**Mechanical Engineering Department**

==**Bachelor Thesis**==

**Cooling And Air Conditioning**


**PROGECT**

# Design And Implementation Of A Refrigerator Utilizing Solar Energy

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**Hebron – Palestine**
**Jun, 2012**

جامعة بوليتكنيك فلسطين

الخليل – فلسطين

دائرة الهندسه الميكانيكية


اسم المشروع

# Design And Implementation Of A refrigerator Utilizing Solar Energy

أسماء الطلبة


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بناءً على نظام كلية الهندسة والتكنولوجيا وإشراف ومتابعة المشرف المباشر على المشروع وموافقة أعضاء اللجنة الممتحنة تم تقديم هذا المشروع إلى دائرة الهندسة الميكانيكية، وذلك للوفاء بمتطلبات درجة البكالوريوس في الهندسة تخصص هندسة التكييف والتبريد.

توقيع المشرف

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توقيع اللجنة الممتحنة

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توقيع رئيس الدائرة

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**Dedication …**

we gift graduation project

to who carry candle of science

to light his avenue

of live

To Dr. Ishaq sider & To all students

& Who wish to look for

The future

To our parents and families

To our friends

to who love the knowledge

# Acknowledgment (شكر والتقدير)

# Abstract

This project aims to design and build a refrigerator which will be used in remote areas. The refrigerator volume and temperature will be estimated for small family food storage, and it consist of one suitable chamber which can serve as a cooler or freezer chamber, and this is related to user needed. The volume of a refrigerator is estimated about (252 L), and the temperature range of a refrigerated space between (-15 to 5) $C^0$.

The refrigerated cycle will be driven by utilizing solar energy (direct energy conversion system 'PV').

The design and selection of cycle component such as : compressor , evaporator , condenser, capillary tube and other component will be done. The working fluid will be selected as a result of comparisons among refrigerants on the basis of many factors i.e. efficiency, environment effect, and cost.

The system will be driven by (PV) solar panel, or from electrical grid. The storage of cold will be done by storing it in a secondary refrigerant. Special Batteries will be used for storing electrical power.

The project will deal with traditional control system (contactors, Relays …etc) and it let enough chance for other project to improve its control system like use modern programmable logic controlling (PLC) system. Also the system may be used as educational unit in laboratories.

The system will be test, and to make measure of a refrigeration capacity ,temperature limits, pressure gages, to make comparisons between theoretical and actual results at the end.


يهدف هذا المشروع إلى تصميم وبناء ثلاجة لخدمة المناطق النائية والبعيدة عن خطوط شبكة الكهرباء العامة،تم تقدير وحساب حجم الثلاجة بحيث تفي باحتياجات أسرة صغيرة الحجم حيث ان حجم الثلاجة من الداخل ( 252 لتر)، تم اختياره بناء على حجم المنتج المراد تخزينه بداخلها.


تتكون الثلاجة من غرفة تبريد واحدة تناسب حجم المنتج المراد تخزينه ، ويمكن ان تستخدم كغرفة للتفريز او التبريد ويتم التحكم في ذلك حسب رغبة وحاجة المستخدم، بحيث تعمل ضمن مدى درجات حرارة (5_15−)درجة مئوية، و يتم تشغيلها باستخدام الطاقة الشمسية ( ألواح الخلايا الشمسية) ، أو مباشرة عن طريق خطوط الشبكة العامة،مع إمكانية تخزين الكهرباء في بطاريات تخزين خاصة. بالإضافة إلى إمكانية تخزين البرودة في مائع تبريد ثانوي للحفاظ على التبريد دون انقطاع و يتم اختياره بناء على عوامل متعددة يتم عملها من خلال مقارنة شاملة بين الأنواع المختلفة لتلك الموائع على أسس متعددة مثل الكفاءة ، أثاره على البيئة والتكلفة ،وبنفس تلك الطريقة يتم اختيار مائع التبريد الأساسي للدورة.
سيتم تصميم و اختيار الأجزاء الميكانيكية والكهربائية للدورة: كالضاغط ،المكثف،المبخر، الأنبوبة الشعرية،الخلايا الشمسية البطاريات وغيرها بناء على حسابات وبرامج خاصة .


كذلك سيتم إجراء التحكم المطلوب في الدورة باستخدام وسائل التحكم المعروفة مثل ( Thermostat, relay contactors...etc)، مع إتاحة الفرصة لاستخدام التحكم المنطقي المبرمج(PLC) في المشاريع القادمة بإذن الله .
سيتم بناء النظام واختباره لعمل مقارنه بين القيم النظرية والقيم الحقيقية .

# Table Of Contents

# List of Tables

| Chapter One | | |
|---|---|---|
| **Table No** | **Title** | **Page** |
| Table(1.1) | Main refrigeration cycle comparison | 4 |
| Table(1.2) | Actual budget table for the project | 5 |
| Table(1.3) | The first semester time plan | 6 |
| Table(1.4) | The second semester time plan | 7 |
| **Chapter Two** | | |
| **Table No** | **Title** | **Page** |
| Table(2.1) | Chamber load | 16 |
| **Chapter Three** | | |
| **Table No** | **Title** | **Page** |
| Table (3.1) | Physical properties of refrigerants | 18 |
| Table (3.2) | comparison between major secondary refrigerant | 20 |
| Table (3.3) | Properties of each point on the cycle | 22 |
| Table( 3.4) | Capillary tube Selection | 36 |
| **Chapter Four** | | |
| **Table No** | **Title** | **Page** |
| Table (4.1) | Table 4.1: Current efficiencies of different PV technology commercial modules | 43 |

# List of Figures

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13 Autocad software

14 DanCap software

**APPENDEX A**

TABLE A-1  maximum and minimum temperature for Hebron city

TABLE A-2   Thermal Conductivity of Materials

TABLE A-3  Properties of common foods

TABLE A-4  Air change per hour

TABLE A-5   Specific heat of packaging material

TABLE A-6  Recommended  Refrigerant Velocities

TABLE A-7   Dimensions and Physical Characteristics of Copper Tube ACR (Air -Conditioning and Refrigeration Fields Service).

TABLE A-8   modified correlation of grim son for heat transfer in tube banks

TABLE A-9   Properties of air at atmospheric pressure

TABLE A-10 ratio of  N rows deep to that for 10rows deep

TABLE A -11  Respiration

# CHAPTER ONE

# INTRODUCTION

# CHAPTER ONE

## Introduction

With scientific and technological development in the field of solar cells, now it be in different applications ,like cooling science  to reach the desired goals , and thereby cooling science develops from better  to best and achieve its goals.

There is a significant problem followed by families who live in remote areas, especially those not reached by the electricity grid in order to preserve their food and drink, So the need call us to  provide a refrigerators operate in the absent of  national electricity grid .

Great dangerous outputs of non-renewable energy which pollute the environment i.e. global warming potential (GWP),and ozone depletion layer (ODP) call us to search for an alternatives  and clean sources of  renewable energy sources ,like solar energy.

### 1.1 Scope of the project

Solar refrigerators are manufacture to serve people who suffer from absent of national electricity grid lines or continuous power cut off   especially for those live  in Gaza  and remote or  'Bedouin 'areas , like some of  the  south areas  people  in Hebron city .This project also has an academic benefit  for (HVAC) students to develop their skills  and ability of combination between practical and theoretical sides.

The refrigerated cycle will be driven utilizing solar energy (direct energy conversion system 'PV'), which is one of the most important source of renewable energy, so it has an environment and economic benefits   referred to community.

The refrigerator volume and temperature will be estimated for small family  food storage .So the volume of the refrigerator was estimated  about (252 L),and the temperature  range of  a refrigerated space  between (-15 to 5)$C^0$ .

### 1.2 About solar-powered refrigerators

After conducting the studies and statistics which keep up with scientific and technological developments in a refrigeration for a promising this research provides a scientific field benefit  in a refrigeration and cooling field using solar cells energy, often known as **photovoltaic's (PV) cells**  , which common use  these days. Some of us use it to power our homes, cell-phone chargers, radios, International Space Station (ISS), and other applications. These technologies use photovoltaic (PV) cells to convert sunlight into

electricity. When sunlight strikes a PV cell, it heats the cell up. Heating the cell causes electrons to knock loose, and these electrons are converted into a stream of electricity, or current. [1]

There are several previous studies and researches were prepared in this field. Some of there will be display in this project, as follow here :

One of the most common solar-powered refrigerators on the market, (NASA) refrigerators, which uses (PV) technology to power there refrigerators in there space travels.

A team of German scientists developed a new technique that make a refrigerators operate by solar-power technology instead of electricity grid lines ; which will bring great benefits to the hot countries ,also in German there is A team of young scientists - who are working with ( Fasenbosc Geladbek inventions center) in a (University of Applied Sciences) in the area (Gelsankrshen) - «Such technology would have tremendous benefits for the hot countries , where are refrigerators and air conditioners, there is vital »also they have developed a prototype will be used soon in a cold storage plant in (Morocco) using solar power technique.[2]

Two Emiratian engineers ; recently graduated from the "Faculty of Technology" in Electrical Engineering Department in ' Abu Dhabi ', United Arab Emirates, enable to accomplish the creation of "solar-powered refrigerator," have a lot of features and uses . It should be noted that the refrigerator invented by engineers' 'Abdullah and Hamad ,it powered by 12 volts and 46 watts, it can be available enough cooling effect for food and medicine storage along 8 continuous hours at10 ° C, and able to turn the cooling system in which to heat what we want done things frozen to be used at a temperature less than it is, note to mention the refrigerator light and easily transported and operated.[3]

In **Gaza** The Ministry of Health has provided several solar-powered refrigerators to save the vaccinations, with the support of "UNICEF" to improve the cold chain for vaccines, to reducing the problem of continuous power cuts off especially in last ware. These refrigerators solved the problem of a large drop in the constant supplied power , pointing that one sunny day equivalent to three days without sun for the work of the refrigerator without any defect in the cooling quality ,also its operating on 12 volt only . [4]

## 1.3 Why vapor compression refrigeration cycle?

There are different types of refrigeration and cooling cycles were discovered and build , such as vapor jet refrigeration cycle , absorption refrigeration cycle, vapor compression refrigeration cycle, and others .Which are considered the main types in the field of cooling

science , (1.1) compare between those three major types mentioned above , in order to select the most appropriate type to be the subject of study , research ,and development in this project

Table (1.1)   Main refrigeration cycle comparison [5]

| Comparison face | Vapor compression refrigerator | Absorption refrigerator | Vapor jet refrigerator |
|---|---|---|---|
| Initial cost | low | high | High |
| Maintenance | continuous | rarely | rarely |
| life | Long life | Short life (corrosive in pipes) | intermediate |
| COP | High nearly (4-5) | Very low (1.1 for $LiBr_2$) | low |
| Heat rejection element | condenser | Condenser ,Analyzer ,Absorber. rectifier. | Condenser, Steam boiler |
| Heat rejection amount | Low(heat rejection factor 1.2) | high(heat rejection factor 2.5) | High( heat rejection 2.7) |
| Refrigerant | Expensive; ie (HCHs ,HCFCs, CFCs) | Cheap and available $(NH_3 ,LiBr_2 ,$ | Cheap and available (water in general) |
| Green house effect | Halocarbon (HCFC)refrigerant produce depletion in Ozone  layer | No  green hose effect | No  green hose effect |

Through the table (1.1), it is difficult to identify which  is  the best type, because   there's many  factors  play rule and  and control . Despite the existence of several problems in the vapor compression system like economical, technical, and environmental problems, it will be considered the best and most appropriate one on the basis of comparison  shown above, by depending on  the two most important factors, like efficiency ,and coefficient of performance (COP) of  cycle . So vapor compression refrigeration cycle is considered as a subject of study in this project.

The   use of solar energy   be a solution of   economic problem   (i.e. operation cost reduction  ). The use of primary refrigerant (R_134a) be   a solution   of environmental problem (environmental friendly), and the use of secondary refrigerant be a solution of the technical problem by keeping cooling effect in the time of compressor shut off by reradiating of cold all time especially at night .

## 1.4 Backgrounds about vapor compression system powered utilizing solar cells energy

These systems like any traditional vapor compression system which compression element (compressor) ,condensing and evaporating units, and expansion devices(capillary tube), in addition there as a solar cell as a power source to run compressor and condenser fan .Also a Special batteries are use to keep necessary operation cooling cycle for a desired needed time .In shortage these component are integrated together in their work to produce desired goals and keep a temperature for cooling space .

## 1.5 The cycle component selection

In next chapters it will be design and select cycle component based on several calculation , designs using catalogs ,and software programs , ( Dan-cap ,solar cell program , cool back,…etc).

