صنايلي

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Palestine Polytechnic University

College of Engineering & Technology

Mechanical Engineering Department

Graduation Project

Plate Ice Maker Problems: Solutions, Analysis, Redesign And Rebuild

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PROJECT NAME

Ice maker problems: Solutions, analysis , Redesign And Rebuilt

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According to the project supervisor and according to the agreement of the Testing committee members, this project is submitted to the Department of Mechanical Engineering at college of engineering and technology in partial fulfillment of the requirements of (B.SC) degree.

Supervisor Signature

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Examinee committee Signature

....

Department Head Signature

....

Dedication

To my parents... who raised me

To my friends... who helped me

To my beloved... who supported me to the end

To all the martyrs who sacrificed their lives for us

To our mothers and fathers

To all Palestinian mothers and fathers

To our great Palestine and Islamic nation

To who love the knowledge
To who carry candle of science
To light his avenue
of life...

To my supervisor Dr. Ishaq Sider

And to all Palestine polytechnic university doctors and engineers

Acknowledgment

My thanks go first to my Supervisor Dr.Ishaq Sider. His guidance and support made this work possible. His constant encouragement, intuitive, wisdom, and resolute leadership were instrumental in completing this work.

I wish to thank Eng. Khaled Sider and Eng. Khulood Al-Jabari. For helping me to do this work.

And finally, my ultimate thanks go to all lecturers, doctors, engineers, and to the great edifice of science, (Palestine Polytechnic University), for their effort and guidance which helped building our characters to become successful Engineers.

Abstract

The purpose of the project is to detect the problems of ice maker that was build before, by doing recalculation and redesign and rebuild to the ice maker system by using components of the refrigeration cycle, compressor, condenser, expansion valve and evaporator, to produce a plate of ice.

The refrigerant R134a was selected as suitable working fluid . This system consists of one cycle, where the cycle operates between (-21 and 35 °C).

theoretical design and building of the system will be made, according to the calculations that will be shown in the next chapters.
This unit will be used in the university laborites to show the process of forming ice

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CHAPTER ONE

INTRODUCTION

1.1 Project Contents(outline)

The introduction of project is divided up in 4 chapters; the chapters follow each other Logically to get the complete idea about the introduction and about the project.

Chapter 1: Introduction to the project with pre studies and why choosing plate ice maker.

Chapter 2: Components of the system.

Chapter 3: Calculation and analysis.

Chapter 4: Pipe design and Selection.

Chapter 5: Electrical control.

1.2 Introduction

This project aims to design and build up ice maker which is expected to produce an Ice plate which can be then divided into ice cubic by using a special producer.

However in the ice maker that was build before two years there are some of problems that appear when they operate the machine.

And we can mentioned the problems as follows:

a) evaporator design , were the area of the evaporator is smaller than needed.
b) water flow rate, the average flow rate of water on the plate of evaporator is high and more than should be.
c) the shape of the evaporator is not true, were the shape is not flat as should be.
d) there is an error in the calculated and design of pipes of the condenser and evaporator.

However I expected to face many problems such as providing the suitable component for the cycle. In the other hand the most important component in the cycle is the evaporator as the evaporator will decide the shape of the ice that will be produced, So I will show in details the calculation and selection of the evaporator shape in the next chapters.

Chapter one has some types of ice makers and why plate ice maker is chosen.

1.3 Project objectives

There are some objectives that can be mentioned as follow:

1) Design and build up ice maker to produce plate ice, according to calculation and selection of every single component of the cycle(or unit).
   The project will be used as a laboratory device to demonstrate the ice making Process and its relation with the processes occurring in the refrigeration cycle.
2) In the project, the refrigerant, cycle components, condenser and evaporator.......etc, will be chosen.
3) Study different types of ice makers especially plate ice maker.
4) Make the electrical control of this unit.

1.4 choosing type of ice maker(or type of ice)

From previous studies ice can be classified to:

1) Flake ice (chip ice)
2) Crushed ice, consist of small pieces made from crushing larger chunk of ice.
3) Fluid ice (or liquid ice), it is aqueous ice mixture that contains >20% of liquid, so it may be handled as fluid (pipe, pumps........etc.).
4) Dry ice, also called carbon dioxide snow, it is solid phase of CO2 .
5) Ice cubes (cubic ice).

Ice makers can be classified into:

1) Block ice maker shown in fig(1.1) can produce ice with 150-170 mm thick.

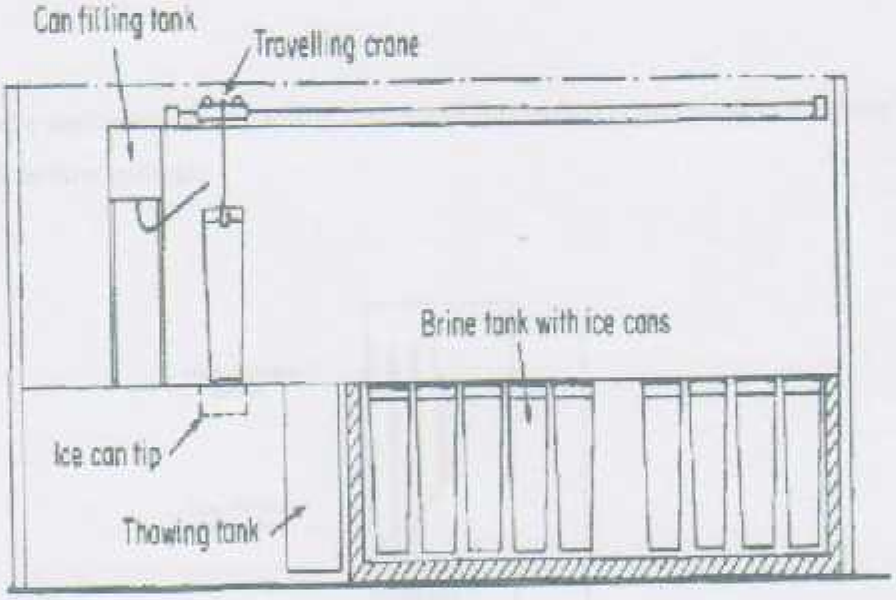


Fig (1.1) : block ice maker

2) Flake ice maker from ice 2 to 3 mm thickness (fig 1.2).

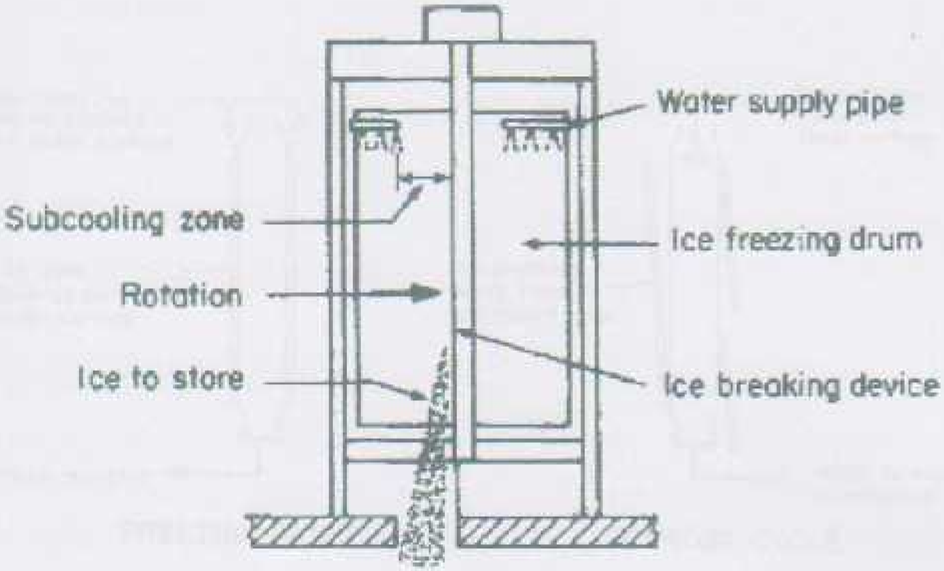


Fig (1.2) : flake ice maker

3) Tube ice maker, shown in fig(1.3), ice in this maker is produced in the form of small hollow cylinder.

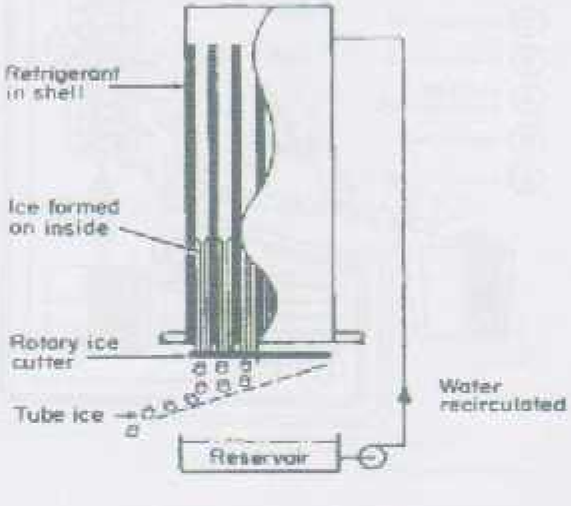


Fig (1.3) : Tube ice maker

4) Plate ice maker, shown in fig (1.4), can be produced plate of ice with thickness of 10 to 12 mm.

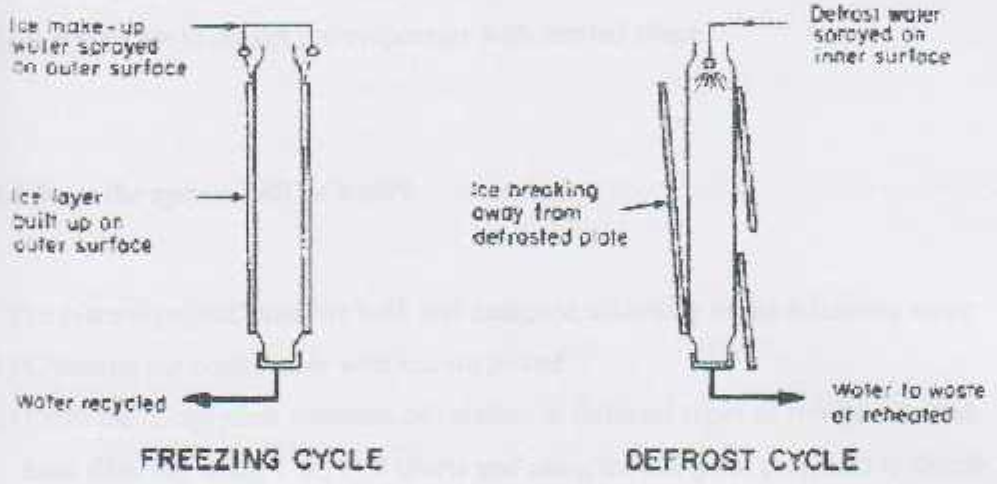


Fig (1.4) : plate ice maker

5) Cubic ice maker. Fig (1.5).

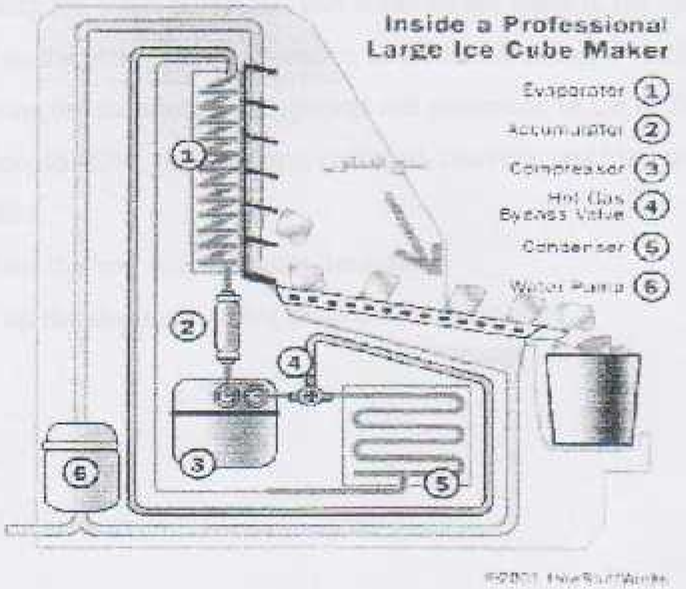


Fig (1.5) : cubic ice maker

The following points show why choosing plate ice maker.

