بسم الله الرحمن الرحيم

Palestine Polytechnic University

College of Engineering & Technology

Mechanical Engineering Department

Graduate Project

Design And Implement Of Two Stage Vapor Compression Refrigeration System For Fresh Frozen Plasma Storage

Project Team

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

توقيع المشرف

....

توقيع اللجنة الممتحنة

....

توقيع رئيس الدائرة

....

Dedication

To my parents… who raised me…

To my friends… who helped me…

To my beloved… who supported me to the end…

To all the martyrs who sacrificed their lives for us

To our mothers and fathers

To all Palestinian mothers and fathers

To our great Palestine and Islamic nation

To our supervisor Dr. Ishaq Sider

To whom their guidance and support made this work possible

Khaled Ayoub Sider — Murad Mohammed Dweik

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Abstract

The purpose of the project is the design and implement of two stage vapor compression refrigeration cycle with intermediate vessel and double throttling , for storage of 10 kg of frozen plasma ; that used in medical applications , and save it at -35 ˚C refrigeration space.

The refrigerant R404A was selected as suitable working fluid . This system consists of two cycles connected in series , where the higher cycle operates between (-8 and 45 ˚C), and the low cycle operates between (-45 and -8 ˚C).

Full theoretical design and building of the system will be made , including the full automation for the cycle and controlling of it's componants.

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1.1 Project Outline

Chapter One:

This chapter is an introduction to the refrigeration systems, it includes the scope of the project, history of two stage cycle , what two stage cycle and its common configurations ,are it's advantages and disadvantages, comparisons between refrigeration cycles, methodology of the project and it contains budget and time tables for the project.

Chapter Two:

Describes the cooling loads and shows their sources and explains how to calculate them.

Chapter Three:

This chapter shows how to select a refrigerant and specifications must be available ,and includes cycle analysis and represented it on (cool back) program .

Chapter Four:

It presents the different types of components of the refrigeration system.

Chapter Five:

It presents the calculations and selection for different types of components of the refrigeration system including compressors ,evaporator ,condenser, vessel and connection pipes .

Chapter six:

This chapter presents the electrical and control cycle design for different types of components of the system.

1.2 Scope of Project:

Multi-stage refrigeration systems are an area of growing industrial importance in large plants. These systems are known to be large power users and represent significant capital investment. energy conservation becoming increasingly important mainly from an environmental perspective, thus it has become necessary to design these systems for optimal performance.[reference 11]

This project aims to design a two stage vapor compression refrigeration cycle with vessel to provide and keep a suitable temperature for storage of 10kg of fresh frozen plasma to keep it in a specific condition, which is below zero temperature by 35 °C , which is used in medical applications, and this large range of temperature can be used for many applications that need a low temperature; such as industrial processing for testing the mechanical parts of the aircraft before taking off to make sure that the execution of these parts at various temperature, biological activity and chemical industry to provide a low- temperature environment for chemical reactions that would race out of control at room temperatures.

1.3 History of two stage refrigeration system

The first extensive calculations, concerning the inter-stage level optimization, by Behringer's (as described by Gosney, (1966))for two- stage ammonia cycles with subcooling and de-superheating at saturation temperature.

Rasi (1955) was the next one who studied this phenomenon and its conditions; his theoretical studies were based on R-12, R-717 and methyl chloride as refrigerants, on saturated suction in both compression stages and total expansion from saturated liquid leaving the condenser, Czaplinski (1959) did some research on ideal cycles later, Baumann and Blass (1961), however, focused their efforts in more realistic cycles; Threlkeld (1966) proved the inadequacy of the geometric mean of pressures as the optimum inter-stage pressure in two-stage refrigerating cycles. De Lepeleire (1973) stated for R-22 and different system configurations, Domanski (1995) found an approximate optimum inter-stage temperature. Zubair et al. (1996) studied a two-stage refrigerating cycle, analysing it through the first and the second thermodynamics laws, and obtained the inter-stage pressure corresponding to a maximum in COP. Ratts and Brown (2000) used the entropy generation minimization method to determine the optimum inter-stage temperature considering only superheating and throttle losses.[reference 12]

1.4 Back ground about two stage refrigeration system

In many cool generation applications in which the temperature difference between evaporation and condensation is below 40 °C simple vapour compression system are appropriate enough, as temperature difference increases , the volumetric efficiency decreases (specially when dealing with reciprocating compressors), the discharge temperature rises and there is an increment in the vapour ratio at the inlet of the evaporator. Hence, for higher temperature difference, advantages of the multi-stage systems should be used.[reference 12]

There are two general types of such systems: cascade and multistage. The multistage system uses two or more compressors connected in series in the same refrigeration system. The refrigerant becomes more dense vapor while it passes through each compressor. Note that a two-stage system can reach a temperature of approximately -65°C and a three-stage about -100°C. cascade refrigeration systems are employed to obtain high temperature differentials between the heat source and heat sink and are applied for temperatures ranging from -70°C to -100°C .[reference 3]

Two stage vapor compression system can be described as two cycles which operate in series, there are a vessel between them , the first compressor pumps up to an intermediate pressure . The second compressor then compresses up to condensing pressures. Compression ratio for each cycle is only equal to the square root of the total compression ratio. And this cycle have greater flexibility to accommodate the variation of refrigeration loads at various evaporating temperatures during part-load operation. The two stage cycle has high initial cost than single stage but it saves on total compressor power consumption as shown in figure (1.1) that explains how making the saved work in a two stage system versus single stage system in T S diagram. The area between points 2 , 3, 4 , 4' represent the saved work in two stage system during running the cycle. This work at a long time economize the compressor power consumption and compensate the high initial cost for this system.

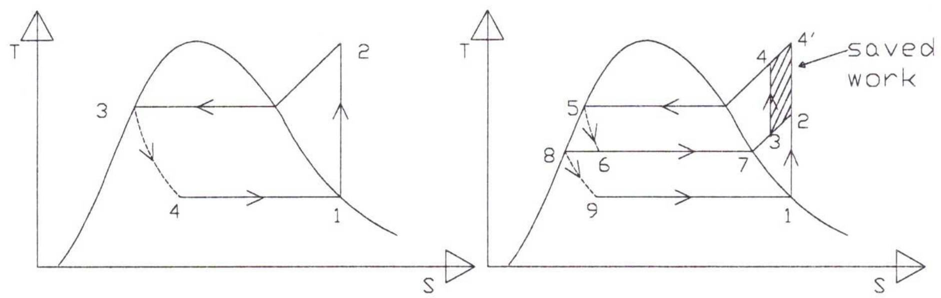


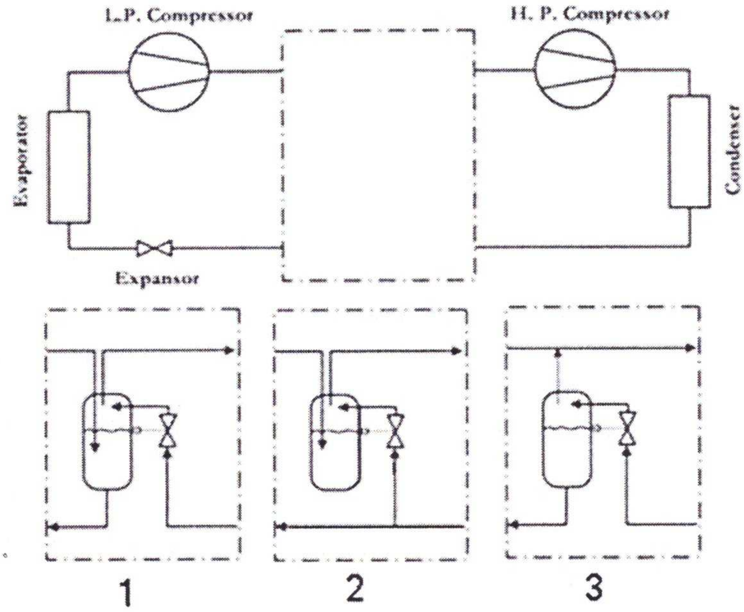
Figure 1.1. saved work in two stage system

Cascade system can be used to obtain evaporator temperature below -70°C but it has higher initial cost than two stage system, and uses two different refrigerants.

1.5 Configurations of two-stage systems

There are many configurations of two stage refrigeration cycles , depends on the mechanical connection of pipes between the main parts in the two stage cycle , and this variation of the connection affects the coefficient of performance (COP) of the cycle , and state of the refrigerant that entering the compressors .

The common configurations for two-stage systems are shown in Figure. 1.2



1- Two stage vapor compression refrigeration cycle(V.C.R.C)with intermediate vessel& double throttling with totally de-superheating & partially injection in vessel
2- Two stage (V.C.R.C) with intermediate vessel& double throttling with totally de-superheating & totally injection in vessel
3- Two stage (V.C.R.C) with intermediate vessel& double throttling with partially de-superheating & totally injection in vessel

Figure 1.2: common configurations for two-stage systems

1.5.1 Comparison Between different configurations

To make comparison between cycles a sample is taken with assumptions to calculate the coefficient of performance of each cycle and determine the better chosen . The assumptions are :

Refrigeration capacity = 5 kW

Evaporator temperature = -40°C

Condenser temperature = 45°C

Refrigerant used is R 404A

1- Two stage vapor compression refrigeration cycle (V.C.R.C) with intermediate vessel& double throttling with totally de- superheating & partially injection in vessel ,figure (1. 3)

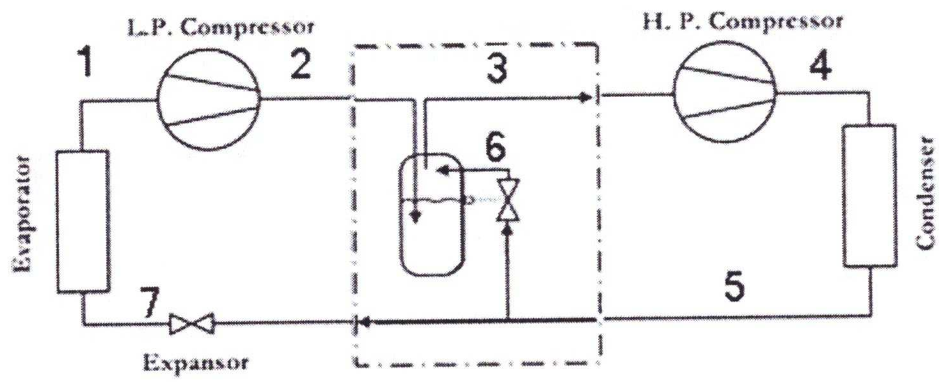


Figure 1-3: Two stage V.C.R.C with intermediate vessel& double throttling with totally de- superheating & partially injection in vessel

(a) schematic diagram.

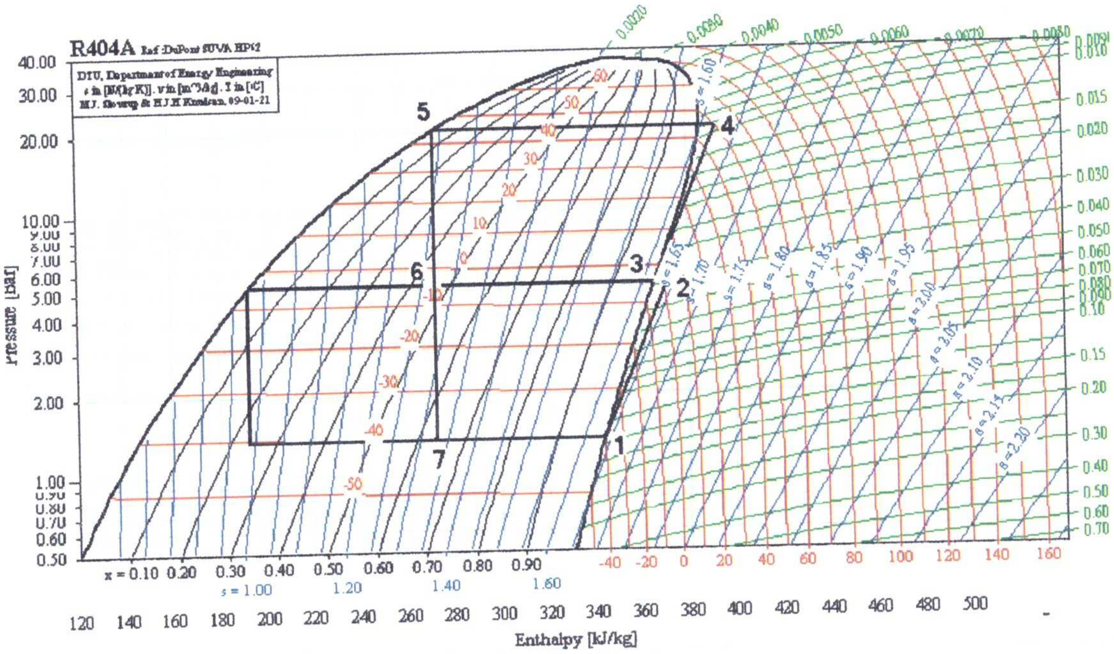


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Figure 1.3:(b) refrigeration cycle.

Table 1.1 Properties of each point on the figure 1.3.b

	T(°C)	h(kJ/kg)	P(bar)
1	-40	343.8	1.33
2	1.7	371	5.2
3	-4.4	365.8	5.2
4	50.5	392.8	20.4
5	45	272	20.4
6	-4.7	272	5.2
7	-40	272	1.33

Where: T:temperature ,h: enthalpy ,P: pressure .

Calculations

$pi = \sqrt{pc * pe}$ ....(1.1)

Where

P_i : intermediate pressure[bar]

p_c : condensing pressure, [bar]

p_e : evaporating pressure, [bar]

p_c= 20.4[bar]

p_e= 1.33[bar]

$pi = \sqrt{20.4 * 1.33}$

P_i= 5.2 [bar]

$\dot{m}_1 = \frac{Qe}{h1-h7}$ ....(1.2)

Where :

$\dot{m}_1$: mass flow rate in the evaporator, [kg/s].

Q_e : refrigeration capacity, [kW]

$\dot{m}_1 = \frac{5}{343.8-272}$

$\dot{m}_1$ =0.069 kg/s

By mass balance of the refrigerant entering and leaving the vessel can obtain this equation

$\dot{m}_2=\dot{m}_1+\dot{m}$ ....(1.3)

where

$\dot{m}_1$: mass flow rate in the evaporator, [kg/s].

$\dot{m}_2$: mass flow rate in the condenser, [kg/s].

$\dot{m}$: mass flow rate in the intermediate line, [kg/s].

