بسم الله الرحمن الرحيم

Palestine Polytechnic University

College of Engineering & Technology

Mechanical Engineering Department

Graduation Project

Design and building of a Cascade Refrigeration System For Plant Tissues Storage

Project Team

Raed Khalil Mohammed **Shadi Tayseer Mesleh**

Tareq Ismael Khlawi

Project Supervisors

Eng. Kazem Osaily

Hebron – Palestine
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Abstract

This project aims for design and building of a Cascade Refrigeration System which works between (35 to -65 °C).

It uses two different refrigerants. Based on a thorough study and analysis of the characteristics of some different refrigerants, we selected refrigerant R-23 and R-22 for this cycle.

This system consists of two separate cycles connected with each other through a heat exchanger, where the higher cycle using refrigerant R-22 operates between (35 to -25 °C), and the low cycle using refrigerant R-23 operates between (-20 to -65 °C).This cycle finds applications in different scientific areas. One of these is the field of conservation biology of endangered plant tissues in order to reproduce to protect and preserve it.

This is the application that we have chosen for its importance in preserving the genetic origins of plants.

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CHAPTER ONE

INDRODUCTION

- History Of Refrigeration
- Importance of project
- Refrigerants
- Refrigeration cycle

1.1 History Of Refrigeration

In the past time man faced many problems to keep his food from spoiling, and the food can't be found at every season, so he invented primeval ways to solve these two main problems.

Hebrews, Greeks, and Romans placed large amounts of snow into storage pits, dug into the ground and insulated with wood and straw. In India, evaporative cooling was employed when a liquid vaporizes rapidly, it expands quickly. The rising molecules of vapor abruptly increase their kinetic energy and this increase is drawn from the immediate surroundings of the vapor. These surroundings are therefore cooled.

The intermediate stage in the history of cooling foods was to add chemicals like sodium nitrate or potassium nitrate to water causing the temperature to fall. Cooling drinks came into vogue by 1600 in France. Instead of cooling water at night, people rotated long-necked bottles in water in which saltpeter had been dissolved. This solution could be used to produce very low temperatures and to make ice. By the end of the 17th century, iced liquors and frozen juices were popular in French society.

Refrigeration is the process of removing heat from an enclosed space, or from a substance, to lower its temperature. A refrigerator uses the evaporation of a liquid to absorb heat.

The first known artificial refrigeration was demonstrated by William Cullen at the University of Glasgow in 1748. However, he did not use his discovery for any practical purpose. In 1805, an American inventor, Oliver Evans, designed the first refrigeration machine. The first practical refrigerating machine was built by Jacob Perkins in 1834; it used ether in a vapor compression cycle. An American physician, John Gorrie, built a refrigerator based on Oliver Evans' design in 1844 to make ice to cool the air for his yellow fever patients. German engineer Carl von Linden, patented not a refrigerator but the process of liquefying gas in 1876 that is part of basic refrigeration technology.

1.2 Importance of project

Attention began this type of cooling systems when needed rights of access to the very low temperatures. Where lies the importance of access to these scores to keep up with technological development and scientific massive occurred in the last quarter of the twentieth century. The process of testing the mechanical parts of the aircraft before takeoff to make sure that the probability of these parts of the circumstances which may be natural to these parts example, the process is no less important than industry. Using this session also in the process of testing clothes men space to see how their ability to maintain their lives. The use of these systems in the fields of medicine, biology and many other fields.

1.3 Refrigerants

The refrigerant is a heat carrying medium which during its cycle in the refrigeration system absorb heat from a low temperature system and discard the heat so absorbed to a higher temperature system.

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 including halo-carbon compounds, hydro-carbon compounds are used for air conditioning and refrigeration applications.

The stability of a refrigerant for a certain application is determined by its physical, thermodynamic, chemical properties and by various practical factors. There is no one refrigerant which can be used for all types of applications, there is no ideal refrigerant. If one refrigerant has a certain good advantage, it will have some disadvantage also. A refrigerant is chosen which has greater advantage and less disadvantage.

1.4 Refrigeration cycle

As we mentioned above people used primeval methods for storing food, then they created the refrigerator which is a mechanical system that uses many elements working together to get the required cooling, this system is known as simple refrigeration system.

The first refrigeration cycle is simple vapour compression refrigeration system, it is an improved type of air refrigeration system in which a suitable working substance, termed as refrigerant is used. It condenses and evaporates at temperatures and pressures close to the atmospheric conditions. The refrigerants usually used for this purpose are ammonia, carbon dioxide and sulphur dioxide. The refrigerant used does not leave the system, but is circulated throughout the system alternately condensing and evaporating. In evaporating, the refrigerant absorbs its latent heat from the brine which is used for circulating it around the cold chamber. While condensing, it gives out its latent heat to the circulating water of the cooler. The vapour compression refrigeration system is, therefore, a latent heat pump as it pumps its latent heat from the brine and delivers it to the cooler. Vapour compression refrigeration systems is now used for all purposes of refrigeration. It is generally used for various purposes from a small domestic refrigerator to a big air conditioning plant.

But some times, the vapour refrigerant is required to be delivered at a very high pressure as in the case of low temperature refrigeration system. In such cases we should compress the vapour refrigerant by employing two or more compressors placed in series. The compression carried out in two or more compressors is called multistage compression.

The cascade cycle is one of multistage applications, this system consists of two or more compression refrigeration systems in series which use two different refrigerants, one of these refrigerants has low boiling temperature.

The principal advantage of the cascade system is that it permits the use of two different refrigerants. The high temperature system uses a refrigerant with high boiling temperature as R-12 or R-22. The low temperature cascade system uses a refrigerant with low boiling temperature such as R-13 or R-23 or R-170. These low boiling temperature refrigerants have extremely high pressure which insures a smaller compressor displacement in the low temperature cascade system and a higher coefficient of performance. The cascade system was first used in 1877 for liquefaction of oxygen, employing sulpher doxide (SO2) and carbon dioxide (CO2) as intermediate refrigerants. The additional advantage of a cascade system over the multistage system is that the lubricating oil from one compressor can't wander to other compressor.

Our project is about cascade refrigeration system (design and building) which is two stage cycle, works between (-65 to 35)°C.

1.5 Project Outline

Chapter One:

This chapter is an introduction to a refrigeration system. it includes the history of refrigeration, refrigerants and refrigeration cycle, what is it ? advantages and disadvantages, what is cascade cycle ? and what are it's advantages and disadvantages ? and it contains time table.

Chapter Two:

This chapter explains why we use cascade refrigeration cycle? And it contains some comparisons between refrigeration cycles, and between cascade system and liquid Nitrogen, and it has cascade applications.

Chapter Three:

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

Chapter Four:

It describes the cooling loads and shows the sources of it and explains how to calculate it.

Chapter Five:

This chapter shows how to select a refrigerants and specifications must be available ,and includes cycle analysis manually and through the use of (cool back).

Chapter Six:

This chapter shows the design for pipes, condenser, evaporator and heat exchanger, and it shows the selection of capillary tube and oil separator.

Chapter Seven:

This chapter contains the electrical construction of the project.

Time table for our project in the previous semester is shown as in Table 1.1

Table 1.1 Time table

Weeks Task	1 2	3 4	5 6	7 8	9 10	11 12	13 14	15 16
Specifying the project								
Searching about sources & references								
Reading about project with details								
Writing ch1 & ch2								
Writing ch3								
Writing ch4 & ch5								
Finishing & Printing								

The time table for our project in the present semester is shown as in Table 1.2

Table 1.2 Time table

Weeks / Task	1 2	3 4	5 6	7 8	9 10	11 12	13 14	15 16
Contacted with beit-jala factory								
Design research								
Pipe design								
Heat exchanger design								
Condenser design								
Evaporator and capillary tube design								
Finishing and printing								

Budget table for our project is shown as in table 1.2

Table 1.2 Budget table

Service	Costs($)
Transportations	100
Inter net service	200
Printing	25
Total price	325$

CHAPTER TWO

CASCADE REFRIGERATION SYSTEM DESCRIPTION AND APPLICATIONS

- Why We Use Cascade Refrigeration Cycle?
- Comparison between refrigeration cycles
- Comparison Between Cascade Refrigeration System And Liquid Nitrogen
- Cascade Refrigeration System Applications

2.1 Why We Use Cascade Refrigeration Cycle?

Overcoming the shortages which are present in the simple refrigeration cycle became hysterics of the refrigeration engineers and they are trying to solve there problems.

Classic single stage refrigeration systems are limited in the lowest temperature they can maintain. The lower the evaporation temperature, the lower the vapour pressure in the evaporator. And with dropping vapour pressure, the compressor sucks in less refrigerant per stroke (at a low pressure, less refrigerant molecules per volume unit are available), reducing capacity to the point where load and capacity are in balance.

To solve this, we could use a refrigerant with lower boiling point, and thus a higher vapour pressure at desired evaporation temperatures, since higher pressures in the evaporator at a certain temperature also means a higher condensing pressure.

And even if a compressor could be found which can handle extremely high condensing pressures, there is a problem with the compression ratio (high side pressure divided by low side pressure). The efficiency of a reciprocating (piston) compressor drops fast when compression ratio increases.

To reach lower temperatures, one could be tempted to put two compressors in series (or use a 2-cylinder one), with the first having an enormous pumping capacity. The first compressor pumps up to an intermediate pressure. The second compressor then compresses up to condensing pressures. Compression ratio for each compressor is only the square root of the total compression ratio.

Also there are a few practical problems. Both compressors spit out oil. Oil from the first compressor will move to the second compressor, oil from the second compressor moves to the first compressor. If this is not in balance, which is very likely, the system will break sooner or later. Also, in this sample the first compressor needs a 10 times larger cylinder than the second one, making it expensive, bulky and noisy. Since the 2-stage method is obviously limited, one could pipe the two compressors in another way and build a cascade system as shown in figure 2.1

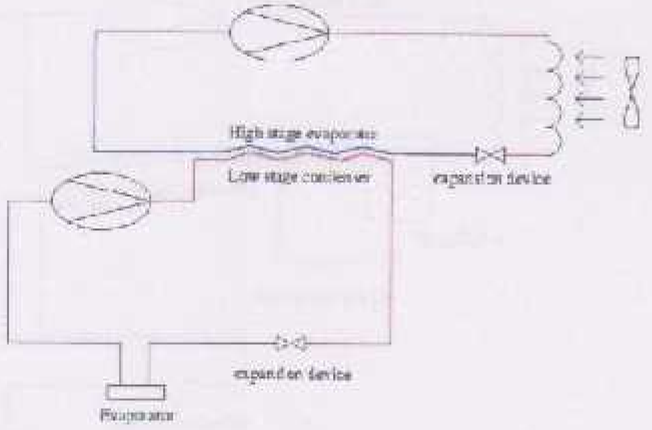


Figure 2.1 cascade system

Here, we still use two compressors, but both in a separate circuit, and with different refrigerants. We have a 'high stage' and a 'low stage'. The high stage condenses at normal temperatures, and evaporates at an arbitrarily low temperature. But instead of cooling a load directly with it, we cool the condenser of the low stage. Since low stage condensing temperature is very low now, we can use a high pressure refrigerant without dealing with insane pressures or compression ratios. Each sub-circuit can be designed to work optimal.

Of course, there are disadvantages also. We still need two expensive compressors, and we need to add a lot of complexity in the form of proper startup sequencing (the low stage can run only if the high stage is down to temperature) and piping. It also increases the size of the system.

To get rid of the expensive and bulky multiple compressor setups and startup sequencing, one could choose an autocascade figure 2.2. The autocascade also uses refrigerants with a different boiling points, but it uses only a single compressor and a device called 'phase separator'. Let's explain it using a picture.

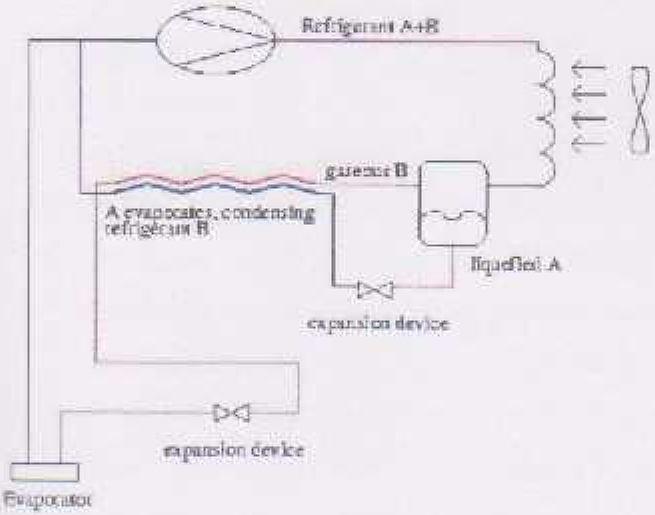


Figure 2.2 autocascade system

It works almost equally to a normal cascade, except that we now only have a single compressor compressing both refrigerants. At a certain pressure, the higher boiling point refrigerant (Refrigerant A) will start condensing to liquid. For refrigerant B pressure is not nearly high enough, so it will remain gaseous in the condenser.

The outlet of the condenser is coupled to a 'phase separator'. This device separates the liquid refrigerant A from the gaseous refrigerant B, coupling them out through separate lines. Then, refrigerant A is evaporated into a similar kind of heat exchanger as used in a 'classic' cascade. This drops the temperature of refrigerant B, allowing it to condense. Refrigerant B is further processed as normal: it is throttled into an evaporator to do work.

The big advantage of the autocascade is the use of a single compressor. This reduces cost and size. The big disadvantage is the special refrigerant mixture needed. Also fine tuning such a system is hard since only coupling a meter set to the system can destroy the delicate balance.

2.2 Comparison Between Refrigeration Cycles

Table 2.1 Comparison Between Refrigeration Cycles

Cycle	Advantages	Disadvantages
Simple refrigeration cycle	1. It has smaller size for a given capacity of refrigeration. 2. It has less running cost. 3. It can be employed over a large range of temperatures. 4. COP is quite high.	1. The initial cost is high. 2. The prevention of leakage of refrigerant is a major problem.
Multi stage refrigeration cycle	1. It improves the volumetric efficiency. 2. It reduces the leak loss. 3. It reaches very low temperatures 4. It provides effective lubrication because of lower temperatures range. 5. COP higher than simple cycle.	1. Its initial cost is higher than simple cycle. 2. It takes more size than simple cycle. 3. It requires more protection devices than simple cycle.
Cascade ref. cycle	1. Different refrigerants are used. 2. Different oils can be used in compressors. 3. reaches temperature less than -75°C.	1. More complicated than multi stage cycle. 2. It costs more than multi stage cycle. 3. overlap of the condensing temperature of the lower cycle with evaporating temperature of the higher cycle.
Autocascade ref. cycle	Cheaper than cascade cycle	Need special refrigerant mixture.

Carefully weighted the advantages and disadvantages of each of the previously described systems (2-stage, cascade and autocascade). Finally we choose the cascade due to the ability to optimize each stage separately. This system use one stage to work from (35 to-25) °C and the second stage works from (-15to -65)°C.

2.3 Comparison Between Cascade Refrigeration System And Liquid Nitrogen

Liquid nitrogen may be used for the same applications like cascade system but it has tow major problems, the first being contaminated by submersed samples so liquid nitrogen not only serves as a refrigerant, but like water can also act as a vehicle for the transmission of viruses, bacteria, fungi, so it will spoil specimen, and second, the observation that vapor phase storage systems may have temperature gradients which rise above the glass transition temperature of water, so refrigeration engineering experts find cascade system more effective than liquid nitrogen.

