Palestine Polytechnic University

Collage of Engineering

Mechanical Engineering Department

Hebron – Palestine

Design and Building of A Chamber To Examine The Effect of Temperature on Properties of Stone & Marble

By

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Submitted to the Collage of Engineering

In partial fulfillment of the requirements for the

Bachelor degree in Mechatronics Engineering.

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August 2017

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Produced by

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Hebron – Palestine

2016-2017

Dedication

To our God………………. For his guidance

To our Teachers ……......... For help us until the beauty end

To our friends ………………. Who give us Positive sentiment & Support

To the family - father, mother, wife's, sons, who have patience in order to get to what we have

To our great Palestine

To our supervisor Eng . Kazem Osaily

To all who made this work is possible & well done .

Acknowledgement

We would like to express our gratitude for everyone who helps us during the graduation project , starting with endless thanks for our supervisor Eng . Kazem Iseili who didn't keep any effort in encouraging us to do a great job , providing our team with valuable information and advices to be better each time .

Thanks for the continuous support and kind communication which great effect regarding to feel interesting about what we are working on .

Thanks are extended to the " Air conditioning and Refrigeration society " for the beneficial lecture provided , and who helped us with them useful lectures , which support our work .

Also our thanks are extended to all instructors and engineers who helped us during the first stages of our graduation project .

Finally , our ultimate thanks go to great edifice of science (Palestine Polytechnic University) for their endless support and their huge effort in providing us with all what we need to build this project .

Abstract

Stone and marble sector is one of the most important industrial and commercial sectors in Palestine since the area is rich with this high quality natural resource. It was given the name of (white gold) for its weight in the general national product and economic returns.

Stone is one of the basic elements in any building for its physical properties and it significant impact in high lighting the architectural and aesthetic shape of building .

The aim of this project is to serve the stone and marble sector in terms of finely building a device that simulate the changing climatic condition stone and marble exposed to (such as temperature) .

As this device is commensurate with global standers in the application of thermal testing of stone and marble .

The system includes refrigeration cycle (heat pump), insulated room with a special thermal specifications , in addition to complete easy to use conditional system which includes a provision for testing standard condition .

It is hoped that this project will serve the practical side of the stone and marble sector , through the provision of this device to conduct tests on samples of different types of stone and marble available in the local market and go out with data tables that define the thermal characteristics of each type of stone and marble .

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Chapter One

Introduction

1.1. Overview

Importance of quality of the stone on the shape and architectural design of buildings, and lack of laboratories or research centers in Palestine that are interested in the study of thermal effects on the stones came this project

Thermal movement occurs when a change occurs in the heat an expansion or contraction of the components of the construction, the main problem arises during differential movement between adjacent and different materials.
Facing all the construction materials thermal movement; however, variable expansion coefficient between the material and therefore the actual movement which is important for buildings also varies .

A number of factors affect the amount of thermal movement occurs in the component or element , Results from to instability of temperature or differential temperature when exposed to sunlight and shadow periods.

Failure due to kinetic theory in articles mechanism depends on the rate of change and movement differential between the colored and dark .dark surfaces absorb more of the components of the colored surfaces heat.

Factors affecting the degree of influence of the thermal motion contain temperature range, degrees of temperature differential and color and installation background, thermal inertia generally, durability and hardness constituents and surrounding structures.

1.2. Project Idea

The project idea emerges to design and build thermal test unit, due to the market needs for this application especially at stone & marble field .

Project discuses the following:

1- Show the effect of temperature difference due to exposure time of stone.
2- Thermal analysis of the cycle .
3- Design of the chamber .
4- Selection of the cycle components .
5- Assembling the cycle components and get ready for testing .
6- Testing the cycle .
7- Sample Testing .

1.3. Project Importance

The significance of the project is the design of the unit serving the stone and marble field , specifically to determine and adjust the quality of the stone when exposed to variable temperatures and difference between night and day and its impact on a stone properties and dimensions.

1.4. Project Objectives

1- Design and building of Refrigeration chamber Operating at - 20 ^{o}C .

2- Community assistance to help stone and marble sector and technical specifications data for different varieties of stone available in the Palestinian market.

3- Support the stone & marble center by this unit , low operating cost, and easy for use . figure (1.2) .

1.5. Simple refrigeration cycle

Simple refrigeration cycle is a simple vapor compression refrigeration system figure (1.1) , in which suitable working refrigerant .

It condenses and evaporates at a temperature and pressure close to atmospheric condition .

It can be used ordinary application of refrigeration and it gives a high coefficient of performance.

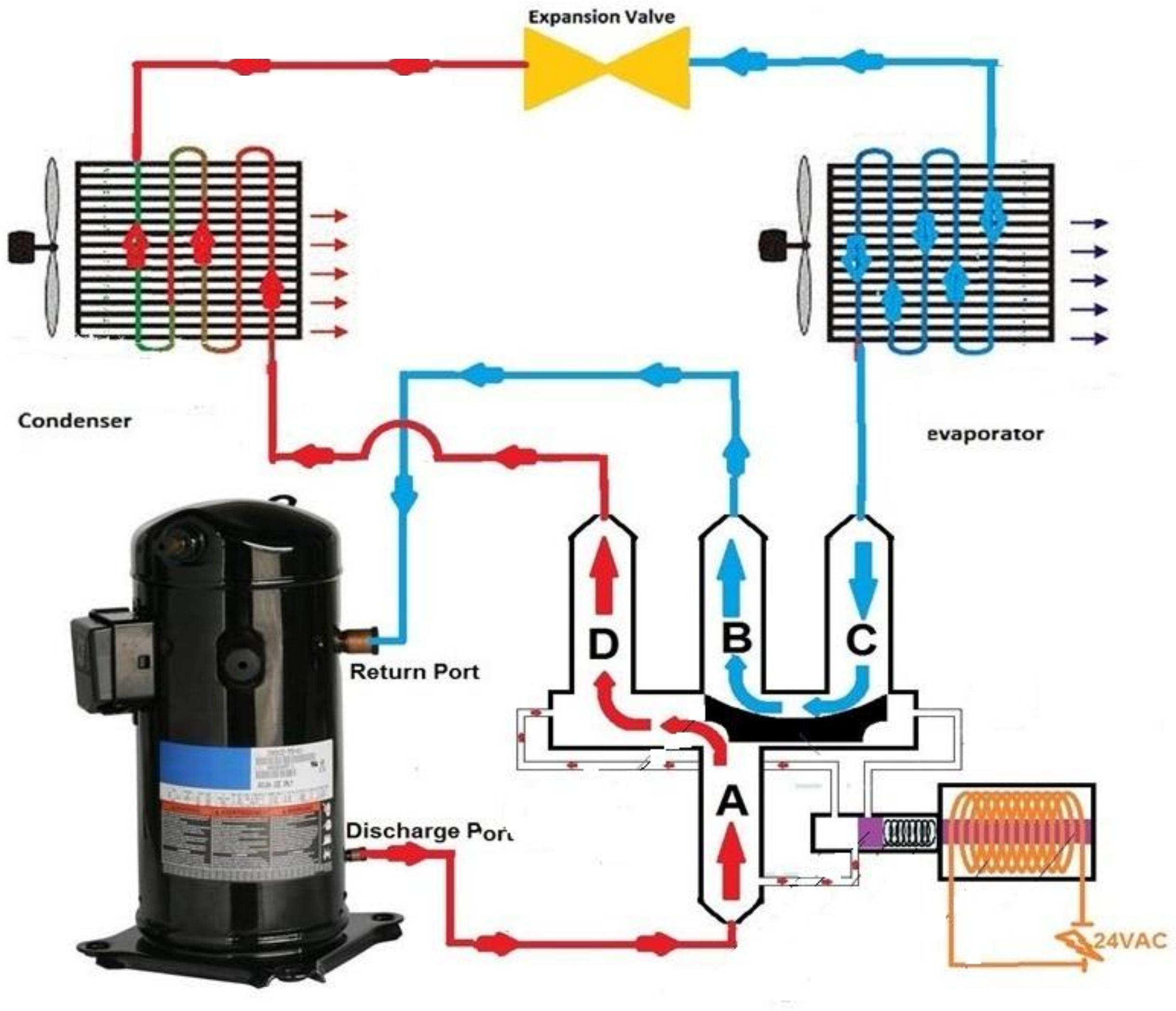

Figure (1.1) Vapor Compression Refrigeration System

Sometimes , the vapor refrigerant is required to be delivered at a very high pressure as in the case of low temperature refrigeration systems , in such cases the vapor refrigerant must be compressed by employing two or more compressors placed in series .

Figure (1.2) Thermal test unit

CHAPTER 2

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 a 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 satisfactory inside conditions .

2.2 load sources

The cooling load seldom results from any one single source of heat .
Rather it is the summation of the heat which usually evolves from several different sources . some of the more common sources of heat that impose the load on refrigerating equipment are :

1- The wall heat gain .
2- The product heat gain .
3- Infiltration heat gain .
4- Packing heat gain .
5- Defrosts heater heat gain .
6- Fan motor heat gain .

Design data of Chamber :(fig 2.1)

Inside design temperature
Storage temperature is $-20\ ℃$.
Outside design
Surrounding Temperature is $35\ ℃$.
Mass of the product $\approx 2.7\ kg$
Cooling time is 12 hours .
Internal Chamber Dimensions (0.6 , 06 , 0.6) meter .
External Chamber Dimensions (0.8 , 0.8 , 0.8) meter.
External Chamber size = 0.6 * 0.6 * 06 = 216 liter .
External Chamber size = 0.8 * 0.8 * 0.8 = 512 liter .
Effective Chamber size = 216 liter

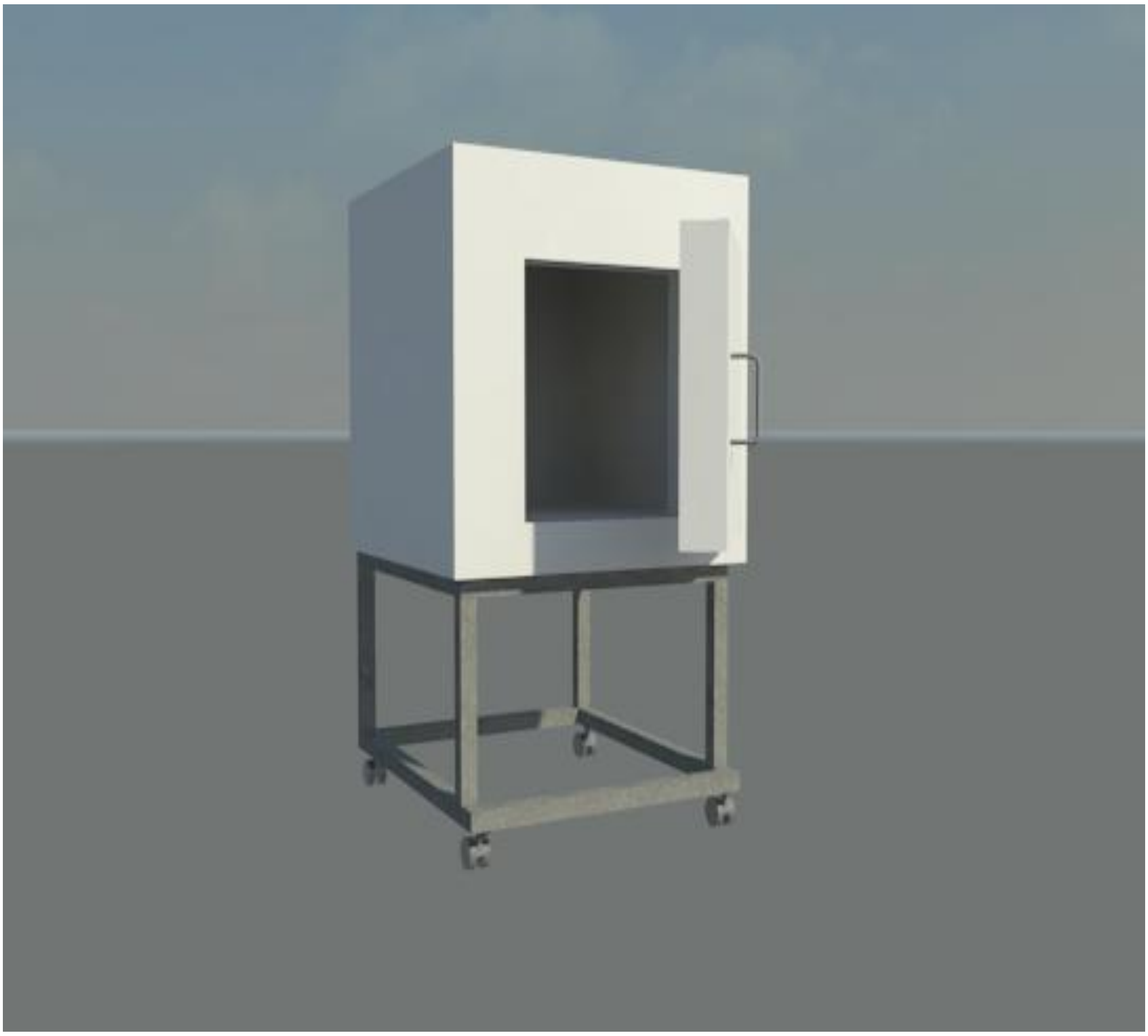

Figure (2.1) chamber design -ref 19

2.2.1 The wall heat gain

The wall heat gain load , sometimes called the leakage load , is a measure of the heat flow rate by conduction through the walls of the refrigerated space from the outside to the inside . since there is no perfect insulation , there is always a certain amount of heat passing from the outside to the inside whenever the inside temperature is below than the outside .
the wall gain load is common to all refrigeration application and is ordinary a considerable part of the total cooling load, commercial storage coolers and residential air conditioning applications are both examples of applications where in the wall gain load often accounts for the greater portion of the total load .

$$Q_{wall} = U * A * \Delta T \quad \text{……………………………………………………….……(2.1)}$$

Where :

A : Outside surface area of the wall [m^2]

U : Overall heat transfer coefficient [$W/_{m^2.°C}$]

ΔT: Temperature differences across the walls [°C]

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

Overall heat transfer coefficient is computed by the following :

$$U = \frac{1}{\frac{1}{h_i}+\frac{\Delta x1}{k_1}+\frac{\Delta x2}{k_2}+\cdots+\frac{1}{h_o}} \quad \text{…………………………………..…………………….…(2.2)}$$

Where :

U : the overall heat transfer coefficient [$W/_{m^2.°C}$]

Δx : the thickness of the layer of the wall [m] .

k : the thermal conductivity of the material [$W/_{m.°C}$] .

h_i: the convection heat transfer coefficient of inside air [$W/_{m^2.°C}$] .