## 1.6 The budget for the project

Table (1.2) Actual budget table for the project

| Task | # | COST (NIS) | Total |
|------|---|------------|-------|
| Compressor | 1 | 450 | 450 |
| Condenser | 1 | 200 | 200 |
| Connecting pipes | 8 m | 1m=40 | 320 |
| Capillary tube | 1 | 25 | 25 |
| Refrigerant R134a | 1 | 200 | 200 |
| Battery | 2 | 650 | 1300 |
| Solar cells | 2 | 1000 | 2000 |
| Building chambers | - | 950 | 950 |
| Regulator(10A) | 1 | 500 | 500 |
| Thermostat | 1 | 40 | 40 |
| Inverter(1440W) | 1 | 1500 | 1500 |
| Filter | 1 | 50 | 50 |
| Sight glass | 1 | 30 | 30 |
| Total | | | 7565NIS |

**1.6 The time planning for the project**

The project plan follows the following time schedule which includes the related tasks of study and system analysis. The following time plans are for the first and second semester

Table (1.3) First semester time plan

| Task/Week | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Choosing project title | ▓ | ▓ | ▓ | ▓ | | | | | | | | | | | | |
| Collecting information | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | | | | | |
| Reading | | | | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | | | | |
| Introduction | | | | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | | | | | | | |
| Load calculations | | | | | ▓ | ▓ | ▓ | ▓ | ▓ | | | | | | | |
| Cycle analysis | | | | | | | ▓ | ▓ | | ▓ | ▓ | | | | | |
| Refrigerant choose | | | | | | | | | ▓ | ▓ | | | | | | |
| Compressor Design and selection | | | | | | | | | ▓ | ▓ | ▓ | | | | | |
| Condenser Design And selection | | | | | | | | | | ▓ | ▓ | ▓ | | | | |
| Evaporator Design and selection | | | | | | | | | | | ▓ | ▓ | ▓ | | | |
| Pipe Design and selection | | | | | | | | | | | | ▓ | ▓ | | | |
| Electric control Design And selection | | | | | | | | | | | ▓ | ▓ | ▓ | ▓ | | |
| Preparing& printing | | | | | | | | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ |
| Project Documentation | | | | | | | | | | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ |

Table (1.4) Second semester time plan

| Task/Week | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Chambers building | ▓ | ▓ | ▓ | ▓ | | | | | | | | | | | | |
| Accessory Selection | | ▓ | ▓ | ▓ | ▓ | ▓ | | | | | | | | | | |
| Electrical & Control building | | | | | | | ▓ | ▓ | ▓ | ▓ | | | | | | |
| Cycle test | | | | | | | | | | | ▓ | ▓ | ▓ | | | |
| Conclusions | | | | | | | | | | | | | ▓ | ▓ | ▓ | |
| Project Documentation | | | | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ | ▓ |

# CHAPTER TWO

# COOLING LOAD

# COOLING LOAD

## 2.1 Introduction

The total heat required to be removed from refrigerated space 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.[6]

## 2.2 Load Sources

The cooling load on refrigerating equipment seldom results from any one single source of heat. Some of the more common sources of heat that impose the load on refrigerating equipment are:-

- The wall heat gain.
- The product heat gain.
- Infiltration heat gain.
- Packing heat gain.

Chamber are working at between two temperature  degrees $(5, -15)C^0$, chamber are located in Hebron city , From  Palestinian codes the Maximum surrounding  temperature for  Hebron city  is 38∘C (TABLE A-1), The required mass product is 8 [L] water ,6 [L] milk ,5[kg] Beans green , 5[kg] Orange, 3[kg] cheese , 1[kg] eggs in chamber cooling and 25 [kg] fish &25 [kg] Poullry in Freezing chamber . Desired cooling time is 16[hr] .

### 2.2.1 The wall heat gain

The quantity of heat transmitted through the wall is a function of three factors whose relationship is expressed in the following equation:[7]

$$Q = U * A * \Delta T \tag{2.1}$$

Where: A: is the outside surface area of the wall [m$^2$], U is the overall heat transfer coefficient [W/m$^2$. ∘C], $\Delta T$ is the temperature difference across the walls [∘C].

$$\Delta T = T_{out} - T_{in} \tag{2.2}$$

Where $T_{in}$ is the refrigeration space temperature, $T_{out}$ is the outside temperature.

Overall heat transfer coefficient is computed by the following:

$$U = \frac{1}{\frac{1}{hi}+\frac{\Delta x1}{k1}+\frac{\Delta x2}{k2}+\frac{\Delta x3}{k3}+\cdots+\frac{1}{ho}}$$  (2.3)

Where : U: is the overall heat transfer coefficient [W/m². C°] , Δx: is the thickness of the layer of the wall [m] , k: is the thermal conductivity of the material [W/m. ∘C] . $h_i$ : is the convection heat transfer coefficient of inside air film =9.37[W/m². °C], $h_o$ : is the convection heat transfer coefficient of outside air film = 22.7[W/m²°C].

All walls are constructed of three layers as shown in figure (2.1) :

- Galvanized steel sheet 0.06 [cm]    k=20[W/m. °C].  From (TABLE A-2)
- Polyurethane 4 [cm]    k=0.025[W/m. °C]. From (TABLE A-2)
- Galvanized steel sheet 0.06 [cm]   k=20[W/m. °C]. From (TABLE A-2)



GALVANIZED
STEEL   x=0.06 cm

Polyurethane
insulation  x=3 cm
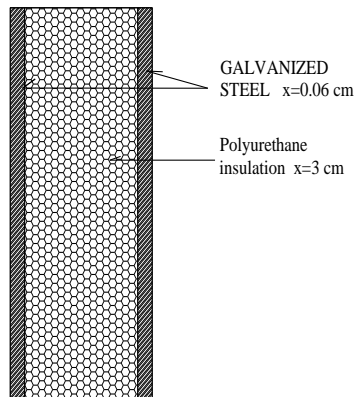
Figure (2.1)  Semitic diagram of constriction wall

Applying equation (2.3), the overall heat  transfer coefficients:

- $U = \frac{1}{\frac{1}{9.37}+\frac{0.0006}{20}+\frac{0.04}{0.025}+\frac{0.0006}{20}+\frac{1}{22.7}} = 0.57\,[W/m^2.\,C°]$

Applying equation (2.1), the wall load for the chamber:

- $Q_{floor} = \{0.57*(1*0.6)*(38- (-15))\}= 18.13$ [W]
- $Q_{roof} = \{0.57*(1*0.6)*(38- (-15))\}= 18.13$ [W]
- $Q_{sides1}= \{0.57*(0.6*0.6)*( 38- (-15))\}= 10.87$ [W]
- $Q_{sides2}= \{0.57*(0.6*0.6)*( 38- (-15))\}= 10.87$ [W]
- $Q_{sides3}= \{0.57*(1*0.6)*( 38- (-15))\}= 18.13$ [W]

10

- $Q_{\text{sides4}} = \{0.57*(1*0.6)*( 38- (-15))\}= 18.14[W]$
- $Q_{\text{all wall}} = 94.26$ [W].


## 2.2.2 The product heat gain


The heat emitted from the product to be stored is very important in case of cold storages. The loads to be considered in the cold storages are divided into the following groups. chilling load above freezing: The product chilling load above freezing depends upon the mass product, mean specific heat of the products above freezing, entering product temperature, final product temperature desired, and the chilling time. The space heat gain from product is computed by the following equation: [6]


$$Q_{ch}= m . C_{p}. \Delta T/\tau \qquad (2.4)$$


Where: $Q_{ch}$ is the quantity of heat [kJ], m: is the mass of the product [kg], $C_{p}$ is the specific heat above\below freezing [kJ/kg.˚C], $\tau$ : desired cooling time [seconds], $\Delta T$: is the change in product temperature [˚C]. The total Cp from (TABLE A-3)


$$\Delta T= T_{o} - T_{ch} \qquad (2.5)$$

Where :
$T_{o}$: is the entering product temperature [˚C].
$T_{ch}$: is the chilling product temperature [˚C].

Applying equation (2.4) to find quantity of heat for the first product (water):

- $CP = 4.18$ [kJ/kg.C°], $T_{o}$=24[°C] , $T_{ch} = 5$ [C °] ,$\tau$ =16hour

- $Q_{ch} = 8*4.18*1000*(24-5)/(16*60*60)$
- $Q_{ch} =11[W]$

Applying equation (2.4), for the second product (Beans green):

- $C_{P} = 3.64$ [kJ/kg.°C] (TABLE A-3) , $T_{o}$= [26.5°C] ,$T_{ch} = $ [5°C] ,t=16 hour

- $Q_{ch} = 5*3.64*1000*(26.5-5)/(16*60*60)$
- $Q_{ch} =7$ [W]

    Applying equation (2.4), for the third product (milk):

- $C_{P} = 3.77$ [kJ/kg.°C] ,$T_{o} = 7.25$[°C] ,$T_{ch} = $ [5°C] ,t=16 hour

- $Q_{ch} = 6*3.77*1000*(7.25-5)/(16*60*60)$
- $Q_{ch} = 0.88[W]$

Applying equation (2.4), for the fourth product (Orange):

- $C_P$ = 3.81 [kJ/kg.°C] ,$T_o$ = 22[°C] ,$T_{ch}$ = [5°C] ,t=16 hour

- $Q_{ch}$ = 5*3.81*1000*(22-5) /(16*60*60)
- $Q_{ch}$ = 6 [W]

Applying equation (2.4), for the fifth product (cheese):

- $C_P$ = 2.68 [kJ/kg.°C] ,$T_o$ = 15[°C] ,$T_{ch}$ = [5°C] ,t=16 hour
- $Q_{ch}$ = 3*2.68*1000*(15-5) /(16*60*60)
- $Q_{ch}$ = 1.4[W]

Applying equation (2.4), for the sixth product (eggs):

- $C_P$ = 3.56 [kJ/kg.°C] ,$T_o$ = 10[°C] ,$T_{ch}$ = [5°C] ,t=16 hour

- $Q_{ch}$ = 2*3.56*1000*(10-5) /(16*60*60)
- $Q_{ch}$ = 0.7 [W]


Applying equation (2.4), for the seventh product (Poullry & fish temperate between (-15 °C):

Fish:
- CP = 3.18 [kJ/kg.C°] (TABLE A-3) ,$T_o$=7[°C] , $T_{ch}$ = -3 [C °], t=16 hour .

- $Q_{ch}$ = 25*3.18*(7-(-3)) = 795 [kJ]


Also, there's another load for freezing (fish) product , calculate use equation (2.6):

$$Q_1 = m * CP * \Delta T \qquad\qquad\qquad Q_2 = m * \Delta h \qquad\qquad \textbf{(2.6)}$$

Where: $Q_1$: is the quantity of heat after freezing [kJ] , $Q_2$ : freezing load , CP: specific heat after freezing [KJ/kg. C°] (TABLE A-3) , $\Delta T$: freezing point to set point , $\Delta h$: latent heat of fusion [kJ/kg].

Applying equation (2.6), for the fish product :

- $Q_1 = 25 * 1.72 * \left(-3 - (-15)\right)$ =516[kJ].    $Q_2 = 25 * 235 = 5875$ [kJ]

- $Q_{\text{total load}} = \dfrac{(Q_{ch} + Q_1 + Q2)}{t} = \dfrac{(795+561+5875)KJ}{16*3600\ s} = 0.12554[KW] = 125.54$ [W]

Poullry:

CP = 3.18 [kJ/kg.C°] (TABLE A-3) ,$T_o$=29.5[°C] , $T_{ch}$ = -3 [C °] .

- $Q_{ch} = 25*3.18*(29.5-(-3)) = 2583.7[kJ]$

Applying equation (2.6), for the Poullry product :

- $Q_1 = 25 * 1.55 * (-3 - (-15)) = 465[kJ].$     $Q_2 = 25 * 246 = 6150\ [kJ]$

- $Q_{total\ load} = \dfrac{(Q_{ch} + Q_1 + Q_2)}{t} = \dfrac{(2583.7 + 465 + 6150\ )}{16*3600} = 0.160[KW] = 160[W]$

## 2.2.3 Infiltration heat gain

In the practical operation of a refrigerated facility, doors must be opened at times in order to move the product in and out. The infiltration load is one of the major loads in the refrigerator. The infiltration air is the air that enters a refrigerated space through cracks and opening of doors. This is caused by pressure difference between the two sides of the doors and it depends upon the temperature difference between the inside and outside air.

The heat gain resulting from air change can be determined by applying the            following equation:

$$Q_{inf} = m^{\cdot}*C_p*(T_o\text{-}T_i\ ) \tag{2.7}$$

$$Q_{inf} = \rho * V^{\cdot}_f*C_p*(T_o\text{-}T_i) \tag{2.8}$$

Where : $\rho$: is the air density [1.25 kg/m$^3$], $C_p$: is the specific heat of the air [kJ/kg.C$^\circ$], $V_f$: is the volumetric flow rate of infiltrated air [m³/s], $T_o$: is the outside temperature [C$^\circ$], $T_i$: is the inside temperature [˚C].

$V^{\cdot}_f$ = number of air change * volume of room.

Number of air change = 0.5 [times /h] (TABLE A-4)

- Volume of room = 0.7*0.6*0.6 = 0.252 m$^3$.
- $V^{\cdot}_f$ = 0.5*0.252 = 0.126 m$^3$/hr.