Also by heat balance of the refrigerants entering and leaving the vessel can obtain this equation

$\dot{m}_2\, h3 = \dot{m}\, h6 + \dot{m}_1\, h2$ ....(1.4)

When $\dot{m}_1 = 0.069$ [kg/s], and substitute equation (1.3) in equation (1.4) ,resulted

$\dot{m} = 0.0038$ [kg/s]

$\dot{m}_2 = 0.0728$ [kg/s]

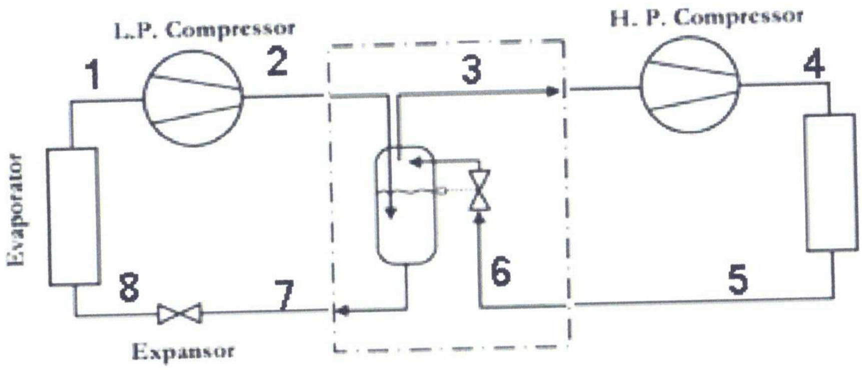
$$COP = \frac{Qe}{m.1(h2-h1)+m.2(h4-h3)} \quad \text{...(1.5)}$$

Where

COP : Coefficient of performance of the cycle

$$COP = \frac{5}{0.069(371-343.8)+0.0728(392.8-365.8)} = 1.3$$

2- Two stage V.C.R.C with intermediate vessel& double throttling with totally de-superheating & totally injection in vessel figure (1. 4)



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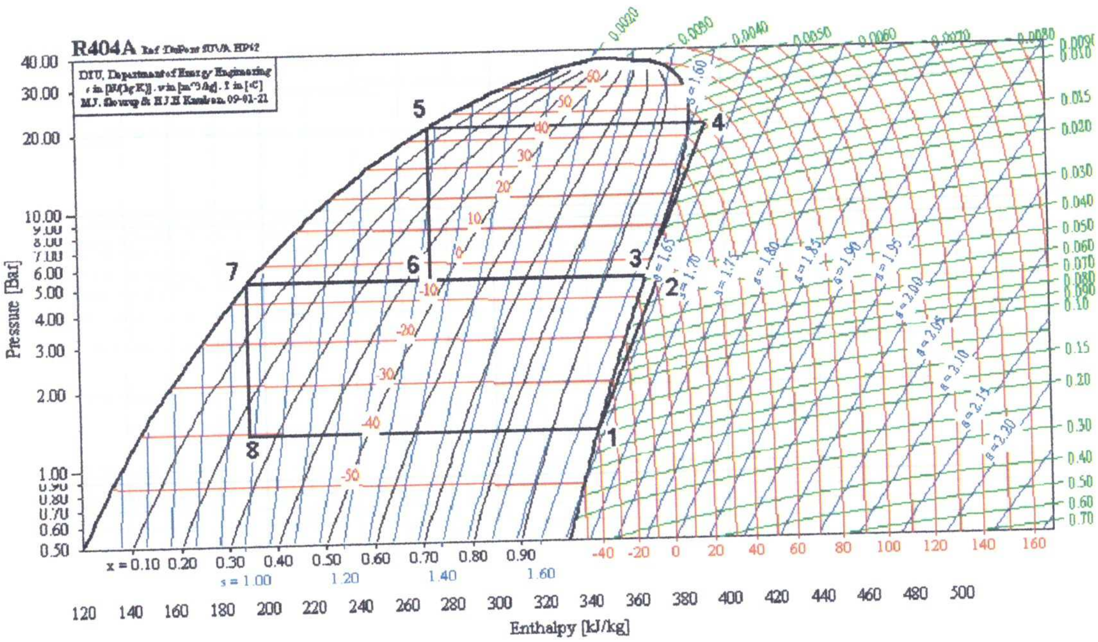

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Figure 1.4 Two stage V.C.R.C with intermediate vessel& double throttling with totally de- superheating & totally injection in vessel: (a) schematic diagram; (b) refrigeration cycle.

Table 1.2 Properties of each point on the figure 1.4.b

	T(°C)	h(kJ/kg)	P(bar)
1	-40	343.8	1.33
2	1.7	371	5.2
3	-4.4	365.8	5.2
4	50.5	392.8	20.4
5	45	272	20.4
6	-4.7	272	5.2
7	-4.7	193	5.2
8	-40	193	1.33

Where: T:temperature ,h: enthalpy ,P: pressure .

Calculations

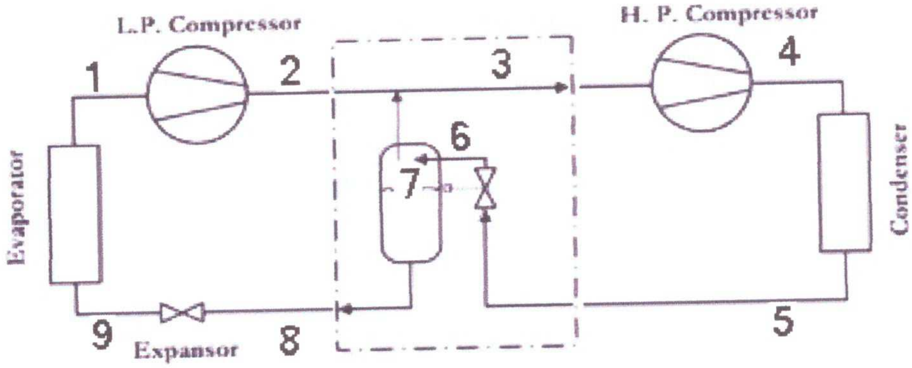
$$m.1 = \frac{5}{343.8-193} = 0.033 \text{ kg/s}$$

By heat balance of the refrigerants entering and leaving the vessel can be calculate the mass flow rate in the high cycle

$\dot{m}_2$=0.0626 [kg/s]

$$COP = \frac{5}{0.033(371-343.8)+0.0626(392.8-365.8)} = 1.9$$

3- Two stage V.C.R.C with intermediate vessel& double throttling with partially de - superheating & totally injection in vessel figure (1. 5)



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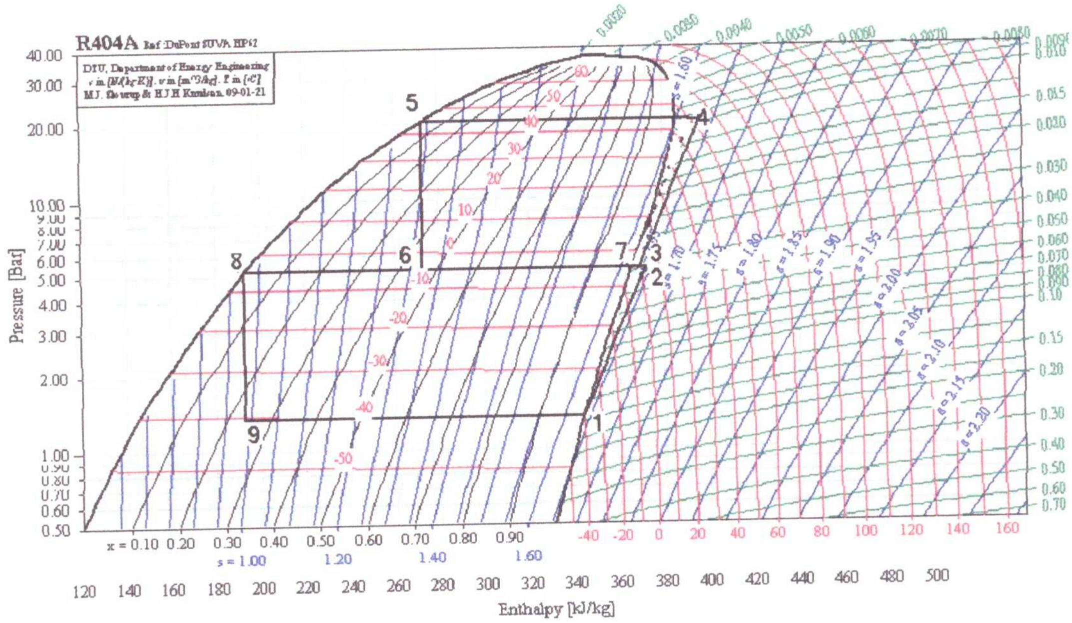

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Figure 1.5 Two stage V.C.R.C with intermediate vessel& double throttling with partially de -superheating & totally injection in vessel: (a) schematic diagram; (b) refrigeration cycle.

Table 1.3 Properties of each point on the figure 1.5.b

	T(°C)	h(kJ/kg)	P(bar)
1	-40	343.8	1.33
2	1.7	371	5.2
3	-1	368.6	5.2
4	53	396.3	20.4
5	45	272	20.4
6	-4.7	272	5.2
7	-4.7	365.8	5.2
8	-4.7	193	5.2
9	-40	193	1.33

Where: T:temperature ,h: enthalpy,P: pressure .

Calculations

$$m.1 = \frac{5}{343.8-193} = 0.033 \text{ kg/s}$$

By heat and mass balance of the refrigerants entering and leaving the vessel can be calculate the mass flow rate in the high cycle and in the intermediate line

$\dot{m}$= 0.027[kg/s]

$\dot{m}_2$=0.06 [kg/s]

$$COP = \frac{5}{0.033(371-343.8)+0.06(396.3-368.6)} = 1.95$$

1.5.2. The project cycle selection

According to the obvious calculations, and the comparison between the two stage cycles for their COP. the chosen of the project cycle was made which is called (two stage vapor compression refrigeration cycle with intermediate vessel and double throttling with partially desuperheating and totally injection in vessel).that shown in figure (1.5) ,figure 1.6 show the cycle with all components and accessories.

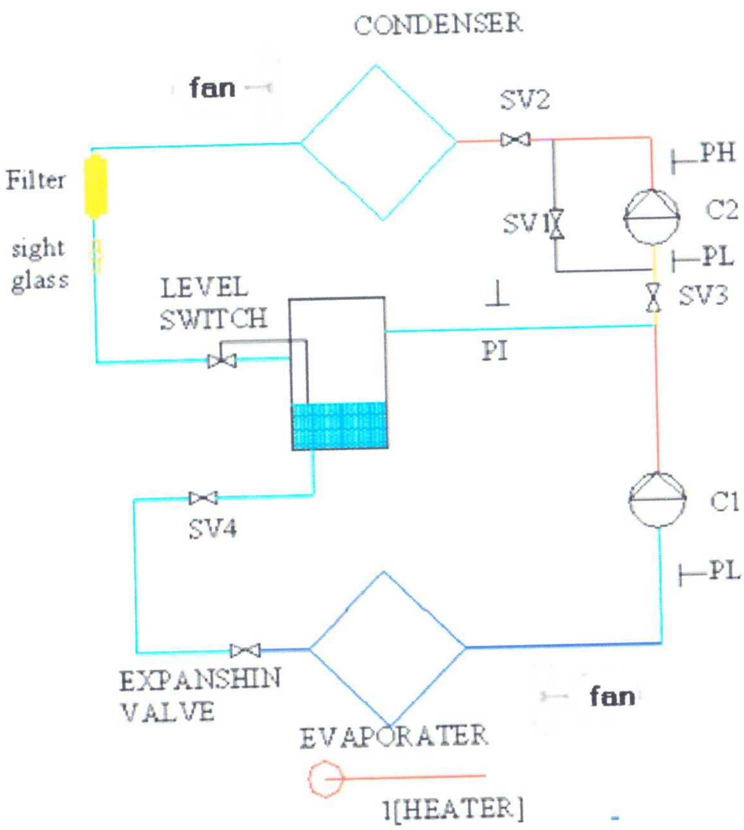

FIGURE 1.6 Tow stage cycle with all components and accessories

1.6 The budget for the project

Table 1.4 Actual budget table for the project

Task	COST (NIS)
Researches	100
Transportations	100
To copy from library	50
Printing papers	150
Reprinting papers	300
components of the project	
Two compressors	1540
Condenser	200
Evaporator	150
Connecting pipes	200
Vessel	200
Expansion valve	150
Refrigerator frame	500
Refrigerant	300
Defrost heaters	200
Control equipments	
pressure switches	2000
temperature switch	
wires	
switches	
Solenoid valves	
filter	
Sight glass	
float valve	
contactors	
Total	**6040**

1.7 The time planning for the project

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

The following time plan is for the first semester

Table 1.5 The first semester time plan

Task/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Collecting Information about the project																
Reading																
Introduction																
Load calculations																
Cycle analysis																
Cycle components																
Project Documentation																

The following time plan is for the second semester

Table 1.6 The second semester time plan

Task/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Compressors selection																
Pipe Design																
Heat Exchangers Design																
Electrical & Control Design																
Accessory Selection																
Cycle building																
Cycle test																
Recommendations																
Conclusions																
Project Documentation																

CHAPTER TWO

COOLING LOAD

CHAPTER TWO

COOLING LOAD

2.1 Introduction

The total heat required to be removed from refrigerated space in order to bring it at the desired temperature and maintain it by the refrigeration equipment is known as cooling load. The purpose of 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 it's refrigeration equipment capacity to produce and maintain satisfactory inside conditions. [reference 13]

2.2 Load Sources

The cooling load on refrigerating equipment 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: [reference 5]

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

- Defrosts' heater heat gain
- Fan motor heat gain

General overview and information

Storage temperature is -35 °C

Maximum surrounding temperature is 38 °C ,table A5

Mass of product is 10[kg]

Desired cooling time is 10[hr]

Inner refrigerator size is (0.55*0.4*0.3) [m^3] . As shown in figure 2.1

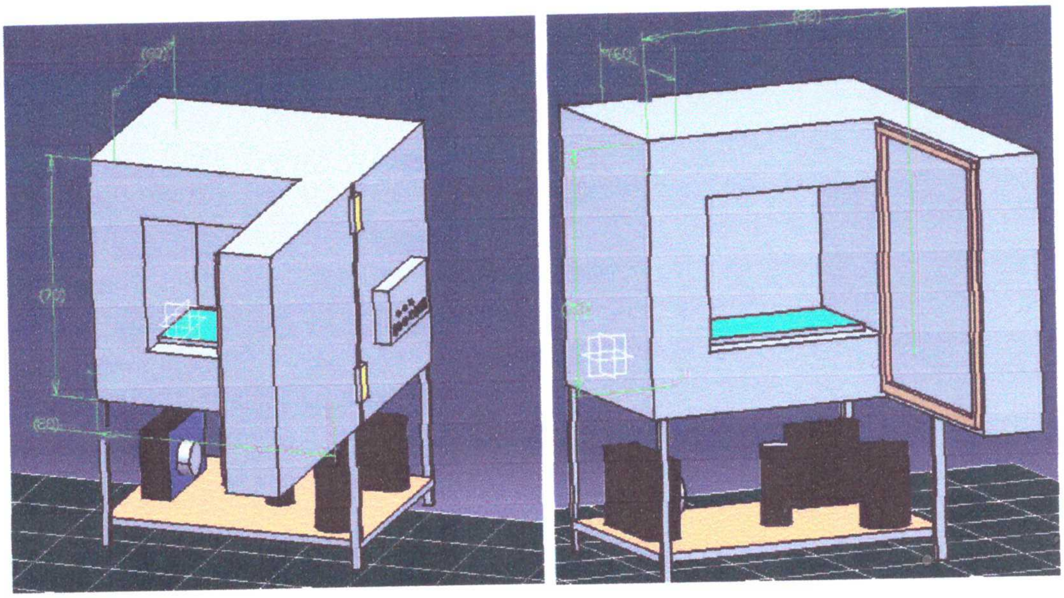

Figure 2.1 refrigerator frame

2.2.1 The wall heat gain

The wall heat gain load, sometimes called the wall leakage load, is a measure of the heat flow rate by conduction through the walls of 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 that of the outside. The wall gain load is common to all refrigeration application and is ordinarily a considerable part of the total cooling load, commercial storage coolers and residential air conditioning applications are both examples of applications wherein the wall gain load often accounts for the greater portion of the total load. [reference 5]

$$Q_{wall}=U * A * \Delta T \qquad (2.1)$$

Where:-

A: out side surface area of the wall [m^2].

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

ΔT: the temperature difference across the walls[$^\circ C$].

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

Where:

T_{in}: the refrigeration space temperature.

T_{out}: the outside temperature.

Overall heat transfer coefficient is computed by the following :

$$U = \frac{1}{\frac{1}{hi}+\frac{\Delta x1}{k1}+\frac{\Delta x2}{k2}+\frac{\Delta x3}{k3}+\cdots+\frac{1}{ho}} \qquad (2.2)$$

Where:

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

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/m^2. $^{\circ}$C].

