Palestine Polytechnic University

College of Engineering & Technology
Mechanical Engineering Department

Graduate Project

Design of Automotive Air Conditioning unit working on R744 (CO_2)

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Hebron – Palestine

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Palestine Polytechnic University
(PPU)
Hebron-Palestine

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Supervisor Signature

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Examine community Signature

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Dedication

We gift this graduation project
To who carry candle of science
To light his avenue
Of live

To all students & who
Wish to look for
The future

To who love the knowledge &
Looking for all is new
In this world

Acknowledgement

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Abstract

In this project we will design an air conditioning unit for private car working on refrigerant R744 (CO_2), because it is an environmental gas. Other refrigerants used in automobile air conditioning like (R12, and R134a) have a dangerous pollution in the world, R12 effects on ozone layer and caused ozone depletion.
R134a can be flammable, and it has high global warming potential effect. CO_2 has no ozone depletion, and it has smaller effect on global warming potential than other gases.

In this project we will talk about CO_2 properties, and the cycle components which we need in the design of the project, and we will do the calculations required to build up the components in the future.

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CHAPTER ONE

INTRODUCTION

Chapter one

Introduction

1.1 General outlook

Today, as we drive our automobiles, many of us, can enjoy the same comfort levels that we are accustomed to at home and at work. With the push of a button or the slide of a lever, we make the seamless transition from heating to cooling and back again without ever wondering how this change occurs.

The rapid advancement of technology in the manufacturing of automobiles had led to a level of aesthetics and functions that makes the owner feel almost at home. The emissions from internal combustion engines and the refrigerants used in the air-conditioning systems increase the rate of pollution and are a leading cause of the destruction of the ozone layer. Furthermore, the CO, NOx emissions greatly contribute to the 'greenhouse' gases responsible for global warming.

With the increase of the level of pollution and the number of automobiles, scientists are becoming ever more concerned with the future health of the world and its inhabitants. With the increase of worry regarding world pollution has lead to an explosion of funding towards the research of more efficient and environmentally friendly systems to develop emissions-free automotive solution. A possible solution to the problems caused by refrigerants used today could be the use of the CO_2 in the A/C systems used in automobiles.

Carbon Dioxide (CO_2) as a refrigerant is environmentally friendly gas whilst maintaining the same efficiently levels as current refrigerants used today. CO_2 is also an inert gas and therefore does not affect the ozone layer in any way. Furthermore, the gas is also readily available in nature and only requires simple purification and containment for this application which is already available in the firefighting industry.

1.2 Previous Studies

Carbon dioxide was a commonly used refrigerant from the late nineteenth and well into the twentieth century. Due to its complete harmlessness it was the generally preferred choice for usage on board ships, while ammonia was more common in stationary applications. By the advent of the 'Freon's' and R-12 in the first place, CO_2 was rapidly abandoned, and it has nearly been forgotten in the course of the last 40-50 years. The main reasons for this development were certainly the rapid loss of capacity at high cooling water temperatures in the tropics, and not less the failure of the manufacturers to follow modern trends in compressor design towards more compact and price effective high speed types. Time is now ripe for a reassessment of this refrigerant for application with present day technology.

Although CO_2 (R-744) was widely used as refrigerant in the early 20th century, its use disappeared from around 1940 with the advent of the fluorocarbon chemicals.

Thus, when Professor Gustav Lorentzen at NTNU/SINTEF in the late 1980s proposed to reconsider the use of CO_2, it had been absent for almost half a century.

Then CO_2 systems have proven to be viable alternatives in several applications in the early 1990s. The European RACE project from 1994 to 1997 included development and testing of car-installed prototype systems, with results confirming the potential for CO2-based car air conditioning.

Then a full-scale laboratory prototype system of 50 kW heating capacity was completed in 1996. From 1996 to 1998 various aspects of heat pumping applications for CO_2 were developed. A 25 kW pilot plant was installed in a food-processing factory in Larvik, Norway in 1999. Several Japanese companies have introduced heat pump water heaters for residential applications into the market during the last years. Europe has also been reported with a possible market launch in 2004.

Over the last years, the German Motor Vehicle Industry Association (VDA) has coordinated development and testing of CO2 systems, and several car manufacturers have had test vehicles on the road since the late nineties.

A research team in Ixetic Company leaded by Willi Parsch developed CO2 compressor. In 1996 the project has been started, through the next four years; the team puts the principal investigation for components (shaft seal, piston ring, efficiency controllability) , from 2000-2005 puts the Definition of final concept, fleet testing. From year 2005 the company started marketing the CO2 compressor to automotives air conditioning industrials.

1.3 Project Objective:

To design an Automotive Air-Conditioning system that utilizes CO_2 as an environmentally friendly refrigerant. In our project we design the prototype of air conditioning unit for a private car.

Figure (1.1) represents the project. And in next chapters we explain more about the project components.

