Palestine Polytechnic University

College of Engineering & Technology

Mechanical Engineering Department

Design and building of Cascade Refrigeration chamber - has a capacity of 30 litter - Operating at -65 C , using R404a and R23 as refrigerants

Produced by

Mohammad YousefAdawiUddaiRasmiSweity

Zaid Jamal Alnazer

Supervisor:

Dr. Ishaq Sider

Hebron – Palestine

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Project Name

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Project Team

Mohammad YousedAdawiUddaiRasmi Al-swity

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بناء 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

توقيع اللجنة الممتحنة

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توقيع رئيس الدائرة

Dedication

To our Families ……………….…… For their support

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

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

To oppressed people throughout the world and their struggle for social justice and egalitarianism

To our great Palestine

To our supervisor DrIshaq Sider

To all who made this work possible

Mohammad YousefAdawiUddaiRasmiSweity

Zaid Jamal Alnazer

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ABSTRACT

Design and building of a cascade refrigeration chamber with capacity of 30 litter operating at -65 ^{o}C, using R-23 and R-404a as refrigerants .

This project can be considered as an implementation of the featured project and sciences theory in a practical model . Cascade refrigeration cycle is one of most suitable cycles for application that needs very low temperature .

The system consists of two separated cycles connected with each other through a heat exchanger, where the higher cycle using refrigerants R-404a operates between (35 to -32^{o}C) , the lower cycle using refrigerants R-23 operates between (-27 to -65^{o}C) this cycle find application in different scientific fields such as storage of blood plasma and storage of tissues which need a very low temperature .

The project is suitable for many fields, The most important field that can benefit from this project is the scientific field . And Commercial field can benefit from this project .

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CHAPTER ONE

INTRODUCTION

1.1. Overview

Creating the proper environment for rapid freezing and storage is absolutely critical . Maintaining the exact temperature range required takes a reliable storage enclosure plus heavy-duty refrigeration system .

As a provide of dependable blood plasma freezers for over 30 years, Master-Bilt supplies a complete package of walk-in panels and refrigeration systems.

Refrigeration systems for blood plasma consist of remote condensing units with matching evaporator coils .

Cascade refrigeration system is one of the most suitable cycles for applications that need very low temperatures, storage blood plasma is one of these applications .

Application which work under very low temperature have evaporating temperature of frozen cabinets that operates at the ranges from (-30°C to -50°C).

In this system, two single-stage units are thermally coupled through cascade condensers , to get a very low temperature, may reach's to -65°C at the end of this cycle.

1.2. Project Idea

The project idea emerges to design and build blood plasma storage unit, due to the market needs for this application especially at medicine field .

Project discuss the following :

1- Show the difference between single stage and two stage refrigeration cycles .
2- Compares between the Refrigerants used in the project and other refrigerants .
3- Thermal analysis of the cycle .
4- Design of the chamber .
5- Selection the cycle components .
6- Assembling the cycle components and get ready for testing .

7- Testing the cycle .

1.3. Project Importance

In the refrigeration field , most of the attention tends to get cooling systems which allows access to very low temperature .

Cascade refrigeration system is one of the applications which made in Developed countries, so, design this cycle is a great achievement .

1.4. Project Objectives

1- Design and building of Cascade Refrigeration chamber Operating at -65^{o}C.

2- Add an applied refrigeration cycle to Educational entities, universities workshops, and will be a praxis which supports Theory courses .

3- Support the market with blood plasma refrigeration system with high (COP), low operating cost, and ease for use .

1.5. Literature Review

People used many methods for storage food , blood plasma , tissues , and other substances . Then they created the refrigerator which used a mechanical systems that uses many elements working together to get the required cooling .

The cascade system was used in 1877 for liquefaction oxygen sulfur dioxide (SO2) and carbon dioxide (CO2) as intermediate refrigerants , now it is used for many applications and fields such as medical fields , biological fields , and industrial fields .

1.5.1 Simple refrigeration cycle

Simple refrigeration cycle is a simple vapor compression refrigeration system , in which suitable working substance termed as 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 , but the single stage refrigeration cycle is limited in the lower evaporation temperature and the lower vapor pressure in the evaporator , and the compressor sucks less refrigerant per stroke so it has a lower volumetric efficiency . This problem caused reducing capacity to the point where load and capacity are in balance [Reference1] .

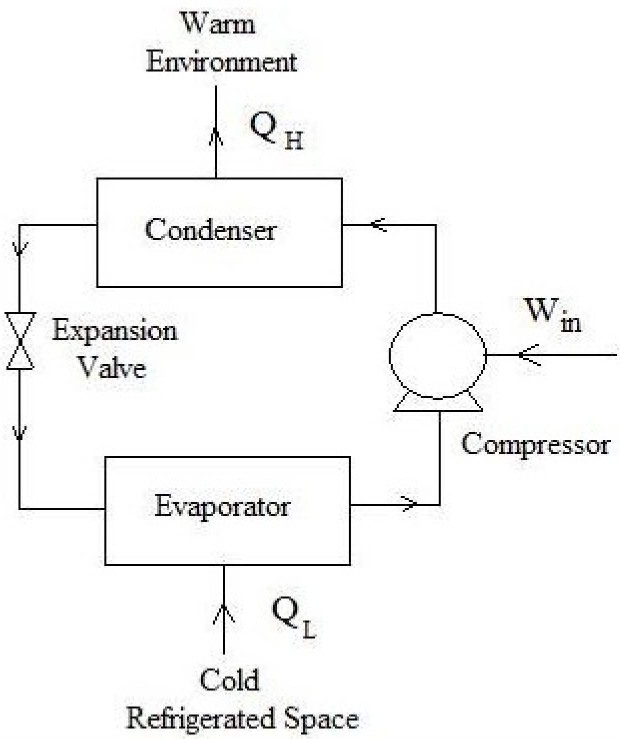

Fig.1.1: Simple vapor compression refrigeration system .

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 . The compression carried by this case is called multistage compression .

1.5.2Multistage refrigeration cycle

Multistage system also used for many refrigeration applications and fields , it can be a suitable choice to solve many problems that appears in single stage system such as to improve a high volumetric efficiency , and reaching a lower temperature . On the other hand the temperature that reachedis Inappropriate for application that need very low temperature reaching to -70°C such as storage blood plasma and tissues , despite this cycle provides effective lubrication because of the lower of temperature- but lubricating oil can be wander between compressors [refarence1].

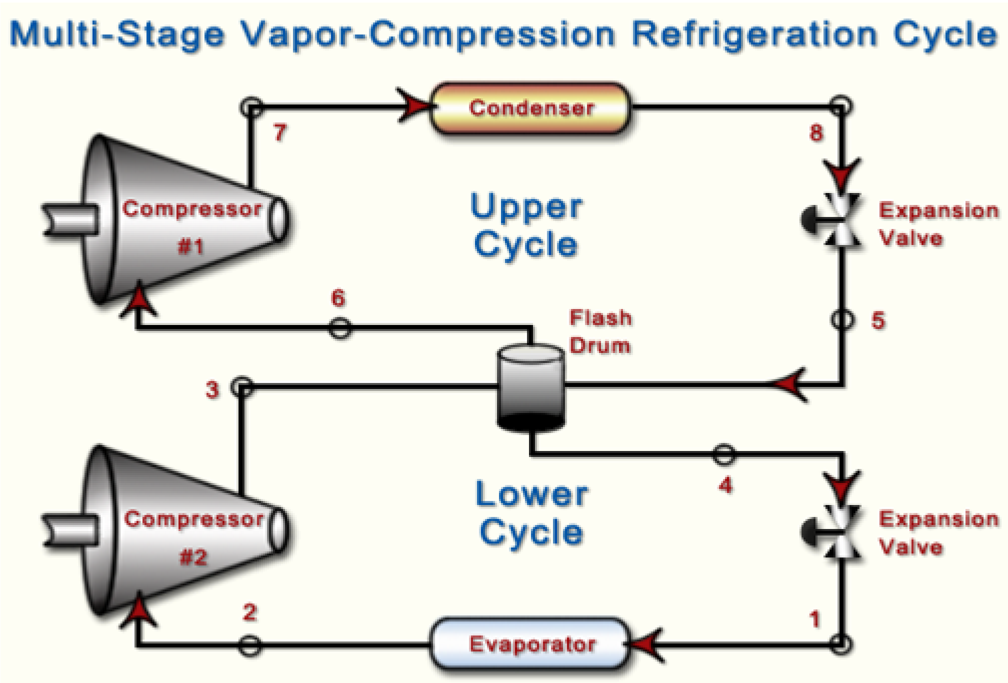

Fig.1.2: Multistage vapor compression refrigeration system .

1.5.3Cascade refrigeration cycle

Cascade system is one of multistage systems , this system consists of two refrigeration systems in series which use two different refrigerants . One of this refrigerants has a low boiling temperature .

The principle of the cascade system is that to permit the use of two different refrigerants . The high temperature cascade cycle uses a refrigerant with high boiling temperature for example R-404a, R-12 , R-22 , and R-134a . The low temperature cascade cycle with low boiling temperature such as R-13 , R-23 , and R-170 , these low boiling temperature refrigerants have extremely high pressure which increase a smaller compressor displacement in the low temperature cascade system and give higher coefficient of performance. The additional advantages of this system over the multistage system is the lubricating oil from one compressor cant wander to the other compressor [Reference 1] .

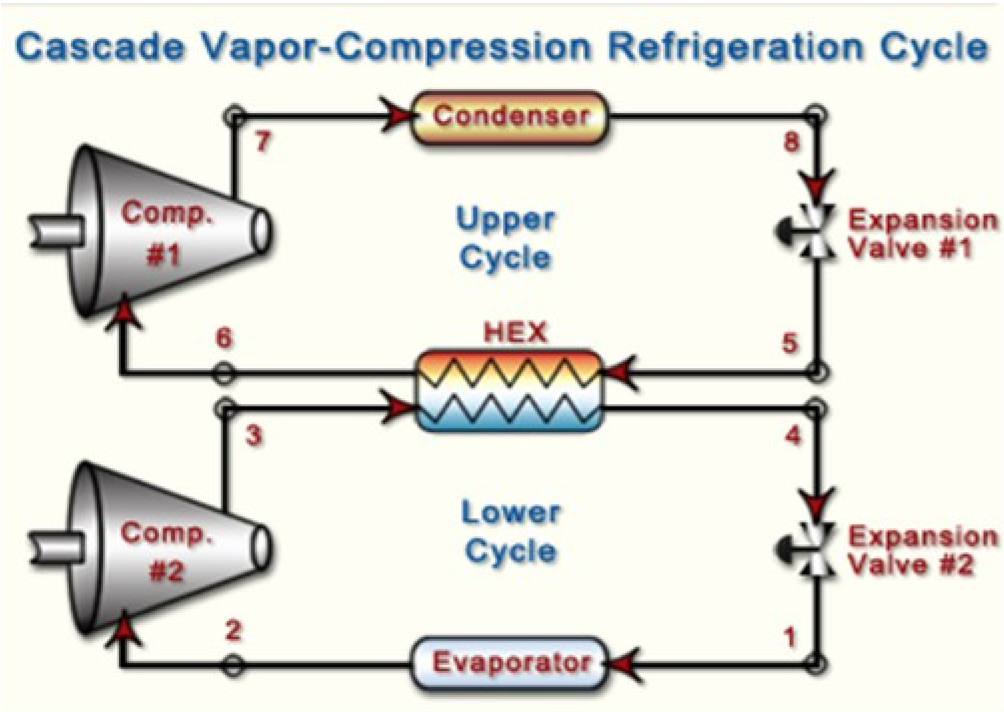

Fig.1.3: Cascade vapor compression refrigeration system .

1.5.4Auto cascade refrigeration cycle

Another type of multistage refrigeration cycle is auto cascade system , it also uses refrigerants with different boiling temperatures , but this system can not be reaching a very low temperature with high evaporating pressure , because of this cycle uses only one compressor . Besides it needs a special substance mixture to used as refrigerants .

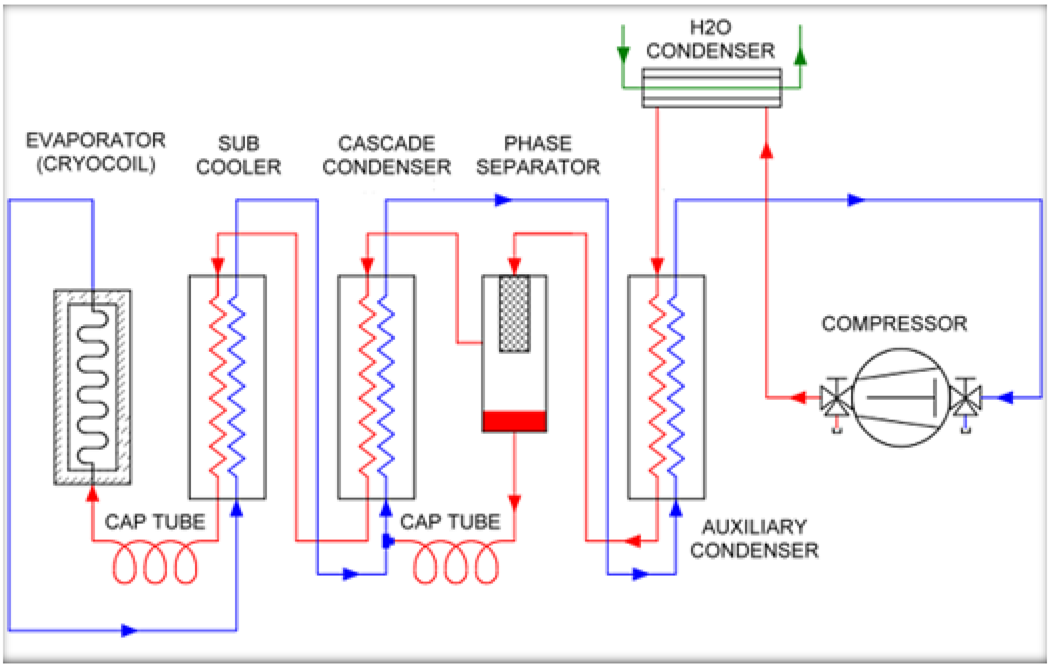

Fig.1.4: Auto Cascade vapor compression refrigeration system .

The following table shows the comparison between the refrigeration cycles which are discusedabove .

Table 1.1 : Comparison between refrigeration cycles

Cycle	Advantages	Disadvantages
Simple refrigeration system	1- It has smaller size for a given capacity of refrigeration 2- Lower running costs 3- Coefficient of performance is quite high	1- High initial costs 2- There is no prevention of leakage 3- The evaporation temperature islow. 4- Low volumetric efficiency .
Multistage refrigeration system	1- Improve high volumetric efficiency 2- Reaching low temperature 3- Provide effective lubrication system	1- High initial cost 2- Take more size than simple cycle 3- Required more protection devices 4- Lubricating oil can

	4- Coefficient of performance higher than the simple cycle	be wander from first compressor to the second one
Cascade refrigeration system	1- Using different refrigerants 2- Using different oil so it can't wander from compressor to the other one 3- Reaching very low temperature 4- Improve high volumetric efficiency 5- Provide effective lubricating system 6- Higher coefficient of performance	1- More complicated than multistage and simple cycle 2- Over lap of the condensed temperature of lower cycle with the evaporating temperature of the higher cycle
Auto cascade system	Cheaper than cascade system	Need special refrigeration mixture

1.5.5. Cascade cycle products

These are specially designed and manufactured for long time storage of biological product, its application can be found in blood banks , hospitals , laboratories and electrical and chemical plants .

1.5.5.1. Ultra-low temperature upright laboratory freezer -86 °C(Haier Co.)

Fig.1.5: Ultra-low temperature upright laboratory freezer.

The following table shows the features of Haier company[reference 2].

Table 1.2 : Features of Haier Product

Type	-86 ULT Freezer
Cabinet Class	Upright
Cooling Type	Direct cooling
Defrost Mode	Manual
Refrigerant	CFC

Advantages :

- Low noise level .
- Energy saving by using door design for the minimum loss of cold temperature during door opining , and using insulation panels .
- High efficiency by using high efficiency refrigeration components such as hermetically sealed compressor and high efficiency oil separator .

- Reliability using network connection with PC for remote monitoring and using temperature recorder .
- Safety using alarm systems .

Disadvantages :

- High cost for using high cost equipments, it cost about (8,995$-10,995$). [Reference 16]
- Using CFC refrigerant can be affecting onenvironment .
- Complex Maintenance and operation .

1.5.5.2. Ultra Low Temperature Freezer (Panasonic Co.)

Fig.1.6: Ultra Low Temperature Freezer

The following table shows the features of Panasonic Product[reference 3] .

Table 1.3 : Features of Panasonic Product

Specifications	Ultra-Low Temperature Freezer
Model	MDF-U700VX
Temperature range	– 50 °C to – 86 °C (in 1 °C increments)
Maximum cooling performance	– 86 °C (Ambient temp. 30 °C)
Compressor	2 compressors, hermetic 1100 W
Refrigerant	HFC
Alarms and safety	High / low temperature, Door ajar, Power failure, Remote alarm contact, Part replacement notification, Fan lock alarm, Refrigeration circuit abnormal alarm
Accessories	1 scraper, Vacuum release port cleaning gear, 1 set of keys

Advantages :

- Low noise level .
- Very low Temperature .
- High efficiency .
- Reliability .
- Safety using alarm systems , and other accessories .
- Using H-F-C refrigerant.