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

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

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

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

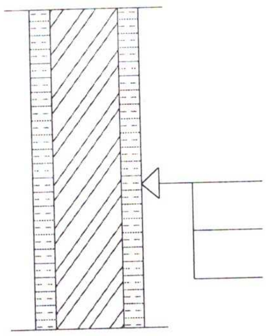

Figure 2.2 walls construction

1-Stainless steel 0.1 [cm]; K=15.6[W/m. $^{\circ}$C]. From Table A1

2-Fiber glass 10 [cm]; K=0.036[W/m. $^{\circ}$C]. From Table A1

3-Stainless steel 0.1 [cm]; K=15.6[W/m. $^{\circ}$C]. From Table A1

$$U = \frac{1}{\frac{1}{50} + \frac{0.001}{15.6} + \frac{0.10}{0.036} + \frac{0.001}{15.6} + \frac{1}{10}}$$

$U = 0.345[W/m^2\ ^{\circ}C]$

$Q_{\text{floor, roof}}$=0.345*(0.754*0.604)*(38- -35)=11.4 [W]

Q_{sides}=0.345*(0.604*0.504)*(38- -35)=7.7 [W]

$Q_{front, behind}$=0.345*(0.754*0.504)*(38- -35)=9.6 [W]

For all walls

$Q_{all\ wall}$ = 2*(11.4+7.7+9.6)

$Q_{all\ wall}$ = 57.4 [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 chilling load above freezing depends upon the mass product, mean specific heat of the products above freezing, entering product temperature , final product temperature desired, and the chilling time. This heat gain can be calculated by the following equation: [reference 13]

Q_{ch}= m.C_p . ΔT……………………………………………………………..(2.3)

Where :

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

m :mass of the product in [kg]

C_p :the specific heat above freezing in[kJ/kg.˚C]

$\Delta T = T_o - T_{ch}$

Where:

T_o : entering product temperature [˚C]

T_{ch} : chilling product temperature [˚C]

C_P = 3.92 [kJ/kg.°C] (table A2)

T_o = 38[°C]

T_{ch} = -0.9 [°C] (table A2)

Q_{ch} = 10*3.92*(38- -0.9)

Q_{ch} = 1525[kJ]

2. Cooling load below freezing: the cooling load below freezing depends upon the mass of product, mean specific heat of the products below freezing, actual storage temperature of the product, desired freezing temperature of the product (refrigerated space temperature), and the cooling time. [reference 13]

$Q_c = m.C_{P'}.\Delta T$ ....(2.4)

Where :

Q_c : cooling product load in [kJ]

m : mass of the product load in [kg]

$C_{P'}$: the specific heat below freezing in[kJ/kg.°C]

$\Delta T = (T_{ch} - T_{Rs})$

Where :

T_{ch}: freezing product temperature [°C]

T_{Rs} : refrigerated temperature [°C]

$C_{p'}$ =2.00 [kJ/kg.°C] table A2

T_{ch} = -0.9 [°C] table A2

T_{RS} = -35 [°C]

$Q_c = m.C_P'.\Delta T$

Q_c = 10*2*(-0.9- -35)

Q_c = 682[kJ]

3. Freezing load: the freezing load depends upon the mass of the product, its latent heat of freezing, and the freezing time. [reference 13]

$Q_f = m.\ H_L$ ....(2.5)

Where

Q_f : freezing load[kJ]

H_L :latent heat for the product [kJ/kg]

H_L=307[kJ/kg](table A2)

$Q_f = 10*307$

$Q_f = 3070$[kJ]

Total product load

$$Q_p = \frac{Qch+Qc+Qf}{\tau} \quad \text{.........(2.6)}$$

Where

τ : desired cooling time in [seconds]

$$Q_p = \frac{1525+682+3070}{10*3600} * 1000$$

Q_p =146.6 [W]

2.2.3 Infiltration heat gain

In the practical operation of a refrigerated facility, doors must be opened at times in order to move the product in and out. The infiltration load is one of the major loads in the refrigerator. The infiltration air is the air that enters a refrigerated space through cracks and opening of doors. This is caused by pressure difference between the two sides of the doors and it depends upon the temperature difference between the inside and outside air, and cooler sizes . [reference 2]

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

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

Where

ρ : air density [1.25 kg/m^3]

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

$V^{\cdot}_f$: the volumetric flow rate of infiltrated air[m^3/s]

T_o : the out side temperature [°C]

T_i :the inside temperature [°C]

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

number of air change = 0.5 [times /h] table A3

volume of room = 0.75*0.6*0.5 =0.225 m^3

$V^{\cdot}_f$= 0.5*0.225 =0.1125 m^3/hr

Q_{inf} = 1.25* (0.1125/3600)*1000*(38- -35)

Q_{inf} = 3[W]

2.2.4 Packaging Heat gain

Many products refrigerated in packages, it could be more than 10% of product's weight. Packages could be plastic, steel , wood , glass or any material that have low specific heat . [reference 13]

Plastic bags used to keep 250 g of liquid plasma ,and packaging it in stainless steel crate. Bags of plasma are arranged above the stainless steel crate. As shown in figure 2.3

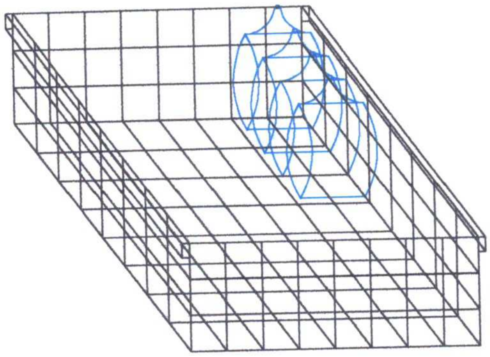

Figure 2.3 stainless steel crate and plastic Bags

$$Q_{pk} = \frac{mpk*Cpk*(To-Ti)}{\tau} * 10^3 \quad \text{.....(2.8)}$$

Where

Q_{pk} : packaging heat load[W]

m_{pk} : mass of product [kg]

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

T_o :out side temperature [°C]

T_i : temperature of the refrigerant space [°C]

τ: desired cooling time in [seconds]

For stainless steel crate

$C_{pk\ steel}$=0.5[kJ/kg°C] (table A4)

$$Q_{pk} = \frac{5*0.5(38--35)}{10*3600} * 10^3$$

$Q_{pk\ steel}$= 5[W]

For plastic bags

$C_{pk\ plastic}$=1.6[kJ/kg°C] (table A4)

$$Q_{pk} = \frac{40*0.2*1.6*(38--35)}{10*3600} * 10^3$$

$Q_{pk\ plastic} = 26[W]$

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

$Q_{pk\ total} = 31[W]$

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 evaporator coil is (0°C) 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]

$Q_{h1} = \eta * P$..(2.9)

Where

P: power of heater , taken 480[W]

η: heater usage factor (0.1 - 0.5) , taken 0.2

$Q_{h1} = 0.2*500 = 100$ [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 around 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.

$Q_{h2}=\eta*P$

Where

P: power of heater , taken 80[W]

η: heater usage factor (0.1 - 0.5) , taken 0.2

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

Total defrosts' heater load

$Q_h=96+16=112[W]$

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

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

$Q_T= 57.4+146.6+3+31+120+25$

$Q_T= 385[W]$

Add 20% upon Q_T as a safety factor

So

Total cooling load = Q_T*1.2

TCL=385*1.2

TCL=462[W]

2.4 Required equipment capacity

After the safety factor has been added, the cooling load is multiplied by 24 hours and divided by the desired operating time in hour for the equipment to determine the Required equipment capacity [reference 5] .

$$Q_e = \frac{TCL\times 24}{operating\ time} \quad \text{.....(2.10)}$$

$$Q_e = \frac{462\times 24}{16}$$

$Q_e = 693[W]$

CHAPTER THREE

REFRIGERANT SELECTION AND CYCLE ANALYSIS

3.1 Refrigerant Selection

A refrigerant is the primary working fluid used for absorbing and transmitting heat in a refrigeration system or heat pump. Refrigerants absorb heat at a low temperature and low pressure and release heat at a higher temperature and pressure.

The natural ice and a mixture of ice and salt were the first refrigerants. In 1834 ammonia, sulpher, methyl chloride and carbon dioxide came into use as refrigerants in compression cycle refrigeration machines. Most of the early refrigerant materials have been discarded for safety reasons or for lack of chemical or thermal stability. In the present days many new refrigerants are used and can be classified into the following five main groups:[reference 3]

- Halocarbons(CFCs): such as R11,R12 .
- Hydrocarbons(HCs): such as R50,R290,R134A.
- Inorganic compounds: such as R718,R744.
- Azeotropic mixtures: such as R502 .
- Nonazeotropic mixtures(zeotropic): such as R404A,R410A

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 refrigerants are expected to meet the following conditions:

Thermodynamic properties

- low boiling point,
- high critical point temperature,
- high latent heat of vaporization,
- low specific heat of liquid,
- positive pressure at evaporator temperature,
- low specific volume at suction pressure and temperature, and
- mixes well with oil.

Thermophysical properties

- high thermal conductivity,
- high convection of heat transfer coefficient, and
- low dynamic viscosity.

Environment friendly

- having zero ozon depletion potential(ODP),
- having low global warming potential(GWP),
- non contribution of the atmosphere's heat retention,
- non corrosive to metal,
- nonacidic in case of a mixture with water or air,
- chemically stable,
- easily detectable in case of leakage,
- nonreactive with the lubricating oils of the compressor,
- non flammable and non explosive, and
- non toxic .

Economics

- high operating efficiency,

- low cost, and
- easy availability.

Table 3.1 Common refrigerants and some of its properties.

Refrigerant R	Sat.temp.@p_{atm} C°	Critical point temp. C°	Latent heat @p_{sat} kJ/kg	ODP (R11)
R717	-33	132.3	1367	0.0
R22	-41	96	234	0.05
R12	-30	112	215.3	1
R134a	-26	101.1	214.8	0.0
R123	27	183.6	170.6	0.02
R11	24	198	180.2	1
R718	100	374	2256	0.0
R404A	-46	72	202.5	0.0
R410A	-52	74.6	269.5	0.0
R407	-38	86.7	235.4	0.0
R507	-40	70.7	196.5	0.0
R744	-48	31	347	0.0

To select refrigerant successfully must be considered the above properties. A comparison between various refrigerants are made and found that, the best refrigerant could be used is R410A, but the small compressor that used this refrigerant are scarce, so will be avoid using this refrigerant. The second best refrigerant is R-404A from zeotropic mixtures family that consist from precise mixtures of substances that have different properties. This substances are (HFC- 143a / HFC- 125 / HFC- 134a) by a concentrations (52 / 44 / 4). If used R407 vacuum happen in the evaporator so may be leakage wet air to the cycle and causing problem. If used R507 the COP of the cycle is less than the COP of R404A because the latent heat is less than it. If used R22 it is possible ,but the environment harmed . so the best and suitable refrigerant could be used are R404A .

3.2 Cycle Analysis

3.2.1 Flow Processes

Figure 3.1a is a schematic diagram of a two-stage compound system with a vessel, and Figure. 3.1b shows the refrigeration cycle of this system.

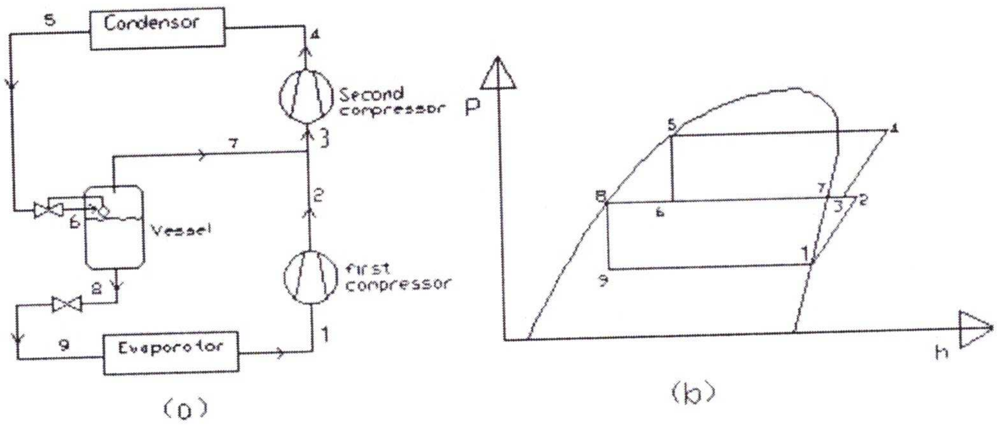

Figure 3.1. Two-stage compound system with a vessel : (a) schematic diagram; (b) refrigeration cycle.

Vapor refrigerant at point 1 enters the first-stage compressed of the first compressor at the saturated state. It is compressed to the intermediate pressure at point 2 and mixes with evaporated vapor refrigerant from the vessel, The mixture then enters the second-stage compressed at point 3. Hot gas, compressed to condensing pressure, leaves the second compressor at point 4. It is then discharged to the condenser, in which the hot gas is desuperheated, and condensed at point 5, After the condensing process, the liquid refrigerant flows through a throttling device, (float valve), at the high-pressure side. a small portion of the liquid refrigerant flashes into vapor in the vessel at point 7, and this latent heat of vaporization cools the remaining liquid refrigerant to the saturation temperature corresponding to the intermediate pressure at point 8. Inside the vessel, the mixture of vapor and liquid refrigerant is at point 6. Liquid refrigerant then flows through another throttling device (thermostatic expansion valve), a small portion is flashed at point 9, and the liquid-vapor mixture enters the evaporator. The remaining liquid refrigerant is vaporized at point 1 in the evaporator. The vapor then flows to the inlet of the first compressor and completes the cycle.

3.2.2 Cycle calculations

3.2.2.1 Intermediate Pressure

A two stage system consists of two compression stages connected in series as shown in figure 3.2,that shown the cycle diagram using R404A .There are three pressures, evaporator pressure, condenser pressure, and intermediate pressure, in the two-stage compression of vapor with ideal compression in both compressors, the optimum intermediate pressure is the geometric mean of the suction and discharge pressure, can be calculated as:

$$pi = \sqrt{pc * pe} \quad \text{.......(3.1)}$$

Where

P_i : intermediate pressure[bar]

p_c : condensing pressure, [bar]

p_e : evaporating pressure, [bar]

p_c= 20.4 [bar]

p_e= 1.055[bar]

$$pi = \sqrt{20.4 * 1.055}$$

P_i= 4.64 [bar]

The compression ratio in each stage can be calculated as,

$$R = \frac{pc}{pi} = \frac{pi}{pe} \quad \text{.......(3.2)}$$

Where

R :compression ratio

$$R = \frac{20.4}{4.64} = \frac{4.64}{1.055} = 4.4$$

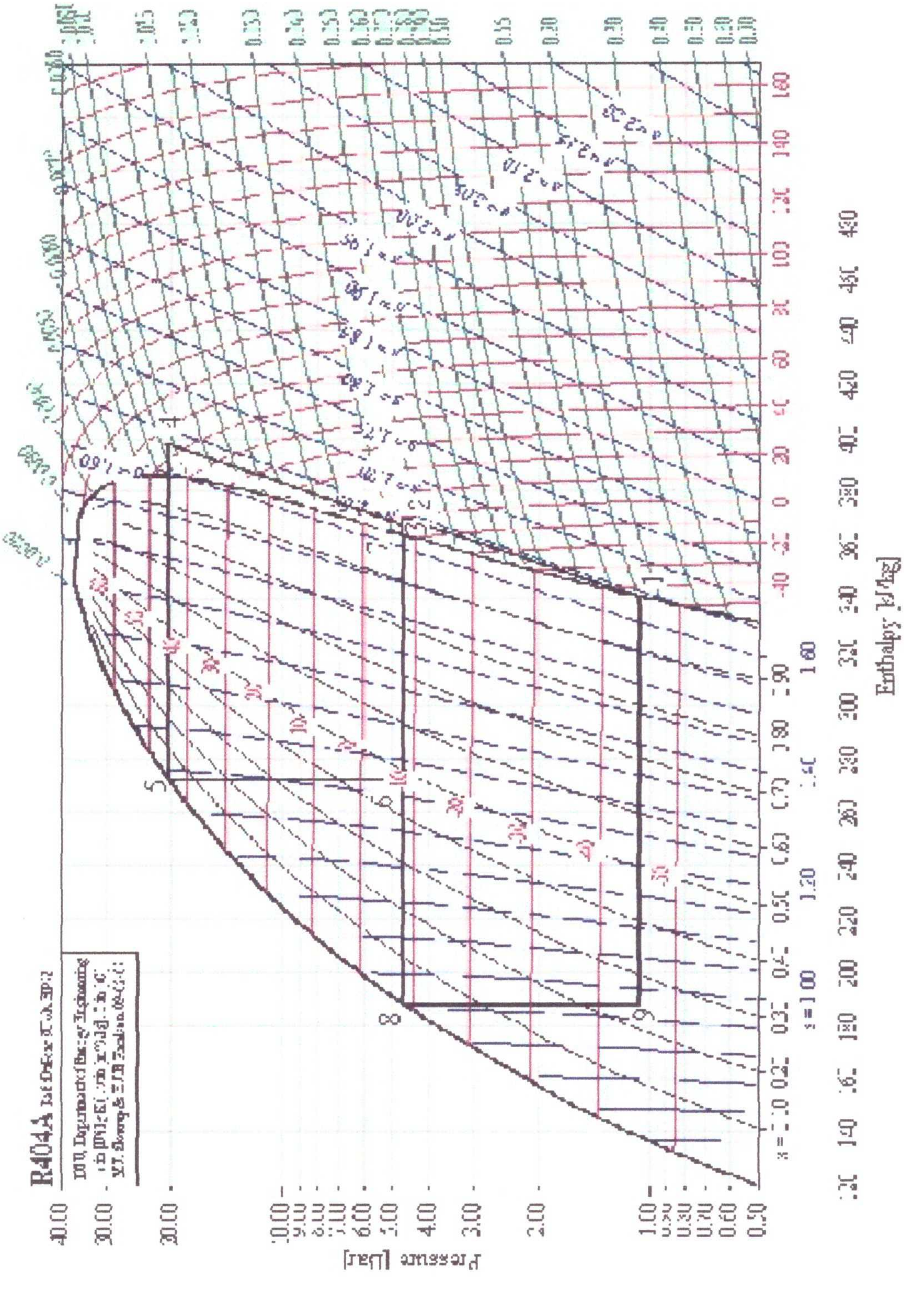

Figure 3.2. P-h diagram for R-404A

Table 3.2 Properties of each point on the cycle

	T(°C)	h(kJ/kg)	P(bar)	υ(m³/kg)
1	-45	340.6	1.055	0.17758
2	-0.6	370	4.64	0.0448
3	-4	367	4.64	0.0435
4	55	397	20.4	0.01
5	45	272.6	20.4	0.001
6	-8	272.6	4.64	0.022
7	-8	363.8	4.64	0.0431
8	-8	188	4.64	0.001
9	-45	188	1.055	0.04

3.2.2.2 Mass flow rate

In the vessel, out of $\dot{m}_2$ of refrigerant flowing through the condenser, $\dot{m}$ of it enter the intermediate line and cools down the remaining portion of liquid refrigerant $\dot{m}_1$ to saturated temperature T8 at intermediate pressure that enter to the evaporator. The mass flow rate entering the evaporator can be calculated as, [reference 1]

$$\dot{m}_1 = \frac{Qe}{qe} \quad \text{.........(3.3)}$$

Where :

$\dot{m}_1$: mass flow rate in the evaporator, [kg/s].