1) The component of the cycle is available.
2) It is easy to connect these components with each other.
3) It is possible to design the evaporator with desired shape.

1.5 How the system will be built?

The system(project) must be built and designed according to the following steps:

1) Choosing the compressor with known power.
2) From the compressor selection calculation at different types of refrigerants had been done (by using T-S , P-V charts and using the coolpack program) to decide the most suitable refrigerant to be used according to it is properties (COP…..etc).
3) Calculation the load of the cycle.
4) By the cycle analysis a suitable condenser had been chosen.
5) Making calculation for the most important component of the cycle which is the evaporator, and choosing the best shape for it, as the shape of evaporator will decide finally the shape of the ice that will from into it.
6) Choosing the water pump that will circulate the water in the cycle and spraying water on the plate of the evaporator, so a suitable nozzle must be chosen.
7) Choosing the monitoring component and protection of the cycle, such as solenoid valve, accumulator, overload, receiver, and high and low pressure switches.
8) Deciding the best way to install the system.
9) Build up the electrical wiring diagram for the system.

The system is expected to work as follow:

1) Refrigeration cycle is on (compressor and fan of condenser are on).

2) Water from the water tank is circulated by the pump over the evaporator and sprayed by the nozzle.

3) The water is cooled down and gradually freezes on the evaporator plate.

4) Ice build up till reaching required thickness.

5) when reaching the required thickness the water recirculating pump and condenser fan are turned off and the harvest of ice starts.

6) Ice harvest takes place by hot gas entering the evaporator directly without been condensed and this step can be done by solenoid valve.

7) As the evaporator plate is slopped, the ice plate formed on the evaporator will fall into the special box by means of gravity.

8) Then a new cycle will begin.

1.6 Time table

The time of the introduction of project is scheduled as follow, table (1.1) shows how scheduled these weeks:

Table 1.1 The first semester time plan

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

Table 1.1 The second semester time plan

Task/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Load calculation																
Pipe design and selection																
the evaporator design																
Connecting the evaporator components																
Connect the evaporator with the cycle																
Operating the cycle																

CHAPTER TWO

COMPONENTS OF THE SYSTEM

Components of the system

2.1 Introduction

This chapter about the components that was used, and about the criteria for Selection each of them.

2.2 Ice maker cycle

The system that was build shown in fig (2.1).

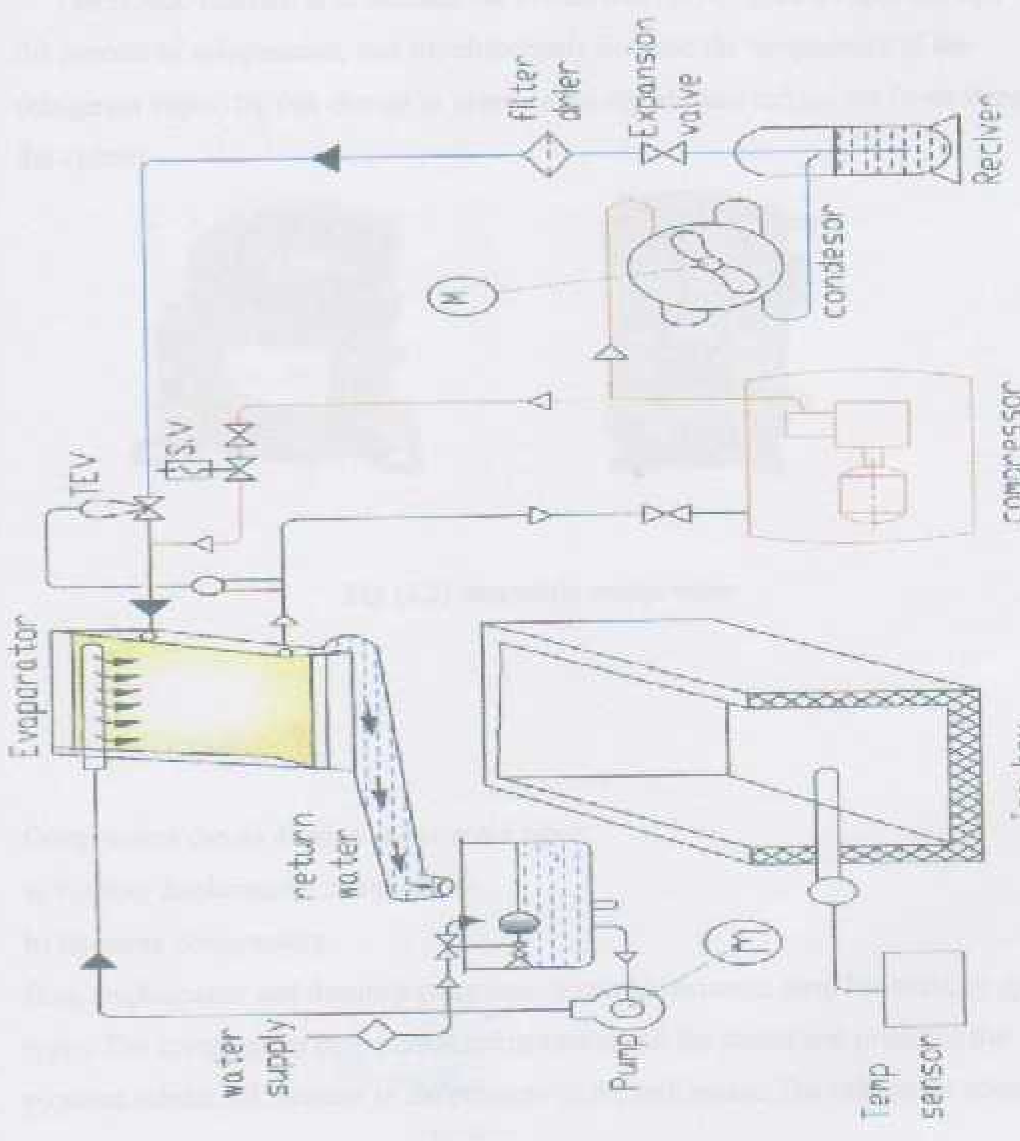


Fig (2.1) ice maker system

2.3 Components of the cycle

2.3.1 Compressor

Fig(2.2),(2.3) shows some types of compressors

Refrigerant compressors are known as the heart of the vapor-compression Refrigeration systems.

In a refrigeration cycle, the compressor has two main function within it. 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.

Fig (2.2) :hermetic compressor

Compressors can be divided in two main types:

a) Positive displacement compressors.

b) Dynamic compressors.

Both displacement and dynamic compressors can be hermetic, semi hermetic, or open types. The compressors both pumps refrigerant round the circuit and produces the required substantial increase in the pressure of the refrigerant. The refrigerant chosen and the operating temperature range needed for heat pumping generally lead to a need for a compressor to provide a high pressure difference for moderate flow rates, and this is most often met by a positive displacement compressor using a reciprocating piston. Other types of positive displacement compressor use rotating vanes or cylinders or intermeshing screws to move the refrigerant. In some large applications, centrifugal or turbine compressors are used, which are not positive displacement machines but accelerate the refrigerant vapor as it passes through the compressors housing. These various compressor types are illustrated in figure(2.3)

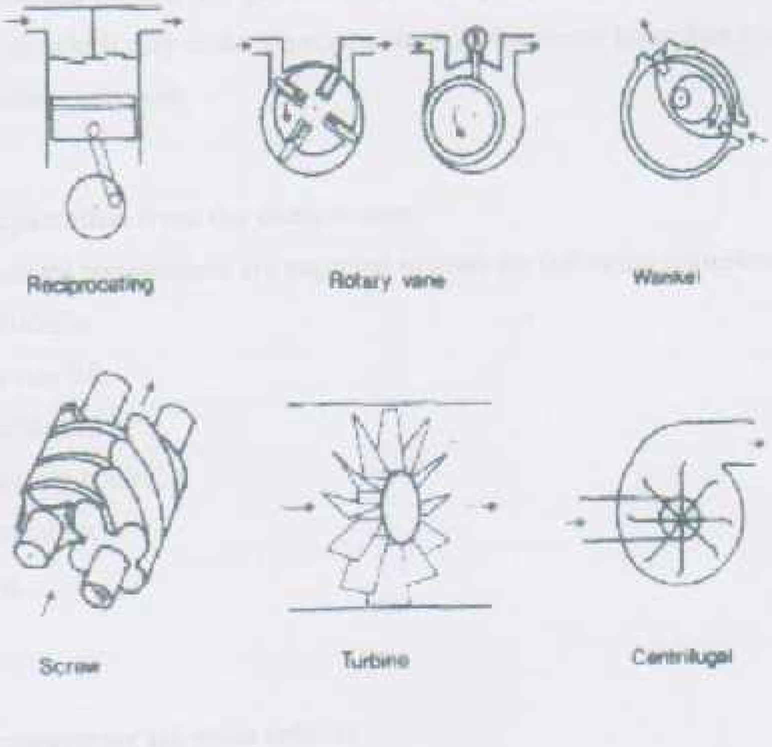


Fig (2.3) : some type of compressor

In this project the compressors that will be used is hermetic (fig 2.2)

Compressors are preferable on reliability grounds to units primarily designed for the smaller range of temperature required in air conditioning or cooling application. In small equipment where cost is a major factor and on-site installation is preferably kept to a minimum, such as hermetically sealed motor/compressor combination, there are no rotating seals separating motor and compressor, and the internal components are not accessible for maintenance, the casing being factory welded.

In these compressors ,which 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, freezer, and air conditioners are of the hermetic type. An internal view of a hermetic type refrigeration compressor is shown in figure (2.4). the capacities of these compressors range are identified with their motor capacities. For example, the compressor capacity ranges from 1/2 HP to 30 HP in household refrigerators. Their revolutions per minute are either 1450 or 2800 rpm. Hermetic compressors can work for along time in small-capacity refrigeration system without any maintenance requirement and without any gas leakage, but they are sensitive to electric voltage fluctuations, which may make the copper coils of the motor burn. The cost of these compressors is very low.

2.3.1.1 expectation from the compressors

The refrigerant compressors are expected to meet the following requirements :

- High reliability .
-Long service life.
-Easy maintenance.
-Easy capacity control.
-Quiet operation.
-Low cost.

2.3.1.2 compressor selection criteria

In the selection of a proper refrigerant compressor, the following criteria are considered:

-Refrigeration capacity .
-Volumetric flow rate.
-Compression ratio, and
-Thermal and physical properties of the refrigerant.

For the ice maker cycle a compressor suggested to be used is single phase hermetic compressor.

2.3.2 Refrigerants

In general, refrigerants are well known as the fluids absorbing heat during evaporation. These refrigerants, which provide a cooling effect during the phase change from liquid to vapor, are commonly used in refrigeration, air conditioning, and heat pump systems, as well as process systems.

2.3.2.1 Selection of Refrigerant

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:

-Ozone and environment friendly.

-Low boiling temperature.

-Low volume of flow rate per unit capacity.

-High heat of vaporization.