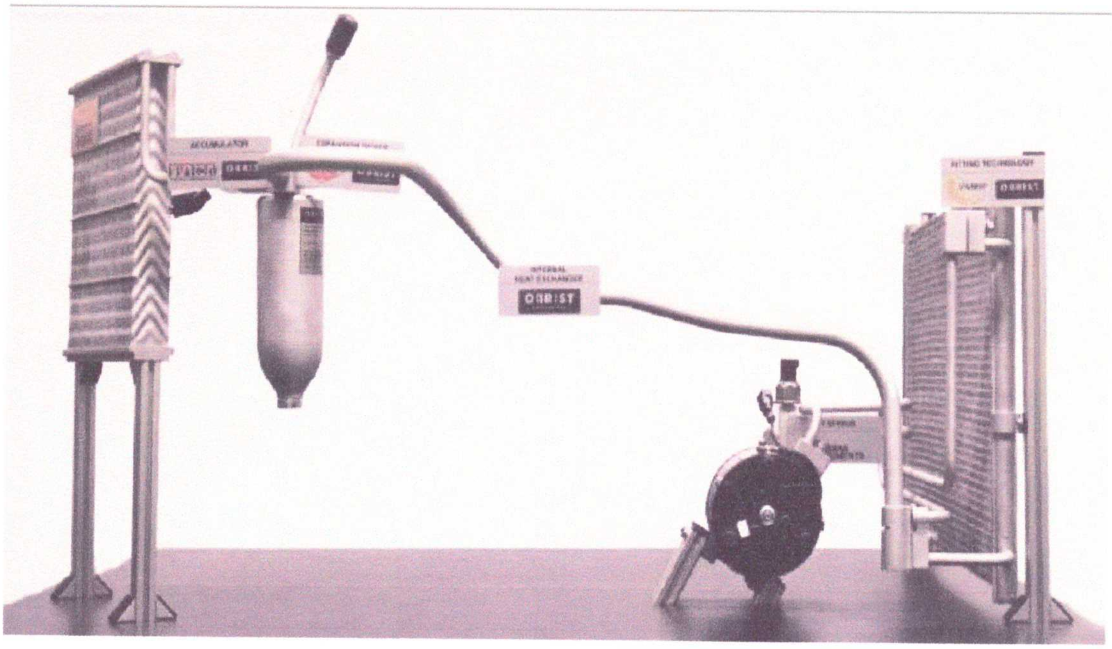


Figure 1.1 The project

1.4 Importance of the project

The importance of this project is:

- It can be used as a learning model for the students who come after us to study air conditioning & refrigeration engineering when the project build in the future.
- To protect the environment from pollution such as, ozone depletion, and Global warming potential.
- There are an economic benefits by saving the energy from fuel consumption .
- Reducing the size of air conditioning parts in the car.

1.5 Project Layout

The layout of this project consist of the following chapters:

Chapter 1: Introduction

- Feasibility research.
- Research past solutions via patent search and online investigation.
- Research current projects by the automotive industry and other universities.
- Research the state of the art of this solution as well researching future developments in relating technologies.
- Time tables and Budget.

Chapter 2: Cycle component

- Definition the refrigerant and mention its properties.
- Explain the Cycle component (compressor, condenser, evaporator,...etc) .

Chapter 3: Calculations

- Calculations of specifications
- Calculation of cooling load inside car.
- Drawing a schematic design of the cycle.
- Drawing the cycle on R744 charts.

Chapter 4: Design and Selection of the pipes in the cycle.

- According to calculations pipes will be selected.

Chapter 5: Design of heat exchangers.

- Gas cooler design.
- Coil design.
- Heat exchanger design.
- Evaporator design.

Chapter 6: Electrical and control design

- Power circuit.
- Control circuit.
- Electrical motor calculation and selection.

1.6 Budget

Table 1.1 Budget

TASK	COST (NIS)
Researches	300
Transportations	200
To copy from library	50
Printing papers	60
Reprinting papers	150
TOTAL	760

1.7 Time Planning

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

The following time plan is for the first semester

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																
Calculations																
Project Documentation																

Figure1.2 The First Time Plan

The following time plan is for the second semester

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

Figure 1.3 The Second Time Plan

CHAPTER TWO

REFRIGERANT AND CYCLE CMPONENTS

Chapter Two

Refrigerant and Cycle Components

2.1 Refrigerant

2.1.1 The natural refrigerant R744 (CO2)

In the early days of refrigeration the two refrigerants in common use were ammonia and carbon dioxide. Both were problematic - ammonia is toxic and carbon dioxide requires extremely high pressures (from around 30 to 200 atmospheres!) to operate in a refrigeration cycle, and since it operates on a Transcritical cycle the compressor outlet temperature is extremely high (around 160°C). When Freon 12 (dichlorodifluoromethane) was discovered it totally took over as the refrigerant of choice. It is an extremely stable, non toxic fluid, which does not interact with the compressor lubricant, and operates at pressures always somewhat higher than atmospheric, so that if any leakage occurred, air would not leak into the system, thus one could recharge without having to apply vacuum.