Disadvantages :

- High cost , it cost about (10,747$-26,999$) .[Reference 17]
- Complex Maintenance .
- Using many types of controller can be complicate the system and rise its cost.
- Complex operation

1.5.5.3. ULTRA.GUARD™ Ultra Low Temperature Freezer (Binder Co.)

Fig.1.7: ULTRA.GUARD™ UltraLow Temperature Freezer

The following table shows the features of Binder product [reference 4] .

Table 1.4 : Features of Binder Product

Specifications	Ultra-Low Temperature Freezer
Temperature range	– 40 °C to – 86 °C
Compressor	2 compressors, hermetic
Refrigerant	CFC
Average temperature variation	2.5

Advantages :

- Easy control , Freezer can be opened and closed effortlessly , Automatic door system .
- Superior energy efficiency
- Quiet operating environment
- Easy operation
- Increased safety features
- Very low Temperature .

- Low noise level .

Disadvantages :

- Complex Maintenance .
- Using C-F-C refrigerants .
- High cost for using high cost equipments , it cost about (15,623$) .[Reference 18]
- There is no prevention of leakage .

Using simple components and equipments ,this project is able to achieve outputs that are close to the achievements of the previous products . Besides , the project can be used in educational field .

1.6. Time Plan

The time plan explains the stages in designing and building the system components. The section includes the first table that shows the activities and task scheduling for the second semester, while the second table shows the tasks for the summer semester, and the last table shows the activities for the 1st semester .

Table 1.5: Timetable for the second semester time (week)

Activity	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Selection of the project															
Search about information															
Search for previous projects															
Search for video for the system in the website															
Cooling Load Calculation															
Cycle analyses															

Components selection															
Search for the components															
Electrical Design															
Chamber Body Design															
Project Documentation															

Table 1.6: Timetable for the summer semester time (week)

Activity	1	2	3	4	5	6	7	8
Search for the components								
Build The System								
Project Documentation								

Table 1.7: Timetable for the first semester time (week)

Activity	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Build The Project															
Testing															
Conclusion and Final Edit															
Project Documentation															

1.7. Risk Management

The implementation of any project may face many risks during each stage of the project, such as: determining and analyzing the system requirements, designing, implementing and testing the whole system. This section illustrates the problems that might occur during the implementation .

The following table lists the types of risk which may impact the projectdevelopment process, showing the risk and its type technology, people, organizational, tools,requirements, and estimation – and a short description about it.

Table 1.8 : Risks Identification and Characterization

Type of Risk	Number of Risk	Possible Risks
Technology	R1	Hardware which is essential for the project may not be delivered on schedule
	R2	Malfunction of hardware parts (Compressors ,Thermal Temperature).
Requirements	R3	There may be a larger number of changes to the requirements than anticipated
Estimations	R4	The time required developing the hardware isUnderestimated.
	R5	The Budget is not sufficient.
Organizational	R6	Time of delivery of the project changed.
People	R7	Illness of one or more team member.

The following table lists the probability and effect of each risk:

Table 1.9 : Risk Probabilities and Analysis :

Risk ID	Risk	Probability	Effects
R1	Hardware which is essential for the project may notbe delivered on schedule	Very high	80%
R2	Malfunction of hardware parts (Compressors ,Thermal Temperature).	Moderate	50%
R3	There may be a larger number of changes to the requirements than anticipated	Low	30%
R4	The time required developing the hardware is Underestimated.	Low	60%
R5	The Budget is not sufficient.	High	70%
R6	Time of delivery of the project changed.	Low	20%
R7	Illness of one or more team member.	Low	50%

The following table shows the strategies that address each risk in this project:

Table 1.10 : Risk Management Strategies :

Risk ID	Strategy
R1	Identify the needed hardware as fast as possible and order them.
R2	Understand almost the exact time needed in developing the hardware.
R3	Understand almost the exact size for the hardware.
R4	Searching for alternative hardware and waiting to get it .
R5	Ask for financial support from the university Ask for financial support from the factories
R6	Ask our friends to help us to delivered it in time
R7	Ask our friends to help us to delivered it in time

1.8. The Budget Of The Project

Table 1.11 : The budget of the project :

Price	Type	Quantity
40	Heater	1
200	Digital thermostat	1
60	Sight glass	1
40	Capillary tube	2
1160	Compressor	2
195	Pipes	15
90	Isolator	2
350	Radiator	1
180	Fan	2
2200	Gas R23	6 KG
70	Thermostat	2
40	Nuts	4
50	Filter drier	1
50	Nedel	1
130	Ex – valve	1
20	Filter	2
80	Pipes	1
450	R404 gas	13.6 Kg
1000	Body	1
2200	Compressors	2
270	Contactor	6
12	Switch 16A	1
14	Led	2
45	Relay with base	1
30	Relay	1
15	Steel Bridge	1
15	Red Switch	1
15	Green Switch	1
9021 Nis	**Total price**	

CHAPTER 2

COOLING LOAD

CHAPTER TWO

COOLING LOAD

2.1 Introduction

The total heat required to be removed from refrigerated space in order to bring it at the desired temperature and maintain it by the refrigeration equipment is known as cooling load . The purpose of 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 and satisfactory inside conditions .[reference 5].

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 . [Reference6] .

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 .

Overview about the Chamber :
Storage temperature is $-65\ ℃$.

Surrounding Temperature is $35\ ℃$.

Mass of the product $\approx 30\ Kg$

Cooling time is ten hours .

Chamber Dimensions (0.7 , 0.7 , 0.6) meter .

Chamber size = 0.7 * 0.7 * 0.6= 294 liter .

Effective Chamber size =200 liter

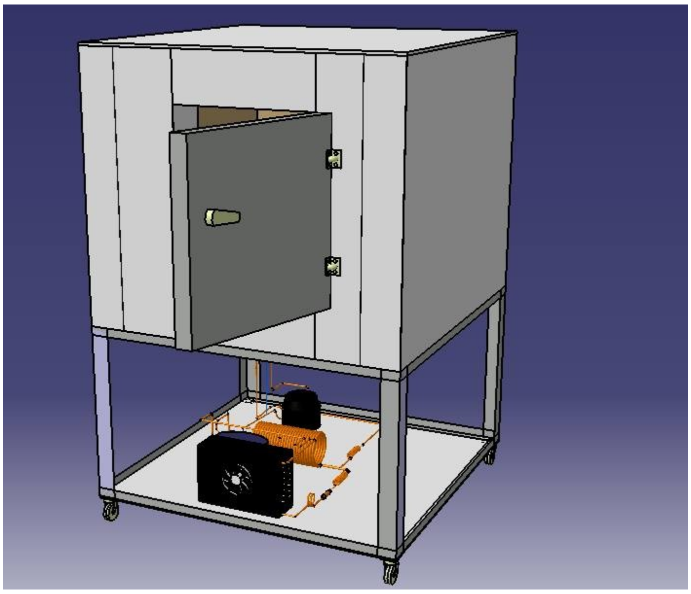

Figure 2.1 The chamber design

2.2.1The 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 . [reference6] .

$Q_{wall} =$ U * A * ΔT ……………………………2.1

Where :-

A : Outside Surface Area of The Wall [m^2]

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

ΔT: the temperature differences across the walls [°C]

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

Where :

T_{in} : -65 °C

T_{out} :35°C

Overall heat transfer coefficient is computed by the following :

$$U = \frac{\mathbf{1}}{\frac{\mathbf{1}}{\boldsymbol{h_i}}+\frac{\boldsymbol{\Delta x1}}{\boldsymbol{k_1}}+\frac{\boldsymbol{\Delta x2}}{\boldsymbol{k_2}}+\cdots+\frac{\mathbf{1}}{\boldsymbol{h_o}}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 2.2$$

Where :

U :the overall heat transfer coefficient [$W/{}^\circ C * m^2$]

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

K : the thermal conductivity of the material [$W/m.{}^\circ C$] .

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

Forced convection by using fan (30 – 100) , taken 50[$W/{}^\circ C.m^2$] .

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

Free convection inside the room (5- 20) , taken 10 [$W/{}^\circ C.m^2$] .

All walls are constructed of three layers as shown in Figure 2.1 .

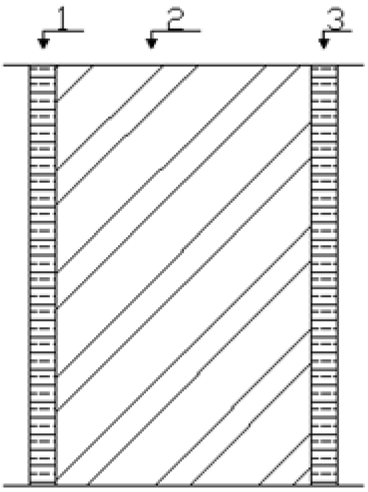

Figure 2.2 Chamber wall layer

1- Galvanizedsteel 0.1 [cm]   , K = 15.60 [$W/m.{}^\circ C$] . From Table A-1 .
2- polytheren 15 [cm]   , K = 0.036 [$W/m.{}^\circ C$] . From Table A-1 .
3- Galvanizedsteel 0.1 [cm] , K = 15.60 [$W/m.{}^\circ C$] . From Table A-1 .

$$U = \frac{\mathbf{1}}{\frac{\mathbf{1}}{\mathbf{50}}+\frac{\mathbf{0.002}}{\mathbf{15.6}}+\frac{\mathbf{0.15}}{\mathbf{0.036}}+\frac{\mathbf{1}}{\mathbf{10}}} = 0.233[\,{}^{W}/_{°C.\,m^2}\,] .$$

$Q_{floor\ \&roof}$ = 0.233 * (0.7 * 0.7) * (35 - -65) = 11.417 W
$Q_{tow\ sides}$ = 0.233 * (0.7 * 0.6) * (35 - -65) = 9.786 W
$Q_{front\ \&behind}$ = 0.233 * (0.7 * 0.6) * (35 - -65) = 9.786 W

$Q_{allwalls}$ = 2 * ($Q_{floor\ \&roof}$ + Q_{sides} + $Q_{front\ \&behind}$)
$Q_{allwalls}$ = 2 * (11.417 + 9.786 + 9.786) = 61.98 [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 :

1- Chilling load above freezing : The product cooling freezing depends upon the mass product , mean specific heat of the products above freezing , entering product temperature , final product temperature desired , and the cooling time . this heat gain can be calculated by the following eqs , : [reference 5]

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 above freezing in [kJ / kg . °C]
ΔT : $T_{out} - T_{ch}$

Where :
T_o : entering product temperature [°C]
T_{ch} : chilling product temperature [°C]
C_p = 3.92 [kJ / kg . °C] (Table A-2)
T_o = 35 [°C]
T_{ch} = -0.9 [°C]
Q_{ch} = 30 * 3.92 * (35 - -0.9)
Q_{ch} = 4221.84[kJ]

2- Cooling load below freezing : the cooling load below freezing depends upon the mass of product , mean specific heat of the product below freezing actual storage temperature of the 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 / °C]
ΔT : ($T_{ch} - T_{Rs}$)

Where :
T_{ch} : freezing product temperature [°C]
T_{Rs} : refrigerated temperature [°C]
C_p = 2 [kJ / kg . °C] table A – 2 .
T_{ch} = - 0.9 [°C] table A -2
T_{Rs}= -65 [°C]

Q_c= m . C_p . ΔT
Q_c= 30 * 2 * (-0.9 –(-65))
Q_c = 3846 [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 = 307 [kJ / kg]
Q_f = 30 * 307
Q_f = 9210[kJ]

Total product load

$Q_p = (Q_{ch} + Q_c + Q_f)$ / C.T

Where :

C.T : desired cooling time in [seconds]

Q_p : ($\frac{(4221.84-3846+9210)}{10*(3600\ )}$)* 1000

Q_p = 266.27[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 sizes . [reference7]

$$Q_{inf} = m \cdot C_p \cdot (T_o - T_i)$$

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

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 = 0.7 * 0.7 * 0.6 = 0. 294m^3

$V_f =$ 0.294 * 0.5 = 0.147m^3/ hr

Q_{inf} = 1.25 * (0.147 / 3600) * 1000 * (35 - -65)

Q_{inf} = 5.1[W]

2.2.4 Packaging Heat gain

Many products refrigerated in packages , it could be more than 10% of products weight . Packages could be plastic , steel , wood , glass or any material that have low specific heat . [reference5]

Plastic bags used to keep 250 g of liquid plasma , and packaging it in Galvanized steel crate . Bags of plasma are arranged above the Galvanized steel crate .

$$Q_{pk} = \frac{m_{pk} * C_{pk} *(T_o - T_i)}{\tau} * 10^3$$

Where :

Q_{pk} : packaging heat load [W]

m_{pk} : mass of product [kg]

C_{pk} : packaging material specific heat [J / kg . ℃]

T_o : outside temperature [℃]

T_i : temperature of the refrigerated space [℃]

τ : desired cooling time in [seconds]

- For Galvanized steel crate :

C_{pk} steel = 0.5 [kJ / kg . ℃] (table A-4)

$$Q_{pk} = \frac{5*0.5*(35 - -65)}{10*3600} * 10^3$$

Q_{pk} = 6.94[W]

- For plastic bags :

C_{pk} = 1.6 [kJ/kg . ℃] From Table A-4

$$Q_{pk} = \frac{0.2*1.6*(35 - -65)}{10 *3600} * 1000$$

Q_{pk} = 0.88[W]

$Q_{pk\ total} = Q_{pk\ steel} + Q_{pk\ plastic}$

$Q_{pk\ total}$ = 6.94 + 0.88 = 7.82[W]

2.2.5 Defrosts heater heat gain

The process of removing frost from the evaporator and around the door is called defrosting .

2.2.5.1 Defrosts heater heat gain of the evaporator

If the surface temperature of the evaporator coil is (0℃) and lower , frost accumulates on the coil surface . Because frost impedes air passage and reduces the rate of heat transfer of the coil , it must be removed periodically . An electric heating element is used as a simple and effective way to defrost the coil .[reference 1] .

$Q_{h1} = \varphi * P$

Where :

P : Power of heater , taken 500 [W]

φ : heater usage factor (0.1 – 0.5) , taken 0.2

$Q_{h1} = P * \varphi = 0.2 * 500 = 100$ [W]

2.2.5.2 Defrosts heater heat gain around the door

The function of the heater around the door is to prevent frost from forming the door , making it difficult to open , and to prevent condensation phenomena around the door that happen because high temperature difference between the refrigeration space and the surrounding .

$Q_{h2} = \varphi * P$

Where :

P : power of heater . taken 80 [kW]

φ : heater usage factor (0.1 – 0.5) , taken 0.2

$Q_{h2} = 0.2 * 80 = 16$ [W]

2.2.6 Fan motor heat gain

The evaporator fan motor release a heat , this heat relatively equal the power of the motor

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

2.3 Total cooling load

The total cooling load is the summation of the heat gains

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

Q_T = 61.98 + 266.27 + 5.1 + 7.82 + 116 + 25

Q_T = 482.17[W]

Add 40 % as safety of factor

Total cooling load = Q_T * 1.4

Total cooling load = 482.17 * 1.4 = 675 [W]

CHAPTER 3

COMPONENTS OF CASCADE REFRIGERATION SYSTEM

CHAPTER THREE

COMPONENTS OF CASCADE REFRIGERATION SYSTEM

3.1. Introduction

There are several mechanical components required in cascade refrigeration system .This part of project discuss the five major components of the system and som auxiliary equipments working with these major components [reference 8].

The major components of the cascade refrigeration system are as follows :

- Compressor,
- Condenser,
- Evaporator
- Throttling device, and
- Heat exchanger,

3.2. Compressor

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
- Simihermetic compressor
- Open compressor

In this project we will use hermetic compressor .

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) [reference 8].

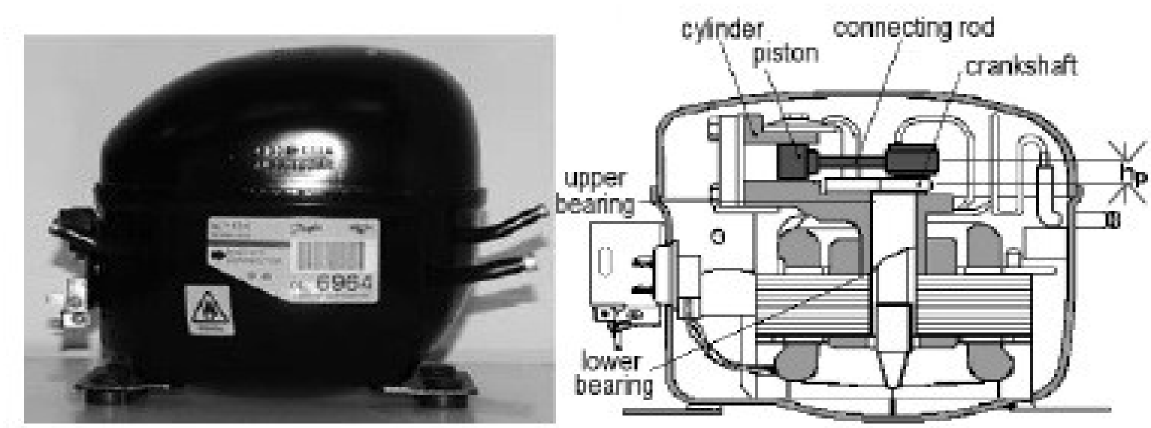

Figure 3.1 Hermetic Compressor

3.2.2. Expectations From The 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

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 [reference 5].