-Non-flammable and non explosive.

-Non corrosive and non toxic.

-Non acidic in case of mixture with water or air.

-Chemically stable.

-Suitable thermal and physical properties (e.g., thermal conductivity, viscosity).

-Commercially available.

-Easily detectable in case of leakage, and

-Low cost.

The table below shows a comparison among R12, R22 and R134

Table (2.1): comparison between refrigerants operating at the same conditions

	R12	R22	R134a
Name	Dichlorodifluoromethane	Chlorodifluoromethane	Tetrafluoroethane
Formula	CCL2F2	CHClF2	CH2FCF3
Evaporator temp.(C)	-21	-21	-21
Condenser temp.(C)	35	35	35
V(m3/s)	0.001	0.00058	0.00109
m(Kg/s)	0.009	0.0061	0.0072
Cop	3.5	3.35	3.47
Qcond (KW)	1.26	1.231	1.253
Qevap (KW)	0.981	0.933	0.972
Pcomp (KW)	0.280	0.280	0.280

* COP for refrigerant R12 is the highest, but R12 is phase out because it is high ozone depletion potential (ODP).

The table below shows some properties of R12, R22 and R134a refrigerants

Table (2.2) properties of refrigerants

	R12	R22	R134a
Name	Dichlorodifluoromethane	chlorodifluoromethane	Tetrafluoroethane
formula	CCL2F2	CHCL2F	CH2FCF3
Boiling point(C)	-29.8	-40.7	-26.3
At 100 kPa			
Freezing point(C) at 100kpa	-29.8	-160	-103.3
Critical temperature(Tc) (C)	112	96.2	101.1
Critical pressure (Pc)MPa	4.170	4.936	4.060
Triple point temperature (Tt) (C)	-157	-157.39	-103.3
Specific volume at Patm. For sat. liquid m3/kg	0.6720	0.7079	0.7264
Molecular mass g/mol	120.91	86.468	102.03
Ozone depletion potential	0.82	0.05	0
Sat. pressure at +35 C MPa	0.84	1.354	0.886
lubricants	Mineral	Mineral alkyl benzene	Alkyl benzene poly ester

*** Leak detection for refrigerant:**

- Bubble test (Soap Solution)
- Water Immersion Method.
- Dye Interception Method.
- Electronic Leak Detectors.
- Ultrasonic Leak Detectors.

Leak in refrigeration units can be detected by a number of methods. The electronic detector is widely used in the manufacture and assembly of refrigeration equipment. The instrument is used to detect leaks in refrigerants except refrigerant (R14). It is not recommended for use in atmospheres containing explosive or flammable vapors.

According to the above data the chosen refrigerant is R134a.

2.3.3 Condenser

There are several condensers to be considered when making a selection for installation. They are air-cooled, water-cooled, shell and tube, shell and coil, tube within a tube, and evaporative condensers. Each type of condenser has its own unique application. Some determining factors include the size and the weight of the unit, weather conditions, location (city or rural), availability of electricity, and availability of water. A wide variety of condenser configurations are employed in the process industry.

Fig (2.4) : Compressor with air cooled condenser

Selection of condenser type is not easy and depends on the following criteria:

- Condenser heat capacity.
- Condensing temperature and pressure.
- The flow rates of refrigerant.
- Design temperature of air.
- Operation period, and
- Climatic conditions.

Condenser is of two type air cooled condenser and water cold condenser, but here the concern is on air cooled one.

The fig. below shows Air-cooled condensers.

Fig (2.5) : Air cooled condenser

The air-cooled condensers find applications in domestic, commercial, and industrial refrigerating, chilling, freezing and air conditioning systems with a common capacity of 20-120 tons. The centrifugal fan air-cooled condensers (with a capacity of 3-100 tons) are particularly used for heat recovery and auxiliary ventilation application. 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 steam.

The advantages of air-cooled condensers are:

- No water requirement.
- Standard outdoor installation.
- Elimination of freezing, scaling, and corrosion problems.
- Elimination of water piping, circulation pumps, and water treatment.
- Low installation cost, and
- Low maintenance and service requirement.

On other hand, they have some disadvantages, as given below:

- High condensing temperature.
- High refrigerant cost because of long piping runs.
- High power requirements per KW of cooling.
- High noise intensity, and
- Multiple units required for large-capacity systems.

The condenser that will be used in this system is air-cooled condenser.

2.3.4 Condenser fan

It is used to increase the efficiency of the condenser by making force convection between the air and the surface of condenser pipe.

Fig (2.6) : Condenser fan

2.3.5 Throttling Devices

Throttling device is important to increase the velocity of refrigerant and decreasing its pressure. Fig(2.7) shows the principle of the device.

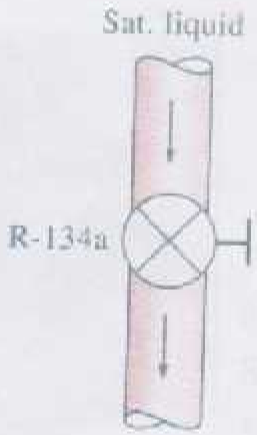


Fig (2.7) : Throttling device

In practice, throttling devices called either expansion valves or throttling valves, it is 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 refrigerant space. The most common throttling devices are as follows:

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

2.3.5.1 Thermostatic expansion valves

The fig(2.8) below shows expansion valve

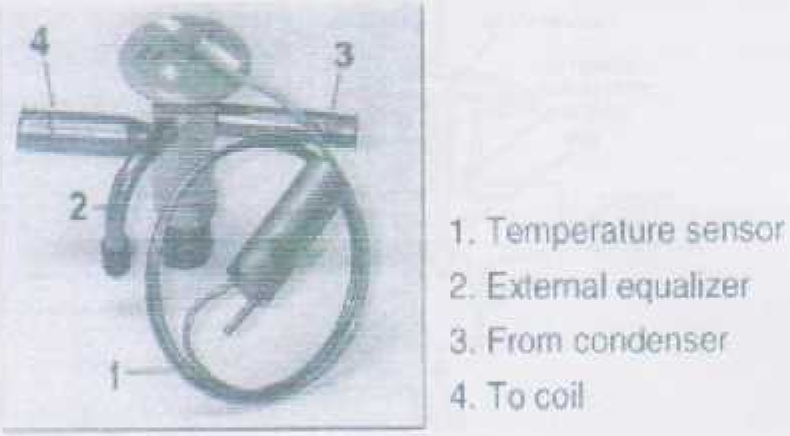


Fig (2.8) : expansion valve

The thermostatic expansion valves are essentially reducing valves between the high pressure side and the low-pressure side of the system. these valves, which are the most widely used devices, automatically control the liquid-refrigerant flow to the evaporator at a rate that matches the system capacity to the actual load.

They operate by sensing the temperature of the superheated refrigerant vapor leaving the evaporator.

For a given valve type and refrigerant, the associated orifice assembly is suitable for all version of the valve body and in all evaporating temperature ranges.

When the thermostatic expansion valves is operating properly, the temperature at outlet side of the valve is much lower than that at the inlet side.

If this temperature difference does not exist when the system is working, the valve seat is probably dirty and clogged with foreign matter.

Once a valve is properly adjusted, further adjustment should not be necessary. The major problem can usually be traced to moisture or dirt collection at the valve seat and orifice.

Figure (2.9) shows a common type of expansion valve.

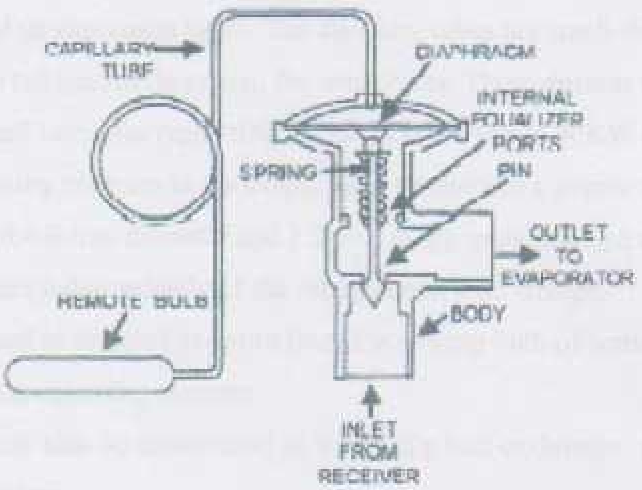


Fig (2.9) : thermostatic expansion valve

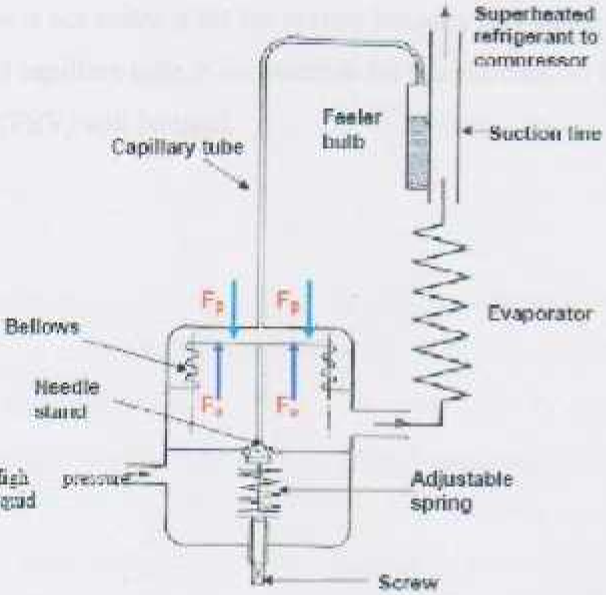


Figure (2.10): schematic of thermostatic expansion valve

2.3.5.2 Capillary tubes

The capillary tube is the simplest type of refrigerant flow control device 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 which are widely used in small hermetic-type refrigeration systems (up to 30KW capacity), 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.

These tubes are used to transmit pressure from the sensing bulb of some temperature control device to the operating element.

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

Other considerations in determining capillary tube size include condenser efficiency and evaporator size.

Capillary tubes are most effective when used in small capacity systems.

The capillary tube is not suitable for the system because harvesting the ice system will be defrosting, and capillary tube is not suitable for this process, so thermostatic expansion valve (TEV) will be used.

2.3.6 Receiver tank

The receiver tank (fig. 2.11) acts as a temporary storage space and surge tank for the liquid refrigerant. The receiver also serves as a vapor seal to keep vapor out of the liquid line to the expansion valve. A pressure drop in the liquid line of a refrigeration system may cause the liquid refrigerant to flash to gas. Receivers are constructed for either horizontal or vertical installation.

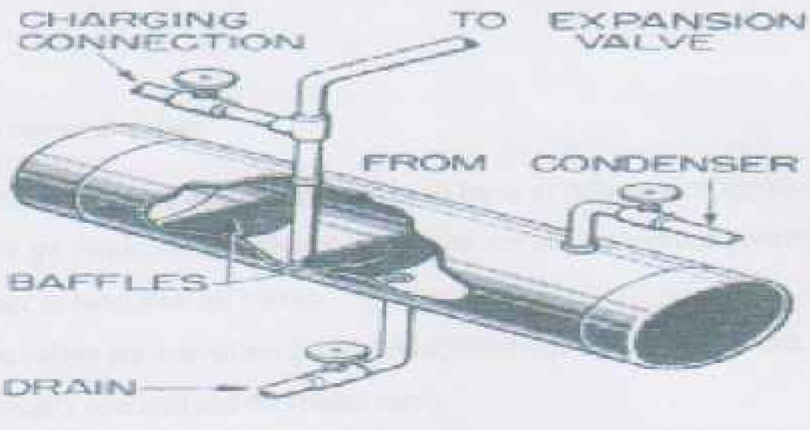


Fig (2.11) : Receiver tank

2.3.7 Accumulators

It is well known that compressors are designed to compress vapor, not liquid. Many refrigeration systems are subjected to the return of excessive quantities of liquid refrigerant to the compressor. Liquid refrigerant returning to the compressor dilutes the oil, washes out the bearings, and in some cases causes complete loss of oil in the compressor crankcase. This condition is known as oil pumping or slugging, and results in broken valve reeds, piston, rods, crankshafts, and the like.