Unfortunately when the refrigerant does ultimately leak and make its way up to the ozone layer the ultraviolet radiation breaks up the molecule releasing the highly active chlorine radicals, which help to deplete the ozone layer. Freon 12 has since been banned from usage on a global scale, and has been essentially replaced by chlorine free R134a (tetraflouro-ethane) - not as stable as Freon 12, however it does not have ozone depletion characteristics.

Recently, however, the international scientific consensus is that Global Warming is caused by human energy related activity, and various man made substances are defined on the basis of a Global Warming Potential (GWP) with reference to carbon dioxide (GWP=1). R134a has been found to have a GWP of 1300 and in Europe, within a few years, automobile air conditioning systems will be barred from using R134a as a refrigerant.

The new hot topic is a return to carbon dioxide as a refrigerant. The previous two major problems of high pressure and high compressor temperature are found in fact to be advantageous. The very high cycle pressure results in a high fluid density throughout the cycle, allowing miniaturization of the systems for the same heat pumping power requirements. Furthermore the high outlet temperature will allow instant defrosting of automobile windshields (we don't have to wait until the car engine warms up) and can be used for combined space heating and hot water heating in home usage.

R744 is the chemical reference for carbon dioxide (CO_2) used as refrigerant. It is a naturally occurring substance that can be applied as a working fluid in different heating and cooling applications; due to its excellent heat transfer properties and its high volumetric cooling capacity.

Carbon dioxide found popularity as a refrigerant for comfort cooling in the 1920s, before being replaced by more convenient CFCs (chlorofluorocarbons) R12 in the 1930s and then HFCs (hydroflurocarbons) in the 1990s.

2.1.2 Main characteristics of R744

- Non-toxic.
- Non-flammable.
- Non-ozone-depleting.
- Environmentally friendly, with a Global Warming Potential = 1.
- Low cost and good availability.
- Pressure close to the economically optimal level.
- Complete compatibility with normal lubricants and common machine Construction materials.
- Thermal stability.
- Simple operation and service, no 'recycling' required, very low price.

2.1.3 The main benefits of R744 in commercial refrigeration

R744-based commercial refrigeration feature:

- Reduced emissions of greenhouse gases
- Space savings for piping arrangements (self-contained system)
- Reduced costs of piping arrangements and insulation
- Energy efficiency

2.1.4 High operating pressure of R744

CO_2 has critical temperature lower than that of HFC-134a and a critical pressure higher than that of HFC-134a. Therefore, in air conditioning systems adopting CO_2 refrigerant, a high pressure side temperature exceeds the critical point. This results in a high operation pressure that 7 to 10 times larger than that of HFC-134a.

Another properties of CO_2 (R744) represented in table 2.1.

Table 2.1 properties of R744

	R744
Operational mode	Transcritical
High pressure (bar)	60–140
Low pressure (bar)	35–50
Max gas temperature	Up to 180°C
A/C line diameters	10–12
A/C line fittings type	Axial metal
Flexible hose material	Steel
Compressor displacement (cm^3)	20–33
Compressor housing diameter (mm)	100–120
Expansion device	Electronic valve or mechanical orifice
Front end heat exchanger	Gas cooler

2.1.5 CO_2 Charts

- **Carbon dioxide pressure-temperature phase diagram**

At −78.5° C, carbon dioxide changes directly from a solid phase to a gaseous phase through sublimation, or from gaseous to solid through deposition. Solid carbon dioxide is normally called "dry ice", a generic trademark. It was first observed in 1825 by the French chemist Charles Thilorier. Dry ice is commonly used as a versatile cooling agent, and it is relatively inexpensive. As it warms, solid carbon dioxide sublimes directly into the gas phase, making its use convenient as it leaves no liquid. It can often be found in groceries and laboratories, and it is also used in the shipping industry. The largest non-cooling use for dry ice is blast cleaning.

Liquid carbon dioxide forms only at pressures above 5.1 atm; the triple point of carbon dioxide is about 518 kPa at −56.6°C (See phase diagram, below). The critical point is 7,821 kPa at 31.1°C. Figure 2.1 represents Carbon dioxide pressure-temperature phase diagram.