2.4 Cascade Refrigeration system Applications

Cascade refrigeration system is used in the following fields:

- Industrial fields
- Biological fields
- Medical fields

2.4.1 Industrial fields

many cryogenic tests can be used in industrial fields, such as tests used to determine the performance of mechanical and electrical equipment, or it could be used to test space men clothes to ensure it will stand in very low temperatures in space which may be less than -100 °C. And it could be used to condense volatile organic compounds. Other low-temperature applications may be used, in a chemical industry to provide a low- temperature environments for chemical reactions that would race out of control at room temperatures.

2.4.2 Biological fields

Most applications in this fields are related to the process of plan tissues storage of endangered species for future reproduction process. These tissues are preserved by cooling to low sub zero temperatures, at these low temperature any biological activity including chemical reactions that would lead to tissues death are effectively stopped, these tissues are preserved below -60 °C.

2.4.3 Medical fields

We can also use cascade systems to preserve sperms, and more recently skin, emproys, heart valves, etc....

In our project we will use our system for preserving plant tissues.

CHAPTER THREE

COMPONENTS OF CASCADE REFRIGERATION SYSTEM

- Introduction
- Compressors
- Condensers
- Evaporators
- Throttling Devices
- Heat exchanger
- Auxiliary components

3.1 Introduction

There are several mechanical components required in a cascade refrigeration system. In this part, we discuss the five major components of a system and some auxiliary equipment working with these major components.

Major components of a cascade refrigeration system are as follows:

- compressor,
- condenser,
- evaporator,
- throttling device, and
- heat exchanger.

3.2 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:

- Hermetic compressor,
- Semihermetic compressor, and
- Open compressor.

We will use hermetic compressor.

3.2. 1 Hermetic compressors

These compressors, are available for small capacities, motor and drive are sealed in compact welded housing. The refrigerant and lubricating oil are contained in this housing. Almost all small motor-compressor pairs used in domestic refrigerators, freezers, and air conditioners are of the hermetic type. Their revolutions per minute are either 1450 or 2800 rpm. Hermetic compressors can work for a long time in small-capacity refrigeration 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 3.1 shows hermetic compressor).

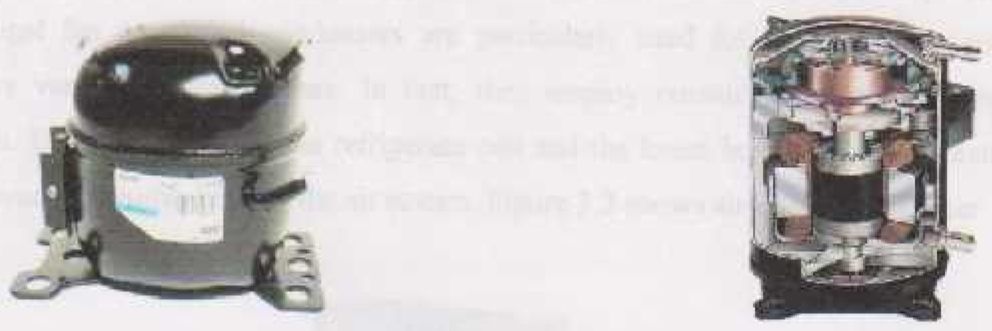


Figure 3.1 hermetic compressors

3.2.2 Expectations From The Compressors

The refrigerant compressors are expected to meet the following requirements:

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

3.3 Condensers

A condenser is a major system component of 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. There are many types and we will use air-cooled condenser type.

3.3.1 Air-cooled condensers

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 3.2 shows air-cooled condenser.

Figure 3.2 air-cooled condenser

3.4 Evaporators

The evaporator is an important device used in the low pressure side of a refrigeration system. The liquid refrigerant from the expansion valve enters into the evaporator where it boils and changes into vapour. The function of an evaporator is to absorb heat from the surrounding location or medium which is to be cooled, by means of a refrigerant. The temperature of the boiling refrigerant in the evaporator must always be less than that of the surrounding medium so that the heat flows to the refrigerant.

3.4.1 Bare Tube Coil Evaporator

The simplest type of evaporator is the bare tube coil evaporator, as shown in Figure 3.3. The bare tube coil evaporators are also known as prime-surface evaporators. Because of its simple construction, the bare tube coil is easy to clean and defrost. A little consideration will show that this type of evaporator, offers relatively little surface contact area as compared to other types of coils. The amount of surface area may be increased by simply extending the length of the tube, but there are disadvantages of excessive tube length. The effective length of the tube is limited by the capacity of expansion valve. If the tube is too long for the valve's capacity, the liquid refrigerant will tend to completely vaporize early in its progress through the tube, thus leading to excessive superheating at the outlet. The long tubes will also cause considerably greater pressure drop between the inlet and outlet of the evaporator. This results in a reduced suction line pressure.

The diameter of the tube in relation to tube length may also be critical. If the tube diameter is too large, the refrigerant velocity will be too low and the volume of refrigerant will be too great in relation to the surface area of the tube to allow complete vaporization. This, in turn, may allow liquid refrigerant to enter the suction line with possible damage to the compressor (slugging). On the other hand, if the diameter is too small, the pressure drop due to friction may be too high and will reduce the system efficiency. The bare tube coil evaporators may be used for any type of refrigeration requirement.

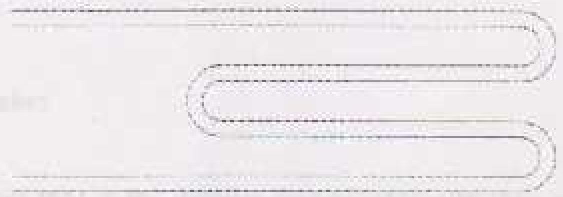


Figure 3.3 Bare Tube Coil Evaporator

3.5 Throttling Devices

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

- thermostatic expansion valves,
- constant pressure expansion valves,
- float valves, and
- capillary tubes.

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

3.5.1 Capillary tubes

The capillary tube is the simplest type of refrigerant flow control device and its shown in Figure 3.4 and may be used in place of an expansion valve. The capillary tubes are small-diameter tubes through which the refrigerant flows into the evaporator. These devices, reduce the condensing pressure to the evaporating pressure in a copper tube of small internal diameter (0.4-3 mm diameter and 1.5-5 m long), maintaining a constant evaporating pressure independently of the refrigeration load change. A capillary tube may also be constructed as a part of a heat exchanger, particularly in household refrigerators.

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

Figure 3.4 Capillary tubes

3.6 Heat exchanger

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

Figure 3.5 Double pipe heat exchanger

3.7 Auxiliary component

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

3.7.1 Oil Separator

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

3.7.2 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 3.6a 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 again.

The purpose of high-pressure control is to stop the compressor when the discharge pressure of the hot gas approaches a dangerous level. Figure 3.6b 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.

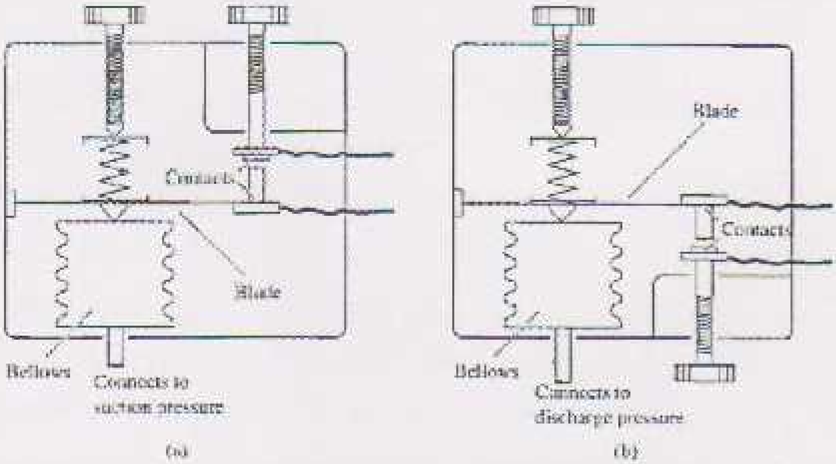


Figure 3.6 (a) Low pressure control and (b) high pressure control

CHAPTER FOUR

COOLING LOAD

- Introduction
- Load sources
- Load Calculation
- Total cooling load
- Required Equipment Capacity

4.1 Introduction

The total heat required to be removed from refrigerated space in order to bring it at the desired temperature 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.

4.2 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:

- The wall heat gain.
- The product heat gain.
- Infiltration heat gain.
- Packing heat gain .
- Doors' heater gain.

4.2.1 The wall heat gain load

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 tempetature 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.

$$Qwall = A \times U \times \Delta T \quad (4.1)$$

Where:-

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

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

ΔT: the temperature difference across the walls[C].

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

Where:

T_{in}: refrigerant space temperature.

T_{out}: outside temperature.

Overall heat transfer coefficient:

$$U = \frac{1}{\frac{1}{h_i} + \frac{\Delta x_1}{k_1} + \frac{\Delta x_2}{k_2} + \frac{\Delta x_3}{k_3} + ... + \frac{1}{h_o}} \quad (4.2)$$

Where:

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

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

k: the thermal conductivity of the material [W/m. C].

hi: the convection heat transfer coefficient of inside air [W/m^2. C].

ho: the convection heat transfer coefficient of outside air[W/m^2. C].

4.2.2 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.

$Q_{ch} = m\,C_p\,\Delta T$ ....(4.3)

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]

$\Delta T = (T_0 - T_{ch})$

T_0: entering product temperature [˚C]

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

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.

$Q_{bf} = m\,C'_p\,\Delta T$ ....(4.4)

where:

Q_{bf}: cooling product load below freezing in [kJ]

m: mass of the product in [kg]

C'_p: the specific heat below freezing in [kJ/kg.˚C]

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

T_{bf}: below freezing product temperature [˚C]

T_{Rs}: refrigerated space temperature [˚C]

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

$$Q_{freezing} = m.H_l \qquad (4.5)$$

where

$Q_{freezing}$: freezing load[kJ]

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

Total product load

$$Q_p = \frac{Q_{cb} + Q_f + Q_c}{\tau}[kW] \qquad (4.6)$$

where τ cooling load time in seconds

4.2.3 Infiltration heat gain

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.

$$Q_f = \frac{1250}{3600} \times \dot{V}_f \times (T_o - T_i)[W] \qquad (4.7)$$

wher:

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

T_o : outside temperature [°C]

T_i : inside temperature [°C]

4.2.4 Packaging Heat Load

Many products refrigerated in packages, it could be more than 10% of product's weight. Packages could be plastic, metal, wood and glass. We used plastic tubes for packaging the product. Tubes are arranged in holes existed in a plastic frame as shown in Figure 4.1

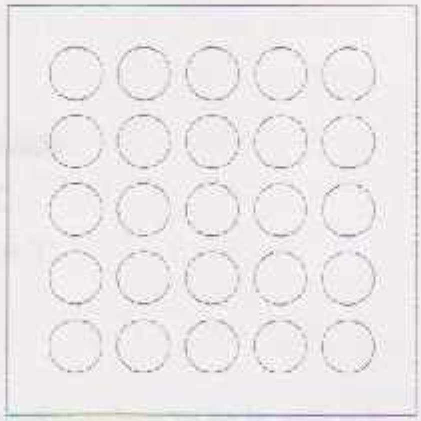


Figure 4.1 tubes plat

$$Q_{pk} = \frac{m_{pk} \times C_{pk} \times (t_o - t_r)}{\tau} \times 10^3 [W] \qquad (4.8)$$

where

Q_{pk} : Packaging Heat Load$[W]$

m_{pk} : mass of products[kg]

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

t_o : temperature of the outside[°C]

t_r : temperature of the refrigerant space[°C]

τ : runing time[hr]

4.2.5 Doors' Heater Gain

Heater used to prevent frost from forming around the door, making it difficult to open , and the heater is located near the door.

$$Q_h = \eta_u \times P [W] \qquad (4.9)$$

where:

P power of heater; heater
power taken as 60[W]

η_u : heater usage factor (0.1 - 0.5)

4.3 Load Calculations

General overview and information

Storage temperature is -65 °C

Surrounding temperature is 35 °C

Mass of product is 2[kg]

Running time is 16[hr]

We take highest product properties values to be in safe side

Figure 4.2 shows our refrigerator frame which is equal to 0.608[m^3]

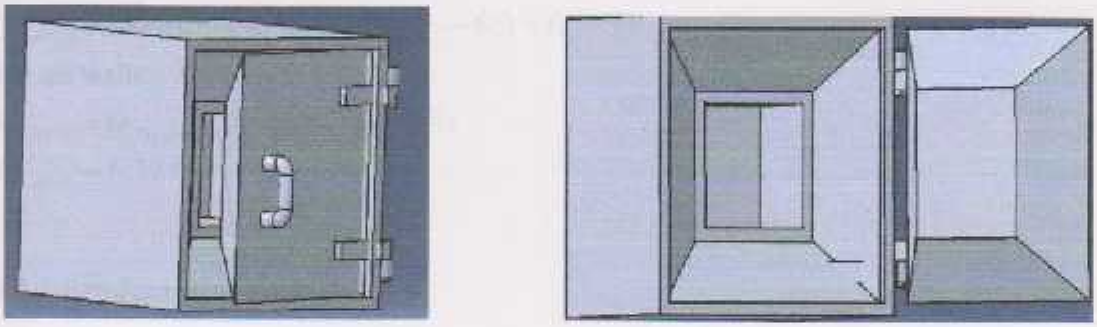


Figure 4.2 refrigerator frame

4.3.1 Walls heat gain

All walls are constructed of three layers as shown in Figure 4.3.

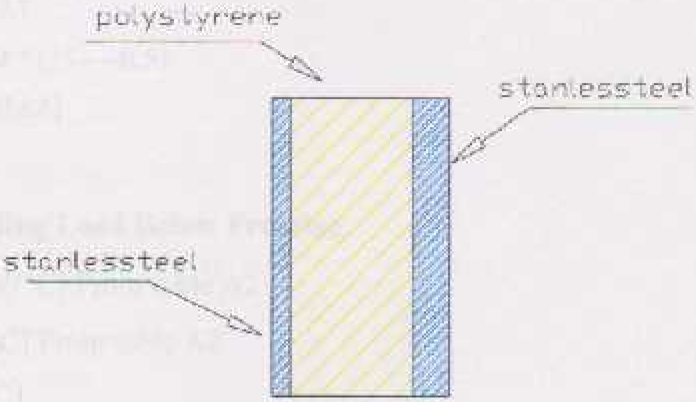


Figure 4.3 walls construction

Stainless steel 0.3 [cm]; k=15.6[W/m. °C]. From Table A1

Polystyrene 15 [cm]; k=0.029[W/m. °C]. From Table A1

Stainless steel 0.1 [cm]; k=15.6[W/m. °C]. From Table A1

h_i: refrigerated space =5[W/m^2. °C].

h_o: inside room =10[W/m^2. °C].

$$U=\frac{1}{\frac{1}{5}+\frac{0.003}{15.6}+\frac{0.15}{0.029}+\frac{0.001}{15.6}+\frac{1}{10}}$$

$$U=0.183[W/m^2.°C]$$

for one wall

$$Q_{wall}=0.183*(0.608*0.608)*(35--65)=6.77\ [W]$$

for all walls

$$Q_{wall\ all}=Q_{for\ one\ wall}*number\ \text{of walls}$$

$$Q_{wall\ all}=6.77*6=40.63\ [W]$$

4.3.2 Product heat gain

4.3.2.1 Chilling Load Above Freezing

C_p =4.06[kJ/kg °C] table A2 and we take it the highest value to be in safe side

T_0 =35[°C] Surrounding temperature

T_{ch} =-0.5 [°C] table A2 and we take it the highest value to be in safe side

$$Q_{ch}=mC_p\Delta T$$

$$Q_{ch}=2*4.06*(35--0.5)$$

$$Q_{ch}=288.26[kJ]$$

4.3.2.2 Cooling Load Below Freezing

C_p^* =2.05[W/°C] From table A2

T_{sf} = -0.5[°C] From table A2

T_{Rs} =-65 [°C]

$$Q_{bf} = m\,C'_p\,\Delta T$$
$$Q_{bf} = 2*2.05*(-0.5--65)$$
$$Q_{bf} = 264.45[kJ]$$

