0.707
2300	0.1538	1.482	7.14	464.0	0.149	6.540	0.710
2400	0.1458	1.574	7.35	504.0	0.161	7.020	0.718
2500	0.1394	1.688	7.57	543.5	0.175	7.441	0.730

Table A-3 Copper Tubes: Types, Standards, Applications, Lengths.

Tube Type	Color Code	Standard	Application ¹	Commercially Available Lengths ²		
				Nominal or Standard Sizes	Drawn	Annealed
TYPE K	Green	ASTM B 88 ³	Domestic Water Service and Distribution, Fire Protection, Solar, Fuel/Fuel Oil, HVAC, Snow Melting, Compressed Air, Natural Gas, Liquefied Petroleum (LP) Gas, Vacuum	STRAIGHT LENGTHS:		
				1/4-inch to 8-inch	20 ft	20 ft
				10-inch	18 ft	18 ft
				12-inch	12 ft	12 ft
				COILS:		
				1/4-inch to 1-inch	—	60 ft
				1 1/4 inch and 1 1/2-inch	—	100 ft
TYPE L	Blue	ASTM B 88	Domestic Water Service and Distribution, Fire Protection, Solar, Fuel/Fuel Oil, HVAC, Natural Gas, Liquefied Petroleum (LP) Gas, Snow Melting, Compressed Air, Vacuum	STRAIGHT LENGTHS:		
				1/4-inch to 10-inch	20 ft	20 ft
				12-inch	18 ft	18 ft
				COILS:		
				1/4-inch to 1-inch	—	60 ft
				1 1/4 inch and 1 1/2-inch	—	100 ft
				2-inch	—	60 ft
TYPE M	Red	ASTM B 88	Domestic Water Service and Distribution, Fire Protection, Solar, Fuel/Fuel Oil, HVAC, Snow Melting, Vacuum	STRAIGHT LENGTHS:		
				1/4-inch to 12-inch	20 ft	N/A
DWV	Yellow	ASTM B 306	Drain, Waste, Vent, HVAC, Solar	STRAIGHT LENGTHS:		
				1 1/4-inch to 8-inch	20 ft	N/A
ACR	Blue	ASTM B 280	Air Conditioning, Refrigeration, Natural Gas, Liquefied Petroleum (LP) Gas, Compressed Air	STRAIGHT LENGTHS:		
				3/8-inch to 4 1/8-inch	20 ft	4
				COILS:		
OXY, MED, OXY/MED, OXY/ACR, ACR/MED	(K)Green (L)Blue	ASTM B 819	Medical Gas Compressed Medical Air, Vacuum	STRAIGHT LENGTHS:		
				1/4-inch to 8-inch	20 ft	N/A

Table A-4 Dimensions and Physical Characteristics of Copper Tube ACR

(Air -Conditioning and Refrigeration Fields Service).

(A: Annealed Temper, D: Drawn Temper).

Nominal or Standard Size, inches		Nominal Dimensions, inches			Calculated Values (based on nominal dimensions)				
		Outside Diameter	Inside Diameter	Wall Thickness	Cross Sectional Area of Bore, sq inches	External Surface, sq ft per linear ft	Internal Surface, sq ft per linear ft	Weight of Tube Only, pounds per linear ft	Contents of Tube, cu ft per linear ft
1/8	A	.125	.065	.030	.00332	.0327	.0170	.0347	.00002
3/16	A	.187	.128	.030	.0129	.0492	.0335	.0575	.00009
1/4	A	.250	.190	.030	.0284	.0655	.0497	.0804	.00020
5/16	A	.312	.248	.032	.0483	.0817	.0649	.109	.00034
3/8	A	.375	.311	.032	.076	.0982	.0814	.134	.00053
	D	.375	.315	.030	.078	.0982	.0821	.126	.00054
1/2	A	.500	.436	.032	.149	.131	.114	.182	.00103
	D	.500	.430	.035	.145	.131	.113	.198	.00101
5/8	A	.625	.555	.035	.242	.164	.145	.251	.00168
	D	.625	.545	.040	.233	.164	.143	.285	.00162
3/4	A	.750	.680	.035	.363	.196	.178	.305	.00252
	A	.750	.666	.042	.348	.196	.174	.362	.00242
	D	.750	.666	.042	.348	.196	.174	.362	.00242
7/8	A	.875	.785	.045	.484	.229	.206	.455	.00336
	D	.875	.785	.045	.484	.229	.206	.455	.00336
1 1/8	A	1.125	1.025	.050	.825	.294	.268	.655	.00573
	D	1.125	1.025	.050	.825	.294	.268	.655	.00573
1 3/8	A	1.375	1.265	.055	1.26	.360	.331	.884	.00875
	D	1.375	1.265	.055	1.26	.360	.331	.884	.00875
1 5/8	A	1.625	1.505	.060	1.78	.425	.394	1.14	.0124
	D	1.625	1.505	.060	1.78	.425	.394	1.14	.0124
2 1/8	D	2.125	1.985	.070	3.09	.556	.520	1.75	.0215
2 5/8	D	2.625	2.465	.080	4.77	.687	.645	2.48	.0331
3 1/8	D	3.125	2.945	.090	6.81	.818	.771	3.33	.0473
3 5/8	D	3.625	3.425	.100	9.21	.949	.897	4.29	.0640
4 1/8	D	4.125	3.905	.110	12.0	1.08	1.02	5.38	.0833