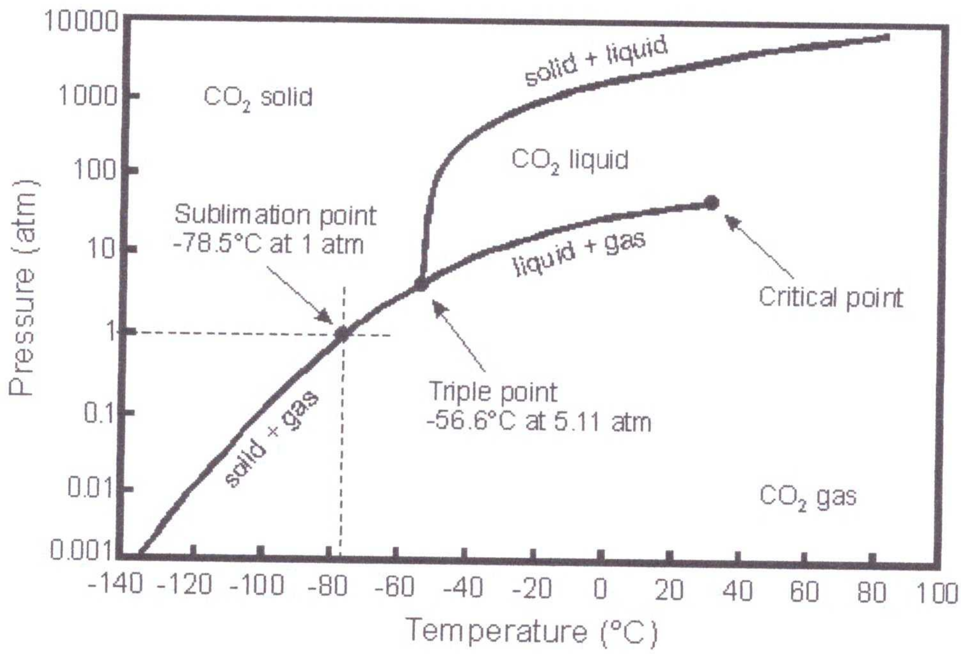


Figure 2.1 Carbon dioxide pressure-temperature phase diagram

- **Pressure- enthalpy diagram**

This diagram show the relationship between pressure (P) and enthalpy (h), at any temperature, from this chart many properties of refrigerant (R744) can be get, such that entropy (s),specific volume, temperature, pressure, and enthalpy. Figure 2.2 show the (P-h) diagram.

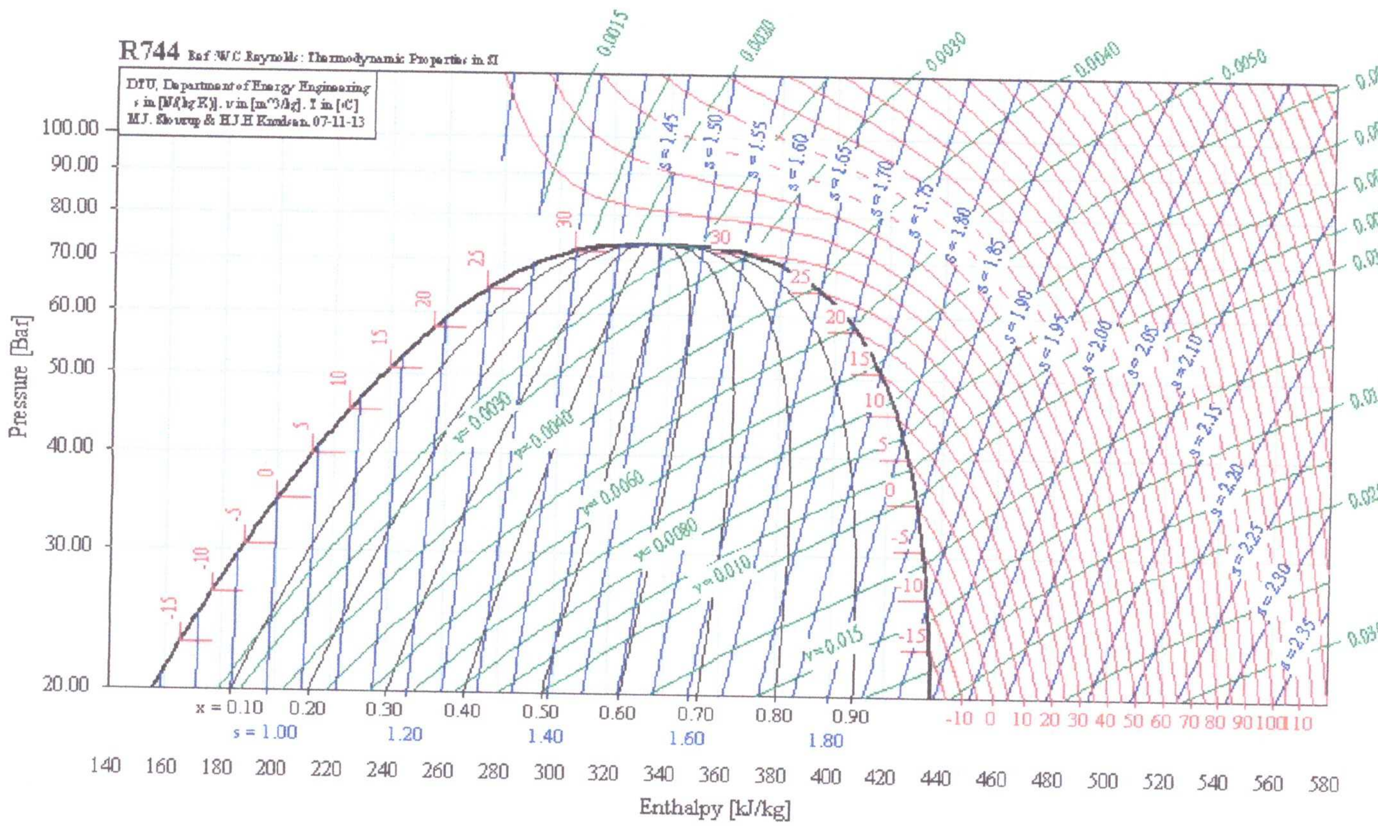


Figure 2.2 P-h diagram of R744 (CO_2)

