

بِسْمِ اللَّهِ الرَّحْمَنِ الرَّحِيمِ



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**Design and Building of ultra-low medical Refrigerator Using
Cascade Refrigeration Cycle Working on Solar Energy**

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Abstract

Design and building of a medical ultra-low cascade refrigeration chamber is performed, the system consists of two refrigeration cycles connected with each other through a heat exchanger, using two different gases to reach ultra-low temperatures to store human (DNA) utilizing solar energy.

تصميم و بناء ثلاجة طبية (cascade) النظام عبارة عن دورتي تبريد متصلة مع بعضها البعض من خلال مبادل حراري باستخدام غازين مختلفين لالتخزين (DNA) الإنسان و تعمل على الطاقة الشمسية.

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Chapter One

Introduction

1.1 Overview

Palestinian medical sector suffers from lack of medical refrigerators to store human "DNA" laboratory specimens which need ultra-low temperatures which can be reached by utilizing cascade refrigeration system.

Cascade refrigeration system is one of the most suitable cycles for applications that need very low storage temperatures where human "DNA" is one of these applications.

Cascade refrigeration cycle consists of two simple refrigeration cycles, the upper cycle has a temperature range between (33 to -30) °C & the lower cycle has a range between (-25 to -70) °C and both cycles are connected through a heat exchanger, where each cycle works on different refrigerant.

As a result of increase demand on renewable energy to reduce environmental pollution and to decrease financial costs, it is decided to design cascade system utilizing solar energy.

1.2 Problem Definition

Human "DNA" needs a very low temperature and due to the lack of medical refrigerators that can be used to store human "DNA", there is a great need for designing and building such a system.

1.3 Motivation

Providing a medical refrigerator for Palestinian hospitals will enhance and raise the level of medical sector.

1.4 DNA Storage Condition

Deoxyribonucleic acid or DNA is a molecule that contains the instructions an organism needs to develop, live and reproduce. These instructions are found inside every cell, and are passed down from parents to their children.

DNA is stored in plastic flasks which placed inside a carton boxes which have a dimensions of (10*10*5) cm. The refrigerator has three chambers; each one has 30 boxes arranged as three rows and five columns, where DNA is dissolved in distributed water in the plastic flasks.



Figure 1.1 plastic flask inside carton box

The weight of an empty flask is 0.8314 [g] as shown in the figure.

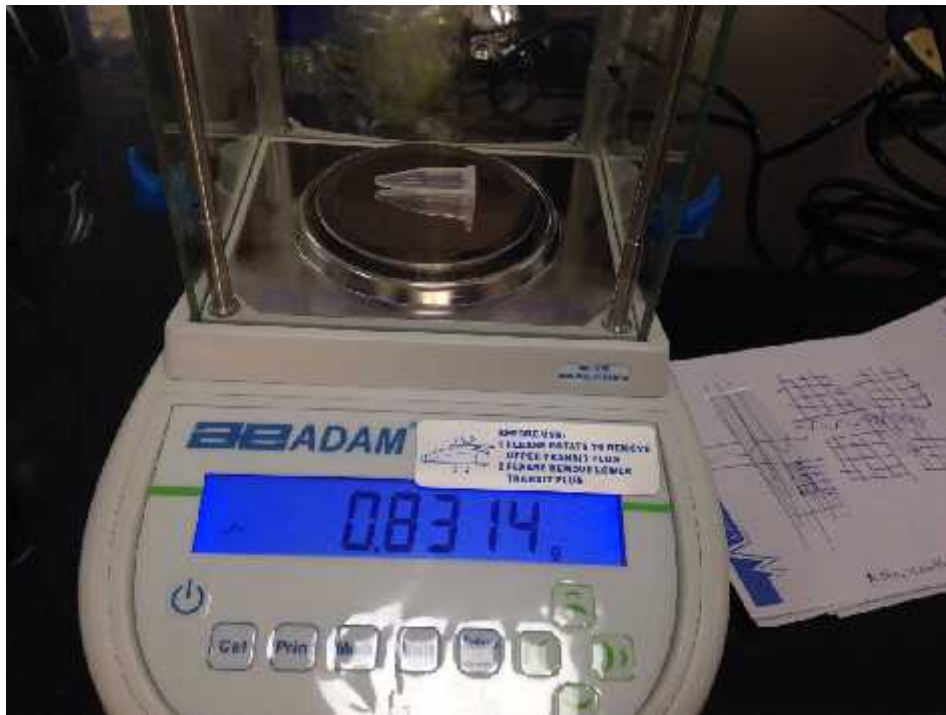


Figure 1.2 empty flask weight

The weight of flask full of DNA is 1.176 [g] as shown in the figure.

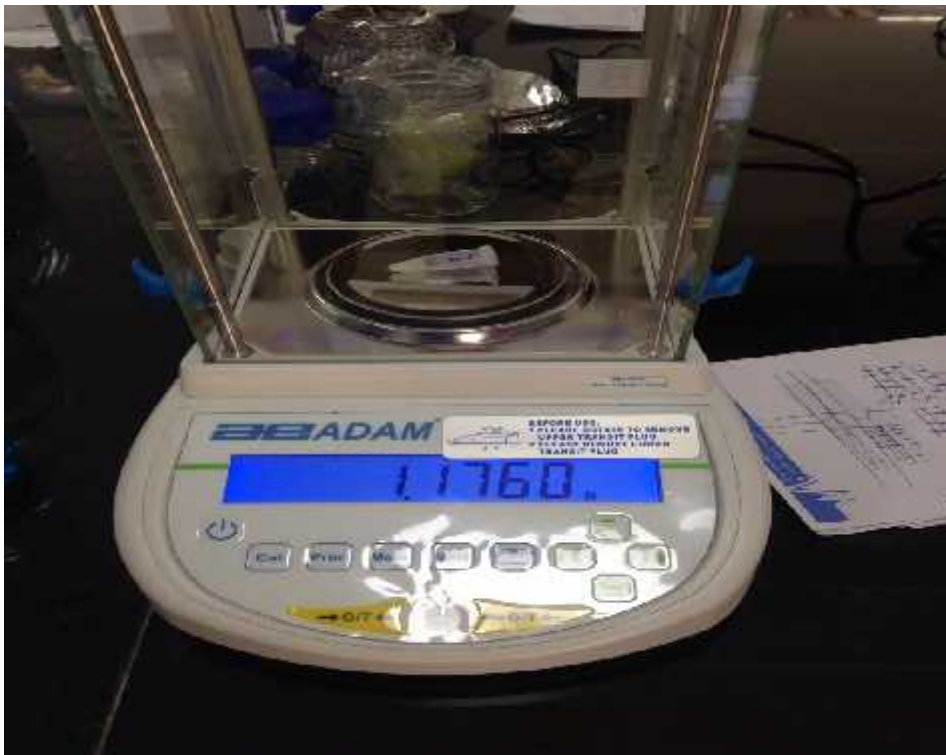


Figure 1.3 flask full of DNA weight

The weight of empty carton box is 88.6 [g] as shown in the figure.



Figure 1.4 empty carton box weight

1.5 Product Mass

Totally mass product consist of the mass of DNA, carton box and plastic

Mass of cartonbox:

Totally number of carton box = number of chamber* number of boxes in each chamber

$$= 3*30 \text{ carton box}$$

$$=90 \text{ carton box}$$

Carton box Weight=88.6[g] from figure 1.5

The total mass of carton box = number of flasks * DNA Weight

$$= 90*88.6$$

$$= 7974.4 \text{ [g]}$$

$$=7.9744 \text{ [kg]}$$

Mass of plastic flask

Totally number of plastic flasks = number of chamber * number of boxes in each chamber * number of flasks in each box

$$=9000 \text{ plastic flask}$$

Totally Plastic flask Weight =number of flasks in each box *Weight of Plastic flask

$$= 9000*0.8314$$

$$= 7482.6 \text{ [g]} = 7.4826 \text{ [kg]}$$

Mass of DNA

DNA Weight =Weight of flask full of DNA- Weight of empty flask

$$= 1.176- 0.831 = 0.3446 \text{ [g]}$$

The total mass of DNA = number of flasks * mass DNAWeight

$$= 9000*0.3446$$

$$= 3101.4 \text{ [g]} =3.1014 \text{ [kg]}$$

The total mass of the product = DNA mass + plastic flask mass + carton box mass

$$=3.1014+7.4826+7.974=18.5584 \text{ [kg]}$$

1.6 Budget of the Project

Component Type	Unit Price[]	Quantity	Total[]
Compressor	500	2	1000
Condenser	300	1	300
Evaporator	400	1	400
Capillary tube	25	2	50
Thermostat	30	1	30
Digital Thermostat	250	1	250
Sight glass	50	2	100
Heater	85	3 [m]	85
Filter drier	30	2	60
Contactora	50	2	100
overload	50	2	100
Relay	30	1	30
Gas R404a	500	6 [kg]	500
Gas R23	2200	13.5 [kg]	2200
Fan motor	60	1	60
Body	500	1	500
Totally price	5765

1.7 Time Plan

Table 1.1 Time table for the first semester time [week]

Activity	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Selection of the project															
Search about information															
Search for helping projects															
Search for helping research															
Cooling Load Calculation															
Cycle analyses															
Components selection															
Electrical Design															

Table 1.2 Time table for the second semester time [week]

Activity	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Search for the Component															
Build the Body															
Build the System															
Testing															

Chapter Two

Cooling Load

2.1 Introduction

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

2.2 Load Sources

Cooling load has different load sources, the most common of it in refrigerator:

- 1- The wall heat gain
- 2- The product heat gain
- 3- Infiltration heat gain
- 4- Packaging heat gain
- 5- Defrost heaters heat gain
- 6- Fan motor heat gain