4.3.2.3 Freezing Load

H_L=321[kJ/kg] From table A2

$$Q_f = m.H_L$$
$$Q_f = 321*2$$
$$Q_f = 642[kJ]$$

Total product load

$$Q_p = \frac{Q_{ch} + Q_f + Q_{bf}}{\tau}$$
$$Q_p = \frac{288.26 + 264.45 + 642}{16\times 3600}$$
$$Q_p = 18.4[W]$$

4.3.3 Infiltration Heat Gain

$\dot{V}_f$ = 0.5[m³/h] From Table A3

$$Q_f = \frac{1250}{3600}\times \dot{V}_f \times (T_o - T_i)$$
$$Q_f = \frac{1250}{3600}\times 0.5\times(35--65) = 17.5[W]$$

4.3.4 Packing Heat Load

C_{pk}=1.6[kJ/kg°C] From table A4

$$Q_{pk} = \frac{m_{pk}\times C_{pk}\times(t_o - t_i)}{\tau}\times 10^3[W]$$
$$Q_{pk} = \frac{0.1\times 2\times 1.6\times(35--65)}{16\times 3600}\times 10^3$$
$$Q_{pk} = 0.5[W]$$

4.3.5 Doors Heater Load

$Q_h = \eta_w \times P$

$Q_h = 0.5 \times 60 = 30[W]$

4.4 Total cooling load .

$Q_T = Q_w + Q_p + Q_{inf} + Q_{pr} + Q_{heater}$

$= 90.3[W]$

Factor of safty we take 10%

TCL=$Q_T + Q_T \times FS$

TCL=$90.3+90.3 \times 0.1 = 100[W]$

4.5 Required Equipment Capacity

$$Q_e = \frac{TCL \times 24}{Runing\ Time} \qquad (4.10)$$

$$Q_e = \frac{100 \times 24}{16}$$

$Q_e = 133.3[W]$

CHAPTER FIVE

CYCLE ANALYSIS

- Refrigerants Selection
- Cycle Analysis

5.1 Refrigerants Selection

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

- low boiling point,
- high critical temperature,
- high latent heat of vaporization,
- low specific heat of liquid,
- low specific volume of vapor,
- non corrosive to metal,
- non flammable and non explosive,
- non toxic,
- low cost,
- easy to liquefy at moderate pressures and temperatures,
- easy of locating leaks by suitable indicator and
- mixes well with oil.

To select refrigerants successfully we must consider the above properties. We made a comparison between various refrigerants and found that, the best refrigerants could be used are R-22 for high stage cycle and R-23 for low stage cycle. In addition to above properties in selection process of R-22 we have to determine the compressor type that works perfectly with R-22, and it has to be cheap and available. And to select the low stage cycle refrigerant we have to consider important factor which is, avoiding vacuum occurrences, in other words (pressure inside cycle must be a little greater than atmospheric pressure). If the atmospheric pressure became greater than inside cycle pressure will destroy the cycle, so we select R-23. R-23 has very low temperature at 1 atm, so R-23 is suitable for our system.

5.2 Cycle Analysis

Figure 5.1 describes Ideal cascade refrigeration cycle

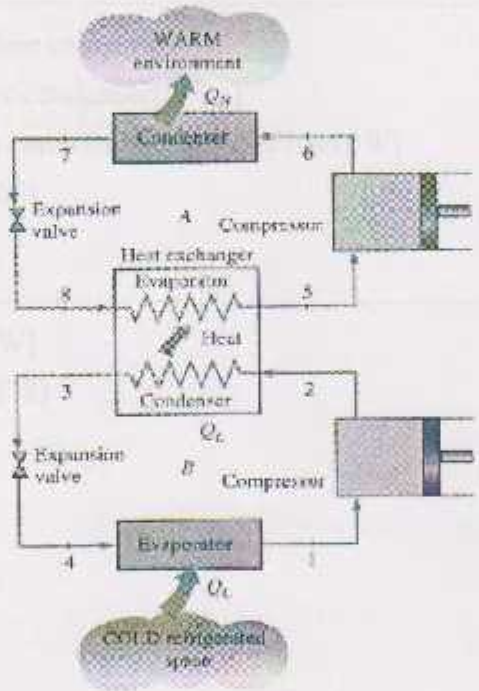


Figure 5.1 Ideal cascade refrigeration cycle

5.2.1 Calculations For Low Stage Using R23

Figure 5.2 shows P-h chart for R23

$$Q_e = m^*_1 q_e \quad \text{.......(5.1)}$$

$$= m^*_1 \times (h_1 - h_4) \quad \text{.......(5.2)}$$

$$m^*_1 = \frac{Q_e}{(h_1 - h_4)}$$

Where:

m^*_1 : mass flow rate[kg/s]

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

q_e : Refrigerating effect[kJ/kg]

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

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

$$m^*_1 = \frac{0.133}{(338-165)} = 0.000768[kg/s]$$

$$\dot{Q}_c = m^{\bullet}_1 \times (h_2 - h_3) \quad (5.3)$$

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

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

$\dot{Q}_c$: Heat transfer rate in condenser(condenser load)[W]

$\dot{Q}_c = 0.000768 \times (370 - 165)$

$\dot{Q}_c = 157.6[W]$

$$W_B = m^{\bullet}_1 \times (h_2 - h_1) \quad (5.4)$$

W_B : Compressor work[W]

$W_B = 0.000768 \times (370 - 338)$

$W_B = 24.57[W]$

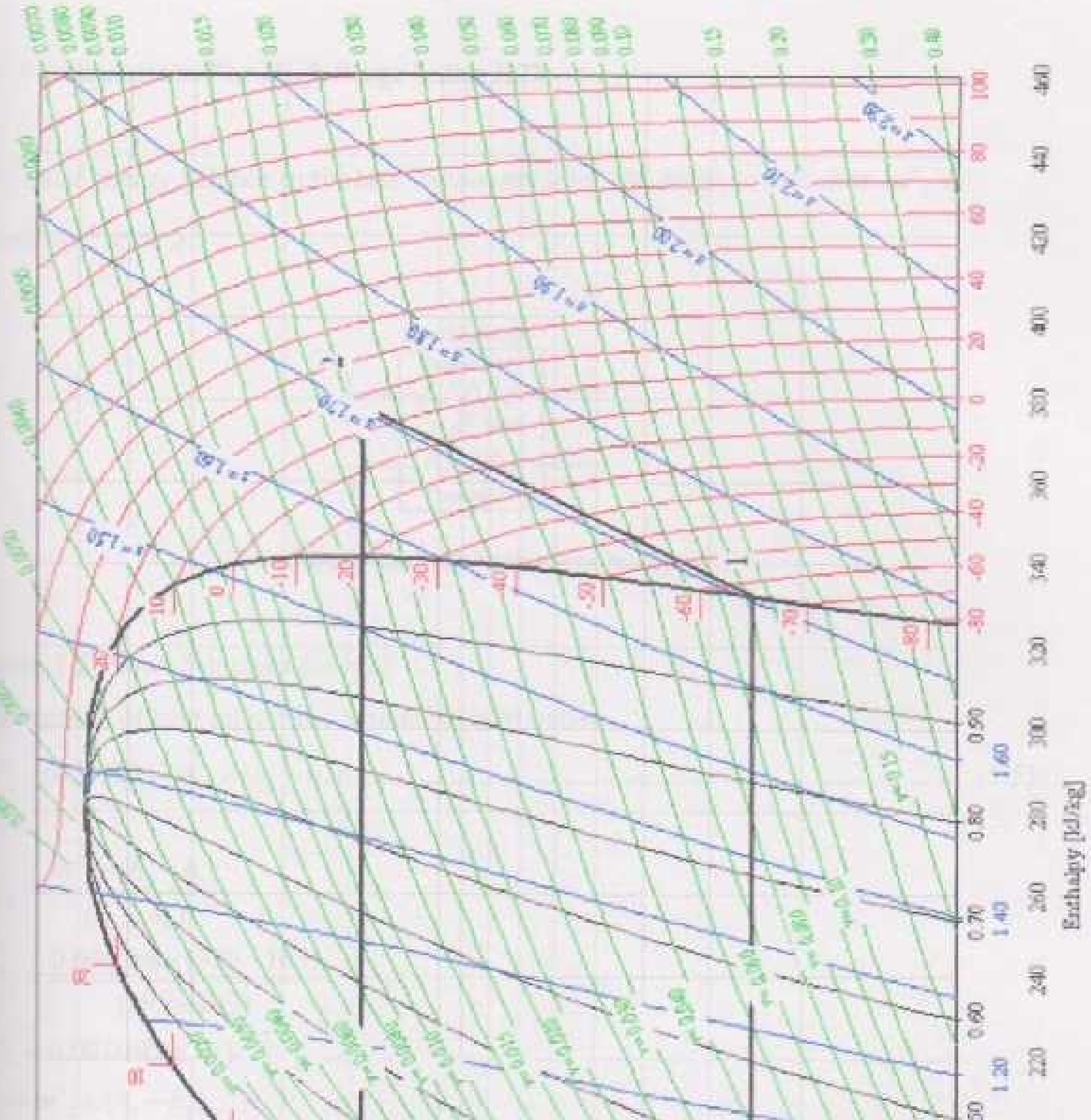


Figure 5.2 P-h chart for R23

5.2.2 Calculations For High Stage Using R22

We made energy balance in the heat exchanger inlet and outlet to calculate $m^{\bullet}_2$ as shown in Figure 5.3

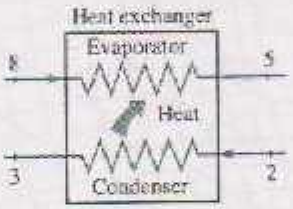


Figure 5.3 heat exchanger balance

Figure 5.4 shows P-h chart for R22

The amount of heat inlet = the amount of heat outlet

$$m^{\bullet}_2 \times (h_5 - h_8) = m^{\bullet}_1 \times (h_2 - h_3) \quad (5.5)$$

So

$$m^{\bullet}_2 = \frac{m^{\bullet}_1 \times (h_2 - h_3)}{(h_5 - h_8)}$$

$$m^{\bullet}_2 = \frac{0.000768 \times (370 - 165)}{(395 - 243)}$$

$$m^{\bullet}_2 = 0.00104[kg/s]$$

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

$$Q_c = 0.00104 \times (444.16 - 243.37)$$

$$Q_c = 208[W]$$

$$W_A = m^{\bullet}_2 \times (h_6 - h_5)$$

$$W_A = 0.00104 \times (444.16 - 395.45)$$

$$W_A = 50.6[W]$$

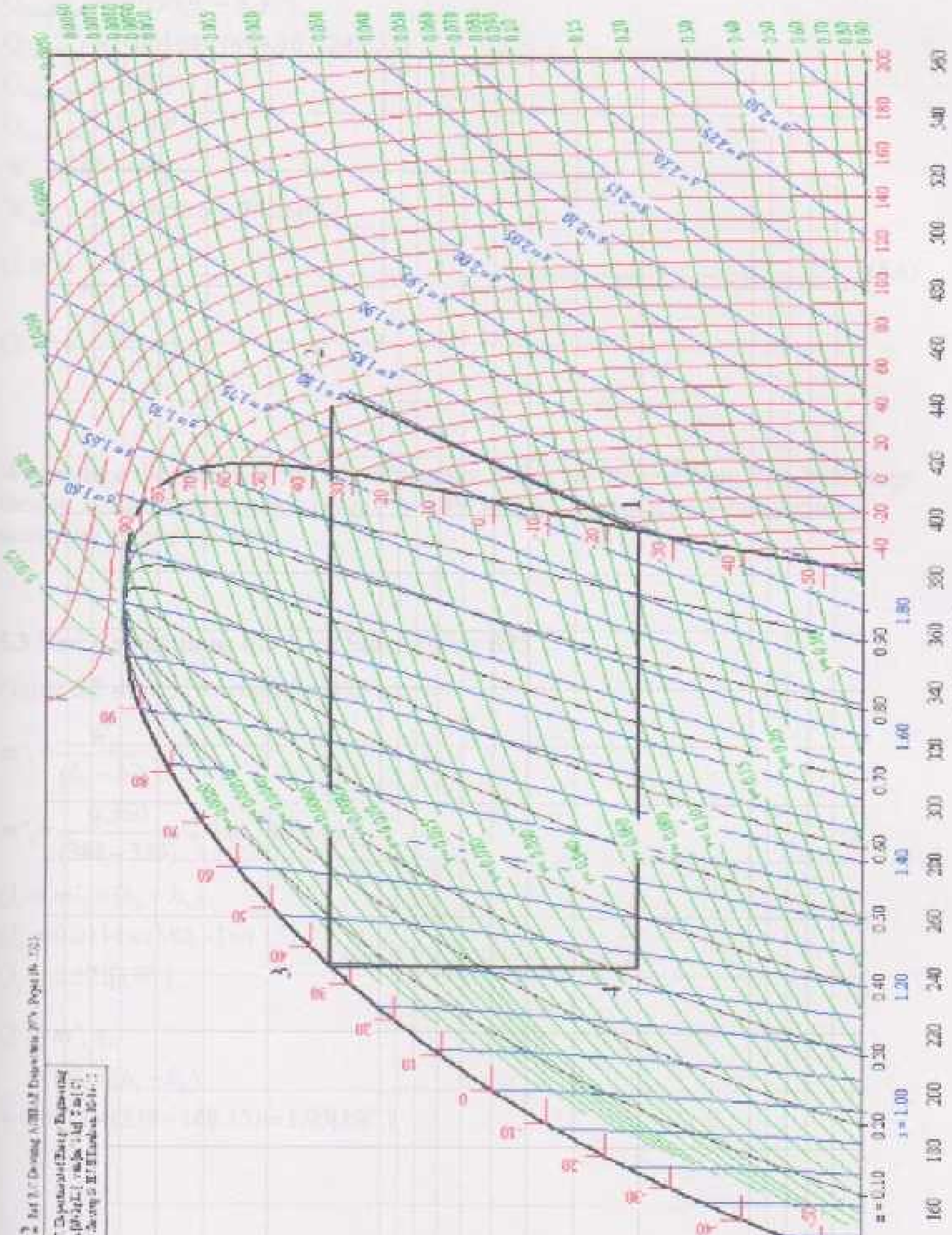


Figure 5.4 P-h chart for R22

5.2.3 Calculation Of Coefficient Of Performance

$$Q_{condenser} = m^{\bullet}_{2} \times (h_6 - h_7)$$

$$Q_{condenser} = 0.00104 \times (444.16 - 243.37)$$

$$Q_{condenser} = 208[W]$$

$$Q_{evaporator} = 133[W]$$

$$W_{total} = W_A + W_B$$

$$W_{total} = 24.56 + 50.6 = 75.16[W]$$

$$COP = \frac{Qe}{W_A + W_B} \qquad (5.6)$$

$$COP = \frac{133}{75.16} = 1.77$$

We changed the design to suit the refrigerator which we brought from Beit-Jalla drugs factory compressor power is given –3/4HP–560[W], we made new calculation according to given power.