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 condenser) .

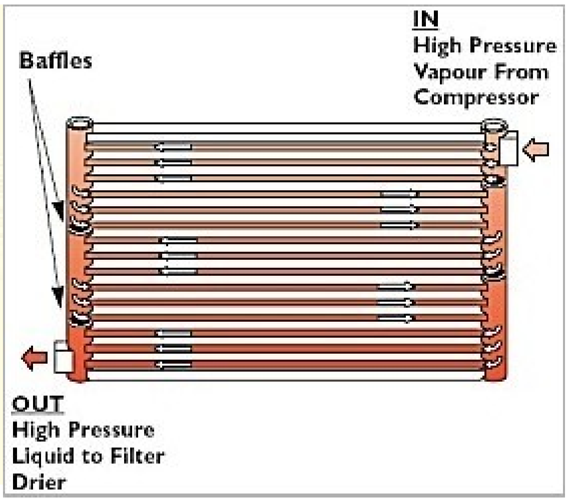

Figure 3.2 Air Cooled Condenser

3.4. Evaporator

Evaporator 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[reference 9] .

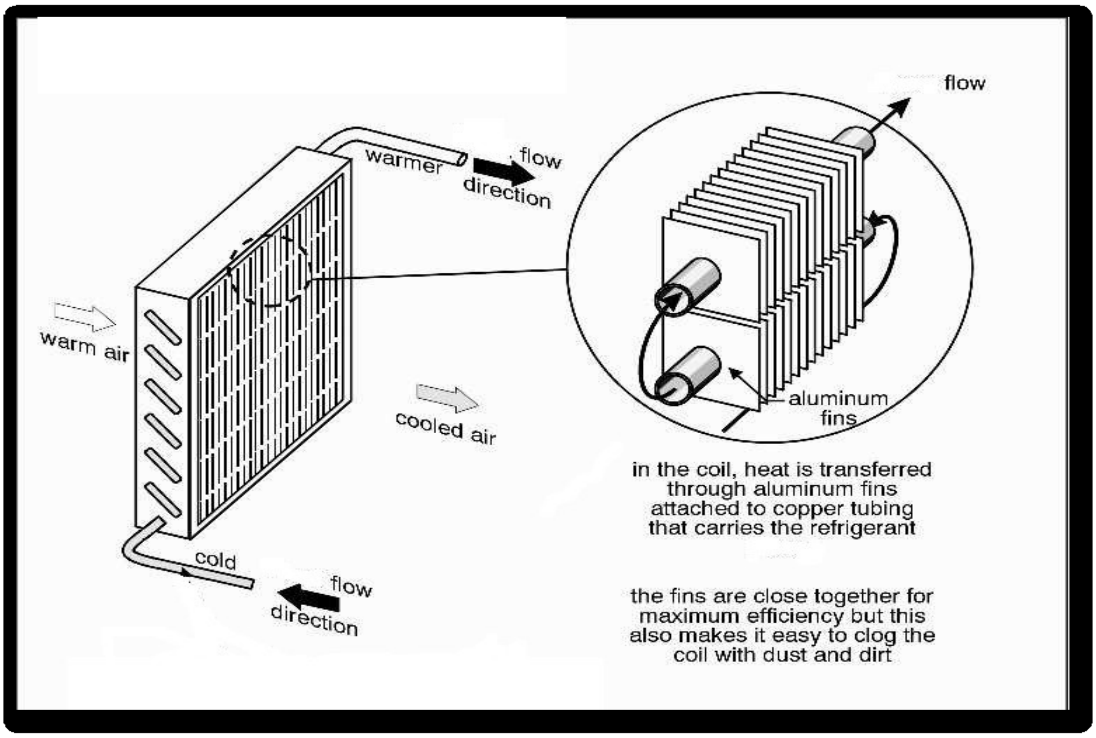

Figure 3.3 Evaporator

3.4.1. Bare Tube Coil Evaporator

The simplest type of evaporator is the bare tube coil evaporator, as shown in figure 3.3. 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 [reference 9].

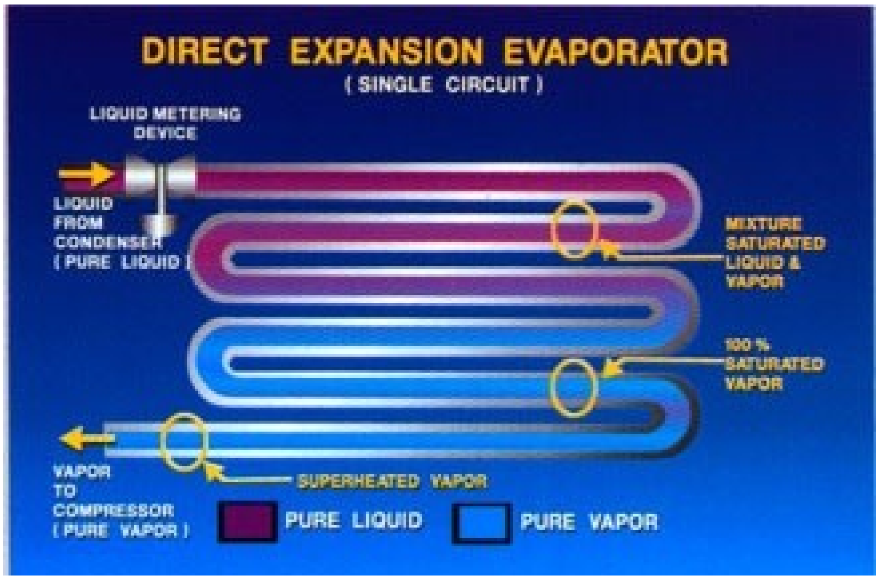

Figure 3.4 Bare Tube Coil Evaporator

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[reference 9] .

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 3.4 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. Heat exchanger

A heat exchanger is a device built for efficient heat transfer from one fluid to another, whether the fluids are separated by a solid wall so that they never mix, or the fluids are directly contacted. They are widely used in petroleum refineries, chemical planets, petrochemical planet, natural gas processing, refrigeration, power planet, air conditioning and space heating. We will use double pipe heat exchanger in this project[reference 9]. (figure 3.5 shows double pipe heat exchanger).

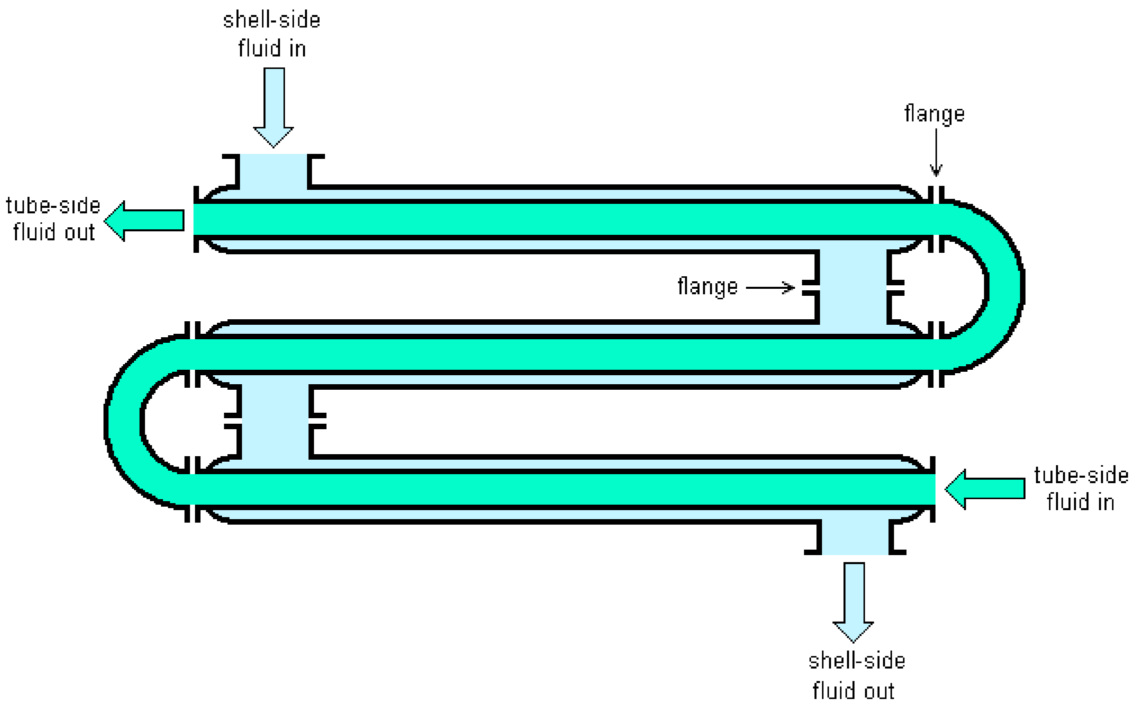

Figure 3.6 Double Pipe Heat Exchanger

3.7. Auxiliary Component

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

3.7.1 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 .

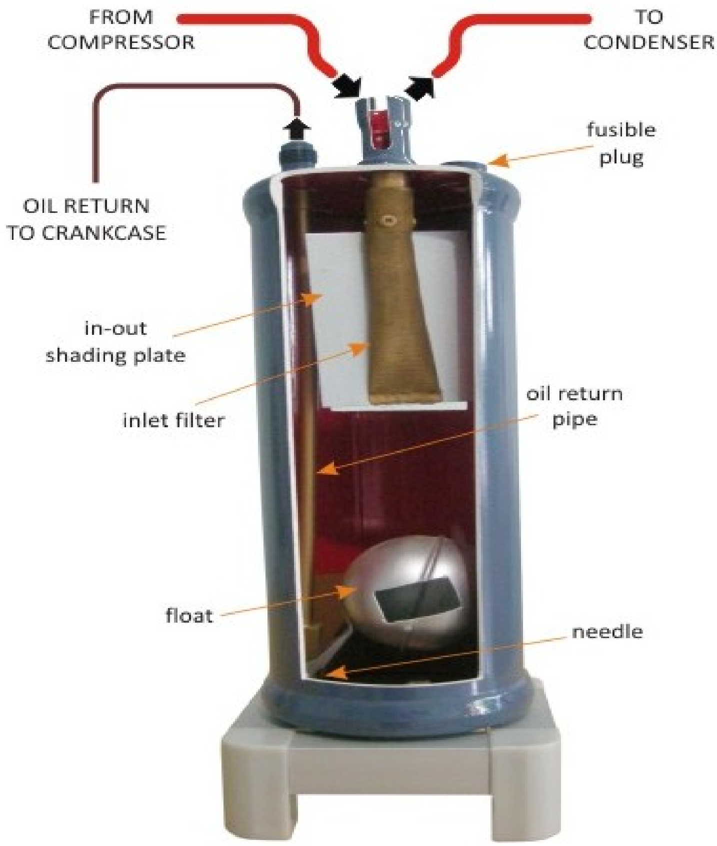

Figure 3.7 Oil separator

3.7.2. 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 3.7(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 start again [reference 9] .

The purpose of high pressure control is to stop the compressor when the discharge pressure of the hot gas approaches a dangerous level. Figure 3.7(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.

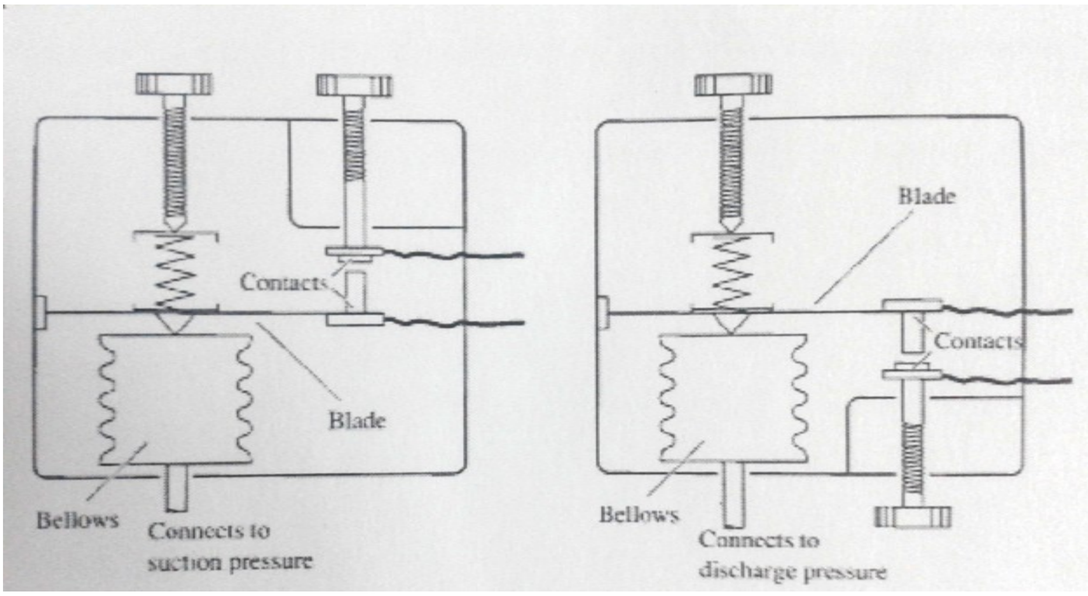

Figure 3.8 (a)Low pressure control (b) high pressure control

Figure 3.9 Low and high pressure control

3.7.3. 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 *setpoint*. 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 statos "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 [reference 1].

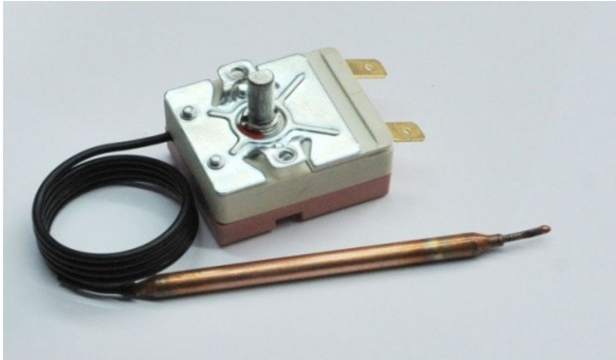

Figure 3.10 Thermostat

CHAPTER 4

CYCLE ANALYSIS

CHAPTER FOUR

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-404a for high stage cycle and R-23 For low stage cycle . In addition to above properties in selection process of R-404a we have to determine the compressor type that works perfectly with R-404a, and it has to be cheap and available . And to select the low stage cycle refrigerant we have to consider important factor which is , avoiding vacuum occurrence, in other words (pressure inside cycle must be a little greater than atmospheric pressure) . If the atmospheric pressure became greater than inside cycle pressure will destroy the cycle, so we select R-23 . Refrigerant 23 has low temperature at 1 atm , so R-23 is suitable for our system [reference 1].

4.2 Cycle Analysis

Figure 5.1 describes Ideal Cascade refrigeration cycle.