2.2.1 Refrigerator Design Data

Design data in July, 2018

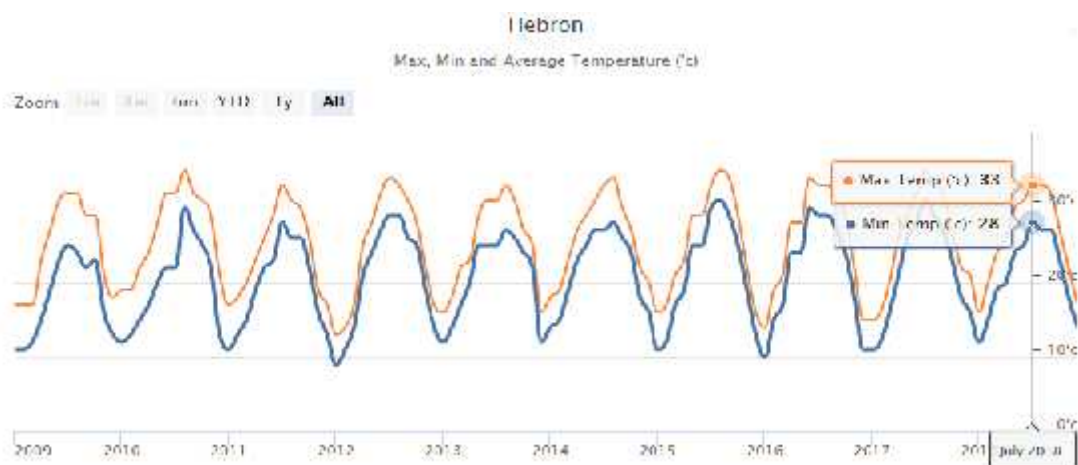


Figure 2.1 Max, Min Temperature

$T_{\text{surr}} = 33 \text{ [}^\circ\text{C]}$ [Reference 2]

DNA Storage temperature is (-70 to -80)[$^\circ\text{C}$]. [Reference 4]

Chamber dimension is (length*width*height) = 40*30*90 [cm]

Mass of the product including (Carton, Plastic and DNA) = 18.5584 [kg]

Cooling time is 12[hours].

2.2.2 The Wall Heat Gain

The wall heat gain load, is a measure of the heat flowrate by conduction through the walls of the refrigerated space from the outside to the inside and it is calculated as

$$Q_{\text{wall}} = U * A * T \dots \dots \dots [2.1] \text{ [Reference 19]}$$

Where:

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

U: Overall heat transfer coefficient [$\text{W}/\text{m}^2 \cdot ^\circ\text{C}$]

T: The temperature difference

$$T = T_{\text{out}} - T_{\text{in}} = (33) - (-70) = 103 \text{ [}^\circ\text{C]}$$

$$\text{Area for wall sides} = (0.9 * 0.3) = 0.27 \text{ [m}^2\text{]}$$

$$\text{Area for front/behind wall} = (0.9 * 0.4) = 0.36 \text{ [m}^2\text{]}$$

$$\text{Area for floor/ceiling} = (0.4 * 0.3) = 0.12 \text{ [m}^2\text{]}$$

The composition of the refrigerator walls

Table 2.1 Characteristics of walls components

Material	Thickness " x" (m)	Thermal conductivity "k"(W/m. $^\circ\text{C}$)
Galvanized steel	0.001	15.6
Polystyrene	0.15	0.03
Stainless steal	0.001	15

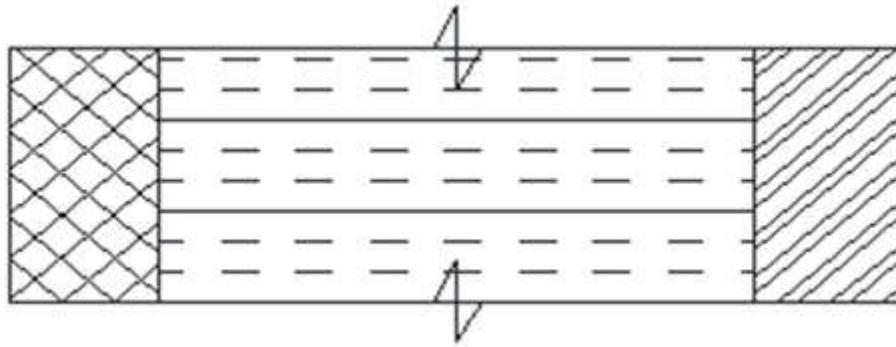


Figure 2.2 Chamber wall layers

Overall heat transfer coefficient

$$U = \frac{1}{\frac{1}{h_i} + \frac{\Delta x_1}{k_1} + \frac{\Delta x_2}{k_2} + \frac{\Delta x_3}{k_3} + \frac{1}{h_o}} \dots\dots\dots [2.2]$$

Where:

$$h_i = 3.2 \text{ [W/m}^2 \cdot \text{°C]} \text{ [reference 13]}$$

$$h_o = 14.3 \text{ [W/m}^2 \cdot \text{°C]} \text{ [References 13]}$$

$$U = 1 / ((1/3.2) + (0.001/15.6) + (0.15/0.03) + (0.001/15) + (1/14.2))$$

$$U = 1/5.37 \text{ [W/m}^2 \cdot \text{°C]}$$

$$U = 0.186 \text{ [W/m}^2 \cdot \text{°C]}$$

$$Q_{\text{floor\&ceiling}} = 0.186 * 0.12 * (33 - -70) * 2 = 4.6 \text{ [W]}$$

$$Q_{\text{front \& behind}} = 0.186 * 0.36 * (33 - -70) * 2 = 13.8 \text{ [W]}$$

$$Q_{\text{two sides}} = 0.186 * 0.27 * (33 - -70) * 2 = 10.3 \text{ [W]}$$

$$Q_{\text{all walls}} = (Q_{\text{floor\&ceiling}} + Q_{\text{two sides}} + Q_{\text{front\& behind}})$$

$$Q_{\text{all walls}} = (4.6 + 13.8 + 10.3) = 28.7 \text{ [W]}$$

2.2.3 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

$$Q_{\text{ch}} = m * C_p * T \dots\dots\dots [2.3]$$

Where:

Q_{ch} : Cooling product load [kJ]

m : Mass of the product equal 3.1 [kg]

C_p : Specific heat for water above freezing and it is equal 4.18 [kJ/kg. °C]

T is $(T_o - T_{ch}) = 33 - 0 = 33$ [°C]

$$Q_{ch} = 3.1 * 4.18 * 33 = 427.6 \text{ [kJ]}$$

2- Cooling load below freezing

$$Q_c = m * C_p * T \dots\dots\dots [2.4]$$

Where:

Q_c : cooling product load [kJ]

m : Mass of the product equal 3.1 [kg]

C_p : Specific heat for water below freezing and it is equal 2.1 [kJ/kg. °C]

T_{ch} = chilling temperature for water and equal 0.0 [°C]

$T = (T_{ch} - T_o) = (0.0 - -70) = 70$ [°C]

$$Q_c = 3.1 * 2.1 * 70 = 455.7 \text{ [kJ]}$$

3- Freezing Load

$$Q_f = m * HL \dots\dots\dots [2.5]$$

Where:

Q_f : freezing load [kJ]

HL : latent heat for water at 0.0°C and equal 334 [kJ/kg]

$$Q_f = 3.1 * 334 \text{ [kJ]}$$

$$Q_f = 1035.86 \text{ [kJ]}$$

Total product load

$$Q_p = (Q_{ch} + Q_c + Q_f) / C.T \dots\dots\dots [2.6]$$

Where:

$C.T$: Desired cooling time 12 [hours]

$$Q_p = 1000 * (427.6 - 455.7 + 1035.86) / (12 * 60 * 60)$$

$$Q_p = 23.3 \text{ [W]}$$

2.2.4 Infiltration Heat Gain

$$Q_{inf} = m * C_p * (T_o - T_i) \dots \dots \dots [2.7]$$

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

Where:

ρ : Air density equal 1.25 [kg/m³] [Reference 5]

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

V_F : The volumetric flow rate of infiltrated air [m³/s]

T_o : The outside temperature equal 33[°C]

T_i : The inside temperature equal -70 [°C]

Number of air change = 0.5 [times/h]

Volume of room = 0.9 * 0.4 * 0.3 = 0.108[m³]

V_F = number of air change * volume of room

V_F = 0.5 * 0.108 = 0.054 [m³/h]

$Q_{inf} = 1.25 * 0.054 * 1000 * (33 - -70)/3600$

$Q_{inf} = 1.931$ [W]

2.2.5 Packaging Heat gain

$$Q_{pk} = \frac{m * C_p * T}{t} \dots \dots \dots [2.8]$$

Where:

Q_{pk} : packaging heat load [W]

m_{pk} : Mass of product [kg]

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

T_o : outside temperature [°C]

T_{in} : temperature of the refrigerated space [°C]

t:desired cooling time [second]

For carton:

$$C_{pk} = 1.7 \text{ [kJ / kg. } ^\circ\text{C]} \text{ [Reference 6]}$$

$$Q_{pk} = 7.974 * 1.7 * (33 - -70) * 1000 / (12 * 60 * 60) \\ = 32.3 \text{ [W]}$$

For plastic

$$C_{pk} = 1.67 \text{ [kJ/kg. } ^\circ\text{C]} \text{ [Reference 6]}$$

$$Q_{pk} = 7.4826 * 1.67 * (33 - -70) * 1000 / (12 * 60 * 60) \\ = 29.79 \text{ [W]}$$

$$Q_{pktotal} = Q_{pk \text{ carton}} + Q_{pkplastic}$$

$$Q_{pktotal} = 32.3 + 29.79 = 62.11 \text{ [W]}$$

2.2.6 Defrosts Heater Heat Gain

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

2.2.6.1 Defrosts Heater Heat Gain of the Evaporator

$$Q_{h1} = P * \varphi \dots \dots \dots [2.9]$$

Where:

P: Power of heater taken 500 [W]

φ : heater usage factor (0.1 – 0.5), taken 0.2 [Reference 5]

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

2.2.6.2 Defrosts Heater Heat Gain around the Door

$$Q_{h2} = P * \varphi \dots \dots \dots [2.10]$$

Where:

P: Power of heater taken 80 [W]

φ : heater usage factor (0.1 – 0.5), taken 0.2 [Reference 5]

$$Q_{h2} = P * \varphi = 80 * 0.2 = 16 \text{ [W]}$$

2.2.7 Fan Motor Heat Gain

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

$$Q_{\text{motor}} = \text{power of motor} = 25 \text{ [W]}$$

2.3 Total Cooling Load

The total cooling load is

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

$$Q_T = 28.7 + 23.32 + 1.93 + 62.11 + 116 + 25$$

$$Q_T = 257.6 \text{ [W]}$$

Add 40 % as safety factor

$$\text{Total cooling load} = Q_T * 1.4$$

$$\text{Total cooling load} = 257.6 * 1.4 = 359.88 \text{ [W]}$$

Chapter Three

Cycle Analysis

3.1 Selection of the Refrigerant

In the selection of an appropriate refrigerant for use in a refrigeration system, there are many criteria to be considered:

- High latent heat of vaporization.
- Non corrosive to metal.
- Nonflammable and non-explosive.
- Nontoxic.
- Low cost.
- Mixes well with oil.
- Easy to liquefy at moderate pressure and temperature.
- Easy of locating leaks by suitable indicator.