5.3 New Calculations For Low Stage Using R23

Figure 5.5 shows P-h chart for R23

$$m^{\bullet}_{1} = \frac{W_B}{(h_2 - h_1)}$$

$$m^{\bullet}_{1} = \frac{0.560}{(388 - 339)} = 0.0114[kg/s]$$

$$Q_c = m^{\bullet}_{1} \times (h_2 - h_4)$$

$$Q_c = 0.0114 \times (388 - 168.15)$$

$$Q_c = 2.521[kW]$$

$$Q_e = m^{\bullet}_{1} q_e$$

$$= m^{\bullet}_{1} \times (h_1 - h_4)$$

$$= 0.0114 \times (339 - 168.15) = 1.95[kW]$$

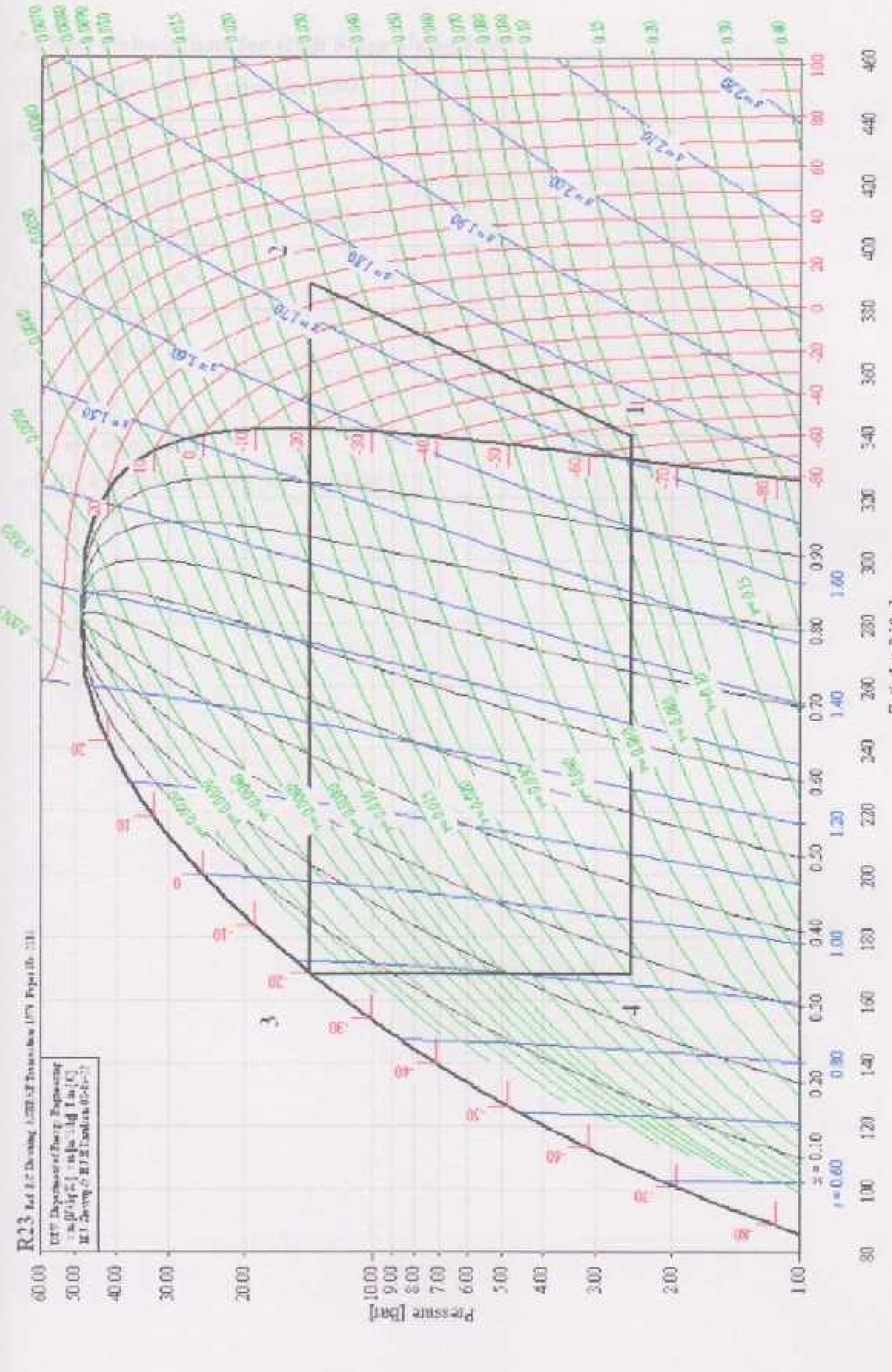


Figure 5.5 P-h chart for R23

5.4 New Calculations for High Stage Using R507

Figure 5.6 shows P-h chart for R507

$$m^{\bullet}_{1}=\frac{W_{B}}{(h_{2}-h_{1})}$$

$$m^{\bullet}_{1}=\frac{0.560}{(399.8-359.52)}=0.01402[kg/s]$$

$$Q_{c}=m^{\bullet}_{1}\times(h_{2}-h_{3})$$

$$Q_{c}=0.0114\times(399.8-244.55)$$

$$Q_{c}=2.173[kW]$$

$$Q_{e}=m^{\bullet}_{1}q_{e}$$

$$=m^{\bullet}_{1}\times(h_{1}-h_{4})$$

$$=0.0114\times(359.52-245.01)=1.616[kW]$$

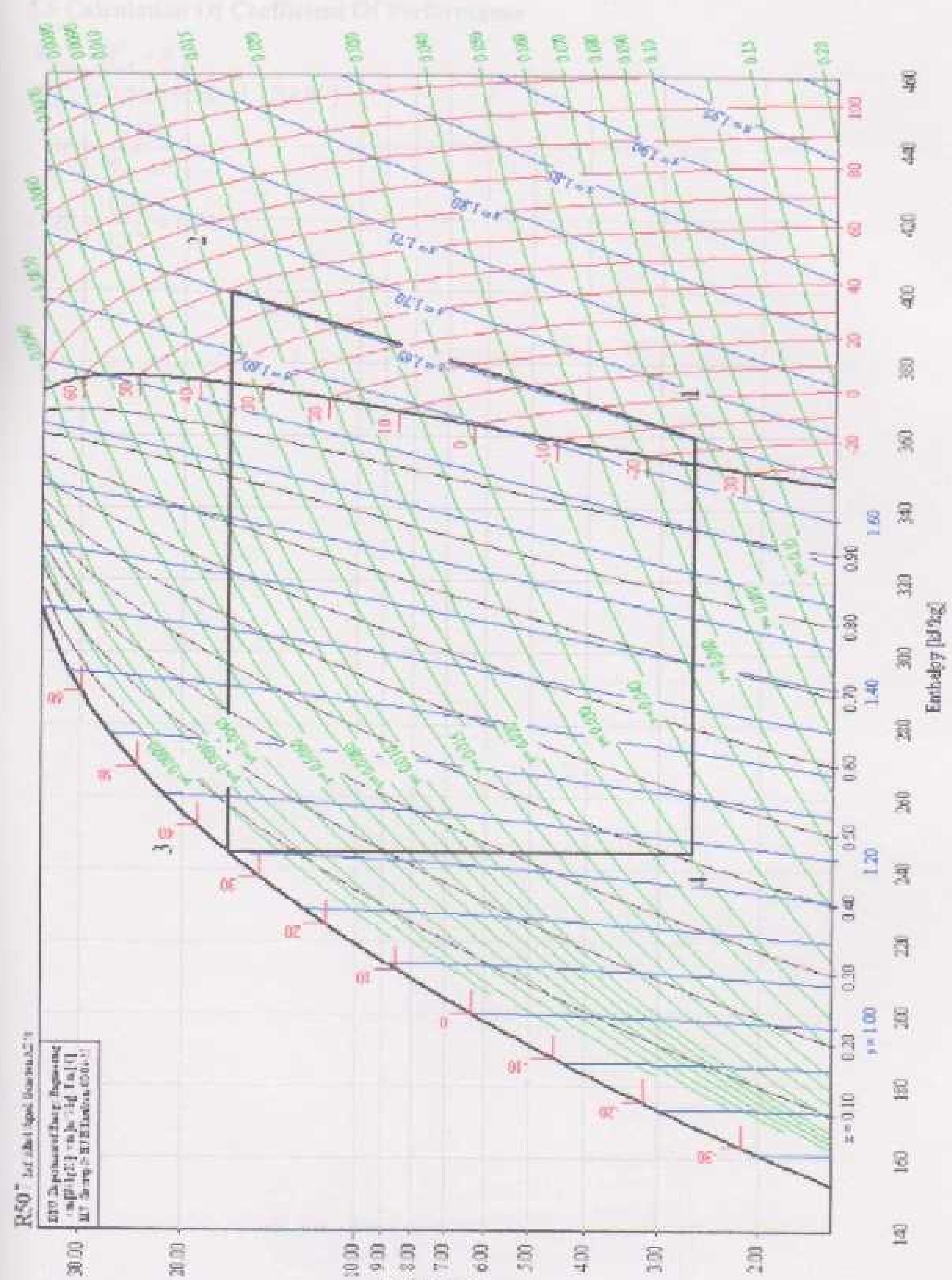


Figure5.6 P-h chart for R507

5.5 Calculation Of Coefficient Of Performance

$$W_{total} = W_A + W_B$$

$$W_{total} = 0.56 + 0.56 = 1.12[kW]$$

$$COP = \frac{Qe}{W_A + W_B}$$

$$COP = \frac{1.95}{1.12} = 1.74$$

CHAPTER SIX

MECHANICAL DESIGN FOR CYCLE

- Pipe Design
- Condenser Design
- Evaporator Design
- Heat Exchanger Design
- Capillary Tube design
- Oil Separator Selection

6.1 Introduction:

Design is an innovative and highly iterative process. It is also decision making process. Decisions sometimes have to be made with too little information. Decisions are sometimes made tentatively, reserving the right to adjust as more becomes known. The point is that the engineering designer has to be personally comfortable with a decision making. It should be a satisfying and welcomed activity. In this chapter we designed the mechanical cycle components, and we satisfy our results.

- Pipe design
- Condenser design
- Evaporator design
- Heat exchanger design
- Capillary tube
- Oil separator selection

6.2 Pipe Design:

Pipe design is very important to avoid explosion in pipe because of pressure and to avoid a noising sound produced from refrigerant flow.
We designed the pipes based on volume flow rate, refrigerant velocity and inside pressure.

6.2.1 Pipe Design For R23

- Suction pipe design
  - Inner diameter calculations

$m = 0.0114[kg/s]$

$P_{kor} = 2.5\times10^{5}[Pa]$

$V = 10[m/s]$

$V = m\times\upsilon$ ....(6.1)

where

V : *Volume flow rate* $[m^{3}/s]$

m : *Mass flow rate* $[kg/s]$

υ : *Specific volume* $[m^{3}/kg]$

$V = 0.0114\times0.092 = 0.001122[m^{3}/s]$

From continuity equation the inner diameter calculated is

$V = A_{in}\times V$ ....(6.2)

A_{in} : Inner area for the pipe$[m^2]$

V : Velocity of the refrigerant in the pipe[m/s]

$$A_{in} = \frac{V}{V}$$

$$A_{in} = \frac{0.001122}{10} = 1.12\times10^{-4}[m^{2}]$$

Then :

$$D_{in} = \sqrt{\frac{4A\,in}{\pi}}$$

$$D_{in} = \sqrt{\frac{4\times1.12\times10^{-4}}{\pi}} = 1.2[cm]$$

➤ Outer diameter calculations

$$\sigma_t = \frac{P_{in}(r_0^2 + r_{in}^2)}{r_0^2 - r_{in}^2} \text{...}(6.3)$$

$$\sigma_t^2 + P_{in} \times \sigma_t + P_{in}^2 = \frac{S_y^2}{n^2} \text{...}(6.4)$$

$where:$

σ_t : Tangential stress[Pa]

P_{in} : Inside pressure[Pa]

S_y : Yield strength[Pa]

n: Factor of safty(we take it 8 From Mechanical Engineerig Design Book)

r_0 : Outer radius[m]

r_{in} : inner radius[m]

$$\sigma_t^2 + 2.5\times10^5\sigma_t + (2.5\times10^5)^2 = \frac{(70\times10^6)^2}{8^2}$$

$$\sigma_t = 8.871[MPa]$$

$$8.871\times10^6 = \frac{2.5\times10^5(r_0^2 + 0.006^2)}{r_0^2 - 0.006^2}$$

$$r_0 = 0.6172[cm]$$

$$D_0 = 1.234[cm]$$

$$t = D_0 - D_{in}$$

$$t = 1.234 - 1.2 = 0.034[cm]$$

- Discharge pipe design
  - Inner diameter calculations

$P_w = 14.14[bar]$

$V = 15[m/s]$

$A_w = \frac{0.001122}{15} = 7.48 \times 10^{-5}[m^2]$

$D_w = \sqrt{\frac{4 \times 7.48 \times 10^{-5}}{\pi}} = 0.97[cm]$

  - Outer diameter calculations

$\sigma_t^2 + 14.14 \times 10^5 \sigma_t + (14.14 \times 10^5)^2 = \frac{(70 \times 10^6)^2}{8^2}$

$\sigma_t = 7.96[MPa]$

$7.96 \times 10^6 = \frac{14.14 \times 10^5 (r_0^2 + 0.485^2)}{r_0^2 - 0.485^2}$

$r_0 = 0.58[cm]$

$D_0 = 1.16[cm]$

$t = 1.16 - 0.97 = 0.19[cm]$

6.2.2 Pipe Design For R507

- Suction pipe design
  - Inner diameter calculations

$P_{in} = 2.6[bar]$

$V = 0.01402 \times 0.077 = 1.081 \times 10^{-3}[m^3/s]$

$V = 10[m/s]$

$$A_{in} = \frac{1.081\times10^{-3}}{10} = 1.081\times10^{-4}[m^2]$$

$$D_{in} = \sqrt{\frac{4\times1.081\times10^{-4}}{\pi}} = 1.17[cm]$$

➢ Outer diameter calculations

$$\sigma_t^2 + 2.6\times10^5\sigma_t + (2.6\times10^5) = \frac{(70\times10^6)^2}{8^2}$$

$$\sigma_t = 7.447[MPa]$$

$$7.447\times10^6 = \frac{2.6\times10^5(r_0^2+0.585^2)}{r_0^2-0.585^2}$$

$$r_0 = 0.61[cm]$$

$$D_0 = 1.22[cm]$$

$$t = 1.22-1.17 = 0.05[cm]$$

- Discharge pipe design

➢ Inner diameter calculations

$$P_{in} = 16.6[bar]$$

$$\dot{V} = 1.081\times10^{-3}[m^3/s]$$

$$V = 15[m/s]$$

$$A_{in} = \frac{1.081\times10^{-3}}{15} = 7.21\times10^{-5}[m^2]$$

$$D_{in} = \sqrt{\frac{4\times7.21\times10^{-5}}{\pi}} = 0.96[cm]$$

- Outer diameter calculations

$\sigma_t = 7.447[MPa]$

$$7.8\times10^{6}=\frac{16.6\times10^{5}(r_0^2+0.48^2)}{r_0^2-0.48^2}$$

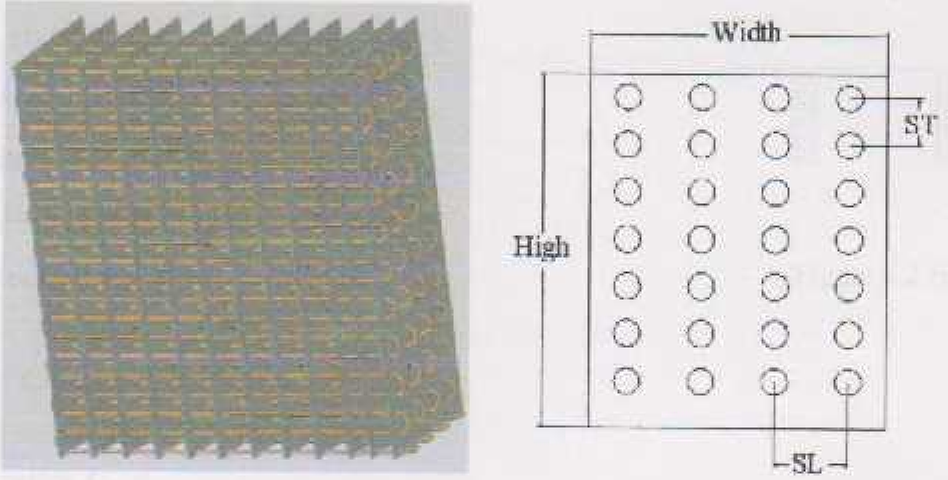
$r_0 = 0.6[cm]$

$D_0 = 1.2[cm]$

$t = 1.2-0.96 = 0.24[cm]$

6.3 Condenser Design

Figure 6.1a shows the condenser in 3D and 6.1b shows the side view of condenser



(a) (b)