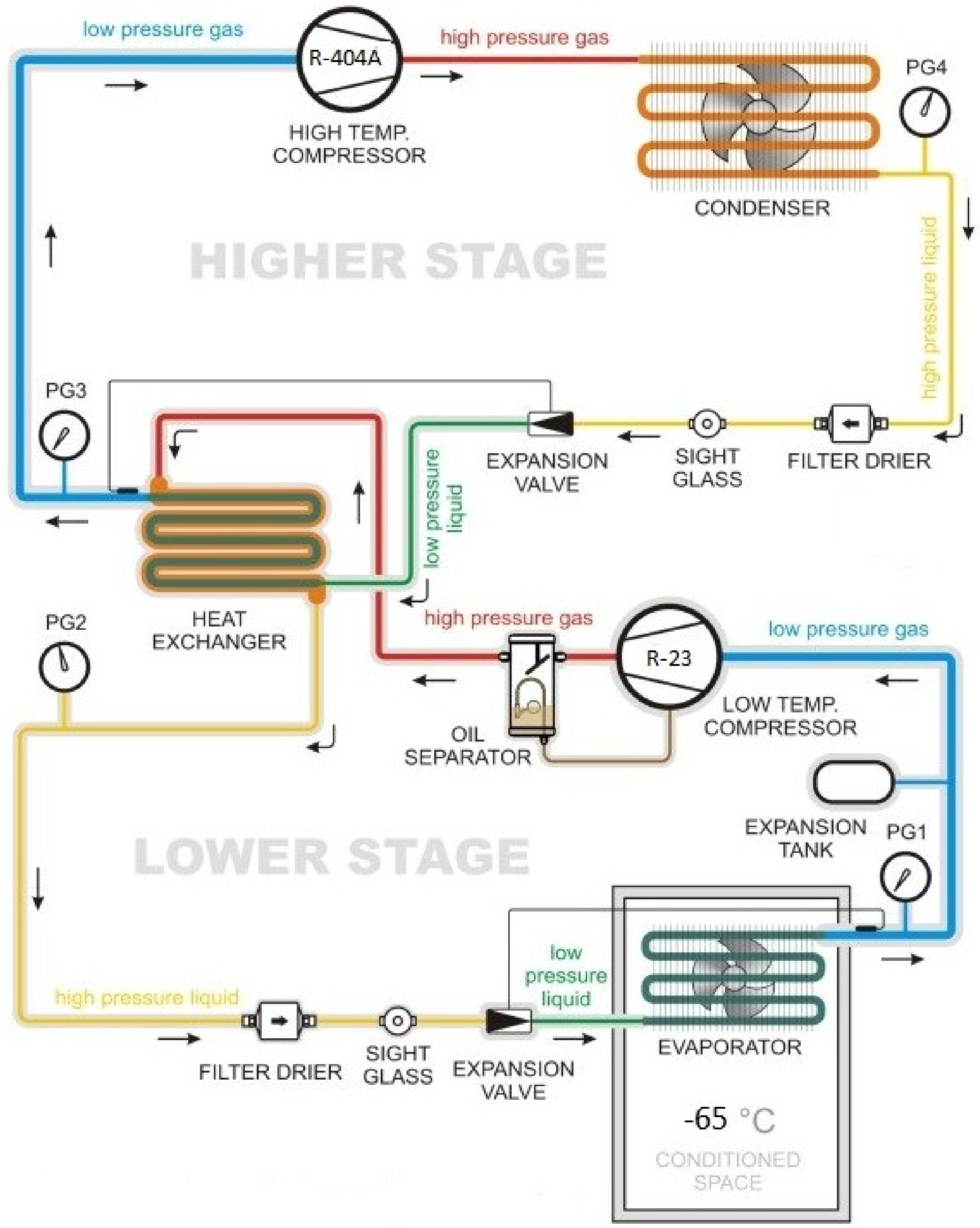

Figure 4.1 Cascade Refrigeration Cycle

4.2.1 Calculation For Low Stage Using R-23

Figure 4.2 Shows P-h chart for R-23[reference 10] .

$Q_e = \dot{m}_1 .\ q_e$ ……………………………………..…………..(4.1)

$Q_e = \dot{m}_1 \times (h_1 - h_4)$ ……………………………………..(4.2)

$\dot{m}_1 = \frac{Qe}{(h1-h4)}$

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]
h_4 : Enthalpy at point before evaporator [kJ/kg]

$\dot{m}_1 = \frac{0.675}{332.402-158.481} = 0.00388$ [kg/s]

$Q_c = \dot{m}_1 \times (h_2 - h_3)$ ……………………………………..(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]

$Q_c = 0.00388 \times (371.737 - 158.481)$
$Q_c = 0.827$ [KW] = 827 [W]

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

W_{c1} : low Compressor Work [W]

$W_{c1} = 0.00388 \times (371.737 - 332.402)$
$W_{c1} = 0.152$ [KW] = 152 [W]

In Hours Power = 152/746 = 0.2

That mean that we need 0.25 HP compressor , but we choose one third HP compressor because we choose compressor 404a for this refrigerant , which ΔP for R23 is bigger than ΔP for R404a .

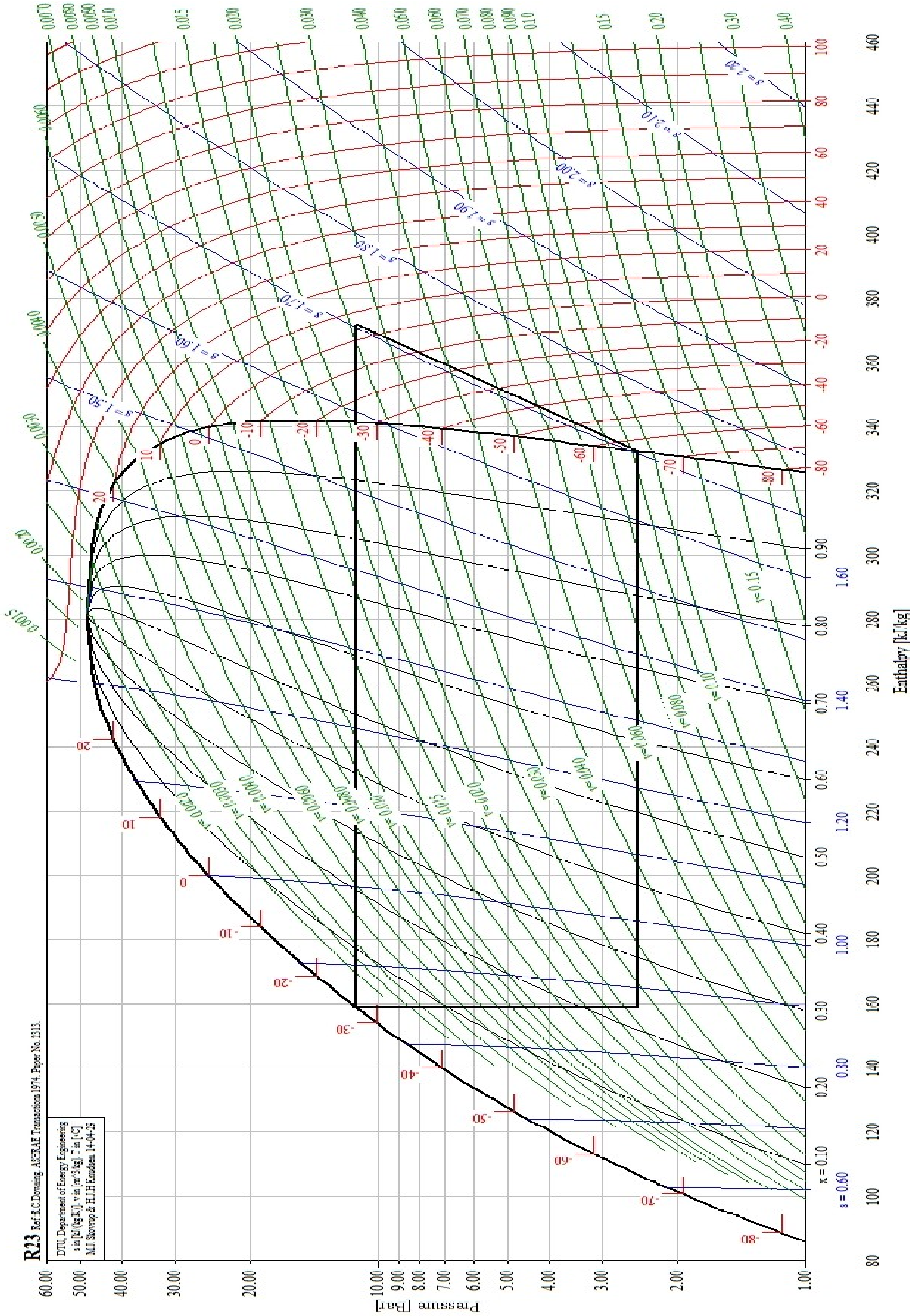

Figure 4.2 P-h chart for R-23

Values at points 1-6,15 for the selected one stage cycle

Point	T	P	v	h	s
	[°C]	[bar]	[m^3/kg]	[kJ/kg]	[kJ/(kg K)]
1	-65.000	2.481	0.092176	332.402	1.7040
2	5.247	11.198	0.026143	371.737	1.7040
3	5.247	11.198	0.026143	371.737	1.7040
4	-27.000	11.198	N/A	158.481	N/A
5	N/A	2.481	N/A	158.481	N/A
6	-65.000	2.481	0.092173	332.402	1.7040
15	N/A	11.198	N/A	158.481	N/A

Figure 4.3 properties of low cycle

4.2.2 Calculation For High Stage Using R-404a

We made energy balance in the heat exchanger inlet and outlet to calculate m^*_2 as shown in Figure 4.3 .[reference 10]

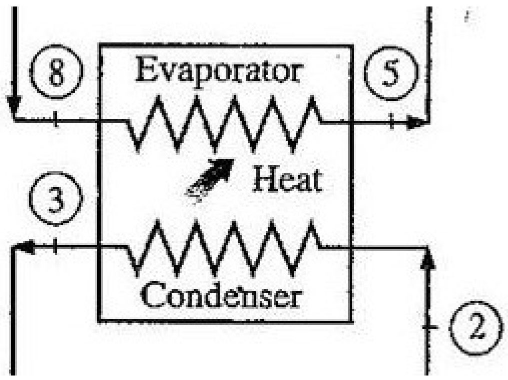

Figure 4.4 heat exchanger balance

Figure 4.5 shows P-h chart for R-507

The amount of heat inlet = the amount of heat outlet

$\dot{m}_2 \times (h_5 - h_8) = \dot{m}_1 \times (h_2 - h_3)$ ……………………………………..(4.5)

So,

$$\dot{m}_2 = \dot{m}_1 \frac{(h2-h3)}{(h5-h8)}$$

$\dot{m}_2 = 0.00388 \times \frac{(371.737 - 158.481)}{(350.304 - 254.208)}$

$\dot{m}_2 = 0.0086$ [Kg/s]

$Q_c = \dot{m}_2 \times (h_6 - h_7)$

$Q_c = 0.0086 \times (392.011 - 254.208)$

$Q_c = 1.185$ [KW] = 1185 [W]

$W_{c2} = \dot{m}_2 \times (h_6 - h_5)$

W_{c2} : High Compressor Work [W]

$W_{c2} = 0.0086 \times (392.011 - 350.304)$

$W_{c2} = 0.358$ [KW] = 358 [W]

In Hours Power = 358/746 = 0.5 HP

That mean that we need 0.5 HP compressor , but we choose Three quarters HP compressor because we took safety factor of 1.2 to be in safe side .

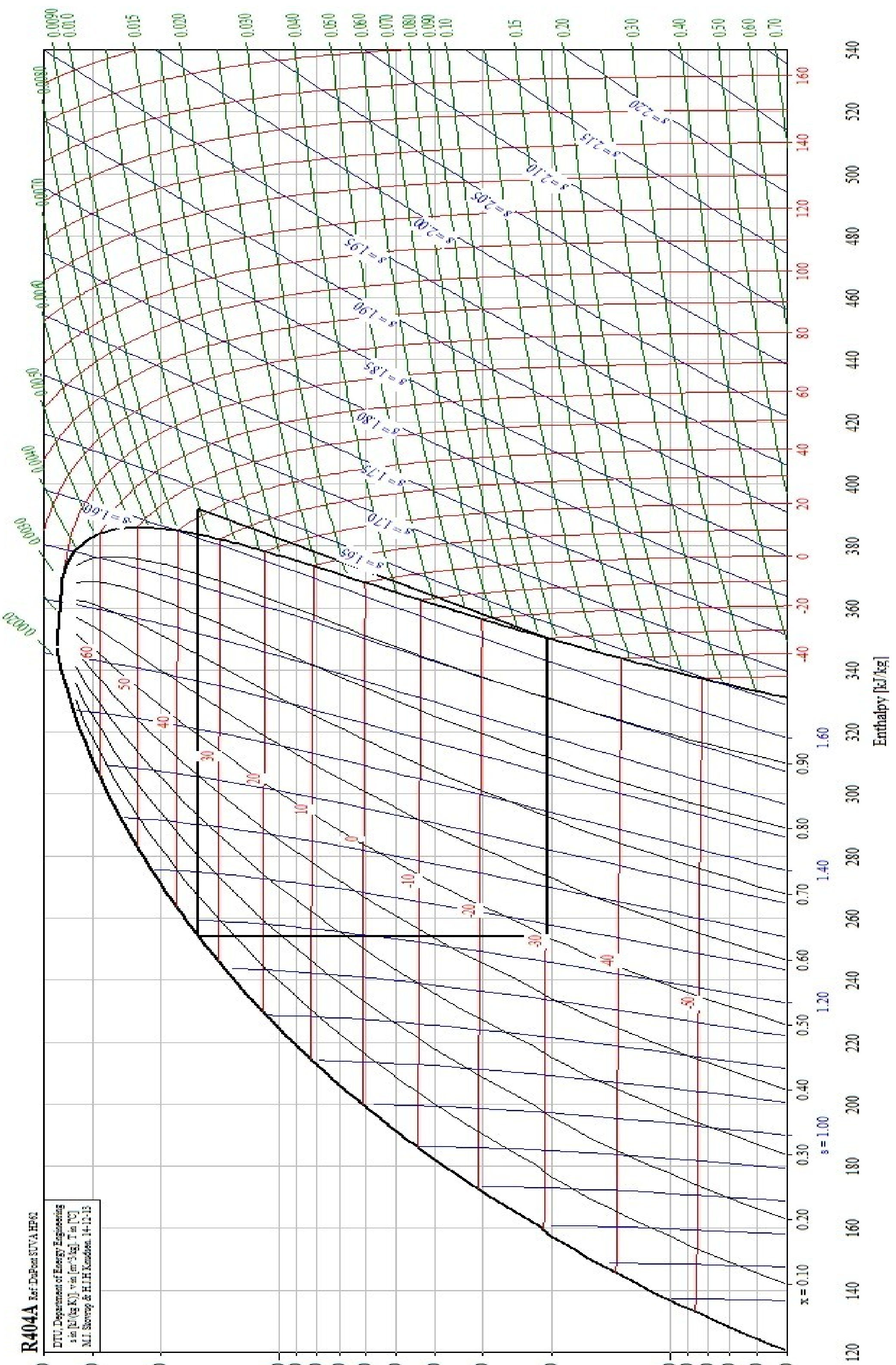

Figure 4.5 P-h chart for R-404a

Values at points 1-6,15 for the selected one stage cycle

Point	T	P	v	h	s
	[°C]	[bar]	[m^3/kg]	[kJ/kg]	[kJ/(kg K)]
1	-29.938	2.045	0.095129	350.304	1.6302
2	42.350	16.065	0.012375	392.011	1.6302
3	42.350	16.065	0.012375	392.011	1.6302
4	34.676	16.065	N/A	254.208	N/A
5	N/A	2.045	N/A	254.208	N/A
6	-29.938	2.045	0.095129	350.304	1.6302
15	N/A	16.065	N/A	254.208	N/A

Figure 4.6 properties of high cycle

4.2.3 Calculation Of Coefficient Of Performance (COP)

$Q_{condenser} = \dot{m}_2 \times (h_6 - h_7)$
$Q_{condenser} = 0.0086 \times (\ 392.0.11 - 254.208\ )$
$Q_{condenser} = 1.185$ [KW] = 1185 [W]

$Q_{evaporater} = 675$ [W]

$W_{total} = W_{c1} + Wc_2$
$W_{total} = 152 + 358$

$W_{total} = 510$[W]

$$COP = \frac{Qe}{Wc1+Wc2} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4.6)$$

$$COP = \frac{675}{510} = 1.32$$

CHAPTER 5

Cycle Design

CHAPTER FIVE

CYCLE DESIGN

5.1 Compressor Calculation And Selection

5.1.1 Calculation For Low Compressor

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

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

$\eta_c = 1 – 0.02\ [(\frac{11.198}{2.481})^{1/1}-1] = 93\%$

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]

$\eta_h = \frac{208}{246} = 84.5\%$

η_v= 93% * 84.5% = 78.5%

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

$\dot{V}_{theo} = \dot{m}_1 * \nu$ ………..……………………………………………………….. (5.4)

Where :

$\dot{V}_{theo}$: 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]

$\dot{V}_{theo}$= 0.00388 * 0.092176 = 3.57 * 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]

$\dot{V}_{act} = \frac{0.000357}{0.785}$=4.54 * 10^{-4} [m^3/s]

The main consider to select the compressor is the actual volumetric flow rate , so we will chose a compressor that satisfy it in the next semester .

5.1.2 Calculation For high compressor

$\eta_v = \eta_c * \eta_h$

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

$\eta_c = 1 - 0.02\,[(\frac{16.065}{2.045})^{1/1} - 1\,] = 86.28\%$

$\eta_h = \frac{243}{308} = 78.8\%$

$\eta_v = 86.28\% * 78.8\% = 67.98\%$

$\dot{V}_{theo} = \dot{m}_2 * v$

$\dot{V}_{theo} = 0.0086 * 0.0956 = 8.22 * 10^{-4}$ [m^3/s]

$\dot{V}_{act} = \frac{\dot{V}_{the.}}{\eta v}$

$\dot{V}_{act} = \frac{0.000822}{0.6798} = 1.21 * 10^{-3}$ [m^3/s]

We will chose the compressor in the next semester .

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 . [reference13]

5.2.1 Introduction

To calculate the inner diameter for the pipe can be used the following steps [reference14] .