To select refrigerants successfully it is must be consider the above properties. Gas R-404a selected for upper cycle and gas R-23 selected for lower cycle.

3.2 Cycle Analysis

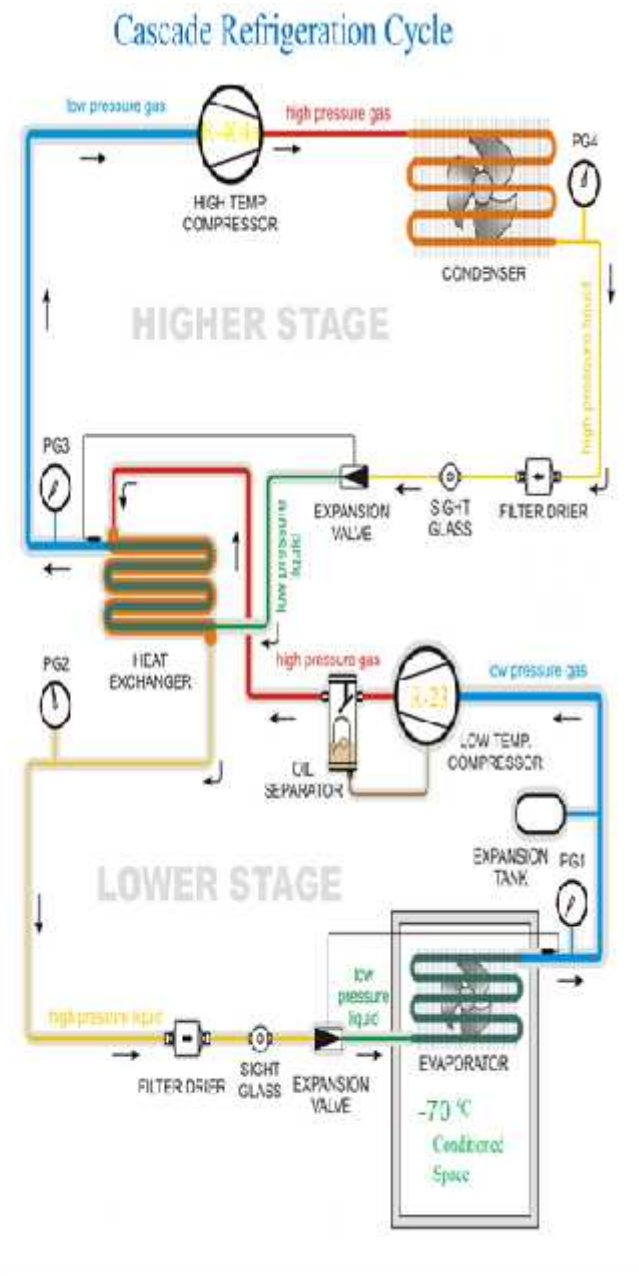


Figure 3.1 Cascade Refrigeration Cycle

3.3 Ideal Cycle Design

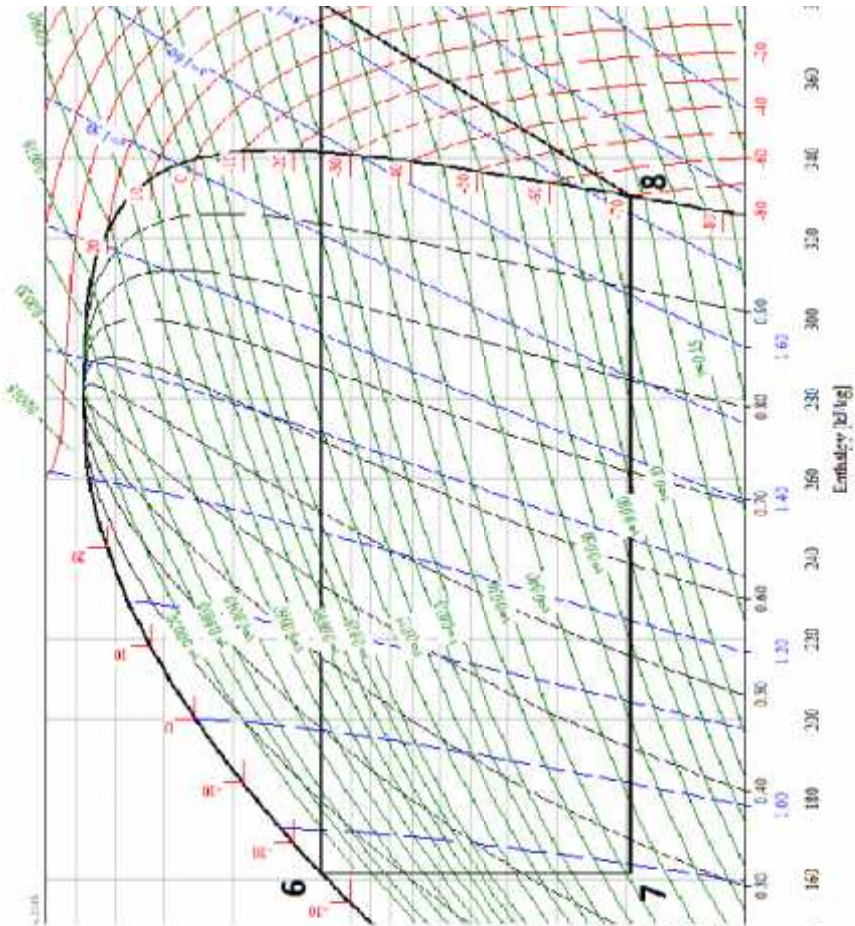


Figure 3.2 P-h Diagram for R-23

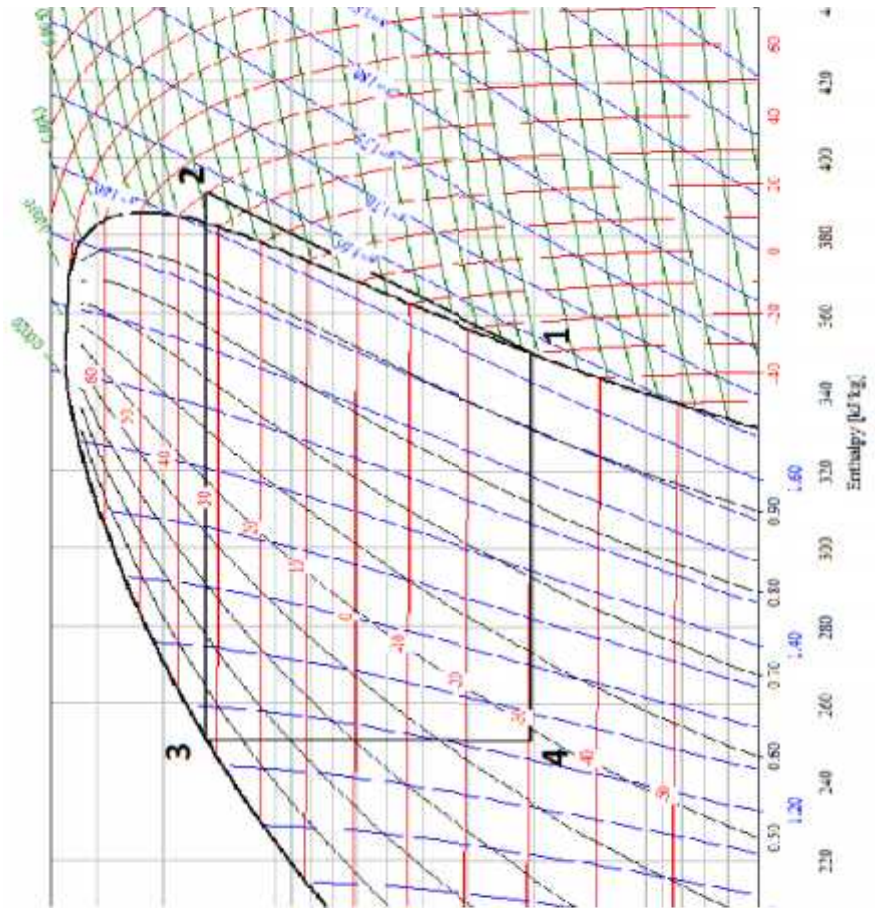


Figure 3.3 P-h Diagram for R-404a

Properties of lower cycle by using cool pack software program. [reference 8].

Table 3.1 Properties of Lower Cycle

points	T [°C]	P [bar]	v [m ³ /kg]	h [kJ/kg]	s [kJ/kg.°k]
5	14.595	11.955	0.025505	378.914	1.7225
6	-25	11.955	N/A	161.394	0.857
7	-70	1.941	0.03152	161.394	0.89
8	-70	1.941	0.11635	330.619	1.7225

Properties of upper cycle by using cool pack software program. [reference 8].