Applied the equation (2.8), the $Q_{inf}$ for the    product **n**: (water, milk, Beans green, eggs, cheese, Orange.):

- $Q_{inf}$ = n* 1.25* (0.126/3600)*1000*(38- 5)
- $Q_{inf}$ = 7 *1.44 = 10.1[W].

### 2.2.4 Packaging Heat gain

Many products refrigerated in packages. It could be more than 10% of product's weight. Packages could be plastic, steel , wood , glass ,carton or any material that has low specific heat. The milk are stored in carton with 100g ,each Beans green and Orange has 1 box . Plastic box mass has 0.5 kg, each one can hold 5 kg of Beans green or 5kg of Orange, 8 L water are stored in plastic Bottles with 50g , 3kg of cheese are stored in 3 Plastic box with mass of 100g , eggs will store in two plastic boxes with mass of 20 g, fish and Poullry will store in two plastic boxes with mass of 0.5Kg .

.

The heat gain resulting from Packaging can be determined by applying the following equation:[ 6]

$$Q_{pk} = \frac{Mpk*Cpk*(To-Ti)}{\tau} * 10^3 \qquad (2.9)$$

Where :$Q_{pk}$ is the packaging heat load, $m_{pk}$ is the mass of packaging material [kg], $C_{pk}$ is the packaging material specific heat [J/kg.C°], $T_{pk}$ is the packaging material temperature [°C], $T_i$ is the temperature of the refrigerated space [°C], $\tau$: is the desired cooling time [seconds].

$C_{pk\ plastic}$=1.6[kJ/kg. C°] (TABLE A-5), Applying equation (2.9), to find $Q_{pk}$ for the (Beans green , Orange):

- $Q_{pk} = 2 * (\frac{0.5*1*1.6*(24-5)}{16*3600}) * 10^3)$
- $Q_{pk\ plastic}$=0.53[W]

Applying equation (2.9), to find $Q_{pk}$ for (water):

- $Q_{pk} = \frac{0.05*6*1.6*(24-5)}{16*3600} * 10^3$
- $Q_{pk\ plastic} = 0.16$ [W]

Applying equation (2.9), to find $Q_{pk}$ for (milk):

- $Q_{pk} = \frac{0.1*6*0.34*(24-5)}{16*3600} * 10^3$
- $Q_{pk\ plastic} = 0.1$ [W]

Applying equation (2.9), to find $Q_{pk}$ for (cheese):

- $Q_{pk} = \frac{0.1*3*1.6*(24-5)}{16*3600} * 10^3$
- $Q_{pk\ plastic} = 0.158$ [W]

Applying equation (2.9), to find $Q_{pk}$ for (egges):

- $Q_{pk} = \dfrac{0.02*2*1.6*(24-5)}{16*3600} * 10^3$
- $Q_{pk\ plastic} = 0.01[W]$

Applying equation (2.9), to find $Q_{pk}$ for (Fish):

- $Q_{pk} = \dfrac{0.5*3*1.6*(24-5)}{16*3600} * 10^3$
- $Q_{pk\ plastic} = 0.792\ [W]$

Applying equation (2.9), to find $Q_{pk}$ for (, Poullry):

- $Q_{pk} = \dfrac{0.5*3*1.6*(24-5)}{10*3600} * 10^3$
- $Q_{pk\ plastic} = 0.792\ [W]$


### 2.2.5 Respiration gain

Fruits and vegetable are still alive after harvesting and continue to undergo changes while in storage; the more important of these changes are produced by respiration process during which oxygen from the air combines with the carbon rates in the plant tissue and results in the release of carbon dioxide and heat .The heat released is called respiration heat which is found by the following equation: - [6]


**$Q_R$=m*respiration rate** (2.10)

Where:- $Q_{R:}$ the respiration heat load, m: is the mass of product [Kg]. Respiration rate[W/Kg] which equal 0.019[W/Kg]for Orange at 5°C and 0.09 [W/Kg] for beans green at 5°C (TABLE A -11). [6].

Applying equation (2.10), to find $Q_R$ for the (Orange):

$Q_{R\ orange}$ =5*0.019= 0.1[W]

Applying equation (2.10), to find $Q_R$ for the (green banes):

$Q_{R\ green\ beans}$ =5*0.09= 0.45[W]

## 2.3 Total cooling load

The total cooling load is the summation of the heat gains

$$Q_T = Q_w + Q_p + Q_{inf} + Q_{pk} + Q_R \qquad (2.11)$$

Table 2.1 Chamber load

| Chamber | $Q_w$[W] | $Q_p$[W] | $Q_{inf}$ [W] | $Q_{pk}$[W] | $Q_R$[W] | $Q_T$[W] |
|---------|----------|----------|---------------|-------------|----------|----------|
| Chamber | 94.62 | 312.54 | 10.1 | 2.546 | 0.55 | 420.34 |

It is common practice to add 10%-12% as a factor of safety as general rule 10% is used:[6]

- $Q_{T(1)}$=420.34+(420.34*0.10) = 462.4 [W]

## 2.4  Required equipment capacity

After the safety factor has been added, the cooling load is multiplied by 24 hours and divided by the desired operating time in hour to determine the average cooling load which is used as basis for equipment selection:[6]

$$Q_a = \frac{TCL \times 24}{operating\,time} \qquad (2.13)$$

Operating time = 16 hour

$$Q_a = \frac{462.4 \times 24}{16} = 693.6[W]$$

# CHAPTER THREE


# DESIGN AND SELECTION OF CYCLE COMPONENT

## 3.1  Refrigerant Selection

A  refrigerants  are classified into two  main types ,one is a primary refrigerant   ,other is a secondary refrigerant .It absorbs and transports heat from a cooling space.

A primary refrigerant  are classified into five groups, one of these groups named HFC, some of its refrigerant R 134a ($CH_2CFC_3$), table (3.1) shows R134a properties and compare them with other refrigerants  like  (R12, R22). [8]

### 3.1.1   a primary Refrigerant

Refrigerants are used for the same purpose, but some of it give better results than others, so what is required of a good refrigerant? A good refrigerant should have the following properties:

❖  Economical properties.
▪   High operating efficiency.
▪   Low cost.
▪   Availability.

❖  Thermodynamic properties

▪   Low boiling point.
▪   High critical point temperature.
▪   High latent heat of vaporization.
▪   Low specific heat of liquid.
▪   Mixes well with oil.

❖  Environment friendly

▪   Having zero ozone depletion potential(ODP).
▪   Having low global warming potential(GWP).
▪   Non corrosive to metal.
▪   Easily detectable in case of leakage.
▪   Non flammable and non explosive.
▪   Non toxic.

Table (3.1): physical properties of refrigerants (the properties at 1 atm)

| Properties | R 12 | R 134a | R 22 |
|---|---|---|---|
| Molecular formula | $CCl_2F_2$ | $C_2H_2F_4$ | $CHClF_2$ |
| boiling point(°C) | −29.8 | -26.3 | -40.8 |
| Melting point (°C) | ---- | -101 | -157 |
| critical point (°C) | 112 | 101.1 | 96.2 |
| toxic | Non | Non | Non |
| flammable | Non | Non | Non |
| ODP | 1.0 | 0.0 | 0.05 |
| GWP | 8100 | 1,300 | 1700 |

From table (3.1),boiling and melting temperature for R-134a is higher than R-12 and R-22 .All refrigerants are nontoxic, non flammable and low ODP "ozone depletion potential ",but GWP "global warming potential " for R-134a is lower than other refrigerants.

So, R134a is best suited for refrigeration system because R-134a is more friendly than R-12 and R-22.[8]

## 3.1.2 Secondary refrigerant selection

A secondary   refrigerant is the fluid that chilled by conducting the primary refrigerant, it performs the refrigeration process by absorbing cold  from primary refrigerant

The terms secondary refrigerant, brines, and antifreezes are sometimes used Interchangeably   . This section will use secondary coolant as the general description of a fluid with freezing temperatures below those of water. Brines are one class of secondary refrigerant formed by the aqueous (with water) solution of a salt.

The number of candidates for secondary refrigerant is large, and this section  Will attempt to give key data on a wide selection. Of this large number of Possibilities, certain choices are more popular than others, so additional Characteristics will be provided. Key data include the following factors:

• Lowest solidification temperature
• Flammability
• Compatibility with food
• Corrosion tendency and inhibition possibilities

If a candidate possesses satisfactory key data to pass the first stage of the selection process, additional characteristics should be evaluated, such as the thermal and fluid properties, including the following:

• Viscosity
• Specific heat
• Specific gravity or density
• Thermal conductivity

Secondary coolants to be considered in this section include the following:

• Chloride–calcium chloride
• Chloride–sodium chloride
• Glycol–propylene glycol

The selection of desired secondary refrigerant will done according table (3.2), by making compassion including major properties

Table ( 3-2 ) comparison between major secondary refrigerant:[9]

| Fluid | Sodium Chloride ( NaCL) | Calcium Chloride(CaCl$_2$) | Ethylene Glycol |
|---|---|---|---|
| X [*vol* %]concentration of solute ,percent by mass | 18.50 | 18.50 | 30.00 |
| T [°C] | -14.68 | -13.06 | 0.00 |
| Density [kg/m^3] | 1150.70 | 1177.09 | 1051.75 |
| Specific heat [kJ/(kg K)] | 3.438 | 3.121 | 3.590 |
| Conductivity [W/(m K)] | 0.526 | 0.526 | 0.429 |
| Dynamic viscosity [10^-5 Pa·s] | 465.672 | 524.302 | 415.220 |
| Kinematic viscocity [cSt] | 4.047 | 4.454 | 3.948 |
| Freeze point [°C] | -14.85 | -15.72 | -15.79 |

The two important brines for refrigeration service are aqueous solutions of either of two salts—sodium chloride (NaCl) or calcium chloride (CaCl$_2$). NaCl brine is probably the most economical of any available secondary refrigerant, and it can be applied in contact with food and in open systems because of low toxicity.

Furthermore, NaCl brine is nonflammable and has favorable thermodynamic and transport properties. The freezing point, density, viscosity, specific heat, and thermal conductivity of NaCl brine are shown in table (3.1) above. These properties result in high heat-transfer coefficients, although not as good as CaCl2 brine.

Steel, aluminum, and copper are compatible with both glycols, but galvanized steel should be avoided as is true of calcium chloride brine ,Neither  NaCl nor CaCl$_2$ brines should be used with galvanized steel, because the brines attacks the zinc in the galvanized coating. CaCl$_2$ brine is corrosive to the metals of its containment system. For this reason inhibitors must be added to the brine and a maintenance plan should include periodic checks of the brine. [10]

So, NaCL will choose as  a secondary refrigerant in   this project  .