$Q = \dot{m} * \nu$ …………………………………………………………………… (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 [reference13]

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

5.2.2 Low Cycle Pipe Calculation

♦ Suction line pipe

Q = $\dot{m}_1 * v_1$

Q = 0.00388 * 0.092176 =3.578 * 10^{-4} [m^3 / s]

A $= \frac{Q}{V} = \frac{3.578 * 10^{-4}}{10}$ = 3.578 * 10^{-5} [m^2]

$d_i = \sqrt{\frac{3.578 * 10^{-5} * 4}{\pi}}$ = 6.75 * 10^{-3}[m]

r_i = 3.375[mm]

$\frac{700}{8} = \sqrt{{\delta_t}^2 + 2.481\, \delta_t + 2.481^2}$

δ_t = 86.26[bar]

$86.26 = \frac{2.481\, ({r_o}^2 + (3.375 * 10^{-4})^2)}{{r_o}^2 - (3.375 * 10^{-4})^2}$

r_o = 3.468 [mm]

T = $r_o - r_i$

T =3.468 – 3.375 = 0.093 [mm]

wheret is the thickness

♦ The inner and outer radius in inch is

$r_i = r_i inmm$ / 25.4 = 0.132 inch

$r_o = r_o inmm$ / 25.4 = 0.136 inch

♦ Discharge line pipe

$Q = \dot{m}_1 * v_2$

$Q = 0.00388 * 0.026143 = 1.01 * 10^{-4}$ [m^3 / s]

$A = \frac{Q}{V} = \frac{1.01 * 10^{-4}}{15} = 6.73 * 10^{-6}$[m^2]

$d_i = \sqrt{\frac{6.73 * 10^{-6} * 4}{\pi}} = 2.92 * 10^{-3}$[m]

$r_i = 1.46$[mm]

$\frac{700}{8} = \sqrt{{\delta_t}^2 + 11.198\,\delta_t + 11.198^2}$

$\delta_t = 82.08$ [bar]

$82.08 = \frac{11.198\,({r_o}^2 + 1.46^2)}{{r_o}^2 - 1.46^2}$

$r_o = 1.54$ [mm]

T =1.58– 1.46 = 0.12 [mm]

♦ The inner and outer radius in inch is

$r_i = r_i inmm$ / 25.4 = 0.057 inch

$r_o = r_o inmm$ / 25.4 = 0.062 inch

♦ Liquid line pipe

$Q = \dot{m}_1 * v_2$

$Q = 0.00388 * 0.001 = 3.88 * 10^{-6}$ [m^3 / s]

$A = \frac{Q}{V} = \frac{3.88 * 10^{-6}}{10} = 3.88 * 10^{-7}$ [m^2]

$d_i = \sqrt{\frac{3.88 * 10^{-7} * 4}{\pi}} = 7.03 * 10^{-4}$[m]

r_i = 0.35[mm]

$$\frac{700}{8} = \sqrt{{\delta_t}^2 + 11.198\,\delta_t + 11.198^2}$$

δ_t = 82.08 [bar]

$$82.08 = \frac{11.198\,({r_o}^2 + 0.35^2)}{{r_o}^2 - 0.35^2}$$

$r_o = 0.4$ [mm]

t = 0.4 –0.35 = 0.05 [mm]

♦ The inner and outer radius in inch is

$r_i = r_i inmm$ / 25.4 = 0.0137 inch

$r_o = r_o inmm$ / 25.4 = 0.0157 inch

5.2.3 HighCycle Pipes Calculation

♦ Suction line pipe

Q = $\dot{m}_2 * \nu_1$

Q =0.0086 * 0.09563 =8.22 * 10^{-4} [m^3 / s]

$$A = \frac{Q}{V} = \frac{8.22 * 10^{-4}}{10} = 8.22 * 10^{-5} \; [\, m^2 \,]$$

$$d_i = \sqrt{\frac{8.22 * 10^{-5} * 4}{\pi}} = 0.01 \; [\text{ m }]$$

r_i = 5[mm]

$$\frac{700}{8} = \sqrt{{\delta_t}^2 + 1.973\,\delta_t + 1.973^2}$$

δ_t = 86.519[bar]

$86.519 = \frac{1.973\ (r_o{}^2 + 5\ ^2)}{r_o{}^2 - 5\ ^2}$

$r_o\ =\ 5.23$[mm]

t = 5– 5.23 =0.23 [mm]

♦ The inner and outer radius in inch is

$r_i\ = r_i inmm$ / 25.4 = 0.196 inch

$r_o\ = r_o inmm$ / 25.4 = 2.04 inch

♦ Discharge line pipe

Q = $\dot{m}_2 *\ v_2$

Q = 0.0086 * 0.011348 = $9.76 * 10^{-4}$ [m^3 / s]

A = $\frac{Q}{V} = \frac{9.76 * 10^{-4}}{15}$ = 6.5* 10^{-5} [m^2]

$d_i = \sqrt{\frac{6.5 * 10^{-5} * \ 4}{\pi}}$ = 9.1 * 10^{-3}[m]

r_i = 4.55[mm]

$\frac{700}{8} = \sqrt{\delta_t{}^2\ +\ 18.795\ \delta_t\ +\ 18.795^2}$

δ_t = 78.605[bar]

$78.605 = \frac{18.795(r_o{}^2 + 4.55^2)}{r_o{}^2 – 4.55^2}$

$r_o\ = 4.67$ [mm]

t = 4.67–4.55 = 0.12 [mm]

♦ The inner and outer radius in inch is

$r_i\ = r_i inmm$ / 25.4 = 0.18 inch

$r_o\ = r_o inmm$ / 25.4 = 0.184 inch

♦ Liquid line pipe

Q = $m_i * v_2$

Q =0.0086 * 0.001 = $8.6 * 10^{-6}$ [m^3 / s]

A = $\frac{Q}{V} = \frac{8.6 * 10^{-6}}{10} = 8.6 * 10^{-7}$ [m^2]

$d_i = \sqrt{\frac{8.6 * 10^{-7} * 4}{\pi}} = 1.046 * 10^{-3}$[m]

r_i = 0.523[mm]

$\frac{700}{8} = \sqrt{{\delta_t}^2 + 18.795\, \delta_t + 18.795^2}$

δ_t = 78.6[bar]

$888 = \frac{18.795\, ({r_o}^2 + 0.523^2)}{{r_o}^2 - 0.523^2}$

$r_o = 0.66$[mm]

t = 0.66– 0.523 =0.137 [mm]

♦ The inner and outer radius in inch is

$r_i = r_i inmm$ / 25.4 = 0.02 inch

$r_o = r_o inmm$ / 25.4 = 0.026 inch

5.3 Evaporator calculations and selection

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

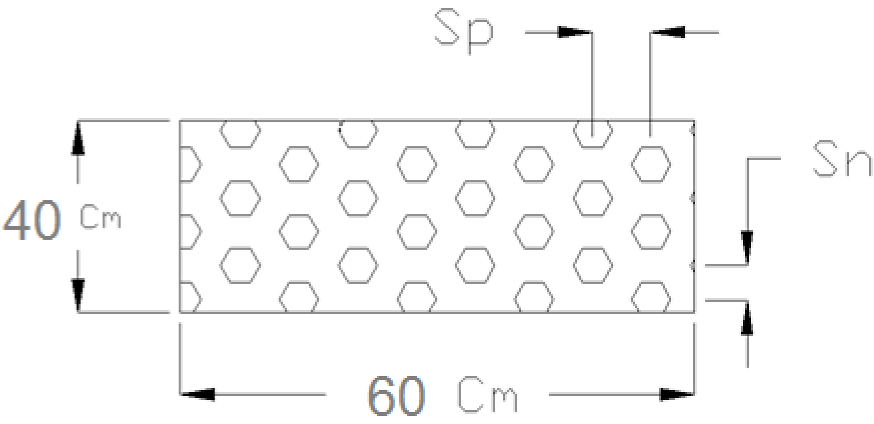

Figure 5.1 evaporator side view

Evaporator length L_e = 60[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]

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

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

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

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

Figure (5.2) show the fin elements

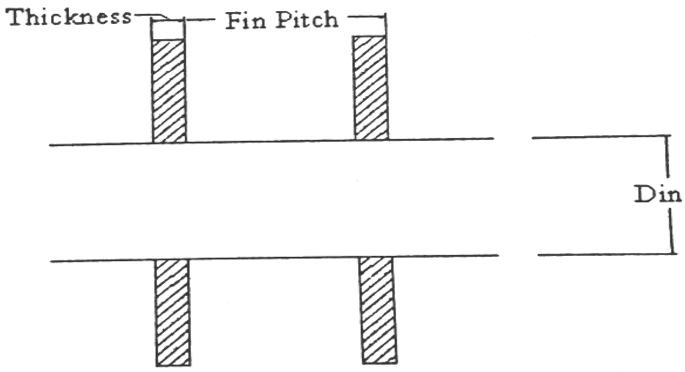

Figure 5.2 Fine elements for evaporator

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 = 89 fins

Fin pitch = $P_f = \frac{53}{89} = 0.59$ [cm]

Fine thickness = $t_f = 0.03$ [cm]

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

$t_b = 0.59 - 0.03 = 0.56$ [cm]

Design the evaporator required 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 .

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

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 [^oC]

T_∞ : free air temperature [^oC]

The heat transferred through the evaporator was determined in chapter 2 and its value wasQ_e= 675[W].

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

$$h_o = \frac{N_u * K}{D} \text{ ....(5.17)}$$

Where

Nu :nusselts number

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

D : Outer diameter of evaporator [m]

The nusselts 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

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

Where

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

V_{max} : maximum velocity of air between the evaporator rubes [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]

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

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

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

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]

$V_\infty = \frac{0.02359}{0.0125} = 1.887\text{m/s}$

$V_{max} = 1.887 * \frac{0.025}{0.025-0.01} = 3.145$ [m/s]

The properties of air are evaluated at the film temperature , which at entrance to the tube bank is [reference 14] .

$$Tf = \frac{Tw + T\infty}{2} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (5.20)$$

Where

Tf : film temperature [˚C]

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

T∞ : free air temperature [˚C]

$$Tf = \frac{-65 + -35}{2} = -50\ [˚C] = 223[˚K]$$

Then from table A-9

ρ=1.5 [Kg/m^3]

$\mu = 1.516 * 10^{-5}$ [Kg/m.s]

k = 0.02 [W/m.˚C]

Pr =0.73

$$Re = \frac{1.5*3.145*0.01}{1.516*10^{-5}} = 3111.8$$

$Nu = 0.3\ (3111.8)^{0.595} * (0.73)^{1/3} = 32.35$

$$h_o = \frac{32.35 * 0.02}{10*10^{-3}} = 64.7\ [W/m^2 °C]$$

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

$$q_{total} = q_{fin\ act} + q_{original} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (5.21)$$

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

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

Where

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

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

T_w : outer evaporator wall temperature [˚C]

T_∞ : free air temperature

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

$q_{original} = 64.7 * 1.7 * 10^{-4} * (-35 - -65) = 0.33$ [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]

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

$q_{fin} = 64.7 * 10 * 10^{-4} (-35 - -65) = 1.9$ [W]

Fin efficiency calculation :

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

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

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

$$m=\sqrt{\frac{h\,P}{k\,A}}\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(5.24)$$

Where

h: external convection heat transfer coefficient [$W/m^2{}^\circ C$]

k: thermal conductivity of aluminum fin , [$W/m^\circ C$]

P : perimeter of the fin[m]

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

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

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

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

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

So

The heat transfer flow from the fin is

$$q_{finact}=q_{fin}*n_f\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(5.25)$$

$q_{finact} = 1.9 * 0.93 = 1.767$ [W]

Now the total heat transfer from the element is

$q_{total} = 1.767 + 0.33 = 2.09$ [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:

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

$$n = \frac{756.568}{2.09} = 362 \text{ elements}$$

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 = 89*2=178 elements [that is not enough] . So we can use same evaporator with 5 rows .

Number of elements = 89 * 5 = 445 and that's is enough .

5.4 Condenser calculations and selection

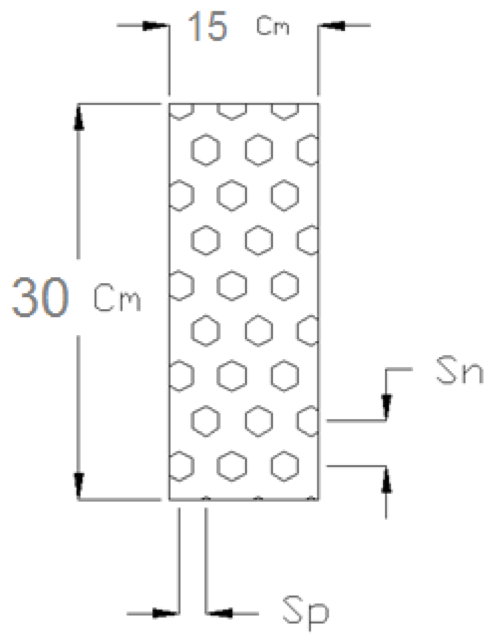

Figure 5.3 Condenser side view

Figure 5.3 description the available condenser , and the following is the geometrical data

Condenser length L_c = 38 [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]

$$S_n = \frac{condenser\ \ height}{numperofrows} \quad \ldots\ldots(5.24)$$

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

$$S_p = \frac{condenserwidt\ \ h}{numperofcolumn} \quad \ldots\ldots(5.25)$$

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

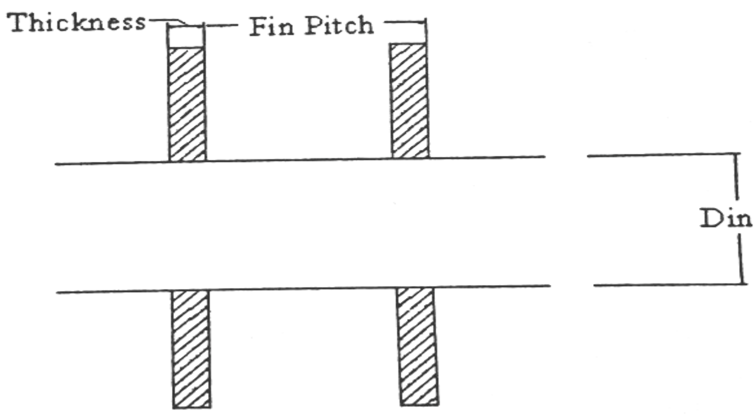

Figure 5.4 fin elements for condenser

Figure 5.4 show the fin elements

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 = 94 fins

Fin pitch $= P_f = \frac{condenserleingt\ \ h}{numperoffins}$ ………………………………………………..(5.28)

$= \frac{38}{94} = 0.4$ [cm]

Fin thickness $= t_f = 0.02$ [cm]

Bare tube thickness $= t_b = P_f – t_f$ ………………………………………………..(5.29)

$= 0.4– 0.02 = 0.38$ [cm]

The project need one condenser in the high stage cycle .

♦High region condenser

$Q_c = \dot{m}_2 \times (h_6 – h_7)$…………………………………………………………….(5.30)

Where :

$\dot{m}_2$: mass flow rate in the condenser

h_6 : Enthalpy at point before compressor [KJ/Kg]

h_7 : Enthalpy at point after condenser with desuperheating [KJ/Kg]

Q_c : Heat transfer rate in condenser (condenser load) [W]

$h_7 = 254.208$ [KJ/Kg] , $h_6 = 392.011$

$Q_c = 0.0086\times(392.011 – 254.208) = 1.18$ [KW]

$Q_c = h_o * A * (T_w – T_\infty)$ …………………………………………………….…(5.31)

Where :

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

A : Surface area of heat transfer [m2]

T_w: Outer condenser wall temperature

T_∞ : Free air temperature

S_p / D = 19 /10 = 1.9 [cm]

S_n / D = 26 / 10 = 2.6 [cm]

From staggered arrangement tube banks table [Table A8]

C = 0.521

N = 0.565

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

$$V_{max} = \frac{V\infty\,(\frac{Sn}{2})}{\sqrt{[(\frac{Sn}{2})^2 + Sp^2]} - D} \quad \text{……(5.32)}$$

$$V\infty = \frac{V}{A} \quad \text{……(5.33)}$$

Where :

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

V= 700 [m3/h] = 0.19444 [m^3 / s] From fan manufacture company .