Figure 6.1 condenser

Condenser length L_c =20 [cm]

Condenser height H_c =18 [cm]

Condenser width W_c =15[cm]

S_T: Transverse tube spacing [m]

S_L: Longitudinal tube spacing [m]

$$S_T = \frac{condenser\ height}{number\ of\ rows} = \frac{0.18}{7} = 0.024\ [m]$$

$$S_L = \frac{condenser\ width}{number\ of\ column} = \frac{0.15}{4} = 0.0375[m]$$

Figure 6.2 shows the fins

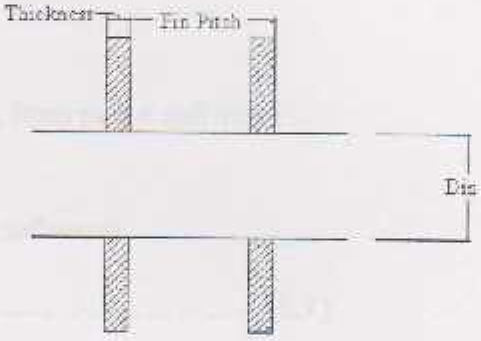


Figure 6.2 fins

$fin\ length = L_f = (S_T - D_o)$

$L_f = (0.024 - 0.006) = 0.018[m]$

$Fin\ width = W_f = (S_L - D_o)$

$W_f = (0.0375 - 0.006) = 0.0315[m]$

$Fin\ thickness = t_f = 0.2[mm]$

Fin pitch $= 2[mm]$

$bare\ tube\ length\ L_{bare\ tube} = P_f - t_f$

$L_{bare\ tube} = 2 - 0.2 = 1.8[mm]$

- Condenser diameter calculation

$$\dot{m} = \rho \times V \times A_{in}$$

$$\rho_{gound\ m} = 33.2[kg/m^3]$$

$$A_{in} = \frac{0.01402}{15 \times 33.2} = 2.81 \times 10^{-5}[m^2]$$

Then :

$$D_{in} = \sqrt{\frac{4A\dot{m}}{\pi}}$$

$$D_{in} = \sqrt{\frac{4 \times 2.81 \times 10^{-5}}{\pi}} = 6[mm]$$

From copper tube handbook we calculated outer diameter D_o=8.3[mm]

- Air side heat transfer coefficient

$$h_a = \frac{j \times C_p \times G_a}{Pr^{2/3}} \qquad (6.6)$$

Where:

h_a : air heat transfer coefficient [kJ/ kg]

j : Colburn factor

C_p: air specific heat=1.0076 [kJ/ kg K](@41. 3 °C and 1 bar, from peace software)

G_a : air mass flux [kg /sm^2]

P_r:Prantdl number =0.711(@41.3 °C and 1 bar, from peace software)

$$G_a = \frac{\dot{m}_a}{A_{flux}} \qquad (6.7)$$

Where:

A_{flux} : Air flux area [m^2].

We take control volume to calculate air mass flow rate.

$$\dot{m}_a = \frac{\dot{m}_r(h_{r,in} - h_{r,out})}{h_{a,out} - h_{a,in}} \qquad (6.8)$$

Where:

$\dot{m}_a$:Air mass flow rate [kg/ s]

$\dot{m}_r$: Refrigerant mass flow rate [kg/ s]

$h_{r,out}$: Refrigerant enthalpy at the outlet of the condenser [kJ/ kg]

$h_{r,in}$: Refrigerant enthalpy at the inlet of the condenser [kJ/ kg]

$h_{a,in}$: Air enthalpy at the inlet of the fins [kJ/kg]

$h_{a,out}$: Air enthalpy at the outlet of the fins [kJ/kg]

We used Psychrometric chart software to calculate air enthalpy

$h_{a,in} = 128.5[kJ/kg]$ $(at\ 35\ °C dbt\ \&\ 30°C wbt)$

$h_{a,out} = 165.5[kJ/kg]$ $(at\ 48°C\ dbt\ \&\ 43°C wbt)$

$$\dot{m}_a = \frac{2.172}{165.5 - 128.5} = 0.0587[\text{kg/s}]$$

$$A_{\text{flux}} = L_C \times H_C = 0.20 \times 0.18 = 0.036[m^2]$$

Then:

$$G_a = \frac{0.0587}{0.036} = 1.63[kg/m^2s]$$

$$j = 0.24Re^{-0.409}\left(S_T/_{S_L}\right)^{0.425}\left(L_{bare\ \ tube}/_{D_0}\right)^{-0.035} \quad \ldots\ldots\ldots\ldots\ldots\ldots (6.9)$$

Re: Reynolds number

$$R_e = \frac{\rho_a \times v \times D_h}{\mu_a} \quad \ldots\ldots\ldots\ldots\ldots\ldots (6.10)$$

Where:

D_h: hydraulic diameter[m]

μ_a: Dynamic viscosity of air =19.33× $10^{-6}[kg/ms]$ (@41. 3 °C and 1 bar, from peace software)

$$D_h = \frac{2(L_{bare\ \ tube} \times L_f)}{(L_{bare\ \ tube} + L_f)} \quad \ldots\ldots\ldots\ldots\ldots\ldots (6.11)$$

$$D_h = \frac{2(1.8 \times 10^{-3} \times 0.018)}{(1.8 \times 10^{-3} + 0.018)} = 3.3 \times 10^{-3}[m]$$

$$V = \frac{\rho \times A_{\text{flux}}}{\dot{m}_a} \quad \ldots\ldots\ldots\ldots\ldots\ldots (6.12)$$

Where:

$\dot{m}_a$: Air mass flow rate [m³/s]

ρ_a: Air density [Kg/m³]

ρ_a=1.1032[Kg/m³](@41.5 °C and 1 bar from peace software)

V: Air velocity[m/s]

Then:

$$V = \frac{1.1023 \times .2 \times .18}{0.0587} = 0.644[m/s]$$

Then:

$$R_e = \frac{1.1032 \times 3.3 \times 10^{-3} \times 0.644}{19.3368 \times 10^{-6}} = 276.6$$

Then the Colburn factor is equal

$$j = 0.24 * 276.6^{-0.409}\left(^{0.024}/_{0.0375}\right)^{0.425}\left(^{1.8}/_{8.3}\right)^{-0.035} = 0.028$$

Then air convection heat transfer coefficient equal

$$h_a = \frac{0.028 \times 1.0076 \times 1.63 \times 1000}{0.711868^{2/3}} = 58[W/m^2C]$$

We take a part from a condenser and we made our calculations for this part to specify number of parts. The part consist of one plate fin and one bare tube. So the calculations as follow:

- Calculations for plate fin

$$q_{fin} = h_a \times A_{fin} \times (T_b - T_\infty) \qquad (6.13)$$

Where:

A_{fin} : Plate fin area[m^2]

T_b: Pipe wall temperature at inlet of condenser 58[°C]

T_∞: Surrounding surface temperature 48[°C]

$$A_{fin} = [(W_C \times L_C) - (A_{p,out} \times N_{pip})] \qquad (6.14)$$

Where:

$A_{p,out}$:Outer cross section area condenser pipe [m^2]

N_{pip}: number of pip

$$A_{fin} = 2[(0.18 * 0.15) - (\pi(6 \times 10^{-3})^2 \times 28)] = 0.0532[m^2]$$

$$q_{fin} = 58 \times 0.0532 \times 10 = 30.85[W]$$

- Calculations for bare tube

$$q_{un\ fin} = h_a \times A_{bare\ part} \times (T_b - T_{so}) \quad \ldots\ldots (6.15)$$

Where:

$A_{bare\ tube}$: Bare tube heat transfer area[m^2]

$$A_{bare\ tube} = N_{pip} \times D_0 \times \pi \times L_{bare\ tube} \quad \ldots\ldots (6.16)$$

$$A_{bare\ part} = 28 \times 8.3 \times 10^{-3} \times \pi \times 1.8 \times 10^{-3} = 1.314 \times 10^{-3} [m^2]$$

$$q_{un\ fin} = 58 \times 1.314 \times 10^{-3} \times 10 = 0.762 [W]$$

$$q_{Total} = q_{un\ fin} + q_{fin} \quad \ldots\ldots (6.17)$$

$$q_{Total} = 0.762 + 30.85 = 31.6 [W]$$

$$N_{Part} = \frac{Q_c}{q_{Total}} \quad \ldots\ldots (6.18)$$

$$N_{Part} = \frac{2173}{31.6} = 72\ Part$$

- Fin efficiency calculation:

$$\eta_f = \frac{\tanh(m\acute{L}_f)}{m\acute{L}_f} \quad \ldots\ldots (6.19)$$

$$\eta_f = \frac{\tanh(m\acute{L}_f)}{m\acute{L}_f}$$

$$\acute{L}_f = \left(\frac{L_f}{2}\right)\left[1 + 0.35 \ln \frac{(\frac{D_0}{2} + \frac{L_f}{2})}{\frac{D_0}{2}}\right] \quad \ldots\ldots (6.20)$$

$$\acute{L}_f = \left(\frac{0.018}{2}\right)\left[1 + 0.35 \ln \frac{(\frac{0.0083}{2} + \frac{0.018}{2})}{\frac{0.0083}{2}}\right] = 0.0126 [m]$$

$$m = \sqrt{\frac{2h}{kt_f}} \quad \ldots (6.21)$$

$$m = \sqrt{\frac{2 \times 58}{202 \times 0.2 \times 10^{-3}}} = 53.58$$

$$\eta_f = \frac{\tanh(53.58 \times 0.0126)}{53.58 \times 0.0126} = 0.74$$

- Overall efficiency calculation:

$$\eta_o = 1 - \frac{A_{fin,T}}{A_{total}}(1 - \eta_f) \quad \ldots (6.22)$$

$$A_{total} = A_{fin,T} + A_{unfin,T} + A_{Filt,T} \quad \ldots (6.23)$$

$$A_{fin,T} = A_{fin} \times N_{part}$$

$$A_{fin,T} = 0.0532 \times 72 = 4.83[m^2]$$

$$A_{unfin,T} = A_{unfin} \times N_{part}$$

$$A_{unfin,T} = 1.314 \times 10^{-3} \times 72 = 0.0946[m^2]$$

$$A_{Filt,T} = \pi\left(\frac{D_0 + S_T}{2}\right) + .02 \times \pi D_0 \times N_{Filt} \quad \ldots (6.24)$$

$$A_{Filt,T} = \pi\left(\frac{8.3 \times 10^{-3} + 0.024}{2}\right) + .02 \times \pi \times 8.3 \times 10^{-3} \times 27 = 0.065[m^2]$$

$$A_{total} = 4.83 + 0.0946 + 0.06 = 5[m^2]$$

$$\eta_o = 1 - \frac{3.83}{5}(1 - 0.74) = 0.80$$

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

➢ One phase calculation

$$U_{sp}A_{sp} = \frac{Q_{sp}}{LMTD_{sp}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (6.25)$$

By using LMTD for one phase

$$LMTD_{sp} = \frac{(T_{ref,in} - T_{a,out}) - (T_{ref,out} - T_{a,in})}{\ln\frac{(T_{ref,in} - T_{a,out})}{(T_{ref,out} - T_{a,in})}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (6.26)$$

$$LMTD_{sp} = \frac{(58-48)-(35-32)}{\ln\frac{(58-48)}{(35-32)}} = 5.814[°C]$$

$$Q_{sp} = \dot{m}_r(h_2 - h_3) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (6.27)$$

$$Q_{sp} = 0.01402(399.35 - 374) = 0.335[\text{kW}]$$

$$A_{sp} = A_{total}\left(\frac{Q_{sp}}{Q_{total}}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (6.28)$$

Where:

A_{sp}: Single phase heat transfer surface area$[m^2]$

A_{total} : Total heat transfer surface area$[m^2]$

Q_{sp}: Single phase heat transfer rate[kW]

Q_{total} : Total heat transfer rate (condenser load)[kW]

$$A_{sp} = 5\left(\frac{0.355}{2.173}\right) = 0.8168[m^2]$$

So

$$U_{sp}A_{sp} = \frac{0.355}{5.814} = 0.061[kW/°C]$$

$$U_{sp} = \frac{U_{sp}\,A_{sp}}{A_{sp}}$$

$$U_{sp} = \frac{0.061 \times 1000}{0.8168} = 74[W/m^{2}°C]$$

1. Air side resistance calculations:

$$R_1 = \frac{1}{h_a \times \eta_0} \quad \text{(6.29)}$$

$$R_1 = \frac{1}{58 \times 0.80} = 0.0168\ [m^2{}^\circ C/W]$$

2. Air side fouling resistance

$$R_2 = \frac{1}{h_{f,a}\eta_0} \quad \text{(6.30)}$$

Where:

$h_{f,a}$: Fouling factor for air $[W/m^2C]$ =6500 (From Chemical Engineering Book)

$$R_2 = \frac{1}{6500 \times 0.80} = 1.923 \times 10^{-4} [m^2{}^\circ C/W]$$

3. Wall resistance

$$R_3 = \frac{A_{sp,0}(\ln\frac{D_0}{D_i})}{2\pi K_W L_{sp}} \quad \text{(6.31)}$$

$A_{sp,o}$: Outside Single phase heat transfer surface area$[m^2]$

D_0: Outside diameter of condenser pipe[mm]

D_i: Inside diameter of condenser pipe[mm]

K_W: Thermal conductivity of pipe material[W/m·C]

L_{sp}: Single phase pipe length[m]

$$R_3 = \frac{0.8168(\ln\frac{8.3}{6})}{2\pi \times 202 \times L_{sp}} = \frac{2.0823 \times 10^{-4}}{L_{sp}} [m^2{}^\circ C/W]$$

4. Fin tube contact resistance

$$R_4 = \frac{1}{h_c(\frac{A_{sp,bare\ tube}}{A_{sp}})} \quad (6.32)$$

Where:

h_c: Fin-tube contact conductance [W /m^2·C]= 4×10^5 (From Heat Transfer Book)

$A_{sp,bare\ tube}$: Single phase bare tube surface area$[m^2]$

$$A_{sp,bare\ tube} = (A_{total} - A_{fins})\left(\frac{\dot{Q_{sp}}}{\dot{Q}_{total}}\right) \quad (6.33)$$

$$A_{sp,bare\ tube} = (5 - 4.83)(\frac{0.355}{2.173}) = 0.208[m^2]$$

$$R_4 = \frac{1}{4 \times 10^5(\frac{0.208}{0.8168})} = 9.81 \times 10^{-6}[m^2{}^{\circ}C/W]$$

5. Refrigerant side fouling resistance

$$R_5 = \frac{1}{h_{f,ref}\left(\frac{A_{sp,i}}{A_{sp}}\right)} \quad (6.34)$$

Where:

$h_{f,ref}$: Air-side fouling conductance [W /m^2 ·C]=5000 (From Chemical Engineering Book)

$$R_5 = \frac{1}{5000\left(\frac{\pi \times 6 \times 10^{-3} \times L_{sp}}{0.8168}\right)} = \frac{8.66 \times 10^{-3}}{L_{sp}}[m^2c/W]$$

6. Internal resistance

$$R_6 = \frac{1}{h_{ref}(\frac{A_{sp,i}}{A_{sp}})} \quad (6.35)$$

Where:

h_{ref} : One phase convection heat transfer coefficient$[W/m^{2}{}^\circ C]$

- ➢ One phase conviction heat transfer coefficient calculation:
- ▪ Pipe wall Temperature $T_w = 42.9[^\circ C]$
- ▪ The reference temperature of the refrigerant is calculated by the following correlation :

$$T_{refernce} = \frac{T_w + T_{average}}{2} \qquad (6.36)$$

$$T_{average} = \left(\frac{T_{cond,i} + T_{cond,o}}{2}\right) \qquad (3.37)$$

$$T_{average} = \frac{58 + 35}{2} = 46.5^\circ C$$

Then,

$$T_{refernce} = \frac{42.9 + 46.5}{2} = 44.7^\circ C$$

Reynolds number calculation inside condenser pipe at reference temperature.

$$Re_{ref} = \frac{\rho_{ref} \times V_{ref} \times D_{ref}}{\mu_{ref}} \qquad (6.38)$$

Re_{ref} : Reynolds number at reference temperature for the refrigerant in pipe.