Q_e : refrigeration capacity, [kW]

$q_e = (h1\text{-}h9)$.........(3.4)

Where:

q_e : refrigeration effect, [k J/kg]

h1:enthalpy of saturated vapor leaving evaporator, [kJ /kg]

h9 :enthalpy of refrigerant entering evaporator, [kJ /kg]

$q_e = 340.6\text{-}188 \quad = 152.6$[kJ /kg]

$$\dot{m}_1 = \frac{0.693}{152.6}$$

$\dot{m}_1 = 0.00454$ [kg/s]

The mass flow rate of the refrigerant in the condenser $m^{\cdot}_2$ equal sum of mass flow rate in the evaporator $m^{\cdot}_1$ and the intermediate line flow rate $m^{\cdot}$

$m^{\cdot}_2=m^{\cdot}_1+m^{\cdot}$ ....(3.5)

If the heat loss from the insulated vessel to the ambient air is small, it can be ignored. Heat balance of the refrigerants entering and leaving the vessel, as shown in Fig. 3.3. a, gives [reference 1]

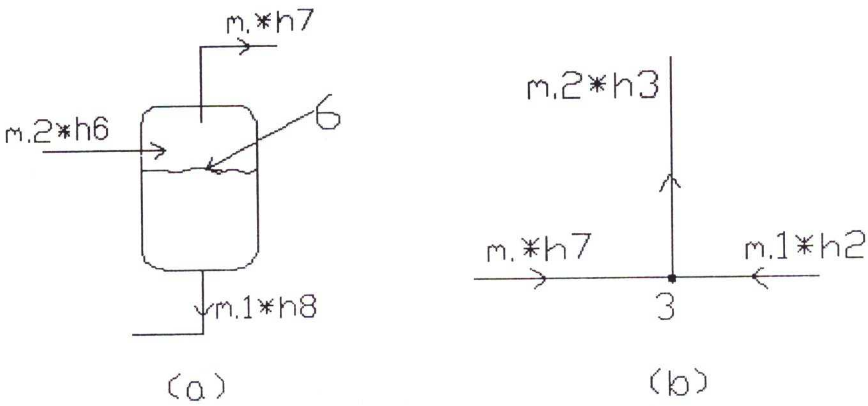

Figure 3.3. Heat balance of entering and leaving refrigerants: (a) in the vessel ; (b) at the mixing point 3 before entering the second compressor.

Sum of heat energy of refrigerant entering vessel = Sum of heat energy of refrigerant leaving vessel [reference 1]

that is,

$m^{\cdot}_2\ h6= m^{\cdot}\ h7+m^{\cdot}_1\ h8$..(3.6)

where

- $m^{\cdot}_1$: mass flow rate in the evaporator, [kg/s].
- $m^{\cdot}_2$: mass flow rate in the condenser, [kg/s].
- $m^{\cdot}$: mass flow rate in the intermediate line, [kg/s].
- h6 : enthalpy of the saturated liquid refrigerant entering the vessel at point 6, [kJ/kg]
- h7 : enthalpy of saturated vapor refrigerant from vessel at point 7, [kJ /kg]
- h8 :enthalpy of saturated liquid from vessel at point 8, [kJ /kg]

When $\dot{m}_1 = 0.00454$ [kg/s], and substitute equation (3.5) in equation (3.6) ,resulted

$\dot{m} = 0.0042$ [kg/s]

$\dot{m}_2=0.00454+0.0042$

$\dot{m}_2=0.00874$[kg/s]

3.2.2.3 Enthalpy of vapor mixture entering second compressor

Ignoring the heat loss from mixing point 3 to the surroundings, we see that the mixing of the gaseous refrigerant discharged from the first compressor at point 2 and the vaporized refrigerant from the vessel at point 7 is an adiabatic process. The heat balance at the mixing point before the second compressor, as shown in Fig. 3.3.b, is given as [reference 1]

$\dot{m}_2\, h3= \dot{m}_1\, h2 +\dot{m}\, h7$ ....(3.7)

where:

h2 : enthalpy of gaseous refrigerant discharged from first compressor, [kJ /kg]

h3 : enthalpy of mixture at point 3, [kJ /kg]

0.00874*h3 =0.00454*370+0.0042*363.8

h3= 367[kJ/kg]

at isentropic process , h4 =397[kJ/kg]

3.2.2.4 Condenser load

the condenser load can be calculated as

$Q_c = \dot{m}_2\,(h4-h5)$ ....(3.8)

Where

Q_c : condenser load [kW]

h4 : enthalpy of the hot gas discharged from the second compressor, [kJ /kg].

$Q_c = 0.00874(397-272.6)$

$Q_c = 1.087$ [kW]

3.2.2.5 Work of compressors

The work of the compressors can be calculated as

$W_L = \dot{m}_1(h2-h1)$ ....(3.9)

Where

W_L : low cycle compressor work [kW]

$W_L = 0.00454(370-340.6)$

$W_L = 0.133[kW]$

$W_H = \dot{m}_2(h4-h3)$ ....(3.10)

Where

W_H : high stage compressor work [kW]

$W_H = 0.00874(397-367)$

$W_H = 0.262[kW]$

3.2.2.6 Coefficient of performance

$$COP = \frac{Qe}{Wtotal} \qquad (3.11)$$

Where

COP : Coefficient of performance of the cycle

W_{total} : Total work input to the compressors , [kW]

$W_{total} = W_L + W_H$ ....(3.12)

$W_{total} = 0.133+0.262$

$W_{total} = 0.395$ [kW]

$$COP = \frac{0.693}{0.395}$$

$$COP = 1.75$$

CHAPTER FOUR

CYCLE COMPONENTS

CHAPTER FOUR

CYCLE COMPONENTS

4.1 Introduction

There are several mechanical components required in two stage vapor compression refrigeration system. In this part, we discuss the seven major components of a system and some auxiliary equipment working with these major components. [reference 3]

Major components of the two stage refrigeration system are as follows:

- Two compressors
- condenser
- evaporator
- throttling devices
- solenoid valve
- vessel
- connecting pipes

4.2 Compressors

In a refrigeration cycle, the compressor has two main functions within the refrigeration cycle. One function is to pump the refrigerant vapor from the evaporator so that the desired temperature and pressure can be maintained in the evaporator. The second function is to increase the pressure of the refrigerant vapor through the process of compression, and simultaneously increase the temperature of the refrigerant vapor. By this change in pressure the superheated refrigerant flows through the system. Refrigerant compressors, which are known as the heart of the refrigeration systems, can be divided into three main categories: [reference 3]

- Hermetic compressor.
- Semi hermetic compressor.
- Open compressor.

The refrigerant compressors are expected to meet the following requirements: [reference 3]

- high reliability,
- long service life,
- easy maintenance,
- quiet operation,
- compactness, and
- low cost

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

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

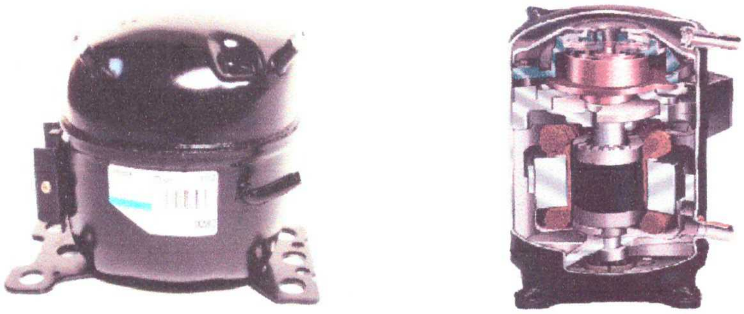

Figure 4.1 hermetic compressors

4.3 Condensers

A condenser is a major system component of a refrigeration system. It is also an indirect-contact heat exchanger in which the total heat rejected from the refrigerant is removed by a cooling medium, usually air or water. As a result, the gaseous refrigerant is cooled and condensed to liquid at the condensing pressure. [reference 13]

4.3.1 Classification of condensers:

Based on the external fluid, condensers can be classified as: [reference 8]

a) Air cooled condensers,

b) Water cooled condensers, and

c) Evaporative condensers.

Air-cooled condenser type was selected

4.3.2 Air-cooled condensers

As the name implies, in air-cooled condensers air is the external fluid, i.e., the refrigerant rejects heat to air flowing over the condenser. Air-cooled condensers can be further classified into natural convection type or forced convection type. [reference 14]

Forced convection type was selected

4.3.3 Forced convection condenser

In forced convection type condensers, the circulation of air over the condenser surface is maintained by using a fan or a blower. These condensers normally use fins on air-side for good heat transfer. The fins can be either plate type or annular type. [reference 14]

The air-cooled condensers find applications in domestic, commercial, and industrial refrigerating, chilling, freezing, and air conditioning systems, The centrifugal fan air-cooled condensers are particularly used for heat recovery and auxiliary ventilation applications. In fact, they employ outside air as the cooling medium. Fans draw air past the refrigerant coil and the latent heat of the refrigerant is removed as sensible heat by the air stream. Figure 4.2 shows air-cooled condensers[reference 3]

Figure 4.2 air-cooled condenser

4.4 Evaporators

An evaporator, like condenser is also a heat exchanger. In an evaporator, the refrigerant boils or evaporates and in doing so absorbs heat from the substance being refrigerated. The name evaporator refers to the evaporation process occurring in the heat exchanger. [reference 14]

The evaporator may be classified as natural convection type or forced convection type. In forced convection type, a fan or a pump is used to circulate the fluid being refrigerated and make it flow over the heat transfer surface, which is cooled by evaporation of refrigerant. In natural convection type, the fluid being cooled flows due to natural convection currents arising out of density difference caused by temperature difference. The refrigerant boils inside tubes and evaporator is located at the top. The temperature of fluid, which is cooled by it, decreases and its density increases. It moves downwards due to its higher density and the warm fluid rises up to replace it. [reference 14]

Air forced convection evaporator was selected .

4.5 Throttling Devices

An expansion device is another basic component of a refrigeration system. The basic functions of an expansion device used in refrigeration systems are to: [reference 14]

1. Reduce pressure from condenser pressure to evaporator pressure, and
2. Regulate the refrigerant flow from the high-pressure liquid line into the evaporator at a rate equal to the evaporation rate in the evaporator .

Under ideal conditions, the mass flow rate of refrigerant in the system should be proportional to the cooling load. Sometimes, the product to be cooled is such that a constant evaporator temperature has to be maintained. In other cases, it is desirable that liquid refrigerant should not enter the compressor. In such a case, the mass flow rate has to be controlled in such a manner that only superheated vapour leaves the evaporator. Again, an ideal refrigeration system should have the facility to control it in such a way that the energy requirement is minimum and the required criterion of temperature and cooling load are satisfied. Some additional controls to control the capacity of compressor and the space temperature may be required in addition, so as to minimize the energy consumption. [reference 14]

The expansion devices used in refrigeration systems can be divided into fixed opening type or variable opening type. As the name implies, in fixed opening type the flow area remains fixed, while in variable opening type the flow area changes with changing mass flow rates. There are basically seven types of refrigerant expansion devices. These are: [reference 14]

1. Hand (manual) expansion valves
2. Capillary Tubes
3. Orifice
4. Constant pressure or Automatic Expansion Valve (AEV)
5. Thermostatic Expansion Valve (TEV)
6. Float type Expansion Valve
    a) High Side Float Valve
    b) Low Side Float Valve
7. Electronic Expansion Valve

Of the above seven types, capillary tube and orifice belong to the fixed opening type, while the rest belong to the variable opening type. Of the above seven types, the hand operated expansion valve is not used when an automatic control is required. The orifice type expansion is used only in some special applications. [reference 14].

The thermostatic expansion valve was chosen to be used before the evaporator ,and the float valve to be used before the vessel .

4.5.1 Thermostatic Expansion Valve (TEV)

Thermostatic expansion valve is the most versatile expansion valve and is most commonly used in refrigeration systems. A thermostatic expansion valve maintains a constant degree of superheat at the exit of evaporator; hence it is most effective for dry evaporators in preventing the slugging of the compressors since it does not allow the liquid refrigerant to enter the compressor. [reference 14] For this reason the thermostatic expansion valve was selected in this project .

The essential objective of the expansion valve is to regulate a flow rate of refrigerant to the evaporator that matches the rate boiled off. The valve accomplishes this objective by controlling the amount of refrigerant superheat leaving the evaporator.

As shown in figure 4.3. The valve stem is positioned by the pressure difference on opposite sides of the diaphragm. The pressure under the diaphragm is provided by the refrigerant at the entrance to the evaporator, and the pressure on the top side of the diaphragm by what is called the power fluid. In the basic expansion valve, the power fluid is the same refrigerant used in the system. It is in vapor form, except for a small amount of liquid in the sensing bulb. [reference 14]

A slight force exerted by the spring on the valve stem keeps the valve closed until the pressure above the diaphragm overcomes the combined forces of the spring and the evaporator pressure. For the pressure above the diaphragm to be higher than the evaporator pressure below the diaphragm, the power fluid temperature must be higher than the saturation temperature in the evaporator. The suction gas must, therefore, be superheated to bring the power fluid up to the valve-opening pressure. [reference 14]

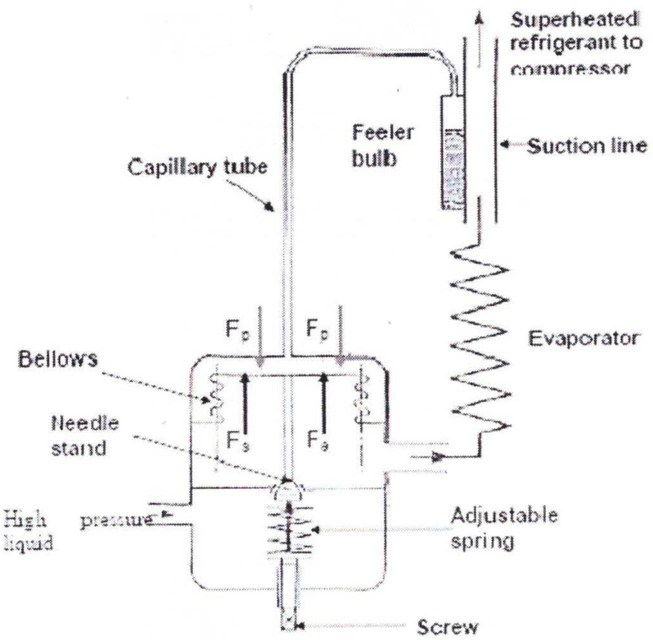

Figure 4.3schematic of an thermostatic expansion valve

4.5.2 float expansion valve

Float type expansion valves are normally used with flooded evaporators in large capacity refrigeration systems. A float type valve opens or closes depending upon the liquid level as sensed by a buoyant member, called as float. The float could take the form of a hollow metal or plastic ball, a hollow cylinder or a pan. Thus the float valve always maintains a constant liquid level in a chamber called as float chamber. Depending upon the location of the float chamber, a float type expansion valve can be either a low-side float valve or a high-side float valve. [reference 14]

4.5.2.1 low-side float valves:

Figure 4.4 shows the schematic of a low-side float valve. As shown in the figure, a low-side float valve maintains a constant liquid level in a flooded evaporator or a float chamber attached to the evaporator. When the load on the system increases, more amount of refrigerant evaporates from the evaporator. As a result, the refrigerant liquid level in the evaporator or the low-side float chamber drops momentarily. The float then moves in such a way that the valve opening is increased and more amount of refrigerant flows into the evaporator to take care of the increased load and the liquid level is restored. The reverse process occurs when the load falls, i.e., the float reduces the opening of the valve and less amount of refrigerant flows into the evaporator to match the reduced load. As mentioned, these valves are normally used in large capacity systems and normally a by-pass line with a hand-operated expansion is installed to ensure system operation in the event of float failure. [reference 2]

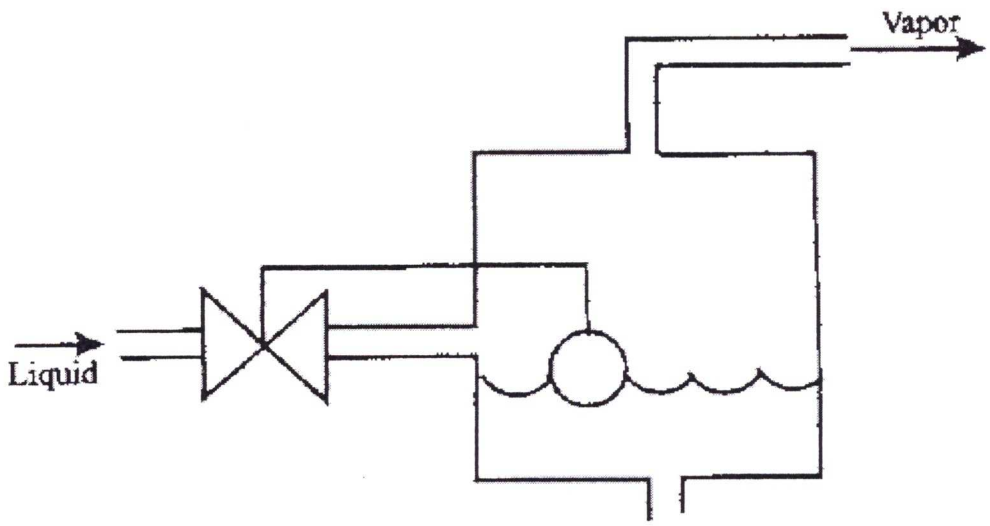

Figure 4.4 : Schematic of a low-side float valve

In the project the level switch was selected with the solenoid valve to work to gather as a float expansion valve. When the liquid level drops under specific level the level switch will give on condition as electrical signal, this signal reserved by the solenoid valve which is opened .The reversed process happened when the liquid level raised over the specific level . The figure 4.5 shows the level switch state. [reference 15]

Figure 4.5 level switch

4.6 Vessel (flash tank)

In compound systems, the vessel are used to subcool liquid refrigerant to the saturated temperature corresponding to the intermediate pressure by vaporizing part of the liquid refrigerant. The vessel are used to desuperheat the discharge gas from the low-stage compressor and, more often, to subcool also the liquid refrigerant before it enters the evaporator. [reference 1]

Suitable equipment to partially expand the refrigerant and then remove the flash gas is shown schematically in Figure 4.6. Liquid refrigerant from the condenser or high-pressure receiver passes through a level-control valve(float valve). The liquid, being more dense than the vapor, separates and flows on to the expansion valve of the evaporator. A second compressor draws off the vapor from the separating vessel or flash tank and the discharge of the first compressor and compresses it to the condensing pressure. [reference 14]

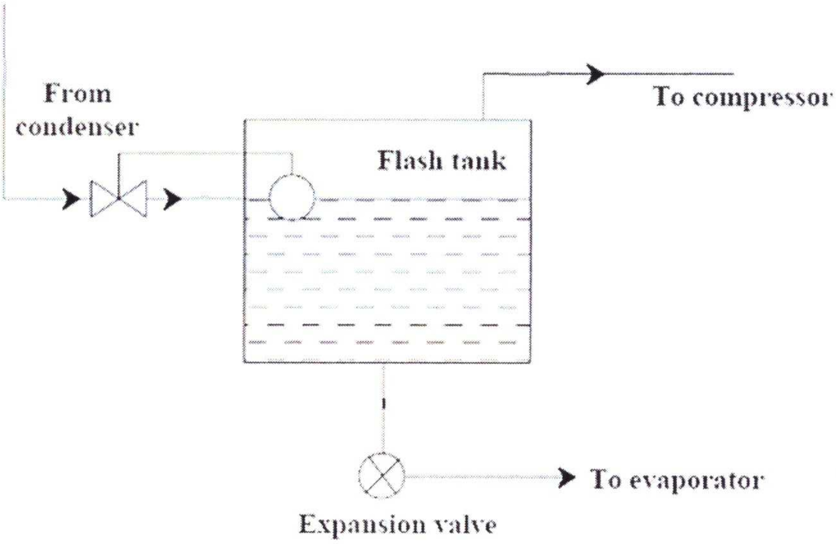

Figure 4.6 vessel

4.7 Connecting pipes

Refrigerant pipes transport refrigerant through the compressor, condenser, expansion valve, and evaporator to provide the required refrigeration effect. For halocarbon refrigerants, the refrigerant pipes are usually made of copper. However, if ammonia is used as the refrigerant, the pipes are always made of steel. [reference 1]

Three types of copper tubing are used for refrigerant piping. Type K is heavy-duty, and type M is lightweight. Type L is the standard copper tubing most widely used in refrigeration systems. Copper tubing installed in refrigeration systems should be entirely free of dirt, scale, and oxides. The open ends of clean new tubes should be capped to keep contaminants out. [reference 1]

4.8 Auxiliary component

The auxiliary components is very important in the refrigeration system its working together with main components allowing system works very well, and we will discuss some auxiliaries in the following. [reference 13]

4.8.1 Filter Dryer

Moisture may freeze in the expansion valve, especially in a low temperature system, and impair expansion valve operation. Foreign matter in the refrigeration system may also damage the expansion valve, control valves, and compressor. Moisture and foreign matter must be removed from the refrigeration system by means of a filter dryer. Figure 4.7 shows a filter dryer. It contains a molded, porous core made of material with high moisture affinity, such as activated alumina, silica gel , and acid-neutralizing agents to remove moisture and foreign matter from the refrigerant. The core should be replaced after a certain period of operation. The filter dryer is usually installed at the liquid line immediately before the expansion valve [reference 1].

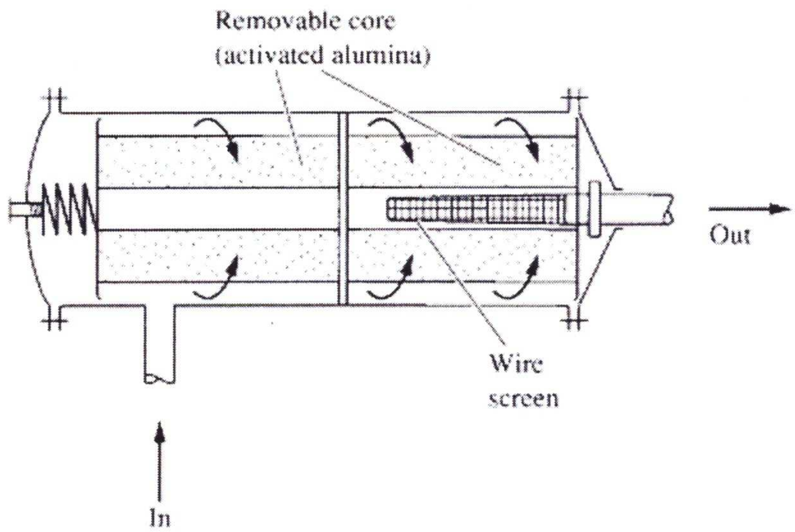

Figure 4.7 filter dryer

4.8.2 Sight Glass

A sight glass is a small glass port used to observe the condition of refrigerant flow in the liquid line. The sight glass is located just before the expansion valve. Bubbles seen through the sight glass indicate the presence of the flash gas instead of liquid refrigerant. The presence of flash gas always indicates that the evaporator's capacity is reduced because of a shortage of refrigerant or insufficient subcooling. [reference 1]

Figure 4.8 : sight glass

CHAPTER FIVE

COMPONENTS CALCULATIONS AND SELECTION

CHAPTER FIVE

COMPONENTS CALCULATIONS AND SELECTION

5.1 Compressors Calculations And Selection

5.1.1 for low compressor

To determine the volumetric efficiency for the compressor can be used the equation

$\eta_v = \eta_c * \eta_h$..(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 ,[reference 6]:

$\eta_c = 1 - c[(\frac{PH}{PL})^{1/n} - 1]$ ....(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 7]:

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

PH: High pressure of the cycle

PL :Low pressure of the cycle

η_c =1- 0.02[$(\frac{4.64}{1.055})^{1/1}$ -1] = 93.2 %

The volumetric efficiency due to the heating in compressor is, [reference 6]:

$$\eta_h = \frac{\text{Tevap}}{\text{Tcond}} \qquad (5.3)$$

Where :

Tevap. :evaporator temperature [°K]

Tcond. :condenser temperature [°K]

$$\eta_h = \frac{228}{265} = 86\%$$

η_v =93.2%*86% =0.8%

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

$$V^{\cdot}_{theo} = m^{\cdot} * \upsilon \qquad (5.4)$$

Where :

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

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

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

$V^{\cdot}_{theo}$= 0.00454*0.17758 =$8*10^{-4}$ [m^3/s]

To determine the actual volume flow rate by the equation, [reference 6]:

$$V^{\cdot}_{act} = \frac{V.theo}{\eta v} \qquad (5.5)$$

Where :

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

$$V^{\cdot}_{act} = \frac{0.0008}{0.8} = 1*10^{-3} [m^3/s]$$

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

From Tecumseh company catalog we chosed the compressor with code number CAJ2446Z which have displacement 26.2 cm^3 per revolution and 3000 RPM

So

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

$V^{\cdot}_{act} = V^{\cdot}_{theo} * \eta_v$ ....(5.6)

Where :

$V^{\cdot}_{act}$: actual volumetric flow rate for the compressor $[m^3/s]$

$V^{\cdot}_{theo}$:theoretical volumetric flow rate for the compressor $[m^3/s]$

η_v: volumetric efficiency

$$V^{\cdot}_{theo} = 26.2 * 10^{-6} *(3000/60) = 1.31*10^{-3}\ [m^3/s]$$

$$V^{\cdot}_{act} = 1.27*10^{-3} * 0.8 = 1.048*10^{-3}\ [m^3/s]$$

5.1.2 For high compressor

$\eta_v = \eta_c * \eta_h$

$\eta_c = 1 - 0.02[\ (\frac{20.4}{4.64})^{1/1} - 1] = 93\ \%$

$\eta_h = \frac{265}{318} = 83\%$

$\eta_v = 93\% * 83\% = 77\%$

$V^{\cdot}_{theo} = m^{\cdot}_2 * \upsilon$

$V^{\cdot}_{theo} = 0.00874*0.043 = 3.7*10^{-4}\ [m^3/s]$

$V^{\cdot}_{act} = \frac{0.00037}{0.77} = 4.8*10^{-4} [m^3/s]$

From Tecumseh company catalog we chosed the compressor with code number CAE2420Z which have displacement 12.6 cm^3 per revolution and 3000 RPM

So

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

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

$V^{\cdot}_{theo} = 12.6 * 10^{-6} * (3000/60) = 6.3*10^{-4}$ [m^3/s]

$V^{\cdot}_{act} = 6.3*10^{-4} * 0.77 = 4.851*10^{-4}$ [m^3/s]

5.2 Pipe Design and Selection

5.2.1 Introduction

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

To calculate the inner diameter for the pipe can be used the following steps:[reference8]

$Q = m^{\cdot} * \upsilon$ ....(5.7)

Where:

Q: flow rate[m^3/s]

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

υ : specific volume [m^3/kg] ,[table 3.2]

$Q = V*A$ ....(5.8)

Where :

V: velocity of refrigerant[m/s] . [table A-6]

A: cross sectional area[m^2]

$A = \pi d_i^2 /4$ ....(5.9)

Where :

d_i : inner diameter [m]

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

To calculate the outer diameter for the pipe can be as the following:[reference 9]

$$б_t = \frac{Pin(ro^2 + ri^2)}{(ro^2 - ri^2)} \quad \text{..........(5.10)}$$

where :

$б_t$:tangential stress [Mpa]

P_{in} : inner pressure [Mpa]

r_o : outer radius [m]

r_i : inner radius [m]

$б_t$ can be calculated from the following equation:[reference 9]

$$\frac{SY}{n} = \sqrt{б t^2 + Pin\ б t + Pin^2} \quad \text{..........(5.11)}$$

Where :

S_Y: yield strength [Mpa],[70 Mpa for copper],[reference 10]

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

now by using equation (5.10) can be calculate the outer radius for the pipe

$t = r_o - r_i$..........(5.12)

where :

t :thickness of the pipe [mm]

5.2.2 Low Cycle Pipes

- Suction line pipe

$Q = m^{\cdot}_1 * \upsilon_1$

$Q = 0.00454*0.17758 = 8*10^{-4}$ [m³/s]

$$A = \frac{Q}{V} = \frac{0.0008}{10} = 8*10^{-5}[m^2]$$

$$d_i = \sqrt{\frac{0.00008*4}{\pi}} = 10*10^{-3}\ [m]$$

$$\frac{700}{8} = \sqrt{6t^2 + 1.055\ 6t + (1.055)^2}$$

$б_t = 87$ [bar]

$$87 = \frac{1.055(ro^2 + 0.005^2)}{(ro^2 - 0.005^2)}$$

r_o =5.06 [mm]

t = 5.06-5 =0.06 [mm]

The inner and outer diameter in inch is

$d_{i,\ inch} = d_{i,\ mm}/25.4 = 10/25.4 = 0.393$ [inch]

$d_{o,\ inch} = d_{o,\ mm}/25.4 = 10.12/25.4 = 0.398$ [inch]

By referring to copper hand book (table A-7), the suitable type selected is ACR type (Air-conditioning and Refrigeration Field Service), and according to pervious calculations, the nominal or standard size (inches) for this section is 1/2 D which has outer diameter 0.5 inch, and inside diameter 0.45 inch .

- discharge line pipe

$Q = \dot{m}_1 * \upsilon_2$

$Q = 0.00454*0.0448 = 2*10^{-4}\ [m^3/s]$

$$A = \frac{Q}{V} = \frac{0.0002}{12} = 1.66*10^{-5}[m^2]$$

$$d_i = \sqrt{\frac{0.0000166*4}{\pi}} = 4.6*10^{-3}\ [m]$$

$$\frac{700}{8} = \sqrt{бt^2 + 4.64\,бt + (4.64)^2}$$

$б_t = 85$ [bar]

$$85 = \frac{4.64(ro^2 + 0.0023^2)}{(ro^2 - 0.0023^2)}$$

$r_o = 2.43$ [mm]

t = 2.43-2.3 =0.13 [mm]

The inner and outer diameter in inch is

$d_{i,\ inch}$ =4.6/25.4=0.181 [inch]

$d_{o,\ inch}$ = 4.86/25.4=0.191 [inch]

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

- liquid line pipe

$Q = m^{\cdot}_1 * \upsilon_8$

$Q = 0.00454*0.001 = 4.54*10^{-5}$ [m^3/s]

$$A = \frac{Q}{V} = \frac{0.0000454}{1} = 4.54*10^{-5} [m^2]$$

$$d_i = \sqrt{\frac{0.0000454*4}{\pi}} = 7.6*10^{-3} [m]$$

$$\frac{700}{8} = \sqrt{бt^2 + 4.64\,бt + (4.64)^2}$$

$б_t = 85$ [bar]

$$85 = \frac{4.64(ro^2 + 0.0038^2)}{(ro^2 - 0.0038^2)}$$

$r_o = 4.0134$ [mm]

t = 4.0134-3.8 =0.2134 [mm]

The inner and outer diameter in inch is

$d_{i,\ inch}$ =7.6/25.4=0.299 [inch]

$d_{o,\ inch}$ = 8.068/25.4=0.316 [inch]

According to pervious calculations, the nominal or standard size (inches) for this section is 3/8D which has outer diameter 0.375 inch, and inside diameter 0.311 inch .