A = Cross sectional area of condenser [m^2]

A = 0.3 * 0.26 = 0.078 [m^2]

$$V\infty = \frac{V}{A} = \frac{0.19444}{0.078} = 2.492 \text{ [m/s]}$$

$$V_{max} = \frac{2.492 * (\frac{0.026}{2})}{\sqrt{[(\frac{0.026}{2})^2 + 0.019^2]} - 0.01} = 2.48 \text{ [m/s]}$$

The properties of the air are evaluated at the film temperature , which at entrance to the tube bank is :

$Tf = \frac{Tav.w + T\infty}{2}$ ……………………………………………………………..……..(5.34)

Where :

Tf : Film temperature [^oC]

T∞ : Free air temperature [^oC]

Tav.w :Avarege wall surface temperature [^oC]

$Tav.w = \frac{45+55}{2} = 50$ [^oC]

$Tf = \frac{50+38}{2} = 44$ [^oC]

Then from Table A-9

$\mu = 1.85 * 10^{-5}$ [kg / m^3]

$\rho = 1.1176$ [kg / m^3]

K = 0.0275 [W / m.^oC]

Pr = 0.704

So ,

$Re = \frac{1.1176*2.48*0.01}{1.85*10^{-5}} = 1497.4$

$Nu = 0.521 * (1497.4)^{0.565} * (0.73)^{1/3} = 29.2$

$ho = \frac{29.2*0.0275}{10*10^{-3}} = 80.3$ [W/m^2 ^oC]

This is the heat transfer coefficient that would be obtained if there 10 rows of tubes in the direction of the flow . Because there are only three rows , this value must be multiplied by the factor of 0.83 , as determined from Table A-10 [Reference 14]

So, ho = 80.3 * 0.38 = 66.6 [$W/m^{2o}C$]

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

$q_{original}$ can be calculated by the equation

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

$A_{original} = \pi DL = 3.14 * 10 * 10^{-3} * 3 * 10^{-3} = 94.24 * 10^{-6} [m^2]$

$q_{origenal} = 66.6 * 94.24 * 10^{-6} * (50 – 38) = 0.0753 [W]$

q_{fin} can be calculated by the equation :

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

$A_{fin} = 2 (Sn * (Sp – A_{pip})) = 2 * [0.026 * 0.019 - \frac{\pi}{4} (0.01)^2] = 8.31 * 10^{-4} [m^2]$

qfin = 66.6 * 8.3 * 10-4 * (50-38) = 0.6633 [W]

Fin efficiency calculation :

$$`L_f = \left(\frac{0.016}{2}\right)\left[1 + 0.35 \ln \frac{(\frac{0.01}{2}+\frac{0.016}{2})}{\frac{0.01}{2}}\right] = 0.01[\,m]$$

$$m = \sqrt{\frac{66.6 * 0.052}{202 * 5.2 * 10^{-6}}} = 57.4$$

$$nf = \frac{\tan h(57.4 * 0.01)}{57.4 * 0.01} = 0.90$$

So the actual heat transfer flow through the fins is :

$q_{fin\ act} = 0.633 * 0.9 = 0.597$ [W]

Now the total heat transfer from the element is

$q_{total} = 0.597 + 0.0753 = 0.6723$ [W]

Now the number of elements that need to perform the condenser load to condensation process can be determined by the following equation :

$$n = \frac{Qcondensation}{Qtotal} \quad ………………………………………………………..………(5.35)$$

$= \frac{94}{0.6723} = 140\ elements$

5.5 Heat Exchanger design

The prime objective in the design of a heat exchanger is to determine the surface area required for specified duty (rate of heat transfer) using the temperature differences variable , diameter of the pipe and length will be calculate[reference 14] .

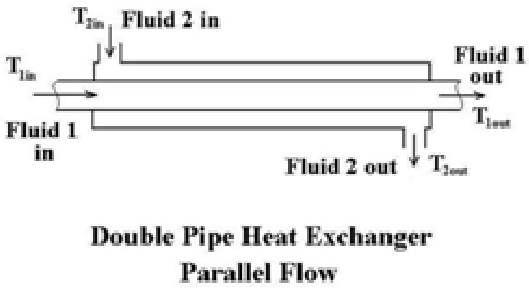

Figure 5.5 Double pipe heat exchanger parallel flow

Heat exchanger diameter selection :

Diameter R23 selection :

Di = 0.5 " = 12.5 [mm]

Ai = 1.23×10^{-4} [m^2]

Diameter R404selection :

Do = 1 " = 25 [mm]

Ao = 4.9×10^{-4} [m^2]

Condensation conviction heat transfer coefficient :

$$h_c = 0.725\ x\ \left[\frac{\rho\ (\rho-\rho v)g\ h_{fg}\,k^3}{\mu\ D\ \Delta T}\right]^{1/4} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(5.36)$$

Where :

ρ : Refrigerant Density

v : Specific Volume

g :Acceleration of gravity

k : Thermal conductivity

D : Inner Diameter

μ : Viscosity

$$h_c = 0.725\ x\ \left[\frac{40.4\ (40.4-1)9.81\ x\ 210.4\ x\ 10^3 x\ \ 386^3}{0.11\ x\ 10^{-3}\ x\ 12.5\ x\ 33}\right]^{1/4}$$

h_c= 1841.2 [W/mC^o]

Boiling conviction heat transfer coefficient :

$$h_b = 2.7\ x\ \left[\frac{V\ Kv\ pv\ \lambda}{Do\ \Delta T}\right]^{1/4} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(5.37)$$

Where :

ρ : Refrigerant Density

V : Specific Volume

g :Acceleration of gravity

k : Thermal conductivity

D : Outer Diameter

λ : Latent Heat

$$h_b = 2.7\ x\ \left[\frac{0.03961\ x\ 0.08\ x\ 25.25\ x\ 23}{25\ x\ 10^{-3}\ x\ 4}\right]^{1/4}$$

h_b= 1158.26 [W/mC^o]

Overall Heat-Transfer Coefficient calculation For Outer Area :

$$\mathbf{Uo} = \frac{1}{\frac{Ao}{Ai\ hi} + \frac{Ao\ ln\ (\frac{ro}{ri})}{2\pi kL} + \frac{1}{ho}} \quad \text{...(5.38)}$$

$$\mathbf{Uo} = \frac{1}{\frac{4.9}{1.23\ x\ 1841.2} + 0 + \frac{1}{1158.26}}$$

Uo = 327 W/m^2 C^o

Heat exchanger length calculations :

$$\text{LMTD} = \frac{(T_{R23,i} - T_{R404,o}) - (T_{R23,o} - T_{R4047,i})}{\ln\frac{(T_{R23,i} - T_{R404,o})}{(T_{R23,o} - T_{R404,i})}} \quad \text{..(5.38)}$$

Where :

LMTD : log mean temperature difference

$T_{R23,i}$: temperature of the refrigerant R23 that entering the heat exchanger .

$T_{R507,o}$: temperature of the refrigerant R507 that leaving the heat exchanger .

$T_{R23,o}$: temperature of the refrigerant R23that leaving the heat exchanger .

$T_{R507,i}$: temperature of the refrigerant R507 that entering the heat exchanger .

$$\text{LMTD} = \frac{(5 - -20\ ) - (\ -27 - -32)}{\ln\frac{(5 - -20\ )}{(\ -27 - -32)}} = 13.3[\ ^oC\ ]$$

$$A = \frac{Qc}{U\ x\ LMDT} \quad (5.39)$$

Where :

U : the overall heat transfer coefficient .

A : the overall cross-sectional area for heat transfer.

Q : the rate heat of transfer (Qc for low cycle) .

LMDT : Log Mean Temperature Difference.

$$A = \frac{675}{(327\ x\ 13.3\ x\ 10^2)} = 1.55\ x\ 10^{-3}\ [\ m^2\ ]$$

$$A = \frac{\pi\ x\ Do^2\ x\ L}{4} = \text{Ao x L}$$

L = A / Ao

$L = 1.55 \text{ x } 10^{-3} / 4.9 \text{ x } 10^{-4}$

L = 3 [m]

5.6 Capillary tube

After searching in the market and internet websites the following dimensions has been selected .

* For the higher cycle :

D = 0.048 inch .
L = 5 meter .

* For the lower cycle :

D = 0.048 inch .
L = 3 meter .