Table 3.2 Properties of Upper Cycle

points	T [°C]	P [bar]	v [m ³ /kg]	h [kJ/kg]	s [kJ/kg.°k]
1	-29.938	2.045	0.095129	350.304	1.6302
2	40.118	15.281	0.013070	391.014	1.6302
3	32.669	15.281	N/A	250.721	1.17
4	-29.938	2.045	0.04573	250721	1.22

3.3.1 Calculations for Lower Stage Using R-23

$$Q_e = \dot{m}_1 * q_e \dots\dots\dots (3.1)$$

$$Q_e = \dot{m}_1 * (h_7 - h_8) \dots\dots\dots (3.2)$$

$$\dot{m}_1 = \frac{Q_e}{h_7 - h_8}$$

Where:

\dot{m}_1 : Mass flow rate for lower cycle [kg/s]

Q_e : Heat transfer rate in evaporator (evaporator load) [W]

q_e : Refrigeration effect [kJ/kg]

h_8 : Enthalpy at point inlet compressor [kJ/kg]

h_7 : Enthalpy at point inlet evaporator [kJ/kg]

$$1 = \frac{0.35988}{330.6 - 161.4}$$

$$1 = 0.002126 \text{ [kg/s]}$$

$W_{in(1)}$: Lower compressor work [kJ/kg]

$$W_{in(1)} = (h_5 - h_8) \dots \dots \dots (3.3)$$

$$W_{in(1)} = (379 - 330.6)$$

$$W_{in(1)} = 48.4 \text{ [kJ/kg]}$$

P_1 : Lower compressor power [W]

$$P_1 = W_{in(1)} * 1 \dots \dots \dots (3.4)$$

$$P_1 = 48.4 * 0.002126 = 0.1028 \text{ [kW]} = 102.8 \text{ [W]}$$

Using energy balance in heat exchanger to calculate 2

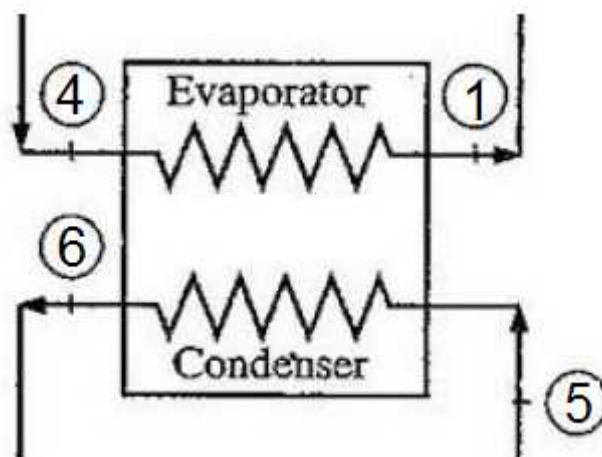


Figure 3.4 Heat Exchanger Balance

$$\dot{m}_1(h_5 - h_6) = \dot{m}_2(h_1 - h_4) \dots \dots \dots (3.6)$$

$$\dot{m}_2 = \frac{\dot{Q}_{c2}}{q_e}$$

Where:

\dot{m}_2 : Mass flow rate for upper cycle [kg/s]

h_5 : Enthalpy at point outlet compressor [kJ/kg]

h_6 : Enthalpy at point outlet condenser [kJ/kg]

h_1 : Enthalpy at point inlet compressor [kJ/kg]

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

$$\dot{m}_2 = \frac{217.52 + 0.002126}{99.592}$$

$$\dot{m}_2 = 0.004645 \text{ [kg/s]}$$

3.3.2 Calculations for Upper Stage Using R-404a

$$Q_C = \dot{m}_2(h_2 - h_3) \dots \dots \dots (3.7)$$

Where:

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

h_2 : Enthalpy at point outlet compressor [kJ/kg]

h_3 : Enthalpy at point outlet condenser [kJ/kg]

$$Q_C = 0.004645 * (391.014 - 250.721)$$

$$Q_C = 0.6516 \text{ [kW]} = 651.6 \text{ [W]}$$

$W_{in(2)}$: Upper compressor work [kJ/kg]

$$W_{in(2)} = (h_2 - h_1) \dots \dots \dots (3.8)$$

$$W_{in(2)} = (391.014 - 350.304)$$

$$W_{in(2)} = 40.71 \text{ [kJ/kg]}$$

P_2 : Upper compressor power [W]

$$P_2 = W_{in(2)} * \eta_2 \dots \dots \dots (3.9)$$

$$P_2 = 40.71 * 0.004645 = 0.189 \text{ [kW]} = 189 \text{ [W]}$$

3.3.3 Calculations of Coefficient of Performance (COP)

$$Q_e = 359.88 \text{ [W]}$$

$$P_{total} = P_1 + P_2 = 102.8 + 189$$

$$P_{total} = 291.8 \text{ [W]}$$

$$COP = \frac{Q_e}{P_{total}}$$

$$COP = 359.88 / 291.8$$

$$COP = 1.233$$

3.4 Compressor Design

A compressor is a positive displacement machine, in which the density of the gas entering the compression chamber has a major influence on the mass flow rate. The gas density at the suction chamber is influenced by the evaporating pressure as well as the suction temperature, which is increased by heat transfer in the suction system. Besides suction superheating, other well-known effects reduce the overall compressor mass flow rate, such as the clearance residual mass, in-cylinder superheating and leakage through the gap between cylinder and piston.

3.4.1 Compressor Calculations and Selections

3.4.1.1 Volumetric efficiency

$$\eta_v = \eta_c * \eta_h$$

η_v : volumetric efficiency.

η_c : volumetric efficiency due to clearance volume in compressor.

η_v : volumetric efficiency due to heating occurs in compressor.

$$\eta_v = 1 - c [(p_h/p_l)^{1/n} - 1]$$

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.

n : exponential coefficient of expansion for refrigerant , $n = 1$

P_H : High pressure of the cycle.

P_L : Low pressure of the cycle.

$$\eta_v = T_{\text{evap.}} / T_{\text{cond.}}$$

Where:

$T_{\text{evap.}}$: Evaporator temperature [$^{\circ}\text{K}$]

$T_{\text{cond.}}$: Condenser temperature [$^{\circ}\text{K}$]

For low stage compressor:

$$\begin{aligned} \eta_v &= 1 - c [(p_h/p_l)^{1/n} - 1] \\ &= 1 - 0.02[(11.955/1.941)^1 - 1] \\ &= 89.6 \% \end{aligned}$$

$$\begin{aligned} \eta_v &= T_{\text{evap.}} / T_{\text{cond.}} \\ &= 203/248 \\ &= 81.85 \% \end{aligned}$$

$$\begin{aligned} \eta_v &= \eta_c * \eta_h \\ &= 0.896 * 0.8185 \\ &= 73.3 \% \end{aligned}$$

For high stage compressor:

$$\begin{aligned}c &= 1 - c [(p_h/p_l)^{1/n} - 1] \\ &= 1 - 0.02 [(15.281/2.045)^1 - 1] \\ &= 87.05 \%\end{aligned}$$

$$\begin{aligned}h &= T_{\text{evap.}} / T_{\text{cond.}} \\ &= 243/306 \\ &= 79.41 \%\end{aligned}$$

$$\begin{aligned}v &= c * h \\ &= 0.8705 * 0.7941 \\ &= 69.12 \%\end{aligned}$$

3.4.1.2 Volume flow rate

The theoretical volume flow rate (V) of the compressor

$$V_{\text{theo}} = \dot{m} * v$$

Where:

V_{theo} : theoretical volume flow rate of the compressor [m^3/s]

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

v : specific volume at the inlet of compressor [m^3/kg]

To determine the actual volume flow rate

$$V_{\text{act}} = V_{\text{theo}} * v$$

Where:

V_{act} : actual volumetric flow rate [m^3/s]

For low stage compressor:

$$\begin{aligned}V_{\text{theo}} &= v_1^* \\ &= 0.002126 * 0.116353 \\ &= 2.47 * 10^{-4} \text{ [m}^3\text{/s]}\end{aligned}$$

$$\begin{aligned}V_{\text{act}} &= V_{\text{theo}} * v \\ &= 2.47 * 10^{-4} * 0.733 \\ &= 1.81 * 10^{-4} \text{ [m}^3\text{/s]}\end{aligned}$$

For high stage compressor:

$$\begin{aligned}V_{\text{theo}} &= v_2^* \\ &= 0.004645 * 0.095129 \\ &= 4.4 * 10^{-4} \text{ [m}^3\text{/s]}\end{aligned}$$

$$\begin{aligned}V_{\text{act}} &= V_{\text{theo}} * v \\ &= 4.4 * 10^{-4} * 0.6912 \\ &= 3.05 * 10^{-4} \text{ [m}^3\text{/s]}\end{aligned}$$