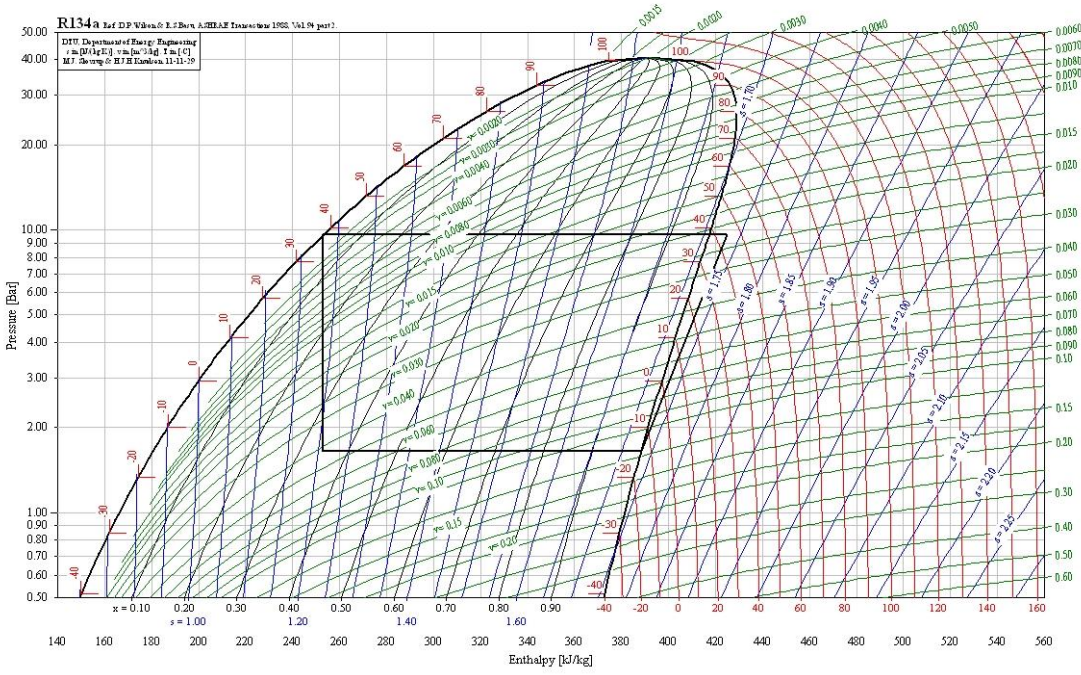## 3.2  Cycle analysis and calculation



Figure (3-1) . P-h diagram of R134a refrigeration system

### 3.2.1 General information

The schematic of such a system and corresponding operating cycle on P-h diagram are shown in Fig (3.1).As shown in the figure the system consists of a single compressor   , a single condenser ,and single evaporators.
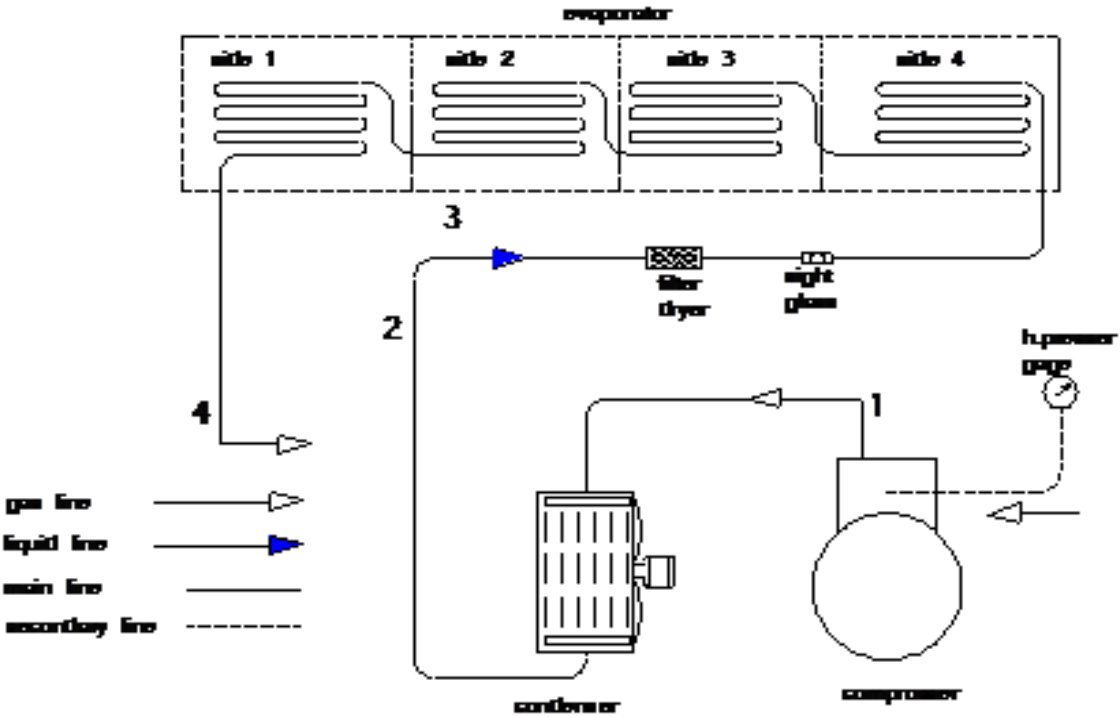


Figure (3.2)  Semitic diagram of  refrigeration cycle

1. Refrigerant enters the compressor at point   (1 ) at the super heated state. It is compressed to the high pressure at point ( 2).
2. Hot gas enters the condenser, in which it is de-superheated and sub cooled at point (3) .
3. After the condensing process, the liquid refrigerant flows through a throttling device (thermostatic expansion valve).
4. Refrigerant is divided into three evaporators, which enters evaporators at low temperature and low pressure at points ( 4 ) The  liquid refrigerant is vaporized at points in the evaporator. The vapor then flow to the inlet of the compressor at point 1 and completes the cycle.

Table (3.3) Properties of each point on the cycle

| # | T(°C) | P(bar) | h(kJ/kg) | υ(m³/kg) |
|---|-------|--------|----------|----------|
| 1 | -15 | 1.64 | 388.32 | 0.012 |
| 2 | 44.65 | 9.63 | 425.2 | 0.022 |
| 3 | 38 | 9.63 | 253.2 | N/A |
| 4 | -15 | 1.64 | 253.2 | 0.12 |

## 3.2.2 Mass flow rate

In this cycle evaporator   has a mass of refrigerant that flows through it .To find the mass flow rate we use the following equation: [11]

$$\mathbf{m^{\cdot}} = \frac{\mathbf{Qe}}{\mathbf{qe}}$$      **(3.1)**

Where:   $m^{\cdot}_1$  is the mass flow rate in the evaporator, [kg/s] , $Q_e$ : refrigeration capacity, [kW] , $q_e$= refrigeration effect[10].

$$\mathbf{q_e} = \Delta\mathbf{h}$$      **(3.2)**

Where $\Delta h$ is the difference between the enthalpy of saturated vapor leaving evaporator and the enthalpy of refrigerant entering evaporator: [k J/kg]   , Applying equation (3.2), the  $q_e$ for the  refrigerated space is:

- $q_e$ = $h_1$- $h_4$ = 388.3 – 253.2=135.1  [kJ /kg]

Applying equation (3.1) the mass flow rate is:

- $m^{\cdot} = \frac{0.693.4}{135.1}$
- m = 0.005314 [kg/s]

### 3.2.3 Condenser load

After calculating the total mass flow, condenser load is found by the following equation:[11]

$$Q_c = m^{\cdot} (h_2 - h_3) \qquad\qquad (3.3)$$

Where: $Q_c$ is the condenser load [kW], $h_2$ is the enthalpy of the hot gas discharged from the compressor [kJ /kg], $h_3$ is the enthalpy of the liquid discharged from the condenser [kJ /kg]. Applying equation (3.3) and table (3.3) the condenser load is :

- $Q_c = 0.005314\ (425.2\text{-}253.1)$
- $Q_c = 0.914\ [kW]$

### 3.2.4  Power of compressors

To find the compressor power , we use the following equation:[ 11]

$$P = m^{\cdot} *(h_2 - h_1) \qquad\qquad (3.4)$$

Where P: is the compressor power [kW], $h_2$: is the enthalpy of the hot gas discharged from the compressor [kJ /kg] ,$h_1$: is the enthalpy of the gas discharged from the evaporator [kJ /kg].

Applied equation (3.4) the power of compressor is :

$P = 0.005314\ (425.2 - 388.3)$
$P = 0.196\ [kW]$
$P = 196\ [W] \qquad\qquad P = 1\backslash3\ hp = 248.6[W]$

### 3.2.5 Coefficient of performance

$$COP = \frac{Qe}{W} \qquad\qquad (3.5)$$

Where COP is the Coefficient of performance of the cycle, P is the compressor power [kW], $Q_{e\ total}$ is the total cooling  capacity [KW]. [6]

- $COP = \dfrac{0.6936}{0.196}$
- $COP = 3.54$

To ensure of the accuracy of the calculation a software program (Cool Pack) was used ,a single stage compression cycle at the lowest evaporating temperature   and at the same

condensing temperature give a cooling capacity(0.696 [kW]) ,compressor power ( 0.198[kW]), condensation capacity (1 [kW]) , refrigerant mass flow (0.0048 [kg/s]) , and COP( 3.6)  .

## 3.3  Mechanical Cycle Components

Mechanical   and electrical components are required to work integrally together in this refrigeration system. In this part, discuss the main and auxiliary components of   a system to make design   and selection of these components

The main components of the  refrigeration system are :

- Compressor.
- Condenser.
- Evaporator
- Connecting pipes.
- Throttling device(capillary tubes).

### 3.3.1 Compressor

The refrigeration compressor (vapor pump) is the heart of the refrigeration system. It moves the vapor to high pressure side of the system. It maintains the low side pressure at which the refrigerant evaporates, and the high side pressure at which it condenses, it supplies the pressure difference necessary to keep the system refrigerant flowing through the system. [12]

There are many types of compressors; one of them is called hermetic compressor. In this compressor, a motor and a compressor are connected together. The reason for choosing this compressor is that:

- It has a long life, so the life of the equipment should be even longer.

### 3.3.1.1 Compressor  calculation:

The actual volumetric flow rate for the compressor must be calculated to determine the type of compressor. To find this value; volumetric efficiency and the theoretical volume flow rate of the compressor must be calculate.

### 3.3.1.2 The volumetric efficiency

First, the volumetric efficiency must be determined using equation (3.6) is  [8]

$$\eta_v = \eta_c * \eta_h \tag{3.6}$$

Where: $\eta_v$ is the volumetric efficiency, $\eta_c$ is the volumetric efficiency due to clearance volume in compressor, $\eta_h$ is the volumetric efficiency due to heating occurring in compressor.

But the volumetric efficiency due to the clearance volume in compressor is calculated by equation (3.7), [8]:

$$\eta_c = 1 - c[\ (\frac{PH}{PL})^{1/n} - 1]$$ (3.7)

Where c: is the 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 [reference 8], n is the exponential coefficient of expansion for refrigerant, n = 1. [Reference8], PH is the high pressure of the cycle; PL is the low pressure of the cycle. Applied equation (3.7) the volumetric efficiency is:

- $\eta_c = 1 - 0.02[\ (\frac{9.63}{1.64})^{1/1} - 1] = 90.2\%$

Equation (3.8) is used to find the volumetric efficiency due to the heating in compressor [7]:

$$\eta_h = \frac{Tevap}{Tcond}$$ (3.8)

Where $T_{evap}$ : is the evaporator temperature [°K] , $T_{cond}$ is the condenser temperature [ °K] :

- $\eta_h = \frac{258}{311} = 83\%$

Applying equation (3.6), the volumetric efficiency is :

- $\eta_v = 90.2\% * 83\% = 75\%$

### 3.3.1.3 The theoretical volume flow rate

The theoretical volume flow rate (V˙) of the compressor can be calculated by using equation (3.9),[8]:

$$\dot{V}_{theo} = \dot{m} * \upsilon$$ (3.9)

Where: - $\dot{V}_{theo}$ is the theoretical volume flow rate of the compressor [m³/s], m˙ is the mass flow rate of refrigerant [kg/s], $\upsilon$ is the specific volume at the inlet of compressor [m³/kg] which =0.012 from the PH diagram. Applied equation (3.9):

- $\dot{V}_{theo} = 0.005314 * 0.012 = 6.38 * 10^{-5}$ [m³/s]

To determine the actual volume flow rate by the equation (3.10), [8]:

$$\dot{V}_{act} = \frac{V.theo}{\eta v}$$ (3.10)

Where $\dot{V}_{act}$ : is the actual volumetric flow rate[m³/s], applied equation (3.10) the actual volumetric flow rate is  :

- $\dot{V}_{act} = \frac{6.38 * 10^{\wedge}10 - 5}{0.75} = 8.51 * 10^{-5}$[m³/s]

### 3.3.1.4 The compressor selection

The main consideration to select the compressor is the actual volumetric flow rate, so we chose a compressor that satisfies it, danfoss compressor is selected for this project from AC compressor catalog , we select a AC compressor type **SC12G** .Which has displacement of 3.5 $cm^3$ per revolution and 3500 RPM .

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

$$\mathbf{V^{\cdot}_{act} = V^{\cdot}_{theo} * \eta_v} \tag{3.11}$$

Where: $-V^{\cdot}_{act}$ is the actual volumetric flow rate for the compressor $[m^3/s]$, $V^{\cdot}_{theo}$ is the theoretical volumetric flow rate for the compressor $[m^3/s]$, $\eta_v$: is the volumetric efficiency.

- $V^{\cdot}_{theo} = 3.5 * 10^{-6} * (3500/60) = 2.04 * 10^{-4} \, [m^3/s]$
- $V^{\cdot}_{act} = = 2.04 * 10^{-4} * 0.75 = 1.53 * 10^{-4} \, [m^3/s]$

### 3.3.2 Condenser

A condenser design was manufactured. The condenser was made from a (10mm) diameter, copper tube, the following is geometrical data, condenser:

- condenser length $L_c$ =30 [cm]
- condenser height $H_c$ =20 [cm]
- condenser width $W_c$ =7[cm]

$S_n$: Transverse tube spacing [m]

- $S_n = \frac{condenserheight}{numberofrows} = \frac{0.20}{10} = 0.02 \, [m]$

$S_p$: Longitudinal tube spacing [m]

- $S_p = \frac{evaporator \quad width}{numberofcolumn} = \frac{0.07}{4} = 0.0175[m]$

$finlength = L_f = (S_n - D_0)$

- $L_f = (0.020 - 0.01) = 0.01[m]$

$Finwidth = W_f = (S_p - D_o)$

- $W_f = (0.0175 - 0.01) = 0.0075[m]$

- number of fins in a row =100 fins
- fin pitch $= P_f = \frac{20}{100} = 0.2$ [cm]
- $Finthickness = t_f = 0.01[cm]$
- $baretubethickness = t_b = P_f - t_f$

- $t_b = 0.2 - 0.01 = 0.19[\text{cm}]$

Condenser has two regions; first region is (desuperheating) region then is (mixture) region, now desuperheating must be calculated:

### 3.3.2.1 Desuperheating region

Applying equation (3.12) the Q $_{desuper}$ is [8]:

$$\text{Q }_{\textbf{desuper}} = \dot{\textbf{m}}*(\textbf{h}_{\textbf{in}}\textbf{-  h}_{\textbf{sat. vap}})\tag{3.12}$$

Where Q $_{desuper}$ : condenser load from desuperheating region [W], ṁ: mass flow rate in the condenser and calculated in chapter three , [kg/s], $h_{in}$ :enthalpy of refrigerant entering the condenser, [kJ /kg], $h_{sat.vap.}$: enthalpy of saturated vapor leaving the desuperheating region in condenser [kJ /kg]so:

- Q $_{desuper}$ = 0.005314 (425.3- 420) = 28.2[W]

From (table A-8) value of C, N are determined by $S_p$/D and $S_n$/D, so the value $S_p$/D= 17.5/10 =1.75, $S_n$/D = 20/10 =2 :

- C =0.183
- N=0.667

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

$$\textbf{V}_{\textbf{max}} = \frac{\textbf{V}\infty \left(\frac{\textbf{Sn}}{2}\right)}{\sqrt{\left[\left(\frac{\textbf{Sn}}{2}\right)^2+\textbf{Sp}^2\right]-\textbf{D}}}\tag{3.13}$$

$$\textbf{V}\infty = \frac{\dot{\textbf{V}}}{\textbf{A}}\tag{3.14}$$

Where: $V_\infty$ is the free air velocity entering the condenser [m/s] , A area of condenser.