D_{ref} : Condenser pipe diameter =6[mm]

μ_{ref} : Viscosity of the refrigerant R507 in the refrigerant pipe at $T_{refernce}$ $[kg/ms]$

ρ_{ref} : Densityf of the refrigerant R507 in the refrigerant pipe at $T_{referrnce}$ $[m^3/kg]$

- ✓ All properties we take it from cool pack software at $T_{refernce}$ and 16.6 bar

$$Re_{ref} = \frac{95.4 \times 15 \times 0.006}{1.47 \times 10^{-5}} = 343662$$

Reynolds number at wall temperature for the refrigerant that flows inside the refrigerant pipe is calculated by the following equation:

$$Re_{refw} = \frac{\rho_{refw} \times u_{ref} \times d_{ref}}{\mu_{refw}}$$

Re_{refw} : Reynolds number calculation inside condenser pipe at wall temperature

d_{ref} : Condenser pipe diameter [m].

μ_{refw} : Refrigerant R507 Viscosity at T_{wall} $[\frac{kg}{(m.s)}]$.

By using Peace Software the viscosity and value of the carbon dioxide refrigerant that flows inside the refrigerant pipe at T_{wall} and 16.6 bar equal:

$$\mu_{refw} = 1.474 \times 10^{-5} \, [Kg/m.s]$$

$$\rho_{refw} = 94.9 [Kg/m^3]$$

So Reynolds number at wall temperature equal:

$$Re_{refw} = \frac{94.9 \times 15 \times 0.00035}{1.474 \times 10^{-5}} = 338008$$

Calculation of Nusselt number for the refrigerant inside pipe to estimate Nusselt number for the refrigerant.

$$Nu_r = \left(\frac{Nu_{rw} + Nu_{rb}}{2}\right)\frac{K_{rw}}{K_{rb}} \quad ,$$

where:

Nu_r : Refrigerant Nusselt number for the refrigerant.

Nu_{rb} : Refrigerant Nusselt number at bulk temperature.

Nu_{rw} : Refrigerant Nusselt number at wall temperature.

k_{rb} : Refrigerant thermal conductivity at reference (bulk) temperature [W/m.K].

k_{rw} : Refrigerant thermal conductivity at wall temperature [W/(m.K)].

D_{ref} : Diameter of refrigerant pipe [m].

1. To find Nusselt number at wall temperature for the refrigerant that flows inside the refrigerant pipe, the following equation is to be used:

$$Nu_{rw} = \frac{(f_{rw}/8)(\mathrm{Re}_{refw}-100)\mathrm{Pr}_{refw}}{107+12.7(f_{rw}/8)^{1/2}(\mathrm{Pr}_{refw}{}^{2/3}-1)} \quad (6.39)$$

Where:

f_{rw} : Friction factor at wall temperature for the refrigerant in the refrigerant pipe dimensionless number.

Pr_{rw} : Prandtle number at wall temperature for the refrigerant in the refrigerant pipe dimensionless number.

- To find Friction factor at wall temperature for the refrigerant that flows inside the refrigerant pipe, the following equation is to be used:

$$f_{rw} = (0.79\ln(\mathrm{Re}_{rw}) - 1.64)^{-2} \quad (6.40)$$

If using the value of Reynolds number at wall temperature for the refrigerant in the refrigerant pipe which is calculated previously, the friction factor will be equal:

$f_{rw} = (0.79\ln(338008) - 1.64)^{-2} = 0.0114$

$\mathrm{Pr}_{refw} = 1.22$ This number exists within Gnielinski range of Prandtle number.

Then the nusselt number for the refrigerant at wall temperture equal:

$$Nu_{rw} = \frac{(0.0114/8)(388008-1000)1.04}{1.07+12.7(0.011/8)^{1/2}(1.22^{2/3}-1)} = 592$$

2. To find Nusselt number at reference temperature for the refrigerant inside pipe, the following equation is to be used:

$$Nu_{rp} = \frac{(f_{rb}/8)(\mathrm{Re}_{refb} - 100)\,\mathrm{Pr}_{refb}}{107 + 12.7(f_{rb}/8)^{1/2}(\mathrm{Pr}_{refb}{}^{2/3} - 1)}$$

Where:

f_{rp}: Friction factor at reference temperature for the refrigerant inside pipe

Pr_{rp} : Prandtle number at reference temperature for the refrigerant inside pipe.

- ➢ To find Friction factor at reference temperature for the refrigerant inside pipe, the following equation is to be used:

$$f_{rp} = (0.79 \ln Re_{rp} - 1.64)^2$$

If using the value of Reynolds number at reference temperature for the refrigerant inside pipe which is calculated previously, the friction factor will be equal:

$$f_{rb} = (0.79 \ln(343662) - 1.64)^{-2} = 0.0114$$

$\mathrm{Pr}_{refb} = 1.24$ This number exists within Gnielinski range of Prandtle number.

Then the nusselt number of the refrigerant at reference temperture equal:

$$Nu_{rp} = \frac{(0.0117/8)(343662 - 1000)1.24}{1.07 + (0.0116/8)^{1/2}(1.24^{2/3} - 1)} = 653$$

K_{rw} =0.0126 [W/m.°C] .

K_{rp} =0.0127[W/m.°C] .

Then Nusselt number for the refrigerant inside pipe. equal:

$$Nu_r = (\frac{Nu_{rw} + Nu_{rb}}{2})\frac{K_{rw}}{K_{rb}}$$,

$$Nu_r = (\frac{592 + 653}{2})\frac{0.0126}{0.0125} = 627$$

Calculation of convection heat transfer coefficient

$$\alpha_r = \frac{Nu_r}{d_{ed}} K_{th}$$

Where:

α_r : Convection heat transfer coefficient for the refrigerant [W/(m^2.K)].

$$\alpha_r = \frac{627}{0.0035} \times 0.0125 = 2242W/(m^2.s)$$

$$R_6 = \frac{1}{2243(\frac{\pi \times 6 \times 10^{-3} \times L_{sp}}{0.8168})} = \frac{0.0289}{L_{sp}} [m^2C/W]$$

$$U_{sp} = \frac{1}{R_T} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (6.41)$$

$$R_T = R_1 + R_2 + R_3 + R_4 + R_5 + R_6$$

$$R_T = 0.0168 + 1.923 \times 10^{-4} + 9.81 \times 10^{-6} + \frac{2.082 \times 10^{-4} + 8.66 \times 10^{-3} + .019}{L_{sp}}$$

$$R_T = 6.46 \times 10^{-4} + \frac{0.0244}{L_{sp}}$$

Then:

$$U_{sp} = \frac{0.017 L_{sp}}{0.017 \times 0.0278}$$

$$74 = \frac{0.017 L_{sp}}{0.017 \times 0.0278}$$

$$L_{sp} = 2.06[m]$$

➢ Two phase calculation:

$$U_{tp}A_{tp} = \frac{Q_{tp}}{LMTD_{tp}}$$

By using LMTD for two phase

$$LMTD_{tp} = \frac{(T_{a,out} - T_{ref,out}) - (T_{ref,out} - T_{a,in})}{\ln\frac{(T_{a,out} - T_{ref,out})}{(T_{ref,out} - T_{a,in})}}$$

$$LMTD_{tp} = \frac{(48-35)-(35-32)}{\ln\frac{(48-35)}{(35-32)}} = 6.814[°C]$$

$$Q_{tp} = \dot{m}_r(h_3 - h_4)$$

$$Q_{tp} = 0.01402(h_3 - h_4) = 1.815[kW]$$

$$A_{tp} = A_{total}\left(\frac{Q_{tp}}{Q_{total}}\right)$$

Where,

A_{tp} : Two phase heat transfer surface area$[m^2]$

Q_{tp} : Two phase heat transfer rate[kW]

Q_{total} : Total heat transfer rate (condenser load)[kW]

$$A_{tp} = 5\left(\frac{1.815}{2.173}\right) = 4.176[m^2]$$

So

$$U_{tp}A_{tp} = \frac{1.815}{4.17} = 0.435[kW/°C]$$

$$U_{tp} = \frac{U_{tp}\,A_{tp}}{A_{tp}}$$

$$U_{tp} = \frac{0.435\times1000}{4.176} = 63.7[W/m^{2}°C]$$

1. Air side resistance

Air side resistance in two phase is equal to air side resistance in single phase.

$$R_1 = \frac{1}{58 \times 0.80} = 0.0168\ [m^2{}^\circ C/W]$$

2. Air side fouling resistance

Air side fouling resistance in two phase is equal to air side fouling resistance in single phase.

$$R_2 = 1.923 \times 10^{-4} [m^2C/W]$$

3. Wall resistance

$$R_3 = \frac{A_{tp,o}(\ln \frac{D_0}{D_i})}{2\pi K_p L_{tp}}$$

A_{tp}: Two phase heat transfer surface area$[m^2]$

L_{tp}: Two phase pipe length[m]

$$R_3 = \frac{4.176(\ln \frac{8.3}{6})}{2\pi \times 338 \times L_{tp}} = \frac{1.067 \times 10^{-3}}{L_{tp}} [m^2{}^\circ C/W]$$

4. Fin tube contact resistance

$$R_4 = \frac{1}{h_c(\frac{A_{tp,bare\ tube}}{A_{tp}})}$$

Where:

$A_{tp,bare\ tube}$: Two phase bare tube surface area$[m^2]$

$$A_{tp,bare\ tube} = (A_{total} - A_{fins})(\frac{Q_{tp}}{Q_{total}})$$

$$A_{tp,bare\ tube} = (5 - (0.0532 \times 72))(\frac{1.185}{2.173}) = 1.0065[m^2]$$

$$R_4 = \frac{1}{4 \times 10^5 (\frac{1.0065}{4.176})} = 7.836 \times 10^{-4} [m^{2\circ}C/W]$$

5. Refrigerant side fouling resistance

$$R_5 = \frac{1}{h_{f,ref} \left(\frac{A_{tp,i}}{A_{tp}}\right)}$$

$$R_5 = \frac{1}{5000 \left(\frac{\pi \times 9.7 \times 10^{-3} \times L_{tp}}{5.864}\right)} = \frac{0.044}{L_{tp}} [m^{2\circ}C/W]$$

6. Internal resistance

$$R_6 = \frac{1}{\acute{h}_{ref} (\frac{A_{tp,i}}{A_{tp}})}$$

Where:

$\acute{h}_{ref}$: Two phase convection heat transfer coefficient$[W/m^{2\circ}C]$

- ➢ Two phase convection heat transfer coefficient calculation:

Calculation of Reynolds number for the refrigerant inside pipe.

Reynolds number at reference temperature for the refrigerant inside pipe is calculated by the following equation:

$$Re_{ref} = \frac{\rho_{ref} \times V_{ref} \times D_{ref}}{\mu_{ref}}$$

Re_{ref} : Reynolds number at reference temperature for the refrigerant inside pipe.

D_{ref} : Refrigerant pipe diameter [m].

μ_{ref}: Viscosity of the refrigerant inside pipe at $T_{refernce}$ [kg/ms].

- ✓ All properties we take it from Cool Pack Software at $T_{refernce}$ and 16.6 bar

$\rho_{ref} = 1112[kg/m^3]$

$\mu_{ref} = 0.000146[kg/ms]$

So Reynolds number at reference temperature equal:

$$Re_{ref} = \frac{1112 \times 15 \times 0.0083}{0.000146} = 118181$$

Calculation of convection heat transfer coefficient for the refrigerant in the refrigerant pipe

The convection heat transfer coefficient for the refrigerant that flows inside the refrigerant pipe is calculated by the following equation:

$\propto_{tp} = F \times \propto_L$

Where,

$\propto_{tp}$: Convection heat transfer coefficient for the refrigerant [W/(m^2.K)].

F: Dittus –Boelter factor

Then the convection heat transfer coefficient for the liquid refrigerant inside pipe equal:

$$\propto_L = 0.023 \frac{K_l}{D_i} Re_L{}^{0.8} Pr_L{}^{0.4}$$

Where:

k_L: Liquid conductivity[W/m·C]

k_L=0.0566[W/m·C]

$Pr_L = 1.25$

Then the convection heat transfer coefficient for the liquid refrigerant in the refrigerant pipe equal:

$$\propto_L = 0.023 \frac{0.0566}{0.0083} 118181^{0.8} 1.25^{0.4} = 15767[kJ/kg]$$

$F = 1 + 1.925 X_{tt}^{-0.83}$

Where:

X_{tt}: Lockhart Martinelli factor

$$X_{tt} = (\frac{1-x}{x})^{0.9}(\frac{\rho_v}{\rho_l})^{0.5}(\frac{\mu_l}{\mu_v})^{0.1} \quad \text{(6.42)}$$

Where:

x: Quality of saturated R507 liquid vapor mixture .