3.5 Theoretical Cycle Design

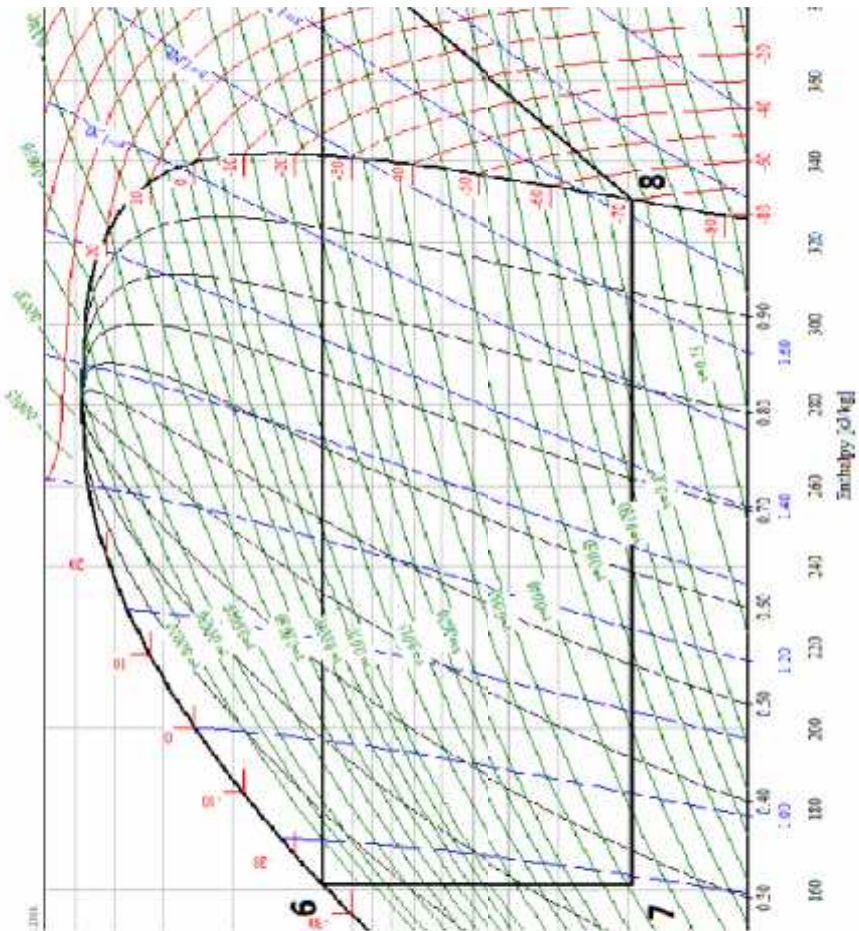


Figure 3.5 P-h Diagram for R-23

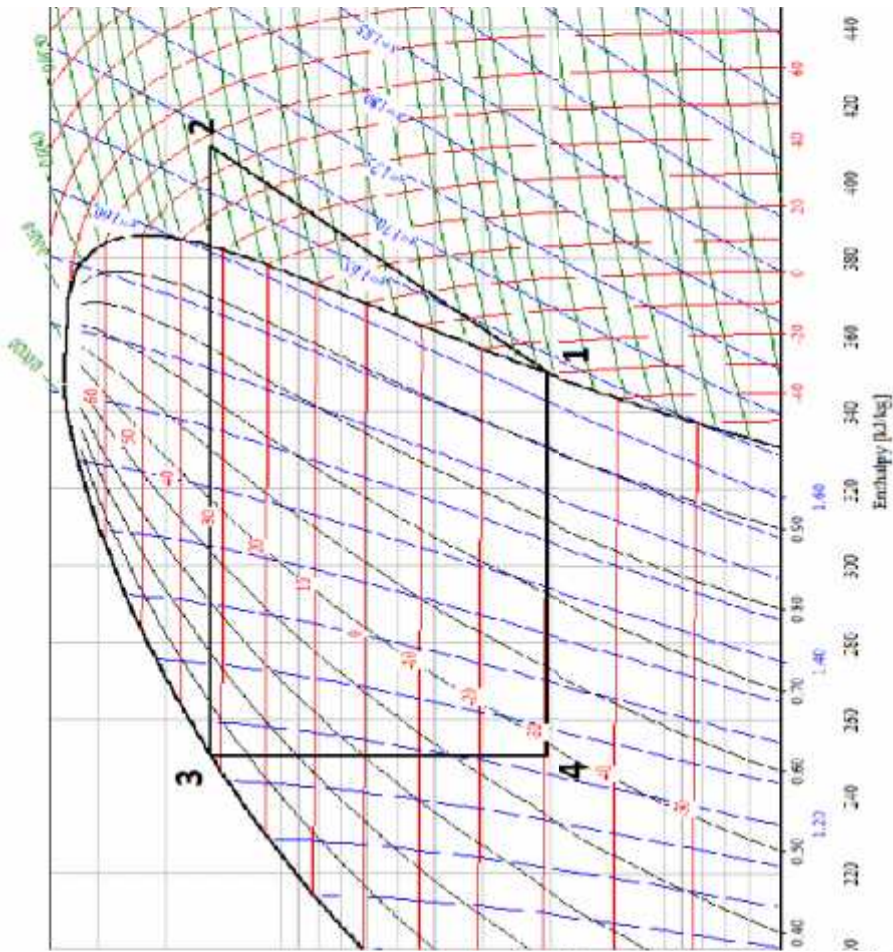


Figure 3.6 P-h Diagram for R-404a

Properties of lower cycle by using cool pack software program. [reference 8].

Table 3.3 Properties of Lower Cycle

points	T [°C]	P [bar]	v [m ³ /kg]	h [kJ/kg]	s [kJ/kg.°k]
5	35	11.955	0.02808	396.506	1.7816
6	-25	11.955	N/A	161.394	0.857
7	-70	1.941	0.03152	161.394	0.89
8	-70	1.941	0.11635	330.619	1.7225

Properties of upper cycle by using cool pack software program. [reference 8].

Table 3.4 Properties of Upper Cycle

points	T [°C]	P [bar]	v [m ³ /kg]	h [kJ/kg]	s [kJ/kg.°k]
1	-29.938	2.045	0.095129	350.304	1.6302
2	57.129	15.281	0.014671	409.305	1.6871
3	32.669	15.281	N/A	250.721	1.17
4	-29.938	2.045	0.04573	250721	1.22

3.5.1 Calculations for Lower Stage Using R-23

$$Q_e = 359.88 \text{ [W]} = 0.48 \text{ [Hp]}$$

It decided to select evaporator 1/2 [Hp]

$$Q_e = \dot{m}_1 \cdot q_e \dots\dots\dots (3.1)$$

$$Q_e = \dot{m}_1 \cdot (h_7 - h_8) \dots\dots\dots (3.2)$$

$$\dot{m}_1 = \frac{Q_e}{h_8 - h_7}$$

Where:

\dot{m}_1 : Mass flow rate for lower cycle [kg/s]

Q_e : Heat transfer rate in evaporator (evaporator load) [W]

q_e : Refrigeration effect [kJ/kg]

h_8 : Enthalpy at point inlet compressor [kJ/kg]

h_7 : Enthalpy at point inlet evaporator [kJ/kg]

$$\dot{m}_1 = \frac{0.35988}{330.6 - 161.4}$$

$$\dot{m}_1 = 0.002126 \text{ [kg/s]}$$

$W_{in(1)}$: Lower compressor work [kJ/kg]

$$W_{in(1)} = (h_5 - h_8) \dots \dots \dots (3.3)$$

$$W_{in(1)} = (396.506 - 330.6)$$

$$W_{in(1)} = 65.906 \text{ [kJ/kg]}$$

P_1 : Lower compressor power [W]

$$P_1 = W_{in(1)} * \dot{m}_1 \dots \dots \dots (3.4)$$

$$P_1 = 65.906 * 0.002126 = 0.140 \text{ [kW]} = 140 \text{ [W]} = 0.187 \text{ [Hp]}$$

It decided to select compressor 1/4 [Hp]

Using energy balance in heat exchanger to calculate \dot{m}_2

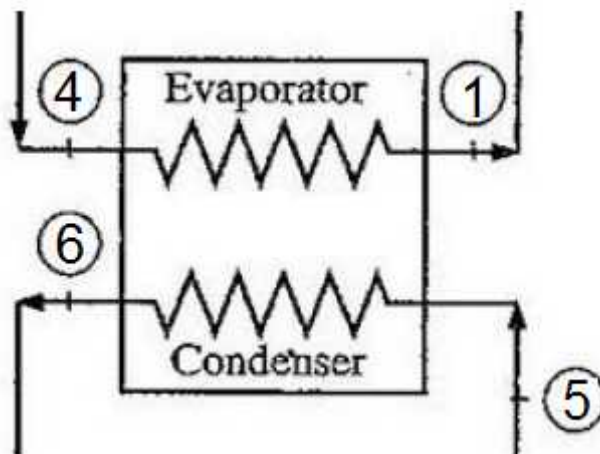


Figure 3.7 Heat Exchanger Balance

$$\dot{m}_1(h_5 - h_6) = \dot{m}_2(h_1 - h_4) \dots \dots \dots (3.6)$$

$$\dot{m}_2 = \frac{\dot{Q}_c \cdot \dot{m}_1}{\dot{Q}_e}$$

Where:

\dot{m}_2 : Mass flow rate for upper cycle [kg/s]

h_5 : Enthalpy at point outlet compressor [kJ/kg]

h_6 : Enthalpy at point outlet condenser [kJ/kg]

h_1 : Enthalpy at point inlet compressor [kJ/kg]

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

$$\dot{m}_2 = \frac{235.11 + 0.002126}{99.592}$$

$$\dot{m}_2 = 0.005012 \text{ [kg/s]}$$

3.5.2 Calculations for Upper Stage Using R-404a

$$Q_C = \dot{m}_2(h_2 - h_3) \dots \dots \dots (3.7)$$

Where:

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

h_2 : Enthalpy at point outlet compressor [kJ/kg]

h_3 : Enthalpy at point outlet condenser [kJ/kg]

$$Q_C = 0.005012 \cdot (409.305 - 250.721)$$

$$Q_c = 0.795 \text{ [kW]} = 795 \text{ [W]} = 1 \text{ [Hp]}$$

It decided to select condenser 1 [Hp]

$W_{in(2)}$: Upper compressor work [kJ/kg]

$$W_{in(2)} = (h_2 - h_1) \dots \dots \dots (3.8)$$

$$W_{in(2)} = (409.305-350.304)$$

$$W_{in(2)} = 59 \text{ [kJ/kg]}$$

P_2 : Upper compressor power [W]

$$P_2 = W_{in(2)} * \eta_2 \dots \dots \dots (3.9)$$

$$P_2 = 40.71 * 0.005012 = 0.295 \text{ [kW]} = 295 \text{ [W]} = 0.395 \text{ [Hp]}$$

It decided to select compressor 1/2 [Hp]

3.5.3 Calculations of Coefficient of Performance (COP)

$$Q_e = 359.88 \text{ [W]}$$

$$P_{total} = P_1 + P_2 = 140 + 295$$

$$P_{total} = 435 \text{ [W]}$$

$$COP = \frac{Q_e}{P_{total}}$$

$$COP = 359.88 / 435$$

$$COP = 0.827$$

Chapter Four

Components of Cascade Refrigeration system

4.1 Introduction

The refrigeration cycle can be described with four main components: compressor, condenser (high-temperature heat rejection), expansion device, and evaporator (low-temperature heat addition) and it's also contains some accessories (Heat exchanger, sight glass, filter, etc...).