Table A-5 Modified Correlation of Grimson for Heat Transfer in Tube Banks of 10 Rows or More

S _T /d								
S _L /d	1.25		1.5		2.0		3.0	
	C	N	C	N	C	N	C	N
In line								
1.25	0.386	0.592	0.305	0.608	0.111	0.704	0.0703	0.752
1.5	0.407	0.586	0.278	0.620	0.112	0.702	0.0753	0.744
2.0	0.464	0.570	0.332	0.602	0.254	0.632	0.220	0.648
3.0	0.322	0.601	0.396	0.584	0.415	0.581	0.371	0.608
Staggered								
0.6	–	–	–	–	–	–	0.236	0.636
0.9	–	–	–	–	0.495	0.571	0.445	0.581
1.0	–	–	0.552	0.558	–	–	–	–
1.125	–	–	–	–	0.531	0.565	0.575	0.560
1.25	0.575	0.556	0.561	0.554	0.576	0.556	0.579	0.562
1.5	0.501	0.568	0.511	0.562	0.502	0.568	0.242	0.568
2.0	0.448	0.572	0.462	0.568	0.535	0.556	0.498	0.570
3.0	0.344	0.592	0.395	0.580	0.488	0.562	0.467	0.574

Table A-6 Ratio of h for Rows Deep to that for 10 Rows Deep

N	1	2	3	4	5	6	7	8	9	10
Ratio for staggered tubes	0.68	0.75	0.83	0.89	0.92	0.95	0.97	0.98	0.99	1.0
Ratio for in-line tubes	0.64	0.80	0.87	0.90	0.92	0.94	0.96	0.98	0.99	1.0

Table A-7 Constant.

Re_{df}	C	n
0.4–4	0.989	0.330
4–40	0.911	0.385
40–4000	0.683	0.466
4000–40,000	0.193	0.618
40,000–400,000	0.0266	0.805

Table A-8 Recommended Refrigerant Velocities for R134a.

Lines	Refrigerant Velocity [$m\ s^{-1}$]
Suction lines	4.5 – 20
Discharge lines	10 – 18
Liquid lines	< 1.5

CHAPTER SEVEN

RESULTS

CHAPTER SEVEN

RESULTS

7.1 Introduction

After the device was built, it must be tested. The device was tested in several days and at different conditions and it gave the expected results.

7.2 Results Obtained

In table (7.1) five results were chosen from five separate days at different temperature and humidity conditions in winter. Results were satisfying and as it was expected. The fifth result in table (7.1) was taken on the 17th of January at which the relative humidity was 80% and temperature was 6 °C, thus the dew point temperature was -0.1 °C and as it was expected that the device will not harvest water because the dew point is lower than the temperature of the condenser.

Table 7.1 Testing and Results

Date	RH%	Temp(°C)	Dew Point(°C)	Amount (L/hr)
07/12/2014	64	22	14.9	1.5
08/12/2014	71	21	17.6	1.5
13/12/2014	74	19	14.7	1.25
03/01/2015	78	16	8.3	0.8
17/01/2015	80	6	-0.1	-