5.2.3 High Cycle Pipes

- Suction line pipe

$Q=\dot{m}_2 * \upsilon_3$

$Q= 0.00874*0.0435 =3.8*10^{-4}$ [m^3/s]

$$A=\frac{Q}{V}=\frac{0.00038}{12}=3.16*10^{-5}[m^2]$$

$$d_i=\sqrt{\frac{0.0000316*4}{\pi}}=6.34*10^{-3}\ [m]$$

$$\frac{700}{8}=\sqrt{6t^2+4.64\,6t+(4.64)^2}$$

$6_t = 85$ [bar]

$$85=\frac{4.64(ro^2+0.00317^2)}{(ro^2-0.00317^2)}$$

r_o =3.36 [mm]

t = 3.36-3.17 =0.19 [mm]

The inner and outer diameter in inch is

$d_{i,\ inch}$ =6.34/25.4=0.248 [inch]

$d_{o,\ inch} = d_{o,\ mm}$/25.4 =6.72/25.4=0.264 [inch]

According to pervious calculations, the nominal or standard size (inches) for this section is 5/16D which has outer diameter 0.312inch, and inside diameter 0.248 inch .

- discharge line pipe

$Q=m_2^{\cdot}*\upsilon_4$

$Q= 0.00874*0.01 =0.874*10^{-4}$ [m^3/s]

$$A=\frac{Q}{V}=\frac{0.0000874}{13}=6.72*10^{-6}[m^2]$$

$$d_i=\sqrt{\frac{0.00000672*4}{\pi}}=3*10^{-3}\ [m]$$

$$\frac{700}{8}=\sqrt{бt^2+20.4\,бt+(20.4)^2}$$

$б_t = 75.5$ [bar]

$$75.5=\frac{4.64(ro^2+0.0015^2)}{(ro^2-0.0015^2)}$$

r_o =1.76 [mm]

t = 1.76-1.5 =0.26 [mm]

The inner and outer diameter in inch is

$d_{i,\ inch}$ =3/25.4=0.118 [inch]

$d_{o,\ inch} = d_{o,\ mm}$ /25.4 =3.52/25.4=0.138 [inch]

According to pervious calculations, the nominal or standard size (inches) for this section is 3/16D which has outer diameter 0.187inch, and inside diameter 0.128 inch

- liquid line pipe

$Q=m_2^{\cdot}*\upsilon_5$

$Q= 0.00874*0.001 =8.74*10^{-6}$ [m^3/s]

$A=\frac{Q}{V}=\frac{8.74*10-6}{1.2}$ = 7.283*10^{-6}[m^2]

$d_i=\sqrt{\frac{7.283*10-6*4}{\pi}}$ = 3*10^{-3} [m]

$$\frac{700}{8}=\sqrt{бt^2+20.4\,бt+(20.4)^2}$$

$б_t$ = 75.5 [bar]

$75.5=\frac{4.64(ro^2+0.0015^2)}{(ro^2-0.0015^2)}$

r_o =1.76 [mm]

t = 1.76-1.5 =0.26 [mm]

The inner and outer diameter in inch is

$d_{i,\ inch}$ =3/25.4=0.118 [inch]

$d_{o,\ inch}$ = $d_{o,\ mm}$ /25.4 =3.52/25.4=0.138 [inch]

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

5.2.4 Intermediate Line Pipe

Q=m˙ *υ_7

Q= 0.0042*0.0431 =1.81*10^{-4} [m^3/s]

$A=\frac{Q}{V}=\frac{0.000181}{12}$ = 1.5*10^{-5}[m^2]

$d_i=\sqrt{\frac{0.000015*4}{\pi}}$ = 4.3*10^{-3} [m]

$$\frac{700}{8}=\sqrt{бt^2+4.64\,бt+(4.64)^2}$$

$б_t = 85$ [bar]

$$85 = \frac{4.64(ro^2 + 0.00215^2)}{(ro^2 - 0.00215^2)}$$

$r_o = 2.27$ [mm]

t = 2.27-2.15 =0.12 [mm]

The inner and outer diameter in inch is

$d_{i,\,inch} = 4.3/25.4 = 0.172$ [inch]

$d_{o,\,inch} = d_{o,\,mm}/25.4 = 4.54/25.4 = 0.178$ [inch]

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

5.3 Evaporator Calculations and selection

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

evaporator length Lc =53 [cm]

evaporator height H_c =5 [cm]

evaporator width W_c =25[cm]

S_n: Transverse tube spacing [m]

S_p: Longitudinal tube spacing [m]

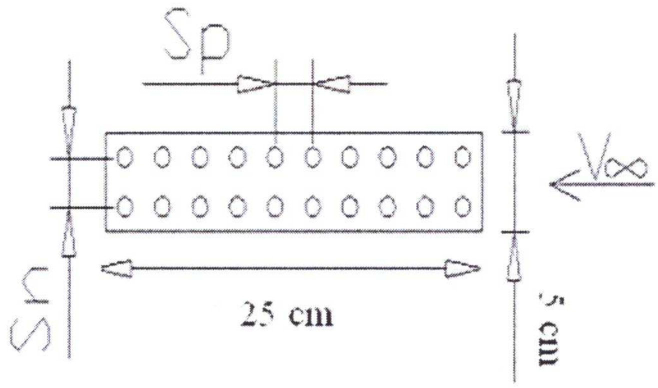

Figure 5.1 evaporator side view

$$S_n = \frac{\text{evaporator height}}{\text{number of rows}} = \frac{0.05}{2} = 0.025\ [\text{m}]$$

$$S_p = \frac{\text{evaporator width}}{\text{number of column}} = \frac{0.25}{10} = 0.025[\text{m}]$$

$$fin\ length\ = L_f\ = (S_n - D_0)$$

$$L_f = (0.025 - 0.01) = 0.015[m]$$

$$\text{Fine width} = W_f = (S_p - D_o)$$

$$W_f = (0.025 - 0.01) = 0.015[\text{m}]$$

Figure (5.2) show the fin elements

number of fins in row =89 fins

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

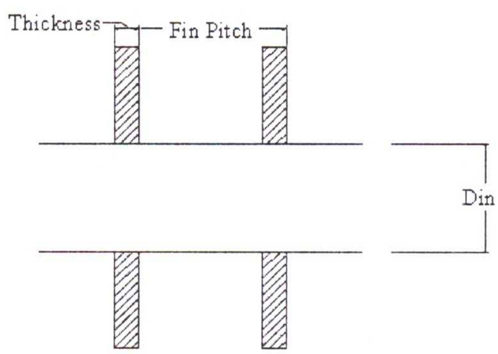

figure 5.2 fin elements for evaporator

$\text{Fine thickness} = t_f = 0.03[\text{cm}]$

$\text{bare tube thickness} = t_b = P_f - t_f$

$t_b = 0.59 - 0.03 = 0.56[\text{cm}]$

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

$Q = h_o * A * (T_w - T_\infty)$..(5.14)

Where

Q: heat transfer through the evaporator [W]

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

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

T_w: outer evaporator wall temperature [^oC]

T_∞: free air temperature

The heat that transferred through the evaporator was determined in chapter three and its value was $Q_e = 693$ [W]

To determine the external convection heat transfer coefficient can be as the following equations [reference 8]

$h_o = \frac{Nu * k}{D}$..(5.15)

where

Nu: nusselts number

K: thermal conductivity of air at entrance of evaporator[W/m^oC]

D: outer diameter of evaporator [m]

The Nusselts number can be calculated by the equation [reference 8]

$Nu = C(Re)^N Pr^{1/3}$..(5.16)

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 8]

$S_p/d = 25/10 = 2.5$

$S_n/d = 25/10 = 2.5$

From In line arrangement tube banks table

C =0.3

N=0.595

$$Re = \frac{\rho * Vmax * D}{\mu} \quad \text{..........(5.17)}$$

Where

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

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

D :outer diameter of the evaporator tubes [m]

μ :dynamic viscosity of air at film temperature [Pa.s]

For flows normal to in -line tube banks the maximum flow velocity can be calculated as the following [reference 8]

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

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

$$V\infty = \frac{V^{\cdot}}{A} \quad \text{..........(5.19)}$$

Where

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

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

A = 0.55*0.05 =0.0275 [m^2]

$V\infty = \frac{0.2359}{0.0275} = 0.857 m/s$

$V_{max} = 0.857 \frac{0.025}{0.025-0.01}. = 1.4 [m/s]$

the properties of air are evaluated at the film temperature ,which at entrance to the tube bank is [reference 8]

$$Tf = \frac{Tw+T\infty}{2} \quad \text{......(5.20)}$$

Where

Tf : film temperature[°C]

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

T∞:free air temperature[°C]

$$Tf = \frac{-45+-35}{2} = -40 \ [°C] = 233[°K]$$

Then from table A-9

$\rho = 1.5 \ [kg/m^3]$

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

$K = 0.02 \ [W/m.°C]$

$Pr = 0.73$

$Re = \frac{1.5*1.4*0.01}{1.516*10^{-5}} = 1413$

$Nu = 0.3(1413)^{0.595} * (0.73)^{1/3} = 20.2$

$h_o = \frac{20.2*0.02}{10*10^{-3}} = 40.4 \ [W/m^{2o}C]$

In order to calculate the total heat transfer from one element (one fin and one bare tube) the following equation is used [reference 11]:

$$q_{total} = q_{fin\ act} + q_{original} \quad \text{......(5.21)}$$

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

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

Where

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

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

T_w: outer evaporator wall temperature [ºC]

T_∞: free air temperature

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

$q_{original} = 40.4*1.7*10^{-4} *(-35- -45) = 0.068$ [W]

q_{fin} can be calculated by the equation

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

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]

$q_{fin} = 40.4*10*10^{-4}$ (-35- -45) =0.404 [W]

- Fin efficiency calculation:

$$\eta_f = \frac{\tanh(m\grave{L}_f)}{m\grave{L}_f} \quad \text{......} \quad (5.22)$$

$$\grave{L}_f = \left(\frac{L_f}{2}\right)\left[1 + 0.35\ln\frac{(\frac{D_0}{2}+\frac{L_f}{2})}{\frac{D_0}{2}}\right] \quad \text{......} \quad (5.23)$$

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

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

Where

h: external convection heat transfer coefficient [W/m^{2o}C] k: thermal conductivity of aluminum fin , [W/m°C]

P :perimeter of the fin[m]

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

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

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

$$m = \sqrt{\frac{40.4 * 0.05}{202 \times 7.5 \times 10^{-6}}} = 36.5$$

$$\eta_f = \frac{\tanh(36.5 * 0.01)}{36.5 * 0.01} = 0.95$$

So

The heat transfer flow from the fin is

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

$q_{fin\ act}$ = 0.404*0.95= 0.3838[W]

Now the total heat transfer from the element is

q_{total} = 0.3838 + 0.068 =0.4518 [W]

Now the number of elements 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:

$$n = \frac{Qe}{q\ total}$$

$$n= \frac{693}{0.4518} = 1534 \text{ elements}$$

number of elements in available evaporator = N*R

where

N: number of elements in rows

R: number of rows

number of elements in available evaporator = 89*20=1780 elements [that is enough]

5.4 Condenser calculations and selection

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

condenser length L_c =30 [cm]

condenser height H_c =26 [cm]

condenser width W_c =5.7[cm]

S_n: Transverse tube spacing [m]

S_p: Longitudinal tube spacing [m]

$$S_n = \frac{\text{evaporator height}}{\text{number of rows}} = \frac{0.26}{10} = 0.026[\text{m}]$$

$$S_p = \frac{\text{evaporator width}}{\text{number of column}} = \frac{0.57}{3} = 0.019[\text{m}]$$

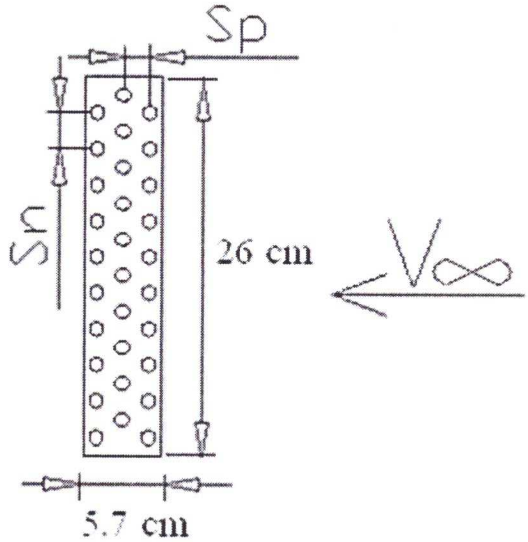

Figure 5.3 condenser side view

Figure 5.3 condenser side view

fin length $= L_f = (S_n - D_0)$

$L_f = (0.026 - 0.01) = 0.016[\text{m}]$

Fine width $= W_f = (S_p - D_o)$

$W_f = (0.019 - 0.01) = 0.009[\text{m}]$

Figure (5.4) show the fin elements

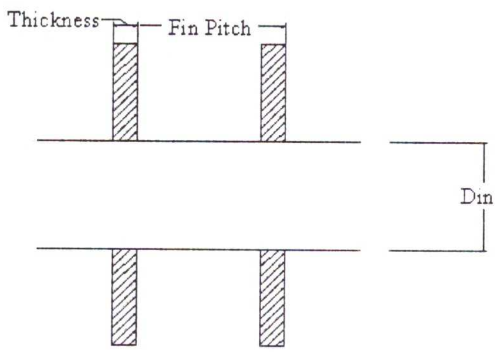

figure 5.4fin elements for condenser

number of fins in a row =94 fins

fin pitch $=P_f=\frac{30}{94} = 0.32$ [cm]

Fine thickness $= t_f = 0.02[\text{cm}]$

bare tube thickness $= t_b = P_f - t_f$

$t_b = 0.32 - 0.02 = 0.3[\text{cm}]$

We take condenser as two regions, one for one phase (desuperheat) region and the other for two phase (mixture) region

➤ Desuperheating region

$Q_{desuper} = m^{\cdot} (h_{in} - h_{sat.\ vap.})$ ....(5.26)

Where

$Q_{desuper}$: condenser load from desuperheating region [W]

$m^{\cdot}$: mass flow rate in the condenser, [kg/s].

h_{in} :enthalpy of refrigerant entering the condenser , [kJ /kg]

$h_{sat.vap.}$: enthalpy of saturated vapor leaving the desuperheating region in condenser , [kJ /kg]

$Q_{desuper} = 0.00874\ (397 - 387) = 87[W]$

$Q_{desuper} = h_o * A * (T_w - T_\infty)$ ....(5.27)

Where

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

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

T_w: outer condenser wall temperature [^oC]

T_∞: free air temperature

$S_p/D = 19/10 = 1.9$

$S_n/D = 26/10 = 2.6$

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

$$V_{max} = \frac{V\infty\left(\frac{Sn}{2}\right)}{\sqrt{\left[\left(\frac{Sn}{2}\right)^2 + Spl^2\right] - D}} \quad \text{...(5.28)}$$

$$V\infty = \frac{V^{\cdot}}{A} \quad \text{...(5.29)}$$

Where

$V^{\cdot}$: flow rate of air in the condenser [m^3/s], $V^{\cdot}$= 700 [m^3/h]=0.19444 [m^3/s]from fan manufacturer company.

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

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

$$V\infty = \frac{0.19444}{0.078} = 2.492 m/s$$

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

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

$$Tf = \frac{Tav.w + T\infty}{2} \quad \text{...(5.30)}$$

Where

Tf : film temperature[°C]

T∞:free air temperature[°C]

Tav.w : average wall surface temperature[°C]

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

$$Tf = \frac{50+38}{2} = 44\ [°C] = 317\ [°K]$$

Then from table A-9

ρ =1.1176 [kg/m^3]

μ= 1.85 *10^{-5} [kg/m.s]

K =0.0275 [W /m.°C]

Pr =0.704

So

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

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

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

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 3rows ,this value must be multiplied by the factor 0.83 ,as determined from table A-10 [reference 8]

so h_o= 80.3*0.83=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 11]:

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

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

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

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

q_{fin} can be calculated by the equation

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

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

$q_{fin} = 66.6*8.3*10^{-4}$ (50-38) =0.6633 [W]

- Fin efficiency calculation:

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

P= 2*t+2*L =2*0.2*10^{-3}+2*0.026 =0.052[m]

A :t*L =0.2*10^{-3}*0.026 =5.2*10^{-6}[m^2]

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

$$\eta_f = \frac{\tanh(57.4 * 0.01)}{57.4 * 0.01} = 0.9$$

So

The actual heat transfer flow through the fin is

$q_{fin\ act} = 0.6633*0.9= 0.597[W]$

Now the total heat transfer from the element is

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

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

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

$$n= \frac{87}{0.6723} =130 \text{ elements}$$

- **Mixture region**

$Q_{mix} = m^{\cdot}\ (h_{sat.vap}- h_{sat.\ liq.})$..(5.31)

Where

Q_{mix} : condenser load from mixture region [W]

$m^{\cdot}$: mass flow rate in the condenser, [kg/s].