- **Temperature –entropy diagram**

This diagram show the relationship between the temperature and the entropy, from this chart many properties of refrigerant (R744) can be get, such that pressure (P),specific volume, and enthalpy. Figure 2.3 show the (T-s) diagram.

- **Enthalpy – entropy diagram**

This diagram show the relationship between the enthalpy and the entropy, from this chart many properties of refrigerant (R744) can be get, such that pressure (P), specific volume, and temperature. Figure 2.4 show the (h-s) diagram.

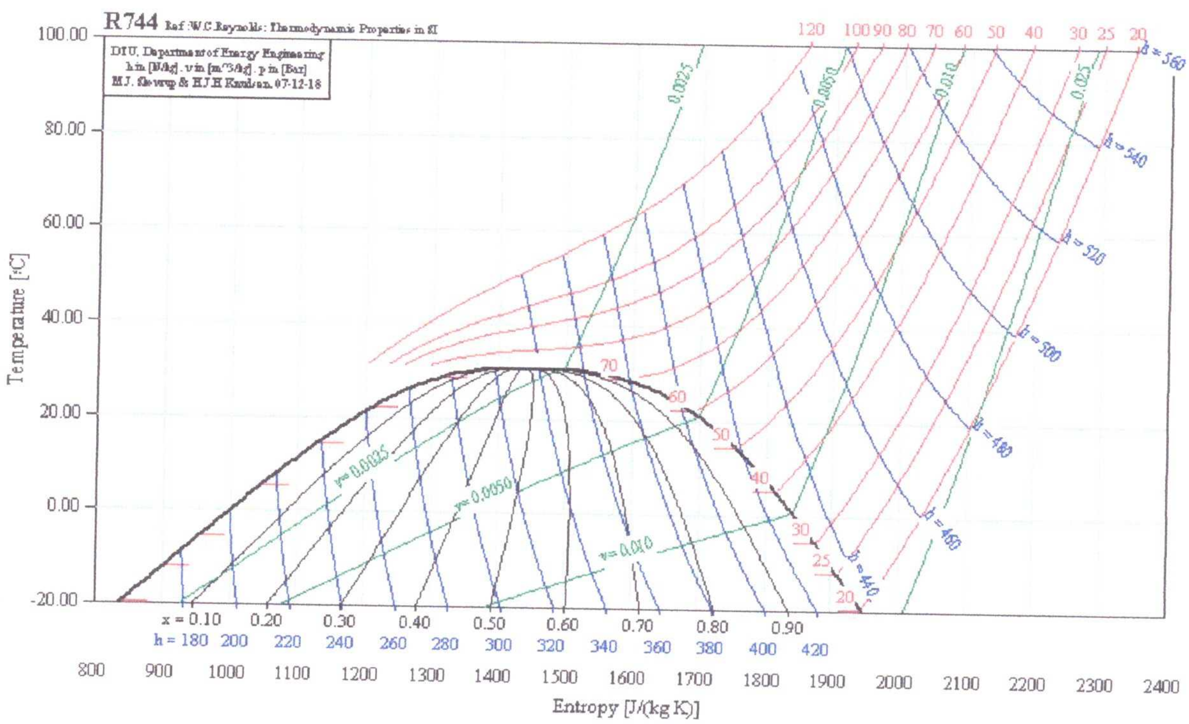


Figure 2.3 T-s diagram of R744 (CO_2)