Forced convection by using fan (30 – 100) , taken 60[$W/_{m^2.°C}$] .

h_o : the convection heat transfer coefficient of outside air [$W/_{m^2.°C}$] .

Free convection inside the room (5- 20) , taken 12 [$W/_{m^2.°C}$] .

All walls are constructed of three layers as shown in Figure (2.2) .

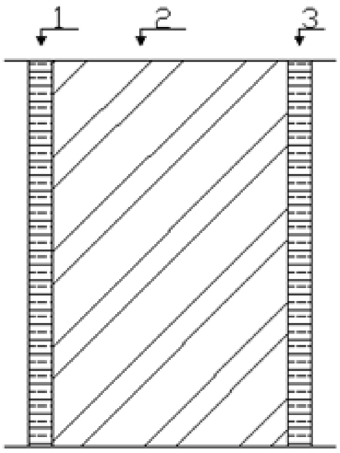

Figure (2.2) Chamber wall layer

Material	**Thinness**	**Thermal conductivity**	**Reference**
Galvanized steel	0.1 [cm]	k = 15.60 [$W/_{m.}$℃]	From Table A-1
Polyrethane	10 [cm]	k = 0.036 [$W/_{m.}$℃]	From Table A-1
Galvanized steel	0.1 [cm]	k = 15.60 [$W/_{m.}$℃]	From Table A-1

$$U = \frac{1}{\frac{1}{60}+\frac{0.001}{15.6}+\frac{0.1}{0.036}+\frac{0.001}{15.6}+\frac{1}{12}} = 0.35 \ [W/_{m^2.} ℃]$$

$Q_{floor\,\&\,roof}$ = 0.35 * (0.8 * 0.8) * (35 – (-20)) = 12.32 W
$Q_{tow\ sides}$ = 0.35 * (0.8 * 0.8) * (35 – (-20)) = 12.32 W
$Q_{front\,\&\,back}$ = 0.35 * (0.8 * 0.8) * (35 – (-20)) = 12.32 W

$Q_{all\ walls}$ = 2 * ($Q_{floor\,\&\,roof}$ + Q_{sides} + $Q_{front\,\&\,behind}$)
$Q_{all\ walls}$ = 2 * (12.32 + 12.32 + 12.32) = 73.92 [W]

2.2.2 Product heat gain

The heat emitted from the sample to be stored is very important in case of thermal testing . The loads to be considered in the thermal testing are divided into the following groups :

1- Chilling load freezing : The product cooling freezing depends upon the mass product , mean specific heat of the products freezing , entering product temperature , final product temperature desired , and the cooling time . this heat gain can be calculated by the following equation:

Q_{ch} = m . c_p . ΔT

Where :

Q_{ch} : Cooling product load in [kJ]

m : mass of the product in [kg]

C_p : the specific heat of freezing in [kJ / kg . ℃]

ΔT : $T_{out} - T_{ch}$

Where :

T_o : entering product temperature [℃]

T_{ch} : chilling product temperature [℃]

C_p = 0.9 [kJ / kg . ℃] (Table A-2)

T_o = 35 [℃]

T_{ch} = -2 [℃]

Q_{ch} = 2.7 * 0.9 * (35 – (-2))

Q_{ch} = 89.91 [kJ]

2- Cooling load below freezing : the cooling load below freezing depends upon the mass of product , desired freezing temperature of the product (refrigerated space temperature) , and the cooling time . [reference 5]

Q_c = m . C_p . ΔT

Where :

Q_c : cooling load in [kJ]

m : mass of the product load in [kg]

Q_p : the specific heat below freezing in [kJ / kg / ℃]

ΔT : ($T_{ch} - T_{Rs}$)

Where :

T_{ch} : freezing product temperature [℃]

T_{Rs} : refrigerated space temperature [℃]

C_p = 0.90 [kJ / kg . ℃] (table A – 1) .

T_{ch} = -2 [℃] table A -2

T_{Rs} = - 20 [℃]

Q_c = m . C_p . ΔT

Q_c = 2.7 * 0.90 * (-2 – (- 20))

Q_c = 43.74 [kJ]

3- Freezing load : the freezing load depends upon the mass of product , its latent heat of freezing , and the freezing time . [reference 5]

Q_f = m . H_L

Where :

Q_f : freezing load [kJ]
H_L : latent heat for the product [kJ / kg]
H_L = 2255 [kJ / kg]
Q_f = 2.7 * 2255
Q_f = 6088.5[kJ]

Total product load

$Q_p = (Q_{ch} + Q_c + Q_f) / t$

Where :

t : desired cooling time in [seconds]

Q_p : ($\frac{(89.91+43.74+6088.5)}{12*(3600)}$) * 1000

Q_p = 144 [W]

2.2.3 Infiltration heat gain

In the practical operation of 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 temperature difference between the inside and outside air , cooler dimensions .

Q_{inf} = m . C_p . (T_o - T_i)

$Q_{inf} = \rho * V_f * C_p * (T_o - T_i)$

Where :

ρ : air density [1.25 kg / m^3]

C_p : the specific heat of the air [1000 J / kg . ℃]

V_f : the volumetric flow rate of infiltrated air [m^3/s]

T_o : the outside temperature [℃]

T_i : the inside temperature [℃]

V_f = number of air change * volume of room

Number of air change = 0.5 [times / h] table A-3

Volume of room = 0.4* 0.4 * 0.6 = 0. 096 m^3

$V_f =$ 0. 096 * 0.5 = 0.048 m^3/ hr

Q_{inf} = 1.25 * (0.048 / 3600) * 1000 * (35 - -20)

Q_{inf} = 1 [W]

2.2.4 Fan motor heat gain

The evaporator fan motor releases heat , this heat is relatively equal the power of the motor

Q_{motor} = power of motor = 25 [W]

2.2.5 Total cooling load

The total cooling load is the summation of the heat gains

$Q_T = Q_w + Q_p + Q_{inf} + Q_{motor}$

Q_T = 73.92 +144 + 1+ 25

Q_T = 243.92 W

Add 40 % as safety of factor

Total cooling load = Q_T * 1.4

Total cooling load = 243.92 * 1.4 = 341.5 [W]

Chapter Three

Components Of Refrigeration System

3.1. Introduction

There are several mechanical components required in a refrigeration system (heat pump system) .

This part of project discuss the major components of the system and some auxiliary equipment working with these major components to get a high efficacy system .

The major components of the refrigeration system are as follows :

- Compressor.
- Condenser.
- Throttling device .
- Evaporator .

3.2. Compressor

This device has two purposes, due to the fact that it separates the low-pressure side of the system from the high-pressure side.

The compressor first removes vapor from the evaporator to keep the evaporator’s boiling point low. Next, the device compresses the low-temperature gas into a small volume, creating a high-temperature, high-pressure gas In refrigeration cycle , the compressor has two main functions within the refrigeration cycle.

One of this 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 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.

Refrigerant compressors, which are knows as the heart of the refrigeration system, can be divided into three main categories: [reference 8]

- Hermetic compressor
- Simi hermetic compressor
- Open compressor

In this project we will use hermetic compressor is to be use.

3.2.1. Hermetic Compressor

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 refrigerator, freezers, and air conditioners are of the hermetic type . Their revolutions per minute are either 1450 or 2800 rpm .

Hermetic compressors can work for a long time in small capacity refrigeration system 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 (3.1) shows hermetic compressor .

Figure (3.1) Hermetic Compressor

The refrigerant compressors are expected to meet the following requirements high reliability, long service life, easy maintenance, quiet operation, compactness, and low cost .

3.3. Condensers

The condenser changes the state of the superheated refrigerant vapor back into a liquid. This is done by creating a high pressure that raises the boiling point of the refrigerant and removes enough heat to cause the refrigerant to condense back into a liquid .

A condenser is a major system component of refrigeration system. It also an indirect contact heat exchanger in which the total heat rejected from the refrigerant is removed by cooling medium, usually air or water. As a result, the gaseous refrigerant is cooled and condensed to liquid at the condensing pressure .

3.3.1. Air cooled condensers

The air cooled condensers find application in domestic, commercial, and industrial refrigerating, chilling, freezing, and air conditioning systems, the centrifugal fan are used in the condenser particularly for heat recovery and auxiliary ventilation applications. In fact, they employ outside air as cooling medium [reference 9].

Fans draw air past the refrigerant coil and the latent heat of the refrigerant is removed as sensible heat by the air stream. Figure (3.2) shows air cooled condensers .

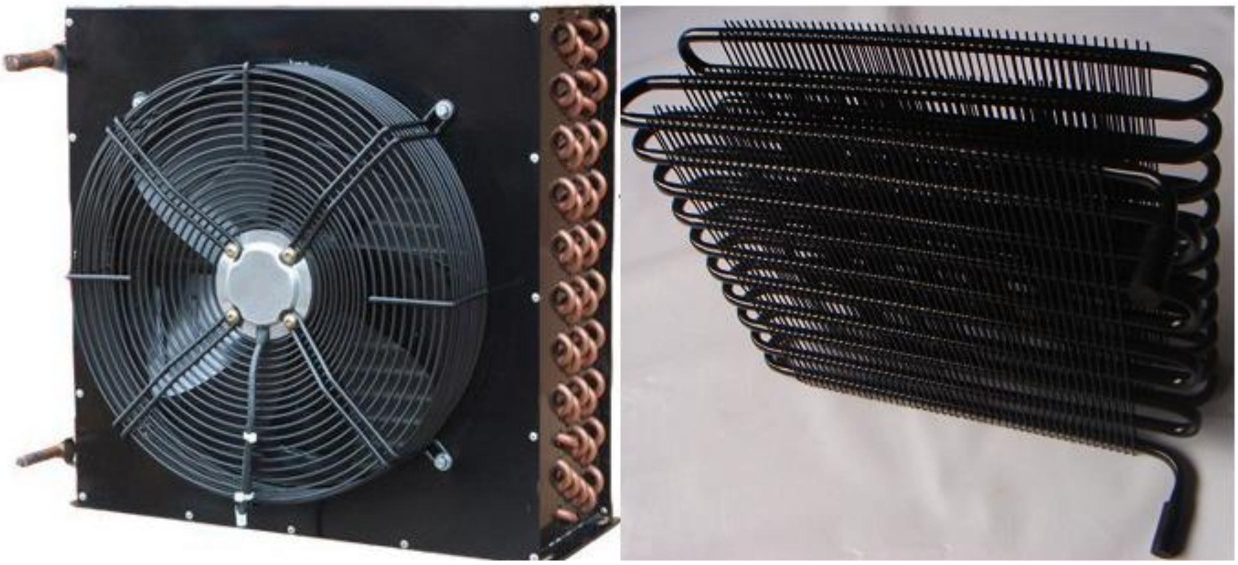

Figure (3.2) Air Cooled Condensers

3.4. Evaporator

Evaporator (Figure 3.3) is an important device used in the low pressure side of a refrigeration system .

The liquid refrigerant from the expansion valve enters into the evaporator where it boils and changes into vapor .