$h_{sat.vap.}$: enthalpy of saturated vapor entering the mixture region in condenser , [kJ /kg]

$h_{sat.liq.}$: enthalpy of saturated liquid leaving the mixture region in condenser , [kJ /kg]

$Q_{mix} =0.00874\ (387-272.6) = 1000[W]$

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

$Tf = \frac{Tw+T\infty}{2}$..(5.32)

Where

Tf : film temperature[°C]

$T\infty$:free air temperature[°C]

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

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

Then from table A-9

ρ =1.13 [kg/m^3]

μ= 1.848 *10^{-5} [kg/m.s]

K =0.0275 [W /m.ºC]

Pr =0.704

$Re = \frac{1.13*2.48*0.01}{1.848*10^{-5}}$ =1516.4

Nu=0.521(1516.4)$^{0.565}$ * (0.73)$^{1/3}$ =29.4

$h_o = \frac{29.4*0.0275}{10*10^{-3}}$ = 80.85 [W/m^{2o}C]

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 3rows ,this value must be multiplied by the factor 0.83 ,as determined from table A-10 [reference 8]

so h_o= 80.85*0.83=67.1[W/m^{2o}C]

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

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

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

$q_{original}$ = 67.1*94.24*10^{-6} *(45-38) =0.04426 [W]

q_{fin} can be calculated by the equation

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

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

q_{fin} = 67.1*8.3*10^{-4} (45-38) =0.39 [W]

➢ Fin efficiency calculation:

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

$$m = \sqrt{\frac{67.1 * 0.052}{202 \times 5.2 \times 10^{-6}}} = 57.63$$

$$\eta_f = \frac{\tanh(57.3 * 0.01)}{57.63 * 0.01} = 0.90$$

So

The actual heat transfer flow through the fin is

$q_{fin\ act} = 0.39*0.90 = 0.351[W]$

Now the total heat transfer from the element is

$q_{total} = 0.351 + 0.04426 = 0.395\ [W]$

Now the number of elements that needed to perform the condenser load from mixture region can be determined by the following equation:

$n = \frac{1000}{0.395} = 2532$ elements

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

$N_{total} = 130 + 2532 = 2662$ elements

number of elements in available condenser = N*R

where

N: number of elements in rows

R: number of rows

number of elements in available condenser= 94*30=2820 elements [that is enough]

5.5 Vessel design

Vessels in industrial refrigeration systems serve either or both of the following functions:

(1) storage of liquid

(2) separation of liquid from vapor.

With a system of multiple components, liquid is likely to move from one condenser or evaporator to another.

Also the liquid content in these components varies with time, so a vessel should be available to provide a reservoir for these changes in liquid content

5.5.1 Calculation of the vessel diameter.

To calculate the vessel diameter of the intermediate vessel the following equation is used:[reference10]

$$D = \sqrt{\frac{4*m^{\cdot}v.max}{\pi*V*\rho v}} \qquad \text{.............(5.33)}$$

Where:

V :allowable velocity of vapor in the intermediate vessel cross section;

V= 0.5 [m/s] [reference10].

ρv :vapor density [kg/m^3]

$m^{\cdot}_{v.max}$: maximum mass flow rate of vapor intering the intermediate vessel at maximum load [kg/s]. $m^{\cdot}$ =0.0042[kg/s].from chapter 3.

To find vapor density ρ_v the following equation is used:

$$\rho_v = \frac{1}{v\ @pint.}$$

Where :-

$v\ @pint.$: is the specific volume of the refrigerant taken at intermediate pressure for the cycle in [m^3/kg].

$$\rho_v = \frac{1}{v\ @pint.} = \frac{1}{0.0431} = 23.2\ [kg/m^3].$$

Back to equation (5.33)

$$D = \sqrt{\frac{4*0.0042}{\pi*0.5*23.2}} = 0.02147[m] = 2.147[cm]$$

5.5.2 Calculation Volume of the vessel

To calculate the volume of the intermediate vessel the flowing equation is used:[reference10]

$V_{int.} = V_1+V_2$. ....(5.35)

Where:

$V_{int.}$: volume of the intermediate vessel[m^3].

V_l : volume of liquid required to drive the circulating pump[m^3].

V_2 : volume of vapor in the intermediate vessel[m^3].

Volume of the liquid in the vessel can be calculated as the following(volume for the cylindrical shape)

$V_l= H_l*\frac{\pi D^2}{4}$ ....(5.36)

Where :

H : the high of the liquid in the vessel.

D: diameter of the vessel .

As a devisor for the vessel refrigeration design the liquid high should be from the range (3-6cm),taken 5cm ,and for high of vapor in the vessel the recommended ratio can be used [1:2,1:3,1:4] ,

Back to equation (5.36)

$V_l= 0.05*\frac{\pi(0.02147)^2}{4} = 0.000018m^3 =0.018L$

Volume of the vapor in the vessel can be calculated as the following equation by using ratio 1:4

$V_2 = H_2 * \frac{\pi D^2}{4}$

Where :

H : the high of the vapor in the vessel.

D: diameter of the vessel .

$V_2 = 0.2 * \frac{\pi (0.02147)^2}{4} = 0.0724$ L

So the volume of the intermediate vessel can calculated as the following

$V_{int.} = 0.018 + 0.0724 = 0.09$ L

CHAPTER SIX

ELECRICAL DESIGN

- Introduction
- Types Of Electrical Circuits
- Components Of Electrical Circuits
- Electrical Description For Two stage system
- Power Circuit

6.1 Introduction:

The refrigerators in general to worked for along period and protect its component needs electrical circuits and some times electronic circuits to controlling an mentoring there system .
So for this project which is designing a refrigerator from two stage its need electrical circuit to control and monitoring it.

6.2 Types Of Electrical Circuits:

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

6.2.1 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.

6.2.2 Power Circuit:

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

6.3 Components Of Electrical Circuits:

6.3.1 Current Relay:

This can best be described 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 contacts 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 filed 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 6.1 shows Current relay.

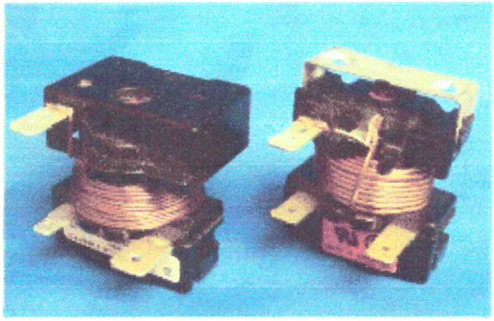

Figure 6.1Current relay

6.3.2 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 much 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 windings 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 contacts open, the voltage and current across the coil will decrease but will maintain a magnetic force strong enough to keep the contacts open until power is disconnected. The contacts will then return to the closed position ready for a restart. . Figure 6.2 shows voltage relay.

Figure 6.2voltage relay

6.3.3 Capacitor Start And Run Motor:

Construction of the capacitor start and run motor is identical to that of the capacitor start motor with the exception that a second capacitor, called a running capacitor, is installed in series with the starting winding but in parallel with the starting capacitor and starting switch. The operation of the capacitor start and run motor differs from that of the capacitor start and split-phase motors in that the starting or auxiliary winding remains in the circuit at all time. At the instant of starting, the starting-and-running capacitors are both in the circuit in series with the auxiliary winding so that the capacity of both capacitors is utilized during the starting period. As the rotor approaches 70% of rated speed, the centrifugal mechanism opens the starting switch and removes the starting capacitor from the circuit, and the motor continues to operate with both main and auxiliary windings in the circuit. The function of the running capacitor in series with the auxiliary winding is to correct the power factor. As a result the capacitor run and start motor not only has a high starting torque but also an excellent running efficiency. Figure 6.3 shows run and start Capacitors.

Figure 6.3 Run and Start Capacitors

6.3.4 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 burn-out. For this reason overload protection is provided in the form of a current and temperature sensitive control which will open the circuit before any damage can occur. Figure 6.4 shows overload.

Figure 6.4 Overload

6.3.5 Thermostat:

A thermostat is a device for regulating the temperature of a system so that the system's temperature is maintained near a desired set point temperature. The thermostat does this by controlling the flow of heat energy into or out of the system. That is, the thermostat switches heating or cooling devices on or off as needed to maintain the correct temperature. Figure 6.5 shows thermostat.

Figure 6.5 Thermostat

6.3.6 Contactors:

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

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

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

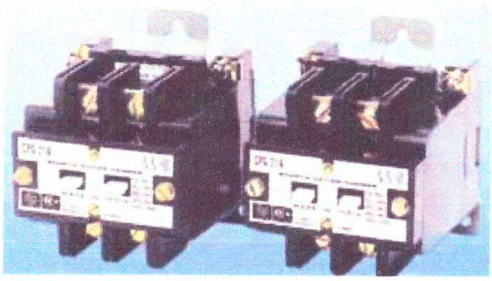

Figure 6.6 Contactor

6.3.7 solenoid valve

Solenoid valves are electrically operated shutoff valves. Probably the most common is the normally closed (NC) valve, but normally open (NO) valves are also available4. With both types, system pressure works to keep the valve closed when that position is desired. Solenoid valves thus can hold against high up stream pressures, but will not restrain much pressure in the reverse direction.

as shown in Figure6.7, the magnetic force developed by the electric coil draws the stem and connected plunger off the valve port when the coil is energized. Some solenoids are designed to allow the stem to start its motion before engaging the plunger which is seated against the system pressure. The momentum of the stem thereby helps open the valve. When the coil is de-energized, the plunger either drops into the closed position by gravity, and/or a light spring assists the closing.[reference8]

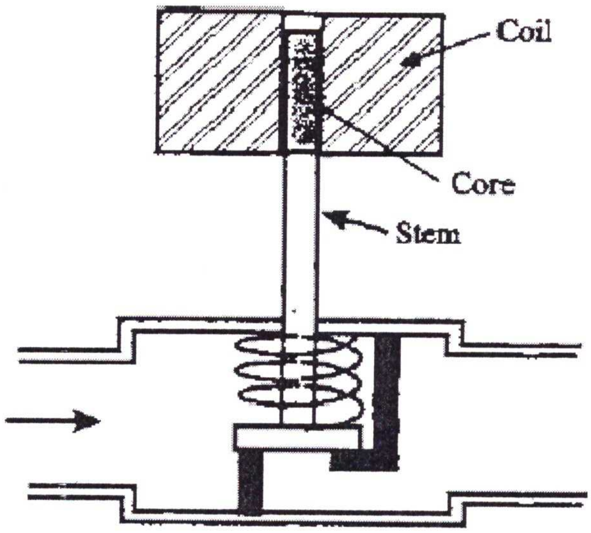

Figure 6.7 solenoid valve

6.3.8 Low-Pressure and High-Pressure Controls

The purpose of low-pressure control is to stop the compressor when the suction pressure drops below a preset value or when the refrigerant flow rate is too low to cool the compressor motor. Figure 6.8a shows a typical low-pressure control mechanism. When the suction pressure falls below a certain limit, the spring pushes the blade downward, opens the motor circuit, and stops the compressor. When the suction pressure increases the bellows expand, thus closing the contact of the motor circuit and restarting the compressor. The two adjusting screws are used to set the cut-out and cut-in pressures. Cut-out pressure is the pressure at which the compressor stops, and cut-in pressure is the pressure at which the compressor starts gain. [reference 1]

The purpose of high-pressure control is to stop the compressor when the discharge pressure of the hot gas approaches a dangerous level. Figure 6.8b shows a typical high-pressure control mechanism. If the discharge pressure reaches a certain limit, the bellows expand so that the blade opens the motor circuit contact and the compressor stops. When the discharge pressure drops to a safe level, the bellows contract and close the contact, and the compressor starts again. As in a low pressure control, two adjusting screws are used to set the cut-out and cut-in pressures.In small refrigeration systems, low-pressure and high-pressure controls are often combined to form a dual-pressure control. [reference 1]

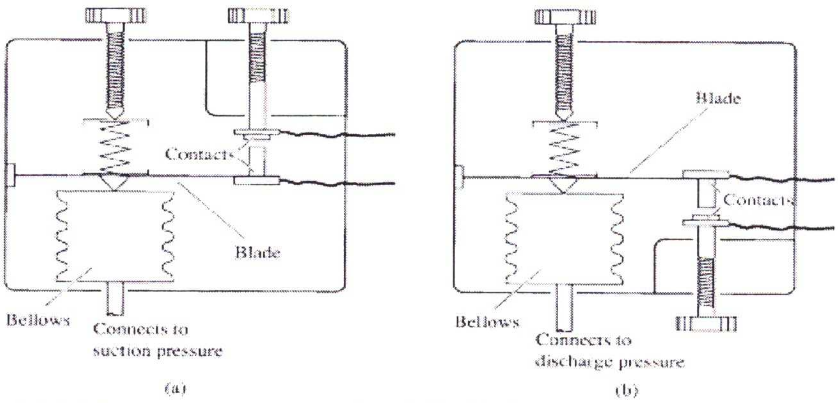

Figure 6.8 (a) Low pressure control and (b) high pressure control

6.3.9 On Delay Timer

on delay timer provides an on delay period between the application of input voltage and energizing a load. The timer is connected to a voltage source and a load. The on delay period begins when voltage is applied. The load remains deenergized until the end of the delay at which time it energizes. It remains energized as long as input voltage is present. When input voltage is removed, the load deenergizes and the timer is reset. Figure 6.9 shows an on delay timer

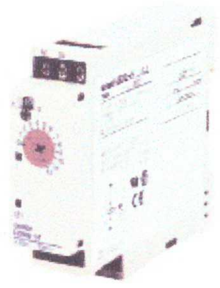

Figure 6.9 on delay timer

6.3.10 Bimetal Defrost Thermostat

It is basically a safety device. It monitors the evaporator temperature . during the defrost cycle, if it sees enough heat and sense the temperature above zero (meaning defrosting is completed) It opens & kills power to the defrost heaters even though the defrost time may not have run out.(controlled by timer) figure 6.10 show bimetal defrost thermostat .

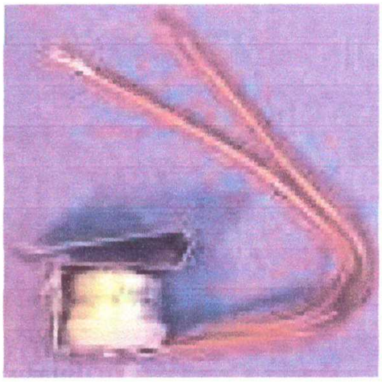

Figure 6.10 bimetal defrost thermostat

6.4 Electrical Description For Two stage cycle:

Two stage 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 temperatures, the high stage cool until it reaches the -8 degrees Celsius, then low stage will assist the high stage to reach -35 degrees Celsius. The electrical control circuit for this mechanical cycle is controlling with contactors. Figure 6.11 show the electrical components in mechanical cycle

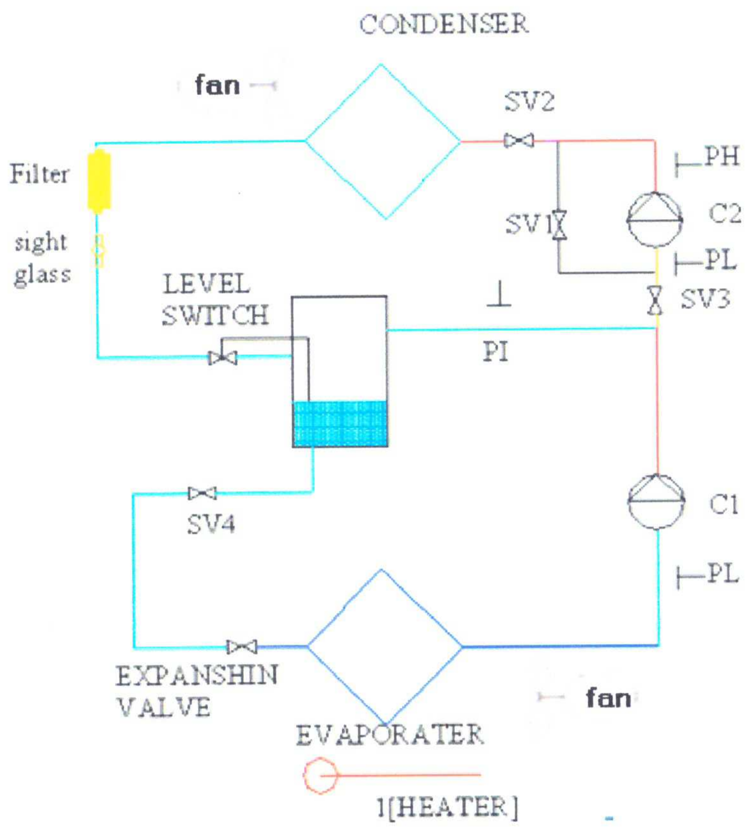

Figure 6.11 Electrical components in mechanical cycle

6.4.1 Control of cycle Using Contactors:

1. Pressing the start button ,thermostat measure the temperature if it high than -35 degrees Celsius give signal to high compressor (C2) in high stage to run with the solenoid valve (SV1),and timer (T1) on delay to count .
2. The timer (T1) run for 3 seconds, after that its closed the solenoid valve (SV1),and open the solenoid valves (SV 2,SV3).
3. The high cycle still run until the intermediate pressure reach the wanted pressure (PI) .
4. After that the intermediate pressure (PI) gives signal for solenoid valve (SV 4) to open with low compressor (C 1) to run.
5. The cycle still worked , after two hours from starting cycle the hibernate clock shut down the cycle, and the heater activated for fifteen minutes to made defrost, after that clock starts the cycle again until reach the specific temperature.
6. If high pressure or low pressure or over load has happen the cycle will be shut down immediately .
7. As monitoring system for the cycle there are a lamps for every compressor lights when its runs and off when its shutdown ,and there are also lamps for pressure switches lights if there pressure failure .