The purpose of the accumulator is to act as a reservoir to temporarily hold the excess oil-refrigerant mixture and to return it at a rate that the compressor can safely handle.

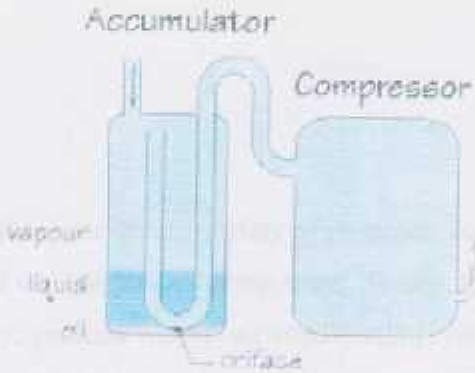


Fig (2.12) : Accumulator

2.3.8 Solenoid valves

Solenoid valves are extensively used in all types of refrigeration applications. These valves are employed as electrically operated line stop valves and perform in the same manner as hand shut-off valves.

These valves are convenient for remote applications due to the fact that these are electrically operated and controlled easily.

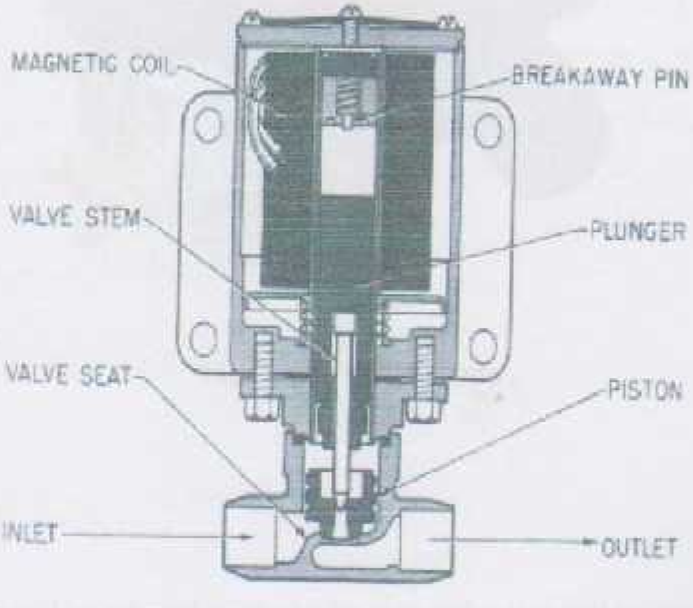


Fig (2.13) : part of solenoid valve

2.3.9 Water pump

pumps are used throughout society for variety of purposes. Early applications includes the use of the windmill or watermill to pump water. Today , the pump is used for irrigation, water supply, gasoline supply, air conditioning systems, refrigeration (usually called a compressor), chemical movement, sewage movement, flood control, marine services, ...etc.

because of the wide variety of applications, pumps have a plethora of shapes and size: from very large to very small, from handling gas to handling liquid, from high pressure to low pressure, and from high volume to low volume.

Fig (2.14) : water pump

The pump that will be selected during to the calculation that will be done in the calculation chapter.

2.3.10 Water tank

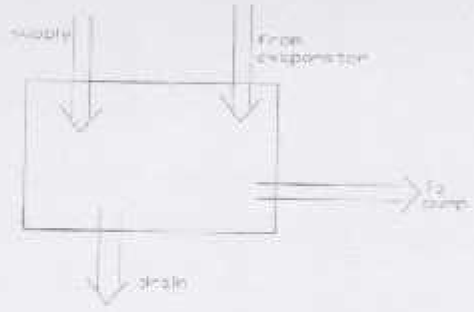


Fig (2.15) : water tank

The water tank shown in the above figure(2.15) is made of thick steel. Two layers of steel, and between these two layers an insulation material will be install to save the cooled water from outside temperatures.

As shown from the figure the tank must have two inputs: one for the water inlet from the supply and one for the water return from the evaporator which is usually cold. However it has two outputs: one for the pump that will circulate water to the cycle, and one for the drain to empty the water from the tank.

2.3.11 Ice box

the ice box must be insulated by some material to prevent ice from melting, so as the water tank it must contain of two layers and insulating material between them. The ice box must be opened from upside to allow for the plated ice to be in.

figure(2.16) shows ice box.

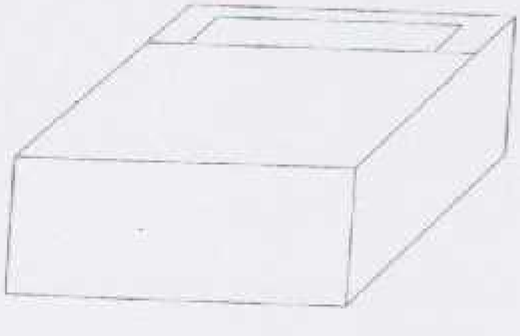


Fig (2.16) : Ice box

CHAPTER THREE

CALCULATION AND CYCLE ANALYSIS

Calculation and cycle analysis

3.1 Calculation criteria

3.1.1 Calculation by selection of compressor

In this method single phase hermetic compressor with power of 3/8 HP(220V, 50HZ) is selected and calculation is done according to it.

As it mentioned in the previous chapter the refrigerant that will be used is R134a calculation will be made for R134a

*** By using R134a**

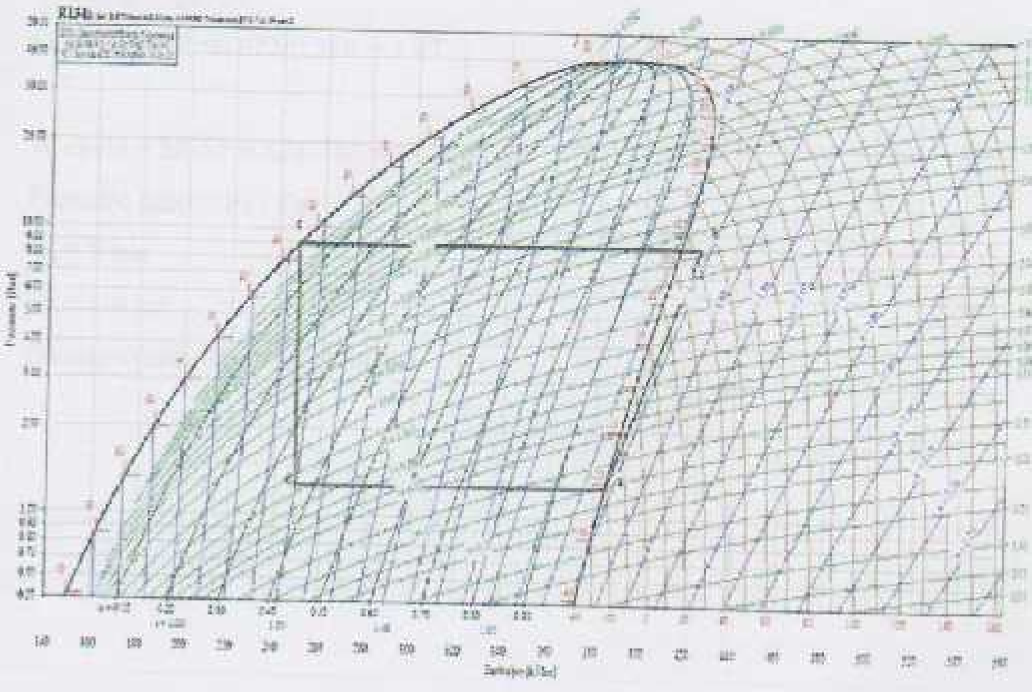


Fig (3.1) : P-h diagram for R134a

W comp = 3/8 * 748 = 0.280 KW

Te = -21C

Tc = 35C

W comp = m*(hb-ha)

0.280 = m*(242-385)

m = 0.072 kg/s

Qe = m*(ha-hd)

= 0.072 * (385-250)

= 0.972 KW

Qc = m*(hb-hc)

= 0.072 * (424-250) = 1.253 KW

W comp = Qc-Qe

COP = Qe/ P =0.972/0.280 = 3.47

W cycle = h2-h1 = 424-385 = 39 KJ/Kg

Pressure ratio = Pc / Pe

Pc = 9 bar

Pe = 1.4 bar

Pressure ratio = 9 / 1.4 = 6.428

*** By using R134a with superheating 10C & sub. Cooling 5C**

h@ point b = 394 KJ/Kg

h@ point c = 436 KJ/Kg

h@ point f = 193 KJ/Kg

h@ point g = 193 KJ/Kg

W comp = 0.280 KW

Te = -21C

Tc = 35C

m = 0.280 / (hc-hb) = 0.280 / (436-394)

= 0.063 kg/s

Qe = m * (hb-hg) = 0.063 * (394-193)

= 1.3467 KW

Qc = m * (hc-hf) = 0.063 * (436-193)

= 1.6281 KW

W comp = Qc-Qe

COP = Qe / P = 1.3467 / 0.280 = 4.81

a-b : super heating

b-c : work compressor

d-e : Q condenser

e-f : sub cooling

g-a : Q evaporator

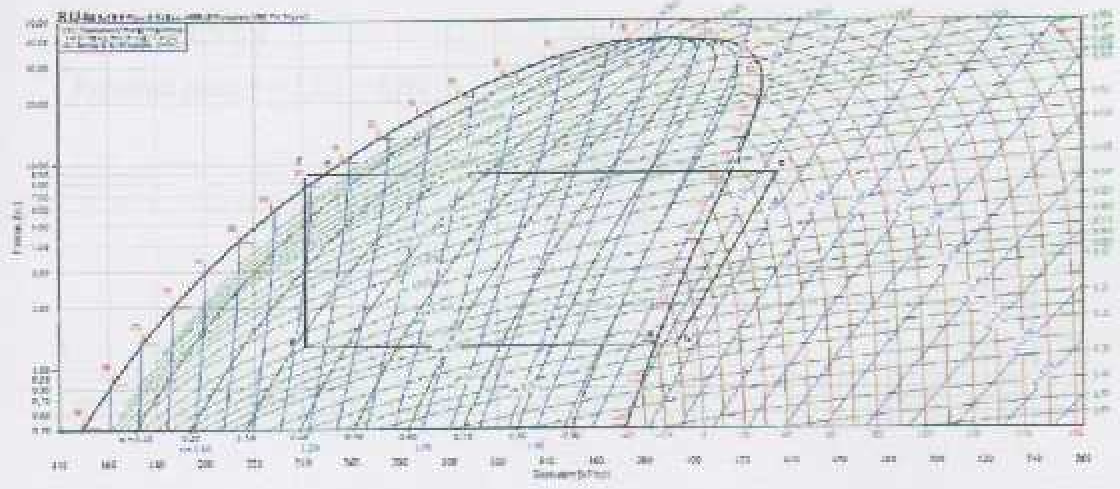


Fig (3.2) : P-h diagram for R134a(super heating 10, sub cooling 5)

*** By using R134a**

@ Te = -21C , Tc = 35C & superheating 20C , sub cooling 5C

h1 = 402 KJ/Kg

h2 = 445 KJ/Kg

h3 = 241 KJ/Kg

h4 = 241 KJ/Kg

W comp = 0.280 KW

m = 0.280 / (h2-h1) = 0.280 / (445-402)

= 0.065 kg/s

Qe = m * (h1-h4) = 0.065 * (402-241)

= 1.064 KW

Qc = m * (h2-h3) = 0.065 * (445-241)

= 1.326 KW

W comp = Qc-Qe

COP = Qe / P = 1.046 / 0.280 = 3.73

W cycle = h2-h1 = 445-402 = 43 KJ/Kg

Pressure ratio = Pcond. / Pevap.