The function of an evaporator is to absorb heat from the surrounding location or medium which is to be cooled, by means of refrigerant .

The temperature of the boiling refrigerant in the evaporator must always be less than the temperature of the surrounding medium so that the heat flows to the refrigerant.

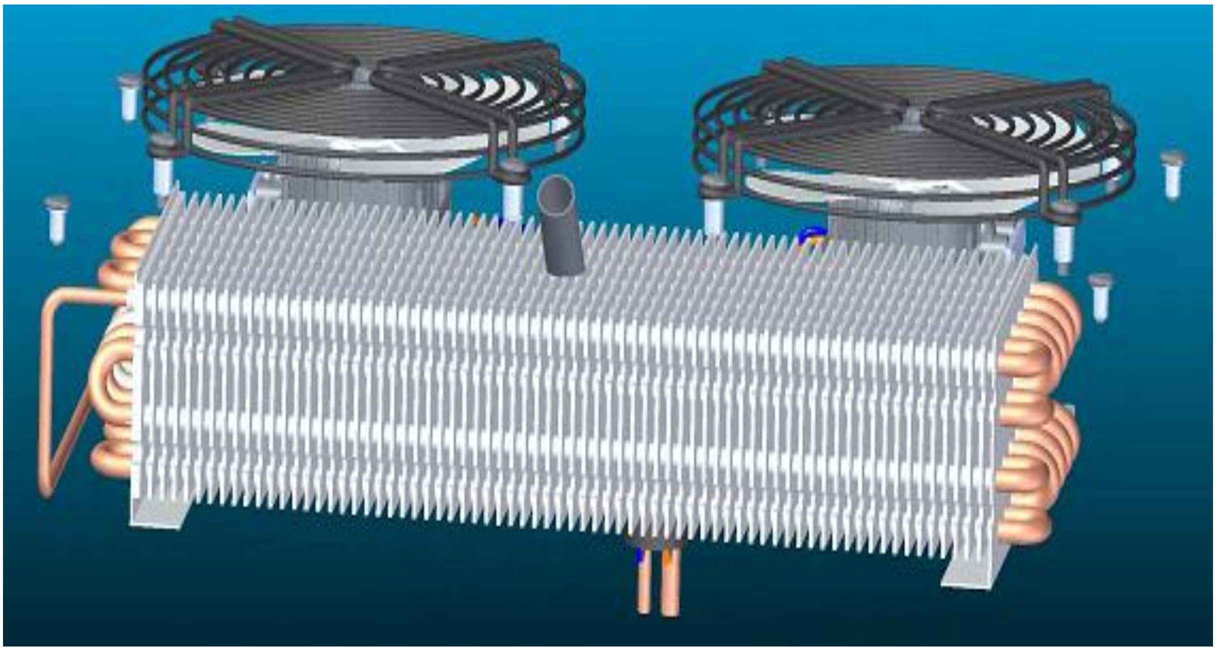

Figure (3.3) Evaporator

3.4.1. Bare Tube Coil Evaporator

The simplest type of evaporator is the bare tube coil evaporator, as shown in figure 3.4. The bare tube coil evaporator are also known as prime surface evaporators. Because of its simple construction, the bare tube coil is easy to clean and defrost.

A little consideration will show that this type of evaporator, offers relatively little surface contact area as compared to other types of coils.

The amount of surface area may be increased by simply extending the length of the tube, but there are disadvantages of excessive tube length. The effective length of the tube is limited by the capacity of expansion valve.

If the tube is too long for the valve's capacity, the liquid refrigerant will tend to completely vaporize early in its progress through the tube, thus leading to excessive superheating at the outlet. The long tubes will also cause considerably greater pressure drop between the inlet and outlet of the evaporator . This results in reduced suction line pressure.

The diameter of the tube in relation to tube length may also be critical. If the tube diameter is too large, the refrigerant velocity will be too low and the volume of refrigerant will be too great in relation to the surface area of the tube to allow complete vaporization. This, in turn, may allow liquid refrigerant to enter the suction line with possible damage to the compressor (slugging).

One the other hand, if the diameter is too small, the pressure drop due to friction may be too high and will reduce the system efficiency. The bare tube coil evaporators may be used for any type refrigeration requirement .

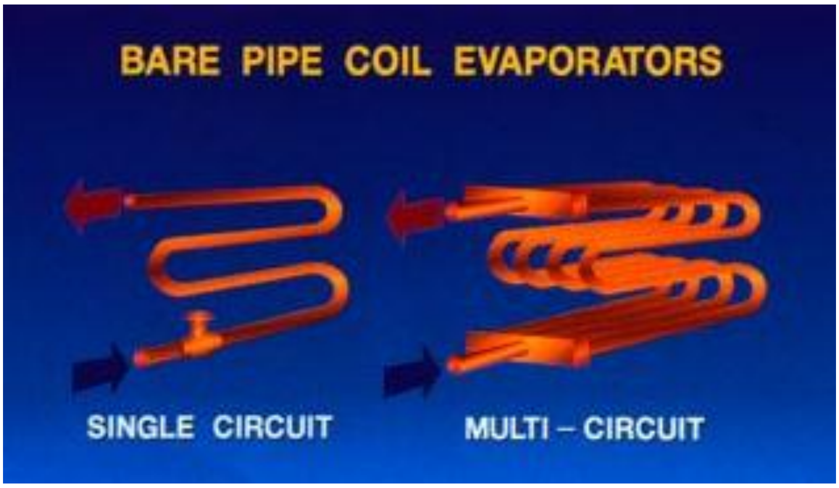

Figure (3.4) Bare Tube Coil Evaporator

3.5. Throttling Devices

In practice, throttling devices, called either expansion valves or throttling valves, are used to reduce the refrigerant condensing pressure (high pressure) to the evaporating pressure (low pressure) by a throttling operation and regulate the liquid refrigerant flow to the evaporator to match the equipment and load characteristics. These devices are designed to proportion the rate at which the refrigerant enters the cooling coil to the rate of evaporation of the liquid refrigerant in the coil , the amount depends – of course – on the amount of heat being removed from the refrigerated space .

The most common throttling devices are as follows :

- Thermostatic expansion valves,
- Constant pressure expansion valves,
- Float valves, and
- Capillary tubes .

Note that a practical refrigeration system may consist of large range of mechanical and electronic expansion valves and other flow control devices for small and large scale refrigeration system, comprising thermostatic expansion valves, solenoid valves and thermostats, modulating pressure regulators, filter driers, liquid indicators, non return valves and water valves, and furthermore, decentralized electronic systems for full regulation and control .

3.5.1. Capillary tubes

The capillary tube is the simplest type of refrigerant flow control device and its shown is (figure 3.5) and may be used in place of an expansion valve.

The capillary tubes are small diameter tubes through which the refrigerant flows into the evaporator.

These devices, reduce the condensing pressure to the evaporating pressure in a copper tube of small internal diameter (0.4 – 3 mm diameter and 1.5 – 5 m length), maintaining a constant evaporating pressure independently of the refrigeration load change.

A capillary tube may also be constructed as a part of a heat exchanger, particularly in household refrigerators .

With capillary tubes, the length of the tube is adjusted to match the compressor capacity. Other consideration in determining capillary tube size include condenser efficiency and evaporator size. Capillary tubes are most effective when used in small capacity systems .

Figure (3.5) Capillary Tube

3.6 Auxiliary Components

The auxiliary components are very important in the refrigeration system, their working together with main components allowing system works very well, and some auxiliaries are discussed in the following sections.

3.6.1 Oil separator

Oil separator provide oil separation and limit oil carry over to approximately 0.0003 – 0.001% of the total amount of refrigerant, depending on various system characteristics, note that all the separators require the mounting of an external float assembly to control return from the separator to the compressor .

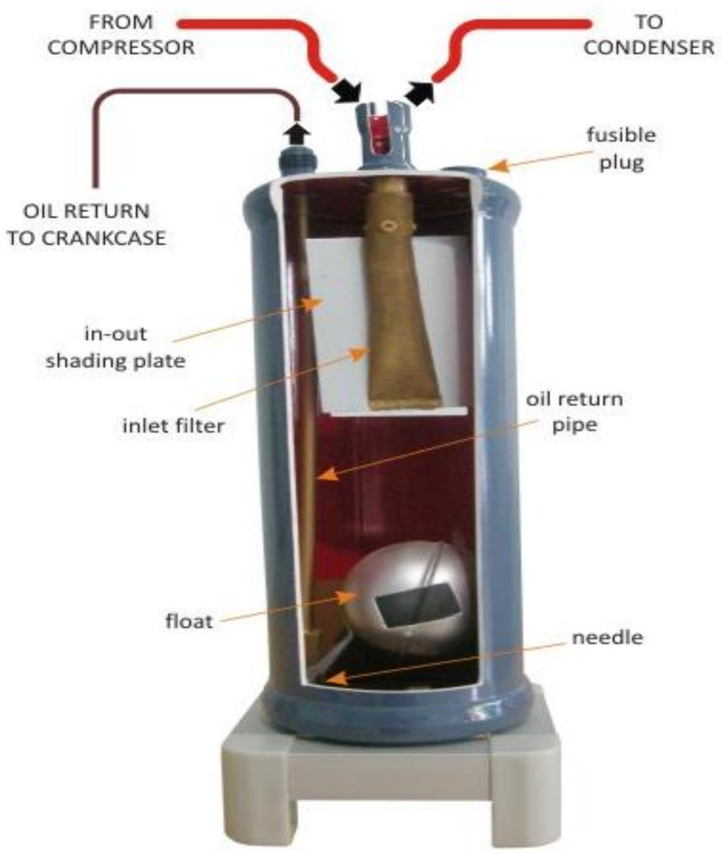

Figure (3.6) Oil separator

3.6.2. Low Pressure And High Pressure Control

The purpose of low pressure control is to stop the compressor when the suction pressure drops below a preset value or when the refrigerant flow rate is too low to cool the compressor motor. Figure 3.6 (a) shows a typical low pressure control mechanism.

When the suction pressure falls below a certain limit, the spring pushes the blade downward, opens the motor circuit, and stops the compressor.

When the suction pressure increases the bellows expand, thus closing the contact of the motor circuit and restarting the compressor.

The two adjusting screws are used to set the cut-out and cut-in pressures

Cut-out pressure is the pressure at which the compressor stops, and the cut-in pressure is the pressure at which the compressor starts again.

The purpose of high pressure control is to stop the compressor when the discharge pressure of the hot gas approaches a dangerous level. Figure 3.6 (b) shows a typical high pressure control mechanism. If the discharge pressure reaches a certain limit, the bellows expand so that the bland opens the motor circuit contact and the compressor stops.

When the discharge pressure drops to the safe level, the bellows contract and close the contact, and the compressor start again.

As in a low pressure control, two adjusting screws are used to set the cut-out and cut-in pressures. In small refrigeration systems, low pressure and high pressure controls are often combined to form a dual pressure control.

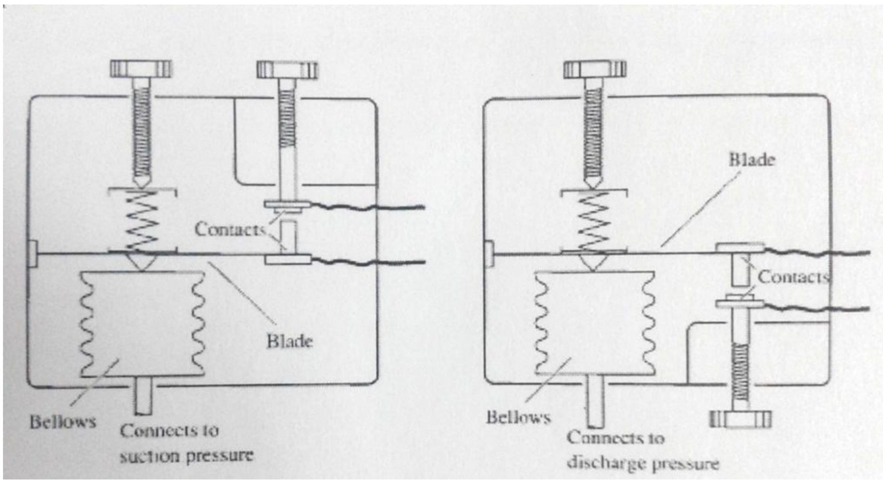

Figure (3.6 - a) low pressure control (3.6-b) high pressure control

Figure (3.6) Low and high pressure control

3.6.3. Thermostat

A thermostat is a component of a control system which senses the temperature of a system so that the system's temperature is maintained near a desired *set point*.
The thermostat does this by switching heating or cooling devices on or off, or regulating the flow of a heat transfer fluid as needed, to maintain the correct temperature. The name is derived from the Greek words thermos "hot" and status "a standing".