Applying equation (3.14), the free air velocity is determined by a known value, $\dot{V}$= 600 [m³/h]=0.1666 [m³/s]from fan manufacturer company.

- A = 0.3*0.20 =0.06 [m²]
- $V_\infty = \frac{\dot{V}}{A} = \frac{0.1666}{0.06} = 2.7$m/s

Applying equation (3.13) the max velocity is :

$$V_{\max} = \frac{2.7\left(\frac{0.020}{2}\right)}{\sqrt{\left[\left(\frac{0.020}{2}\right)^2+0.0175^2\right]-0.01}}=2.66 \text{ [m/s]}$$

$$Tf = \frac{Tw + T\infty}{2} \qquad\qquad (3.15)$$

Where $T_f$ : film temperature[ºC], $T_w$ : wall surface temperature[ºC], assume that it equals the refrigerant temperature, $T_\infty$:free air temperature[ºC].

The properties of air are evaluated at the film temperature, which can calculated by equation (3.15), applying this equation:

- Tav. w $= \frac{45+44.65}{2} = 45$ [ºC]
- Tf $= \frac{45+38}{2} = 41.5$ [ºC] $= 314.5$ [ºK]

Using table A-9, different properties can be determined:

- $\rho$ =1.1254 [kg/m³]
- $\mu$= 1.912 *10⁻⁵ [kg/m.s]
- K =0.0273 [W /m.ºC]
- Pr =0.704

$$Re = \frac{\rho * Vmax * D}{\mu} \qquad\qquad (3.16)$$

Where $\rho$: density of air at film temperature [kg/m³], $V_{max}$: maximum velocity of air between the evaporator tubes [m/s], D :outer diameter of the condenser tubes [m], $\mu$ :dynamic viscosity of air at film temperature [Pa.s].

Applying equation (3.16), Renaldo number is :

- Re $= \frac{1.1254 * 2.66 * 0.01}{1.912 * 10^{-5}} = 1566.$

$$Nu=C(Re)^N Pr^{1/3} \qquad\qquad (3.17)$$

Where Re: Reynolds number, Pr: Prandtl number of air at film temperature,(C,N):constants can be obtained from table (A-8) according the following considerations.

- $S_p/D$= 17.5/10 =1.75
- $S_n/D$ = 20/10 =2 :
- From In line arrangement tube banks table(C =0.183, N=0.667).

Applying equation (3.17), Nusselts numbers :

- Nu=0.183(1566)^{0.667} * (0.704)^{1/3} =25.63

$$h_o= \frac{Nu * k}{D} \qquad\qquad (3.18)$$

28

Where Nu: Nusselts number ,k: thermal conductivity of air at entrance of condenser [W/m ℃], D: outer diameter of evaporator [m]. The Nusselts number can be calculated by the equation. [8]:

Applying equation (3.18), external convection heat transfer coefficients :

- $h_o = \dfrac{25.63*0.0273}{10*10^{-3}} = 70$ [W/m$^2$℃]

The heat transfer coefficient that would be obtained if there are 10 rows of tubes in the direction of the flow ,because there are only 4 rows ,this value must be multiplied by the factor 0.83 ,as determined from (table A-10) [ 8 ]

- $h_o = 70*0.83 = 58.1$ [W/m$^2$℃]

In order to calculate the total heat transfer from one element  (one fin and one bare tube) the equation (3.19) is used [8]:

$$Q_{total} = Q_{fin\ act} + Q_{original} \tag{3.19}$$

Where Q $_{total}$ : the total heat transfer from the element[W], Q $_{fin\ act}$ : actual heat transfer rate per fin[W], Q $_{original}$: heat transfer rate from tube without fin[W], Q $_{original}$ can be calculated by the equation:

$$Q_{original} = h_o * A_{original} * (T_w - T_\infty) \tag{3.20}$$

Where h$_o$: external convection heat transfer coefficient [W/m2℃], A$_{original}$:the outer surface area of bare tube [m$^2$], T$_w$: outer evaporator wall temperature [℃], T$_\infty$: free air temperature:

- $A_{original,} = \pi*D*L = \pi*10*10^{-3}*2*10^{-3} = 62.83*10^{-6}$ [m$^2$]
- $Q_{original} = 58.1*62.83*10^{-6}*(45-38) = 0.0255$ [W]

Q $_{fin}$ can be calculated by the equation (3.21):

$$Q_{fin} = h_o\, A_{fin}(T_w - T_\infty) \tag{3.21}$$

Where:  Q$_{fin}$: theoretical heat transfer rate per fin [W], A$_{fin}$ : surface area for fin [m2]:

- $A_{fin} = 2(S_n*S_P - A_{pipe}) = 2[0.02*0.0175 - \frac{\pi}{4}(0.01)^2] = 0.54*10^{-3}$ [m$^2$]
- $Q_{fin} = 58.1*0.54*10^{-3}(45-38) = 0.22$ [W]

Now, Fin efficiency is calculated by the following equations[7]:

$$\eta_f = \frac{\tanh(m\grave{L}_f)}{m\grave{L}_f} \tag{3.22}$$

$$\grave{L}_f = \left(\frac{L_f}{2}\right)\left[1 + 0.35\ln\frac{(\frac{D_0}{2} + \frac{L_f}{2})}{\frac{D_0}{2}}\right] \tag{3.23}$$

- $\grave{L}_f = \left(\frac{0.01}{2}\right)\left[1 + 0.35\ln\frac{(\frac{0.01}{2} + \frac{0.01}{2})}{\frac{0.01}{2}}\right] = 0.0062[m]$

$$m = \sqrt{\frac{h*P}{kA}} \tag{3.24}$$

Where: h:external convection heat transfer coefficient [W/m².ºC], k:thermal conductivity of cupper fin ,[W/m. ºC], P :perimeter of the fin[m], A :surface area for convection of fin[m²]:

Fin efficiency can be calculated by equation (3.21):

- perimeter of the fin = 2*t+2*L =2*0.1*10⁻³+2*0.022 =0.0442[m]
- Area of fin = t*L =0.1*10⁻³*0.022 =2.2*10⁻⁶[m² ]

Applying equations (3.22) and (3.24):

- $m = \sqrt{\frac{58.1*0.0442}{386\times2.2\times10^{-6}}} = 55$
- $\eta_f = \frac{\tan h(55*0.0062)}{55*0.0062} = 0.90$

So the actual heat transfer flow through the fin is

- Q $_{fin\ act}$ = 0.22*0.90= 0.2[W]

Now the total heat transfer from the element is:

- Q $_{total}$= 0.2 + 0.0255=0.225 [W]

Now the number of elements needed to perform the condenser load to desuperheating process can be determined by the following equation:

$$n = \frac{Qdesuper}{Qtotal} \tag{3.25}$$

- n= $\frac{28.2}{0.225}$ =125.33 elements

### 3.3.2.2 Mixture region

After finishing the desuperheating region, and finding the element number, the mixture region will be calculated and find the number of element, so applying equation (3.26), Q mix is determined :

**Q mix =m· (h$_{sat.vap}$- h$_{sat. liq.}$)**                         **(3.26)**

Where: Q$_{mix}$ condenser load from mixture region [W], ṁ·: mass flow rate in the condenser, [kg/s],h $_{sat.vap}$.: enthalpy of saturated vapor entering the mixture region in condenser, [kJ /kg], h$_{sat.liq.}$: enthalpy of saturated liquid leaving the mixture region in condenser [kJ /kg].

- Q$_{mix}$ = 0.005314 ( 420-253.2) = 886.4[W]

The properties of air are evaluated at the film temperature .It can calculated by equation (3.15), applied this equation :

- $Tf = \frac{45+38}{2}$ =41.5 [ºC] =314.5 [ºK]

    Then from table A-9
- ρ =1.1254 [kg/m$^3$]
- μ= 1.912 *10$^{-5}$ [kg/m.s]
- K =0.0273 [W /m.ºC]
- Pr =0.704

Applying equation (3.16), Renaldo number is :

- Re $= \frac{1.1254*2.66*0.01}{1.912*10^{-5}}$ =1566

Applying equation (3.17), Nusselts numbers :

- Nu=0.183(1566)$^{0.667}$ * (0.704)$^{1/3}$ =25.63

- Applying equation (3.18), external convection heat transfer coefficients :

- h$_o$= $\frac{25.63*0.0273}{10*10^{-3}}$ =70 [W/m$^2$ºC]

This is the heat transfer coefficient that would be obtained if there are 10 rows of tubes in the direction of the flow .Because there are only 4rows ,this value must be multiplied by the factor 0.83 ,as determined from table A-10[reference 8],so:

- h$_o$= 70 *0.83=58.1[W/m$^2$ºC]

In order to calculate the total heat transfer from one element  (one fin and one bare tube) the equation (3.20) is used [8]:

- A$_{original,}$= π*D*L =π *10*10$^{-3}$*2*10$^{-3}$=62.83*10$^{-6}$ [m$^2$]

31

- $Q_{original} = 58.1 \times 62.83 \times 10^{-6} \times (45-38) = 0.0255$ [W]

  $Q_{fin}$ can be calculated by the equation (3.21):

- $A_{fin} = 2(S_n \times S_P - A_{pipe}) = 2[0.02 \times 0.0175 - \frac{\pi}{4}(0.01)^2] = 0.54 \times 10^{-3}$ [m²]
- $Q_{fin} = 58.1 \times 0.54 \times 10^{-3} \times (45-38) = 0.22$ [W]

Fin efficiency can be calculated by equation (3.22):

- $\grave{L}_f = \left(\frac{0.01}{2}\right)\left[1 + 0.35 \ln \frac{(\frac{0.01}{2} + \frac{0.01}{2})}{\frac{0.01}{2}}\right] = 0.0062 [m]$

- perimeter of the fin $= 2 \times t + 2 \times L = 2 \times 0.1 \times 10^{-3} + 2 \times 0.022 = 0.0442$ [m]
- Area of fin $= t \times L = 0.1 \times 10^{-3} \times 0.022 = 2.2 \times 10^{-6}$ [m²]

  Applying equations (3.22) and (3.24):

- $m = \sqrt{\frac{58.1 \times 0.0442}{386 \times 2.2 \times 10^{-6}}} = 55$
- $\eta_f = \frac{\tan h(55 \times 0.0062)}{55 \times 0.0062} = 0.90$

So the actual heat transfer flow through the fin is

- $Q_{fin\ act} = 0.22 \times 0.90 = 0.2$ [W]

Now the total heat transfer from the element is:

- $Q_{total} = 0.2 + 0.0255 = 0.225$ [W]

  Now the number of elements needed to perform the condenser load to Mixture process can be determined by the following equation(3.25):

$n = \frac{Qdesuper}{Qtotal}$

- $n = \frac{886.4}{0.225} = 3939.4$ elements

Now the number of elements needed to perform the total condenser load is:

- $N_{total} = 125.33 + 3939.4 = 4064.77$ elements, Number of elements in available condenser $150 \times 30 = 4500$ elements.