8. For power cycle The evaporator fan run immediately with the low compressor ,and the condenser fan with high compressor.

The above procedure is explained in Figure 6.12

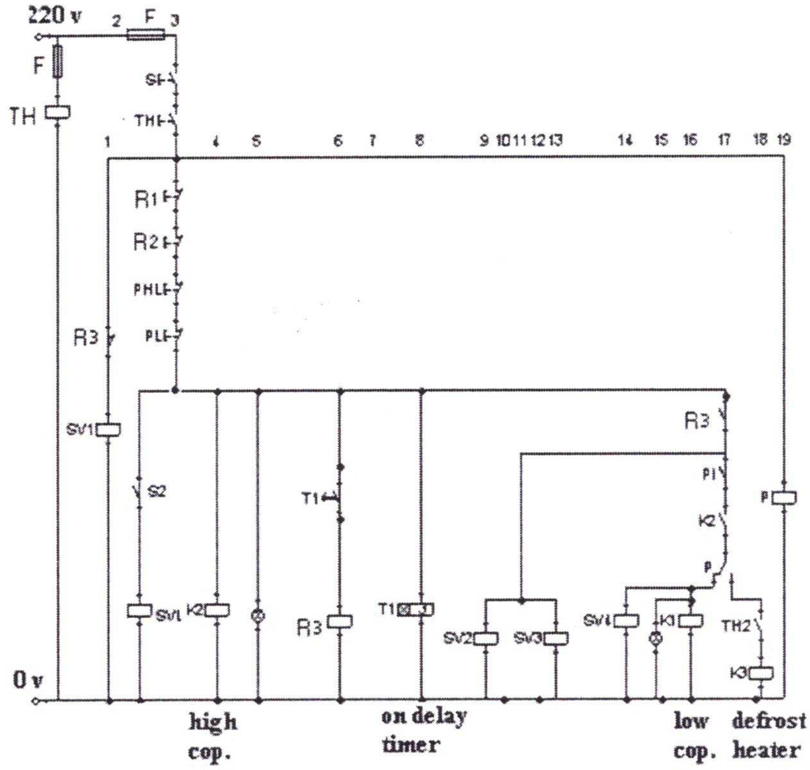

Figure 6.12. controlling with contactors

6.5 Power Circuit:

Power circuit consists of four motors and a defrost heater that work according to the instructions receiver from control circuit, figure 6.13 bellow depicts power circuit

blueprint for the cycle , figure 6.14 shows the electrical construction for compressors.

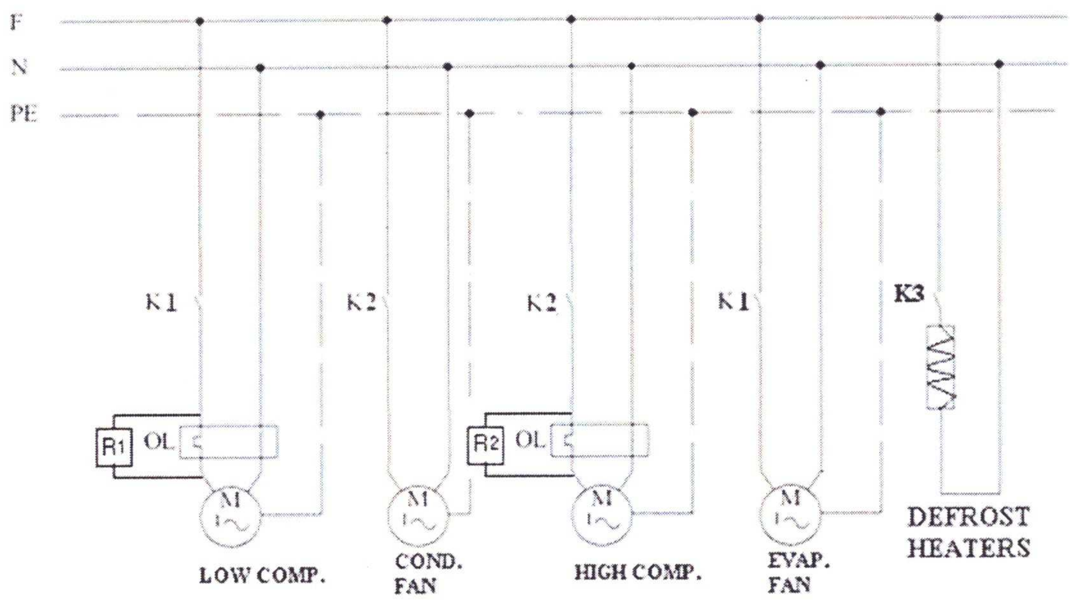

Figure 6.13 power circuit

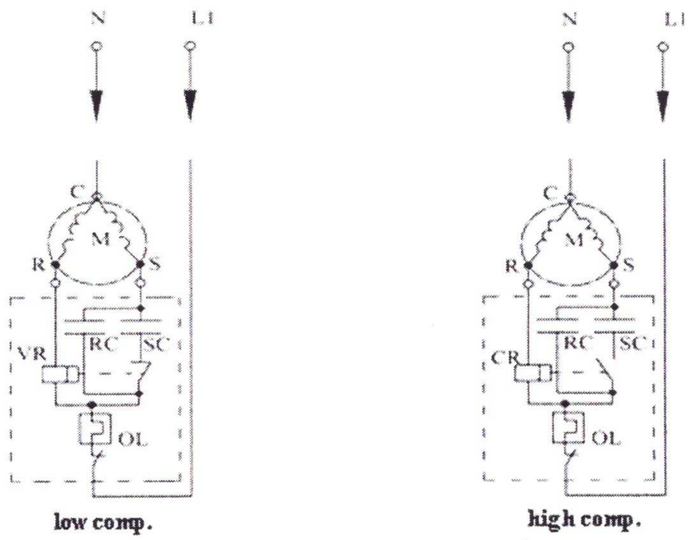

Figure 6.14 Electrical construction for compressors

Table 6.1 Circuit symbols

Table 6.1 Circuit symbols

Symbol	Description
L1	Phase line
N	Neutral line
F	Fuse
F1	Overload
F2	overload
PL1	Low stage pressure switch
PLH2	Low and high pressure switch
T1	On delay timer
TH	thermostat
K1	Start contactor
C1	Compressor 1
C2	Compressor 2
SV	Solenoid valve
PI	Intermediate pressure
R	relay
S	switch
S2	Level switch
p	hibernate clock
RC	Run capacitor
SC	Start capacitor
M	Motor

Chapter seven

7.1 Recommendations

At the end of this project the working team have some recommendations and these recommendations are:

1. The suggested project has useful applications and the procedures will be useful for students.

2. This project can be controlled by PLC ,so its can be worked on by another group of students.

3. The future of most refrigeration and air conditioning applications are depend on the refrigerants ,so it is recommended to observing the refrigerant development.

4. Its needed to connect the graduation projects with the labs to let them provide the good cooperation between them and made a database for websites to use in graduation projects researches in the department.

7.2 Conclusions:

As there were some recommendations, so some Conclusions are obtained at the end of project, and these Conclusions are:

1. This project is friendly to environment, specially the suggested refrigerant (R404a) comparing with usually used refrigerants.

2. The COP of cycle in this project with the suggested refrigerant (R404a) is higher than COP for the some cycle using any one of common refrigerants.

APPENDEX A

TABLE A-1 Thermal Conductivity of Materials

TABLE A-2 Properties of common foods

TABLE A-3 Air change per hour

TABLE A-4 Specific heat of packaging material

TABLE A-5 maximum and minimum temperature for Hebron city

TABLE A-6 Recommended Refrigerant Velocities

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

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

TABLE A-9 Properties of air at atmospheric pressure

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

TABLE A-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 glass fiber	0.039	
	Board or slab		
	Cellular glass	0.058	
	Corkboard	0.043	
	Glass fiber	0.036	
	Expanded polystyrene (smooth)	0.029	
	Expanded polystyrene (cut cell)	0.036	
	Expanded polyurethane	0.025	
Loose fill			
	Milled paper or wood pulp	0.039	
	Sawdust or shavings	0.065	
.	Mineral wool (rock, glass, slag)	0.039	
	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	Stainless steel	15.6	
	Aluminum	202	
	Cooper	386	

TABLE A-2 Properties of common foods and substances

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	R12	R22	8-12
	R404A	R410A	
Discharge	R12	R22	10-18
	R404A	R410A	

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

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

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

	$\frac{S_n}{d}$							
	1.25		1.5		2.0		3.0	
$\frac{S_p}{d}$	C	n	C	n	C	n	C	n
In line								
1.25	0.386	0.592	0.305	0.608	0.111	0.704	0.0703	0.752
1.5	0.407	0.586	0.278	0.620	0.112	0.702	0.0753	0.744
2.0	0.464	0.570	0.332	0.602	0.254	0.632	0.220	0.648
3.0	0.322	0.601	0.396	0.584	0.415	0.581	0.317	0.608
Staggered								
0.6	—	—	—	—	—	—	0.236	0.636
0.9	—	—	—	—	0.495	0.571	0.445	0.581
1.0	—	—	0.552	0.558	—	—	—	—
1.125	—	—	—	—	0.531	0.565	0.575	0.560
1.25	0.575	0.556	0.561	0.554	0.576	0.556	0.579	0.562
1.5	0.501	0.568	0.511	0.562	0.502	0.568	0.542	0.568
2.0	0.448	0.572	0.462	0.568	0.535	0.556	0.498	0.570
3.0	0.344	0.592	0.395	0.580	0.488	0.562	0.467	0.574

TABLE A-9 Properties of air at atmospheric pressure

T (K)	ρ (kg/m³)	c_p (J/kg·K)	μ (kg/m·s)	ν (m²/s)	k (W/m·K)	α (m²/s)	Pr
Air							
100	3.6010	1.0266×10^3	0.6924×10^{-5}	1.923×10^{-6}	0.009 246	0.0250×10^{-4}	0.768
150	2.3675	1.0099	1.0283	4.343	0.013 735	0.0574	0.756
200	1.7684	1.0061	1.3289	7.490	0.018 09	0.1016	0.739
250	1.4128	1.0053	1.5990	11.310	0.022 27	0.1568	0.722
300	1.1774	1.0057	1.8462	15.690	0.026 24	0.2216	0.708
350	0.9980	1.0090	2.075	20.76	0.030 03	0.2983	0.697
400	0.8826	1.0140	2.286	25.90	0.033 65	0.3760	0.689
450	0.7833	1.0207	2.484	31.71	0.037 07	0.4636	0.683
500	0.7048	1.0295	2.671	37.90	0.040 38	0.5564	0.680
550	0.6423	1.0092	2.848	44.27	0.043 60	0.6532	0.680
600	0.5879	1.0551	3.018	51.34	0.046 59	0.7512	0.682
650	0.5430	1.0635	3.177	58.51	0.049 53	0.8578	0.682
700	0.5030	1.0752	3.332	66.25	0.052 30	0.9672	0.684
750	0.4709	1.0856	3.481	73.91	0.055 09	1.0774	0.686
800	0.4405	1.0978	3.625	82.29	0.057 79	1.1951	0.689
850	0.4149	1.1095	3.765	90.75	0.060 28	1.3097	0.692
900	0.3925	1.1212	3.899	99.3	0.062 79	1.4271	0.696
950	0.3716	1.1321	4.023	108.2	0.065 25	1.5510	0.699
1000	0.3524	1.1417	4.152	117.8	0.067 52	1.6779	0.702
1100	0.3204	1.160	4.44	138.6	0.073 2	1.969	0.704
1200	0.2947	1.179	4.69	159.1	0.078 2	2.251	0.707
1300	0.2707	1.197	4.93	182.1	0.083 7	2.583	0.705
1400	0.2515	1.214	5.17	205.5	0.089 1	2.920	0.705
1500	0.2355	1.230	5.40	229.1	0.094 6	3.266	0.705
1600	0.2211	1.248	5.63	254.5	0.100	3.624	0.705
1700	0.2082	1.267	5.85	280.9	0.105	3.977	0.705
1800	0.1970	1.287	6.07	308.1	0.111	4.379	0.704
1900	0.1858	1.309	6.29	338.5	0.117	4.811	0.704
2000	0.1762	1.338	6.50	369.0	0.124	5.260	0.702
2100	0.1682	1.372	6.72	399.6	0.131	5.680	0.703
2200	0.1602	1.419	6.93	432.6	0.139	6.115	0.707
2300	0.1538	1.482	7.14	464.0	0.149	6.537	0.710
2400	0.1458	1.574	7.35	504.0	0.161	7.016	0.718
2500	0.1394	1.688	7.57	543.0	0.175	7.437	0.730

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

N	1	2	3	4	5	6	7	8	9	10
Ratio for staggered tubes	0.68	0.75	0.83	0.89	0.92	0.95	0.97	0.98	0.99	1.0
Ratio for in-line tubes	0.64	0.8	0.87	0.9	0.92	0.94	0.96	0.98	0.99	1.0

References

1- Shan K. Wang, Hand book of Air Conditioning And Refrigeration, 2^{nd} edition

2- Welbert F.Stocker, industrial Refrigeration hand Book

3- Ibrahim Dincer, Refrigeration Systems and Applications, Department of Mechanical Engineering, KFUPM, Saudi Arabia, 2003.

4- Mohammed A. Alsaad And Mahmoud A. Hammad, Heating and Air Conditioning, Third edition, University of Jordan, Amman, Jordan, 1995.

5- Roy J. Dossat, Principles of refrigeration, second edition, SI version, University of Houston, Houston, Texas, USA.

6- Chumak I.G, Nikulshina D.G, Refrigeration system first edition, Vee shya shcola cor, Moscw, 1988.

7- Burmeister, L.C., Elements of Thermal-Fluid System Design, Prentice Hall, 1998.

8- J.P. Holman, Heat Transfer, Ninth Edition, South Methodist University, New York, McGRAW-HILL BOOK COMPANY, USA.

9-Joseph E. Shigley, Charles R. Mischke, Richard G. Budyans, Mechanical Engineering Design, Seventh Edition, University of Michign, and Lowa State University, McGRAW-HILL BOOK COMPANY, USA.

10- Chumak I.G.,Nicoolshian D.G., design of refrigeration inistulation

11-Basel Nashat Asmar, "Control of a two-stage refrigeration system, University of Nottingham", April 1999

12- E. Torrella, R. Llopis and R. Cabello, "Experimental evaluation of the inter-stage conditions of a two-stage refrigeration cycle using a compound compressor", 2008.

13- Raed K. Mohammed, Shadi T. Mesleh And Tareq I. Khlawi , "Design and building cascade refrigeration system for plant tissues storage" , Mechanical engineering department, PPU, Palestine 2008.

14-http://www.onlinefreeebooks.net/engineering-ebooks/manufacture-mechanical-industrial-engineering/refrigeration-and-air-conditioning-pdf.htm

15- http;//www.kelco.com.au

16-blood plasma :http://www.maexamhelp.com/id97.htm

17-Cool pack software.

18-Smart draw software .

19-Autocad software