4.2 Compressor

The function of a compressor is to remove the vapor produced by the evaporator and to deliver it at a required higher pressure. The compressor can be compared to a heart pumping the blood (the refrigerant) inside the body (the compression cycle). In the basic compression cycle, the compressor is positioned between the evaporator and the condenser. Compressors can be installed in either single or multistage configuration, and can be connected to each other in series or in parallel. [Reference 14].

4.2.1 Reciprocating Compressor

Reciprocating compressors (piston compressors), are still widely used but have faced increasing competition from other compressor types in recent decades.

Inside the reciprocating compressor housing, one piston moves up and down in each cylinder. When the piston is at its lowest point, superheated gas enters the compressor through the inlet valves. When the piston moves up, the inlet valve closes and the gas pressure increases, due to the reduced volume. The compressed gas leaves the compressor when the pressure is high enough to open the exit valve. The downward piston action initiates a new intake of gas through the valve.[Reference 14].

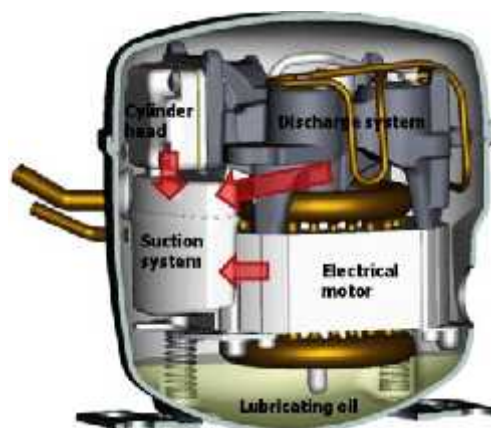


Figure 4.1 Reciprocating Compressor

4.3 Evaporator

Evaporator is a heat transfer device in which a volatile liquid is allowed to vaporize, removing heat from a space or product. Evaporators are used in mechanical refrigeration systems to absorb thermal energy. Air, water and brines are generally used to absorb heat from products, spaces and manufacturing processes to keep the temperature and humidity within design parameters. These fluids transport energy to the evaporator through natural or forced convection currents. [Reference 1].

4.3.1 Finned Tube Evaporator

The finned evaporators are the bare tube type of evaporators covered with the fins. When the fluid (air or water) to be chilled flows over the bare tube evaporator lots of cooling effect from the refrigerant goes wasted since there is less surface for the transfer of heat from the fluid to the refrigerant. The fluid tends to move between the open spaces of the tubing and does not come in contact with the surface of the coil, thus the bare tube evaporators are less effective. The fins on the external surface of the bare tube evaporators increases the contact surface of the of the metallic tubing

with the fluid and increase the heat transfer rate, thus the finned evaporators are more effective than the bare tube evaporators.[Reference 1].

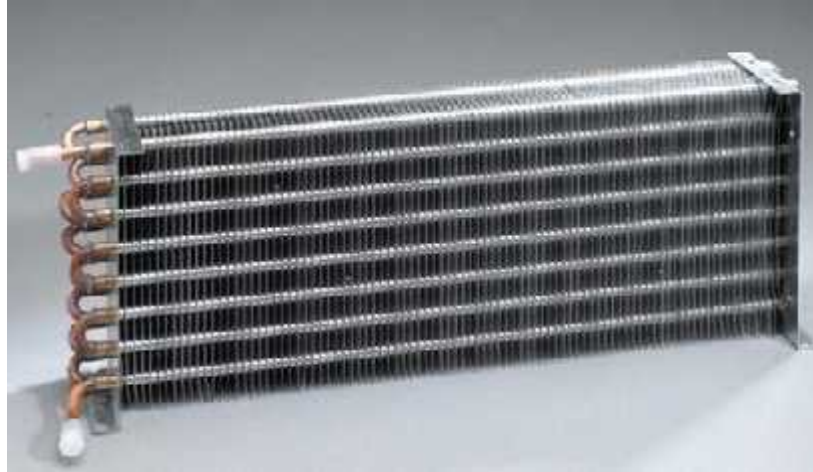


Figure 4.2 Finned Tube Evaporator

4.4 Condenser

Condenser is a heat transfer device use to transfer the heat contained in hot refrigerant vapor to the condensing medium thereby allowing the refrigerant to return to its liquid state so that refrigeration cycle will continue.[Reference 1].

4.4.1 Air Cooled Condenser

The air cooled condensers find application in domestic, commercial, and industrial refrigerating, chilling, freezing, and air conditioning systems, the centrifugal fan are used in the condenser particularly for heat recovery and auxiliary ventilation applications. In fact, they employ outside air as cooling medium [reference 20].

Fans draw air past the refrigerant coil and the latent heat of the refrigerant is removed as sensible heat by the air stream.



Figure 4.3 Air Cooled Condenser

4.5 Expansion Valve

Expansion valve are metering device that are used to modulate the flow of refrigerant into an evaporator. There are seven basic types of metering, or refrigerant flow control device: hand expansion valve, constant pressure expansion valve, thermostatic expansion valve, capillary tubes, low pressure float valves, high pressure float valves, and short tube restrictors.[Reference 1].

4.5.1 Capillary Tube

Instead of the orifice, the capillary is small diameter tubing that offers the restricted flow of the refrigerant. Its internal diameter ranges from 0.020 to 0.090 inches depending upon the capacity of the refrigerating or air-conditioning system. The pressure drop attained through the capillary depends upon its diameter and length. Capillary tubing made of copper is most commonly used, Capillary tubing is used for small refrigerating and air-conditioning systems like household refrigerators, water coolers, deep freezers, window air-conditioners, split air-conditioners, small packaged air-conditioners etc.[Reference 1].



Figure 4.4 Capillary Tube

4.6 Heat Exchanger

A heat exchanger is a device that allows heat from a fluid (a liquid or a gas) to pass to a second fluid (another liquid or gas) without the two fluids having to mix together or come into direct contact.

4.6.1 Double pipe Heat Exchanger

A heat exchanger is a device built for efficient heat transfer from one fluid to another, whether the fluids are separated by a solid wall so that they never mix, or the fluids are directly contacted. They are widely used in petroleum refineries, chemical plants, petrochemical plant, natural gas processing, refrigeration, power plant, air conditioning and space heating. We will use double pipe heat exchanger in this project. [Reference 4].

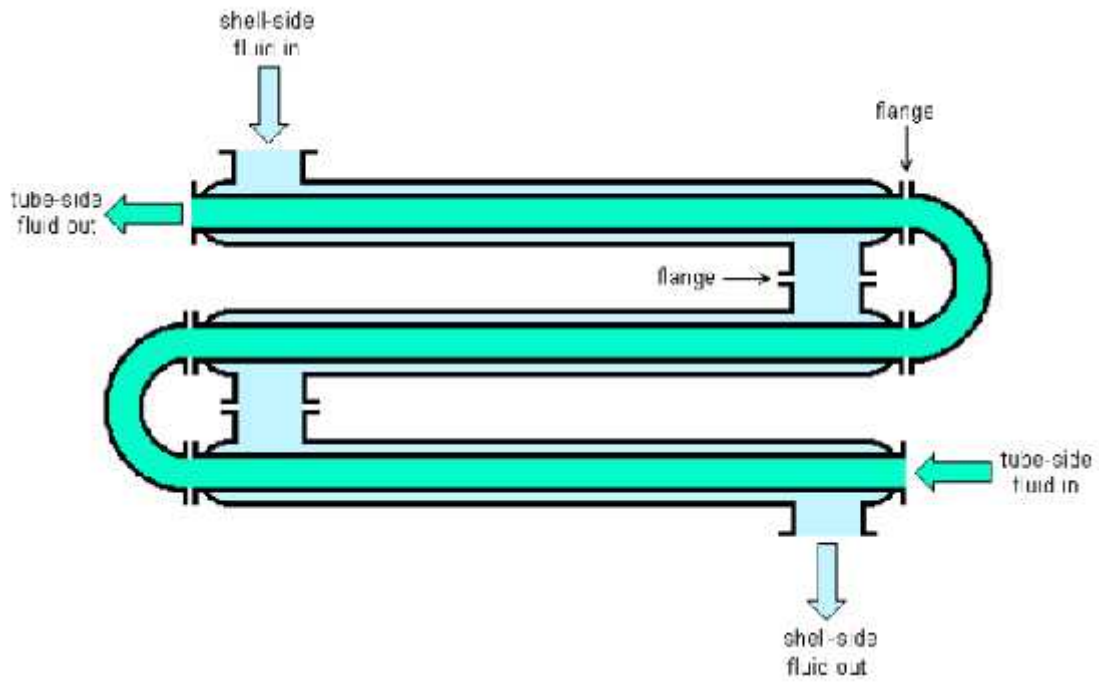


Figure 4.5 Double pipes Heat Exchanger

4.7 Sight Glass

Sight glass is a transparent glass tube or window installed in a tank or boiler system that allows an observer to view what is happening inside the container, ensuring optimal quality and safety. Sight glasses can also serve a number of functions aside from basic viewing, but this is the most common use.



Figure 4.6 Sight Glass

4.8 Thermostat

Thermostats are designed having out-in and cut-out points that correspond to the limits established by the set point plus or minus one-half of the control differential. The cut-in temperature is the control point that commands the compressor to start when the maximum limit of the space or product temperature has been reached. The cut-out temperature is the control point that commands the compressor to stop when the minimum limit space or product temperature has been reached.