Pcond. = 9 bar

Pevap. = 1.3 bar

Pressure ratio = 9 / 1.3 = 6.92

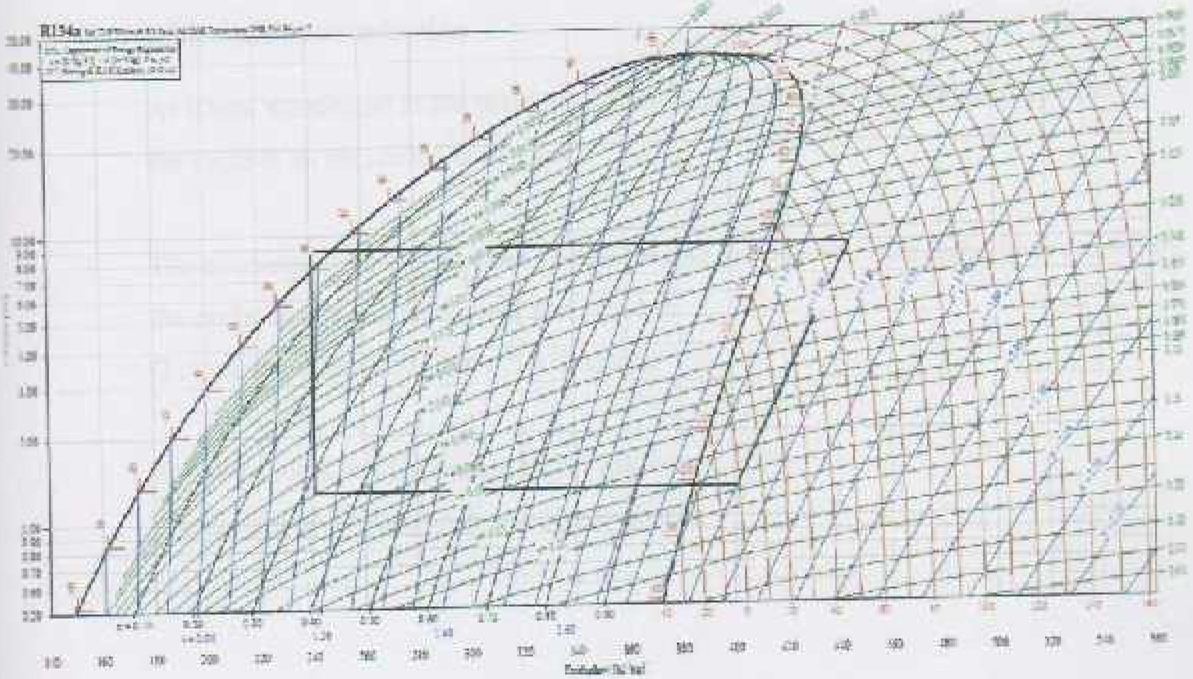


Fig (3.3) : P-h diagram for R134a (at 20 superheating, 5 sub cooling)

From the calculation that was done in the previous pages we note that the refrigerant R134a with superheating 10C and sub cooling 5C have high COP.

So the refrigerant that will be used in the cycle is R134a with superheating 10C and sub cooling 5C and the result (from the P-h diagram) is:

W comp = 0.280 KW

m = 0.063 kg/s

Qe = 1.3467 KW

Qc = 1.6281 KW

COP = 4.81

3.1.2 Condenser selection

As it was mentioned in the previous chapter the condenser type that will be used in the cycle is an air cooled-condenser

The condenser that will be used should be able to reject the heat that will produced by the cycle, and it was calculated and equal to
Q cond = 1.6281 KW (from P-h diagram)

So the selected condenser must eject this heat.
Selection will be done from condenser charts.

Fig (3.4) : Air cooled-condenser

However the condenser have a fan with it, to make forced convection.
The fan usually single phase with power 10-15 W (from previous project), sometimes this fan has multi speeds.

3.1.3 Evaporator design

The evaporator will decide the shape of the ice that will be produced, and this project is concerned with plated ice, so the ice shape must be plate, and the suggestion for these shape as shown in fig(3.5)

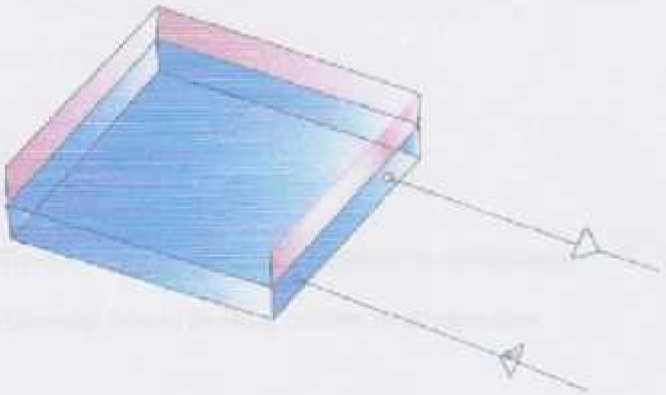


Fig (3.5)A : plate evaporator without tubes

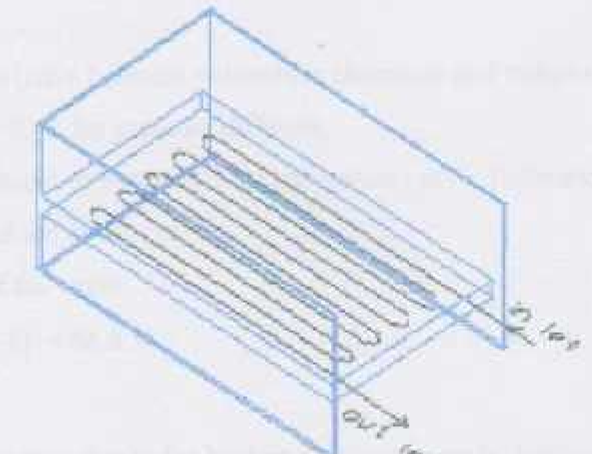


Fig (3.5)B : plate evaporator with tubes into it

Fig (3.6) Suggested shapes for the evaporator

In fig(3.5)A the refrigerant will boil inside the plate.

In fig(3.5)B the refrigerant will boil inside the tubes that installed under the plate.

3.1.3.1 Compressors Calculations And Selection

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

$$\eta_v = \eta_c * \eta_h \quad (3.1)$$

Where:

η_v: volumetric efficiency

η_c: volumetric efficiency due to clearance volume in compressor

η_h: volumetric efficiency due to heating occurs in compressor

The volumetric efficiency due the clearance volume in compressor calculated by equation .[reference 6]:

$$\eta_c = 1 - c[(\frac{PH}{PL})^{1/n} - 1] \quad (3.2)$$

Where :

c : clearance volume (ratio between volumetric clearance and volume of cylinder of the compressor , c = 0.02 for pressure different,

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

PH: High pressure of the cycle

PL :Low pressure of the cycle

$\eta_c = 1 - 0.02[(\frac{8.5}{1.3})^{1/1} - 1] = 88.3$ %

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

$$\eta_h = \frac{Tevap}{Tcond} \quad (3.3)$$

Where :

Tevap. :evaporator temperature [°K]

Tcond. :condenser temperature [°K]

$\eta_h = 252/308 = 82\%$

$\eta_v = 88.3\% * 82\% = 0.724\%$

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

$$V'_{theo} = m' * \upsilon \qquad (3.4)$$

Where :

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

m' :mass flow rate of refrigerant [kg/s]

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

$V'_{theo} = 17.4\ [m^3/s]$

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

$$V'_{act} = \frac{V.theo}{\eta v} \qquad (3.5)$$

Where :

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

$V'_{act} = 17.4 * 0.724 = 12.6[m^3/s]$

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

From Aspera company catalog we chose the compressor with code number T2134A which have displacement 17.4 cm^3

So

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

$$V'_{act} = V'_{theo} * \eta_v \qquad (3.6)$$

Where :

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

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

η_v: volumetric efficiency

V'_{act} comp = V'_{act} cycle

V'_{act} cycle = V'_{theo} / η_v

Where η_v from the side of the cycle

$V'_{theo} = V'_{act}$ cycle * η_v

= 12.6 * 0.724

= 9.1224 [m^3/s]

Qe = m'ref * Δh

= 0.063 * (385 – 250)

Qe = 0.8235 kw

3.1.3.2 Plat evaporator area calculation

Qe = U * A *(ΔT)

Where :

Q : the load in evaporator

➔Q = 0.8235 kw

A : area of the evaporator

➔ A = ????

ΔT : the change in temperature = 21C

U : over all heat transfer coefficients

The calculation of U doing as follow:

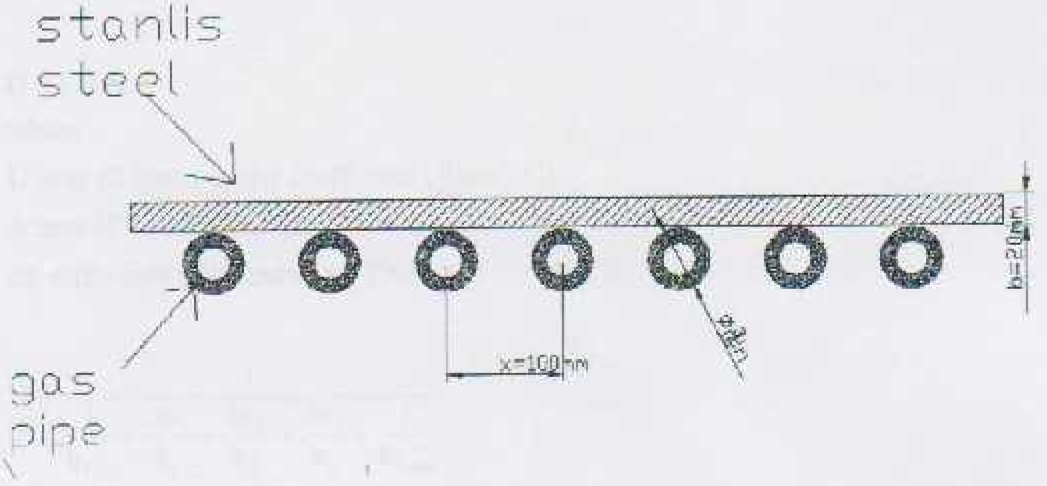


Fig (3.7) : section of plate evaporator (with tubes)

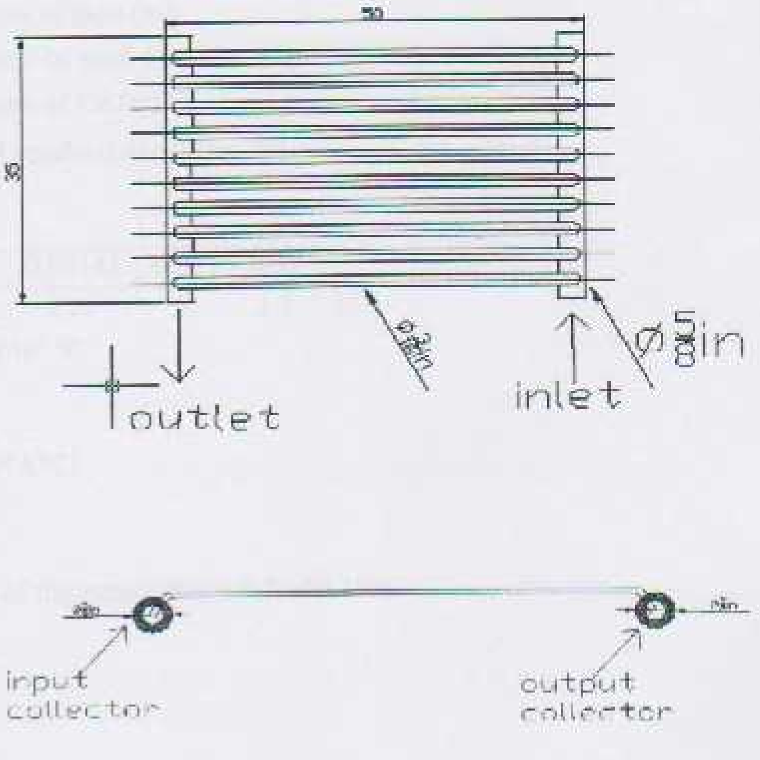


Fig (3.8) : section of the evaporator

To find U value the following equation is applied :

$Q = UA\Delta t$

where :

U:over all heat transfer coefficient ($W/m^2.^\circ C$).