Thermostat used in refrigeration system to senses the evaporating temperature. If this temperature reach the required temperature, the thermostat shut off the compressor, and when this temperature increase again it turn on the compressor .

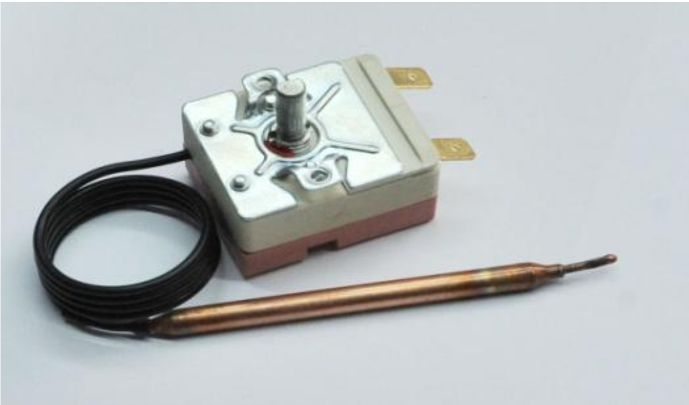

Figure (3.7) Thermostat

3.6.4. Four Way Valve

A heat pump reversing valve is an electro-mechanical 4-way valve that reverses the refrigerant (Freon) flow direction, using an electrical magnet.
When you energize the coil with electricity the coil becomes magnetized, pulling a pin and compressing the spring.
This action opens the valve to let the Freon flow.

When you de-energize the coil the electrical magnet loses its magnetic power, and the compressed spring expands and pushes the pin back to shut off the Freon. it also has:-
capillary tube (1), capillary tube (2), capillary tube (3), slider (4), block (5) ,electrical coil (6), electrical magnet (7), and spring (8) as shown in figure (3.10).

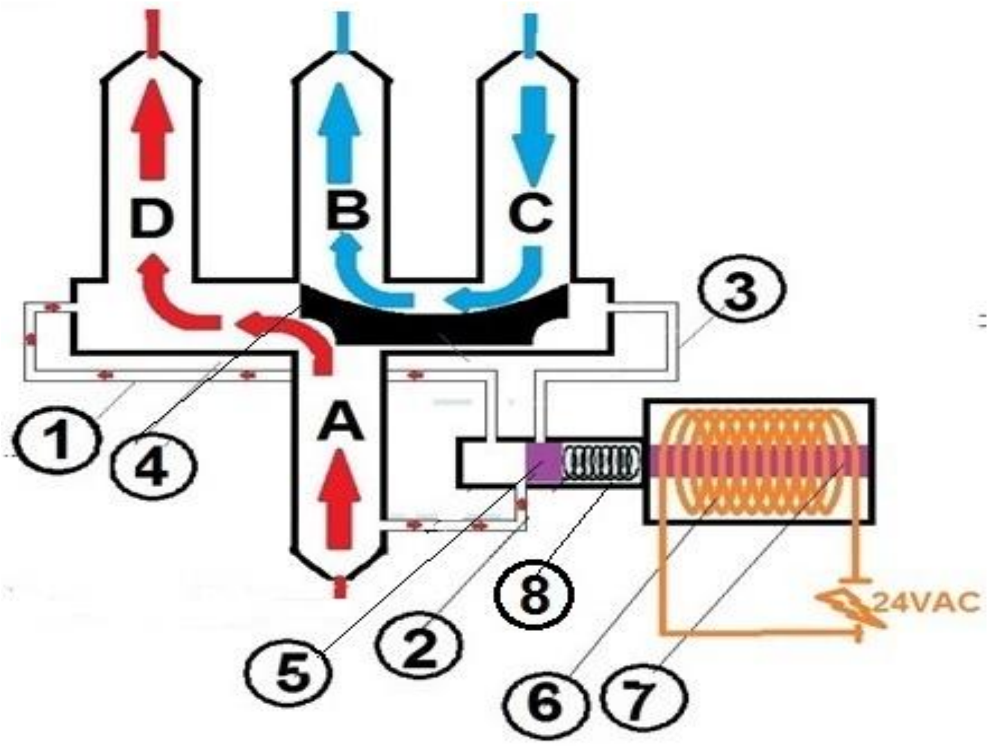

Figure (3.8) Four Way Valve

CHAPTER 4

CYCLE ANALYSIS

4.1 Refrigerant selection

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

- Low boiling point .
- High critical temperature .
- High latent heat of vaporization .
- Low specific volume of vapor .
- Non corrosive to metal .
- Non flammable and non explosive .
- Non toxic.
- Low cost.
- Easy to liquefy at moderate pressure and temperature .
- Easy of locating leaks by suitable indicator .
- Mixes well with oil .

To select refrigerants successfully we must consider the above properties . We made a comparison between various refrigerants and found that , the best refrigerants could be used are R-134a for the cycle . In addition to above properties in selection process of R-134a we have to determine the compressor type that works perfectly with it, and it has to be cheap and available . [reference 1].

4.2 Cycle Analysis

Figure 4.1 describes Ideal refrigeration cycle .

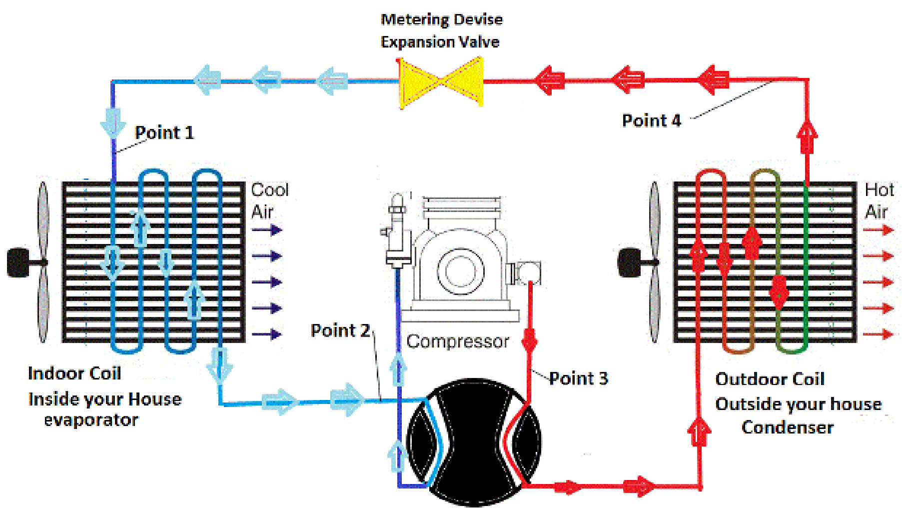

Figure (4.1) Refrigeration Cycle

4.2.1 Calculation Using R-134a

$Q_e = \dot{m}.\ q_e$ ……………………………………..…………..(4.1)

$Q_e = \dot{m} \times (h_1 – h_4)$ ……………………………………..(4.2)

$$\dot{m} = \frac{Qe}{(h1-h4)}$$

where :
$\dot{m}_1$: mass flow rate [kg/s]
Q_e : Heat transfer rate in evaporator (evaporator load) [W]
q_e : Refrigeration effect [kJ/kg]
h_1 : Enthalpy at point before compressor [kJ/kg] figure (2.1)
h_4 : Enthalpy at point before evaporator [kJ/kg]

$$\dot{m} = \frac{0.3415}{385-258} = 0.00269 \text{ [kg/s]}$$

$Q_c = \dot{m} \times (h_2 – h_3)$ ……………………………………..(4.3)

h_2 : Enthalpy at point before compressor [kJ/kg]
h_3 : Enthalpy at point after condenser [kJ/kg]
Q_c : Heat transfer rate in condenser (condenser load) [W]

$Q_c = 0.00269 \times (424 – 258)$
$Q_c = 0.4465$ [kW] = 446.5 [W]

$W_{c1} = \dot{m}_1 \times (h_2 – h_1)$ ……………………………………..(4.4)

W_c : Compressor Work [W]

W_c 0.00269 × (424 – 385)
$W_c = 0.105$ [kW] = 105 [W]

In Hours Power = 105/746 = 0.15 hp

That mean that we need 0.25 HP compressor to cover required load .

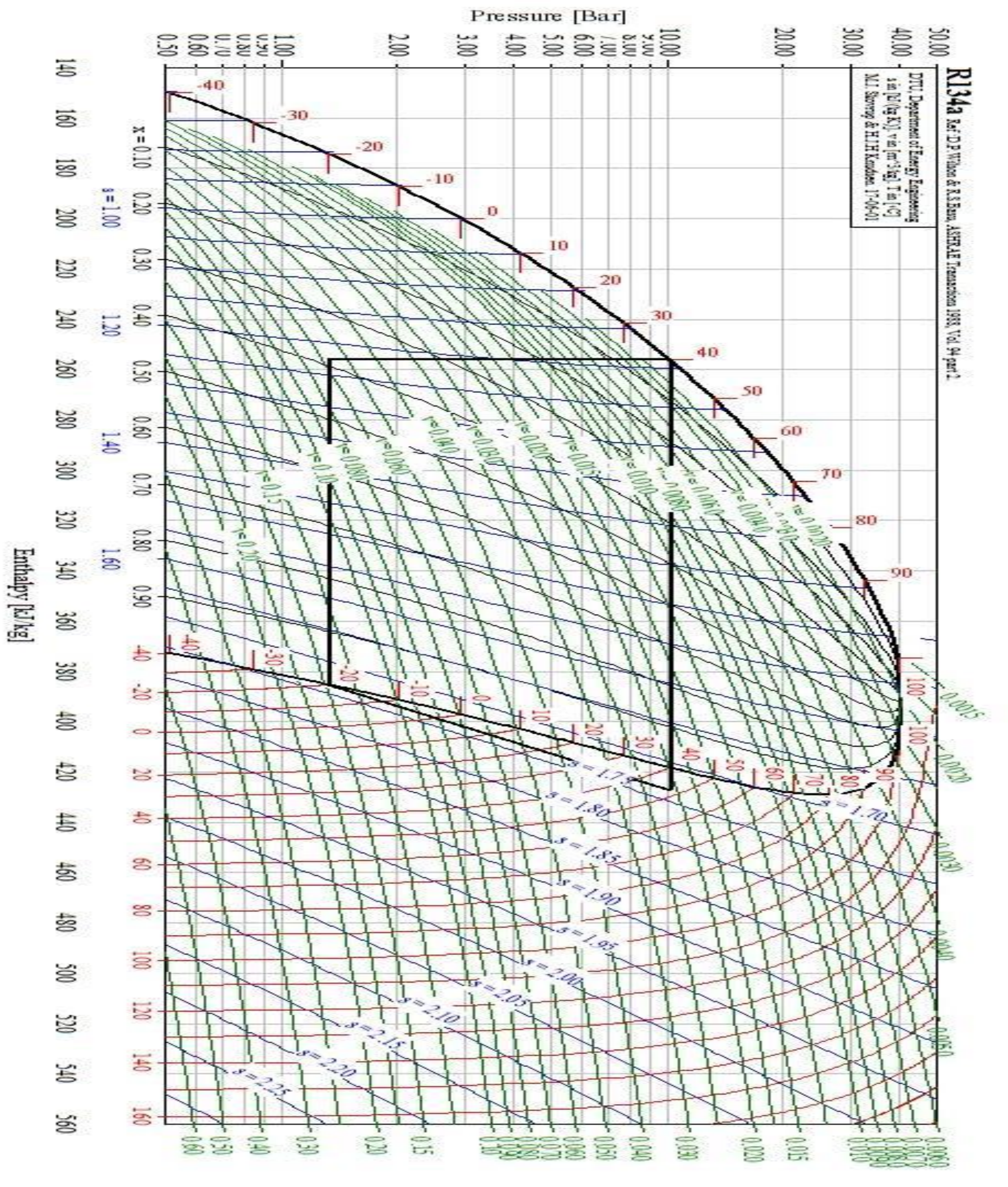

Figure (2.1) Cool pack software

4.2.2 Calculation Of Coefficient Of Performance (COP)

Condenser load

$Q_{condenser} = \dot{m} \times (h_2 - h_3)$
$Q_{condenser} = 0.00269 \times (424 - 258)$
$Q_{condenser} = 0.4465$ [kW] = 446.5 [W]

$Q_{evaporator} = \dot{m} \times (h_1 - h_4)$

$Q_{evaporator} = 0.00269 \times (385 - 258)$
$Q_{evaporator} = 0.3415$[kW] = 341.5 [W]

$COP = \frac{Qe}{Wc}$ ……………………………………..(4.5)

$COP = \frac{341.5}{105} = 3.25$

CHAPTER 5

Cycle Design

5.1 Compressor calculation and selection

5.2 Pipe design and selection

5.2.1 Cycle Pipe Calculation

5.3 Evaporator calculations and selection

5.4 Condenser calculation and selection

CHAPTER FIVE

CYCLE DESIGN

5.1 Compressor Calculation And Selection

To determine the volumetric efficiency for the compressor equation (5.1) can be used .