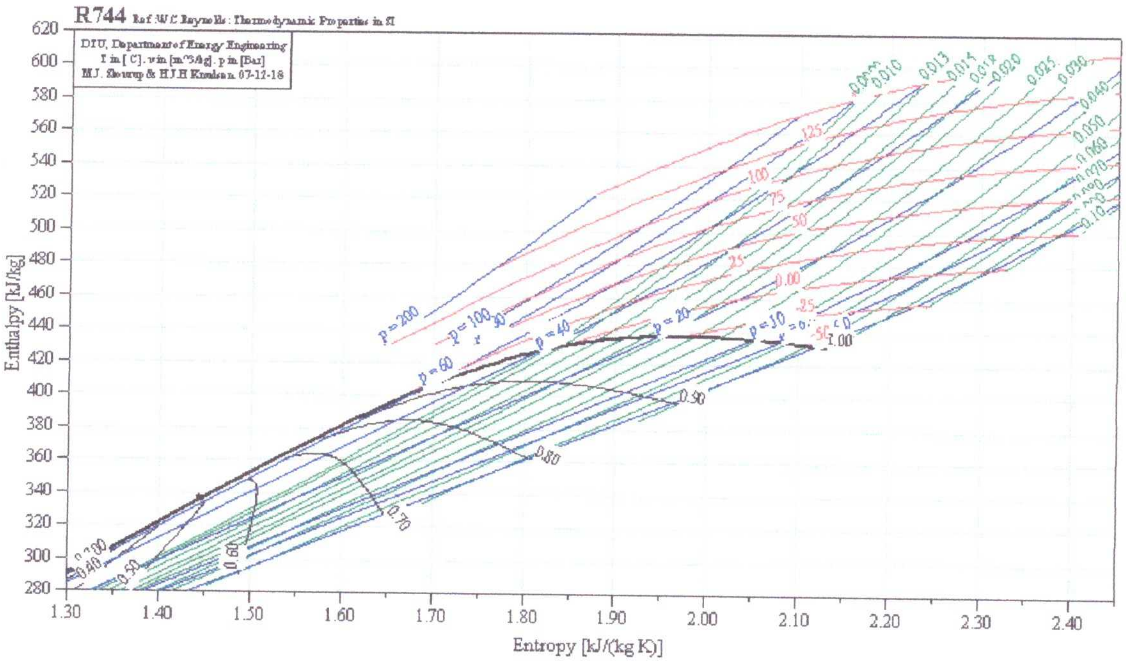


Figure 2.4 h-s diagram of R744 (CO_2)

2.1.6 Comparison with other refrigerant

In table 2.2 we compare between R744 and other refrigerants used in MAC (Mobile Air Conditioning).

Table 2.2 R744 and other refrigerants in MAC.

Refrigerant	R12	R134a	R152a	R744 / CO_2
Natural ?	No	No	No	Yes
Flammable ?	No	No	Yes	No
Combustion Product		Toxic	Toxic	-
Volum. Capacity	0.9	(1)	0.9	6.9
Critical Temp. °C (°F)	112 (234)	101(214)	113(236)	31(88)
ODP	1.0	0	0	0
Pres. at 21°C in bar (70°F) (psia)	5.9 (85)	5.9 (86)	5.3 (77)	59 (852)
GWP (100yr)	7100	1300	140	(1)
	Ozone Depletion 1990	Global Warming 2011		

Where :

ODP: Ozone Depletion Potential.

GWP: Global Warming Potential

2.2 Cycle Components

The components of this project are normally used in any refrigeration or air conditioning unit. Figure (2.5) represents the cycle components in the project, which are compressor, condenser, evaporator, expansion valve, receiver dryer, or accumulator, and fans.

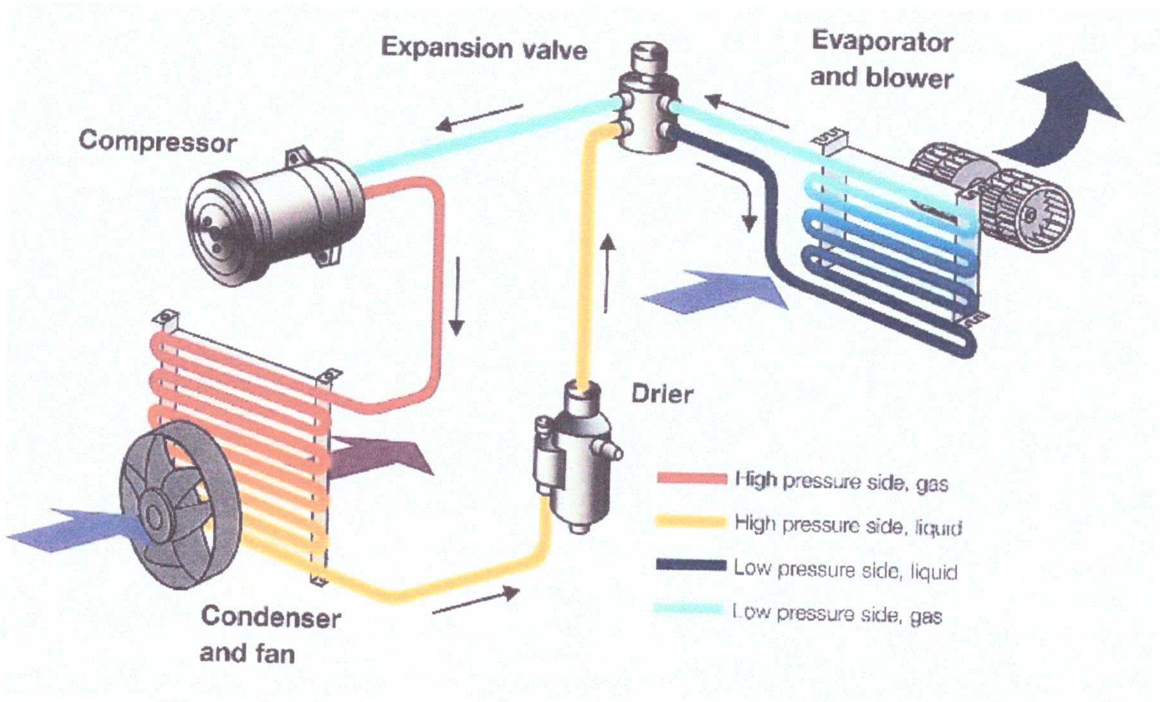


Figure 2.5 Cycle components of automotive air conditioning

2.2.1 Compressor

Commonly referred to as the heart of the system, the compressor is a belt driven pump that is fastened to the engine. It is responsible for compressing and circulating refrigerant gas.