### 3.3.3 Evaporator design

An evaporator design will be manufacture as a close loop copper tube ,without fins . The evaporator was made from copper tub ,the dimension of evaporator is shown below :



( A)                                                    (B)
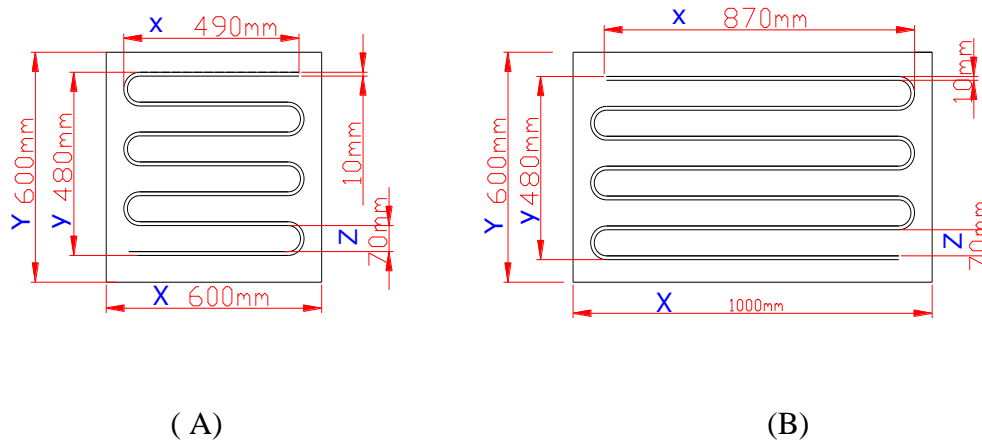
Fig(3.3 ) evaporator schematic .

To calculate pipe area ,it must calculate total area include space area between pipes and so it can subtracted from total area to find pipe area :

$$A_{pipe} = X* Y- (6x*Z). \tag{3.27}$$

Appling equation (3.27) calculate area of (A) side evaporator:

$$A_{pipe} = (0.6*0.6) - (6*0.49*0.07) = 0.1542 \text{ m}^2$$

Appling equation (3.28) calculate length of evaporator:

$$A \text{ pipe } = \pi DL.$$

$$\tag{3.28}$$

$0.1542 = \pi*0.01*L$
$L = 4.6 \text{ m}$

Product the value with safety factor to avoid fittings and elbow by 10%
So, pipe Length in each side L=5 m.

The total pipe length in all tow sides is:

$L_{total} = 2*5 = 10 \text{ m}$

Appling equation (3.27) calculate area of (B) side evaporator:

$$A_{pipe} = (1*0.6) - (6*0.87*0.07) = 0.2346 \text{ m}^2$$

Appling equation (3.28) calculate length of evaporator:

$$A \text{ pipe } = \pi DL.$$

$0.2346= \pi*0.01*L$

$L= 7.3$ m

Product the value with safety factor to avoid fittings and elbow by 10%
So, pipe Length in each side L=8 m.

The total pipe length in all three sides is:

$L_{total}= 3*8= 24$ m

The total pipe length in all sides is:

$L_{total}= 24+10 = 34$ m

## 3.3.4 Pipe Design and Selection

To calculate the inner diameter for the pipe the following steps can be used: [7]

$$Q=m\dot{}*\upsilon \qquad (3.29)$$

Where Q: is the flow rate [$m^3$/s], m˙ is the mass flow rate of refrigerant [kg/s],$\upsilon$ is the specific volume [$m^3$/kg] ,[table 3.3] [reference 8]:

$$Q=V*A \qquad (3.30)$$

Where V: is the velocity of refrigerant[m/s]. [TABLE A-9] ,A is the cross sectional area [$m^2$]

$$A= \pi d_i^2 /4 \qquad (3.31)$$

Where: $d_i$ is the inner diameter [m]:

$$d_i = \sqrt{\frac{A*4}{\pi}} \qquad (3.32)$$

To calculate the outer diameter for the pipe can be as follows[8]

$$\sigma_t =\frac{Pin(ro^2+ri^2)}{(ro^2-ri^2)} \qquad (3.33)$$

Where $\sigma_t$: is the tangential stress [Mpa], $P_{in}$: is the inner pressure [Mpa], $r_o$: is the outer radius [m], $r_i$: is the inner radius [m]. $\sigma_t$: can be calculated from the following equation:[reference 10]:

$$\frac{SY}{n} = \sqrt{\sigma t^2 + Pin\sigma t + Pin^2} \qquad (3.34)$$

34

Where $S_{Y.}$is the yield strength [MPa],[70 MPa for copper ],[8], n is the factor of safety, taken 8 [recommended from cooper hand book]. Now by using equation (3.35) the outer radius of the pipe can be calculated:

**t = r$_o$–r$_i$** (3.35)

Where: - t is the thickness of the pipe [mm]

### 3.3.4.1 Discharge line pipe

Discharge line is the outlet pipe from compressor. To determine the diameter of discharge line, last equation used :

- $Q = 0.005314 * 0.012 = 6.38 * 10^{-5}$ [m$^3$/s]
- $A = \frac{Q}{V} = \frac{0.000938}{10} = 6.38 * 10^{-6}$ [m$^2$]
- $d_i = \sqrt{\frac{0.00000364 * 4}{\pi}} = 2.85 * 10^{-3}$ [m]
- $r_i = 1.425 * 10^{-3}$ [m] = 1.425 [mm]

- $\frac{700}{8} = \sqrt{\sigma t^2 + 1.64 \, \sigma t + (1.64)^2}$
- $\sigma_t = 86.67$ [ bar]

- $86.67 = \frac{1.64 \, (ro^2 + 0.001425^2)}{(ro^2 - 0.001425^2)}$
- $r_o = 1.452$ [mm]
- $t = 1.452 - 1.425 = 0.027$[mm]

The inner and outer diameter in inch is :

- $d_{i, \, inch} = d_{i, \, mm} / 25.4 = 2.85 / 25.4 = 0.112$ [inch]
- $d_{o, \, inch} = d_{o, \, mm} / 25.4 = 2.904 / 25.4 = 0.1143$ [inch]

By referring to copper hand book (TABLE A-7), the suitable type selected is ACR type (Air-conditioning and Refrigeration Field Service), and according to pervious calculations, the nominal or standard size (inches) for this section is 3/16A which has outer diameter 0.187 inch, and inside diameter 0.128 inch .

### 3.3.4.2 Suction line pipe

Inlet pipe for compressor named suction line. Last equation is used to determine the suction line diameter:

- $Q = 0.005314 * 0.022 = 1.17 * 10^{-4}$ [m$^3$/s]
- $A = \frac{Q}{V} = \frac{0.000117}{14} = 8.35 * 10^{-6}$[m$^2$]
- $d_i = \sqrt{\frac{0.00000835 * 4}{\pi}} = 3.26 * 10^{-3}$ [m]
- $r_i = 1.63 * 10^{-3}$ [m] = 1.63 [mm]
-

- $\frac{700}{8} = \sqrt{\delta t^2 + 9.63\delta t + (9.63)^2}$
- $\delta_t = 82.82$ [ bar]
- $82.82 = \frac{9.63(r_o^2 + 0.00163^2)}{(r_o^2 - 0.00163^2)}$
- $r_o = 1.832$ [mm]
- $t = 1.832 - 1.63 = 0.202$[mm]

The inner and outer diameter in inch is
- $d_{i,\,inch} = 3.26/25.4 = 0.128$ [inch]
- $d_{o,\,inch} = 3.664/25.4 = 0.144$ [inch]

According to pervious calculations, the nominal or standard size (inches) for this section is 1/4D which has outer diameter 0.25 inch, and inside diameter 0.19 inch .

## 3.3.5 Capillary tube Design and Selection

By using the software Danfoss capillary tube selector under the following conditions the condenser temperature 45C°, the evaporator's temperature (-15C°), the cooling load (693.6 W), and R143a is the refrigerant used we found that the length and the diameter of the tube as in the table:-

Table 3.4 Capillary tube Selection

|  | L[m] | Φ[m] |
|---|---|---|
| **Chamber** | 1.80 | 0.042 |

# CHAPTER FOUR

# ELECTRICAL AND SOLAR PANEL DESIGN AND SELECTION

# CHAPTER FOUR

# ELECTRICAL DESIGN AND SELECTION

## 4.1 Introduction

This chapter presents the protection of refrigeration cycle. The refrigeration cycle is keeping food at a suitable temperature. If a motor stops at several times due to power failure or a single breakdown, food will be destroyed. So the electric motor must be protected.

## 4.2 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.[13]

### 4.2.1 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. [13]

### 4.2.2 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.[13]

## 4.3 Components of Electrical Circuits

The electrical circuit is built with more conditions. First the requirement is economy and this is important in the installation of any equipment. Second cost is an impotent part of total cost; repair and maintenance costs are also part of total cost. Third the requirement   for driving equipment is simplicity, simple mean can be understood, simple to operate, maintain and repair. There are many components which are used in this project to protect a refrigeration cycle. The components are:

### 4.3.1 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 4.3 shows thermostat. [13]
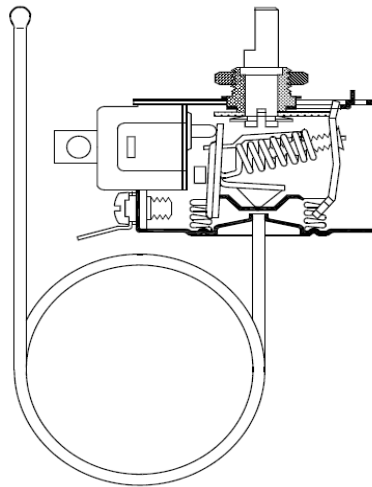


Figure (4.1) Thermostat

### 4.3.2 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. [12]

### 4.3.3 Contactor

Figure (4.4) 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. [12]
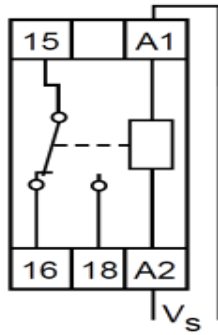
**(a)**  **(b)**

Figure (4.2) (a) a schematic diagram of contactor and (b) contactor device

### 4.3.4 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. [7]

## 4.4 Control circuit



Figure (4.3) control circuit

- When pressing on the main switch  S1 (plug) and  there is no signal from thermostat so compressor will run ,the compressor will continue operate until   signal reached   from thermostat.
- When compressor contact (K1) activate then it will contact fan contactor (K2)and condenser fan will operate.
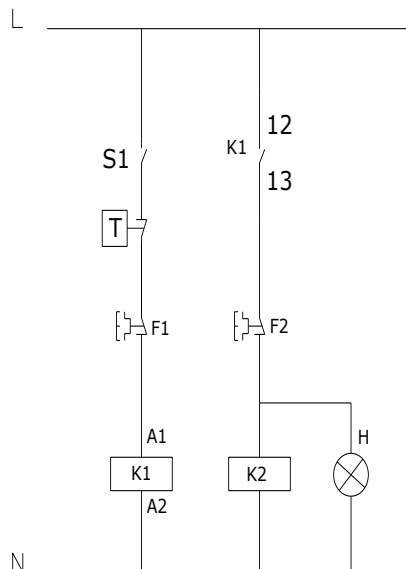- Lamp(H1)will continue light while compressor operate in normal mode.

# 4.5 Solar panel

### 4.5.1 Solar  ' 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 Gigatones  of $CO_2$ 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 viable competitors in the PV marketplace.[14]

### 4.5.2  ( PV ) analyses and principal of operating

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]

## 4.5.3 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:

  i)      single crystalline (sc-Si)
  ii)     multi-crystalline (mc-Si).