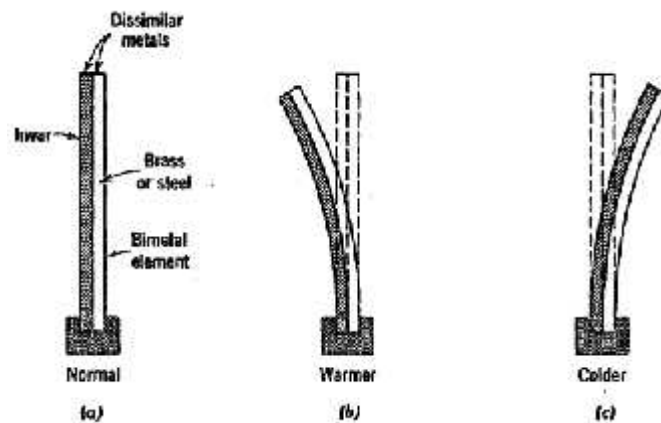


Figure 4.7 Bimetal-type Temperature Sensing Element

Increasing thermal intensity of the bimetal causes it to warp in the direction of the invar as shown in Figure 25-7b. Conversely/decreasing the temperature of the bimetal element causes the strip to warp in the direction of the brass or steel as shown in Figure 25-7c. This temperature actuated motion is utilized to actuate the electrical contacts other controlling mechanisms in thermostats. [Reference 1].



Figure 4.8 Thermostat

4.9 Filter Dryer

The filter-dryer is typically installed in the liquid line of a refrigeration system in order to avoid the high-pressure drop which would result if the unit were installed in the portion of the system in which the refrigerant is in a vapor state. Pressure drop anywhere in the system reduces the refrigeration capacity.

It is conventional in most refrigeration systems to utilize a filter-dryer in the system for filtering particles of solid foreign matter from the fluid and for adsorbing water, acids and other impurities. The solid contaminants might occur as casting sand, copper oxide and metal fine from manufacturing and assembly procedures as well as from wear during operation while chemical reactions within the system may form sludge's and varnishes which plug orifices within the system.[Reference 17]



Figure 4.9 Filter Dryer

Chapter Five

Pipes Design

5.1 Pipes Design and Selections

5.1.1 Low cycle pipe calculations

$$Q = \dot{m} \cdot v$$

Where:

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

v : specific volume [m^3/kg]

For suction line:

$$\begin{aligned} Q &= \dot{m}_1 \cdot v_8 \\ &= 0.002126 \cdot 0.11635 \\ &= 2.473 \cdot 10^{-4} [\text{m}^3/\text{s}] \end{aligned}$$

$$A = Q/V$$

Where:

A: cross sectional area [m^2]

V: velocity of refrigerant [m/s] From Table A-6

$$\begin{aligned} A &= Q/V \\ &= 2.473 \cdot 10^{-4} / 10 \\ &= 2.473 \cdot 10^{-5} [\text{m}^2] \end{aligned}$$

$$A = \frac{d^2}{4}$$

Where:

d inner diameter [m]

$$\begin{aligned} d &= \sqrt{A \cdot 4} \\ &= \sqrt{4 \cdot 2.473 \cdot 10^{-5}} \\ &= 5.6 \cdot 10^{-3} [\text{m}] \end{aligned}$$

$$\begin{aligned}
 r &= d/2 \\
 &= (5.6 \cdot 10^{-3})/2 \\
 &= 2.8 \cdot 10^{-3} [\text{m}]
 \end{aligned}$$

For discharge line:

$$\begin{aligned}
 Q &= v_1 \cdot A_1 \\
 &= 0.002126 \cdot 0.02808 \\
 &= 5.97 \cdot 10^{-5} [\text{m}^3/\text{s}]
 \end{aligned}$$

$$\begin{aligned}
 A &= Q/V \\
 &= 5.97 \cdot 10^{-4} / 15 \\
 &= 3.98 \cdot 10^{-6} [\text{m}^2]
 \end{aligned}$$

$$\begin{aligned}
 d &= \sqrt{(A \cdot 4)} \\
 &= \sqrt{(4 \cdot 3.98 \cdot 10^{-6})} \\
 &= 2.25 \cdot 10^{-3} [\text{m}]
 \end{aligned}$$

$$\begin{aligned}
 r &= d/2 \\
 &= (2.25 \cdot 10^{-3})/2 \\
 &= 1.125 \cdot 10^{-3} [\text{m}]
 \end{aligned}$$

5.1.2 High cycle pipe calculations

For suction line:

$$\begin{aligned}
 Q &= v_2 \cdot A_2 \\
 &= 0.005012 \cdot 0.095129 \\
 &= 4.76 \cdot 10^{-4} [\text{m}^3/\text{s}]
 \end{aligned}$$

$$A=Q/V$$

$$= 4.76 \cdot 10^{-4} / 10$$

$$= 4.76 \cdot 10^{-5} [\text{m}^2]$$

$$d = \sqrt{\frac{A \cdot 4}{\pi}}$$

$$= \frac{\sqrt{4 \cdot 4.76 \cdot 10^{-5}}}{\pi}$$

$$= 7.78 \cdot 10^{-3} [\text{m}]$$

$$r = d/2$$

$$= (7.78 \cdot 10^{-3})/2$$

$$= 3.89 \cdot 10^{-3} [\text{m}]$$

For discharge line:

$$Q = v \cdot A$$

$$= 0.005012 \cdot 0.014671$$

$$= 7.35 \cdot 10^{-5} [\text{m}^3/\text{s}]$$

$$A=Q/V$$

$$= 7.35 \cdot 10^{-5} / 15$$

$$= 4.9 \cdot 10^{-6} [\text{m}^2]$$

$$d = \sqrt{\frac{A \cdot 4}{\pi}}$$

$$= \frac{\sqrt{4 \cdot 4.9 \cdot 10^{-6}}}{\pi}$$

$$= 2.49 \cdot 10^{-3} [\text{m}]$$

$$r = d/2$$

$$= (2.49 \cdot 10^{-3})/2$$

$$= 1.25 \cdot 10^{-3} [\text{m}]$$

Table 5.1 Recommended Refrigerant Velocities [Reference 4]

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

Chapter Six

Electrical Design

6.1 Introduction

An electric circuit is a path in which electrons from a voltage or current source flow. The point where those electrons enter an electrical circuit is called the "source" of electrons. The point where the electrons leave an electrical circuit is called the "return" or "earth ground". The exit point is called the "return" because electrons always end up at the source when they complete the path of an electrical circuit.

6.2 Solar Energy

In the field of solar energy, Palestine is very suitable for producing electricity from the sun, where the number of sunny days is about 300 days a year, allowing to receive solar radiation at high rates in the year.

Which is an effective alternative, in shade of the increasing demand for electric loads, where Palestinians need to find alternatives to electricity, one of those solutions was the importance of generating solar energy.[reference 7]

6.2.1 The basic components of the solar energy system

6.2.1.1 Photovoltaic solar panels

A solar cell, or photovoltaic cell, is an electrical device that converts the energy of light directly into electricity by the photovoltaic effect, which is a physical and chemical phenomenon .it is a form of photoelectric cell, defined as a device whose electrical characteristics, such as current, voltage, or resistance, vary when exposed to light. Individual solar cell devices can be combined to form modules, otherwise known as solar panels.

Photovoltaic solar panel absorb sunlight as a source of energy to generate electricity. A photovoltaic (PV) module is a connected assembly of photovoltaic solar cells. Photovoltaic modules constitute the photovoltaic array of a photovoltaic system that generates and supplies solar electricity in commercial and residential applications. [Reference].



Figure 6.1 Photovoltaic solar panels

Will choose a photovoltaic solar panels because it has a very high performance and efficiency rate.

6.2.1.2 Solar array mounting racks

Solar panels are joined into arrays and commonly mounted in one of three ways: on roofs, on poles in free standing arrays or directly on the ground.

Roof mounted systems are the most common. This approach is aesthetic and efficient. The main drawback of roof mounting is maintenance. For high roofs, clearing snow or repairing the systems can be an issue. Panels do not usually require much maintenance.

Tracking arrays move with the sun. Tracking array move east to west with the sun and adjust their angle to maintain the optimum as the sun moves. [Reference12].



Figure 6.2 Solar array mounting racks

6.2.1.3 Charge controller (charge regulator)

Charge regulator limits the rate at which electric current is added to or drawn from electric batteries. [Reference] It prevents overcharging and may protect against overvoltage, which can reduce battery performance or lifespan, and may pose a safety risk. It may also prevent completely draining ("deep discharging") a battery, to protect battery life. [Reference 12].



Figure 6.3 Charge controller

6.2.1.4 Battery pack

Solar power systems produce electricity during the daytime, when the sun is shining. Our project demands electricity at night and on cloudy days – when the sun isn't shining. To offset this mismatch, batteries can be added to the system. [Reference 12].



Figure 6.4 Battery pack

6.2.1.5 Inverter

Solar panels and batteries produce DC (direct current) power. Standard home appliances use AC (alternating current). An inverter converts the DC power produced by the solar panels and batteries to the AC power required by appliances.[Reference 12].



Figure 6.5 Inverter

6.3 Components of Electrical Circuits

The electrical circuits have many electrical components:

6.3.1 Contactors

Magnetic contactors are a form of electrical relay found on most electrically powered motors. They act as a go-between for direct power sources, and high-load electrical motors in order to homogenize or balance out changes in electrical frequency which may come from a power supply as well as to act as a safeguard. It should be noted that though they are similar in design, magnetic contactors are not circuit breakers. They do not sever the connection between appliance, and power source during a short circuit. They are detachable from a motor so that an operator may work with that motor; disassemble or maintain it, without the possibility of live current still passing through the device. [Reference 10].



Figure 6.6 Contactor

6.3.2 Overload Relays

Thermal overload relays are economic electromechanical protection devices for the main circuit. They offer reliable protection for motors in the event of overload or phase failure. The thermal overload relay can make up a compact starting solution together with contactors. [Reference 11].



Figure 6.7 Overload Relays

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 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. [Reference 5].