$\eta_v = \eta_c * \eta_h$ ………………………………………………………………………….. (5.1)

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 .

The volumetric efficiency due the clearance volume in compressor is calculated by equation (5.2) , [reference 11]

$\eta_c = 1 - c\,[(\frac{PH}{PL})^{1/n} -1\,]$ ……………………………………………………………. (5.2)

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, [reference 12]

n : exponential coefficient of expansion for refrigerant , n = 1 , [reference 7]

PH : High pressure of the cycle .

PL : Low pressure of the cycle .

$\eta_c = 1 – 0.02\,[(\frac{10.2}{1.52})^{1/1} -1\,] = 88.6\%$

The volumetric efficiency due to the heating in compressor is get from equation (5.3), [reference 6]

$\eta_h = \frac{Tevap.}{Tcond.}$ ……………………………………………………………......... (5.3) where :

$T_{evap.}$: evaporator temperature [°K]

$T_{cond.}$: condenser temperature [°K]

$\eta_h = \frac{253}{313}$ = 80.8%

η_v = 88.6% * 80.8% = 71.6%

The theoretical volume flow rate (V) of the compressor can be calculated in equation, [reference 11]

$\dot{V}_{theo} = \dot{m} * \nu$ ……………………………………………………………………... (5.4)

Where :

$\dot{V}_{theoretcal}$: theoretical volume flow rate of the compressor [m^3/s]

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

ν : specific volume at the inlet of compressor [m^3/s] ,[table 3.2]

$\dot{V}_{theoretcal}$ = 0.00269 * 0.13

= $3.45 * 10^{-4}$ [m^3/s]

To determine the actual volume flow rate , using equation (5.5),[reference 6]

$\dot{V}_{act} = \frac{\dot{V}_{the.}}{\eta v}$ ……………………………………………………………….. (5.5)

Where :

$\dot{V}_{act}$: actual volumetric flow rate [m^3/s]

$\dot{V}_{act} = \frac{0.000345}{0.716}$ = $4.82 * 10^{-4}$ [m^3/s]

5.2 Pipe Design and Selection

Pipe Design is very important to avoid explosion in pipe because of pressure and to avoid a noising sound produced from refrigeration flow .

Pipe design depend on volume flow rate , refrigerant velocity and inside pressure . [reference 13]

To calculate the inner diameter for the pipe the following steps are used [reference 14] .

$Q = \dot{m} * \nu$ ………………………………………………………………………… (5.6)

Where :

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

ν : specific volume [m^3 / kg] , (From table 3.2)

Q = A * V ………………………………………………………………………. (5.7)

Where :

A : cross sectional area [m^2]

V : velocity of refrigerant [m/s] , From table 6.2

$A = \pi * \frac{d_i^2}{4}$ ……………..……………………………………………………… (5.8)

Where :

d_i : inner diameter [m]

$$d_i = \sqrt{\frac{A*4}{\pi}}$$

To calculate the outer diameter for the pipe can be as the following [reference 13]

$\sigma_t = \frac{p_{in}*(r_o^2 + r_i^2)}{(r_o^2 - r_i^2)}$ ……………………………………………………………..... (5.9)

$Where$

σ_t : tangential stress [Mpa]

p_{in} : inner pressure [Mpa]

r_o : outer diameter [m]

r_i : inner diameter [m]

σ_t can be calculated from the following equation : [reference 13]

$\frac{SY}{n} = \sqrt{\sigma_t{}^2 + \sigma_t * p_{in} + p_{in}{}^2}$ …………………………………………………..... (5.10)

Where :

SY : yield strength [Mpa] , [70 Mpa for copper] , [reference 13]

n : factor of safety , taken 8 [recommended from cooper hand book] , [reference 13]

From equation (5.10) outer diameter for the pipe can be calculated

t = $r_o - r_i$ ……………………………………………………………….... (5.11)

Where :

t : thickness of the pipe [mm]

5.2.1 Cycle Pipe Calculation

Suction line pipe :

Q = $\dot{m} * \nu$

Q = 0.00269 * 0.13 = $3.45 * 10^{-4}$ [m^3 / s]

A $= \frac{Q}{V} = \frac{3.45 * 10^{-4}}{8} = 3.4 * 10^{-5}$ [m^2]

$d_i = \sqrt{\frac{3.4 * 10^{-5} * 4}{\pi}}$ = 6.63*10^{-3} [m]

r_i = 3.315 [mm]

$\frac{700}{8} = \sqrt{\delta_t^{\,2} + 1.53\,\delta_t + 1.53^2}$

δ_t = 86.72 [bar]

86.72 = $\frac{1.53\left(r_o^{\,2} + (3.315 * 10^{-4})^2\right)}{r_o^{\,2} - (3.315 * 10^{-4})^2}$

$r_o = 3.374$ [mm]

T = $r_o - r_i$

T = 3.374 – 3.315 = 0.059 [mm]

where **T** is the thickness

The inner and outer radius in inch is :

$r_i = r_i\ in\ mm$ / 25.4 = 0.131 inch

$r_o = r_o\ in\ mm$ / 25.4 = 0.133 inch

Discharge line pipe :

Q = $\dot{m} * v_2$

Q = 0.00269 * 0.0225 = 6.05* 10^{-5} [m^3 / s]

A = $\frac{Q}{V}$ = $\frac{6.05 * 10^{-5}}{10}$ = 6.05 * 10^{-6}[m^2]

$d_i = \sqrt{\frac{6.05 * 10^{-6} * 4}{\pi}}$ = 2.78 * 10^{-3} [m]

r_i = 1.46 [mm]

$\frac{700}{8} = \sqrt{\delta_t^{\,2} + 10.2\,\delta_t + 10.2^2}$

δ_t = 81.95 [bar]

81.95 = $\frac{10.2\left(r_o^{\,2} + 1.46^2\right)}{r_o^{\,2} - 1.46^2}$

$r_o =$ 1.65 [mm]

T = 1.65 – 1.46 = 0.19 [mm]

The inner and outer radius in inch is:

$r_i = r_i \; in \; mm$ / 25.4 = 0.057 inch

$r_o = r_o \; in \; mm$ / 25.4 = 0.065 inch

Liquid line pipe :

Q = $\dot{m} * v$

Q = 0.00269* 0.001 = 2.69 * 10^{-6} [m^3 / s]

A = $\frac{Q}{V}$ = $\frac{2.69 * 10^{-6}}{8}$ = 3.36 * 10^{-7} [m^2]

d_i = $\sqrt{\frac{3.36 * 10^{-7} \; * \; 4}{\pi}}$ = 6.54 * 10^{-4} [m]

r_i = 0.33[mm]

$\frac{700}{8} = \sqrt{\delta_t{}^2 + 10.2 \; \delta_t + 10.2^2}$

δ_t = 81.95 [bar]

82.08 = $\frac{10.2 \left(r_o{}^2 + 0.33^2 \right)}{r_o{}^2 - 0.33^2}$

$r_o =$ 0.37 [mm]

t = 0.37 – 0.33 = 0.04 [mm]

The inner and outer radius in inch is:

$r_i = r_i \; in \; mm$ / 25.4 = 0.0129 inch

$r_o = r_o \; in \; mm$ / 25.4 = 0.0146 inch

5.3 Evaporator calculations and selection

Figure (5.1) description of the available evaporator , and the following is geometrical data :

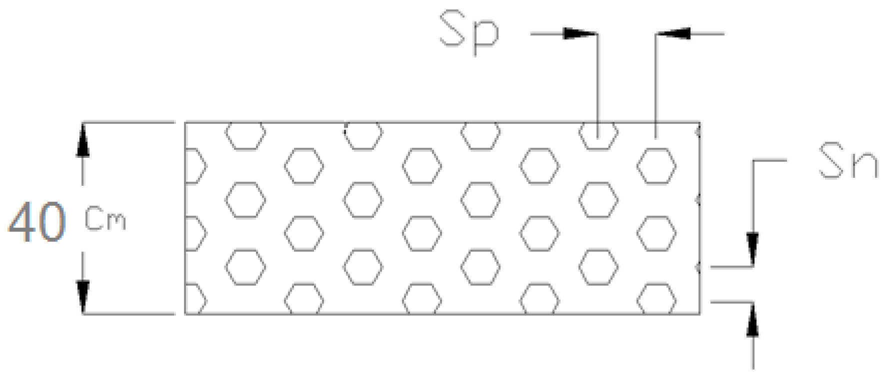

Figure (5.1) evaporator side view

Evaporator length L_e = 50 [cm]

Evaporator height H_e = 40 [cm]

Evaporator width W_e = 12 [cm]

S_n : Transevers tube spacing [m]

S_p : longitudinal tube spacing [m]

$$S_n = \frac{\text{evaporator height}}{\text{number of rows}} \qquad (5.12)$$

$$= \frac{0.40}{2} = 0.2 [\text{ m }]$$

$$S_p = \frac{evaporator\ width}{number\ of\ column} \qquad (5.13)$$

$$= \frac{0.12}{10} = 0.012 [\text{ m }]$$

Figure (5.2) show the fin elements

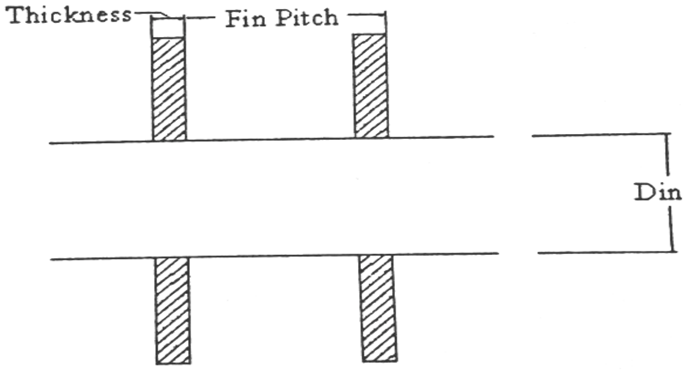

Figure (5.2) Fin elements for evaporator

Fine length = L_f = (S_n - D_o) …………………………………………..(5.14)

L_f = (0.2 – 0.01) = 0.19 [m]

Fine width = W_f = (S_p - D_o) ……………………………..……..…..(5.15)

W_f = (0.012 – 0.01) = 0.002 [m]

Number of fins in row = 90 fins

Fin pitch = $P_f = \frac{53}{90}$ = 0.588 [cm]

Fin thickness = t_f = 0.03 [cm]

Bare tube thickness = t_b = P_f - t_f …………………………………………….(5.16)

t_b = 0.588 – 0.03 = 0.558 [cm]

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

Q = h_o * A * (T_w -T_∞)

Where :

Q : heat transfer through the evaporator [w]

h_o : external convection heat transfer coefficient [W/m^{2o}C]

A : surface area of heat transfer [m^2]

T_w : outer evaporator wall temperature [^{o}C]

T_∞ : free air temperature [^{o}C]

The heat transferred through the evaporator was determined in chapter 2 and its value was Q_e= 341.5 [W].

To determine the external convection heat transfer coefficient can be as the following equations [reference]

$$h_o = \frac{N_u * k}{D} \quad \text{……………………………………………….………(5.17)}$$

Where

Nu : nusselt number

k : Thermal conductivity of air at entrance of evaporator [W /m^{o}C]

D : Outer diameter of evaporator [m]

The Nusselt number can be calculated by the equation [reference 14]

Nu = C * $(Re)^N * P_r^{1/3}$

Where

Re : Reynolds number

Pr : Prandtl number of air at film temperature

C,N : Constants can be obtain from table A-8 according the following considerations [reference 14]

S_p/D = 25/10 = 2.5

S_n/D = 25/10 = 2.5

From In line arrangement tube banks table:

C = 0.3

N = 0.595

$$Re = \frac{\rho * Vmax * D}{\mu} \qquad (5.17)$$

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 [Pa.s]

For flows normal to in –line tube banks the maximum flow velocity can be calculated as the following [reference 14]

$$V_{max} = V\infty \frac{Sn}{Sn-D} \qquad (5.18)$$

Where V∞ is the free air velocity entering the evaporator [m/s], can be calculated by the following equation

$$V\infty = \frac{V'}{A} \qquad (5.19)$$

Where

V′: flow rate of air through the evaporator [m^3/s], V′ = 50[cfm] = 0.02359 [m^3/s] from fan manufacturer company.

A : cross sectional area of evaporator [m^2].

A = 0.25 * 0.05 = 0.0125 [m^2]

$$V_{\infty} = \frac{Tw + T\infty}{2} \qquad (5.20)$$

Where :

T_F : film temperature [˚C]

T_W : wall surface temperature [˚C], assume that it equal the refrigerant temperature.