A:aera of heat trensfer [m^2]

Δt: differance of temperature ($^\circ C$).

$$U = \frac{1}{\frac{1}{h_{f,in}} + \frac{\Delta x_1}{k_1} + \frac{\Delta x_2}{k_2} + \frac{\Delta x_3}{k_3} + \frac{1}{h_{f,out}}}$$

where:

$h_{f,in}$: convection heat transfer coefficient of refrigerantR134a ($W/m^2.^\circ C$).

$h_{f,out}$: convection heat transfer coefficient of air ($W/m^2.^\circ C$).

Δx1: thickness of pipe (m).

Δx2: thickness of steel (m).

Δx3: thickness of ice (m).

k1 : thermal conductivity ($W/m.^\circ C$)

$$U = \frac{1}{\frac{1}{10000} + \frac{0.00145}{120} + \frac{0.001}{20} + \frac{0.02}{2.2} + \frac{1}{30}}$$

$U = 23.5\ W/m^2\ ^\circ C$

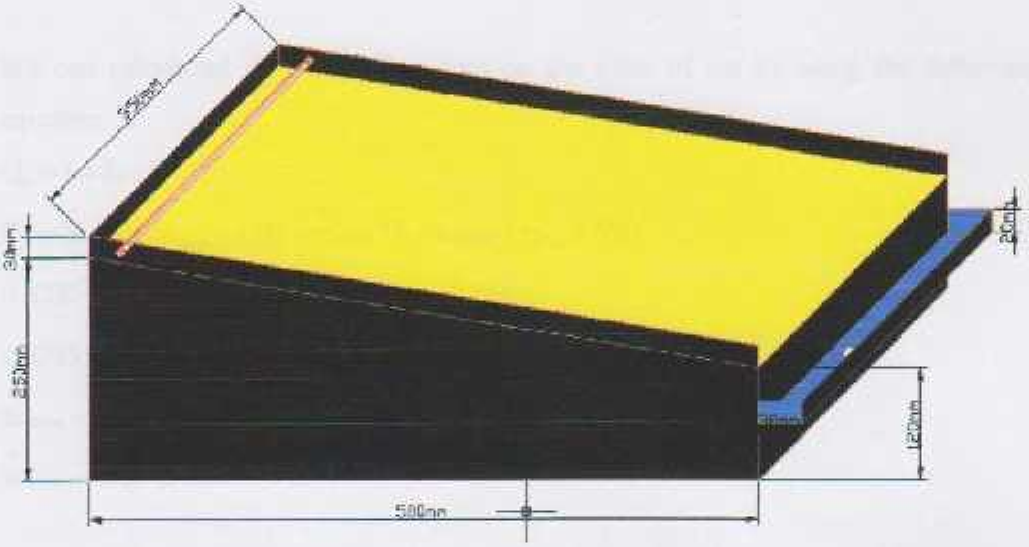
$Q = UA\Delta t$

823.5=23.5*A*21

$A=1.5m^2$

➔ the area of the evaporator = 0.5m*0.35m

The evaporator shape in this project is closed to the following shape:



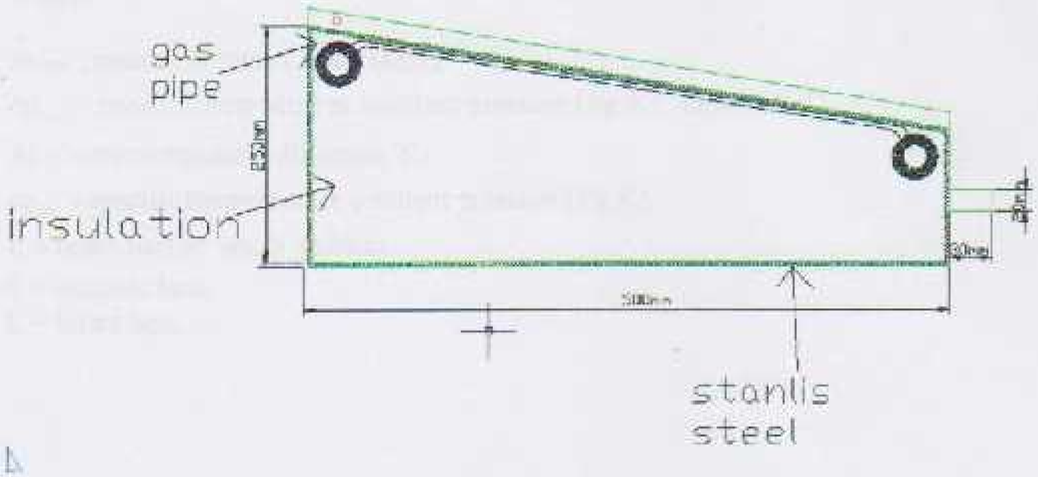


Fig (3.9) : the evaporator shape

The ice plate will be formed in the plate above the pipe and when to harvest the ice a defrosting process starts till sides of ice which have contact with the plate melt and ice fall into ice box by means of gravity

3.1.4Calculation of water flow rate

We can calculated the water flow rate on the plate of ice by using the following equation:

$$Q_e = S + L + S$$

$$Q_e = \dot{m}_{water} \times cp_{water} \times (\Delta t) + \dot{m}_{water} * h + \dot{m}_{water} \times cp_{ice} \times (\Delta t)$$

$$0.8235 = \dot{m}_{water}(cp_{water} * 20 + 334 + cp_{ice} * 10)$$

$$0.8235 = \dot{m}_{water}(4.1813 * 20 + 334 + 2.05 * 10)$$

$$\dot{m}_{water} = 2.465 * 10^{-3} kg/s$$

$$\dot{m}_{water} = 9kg/h$$

where:

$\dot{m}_{water}$: mass flow rateof water (kg/s).

cp_{water} . specific temprature at constant pressure (J/g.K). table A-2.

Δt = water temprature different(°C).

cp_{ice} = specific temprature at constant pressure (J/g.K).

h = latent heat of water (kJ/kg)

S = sensible heat.

L = latent heat.

CHAPTER FOUR

PIPE DESIGIN AND SELECTION

Chapter Four

Pipe Design and Selection

4.1 Introduction

Pipes carrying fluids at high pressure develop both radial and tangential stresses with values that depend on the radius of the element under consideration. In determining the radial stress (σ_r) and the tangential stress (σ_t), by using the assumption that the longitudinal elongation is constant around the circumference of the pipe. By referring to figure (4.1) it can be shown that tangential and radial stresses exist whose magnitudes represented in equation (4.1) and (4.2).

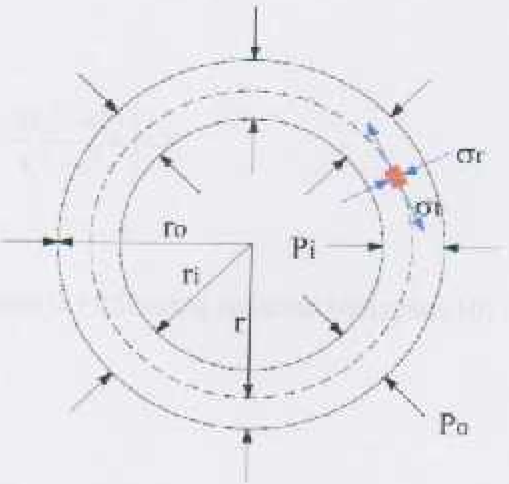


Figure 4.1 stresses, internal and external pressures

$$\sigma_t = \frac{P_i r_i^2 - P_o r_o^2 - r_i^2 r_o^2 (P_o - P_i)/r^2}{r_o^2 - r_i^2} \quad (4.1)$$

$$\sigma_r = \frac{P_i r_i^2 - P_o r_o^2 + r_i^2 r_o^2 (P_o - P_i)/r^2}{r_o^2 - r_i^2} \quad (4.2)$$

Where :

σ_t : tangential stress (MPa).

σ_r : radial stress (MPa).

P_i : inner pressure (Mpa).

P_o : outer pressure (Mpa).

r_i^2 : inner diameter (m).

r_o : outer diameter (m).

r : mean diameter (m).

The design should be taken at inner diameter ($r = r_i$), because it is the critical point (dangerous point). And at ($P_o = 0$) as usual positive indicate tension and negative compression. The equation becomes as the following [reference 10] :

At $r = r_i$, and $P_o = 0$, then:

$$\sigma_t = \left[\frac{r_i^2 + r_o^2}{r_o^2 - r_i^2}\right] \times P_i \quad (4.3)$$

$$\sigma_r = \left[\frac{r_i^2 - r_o^2}{r_o^2 - r_i^2}\right] \times P_i = \left[\frac{-(r_o^2 - r_i^2)}{r_o^2 - r_i^2}\right] \times P_i \quad (4.4)$$

$$\sigma_r = -P_i$$

σ_t can be calculated from the following equation [reference 10] :

$$\sigma' = \sqrt{\sigma_t^2 - \sigma_t \sigma_r + \sigma_r^2} \quad (4.5)$$

$$n = \frac{S_y}{\sigma'} \quad (4.6)$$

Where :

σ' : Von Mises stress (equivalent stress) (Mpa)[reference 10].

n : factor of safty.

if $n > 1$ the system is safe, if $n \leq 1$ the system is failure.

S_y : yield strength (for copper 70 Mpa).

To calculate the thickness of the pipe according to the previous equations, the internal diameter of the pipe should be calculated, by knowledge the velocity of the refrigerant inside pipes . (table A-4).

4.2 Pipe design in high pressure side

After compressor the pressure is 9 bar and pipes in this side must be tolerated this pressure, and the internal diameter can be determined by the following [reference11]

$$Q = VA \tag{4.7}$$

Where :

Q : flow rate (m^3/s).

V : velocity of the refrigerant (m/s)

(see appendix A, table A-4).

A:cross sectional area (m^2).

So

$$Q = \dot{m}\,v \tag{4.8}$$

Where :

$\dot{m}$: mass flow rate of refrigerant (0.063kg/s).

v :specific volume at compressor outlet (0.027 m^3/kg) (table A-3).

$Q = 0.063 \times 0.027 = 1.701 \times 10^{-3}\, m^3 / s.$

By substitution in equation (4.7)

$$A = \frac{Q}{V} = \frac{1.701 \times 10^{-3}}{12} = 1.4175 \times 10^{-4}\, m^2$$

$$A = \frac{\pi d^2}{4} \Rightarrow d = \sqrt{\frac{4 \times 1.4175 \times 10^{-4}}{\pi}} = 11.2 \times 10^{-3}\, m$$

the interanal diameter of the pipe in high pressure side ($d_i = 11.2\,mm$).