Figure 6.8 Capacitor Start and Run Motor

6.4 Electrical Calculations

6.4.1 Electrical Energy Consumption Calculations

Energy consumption calculations:

$$E \text{ [kWh/day]} = P \text{ [W]} \times t \text{ [h/day]} / 1000 \text{ [W/kW]}$$

Where:

E: Energy consumption

P: Compressor power in Watts [W] equal 488.9 [W]

t: times number of usage hours per day and equal 12 [h/day]

$$E = (291.8) * 12 / 1000$$

$$E = 3.5 \text{ [kWh/day]} = 1278 \text{ [kWh/year]}$$

Electricity cost calculation for one year

$$\text{Cost} = E \text{ [kWh/year]} * \text{Price [/kWh]}$$

$$= 1278 * 0.57$$

$$\text{/year]} = 728.5 \text{ [$$

6.4.2 Solar Energy Consumption calculations

Solar Electricity cost calculations for Sunny day in one year is

$$/kWh] * \text{Sunny days in one year [Day/year]} \text{Cost} = E \text{ [kWh/day]} * \text{Cost} \quad [$$

$$= 3.5 * 300 * 0.57$$

$$/year] = 598.5 \quad [$$

Yearly cost = Electricity cost calculation for one year - Solar Electricity cost calculation for Sunny day in one year.

$$\text{Yearly Cost} = 728.5 - 598.5$$

$$/year] \quad \text{Yearly Cost} = 130 \quad [$$

6.5 Control Circuit

Table 6.1 System component

Component name	Quality	Component symbol
Thermostat	2	TH
Switch NO	2	S
Timer	1	T6H
Over load	2	F
Emergency switch	1	EM
Contactora	2	K

Control description

1. Cycle starts after pressing switch (s1) or by the thermostat (TH1) if the temperature more than (-70 c°), then electrical current flows through the emergency switch (EM) and the overload.
2. Then current flow to contactor (K1), so R404a Compressor and condenser fan “on”
3. When the Second Thermostat (TH2) detect that the temperature reach (-30c°), then electrical current flows through the overload to two parallel lines (Contactor K2 and timer).
4. Then the contactor K2 will turn on the R23 compressor and Evaporator fan. The timer start counting for 6 hours, after 6 hours the timer will turn on evaporator heater and shut down contactor K2.
5. When the temperature reach (-70c°), thermostat (TH1) shut down the system.

Table 6.2 component name

Contactor Symbol	Component name
K1	Compressor R404a and condenser fan contactor
K2	Compressor R23 and evaporator fan contactor
TH1	Thermostat (-70c°)
TH2	Thermostat (-30 c°)

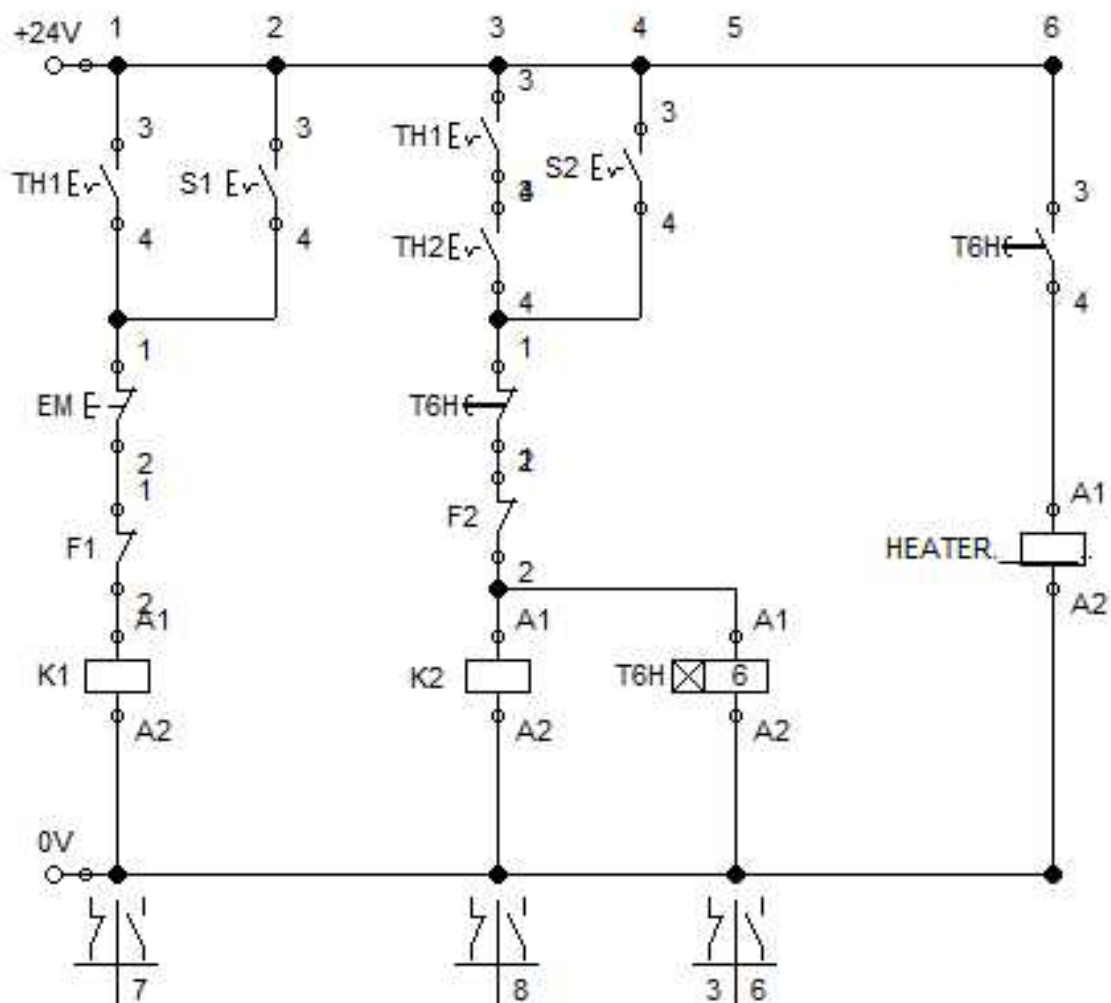


Figure 6.9 Control Circuit

6.6 Power Circuit

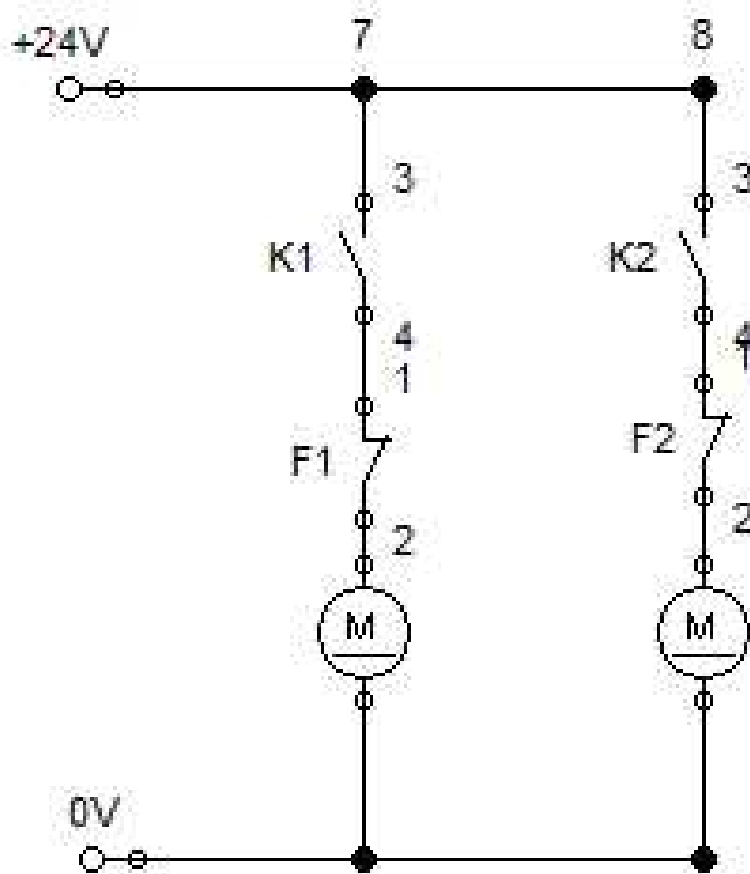


Figure 6.10 Power Circuit

Table 6.3 motor name

Motor symbol	Motor name
M1	R404a Compressor and condenser fan
M2	R23Compressor and Evaporator fan

6.7 Solar energy calculations

Higher cycle compressor power = 295 [W]

Lower cycle compressor power = 140 [W]

Evaporator fan power = 25 [W]

Condenser fan power = 25 [W]

Daily power consumption for higher cycle = unit power * working hours

$$= (295 + 25) * 12 = 3840 \text{ [W/h]}$$

Daily power consumption for Lower cycle = unit power * working hours

$$= (140 + 25) * 12 = 1980 \text{ [W/h]}$$

Total daily power consumption = 1980 + 3840 = 5820 [W/h]

Total daily power consumption with safety factor = 5820 * 1.3

$$= 7566 \text{ [W/h]}$$

Energy panels = Total daily power consumption / rate of solar radiation

$$\text{Energy panels} = 7566 / 5.4 = 1401.11 \text{ [W]}$$

Number of panels = Energy Panels / The capacity of the panel

$$= 1401.11 / 800 = 2 \text{ panels}$$

Battery number = daily power consumption * 1.3 * (365 - sunny days in the year) / (24 * battery ampere)

$$= (7566 * 1.3 * 60) / (24 * 1000) = 25 \text{ battery}$$

Inverter capacity = Total cycle power * 1.3

$$= (295 + 140 + 25 + 25) * 1.3$$

$$= 630.5 \text{ [W]}$$

Chapter Seven

Testing

7.1 Testing for the project

Starting Operation



Figure 7.1 Temperature at starting operation

After one hour



Figure 7.2 Temperature after one hour

After two hour



Figure 7.3 Temperature after two hour

After three hour



Figure 7.4 Temperature after three hour

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Dedication

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