T∞ : free air temperature [˚C]

$T_F = \frac{-20+(-10)}{2} = -15$ [˚C] $= 258$ [K]

Then from table (A-9)

$\rho = 1.4$ [Kg/m^3]

$\mu = 1.599 * 10^{-5}$ [Kg/m.s]

k = 0.02 [W/m.˚C]

Prandtl = 0.725

$Re = \frac{1.4*3.145*0.01}{1.599*10^{-5}} = 2753.6$

Nu= 0.3 (2753.6) 0.595 * (0.725) 1/3 = 30.03

$ho = \frac{30.04 * 0.02}{10*10^{-3}} = 60.08$ [W/m2˚C]

In order to calculate the total heat transfer from one element (one fin and one bare tube) the following equation is used [reference 15] :

$Q_{total} = q_{fin\ act} + q_{original}$ …………………………………………………………..(5.21)

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

$q_{original} = h_0 * A_{origenal} *(T_w - T_\infty)$

Where :

h_0 : external convection heat transfer coefficient [W/m$^{2˚}$C]

$A_{origenal}$: the outer surface area of bare tube [m^2]

T_w : outer evaporator wall temperature [˚C]

T_∞ : free air temperature

$A_{original} = \pi D L = \pi * 10*10^{-3} * 5.6 * 10^{-3} = 1.7*10^{-4}$ [m^2]

$q_{original} = 60.08 *(1.7 * 10^{-4}) * (-10- (- 20)) = 0.102$ [W]

q_{fin} can be calculated by the equation

$q_{fin} = h_0 * A_{fin} * (T_w - T_\infty)$

Where :

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

A_{fin} : surface area for fin [m^2]

$A_{fin} = 2 (S_n * S_p – A_{pipe})$

$=2 [0.025*0.025 - \frac{\pi}{4} (0.01) 2] = 10 *10^{-4}$ [m^2]

$q_{fin} = 60.08 *(10*10^{-4)} (-10- -20) = 0.6$ [W]

Fin efficiency calculation :

$$n_f = \frac{\tan(m`Lf)}{m`Lf} \quad \text{………………………………………..……………………….(5.22)}$$

$$`L_f = (\frac{Lf}{2}) \left[1 + -0.35 \ln \frac{(\frac{D0}{2}+\frac{Lf}{2})}{\frac{D0}{2}}\right] \text{……………………………….……..….(5.23)}$$

$$`L_f = \left(\frac{0.015}{2}\right) \left[1 + 0.35 \ln \frac{(\frac{0.01}{2}+\frac{0.015}{2})}{\frac{0.01}{2}}\right] = 0.01[\ m]$$

$$m = \sqrt{\frac{h P}{k A}} \quad \text{…………………………………………………………………….(5.24)}$$

Where :

h : external convection heat transfer coefficient [W/m^2°C]

k: thermal conductivity of aluminum fin , [W/m°C]

P : perimeter of the fin[m]

A : surface area for convection of fin [m^2]

$P = 2 * t + 2 * L = 2 * 0.3 * 10^{-3} + 2 * 0.025 = 0.05$[m]

$A = t * L = 0.3*10^{-3} * 0.025 = 7.5*10^{-6}$ [m^2]

$m = \sqrt{\frac{64.7 * 0.05}{202 * 7.5 * 10^{-6}}} = 46.2$

$nf = \frac{\tanh(46.2 * 0.01)}{46.2 * 0.01} = 0.93$

So

The heat transfer flow from the fin is

$q_{fin\ act} = q_{fin} * n_f$ ……………………………………………………….(5.25)

$q_{fin\ act} = 1.9 * 0.93 = 1.767$ [W]

Now the total heat transfer from the element is

$q_{total} = 1.767 + 0.102 = 1.9$ [W]

Now the number of element that needed to perform the evaporator load can be determined by dividing the total heat transfer through the evaporator by the element total heat transfer , by using the following equation:

$n = \frac{Qe}{q\ total}$…………………………………………………..…………(5.26)

$n = \frac{341.5}{1.9} = 179.7$ elements

Number of elements in available evaporator = N*R

Where

N: number of elements in row

R : number of rows

Number of elements in available evaporator = 90 * 2 =180 elements.

So we can use same evaporator with 5 rows .

Number of elements = 90 * 5 = 450 .

5.4 Condenser calculations and selection :

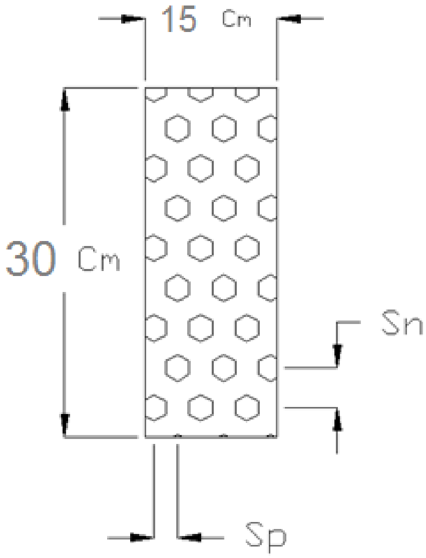

Figure (5.4) Condenser side view

Figure 5.4.1 description the available condenser , and the following is the geometrical data

Condenser length L_c = 40 [cm]

Condenser Height H_c = 30 [cm]

Condenser width W_c = 15 [cm]

S_n : Transverse tube spacing [m]

S_p : Longitudinal tube spacing [m]

$$S_n = \frac{condenser\ height}{numper\ of\ rows} \quad \ldots\ldots(5.27)$$

$$= \frac{0.3}{10} = 0.03\ [m]$$

$$S_p = \frac{condenser\ width}{numper\ of\ column} \quad \ldots\ldots(5.28)$$

$$= \frac{0.15}{10} = 0.015\ [m]$$

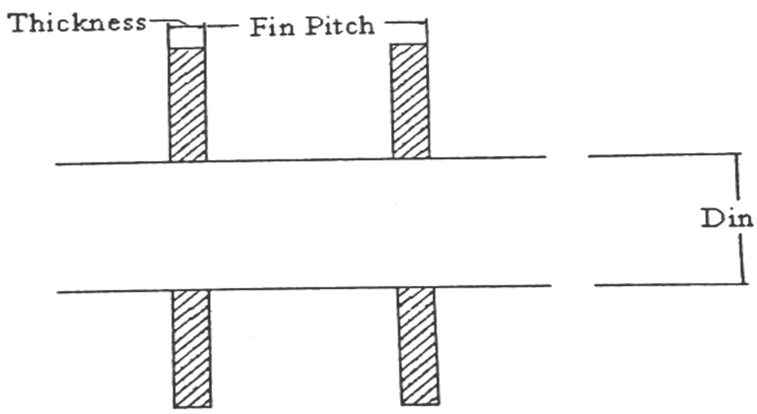

Figure (5.4.1) fin elements for condenser

Fin length = $L_f = (S_n - Do)$ ……………………………………………………(5.26)

= (0.03 – 0.01) = 0.02 [m]

Fin width = $W_f = (S_p - Do)$ ……………………………………………………(5.27)

= (0.015 – 0.01) = 0.005 [m]

Number of fins in a row = 96 fins

Fin pitch = $P_f = \frac{condenser\ leingth}{numper\ of\ fins}$ ………………………………………………..(5.28)

$= \frac{40}{96} = 0.41$ [cm]

Fin thickness $t_f = 0.02$ [cm]

Bare tube thickness tb = $P_f - t_f$ ………………………………………………..(5.29)

= 0.41 – 0.02 = 0.39 [cm]

So the project need one condenser.

CHAPTER SIX

ELECRICAL DESIGN

6.1 Introduction:

The refrigerators in general to worked for a long period and protect its component needs electrical circuits and sometimes electronic circuits to controlling an mentoring there system .

6.1.1 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 voltage in control circuit less or equal in power circuit. The energy consumed to control much less of the energy power circuit.

6.1.2 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 phase.

6.2 Components Of Electrical Circuits:

6.2.1: PTC relay :

A PTC (Positive Temperature Coefficient) relay is a starting device for fridge compressors. It is responsible for powering the start winding for a brief moment to help start up the fridge compressor motor. If your fridge cannot start there is a high probability that the PTC relay is defective. This article presents how to test them to determine whether they need replacing. Luckily, these common fridge parts are relatively inexpensive and easy to replace.

Figure (6.1) PTC relay

6.2.2 Capacitor Start and Run Motor:

Construction of the capacitor start and run motor is identical to that of the capacitor start motor with the exception that a second capacitor, called a running capacitor, is installed in series with the starting winding but in parallel with the starting capacitor and starting switch.

The operation of the capacitor start and run motor differs from that of the capacitor start and split-phase motors in that the starting or auxiliary winding remains in the circuit at all time.

At the instant of starting, the starting-and-running capacitors are both in the circuit in series with the auxiliary winding so that the capacity of both capacitors is utilized during the starting period. As the rotor approaches 70% of rated speed, the centrifugal mechanism opens the starting switch and removes the starting capacitor from the circuit, and the motor continues to operate with both main and auxiliary windings in the circuit.

The function of the running capacitor in series with the auxiliary winding is to correct the power factor. As a result the capacitor run and start motor not only has a high starting torque but also an excellent running efficiency. Figure 6.3 shows run and start Capacitors.

Figure (6.2) Run and Start Capacitors

6.2.3 Overload:

The most common cause of motor failure is overheating. The condition is created when a motor exceeds it is 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. Figure 6.4 shows overload.

Figure (6.3) Overload

6.2.4 Thermostat:

A programmable thermostat is a thermostat which is designed to adjust the temperature according to a series of programmed settings that take effect at different times of the day. Programmable thermostats may also be called setback thermostats or clock thermostats.

Figure (6.4) Thermostat

6.2.5 Contactors:

A contactor is an electrical switch that opens and closes under the control of another electrical circuit. In the original form, the switch is operated by an electromagnet to open or close one or many sets of contacts.

When a current flows through the coil, the resulting magnetic field attracts an armature that is mechanically linked to a moving contact. The movement either makes or breaks a connection with a fixed contact. When the current to the coil is switched off, the armature is returned to a normal position.

Contactors are used to control electric motors, lighting, heating, capacitor banks, and other electrical loads. Figure 6.6 shows contactor.

Figure 6.6 Contactor

6.2.6 Low-Pressure and High-Pressure Controls

The purpose of low-pressure control is to stop the compressor when the suction pressure drops below a preset value or when the refrigerant flow rate is too low to cool the compressor motor. Figure 6.8a shows a typical low-pressure control mechanism.

When the suction pressure falls below a certain limit, the spring pushes the blade downward, opens the motor circuit, and stops the compressor. When the suction pressure increases the bellows expand, thus closing the contact of the motor circuit and restarting the compressor. The two adjusting screws are used to set the cut-out and cut-in pressures.

Cut-out pressure is the pressure at which the compressor stops, and cut-in pressure is the pressure at which the compressor starts gain. [reference 1]

The purpose of high-pressure control is to stop the compressor when the discharge pressure of the hot gas approaches a dangerous level. Figure 6.8b shows a typical high-pressure control mechanism. If the discharge pressure reaches a certain limit, the bellows expand so that the blade opens the motor circuit contact and the compressor stops.

When the discharge pressure drops to a safe level, the bellows contract and close the contact, and the compressor starts again.