The A/C system is split into two sides, a high pressure side and a low pressure side; defined as discharge and suction. Since the compressor is basically a pump, it must have an intake side and a discharge side. The intake, or suction side, draws in refrigerant gas from the outlet of the evaporator. In some cases it does this via the accumulator.

Once the refrigerant is drawn into the suction side, it is compressed and sent to the condenser, where it can then transfer the heat that is absorbed from the inside of the vehicle.

The A/C compressor compresses refrigerant to a high pressure (and since temperature and pressure are relative), high temperature gas which is then sent to the condenser for the process of removing the heat. The fact that this temperature is greater than the outside temperature allows the heat to transfer to the outside air. This then causes the "gaseous" refrigerant to "condense" back to a high pressure liquid. This high pressure liquid is then sent to the restriction in the system (that being an orifice, or a valve of some kind) which lowers the pressure causing the liquid to boil and vaporize (thus the term "evaporator"). It is then in this low pressure gas state that it is capable of absorbing the heat from inside the vehicle and sent back to the compressor for the cycle to repeat itself.

The compressor is lubricated with special oil. It is very important to use the correct oil to the compressor and system. Some seals and gaskets may not work with some oils. Use of the wrong refrigerant or oil can reduce system performance or even cause damage in compressor.

A compressor used in automotive air conditioning is an open type, because the driver is not in the same case of the driven. Figure 2.6 show the car air conditioning compressor (open type).

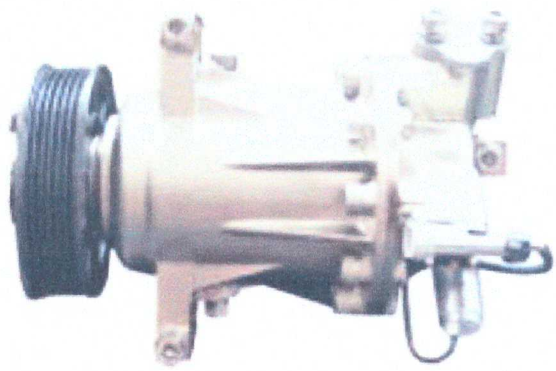


Figure 2.6 open type compressor.

2.2.1.1 Features

- Variable displacement compressor with 6 or 7 pistons.
- External electrical control valve for maximum efficiency and comfort.
- Compact compressor body/packaging for more space in the engine compartment.
- Pressure release device for enhanced safety.

2.2.1.2 Benefits

- Designed to operate with R744 environmentally friendly refrigerant.
- Complies with forthcoming EU-regulations to phase out HFC-R134a.
- Contributes to an efficient A/C system.
- No refrigerant recovery or recycling necessary.
- Optimized fuel consumption.

2.2.1.3 Compressor types using in automobile air conditioning

There are two common type of compressor used in automotive air conditioning system using R744 :

- **Swash compressor**

An inclined plate drive two pistons to compress gas, one of these pistons is in compression stroke, and another piston is in suction stroke at the same position of swash plate, at angle 180^{o} of previous position the first piston become in the suction stroke , and the another piston become in the compression stroke. Figure 2.7 represent the swash compressor.

- **CO2 Scroll Compressor**

This compressor have high volumetric efficiency, low pressure and mechanical losses, and low noise and vibrations. Figure 2.8 shows cross section in scroll compressor for CO_2 air conditioning used in automobile technology.