CHAPTER 6

Drawings OfRefrigerator

6.1 Refrigerator Drawings .

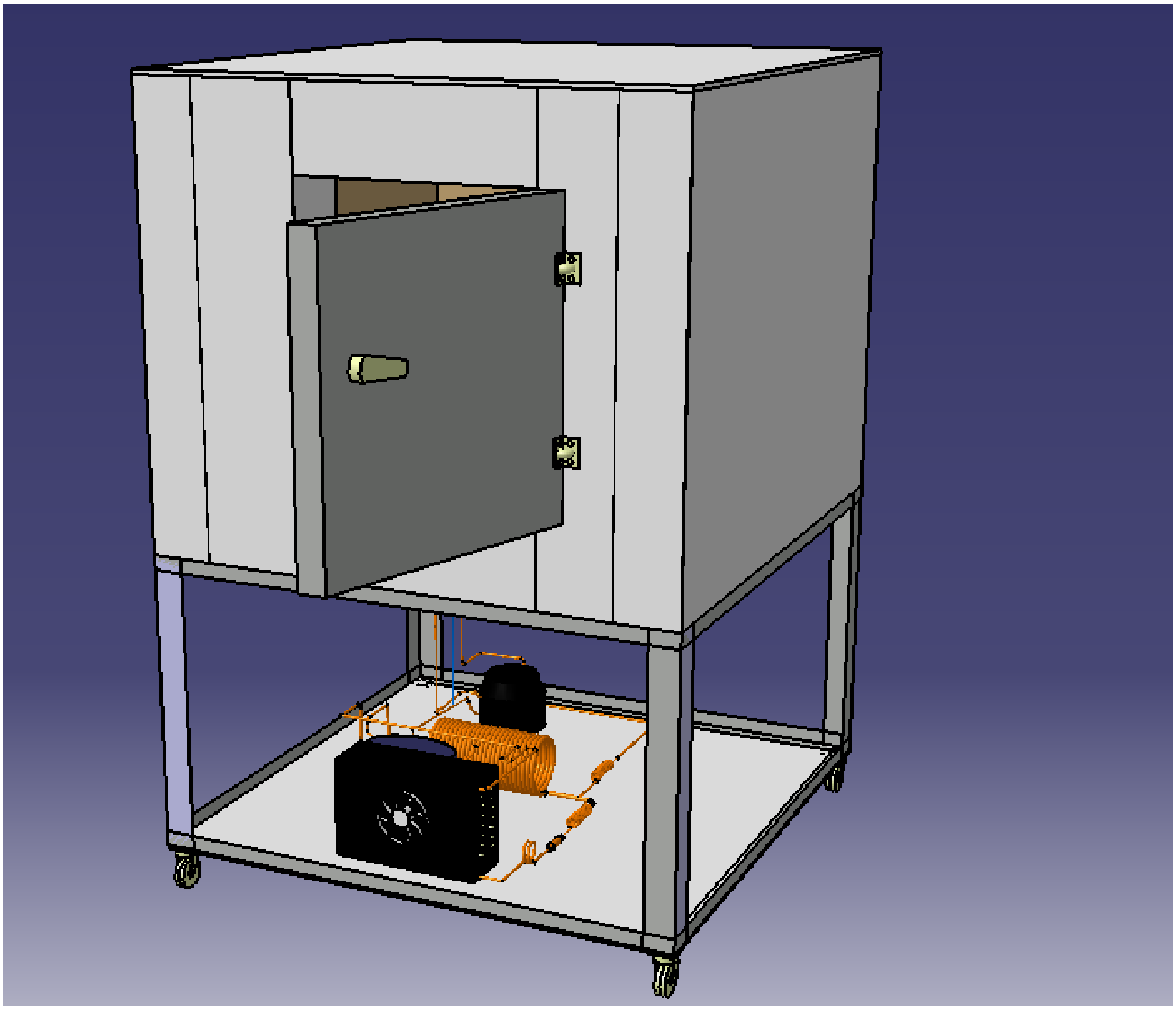

Figure 6.1 Refrigerator Drawing

AutoCAD

6.2 Refrigerator Views .

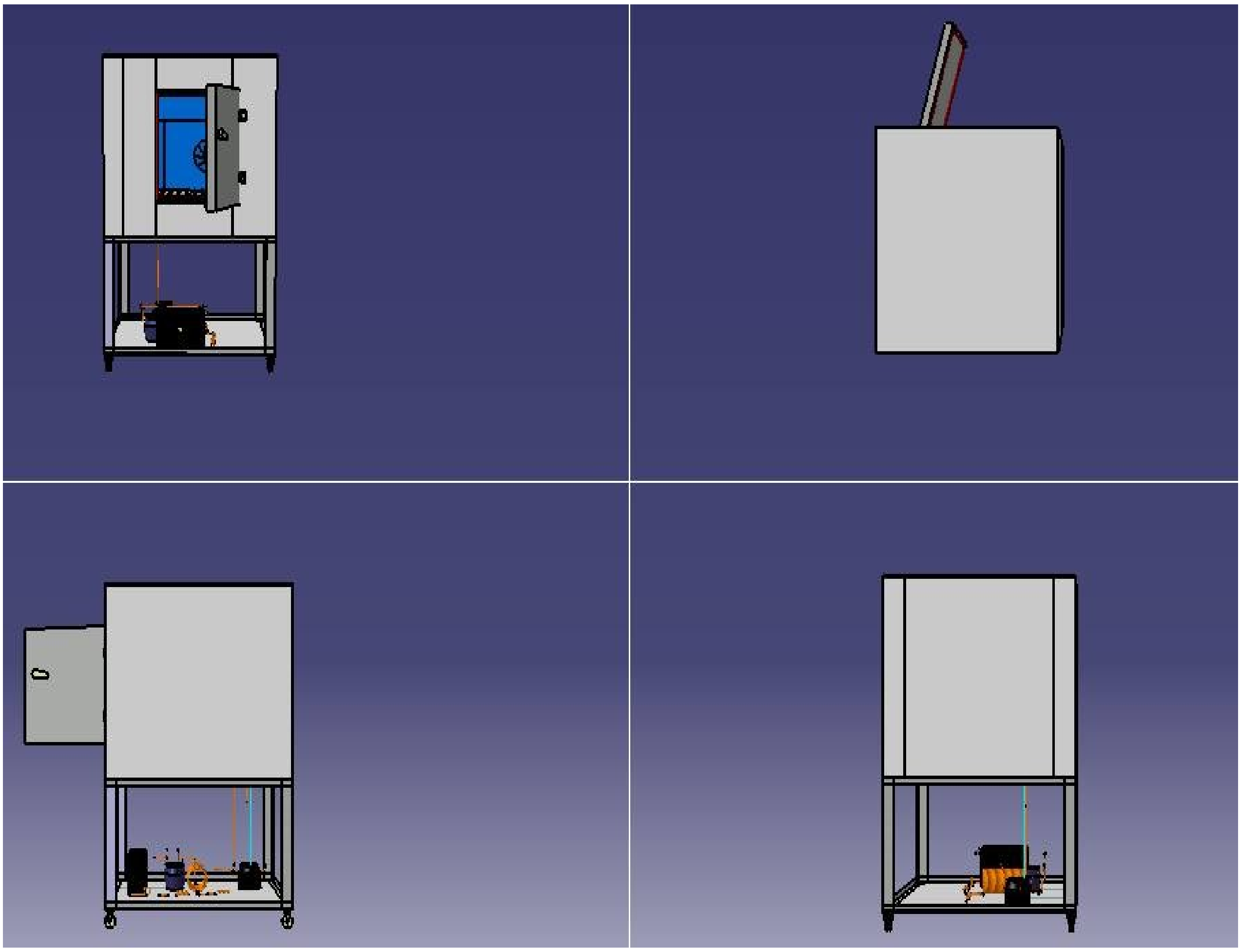

Figure 6.3 Refrigerator Views

AutoCAD

AutoCAD

AutoCAD

AutoCAD

6.3 Chamber Drawings .

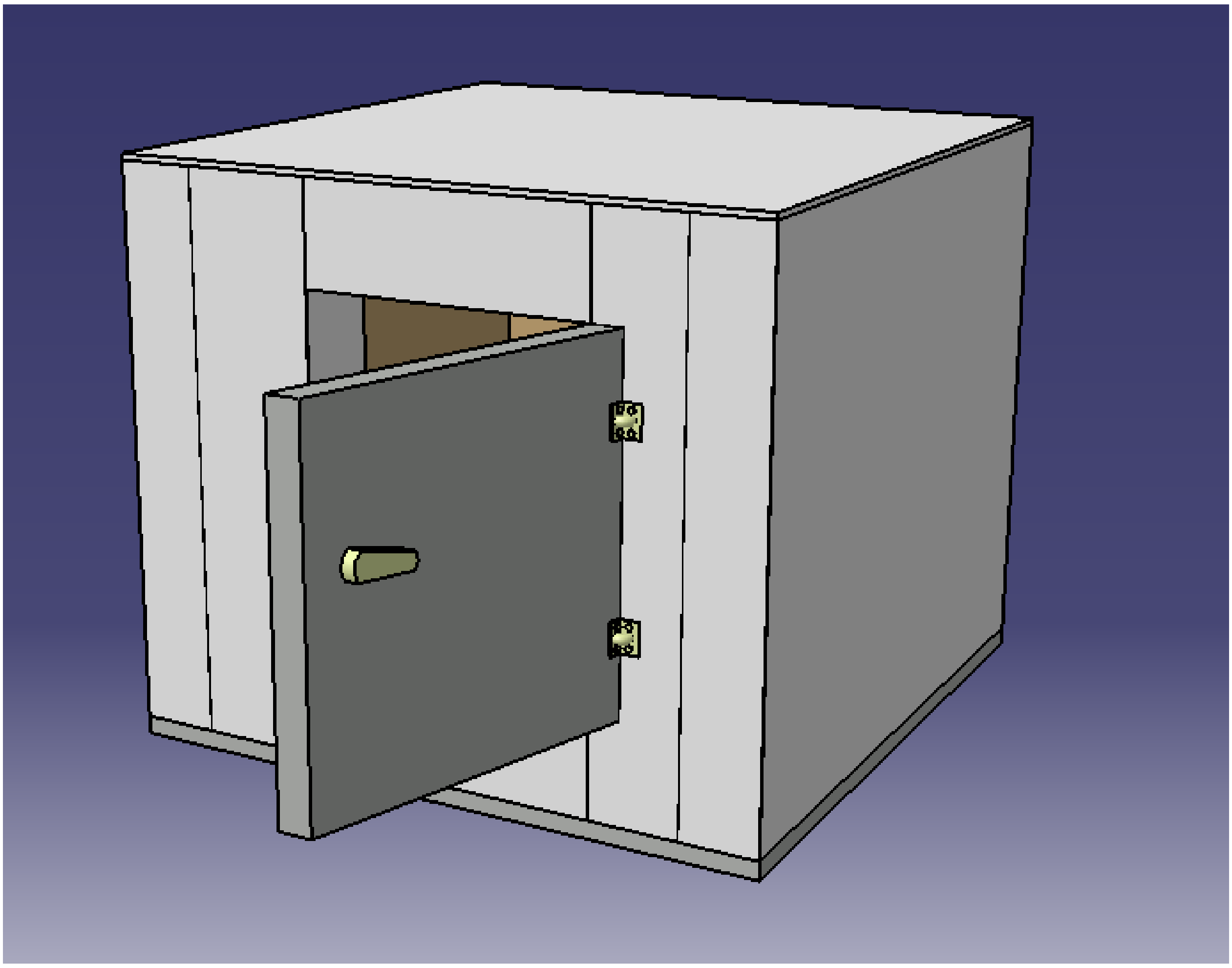

Figure 6.8 Chamber Drawing

AutoCAD

6.4 Door Drawings .

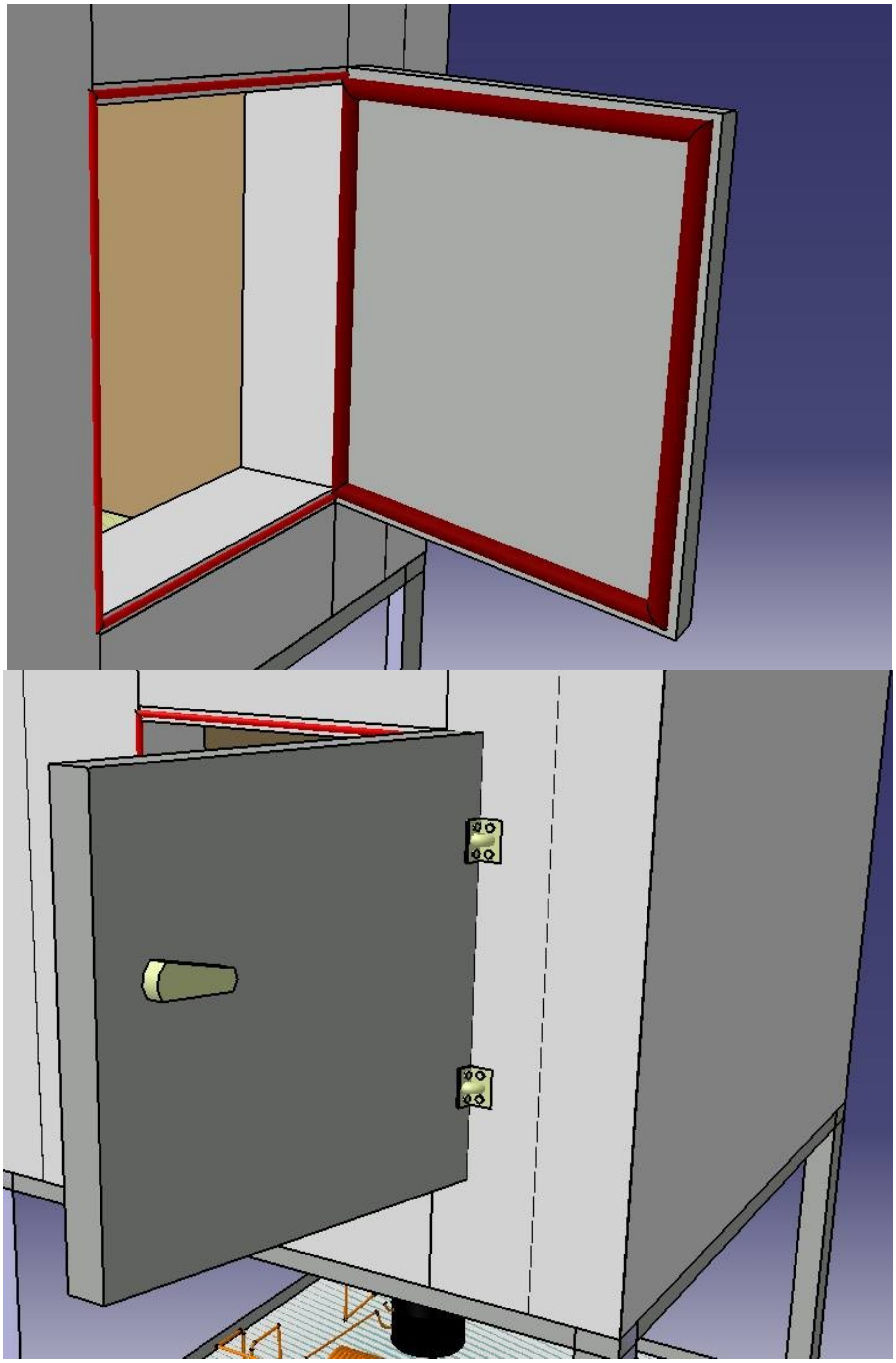

Figure 6.10 Door Drawings

AutoCAD

6.5 Stand Drawings .

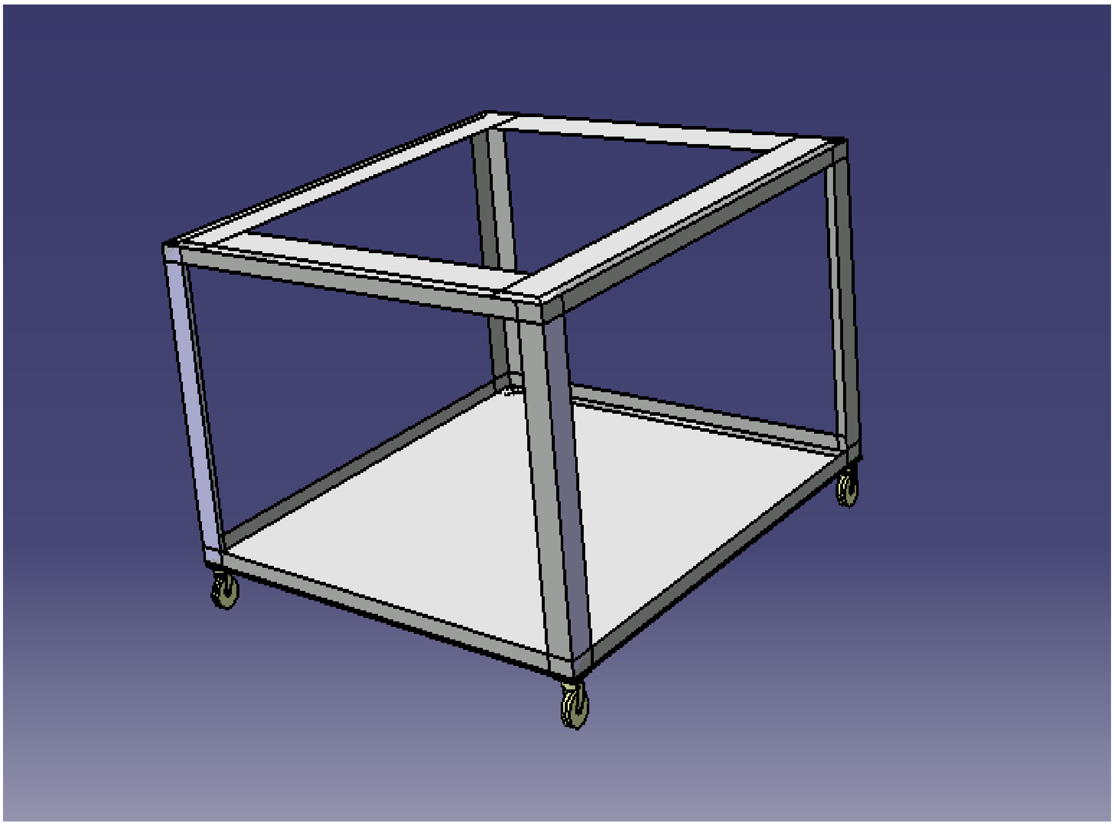

Figure 6.12 Stand Drawing

AutoCAD

6.6 Cycle Drawings .

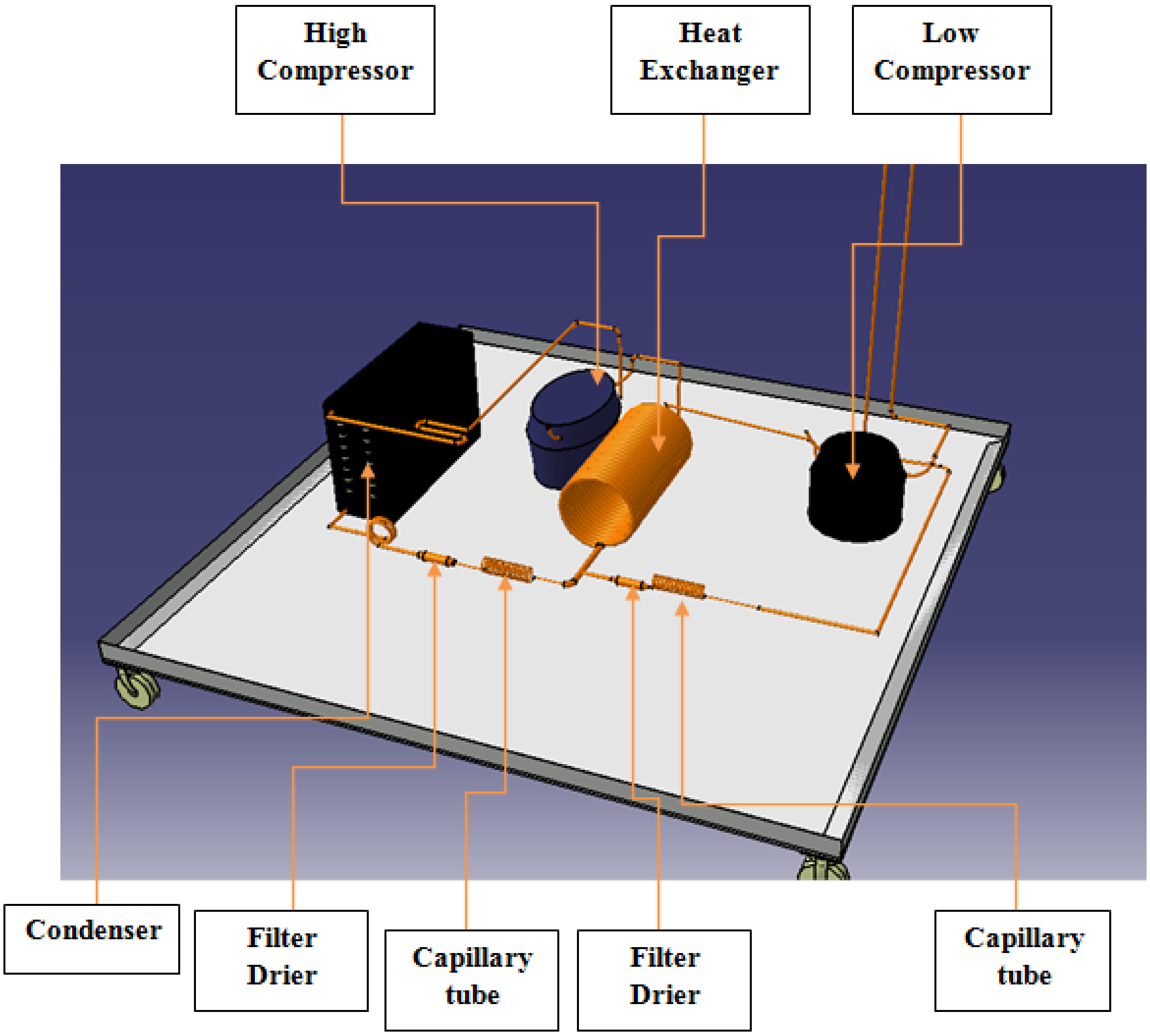

Figure 6.14 Cycle Drawing

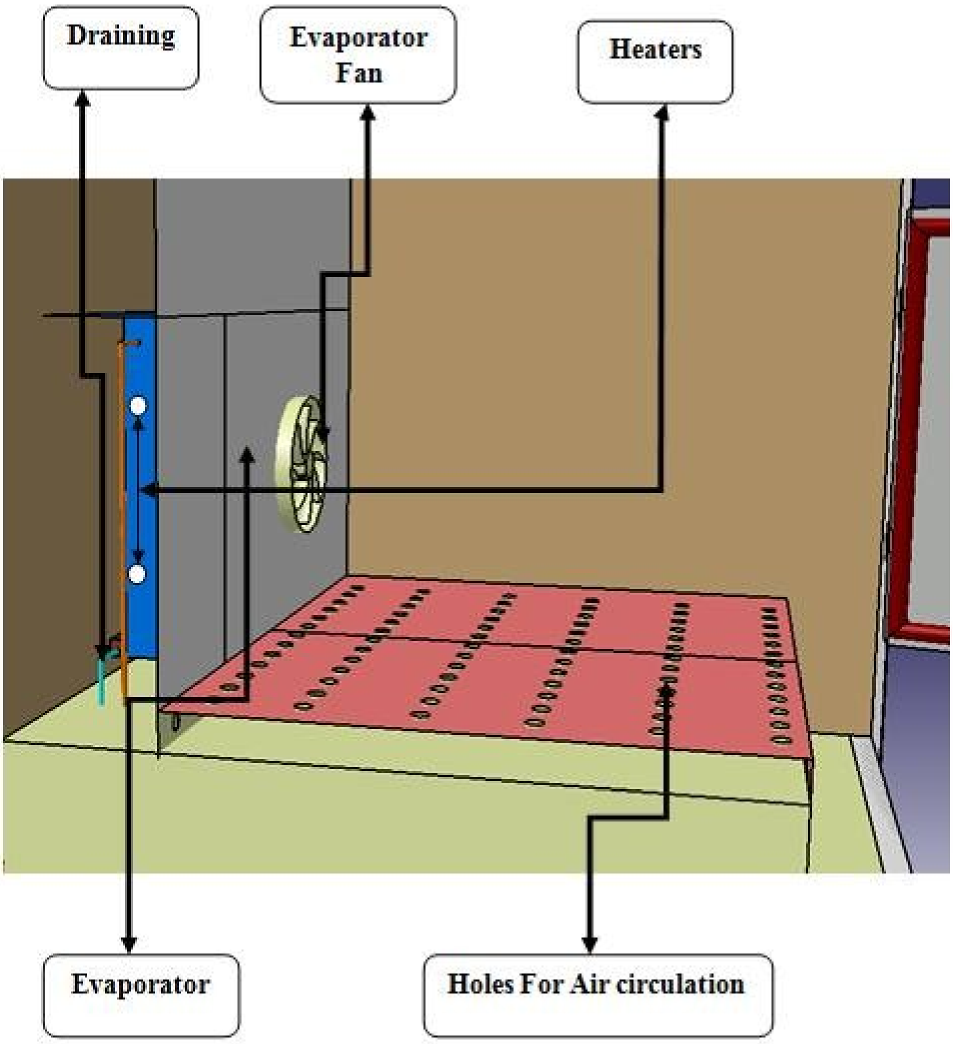

Figure 6.15 Cycle Drawing

CHAPTER 7

ELECTRICAL DESIGN

7.1 Introduction

At the beginning of this chapter we examine the basic equipment for cooling cycles as well as organizations and accessories that enable control over the operation and characteristics of refrigeration units . In order to obtain the highest efficiency and the safe operating conditions for the refrigeration unit and person and to achieve the necessary control requirements .

Require identification requirement for controlling logical sequence for the operation of various organs according to the cooling to be achieved . Even complete control over the process must be added the electrical components needed to ensure the functioning of organs and achieving the desired control requirement .

7.2 Types Of Electrical Circuit

There are tow types of electrical circuits in general . There are power circuit and control circuit . Capacity for small units are usually the control and power of one, either for units with high capacity controlling circuit be controlled separately from the power circuit .

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

7.2.2 **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 of three phases .

7.3 Components Of Electrical Circuits

7.3.1 Current Relay

This component can be describe as a magnetic switch . It comprises a small solenoid coil around a sleeve and an iron core . Inside the sleeve is a plunger to which the switch contact bridge is attached . The contact are normally open, When the coil is energized, a strong magnetic field of force is created because the current will be high during the starting phase. The magnetic force will move the plunger upward and bridge the switch contacts, completing the circuit to the start winding. The run winding is wired through the relay so that it is always in circuit . A high starting current is drawn when the compressor motor starts. The current reduces as the motor gathers speed, the magnetic field through the relay then becomes weaker so that it can no longer hold the contact bridge on to the switch. The plunger then drops down by gravity to open the circuit to the start winding .

It is not uncommon for a start capacitor to be fitted when a current ready is employed. This is wired in series with the start winding figure 7.1 shows current relay .