By substitution in equation (4.6) [reference 10]

$$n = \frac{S_y}{\sigma'}$$

let $n = 2$

$$\sigma' = \frac{70}{2} = 35\,Mpa = 350\,bar$$

By substitution in equation (4.5)[reference 10]

$$\sigma' = \sqrt{\sigma_t^2 - \sigma_t\sigma_r + \sigma_r^2}\ ,\ \sigma_r = -P_i = -9\,bar$$

$$350 = \sqrt{\sigma_t^2 + 9\sigma_t + 81}$$

$$\sigma_t^2 + 9\sigma_t = 122419$$

$$\sigma_t = 349.913\,bar$$

from equation (4.3)[reference 10]

$$\sigma_t = \left[\frac{r_i^2 + r_o^2}{r_o^2 - r_i^2}\right] \times P_i$$

$$r_i = \frac{11.2}{2} = 5.6mm$$

$$349.913 = \frac{(5.6 \times 10^{-3})^2 + r_o^2}{r_o^2 - (5.6 \times 10^{-3})^2} \times 9$$

$$r_o^2 = 7.3 \times 10^{-3}\, m = 7.3mm$$

the thickness (t) of the pipe in high pressure side is

$$t = r_o - r_i = 7.3 - 5.6 = 1.7mm$$

The internal and outer diameter in high pressure side (9 bar) in inch

$$d_{i,inch} = \frac{d_{i,mm}}{25.4} = \frac{11.2}{25.4} = 0.45inch$$

$$d_{o,inch} = \frac{d_{o,mm}}{25.4} = \frac{14.6}{25.4} = 0.58inch$$

4.3 Pipe design in low pressure side

Like equation used in pipe design in high pressure side, it is used for low pressure side (1.4 bar).

Using equation (4.7)

$Q = VA$

equation (4.8)

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

$\dot{m} = 0.063\,\text{kg/s}$

$v = 0.17\ \text{m}^3/\text{kg}$ (table A-3).

$Q = 0.063 \times 0.17 = 10.71 \times 10^{-3}\,\text{m}^3/\text{s}$

$$A = \frac{Q}{V} = \frac{10.71 \times 10^{-3}}{10} = 1.071 \times 10^{-3}\,\text{m}^2$$

$$d_i^2 = \frac{4 \times A}{\pi} \Rightarrow d_i = \sqrt{\frac{4 \times 1.071 \times 10^{-3}}{\pi}} = 11.7 \times 10^{-3}\,\text{m} = 11.7\text{mm}$$

equation (4.6)

$$n = \frac{S_y}{\sigma'}$$

let $n = 2$, S_y 70 Mpa (copper).

$$\sigma' = \frac{70}{2} = 35\,\text{Mpa} = 350\,\text{bar}$$

by substitution in equation (4.5)

$$\sigma' = \sqrt{\sigma_t^2 - \sigma_t\sigma_r + \sigma_r^2}\ ,\ \sigma_r = -P_i = 1.4\,\text{bar}.$$

$$350 = \sqrt{\sigma_t^2 + 42\sigma_t + 1764}$$

$$\sigma_t^2 + 42\sigma_t = 120736$$

$$\sigma_t = 654.21\,\text{bar}$$

from equation (4.3)

$$\sigma_t = \left[\frac{r_i^2 + r_o^2}{r_o^2 - r_i^2}\right] \times P_i$$

$$r_i = \frac{11.7}{2} = 5.85\,\text{mm}$$

$$654.21 = \frac{(5.85\times10^{-3})^2 + r_o^2}{r_o^2 - (5.85\times10^{-3})^2} \times 1.4$$

$$r_o^2 = 7.3\times10^{-3}\,\text{m} = 7.3\,\text{mm}$$

the thickness (t) of the pipe in low pressure side is

$$t = r_o - r_i = 7.3 - 5.85 = 1.45\,\text{mm}$$

The internal and outer diameter in low pressure side (1.4 bar) in inch

$$d_{i,inch} = \frac{d_{i,mm}}{25.4} = \frac{11.7}{25.4} = 0.46\,\text{inch}$$

$$d_{o,inch} = \frac{d_{o,mm}}{25.4} = \frac{14.6}{25.4} = 0.58\,\text{inch}$$

4.4 Pipe Selection

By referring to copper hand book (table 5), the suitable type selected is ACR type (Air-conditioning and Refrigeration Field Service), and according to pervious calculations, the following result is for pipe selection:

4.4.1 In high pressure side

Nominal or standard size (inches) for this section is 5/8 D which has outer diameter 0.625 inch, and inside diameter 0.545 inch, and wall thickness 0.040 inch. (table 6). According to this result the calculations of pipe in high pressure side is very closed to the selection result.

4.4.2 In low pressure side

Nominal or standard size (inches) for this section is 5/8 D which has outer diameter 0.625 inch, and inside diameter 0.545 inch, and wall thickness 0.040 inch. (table 6). According this result the calculations of pipe in low pressure side is near to the this selection data.

4.5 Calculation of evaporator pipes

$$Qe = U * A *(\Delta T)$$

$$= U * 2\pi rL * (\Delta T)$$

Where:

U : over all heat transfer coefficients($w/m^2.c^o$)

A: area of gas pipe

ΔT: the change in temperature(c^o)

r: radius of evaporator pipe(m)

L: the long of pipe(m)

L= Qe/ (U*ΔT*2πr)

L= 823.5 W.m / (23.5w/m^2.c^o*21c^o*2*3.14*0.00775m)

L= 823.5 / 48.04

L= 17m

The area of the evaporator is 0.5m*0.35m and by calculation we find that we need 34 small pipe on long 0.5m for each one, where the diameter of every pipe = 3/16 inch which is equal 4.67mm.

And the diameter of the collector pipes is = 5/8inch

$$Q_{pip} = U * A *(\Delta T)$$

= (U*ΔT*2πrL)