ρ_v: Vapor density[kg/m^3]

ρ_L: Liquid density[kg/m^3]

μ_v: Vapor dynamic viscosity[kg/ms]

μ_L: Liquid dynamic viscosity[kg/ms]

$x = 0.4$

$\rho_v = 94[kg/m^3]$

$\rho_L = 1112[kg/m^3]$

$\mu_v = 1.48 \times 10^{-5}$

$\mu_L = 0.000146$

$$X_{tt} = (\frac{1-0.4}{0.4})^{0.9}(\frac{94}{1112})^{0.5}(\frac{0.000146}{1.48 \times 10^{-5}})^{0.1} = 0.526$$

$$F = 1 + 1.925 \times 0.526^{-0.83} = 4.27$$

Then:

$\alpha_{tp} = 4.27 \times 15767 = 64436$[kJ/kg]

$$R_6 = \frac{1}{64436(\frac{\pi \times 6 \times 10^{-3} \times L_{sp}}{4.176})} = \frac{3.43 \times 10^{-3}}{L_{tp}}[m^2{}^{\circ}C/W]$$

$$R_T = 6.46 \times 10^{-4} + \frac{0.0244}{L_{tp}}$$

$$U_{tp} = \frac{1}{R_T}$$

$$R_T = R_1 + R_2 + R_3 + R_4 + R_5 + R_6$$

$$R_T = 6.46 \times 10^{-4} + \frac{0.0244}{L_{tp}}$$

Then:

$$U_{tp} = \frac{0.0169 L_{tp}}{0.0169 \times 0.0675}$$

$$63.7 = \frac{0.0169 L_{tp}}{0.0169 \times 0.0675}$$

$$L_{tp} = 4.3[m]$$

Total condenser length $= L_{tp} + L_{sp} = 4.3 + 2.06 = 6.36[m]$

6.4 Evaporator Design:

In the evaporator design we determine the heat transfer area and convert it to length

- Evaporator diameter calculation

$$\dot{m} = \rho \times V \times A_{in}$$

$$\rho_{@evap\ in} = 34.7[kg/m^3]$$

$$A_{in} = \frac{0.01147}{10 \times 34.7} = 3.3 \times 10^{-5}[m^2]$$

Then:

$$D_{in} = \sqrt{\frac{4A\,in}{\pi}}$$

$$D_{in} = \sqrt{\frac{4 \times 3.1 \times 10^{-5}}{\pi}} = 7.1[mm]$$

From copper tube handbook we calculated outer diameter D_o=8.1[mm]

We take different temperature between evaporator pipe wall and air is equal 10 [°C]

$$Q = UA\Delta T$$

$$UA = \frac{Q}{\Delta T}$$

$$UA = \frac{1961}{10} = 196.1[W/°C]$$

$$UA = \frac{1}{\frac{1}{\pi L \times D_i \times h_i} + \frac{\ln(D_o/D_i)}{2\pi kL} + \frac{1}{\pi L \times D_o \times h_a}}$$

$$UA = \frac{1}{\frac{1}{\pi L \times 7.1 \times 10^{-3} \times 64436} + \frac{\ln(8.1/7.1)}{2\pi \times 338 \times L} + \frac{1}{\pi L \times 8.1 \times 10^{-3} \times 188.532}}$$

$$196.1 = \frac{L}{0.209}$$

L=41.1[m]

6.5 Heat Exchanger design:

The prime objective in the design of an exchanger is to determine the surface area required for specified duty (rate of heat transfer) using the temperature differences variable. Then we determine the suited heat exchanger length. Figure 6.3 shows double pipe heat exchanger

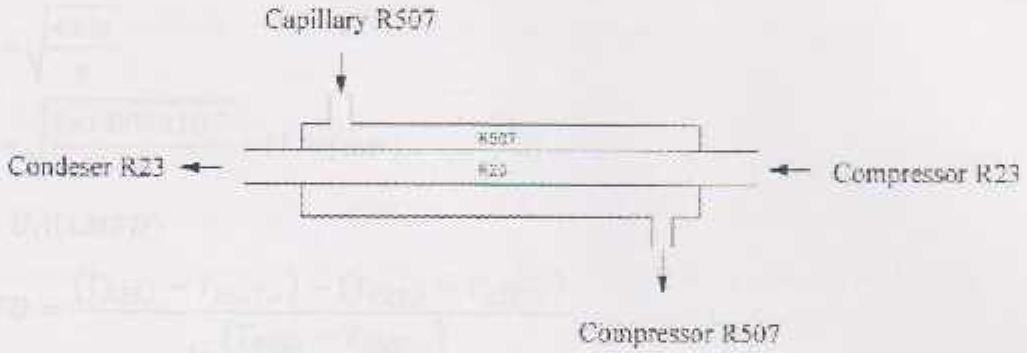


Figure 6.3 Double pipe heat exchanger

- Heat exchanger diameter calculation
- Inner diameter R23 calculation:

$$m = \rho \times V \times A_{in}$$

$$\rho_{@condR23\ in} = 44.247[kg/m^3]$$

$$A_{in} = \frac{0.0114}{15 \times 44.247} = 1.71 \times 10^{-5}[m^2]$$

Then :

$$D_{in} = \sqrt{\frac{4Ain}{\pi}}$$

$$D_{in} = \sqrt{\frac{4 \times 1.71 \times 10^{-5}}{\pi}} = 4.6[mm]$$

- Inner diameter R507 calculation:

$$m = \rho \times V \times A_{in}$$

$$\rho_{@evapR507\ in} = 13.9[kg/m^3]$$

$$A_{in} = \frac{0.01402}{10 \times 13.9} = 1.008 \times 10^{-4}[m^2]$$

Then :

$$D_{in} = \sqrt{\frac{4A\,in}{\pi}}$$

$$D_{in} = \sqrt{\frac{4\times 1.008\times 10^{-4}}{\pi}} = 11.46[mm]$$

$$Q = UA(LMTD)$$

$$LMTD = \frac{(T_{R23,i} - T_{R507,o}) - (T_{R23,o} - T_{R507,i})}{\ln\frac{(T_{R23,i} - T_{R507,o})}{(T_{R23,o} - T_{R507,i})}}$$

$$LMTD = \frac{(29-15) - (-20 - -25)}{\ln\frac{(29-15)}{(-20 - -25)}} = 17.93[C]$$

$$UA = \frac{Q}{(LMTD)}$$

$$UA = \frac{2.52}{17.93} = 146.5[W/m^2]$$

$$UA = \frac{1}{\frac{1}{\pi L\times D_{R23,i}\times h_{R23,i}} + \frac{\ln(D_{R507,i}/D_{R23,i})}{2\pi kL} + \frac{1}{\pi L\times D_{R507,i}\times h_{R507,i}}}$$

$$UA = \frac{1}{\frac{1}{\pi L\times 4.6\times 10^{-3}\times 10000} + \frac{\ln(11.46/4.6)}{2\pi kL} + \frac{1}{\pi L\times 11.46\times 10^{-3}\times 5000}}$$

$$UA = \frac{L}{0.010136}$$

$$L = 1.5[m]$$

6.6 Capillary Tube Selection:

We selected capillary tubes by using Dancap software

- For high stage we selected capillary tube with inner diameter 1.40[mm]

And 0.38[m] length

- For low stage we selected capillary tube with inner diameter 1.50[mm]

And 0.50[m] length

6.7 Oil Seprator Selection:

We selected type 900 series from temprato products, and has a volume 445[cm^3]

CHAPTER SEVEN

ELECRICAL DESIGN

- Introduction
- Types Of Electrical Circuits
- Components Of Electrical Circuits
- Electrical Description For Cascade Cycle
- Power Circuit
- Alarm And Monitoring Systems

7.1 Introduction:

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

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

7.2 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.

7.2.1 Control Circuit:

This circuit is working to influence the controls to follow up the implementation of required control program as defined by introducing elements operating according to the requirements of control thermostat and unequivocal pressure and break convection. Also working to introduce elements of the capacity as the exact timing advance. Often control circuit is working with single phase. and potential voltage in control circuit less or equal in power circuit. The energy consumed to control much less of the energy power circuit.

7.2.2 Power Circuit:

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

7.3 Components Of Electrical Circuits:

7.3.1 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 7.1 shows Current relay.

Figure 7.1Current relay

7.3.2 Potential (Voltage) Relay:

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

The relay has 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 7.2 shows voltage relay.

Figure 7.2 Voltage Relay

7.3.3 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 7.3 shows run and start Capacitors.

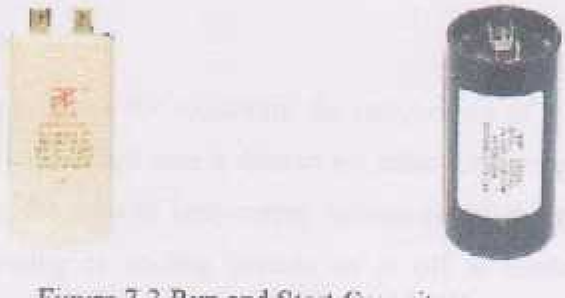


Figure 7.3 Run and Start Capacitors

7.3.4 Overload:

The most common cause of motor failure is overheating. The condition is created when a motor exceeds it is normal operating current flow. The result can be either a breakdown of the motor winding insulation and a short circuit, or a winding 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 7.4 shows overload.

Figure 7.4 Overload

7.3.5 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 7.5 shows thermostat.

Figure 7.5 Thermostat

7.3.6 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 7.6 shows contactor.

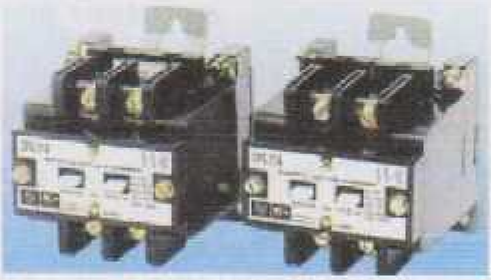


Figure 7.6 Contactor

7.4 Electrical Description For Cascade Cycle:

Cascade refrigeration cycle is one of the cooling multistage, and each stage need a compressor. Controlling the operation of these stages needs an electrical circuit integrated with the mechanical cycle. It is known that each stage of this cycle produces to a certain temperature, to reach very low temperatures, the first stage Cool until it reaches the - 25 degrees Celsius, then second stage will assist the first stage to reach -65 degrees Celsius. If we want to design an electrical control circuit for this mechanical cycle we can use more than one way. In this chapter we discussed two methods for controlling this cycle, which are:-

- Controlling with contactors.
- Controlling with PLC.

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

7.4.1 Control of cycle Using Contactors:

1. Cycle starts after pressing operating pushbutton. Current flows across the emergency switches and through ON/OFF pushbutton.

2. Then electric current flows through starting contactor coil (K), and then to the neutral line.

3. This ensure the arrival of current to the coil of the high cycle Contactor (Kh). The high cycle compressor will run and remain so until reaching the required temperature in the first stage (-25°C).

4. The Contactors (Kh) will operate the condenser fan and the timer (T).

5. The Timer (T) will operate the auxiliary contactor (K2).

6. The Contactor (K2) will operate the low cycle contactor (Kl), then second compressor will run and remain so until reaching the required temperature in the second stage (-65°C).

The above procedure is explained in Figure 7.7.

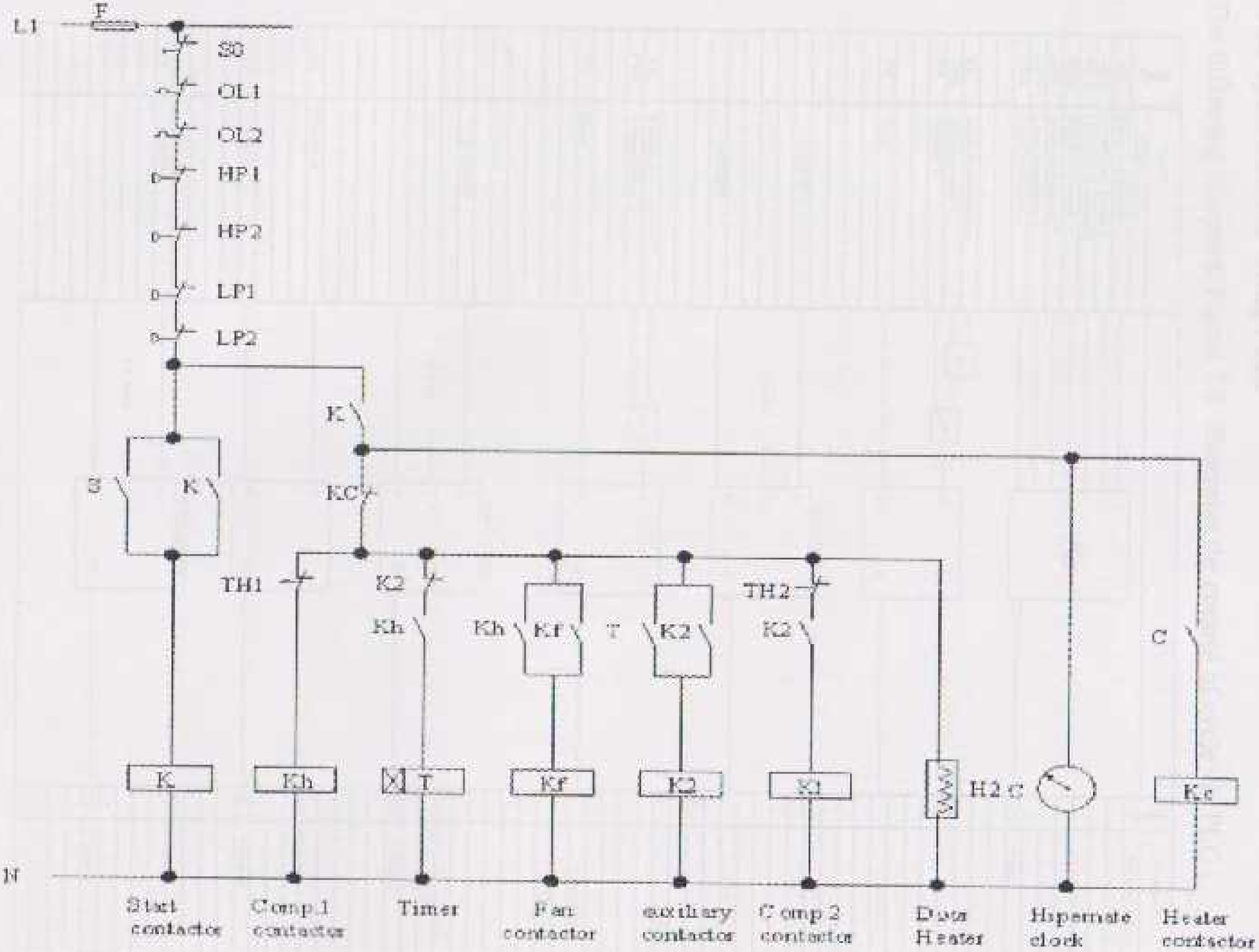


Figure 7.7 Contactor control circuit

7.4.2 Cycle Control Using PLC:

The following blueprint, Figure 7.8 illustrates the control of cycle via PLC .

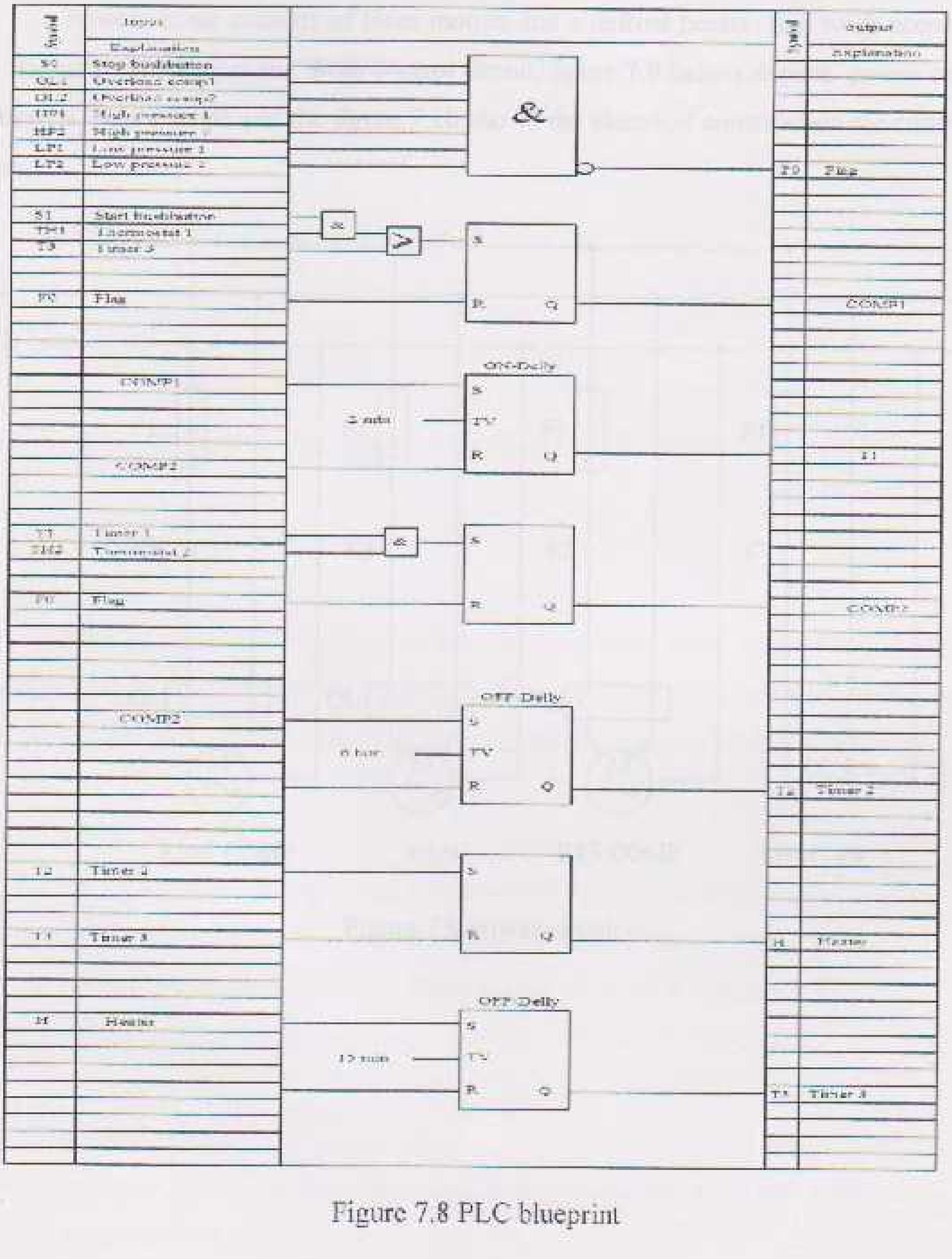


Figure 7.8 PLC blueprint

7.5 Power Circuit:

Power circuit consists of three motors and a defrost heater that work according to the instructions receiver from control circuit, figure 7.9 bellow depicts power circuit blueprint for the cycle and the figure 7.10 shows the electrical construction for compressors.