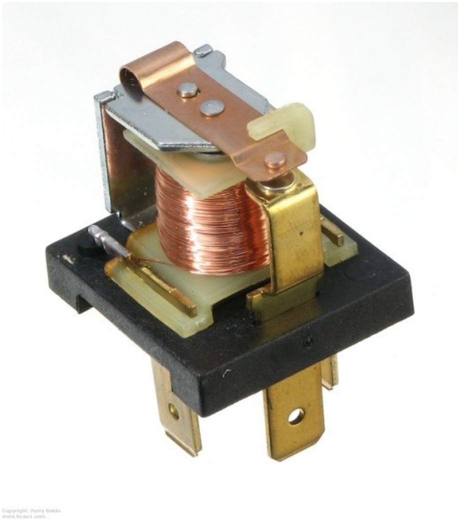

Figure 7.1 Current Relay

7.3.2 Potential (voltage) Relay

This type of relay is used with high starting torque motors . It operates in a similar manner to the current relay except that the switch contact are normally closed . The solenoid coil, once energized , maintains a magnetic force strong enough to open the switch contact and keep them open whilst the compressor is running .

The relay has mush higher design voltage rating than the supply voltage . As the motor approach its design speed, the voltage across the coil can sometimes be more than twice that of the supply voltage. When power is supplied to then circuit, the relay contact are closed. Both motor winding are energized and starting is achieved as the motor incases speed, the voltage in the start winding increases to caused increased in both voltage sand current basing through the coil. When the design voltage the coil is reached, the current creates a strong magnetic force to pull in the plunger and contact bridge to open the start circuit, but allows compressor in the run winding .

When the relay contact open, the voltage and current across the coil will decrease but will maintain a magnetic force strong enough to keep the contact open until power is disconnected. The contact will then return to the closed position ready for a restart. Figure 7.2 shows voltage relay .

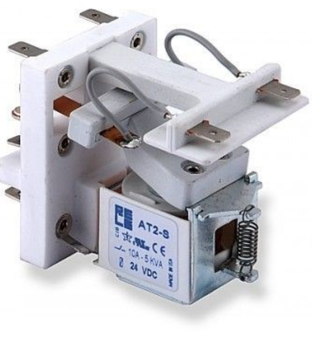

Figure 7.2 Voltage Relay

7.3.3 Capacitor Start And Rum 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 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 7.3 shows run and start capacitor .

Figure 7.3 Run And Start Capacitor

7.3.4 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 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 .figure7.4 shows overload .

Figure 7.4 Overload

7.3.5 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 7.5 shows contactor .

Figure 7.5 Contactor

7.4 Electrical Description For Cascade Cycle

Cascade refrigeration cycle is one of the cooling multistage, and each stage need a compressor. Controlling the operation of these stages needs an electrical circuit integrated with the mechanical cycle . It is known that each stage of this cycle produces to a certain temperature, to reach very low temperature, the first stage cool unit it reaches the -32 degrees Celsius, then second stage will assist the first to reach -65 degrees Celsius. If we want to design an electrical control circuit for this mechanical cycle we can use more than one way. In this chapter we discussed just one method for controlling this cycle, which is controlling with contactors .

Before explaining control process we shall clarify how the cascade cycle work electrically. After pressing start pushbutton to operate the cycle, the first compressor in the high cycle will run. Then second compressor in the low cycle will run. The cycle has a fan which operates when the first compressor start, after four hours from starting cycle the clock shut down the cycle and start the heater for ten minutes to defrost, then the hibernate clock starts the cycle again .

7.4.1 Control Of Cycle Using Contactor

1. Cycle starts after pressing pushbutton, current flows across through ON/OFF pushbutton .

2. Then electric current flows through the protection line (OL1 , OL2 , HP1 , HP2 , LP1 , LP2).

3. Then current flow to contactor (K) after pressing on pushbutton S , and contactor (K) make self holding to keep the cycle ON .

4. The cycle starts with R404a compressorbe ON , but the temperature inside the chamber must be more than -65°C, which know by thermostat (TH1) .

5. when the R404a compressor switch on , the condenser fan will be switch on , and Door heater (KH2) must switched ON too .

6. When the temperature at the high cycle reached to -32°C , Evaporator Fan must switched ON.

7. After the Evaporator Fan be ON, R23 compressor must be switched ON, and timer (C) will switched ON too .

8. The Timer (C) will operate the Defrost Heater (KH1) .

9. When the temperature reached -65°C , it will be operate the on delay Timer (T) .

10. The on delay timer make R23 compressor turned OFF first , after while R404a will be turned OFF .

The above procedure is explained in Figure 7.6 .

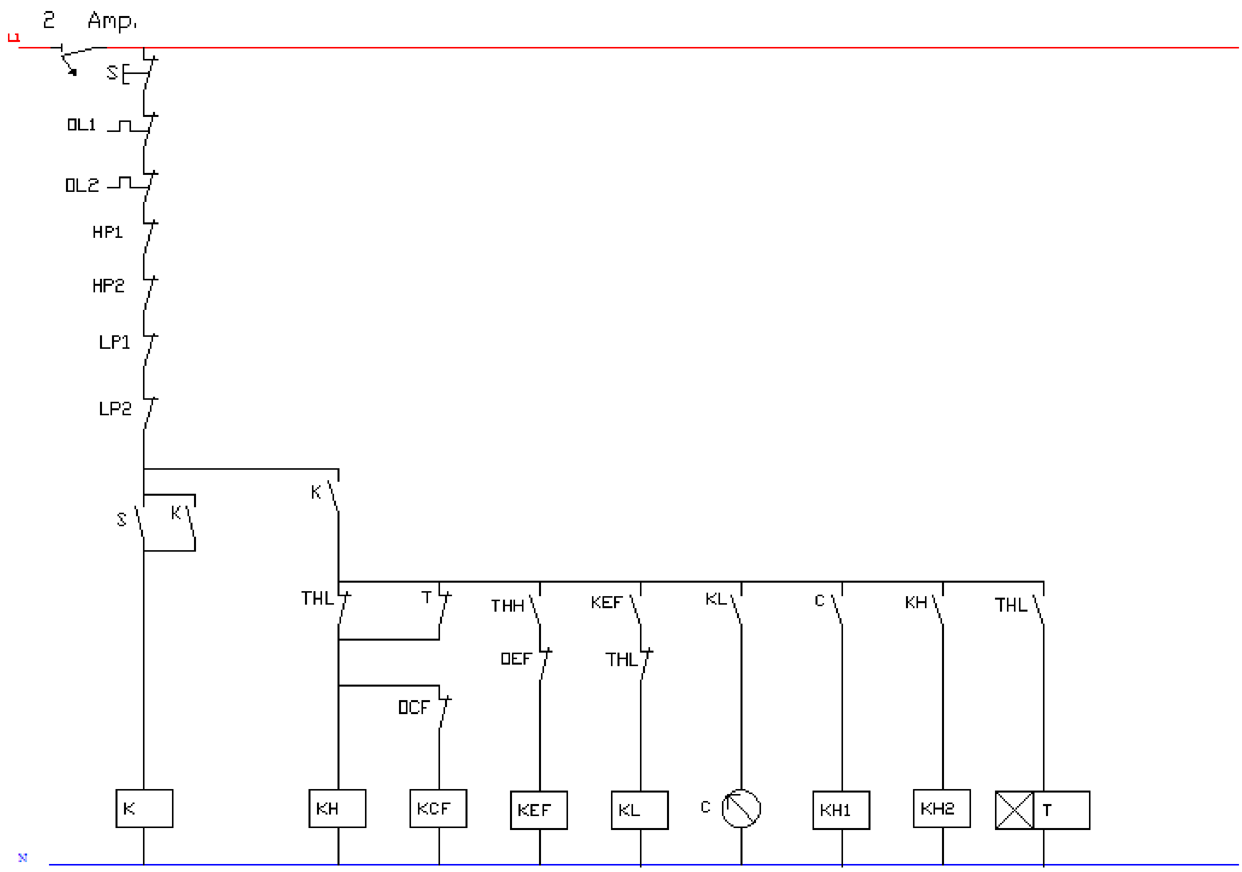

Figure 7.6 Control Circuit

7.5 Power Circuit

Power circuit consists of three motors and a defrost heater that work according to the instructions receiver from control circuit, Figure7.7 bellow depict power circuit blueprint for the cycle .

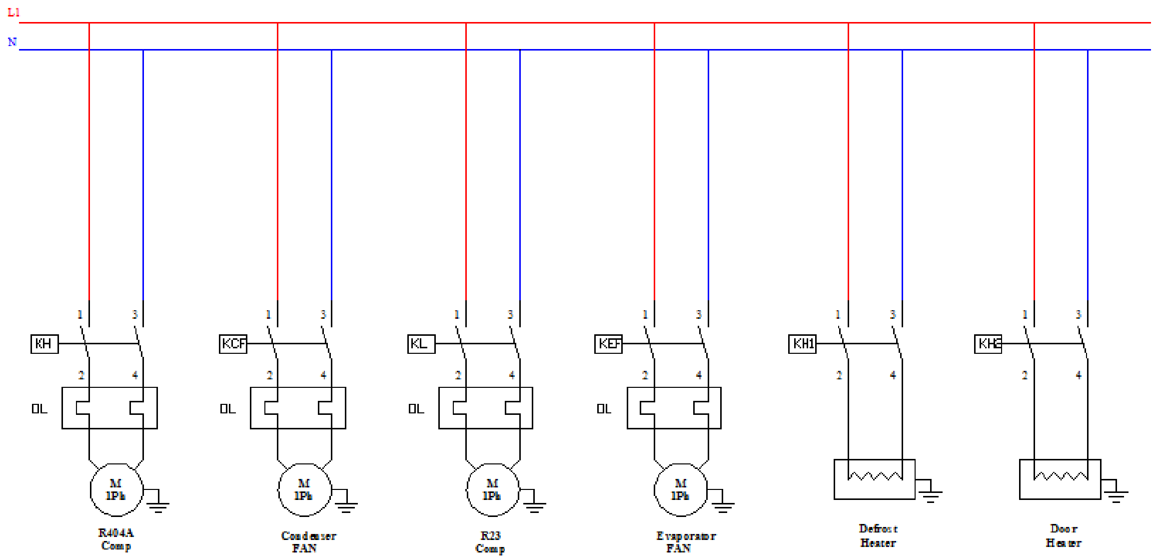

Figure 7.7 Power Circuit

7.6 Alarm And Monitoring System

Alarm and monitoring system are very important parts in electrical circuit, it allows to detect any fault which may occur in the tow stages, in order to handle it easily. In our project we used these tow system through using lamps working as follows :

1) Monitoring circuit :
   a. Lamp lights when the high stage runs and off when it shutdown (HS) .
   b. Lamp lights when the low stage runs and off when it shutdown (LS) .
   c. Lamp lights when the condenser fan runs and off when it shutdown .
   d. Lamp light when the defrost starts and off when it ends (KC) .

2) Alarming Circuit
   a. Lamp lights when pressure increases above required pressure (each stage needs one lamp) (HP1 and HP2) .
   b. Lamp lights when pressure decreases below the required pressure (each stage needs one lamp) (LP1 and LP2) .
   c. Lamp lights when compressor are overload (each stage needs a lamp) (OL1 And OL2).

This is explain in Figure 7.8below .

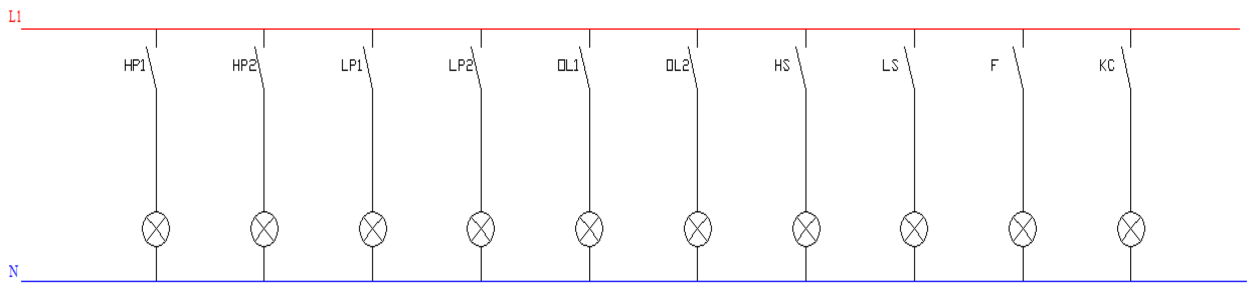

Figure 7.8 Lamp Circuit

Table 7.1 Circuit symbols

Symbol	Description
AMP	Tow Ampere Fuse
S0	Pushbutton
OL1	Over Load For High Compressor
OL2	Over Load For Low Compressor
HP1	High Pressure Cutout For High Compressor
HP2	High Pressure Cutout For Low Compressor
LP1	Low Pressure Cutout For High Compressor
LP2	Low Pressure Cutout For Low Compressor
S	N.O Pushbutton
K	Self Holding Contactor
THL	Low Cycle Thermostat
THH	High Cycle Thermostat
KH	High Compressor Contactor
T	On Delay Timer
OCF	Over Load Condenser Fan
OEF	Over Load Evaporator Fan
KCF	Condenser Fan Contactor

KEF	Evaporator Fan Contactor
KL	Low Compressor Contactor
C	Timer
KH1	Defrost Heater Contactor
KH2	Door Heater Contactor

CHAPTER 8

Testing

8.1 Introduction

After finish any practical project , it must have testing stage .This project pass many tests to compare it with the theoretical design .

8.2 Testing

A- The cascade system operate at the following conditions :

The higher cycle usesR404a .

The lower cycle uses R23.

Using tube in tube heat exchanger with inner diameter of (3/8") and outer diameter of (3/4") .

Using capillary tube with diameter of (0.052") and length of 3 meters .

The higher cycle work normally and the temperature reaches (-15 C^o) .

The suction pressure was (2.75 bar) and the discharge was (7.24bar) .

The Amendment on the higher cycle by the following methods :

1- Reducing the suction pressure of the higher cycle the temperature reaches (-18 C^o) .

2- Declining the capillary tube length from (3 meter to 2 meter) the temperature didn't reaches rather than (-10 C^o) .

3- Charging the cycle by R404a Gradually until the suction pressure reach to (4 bar) and the temperature reach to (-14 C^o) .

4- After isolated all the pipes completely and isolated the heat exchanger by two layers of vidoflex, and charging the cycle by (2 bar) of R404a , the temperature reaches (10 C^o) , then we charge the cycle until the pressure reaches (4bar) and the temperature reaches (-5 C^o) and the higher compressor was very hot that lead to shut down the compressor and the compressor didn't pressurized .

B- The cascade system operate at the following conditions :

The higher cycle used R22 .

The lower cycle used R23.

Using axial tube in tube heat exchanger with inner diameter of (1/4") and outer diameter of (1/2") , and it putted in wooden box and isolated using pressurized polytheren, as shown in the Figure (8.1) below :

Figure 8.1 Heat Exchanger

Using double capillary tube with diameter of (0.038") and length of 95 cm , as showing in the figure (8.2) below :

Figure 8.2 Double Capillary Tube

After charging the higher cycle by R22 with suction pressure (2 bar) the cycle work normally and the temperature reaches (-25 C^o)

The suction pressure of the higher cycle was (2 bar) and the discharge was (13.1 bar)

The lower cycle didn't operate automatically because the higher cycle temperature didn't reach (-20 C^o) as in control design , so it operate manually .

The Amendment on the cascade system by the following methods :

1- Charging the higher cycle by (2.75 bar) of R22 and run it for 1 hour , the temperature reached to (-25 C^o) .
2- Charging the lower cycle by (2.4 bar) of R23 and run it manually for 2 hours , the temperature reached to (-5 C^o) only .
3- Adding isolation material to the refrigeration chamber to decrease the volume which lead to decline the cooling load , but the temperature still (-5 C^o) .
4- Changing the higher cycle compressor from 2/3 hp to 1.5 hp and keep the low cycle compressor 1/3 hp , the ice start تراكم on the high compressor and the lower compressor shut off after 5 min. of running the system , figure (8.3) shown the ice on the higher cycle compressor :

Figure 8.3 High Cycle Compressor

5- After changing the low compressor from 1/3 hp to 1.25, as shown in figure (8.4) the reading was as the following :

Figure 8.4 The Mechanical Cycle

-For the low stage the temperature was ($-54.8C^o$) and the pressure (1.4 bar) .

-For the high stage the temperature was ($-28\ C^o$) and the pressure (2.4 bar) .

Figure 8.5 Reading Temperature

Conclusion and Recommendations :

Through our work on this project we noticed the following points , and found it useful to bring these notes to your attention :

- Cascade system is useful for many field especially for medical field for storing blood plasma, tissues, and bone marrow .
- Hardware lacks in our country can effects on our project duration so you should be ready to find other alternative hardware .
- Heat exchanger is very important part for cascade system so you must design the heat exchanger with high efficiency .
- You can use another alternative compressor for other refrigerant but it must be oil computable with these refrigerant – We use compressor R404a for refrigerant R23 with oil computability (POE) .
- A large attention should be attributed to the electricity side of the system .
- The isolation of the parts that have temperature lower than the surrounding temperature is very important .
- In this project may we have problems in heat exchanger and capillary design .
- The lower compressor must be larger than what we use because it can't bear the high pressure of R23 , and it very small comparing with the system chamber .

Appendix A

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^{o}	Min. Temp. C^{o}
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

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