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

i)      amorphous (a-Si) and micro morph silicon (a-Si/μc-Si).
ii)     Cadmium-Telluride (CdTe).
iii)     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]

### 4.5.4 Types of 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]
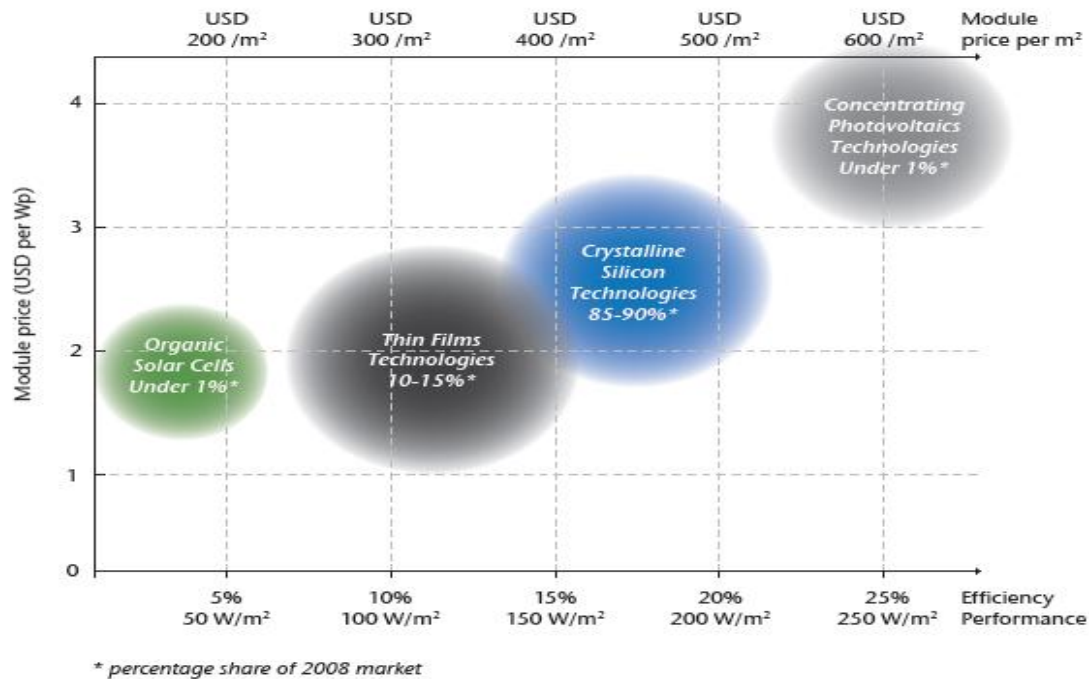


Figure 4.4: Current performance and price of different PV module technologies

Table 4.1: Current efficiencies of different PV technology commercial modules

| Wafer-based c-Si | | Thin films | | |
|---|---|---|---|---|
| sc-Si | mc-Si | a-Si; a-Si/μc-Si | CdTe | CIS/CIGS |
| 14-20% | 13-15% | 6-9% | 9-11% | 10-12% |

With good efficiency and availability in Palestine market **amorphous** PV cells are selected, with Efficiency 10-15%, Model VAF 1400 serial number000010000551 ,made in France.
24 volt ,110 W, two panels, which shown in figure (4.6) below .
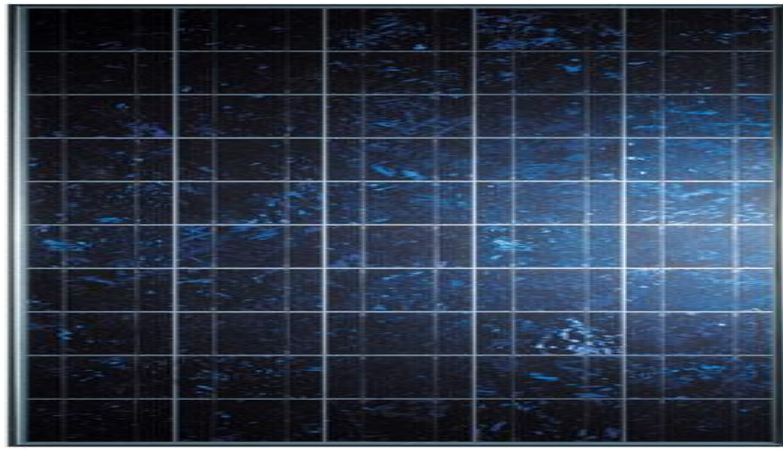
Figure 4.5: amorphous solar panel.

## 4.5.5 Calculation of solar panel
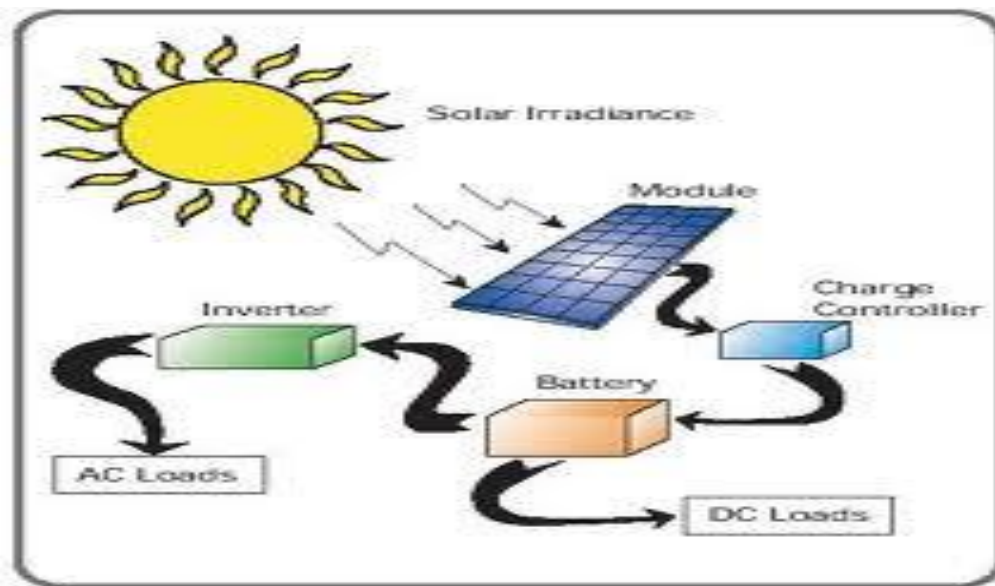


Figure 4.6: convert to solar energy to electrical energy

Load of compressor is 150 w and calculate load of solar panel is :

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

$E_{load} = Q_{comp} *$ time operating **(4.1)**

Appling equation (4.1) :

$E_{load} = 250 * 8$ h/day $= 2000$ w.h/day

$$E_{inverter} = \frac{E_{comp}}{\eta_{inv}} \qquad\qquad\qquad\qquad\qquad \textbf{(4.2)}$$

$$E_{inverter} = \frac{2000}{0.85} = 2353 \text{ w.h/day}$$

$$E_{charge\ controller} = \frac{E_{inverter}}{\eta_{ch}} \qquad\qquad\qquad\qquad \textbf{(4.3)}$$

$$E_{charge\ controller} = \frac{2353}{0.9} = 2614 \text{ w.h/day}$$

$$E_{solar\ panel} = \frac{E_{charge\ controller}}{\eta_{pv}} \qquad\qquad\qquad \textbf{(4.4)}$$

$$E_{solar\ panel} = \frac{2614}{0.1} = 26140 \quad \text{w.h/day}$$

Calculate area of solar panel by, Applying equation (4.4):

$$1\text{m}^2 \longrightarrow 5400[\text{w.h / day}]$$
$$\text{A} \longrightarrow E_{solar\ panel} \qquad\qquad\qquad \textbf{(4.5)}$$

$5400[\text{w.m}^2/\text{day}]$ : annual rate of radiation
A: area of solar panel.

$$1\text{m}^2 \longrightarrow 5400[\text{w.h / day}]$$
$$\text{A} \longrightarrow 26140$$
$$\text{A} = \frac{26140}{5400} = 4 \ [\text{m}^2]$$

## 4.5.6 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 from the photovoltaic(PV) panel in charging the battery according to the battery state. It has 3 stages of charging, as follows:[15]

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 (approximately14.3 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.[15]

**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  .[15]

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



Figure (4.7) PV Charge Controller

## 4.5.7 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 appliances but the usual AC mains power is not available. Examples include operating appliances 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
Transistors as electronic switches. This gives very high conversion efficiency, but the .alternating pulses. output wave form 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 a high frequency DC-DC converter. It then uses a high frequency PWM 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 50Hz sine wave output. This is like a high-voltage version of the single-bit digital to analog conversion process used in many CD players.

After electrical calculation and design , Info Inverter 240020, rating power 2400VA, are selected.



Figure (4.8) PV inverter
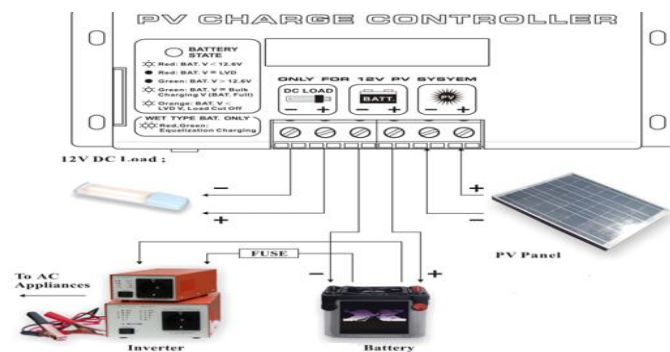


Figure (4.19) Connection of PV Charge Controller and inverter [16]

## 4.5.8 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 250 W, power factor 0.85, voltage of batteries =24 V, supply needed time from batteries =8 hours, So the energy of inverter equal 2353 w.h/day.

$$I = \frac{2353}{24 * 0.85 * 0.8} = 150 \text{ Ah/day}$$

Ah/day  for one battery $= 150$ Ah/day $/2 =75$ Ah/day

So, two (12V, 100Amp) batteries type UBS 100Ah vision 6FM100-X are needed for this operating time (8hr).