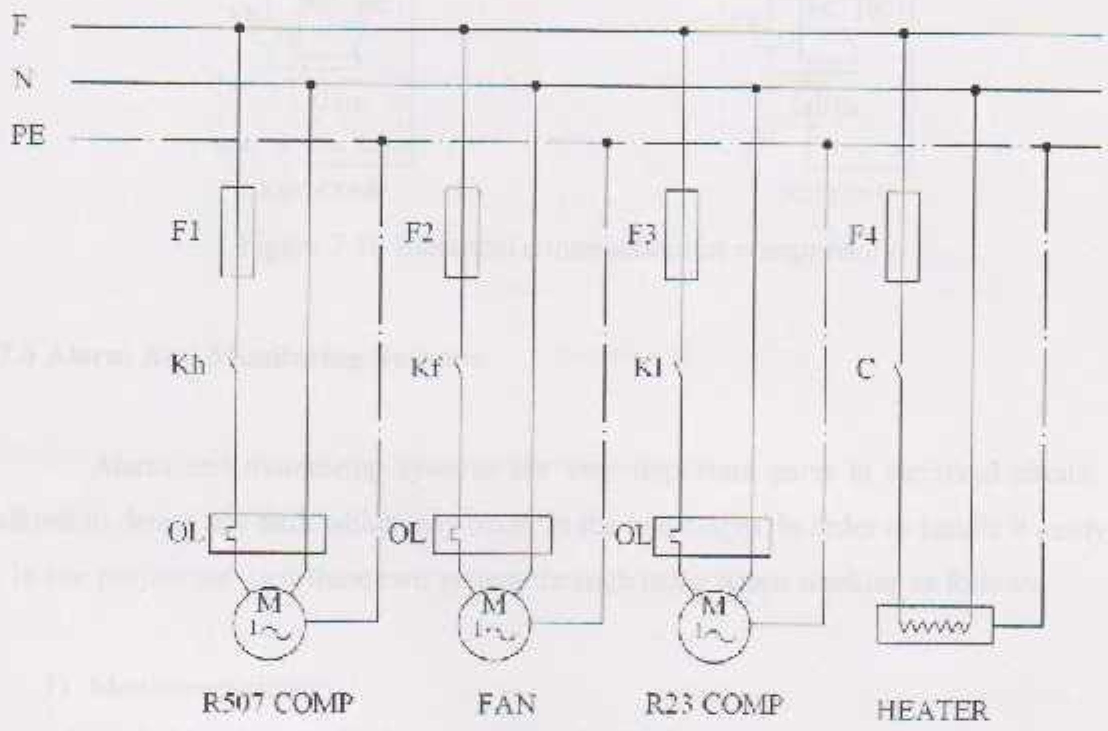


Figure 7.9 power circuit

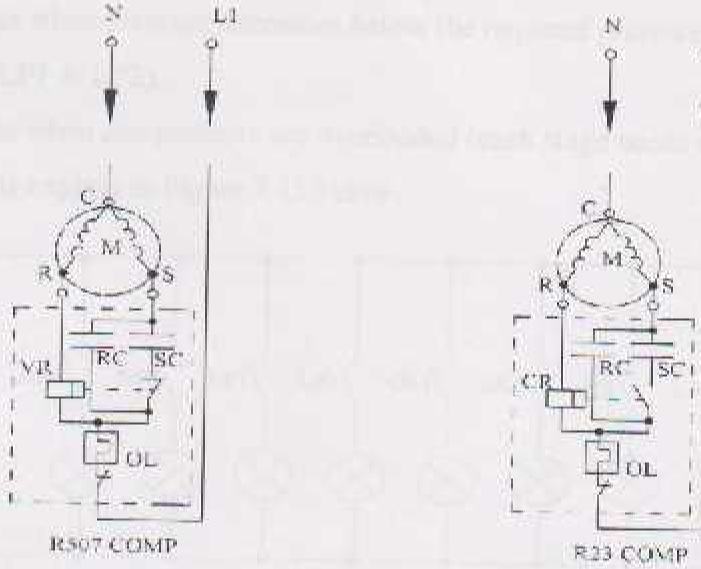


Figure 7.10 Electrical construction for compressors

7.6 Alarm And Monitoring Systems:

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

1) Monitoring circuit:

a-lamp lights when the high stage runs and off when it shutdown (HS).

b-lamp lights when the low stage runs and off when it shutdown (LS).

c-lamp lights when the condenser fan runs and off when it shutdown (F).

d-lamp lights when the defrost starts and off when it ends (Kc).

2)Alarming circuit:

a-lamp lights when pressure increases above required pressure (each stage needs one lamp) (HP1 & HP2).

b-lamp lights when pressure decreases below the required pressure (each stage needs one lamp) (LP1 & LP2).

c-lamp lights when compressors are overloaded (each stage needs a lamp) (OL1 & OL2). This is explain in Figure 7.11 below.

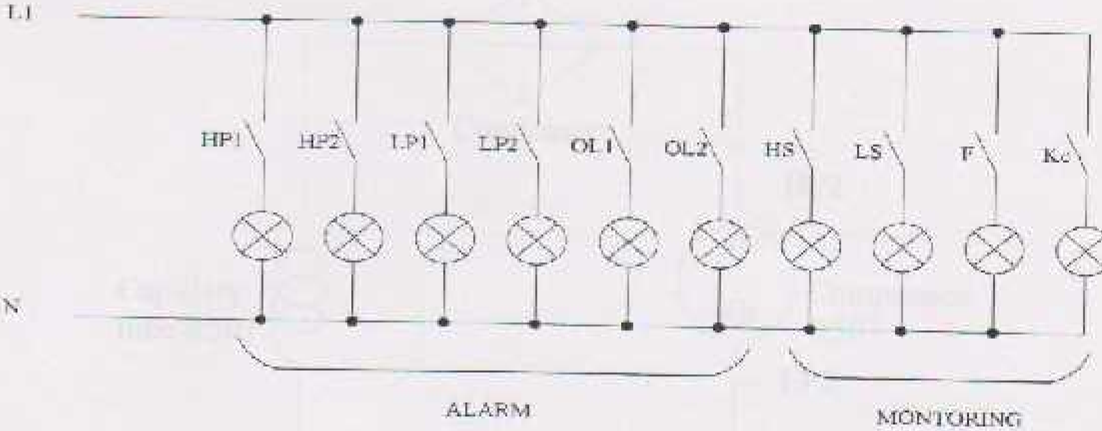
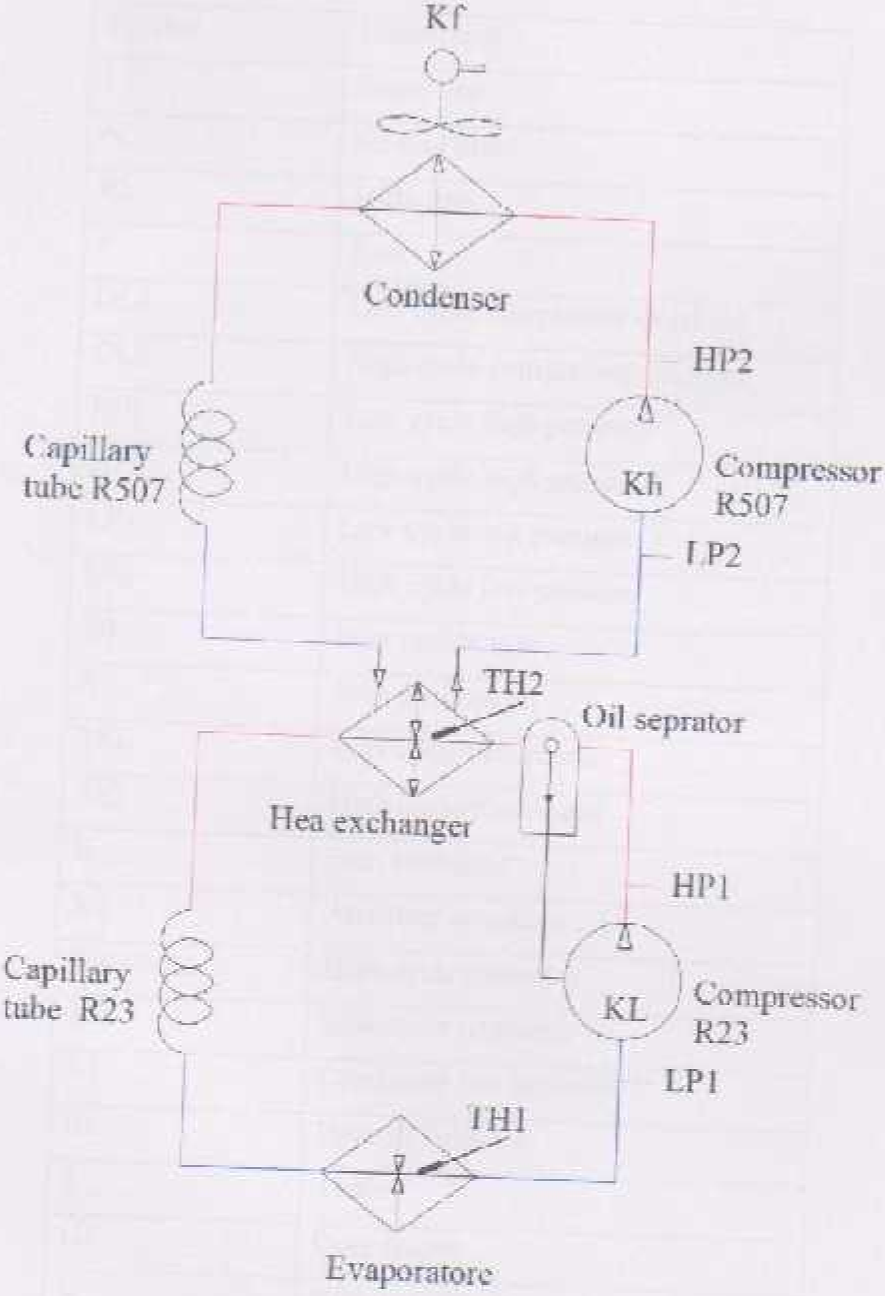


Figure 7.11 Alarm and Monitoring

Figure 7.12 shows electrical components on mechanical cycle.



7.12 Electrical components on mechanical cycle

Table 7.1 Circuit symbols

Symbol	Description
L1	Phase line
N	Neutral line
PE	Earth line
F	Fuse
OL1	Low cycle compressor overload
OL2	High cycle compressor overload
HP1	Low cycle high pressure
HP2	High cycle high pressure
LP1	Low cycle low pressure
LP2	High cycle low pressure
S0	Stop pushbutton
S	Start pushbutton
Th1	Low cycle thermostat
Th2	High cycle thermostat
K	Start contactor
K2	Auxiliary contactor
Kh	High cycle contactor
KL	Low cycle contactor
Kf	Condenser fan contactor
Kc	Heater Contactor
T	Timer
H2	Door heater
C	Hibernate clock
CR	Current relay

Symbol	Description
VR	Voltage relay
RC	Run capacitor
SC	Start capacitor
M	Motor

Recommendations:

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

- Refrigeration and air conditioning workshop lacks appropriate means for refrigerant leak detection.
- Collage library should be enriched with specialized references in the field of thermal sciences and HVAC
- A larger attention should be attributed to the electricity side of HVAC systems.
- Cryogenics a branch of refrigeration that should be introduced in the courses taught in the collage.

Conclusion:

After studying this project deeply the importance of this project in human's life becomes so clear. Such as this project helped and still helping humans to test equipments that people need it permanently in life such as testing planes parts and preserving humans, animals, and plants tissues to rebuilt and reproduce it again thus prevent it from extinction. But lakes in references, sources, parts and refrigerant makes building such as these projects in our country so difficult. Because of that we couldn't surround this project perfectly. Determining over all heat transfer coefficient in two phase is the main problem which we faced in a theory analysis in this project.

So we recommended some advices to consider it, to make studying such these projects easier.

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19- Tables A1, A2 and A3 from principle of refrigeration.

20- Table A4.معمل التبريد التجاري والصناعي. الإدارة العامة لتصميم و تطوير المناهج

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-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
	Cinder aggregate 100 mm		5.11
	Cinder aggregate 200 mm		3.29
	Cinder aggregate 300 mm		3.01
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	

TABLE A-2 Properties of common foods

Food	Water content %(mass)	Freezing point [°C]	Specific heat [kJ/kg.°C]		Latent heat [kJ/kg]
			Above freezing	Below freezing	
Vegetables					
Artichokes	84	-1.2	3.65	1.90	281
Asparagus	93	-0.6	3.96	2.01	311
Beans, snap	89	-0.7	3.82	1.96	297
Broccoli	90	-0.6	3.86	1.97	301
Cabbage	92	-0.9	3.92	2.00	307
Carrots	88	-1.4	3.79	1.95	294
Cauliflower	92	-0.8	3.92	2.00	307
Celery	94	-0.5	3.99	2.02	314
Corn	74	-0.6	3.32	1.77	247
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
Mushrooms	91	-0.9	3.89	1.99	304
Okra	90	-1.8	3.86	1.97	301
Onions, Green	89	-0.9	3.82	1.96	297
Onion, dry	88	-0.8	3.79	1.95	294
Parsley	85	-1.1	3.69	1.91	284
Peas, Green	74	-0.6	3.32	1.77	247
Pepper sweet	92	-0.7	3.92	2.00	307
Potatoes	78	-0.6	3.45	1.82	261
Pumpkins	91	-0.8	3.89	1.99	304
Spinach	93	-0.3	3.96	2.01	311
Tomato .ripe	94	-0.5	3.99	2.02	314
Turnips	92	-1.1	3.92	2.00	307
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
Cherries, sour	84	-1.7	3.65	1.90	281
Cherries, sweet	80	-1.8	3.52	1.85	267
Figs, dried	23	-------	-------	1.13	77
Figs, fresh	78	-2.4	3.45	1.82	261
Grapefruit	89	-1.1	3.82	1.96	297
Grapes	82	-1.1	3.59	1.87	274
Lemons	89	-1.4	3.82	1.96	297
Olives	75	-1.4	3.35	1.78	251
Oranges	87	-0.8	3.75	1.94	291
Peaches	89	-0.9	3.82	1.96	297
Pears	83	-1.6	3.62	1.89	277
Pineapples	85	-1.0	3.69	1.19	284
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

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

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