As in a low pressure control, two adjusting screws are used to set the cut-out and cut-in pressures.In small refrigeration systems, low-pressure and high-pressure controls are often combined to form a dual-pressure control. [reference 1]

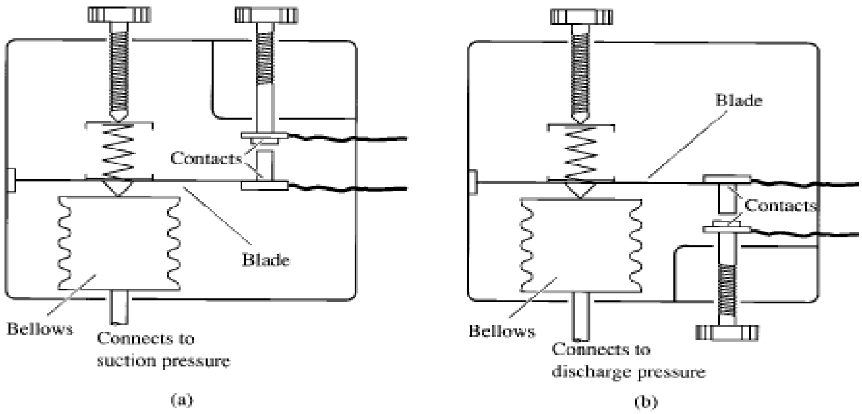

Figure 6.6 (a) Low pressure control and (b) high pressure control

6.3 Control of cycle using programmable board :

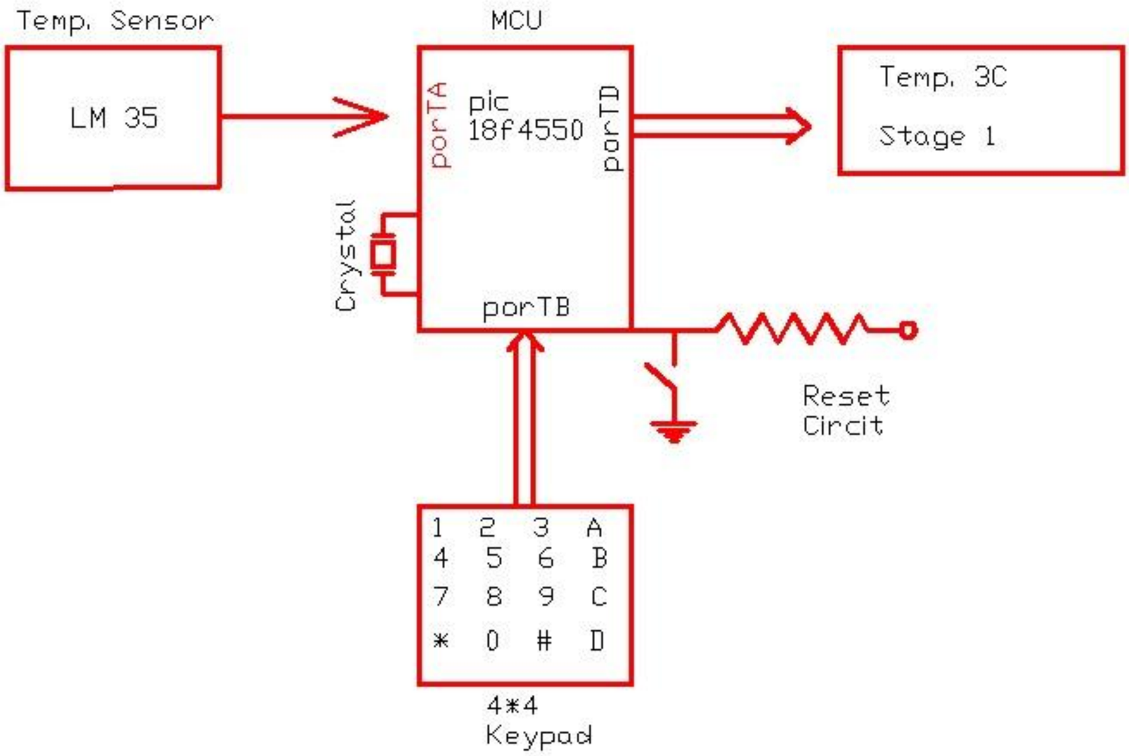

Figure (6.3.a) electronic control board diagram

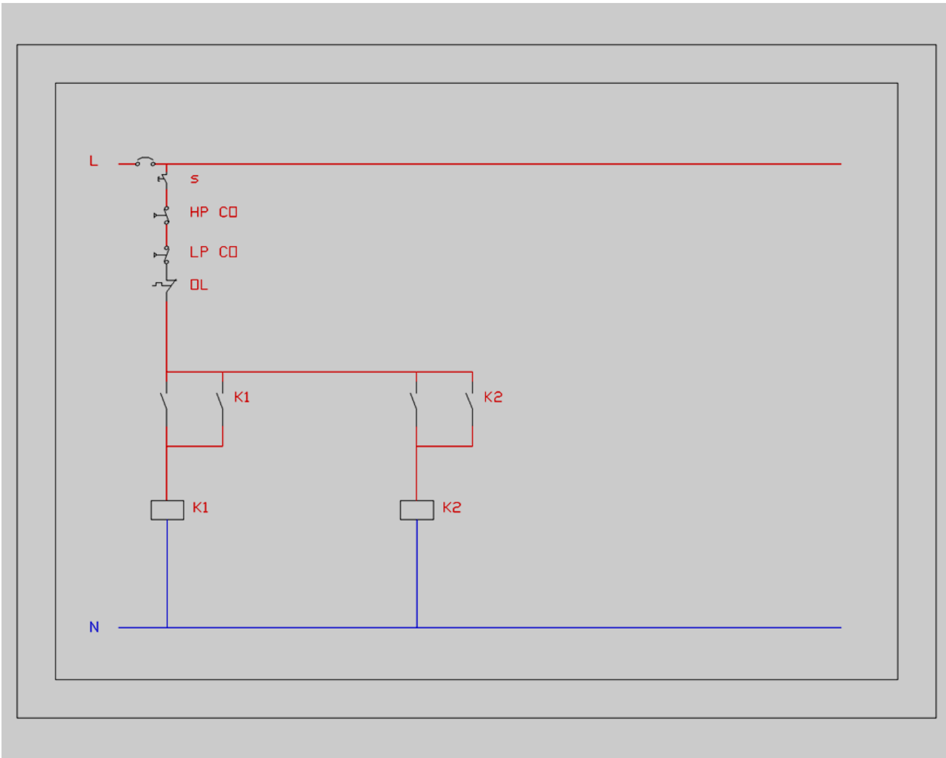

Figure (6.4.b) controlling by programmable board

6.4 Power Circuit:

Power circuit consists of four motors and a defrost reversing valve that work according to the signals from electronic board , figure 6.13 bellow depicts power circuit

blueprint for the cycle , figure 6.5 shows the electrical construction for compressor .

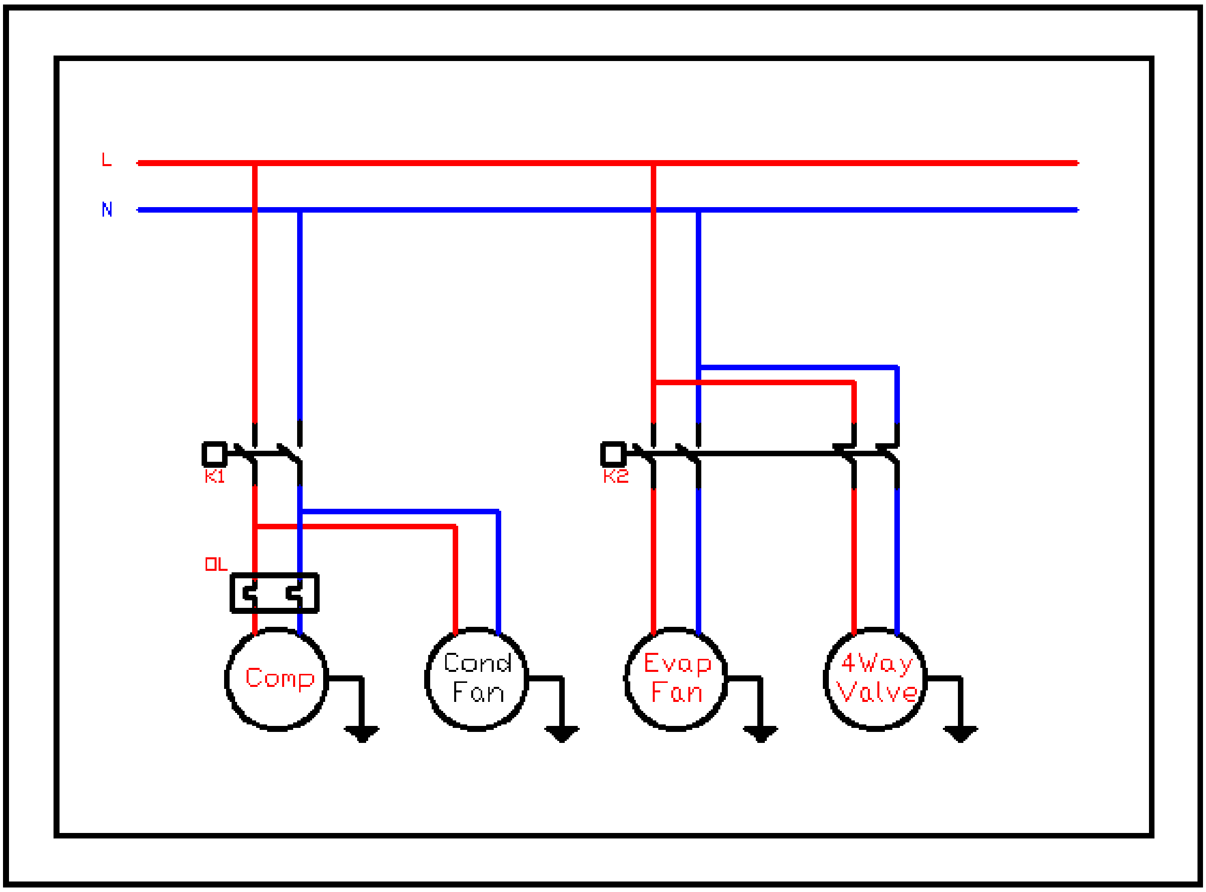

Figure (6.4) Electrical construction for compressor

Table 6.1: Circuit symbols:

Symbol	Description
L1	Phase line
N	Neutral line
LM 35	Temperature sensor
Ol	Overload
LP	Low stage pressure cutout
HP	High pressure cutout
K1	Composer contactor
K2	Fan & reversing valve
Comp	Compressor
S	Switch
RC	Run capacitor
SC	Start capacitor