Figure (4.10) PV storage batteries

# CHAPTER FIVE

# DESIGEN OF REFRAGRATION

# PART A

## DESIGN OF CHAMBER
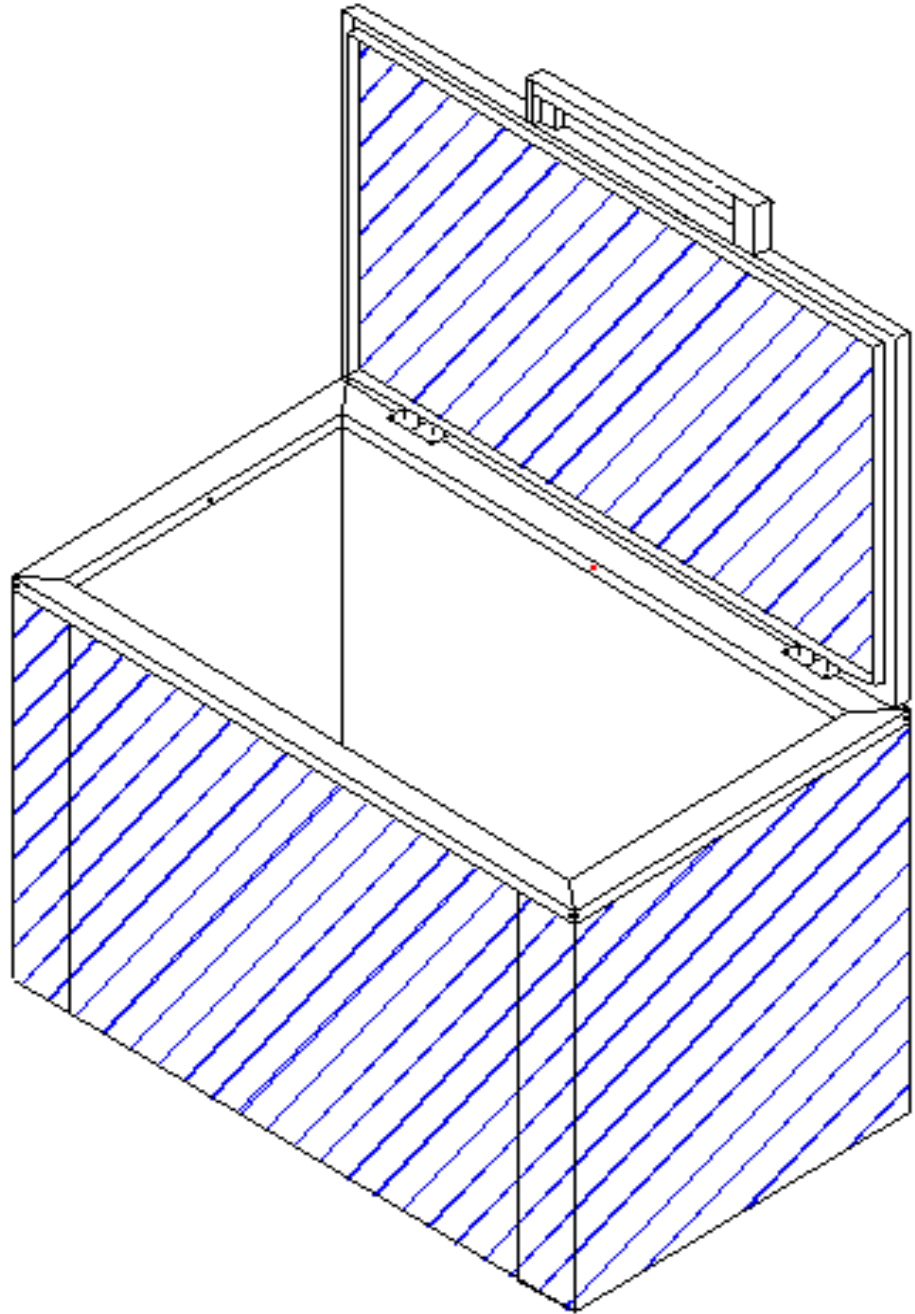
# PART B

## DESIGN OF EQUIPMENT CHAMBER
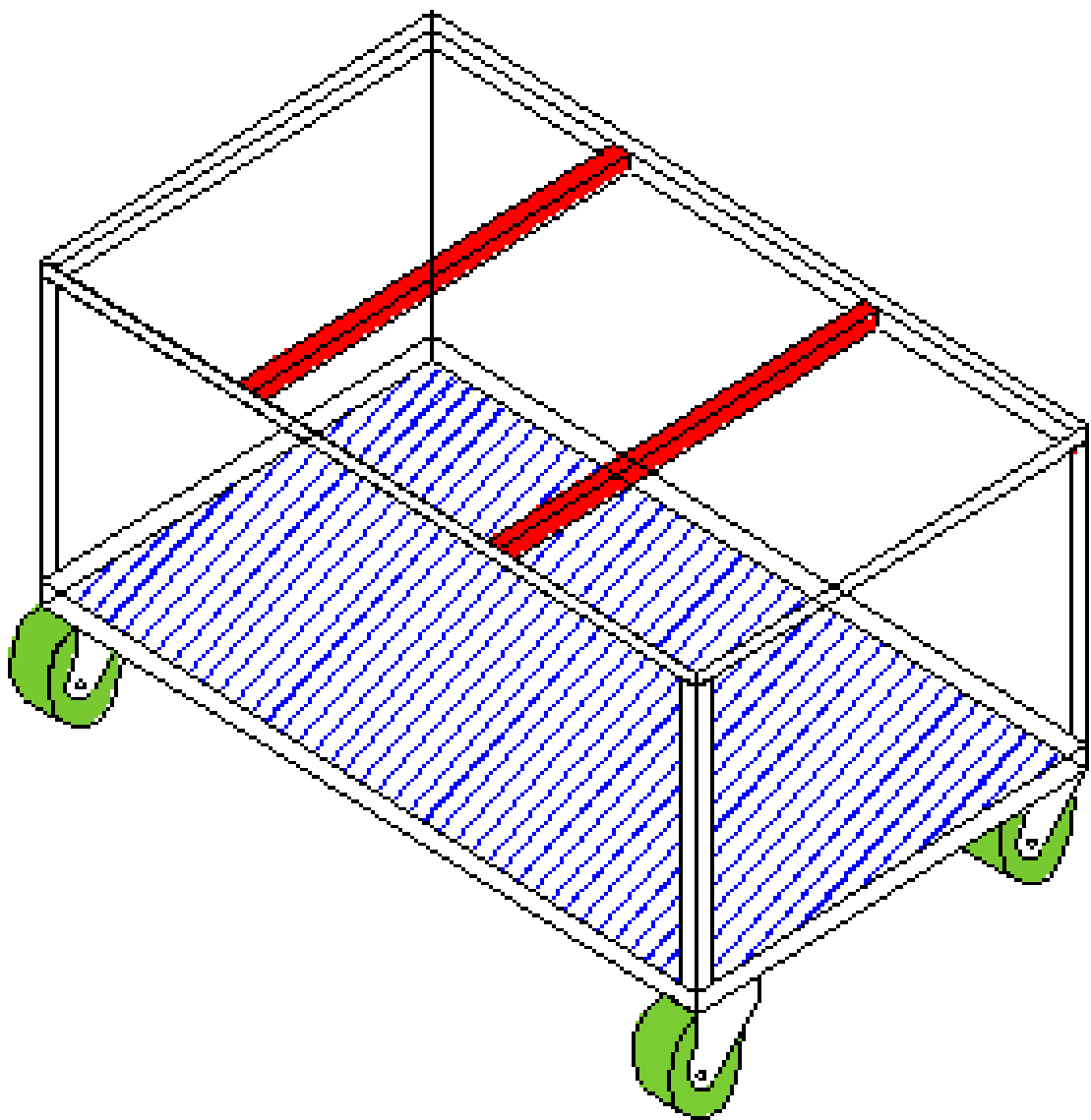
TABLE A-1  maximum and minimum temperature for Hebron city

| Month | Max. Temp. C° | Min. Temp. C° |
|-------|---------------|---------------|
| Jan | 10.3 | 3 |
| Fep | 11.5 | 4.7 |
| Mar | 14.6 | 6.5 |
| Apri | 19.6 | 9.9 |
| May | 25.6 | 13.2 |
| Jun | 26 | 15.8 |
| Jul | 28 | 17 |
| Aug | 38 | 18 |
| Sep | 29 | 15 |
| Oct | 28 | 14 |
| Nov | 22 | 9.9 |
| Dec | 12 | 5.6 |

## TABLE A-2 Thermal Conductivity of Materials

| Material | Description | Thermal Conductivity (k) W/m K | Thermal Conductance (C) W/m$^2$ K |
|---|---|---|---|
| Masonry | Brick, common | 0.72 | |
| | Brick, face | 1.30 | |
| | Concrete, mortar or plaster | 0.72 | |
| | Concrete, sand aggregate | 1.73 | |
| | Concrete block | | |
| | Sand aggregate 100 mm | | 7C.95 |
| | Sand aggregate 200 mm | | 5.11 |
| | Sand aggregate 300 mm | | 4.43 |
| Woods | | | |
| | Maple, oak, similar hardwoods | 0.16 | |
| | Fir, pine, similar softwoods | 0.12 | |
| | Plywood 13 mm | | 9.09 |
| | Plywood 1.9mm | | 6.08 |
| Roofing | Asphalt roll roofing | | 36.91 |
| | Built-up roofing 9 mm | | 17.03 |
| Insulating materials | Blanket or batt, mineral or glass fiber | 0.039 | |
| | Board or slab | | |
| | Cellular glass | 0.058 | |
| | Corkboard | 0.043 | |
| | Glass fiber | 0.036 | |
| | Expanded polystyrene (smooth) | 0.029 | |
| | Expanded polystyrene (cut cell) | 0.036 | |
| | Expanded polyurethane | 0.025 | |
| Loose fill | | | |
| | Milled paper or wood pulp | 0.039 | |
| | Sawdust or shavings | 0.065 | |
| . | Mineral wool (rock, glass, slag) | 0.039 | |
| Glass | Single pane | | 6.42 |
| | Two pane | | 2.61 |
| | Three pane | | 1.65 |
| | Four pane | | 1.19 |
| Metal | Stainless steel | 18 | |
| | Aluminum | 202 | |
| | Cooper | 386 | |
| | Galvanized steel | 20 | |

TABLE A-3 Properties of common foods and substances

| Food | Specific Heat Capacity above Freezing | Specific Heat Capacity below Freezing |
|---|---|---|
| | (KJ/kg$^{o}$C) | (KJ/kg$^{o}$C) |
| Apples | 3.64 | 1.76 |
| Apricots. fresh | 3.68 | 1.8 |
| Avocados | 3.01 | 1.55 |
| water | 4.18 | --- |
| Beef, carcass | 2.85 | 2.01 |
| Beans green | 3.64 | 1.94 |
| Beef, round | 3.1 | 1.59 |
| cheese | 2.68 | 1.51 |
| milk | 3.77 | 2.05 |
| Cucumber | 4.1 | 1.88 |
| oranges | 3.81 | 1.84 |
| Lemon juice | 3.85 | 1.84 |
| eggs | 3.56 | 1.88 |
| Fish | 3.18 | 1.72 |
| Poullry: | 3.18 | 1.55 |

TABLE A-4 Air change per hour

| Kind of room or building | Air Change [times/hr] |
|---|---|
| Room with no windows or exterior door | 0.5 |
| Room with windows or exterior door on one side only | 1.0 |
| Room with windows or exterior door on two side only | 1.5 |
| Room with windows or exterior door on three side only | 2.0 |
| Entrance halls | 2.0 |
| Factories, machine shops | 1.0-1.5 |
| Recreation room, assembly rooms, gymnasium | 1.5 |
| Home, apartment, offices | 1.0-2.0 |
| Class rooms, dining room, lounges, toilets, hospital room, kitchen, laundries, ballrooms, bathrooms | 1.0-2.0 |
| Stores, public buildings | 2.0-3.0 |
| Toilets, auditorium | 3.0 |

TABLE A-5 Specific heat of packaging material

| Packaging Material | Specific Heat [kJ/kg.°C] |
|---|---|
| Wood | 2.3 |
| Stainless steel | 0.5 |
| carton | 0.34 |
| Plastic | 1.6 |
| Aluminum | 0.85 |

Table A-6 Recommended Refrigerant Velocities

| Line | Refrigerant | | Recommended Velocity (m/s) |
|---|---|---|---|
| Suction | R12 | R22 | 8-12 |
| | R404A | R410A | |
| Discharge | R12 | R22 | 10-18 |
| | R404A | R410A | |

Table A-7 Dimensions and Physical Characteristics of Copper Tube ACR (Air -Conditioning and Refrigeration Fields Service).

| 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 |
| | A | .750 | .680 | .035 | .363 | .196 | .178 | .305 | .00252 |
| 3/4 | 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-8  modified correlation of grimson for heat transfer in tube banks

| $\frac{S_p}{d}$ | $\frac{S_n}{d}$ 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 | 0.317 | 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 | 0.542 | 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-9 Properties of air atmospheric pressuer

| $T$ (K) | $\rho$ (kg/m³) | $c_p$ (J/kg·K) | $\mu$ (kg/m·s) | $\nu$ (m²/s) | $k$ (W/m·K) | $\alpha$ (m²/s) | Pr |
|---|---|---|---|---|---|---|---|
| | | | | Air | | | |
| 100 | 3.6010 | $1.0266 \times 10^3$ | $0.6924 \times 10^{-5}$ | $1.923 \times 10^{-6}$ | 0.009 246 | $0.0250 \times 10^{-4}$ | 0.768 |
| 150 | 2.3675 | 1.0099 | 1.0283 | 4.343 | 0.013 735 | 0.0574 | 0.756 |
| 200 | 1.7684 | 1.0061 | 1.3289 | 7.490 | 0.018 09 | 0.1016 | 0.739 |
| 250 | 1.4128 | 1.0053 | 1.5990 | 11.310 | 0.022 27 | 0.1568 | 0.722 |
| 300 | 1.1774 | 1.0057 | 1.8462 | 15.690 | 0.026 24 | 0.2216 | 0.708 |
| 350 | 0.9980 | 1.0090 | 2.075 | 20.76 | 0.030 03 | 0.2983 | 0.697 |
| 400 | 0.8826 | 1.0140 | 2.286 | 25.90 | 0.033 65 | 0.3760 | 0.689 |
| 450 | 0.7833 | 1.0207 | 2.484 | 31.71 | 0.037 07 | 0.4636 | 0.683 |
| 500 | 0.7048 | 1.0295 | 2.671 | 37.90 | 0.040 38 | 0.5564 | 0.680 |
| 550 | 0.6423 | 1.0392 | 2.848 | 44.27 | 0.043 60 | 0.6532 | 0.680 |
| 600 | 0.5879 | 1.0551 | 3.018 | 51.34 | 0.046 59 | 0.7512 | 0.682 |
| 650 | 0.5430 | 1.0635 | 3.177 | 58.51 | 0.049 53 | 0.8578 | 0.682 |
| 700 | 0.5030 | 1.0752 | 3.332 | 66.25 | 0.052 30 | 0.9672 | 0.684 |
| 750 | 0.4709 | 1.0856 | 3.481 | 73.91 | 0.055 09 | 1.0774 | 0.686 |
| 800 | 0.4405 | 1.0978 | 3.625 | 82.29 | 0.057 79 | 1.1951 | 0.689 |
| 850 | 0.4149 | 1.1095 | 3.765 | 90.75 | 0.060 28 | 1.3097 | 0.692 |
| 900 | 0.3925 | 1.1212 | 3.899 | 99.3 | 0.062 79 | 1.4271 | 0.696 |
| 950 | 0.3716 | 1.1321 | 4.023 | 108.2 | 0.065 25 | 1.5510 | 0.699 |
| 1000 | 0.3524 | 1.1417 | 4.152 | 117.8 | 0.067 52 | 1.6779 | 0.702 |
| 1100 | 0.3204 | 1.160 | 4.44 | 138.6 | 0.073 2 | 1.969 | 0.704 |
| 1200 | 0.2947 | 1.179 | 4.69 | 159.1 | 0.078 2 | 2.251 | 0.707 |
| 1300 | 0.2707 | 1.197 | 4.93 | 182.1 | 0.083 7 | 2.583 | 0.705 |
| 1400 | 0.2515 | 1.214 | 5.17 | 205.5 | 0.089 1 | 2.920 | 0.705 |
| 1500 | 0.2355 | 1.230 | 5.40 | 229.1 | 0.094 6 | 3.266 | 0.705 |
| 1600 | 0.2211 | 1.248 | 5.63 | 254.5 | 0.100 | 3.624 | 0.705 |
| 1700 | 0.2082 | 1.267 | 5.85 | 280.9 | 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 | 5.680 | 0.703 |
| 2200 | 0.1602 | 1.419 | 6.93 | 432.6 | 0.139 | 6.115 | 0.707 |
| 2300 | 0.1538 | 1.482 | 7.14 | 464.0 | 0.149 | 6.537 | 0.710 |
| 2400 | 0.1458 | 1.574 | 7.35 | 504.0 | 0.161 | 7.016 | 0.718 |
| 2500 | 0.1394 | 1.688 | 7.57 | 543.0 | 0.175 | 7.437 | 0.730 |

TABLE A-10 ratio of N rows deep to that for 10rows 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 | 0.8 | 0.87 | 0.9 | 0.92 | 0.94 | 0.96 | 0.98 | 0.99 | 1.0 |

TABLE A-11 Respiration gain

| Product | temperature | Watts per kilogram |
|---|---|---|
| Orange | 5°C | 0.019 |
| Green beans | 5°C | 0.09 |