= 23.5*21*0.0468

Qpip = 23w for each pipe

Qtot. = 23*34 = 782 w

This value is closed to the evaporator value.=823w

CHAPTER FIVE

ELECTRICAL CONTROL

This chapter show the electrical and power cycle of the ice maker

5.1 Power circuit of the ice maker

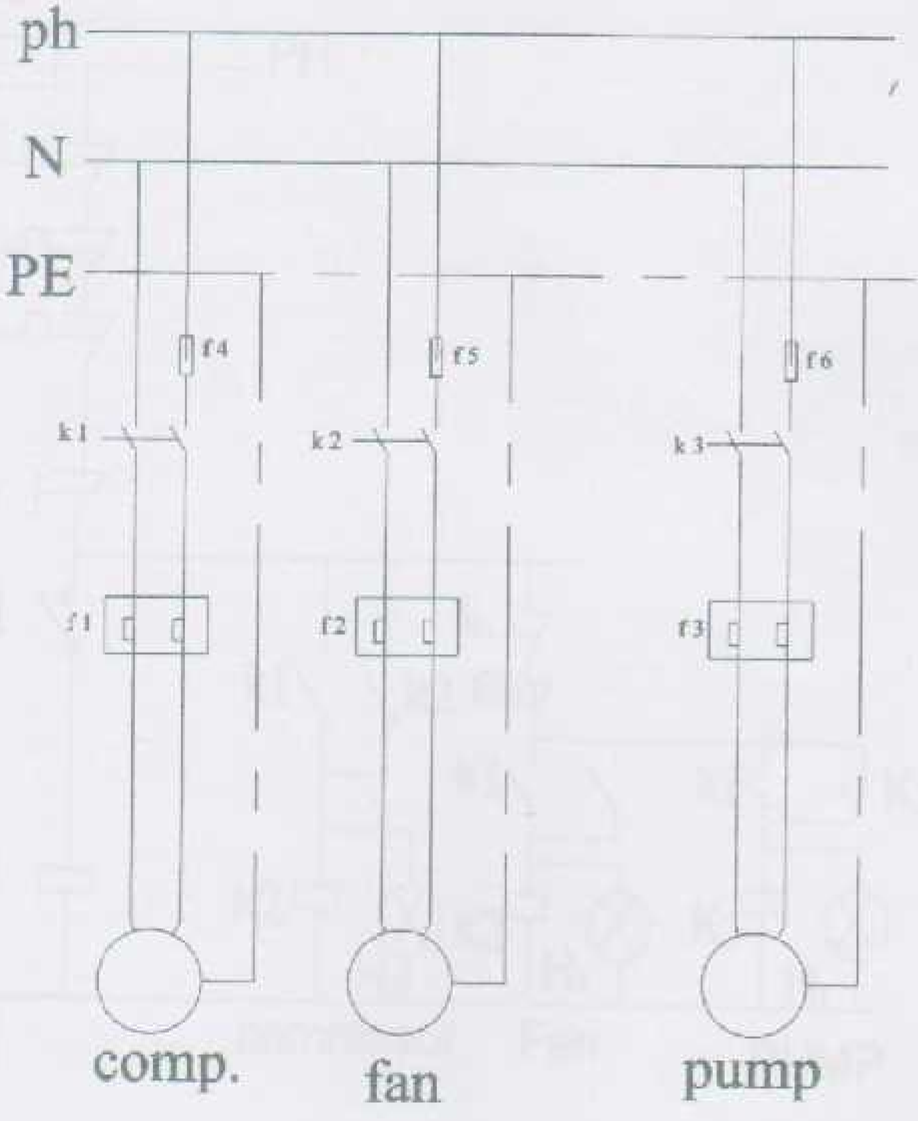


Fig (5.1) power circuit of the ice maker

5.2 Electrical control circuit

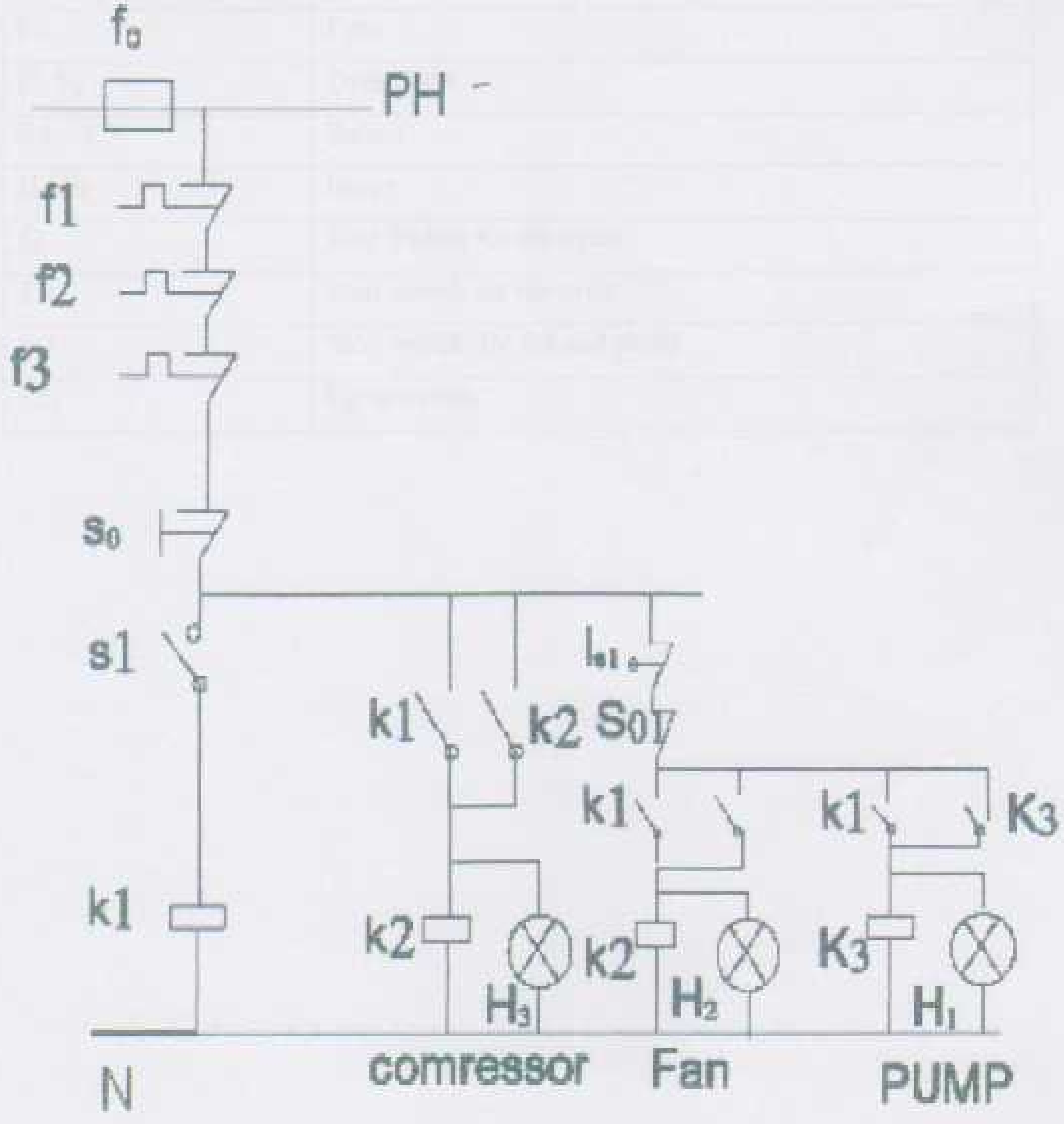


Fig (5.2) Electrical control circuit of the ice maker

Table of symbols

Table (5.1) symbol of the electrical and power circuits

symbols	
F_0	Fuse
$F_1 . F_6$	Over loads
$K1 - K3$	Relays
$H_1 . H_3$	lamps
S_0	Stop Switch for the cycle
S_1	Start switch for the cycle
S_{01}	Stop switch for fan and pump
LS_1	Limit switch

APPENDIX

Table 1

Table of specific heat capacities

	Phase	C_p J/(g·K)	$C_{p,m}$ J/(mol·K)	$C_{v,m}$ J mol^{-1}·K^{-1}	Volumetric heat capacity J/(cm^3·K)
Air (Sea level, dry, 0 °C)	gas	1.0035	29.07	20.7643	0.001297
Air (typical room conditions^)	gas	1.012	29.19	20.85	
Aluminium	solid	0.897	24.2		2.422
Ammonia	liquid	4.700	**80.08**		3.263
Animal (and human) tissue[15]	mixed	3.5	—		3.7*
Antimony	solid	0.207	25.2		1.386
Argon	gas	0.5203	20.7862	12.4717	
Arsenic	solid	0.328	24.6		1.878
Beryllium	solid	1.82	16.4		3.367
Bismuth[16]	solid	0.123	25.7		1.20
Cadmium	solid	0.231	—		—
Carbon dioxide CO_2[14]	gas	0.839*	36.94	28.46	
Chromium	solid	0.449	—		—
Copper	solid	0.385	24.47		3.45
Diamond	solid	0.5091	6.115		1.782
Ethanol	liquid	2.44	112		1.925
Gasoline	liquid	2.22	228		1.64
Glass[16]	solid	0.84			
Gold	solid	0.129	25.42		2.492
Granite[16]	solid	0.790			2.17
Graphite	solid	0.710	8.53		1.534
Helium	gas	5.1932	20.7862	12.4717	
Hydrogen	gas	14.30	28.82		
Hydrogen sulfide H_2S[14]	gas	1.015*	34.60		
Iron	solid	0.450	25.1[citation needed]		3.537
Lead	solid	0.129	26.4		1.44
Lithium	solid	3.58	24.8		1.912
Magnesium	solid	1.02	24.9		1.773
Mercury	liquid	0.1395	27.98		1.888
Methane at 2 °C	gas	2.191			
Nitrogen	gas	1.040	29.12	20.8	
Neon	gas	1.0301	20.7862	12.4717	

Oxygen	gas	0.918	29.38		
Paraffin wax	solid	2.5	900		2.325
Polyethylene (rotomolding grade)[17]	solid	2.3027			
Polyethylene (rotomolding grade)[17]	liquid	2.9308			
Silica (fused)	solid	0.703	42.2		1.547
Silver[16]	solid	0.233	24.9		2.44
Tin	solid	0.227	—		—
Tungsten[16]	solid	0.134	24.8		2.58
Uranium	solid	0.116	27.7		2.216
Water at 100 °C (steam)	gas	**2.080**	**37.47**	**28.03**	
Water at 25 °C	liquid	**4.1813**	**75.327**	**74.53**	**4.1796**
Water at 100 °C	liquid	**4.1813**	**75.327**	**74.53**	**4.2160**
Water at −10 °C (ice)[16]	solid	**2.05**	**38.09**		**1.938**
Zinc[16]	solid	0.387	25.2		2.76

Table 2

The following table shows the latent heats and change of phase temperatures of some common fluids and gases.

Substance	Latent Heat Fusion kJ/kg	Melting Point °C	Latent Heat Vaporization kJ/kg	Boiling Point °C
Alcohol, ethyl	108	-114	855	78.3
Ammonia	339	−75	1369	−33.34
Carbon dioxide	184	−78	574	−57
Helium			21	−268.93
Hydrogen(2)	58	−259	455	−253
Lead[3]	24.5	327.5	871	1750
Nitrogen	25.7	−210	200	−196
Oxygen	13.9	−219	213	−183
R134a		−101	215.9	−26.6
Toluene		−93	351	110.6
Turpentine			293	
Water	334	0	2260 (at 100°C)	100

Table 3 Thermal Conductivity of some materials.

Material	Thermal Conductivity (k) (W/m K)
Cupper pipe	120
Stainless steel	20
ice	2.2
Air	0.025

Table A-4 Recommended Velocities

Line	Refrigerant		Recommended Velocity (m/s)
Suction	R12	R22	8-12
	R134a		10-20
Discharge	R12	R22	10-18
	R134a		12-25
Liquid Between Condenser and Receiver	R12	R22	1-1.25
	R134a		0.5-0.7
Discharge and Suction with Pump in System	R12	R22	0.3-0.5
	R134a		0.6-1.2

Table 5 : Copper tubes: types, standards, Applications, lengths.

Tube Type	Color Code	Standard	Application[1]	Commercially Available Lengths[2] Nominal or Standard Sizes	Drawn	Annealed
TYPE K	Green	ASTM B 88[3]	Domestic Water Service and Distribution, Fire Protection, Solar, Fuel/Fuel Oil, HVAC, Snow Melting, Compressed Air, Natural Gas, Liquified Petroleum (LP) Gas, Vacuum	STRAIGHT LENGTHS:		
				¼-inch to 8-inch	20 ft	20 ft
				10-inch	16 ft	18 ft
				12-inch	12 ft	12 ft
				COILS:		
				¼-inch to 1-inch	—	60 ft
					—	100 ft
				1¼-inch and 1½-inch	—	60 ft
				2-inch	—	40 ft
					—	45 ft
TYPE L	Blue	ASTM B 88	Domestic Water Service and Distribution, Fire Protection, Solar, Fuel/Fuel Oil, Natural Gas, Liquified Petroleum (LP) Gas, HVAC, Snow Melting, Compressed Air, Vacuum	STRAIGHT LENGTHS:		
				¼-inch to 10-inch	20 ft	20 ft
				12-inch	16 ft	18 ft
				COILS:		
				¼-inch to 1-inch	—	60 ft
					—	100 ft
				1¼-inch and 1½-inch	—	60 ft
				2-inch	—	40 ft
					—	45 ft
TYPE M	Red	ASTM B 88	Domestic Water Service and Distribution, Fire Protection, Solar, Fuel/Fuel Oil, HVAC, Snow Melting, Vacuum	STRAIGHT LENGTHS:		
				¼-inch to 12-inch	20 ft	N/A
DWV	Yellow	ASTM B 306	Drain, Waste, Vent, HVAC, Solar	STRAIGHT LENGTHS:		
				1¼-inch to 8-inch	20 ft	N/A
ACR	Blue	ASTM B 280	Air Conditioning, Refrigeration, Natural Gas, Liquified Petroleum (LP) Gas, Compressed Air	STRAIGHT LENGTHS:		
				⅜-inch to 4⅛-inch	20 ft	*
				COILS:		
				⅛-inch to 1⅝-inch	—	50 ft
OXY, MED, OXY/MED, OXY/ACR, ACR/MED	(K)Green (L)Blue	ASTM B 819	Medical Gas, Compressed Medical Air, Vacuum	STRAIGHT LENGTHS:		
				¼-inch to 8-inch	20 ft	N/A

Table 6: Dimensions and Physical Characteristics of Copper Tube ACR (Air - Conditioning and Refrigeration Fields Service).
(A: Annealed Temper, D: Drawn Temper).

Nominal or Standard Size, inches		Nominal Dimensions, inches			Calculated Values (based on nominal dimensions)				
		Outside Diameter	Inside Diameter	Wall Thickness	Cross Sectional Area of Bore, sq inches	External Surface, sq ft per linear ft	Internal Surface, sq ft per linear ft	Weight of Tube Only, pounds per linear ft	Contents of Tube, cu ft per linear ft
1/8	A	.125	.065	.030	.00332	.0327	.0170	.0347	.00002
3/16	A	.187	.128	.030	.0129	.0492	.0335	.0575	.00009
1/4	A	.250	.190	.030	.0284	.0655	.0497	.0804	.00020
5/16	A	.312	.248	.032	.0483	.0817	.0649	.109	.00034
3/8	A	.375	.311	.032	.076	.0982	.0814	.134	.00053
	D	.375	.315	.030	.078	.0982	.0821	.126	.00054
1/2	A	.500	.436	.032	.149	.131	.114	.182	.00103
	D	.500	.430	.035	.145	.131	.113	.198	.00101
5/8	A	.625	.555	.035	.242	.164	.145	.251	.00168
	D	.625	.545	.040	.233	.164	.143	.285	.00162
3/4	A	.750	.680	.035	.383	.196	.178	.305	.00252
	A	.750	.666	.042	.348	.196	.174	.362	.00242
	D	.750	.686	.042	.348	.196	.174	.362	.00242
7/8	A	.875	.785	.045	.484	.229	.206	.455	.00336
	D	.875	.785	.045	.484	.229	.206	.455	.00336
1 1/8	A	1.125	1.025	.050	.825	.294	.268	.655	.00573
	D	1.125	1.025	.050	.825	.294	.268	.655	.00573
1 3/8	A	1.375	1.265	.055	1.26	.360	.331	.884	.00875
	D	1.375	1.265	.055	1.26	.360	.331	.884	.00875
1 5/8	A	1.625	1.505	.060	1.78	.425	.394	1.14	.0124
	D	1.625	1.505	.060	1.78	.425	.394	1.14	.0124
2 1/8	D	2.125	1.985	.070	3.09	.556	.520	1.75	.0215
2 5/8	D	2.625	2.465	.080	4.77	.687	.645	2.48	.0331
3 1/8	D	3.125	2.945	.090	6.81	.818	.771	3.33	.0473
3 5/8	D	3.625	3.425	.100	9.21	.949	.897	4.29	.0840
4 1/8	D	4.125	3.905	.110	12.0	1.08	1.02	5.38	.0833

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