CHAPTER 7

Drawings Of Refrigerator

6.1 Refrigerator Drawings

6.2 Refrigerator Views

6.3 Chamber Drawings

6.4 Door Drawings

6.5 Stand Drawings

6.1 Refrigerator Drawings :

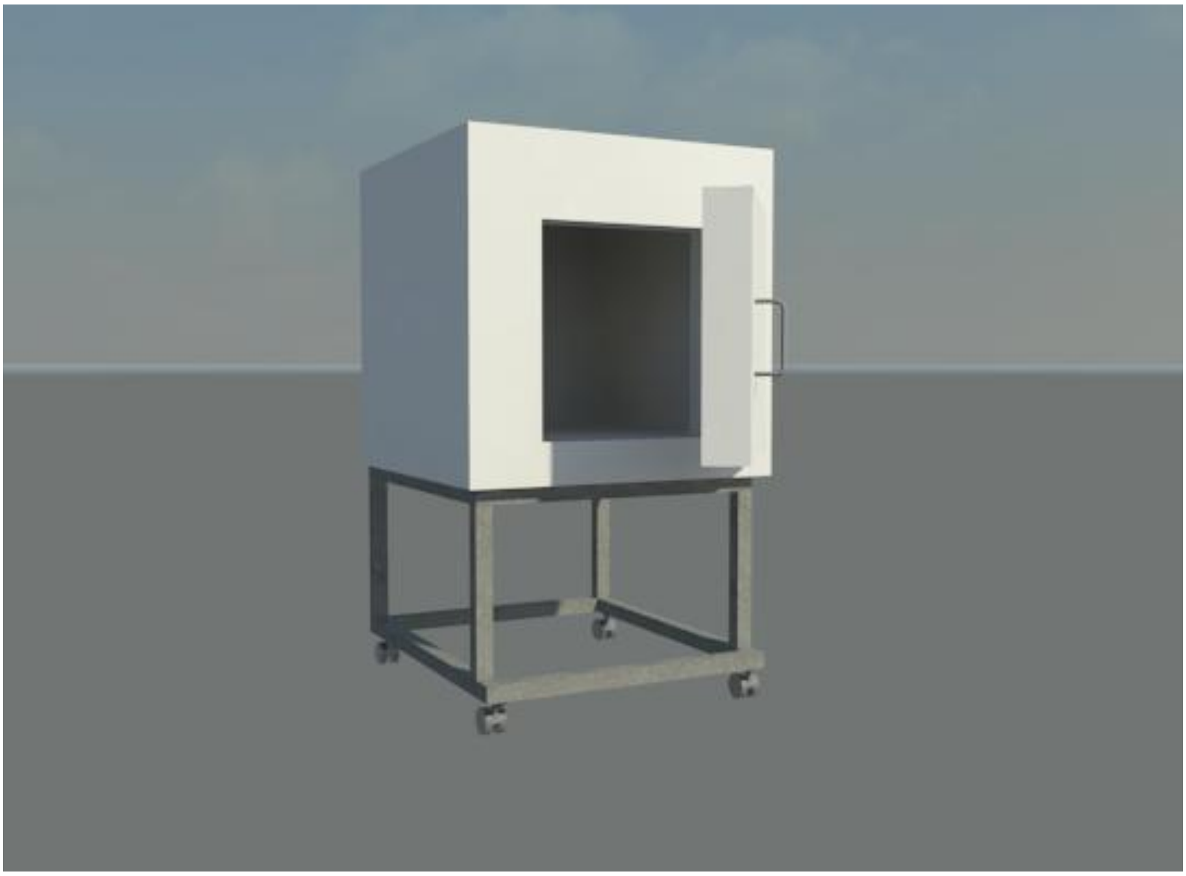

Figure 6.1 Refrigerator Drawing

6.2 Refrigerator Views :

Figure (6.2) Refrigerator Views

6.3 Chamber Drawings .

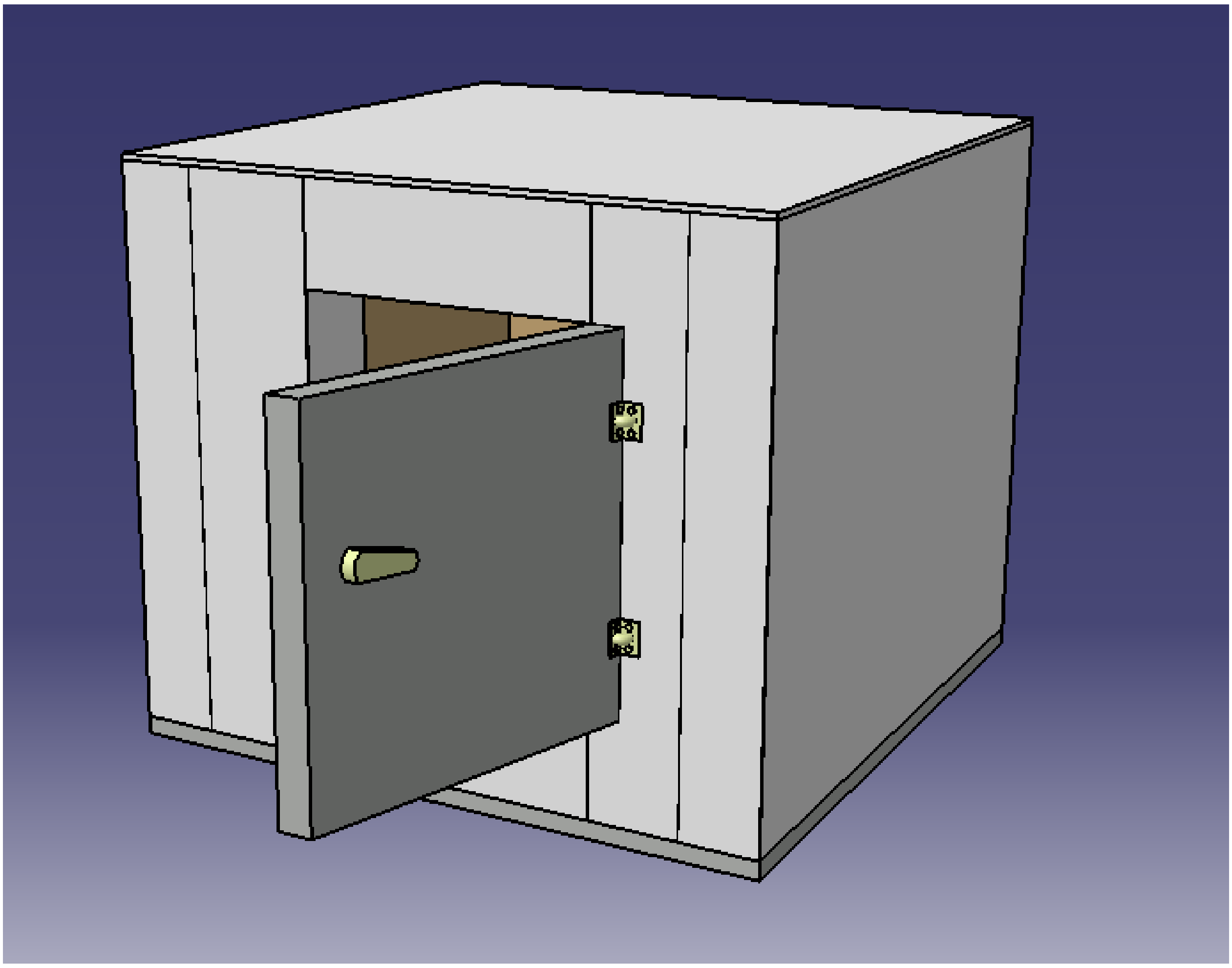

Figure (6.3) Chamber Drawing

6.4 Door Drawings :

Figure (6.4) Door Drawing

6.5 Stand Drawings .

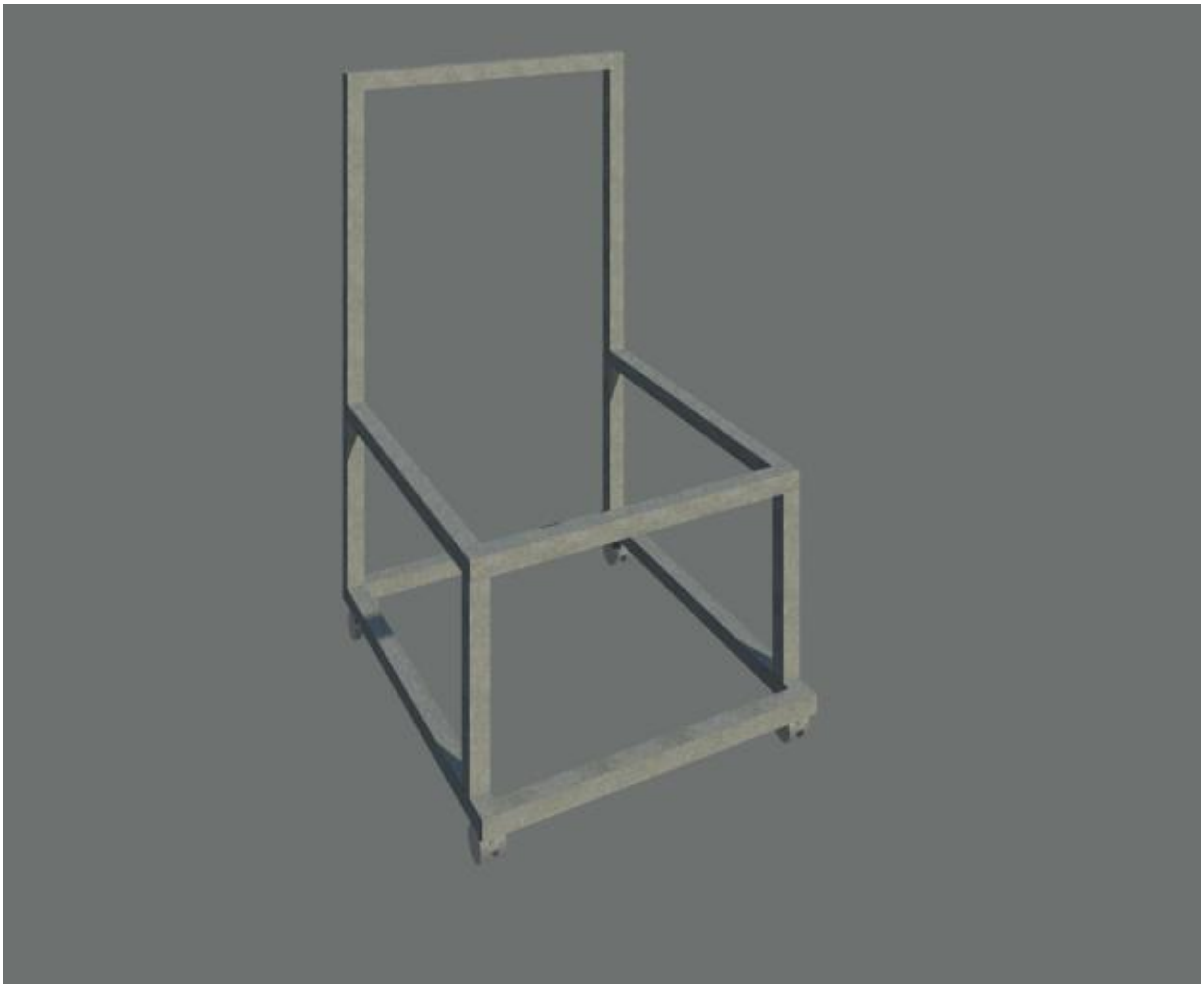

Figure (6.5) Door Drawing

1.7. The Budget Of The Project

Table 1.11 : The budget of the project :

Task	Cost (JD)
Researches	**15**
Transportation	**35**
Printing paper	**25**
Reprinting Paper	**25**
Components of the project	
Compressor	**100**
Condenser	**50**
Evaporator	**50**
Connecting pipes	**20**
Capillary tube	**5**
Refrigerator Frame	**20**
Refrigerant	**20**
Reversing valve	**30**
Control Equipments	
Pressure switches	**200**
Temperature switch	
Wires	
Switches	
Filter	
Sight glass	
Float valve	
Contactors	
Total	**592 JD**

Appendix A

TABLE A-1 Thermal Conductivity Of Materials

TABLE A-2 Properties Of Common Foods

TABLE A-3 Air Change Per Hour

TABLE A-4 Specific Heat Of Packaging Material

TABLE A-5 Maximum And Minimum Temperature For Hebron City

TABLE A-6 Recommended Refrigerant Velocities

TABLE A-1 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		7.95
	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 Polythane	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	
	Redwood bark	0.037	
	Wood fiber (soft woods)	0.043	
Glass	Single pane		6.42
	Two pane		2.61
	Three pane		1.65
	Four pane		1.19
Metal	Galvanized steel	15.6	
	Aluminum	202	
	cooper	386	

TABLE A-2 Properties Of Common Foods

Food	Water content %(mass)	Freezing point [°C]	Specific heat [kJ/kg.°C]		Latent heat [kJ/kg]
			Above freezing	Below freezing	
Vegetables					
Artichokes	84	-1.2	3.65	1.90	281
Asparagus	93	-0.6	3.96	2.01	311
Beans, snap	89	-0.7	3.82	1.96	297
Broccoli	90	-0.6	3.86	1.97	301
Cabbage	92	-0.9	3.92	2.00	307
Carrots	88	-1.4	3.79	1.95	294
Cauliflower	92	-0.8	3.92	2.00	307
Cucumbers	96	-0.5	4.06	2.05	321
Eggplant	93	-0.8	3.96	2.01	311
Horseradish	75	-1.8	3.35	1.78	251
Leeks	85	-0.7	3.69	1.91	284
Lettuce	95	-0.2	4.02	2.04	317
Fruits					
Apples	84	-1.1	3.65	1.90	281
Apricots	85	-1.1	3.69	1.91	284
Avocados	65	-0.3	3.02	1.66	217
Bananas	75	-0.8	3.35	1.78	251
Blueberries	82	-1.6	3.59	1.87	274
Cantaloupes	92	-1.2	3.92	2.00	307
Peaches	89	-0.9	3.82	1.96	297
Pears	83	-1.6	3.62	1.89	277
Plumps	86	-0.8	3.72	1.92	287
Quinces	85	-2.0	3.69	1.91	284
Raisins	18	-------	-------	1.07	60
Strawberries	90	-0.8	3.86	1.97	301
Tangerines	87	-1.1	3.75	1.94	291
Watermelon	93	-0.4	3.96	2.01	311
substances					
Frozen plasma	92	-0.9	3.92	2.00	307
water	100	0.0	4	2.1	2255

TABLE A-3 **Air Change Per Hour**

Kind of room or building	Air Change[m^3/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-4 **Specific Heat Of Packaging Material**

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

TABLE A-5 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-6 Recommended Refrigerant Velocities

Line	Refrigerant		Recommended velocity[m/s]
Suction	R23	R507	10
Discharge	R23	R507	15

References

[1] Shan K. Wang, Hand book of Air conditioning and refrigeration, 2ed edition .

[2] http://www.binder-world.com/en/ultra-low-temperature-freezer/uf-v-series/uf-v-500/

[3] Roy J. Dossat , Principles of refrigeration , second edition , SI version , university of Houston , Houston , Texas , USA .

[4] Welbert F. Stocker , Industrial Refrigeration hand Book .

[5] Ibrahim Dincer , Refrigeration system and Application , Department of Mechanical Engineering , KFUPM , Saudi Arabia , 2003 .

[6] http://www.online freeebooks.net/engineering-e-books/manufacture-mechanical-industrial-engineering/refrigeration-and-air-conditioning-pdf.htm .

[7] Cool pack software .

[8] Chumak I.G, Nikulshina D.G, Refrigeration system first edition , Vee shya shcola cor, Moscw ,1988 .

[9] Burmeister, L.C., Element of thermal-fluid System Design , Prentice Hall, 1998 .

[10] Joseph E. Shigley , Charles R. Mischke .Richard G. Budyans, Mechanical Engineering design , Seven edition , University of Michign, and lowa state university, MCGRAW-HILLBOOK COMPANY, USA .

[11] J.P. Holman, Heat Transfer , Ninth Edition , South Methodist University , New York , MCGRAW-HILLBOOK COMPANY, USA .

[12] http://www.fronine.com.au/haier-freezer.pdf

[13] http://www.wolflabs.co.uk/laboratory-products/freezers-ultra-low-temperature

[14] http://acintergroup.com/index.php/ultra-guardtm-ultra-low-temperature-freezer-uf-v-700-binder.html

[15] AutoCAD software