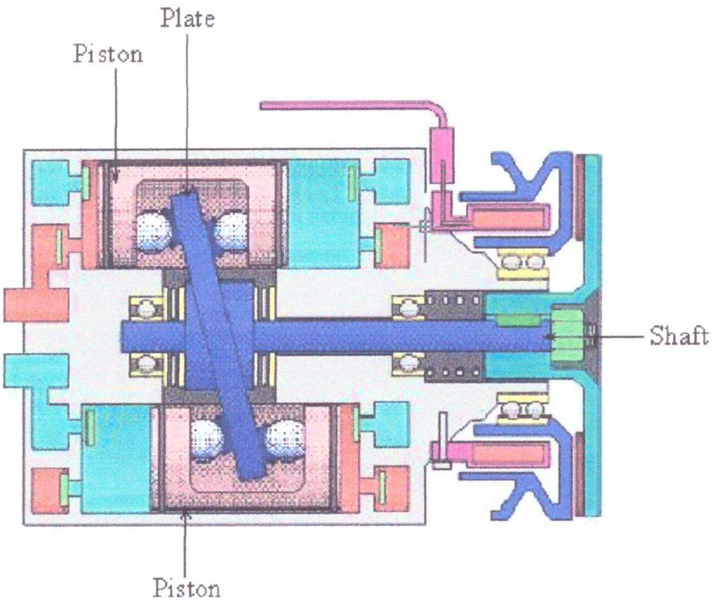


Figure 2.7 swash compressor

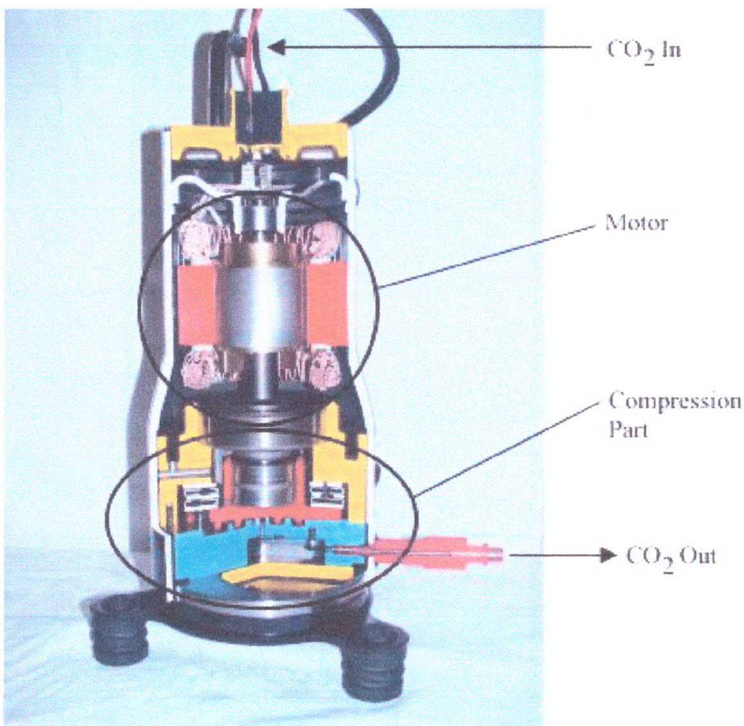


Figure 2.8 cross section in scroll compressor

Figure 2.9 shows the difference in size between CO2 compressor and R134a compressor (these pictures are in the same scale).

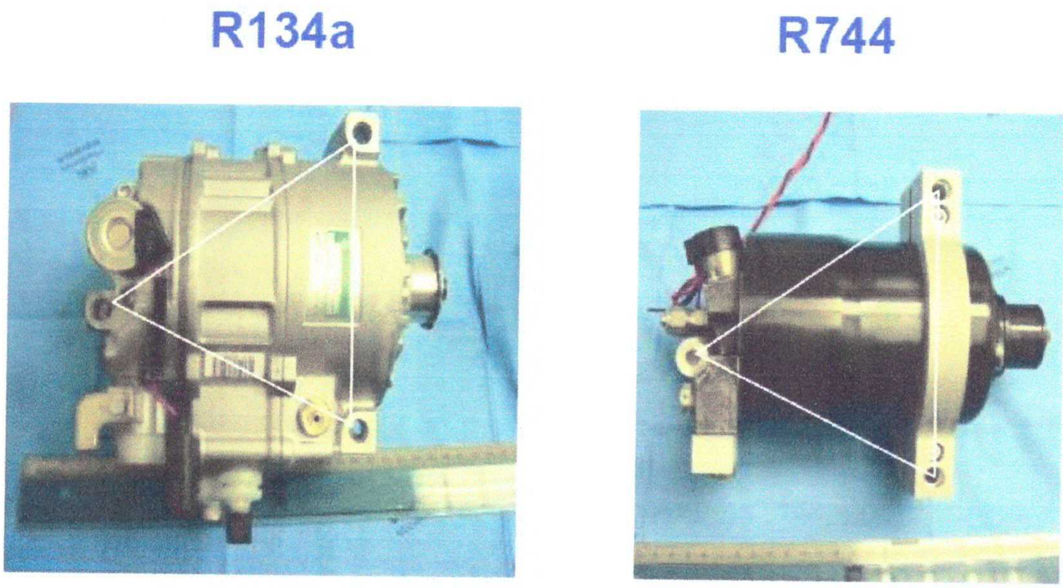


Figure 2.9 CO_2 compressor and 134a compressor

2.2.2 Condenser (Gas cooler)

The automobile air conditioning A/C condenser (gas cooler) is located in front of the radiator (and kind of looks like a radiator too). Through the use of cool air flow provided by the engine fan, the condenser cools the hot gas and converts it to liquid. The liquid is still under considerable pressure and is warm, but not as hot or as high pressure as when it exited the compressor. Output = high pressure (warm) liquid. This is the area in which heat dissipation occurs. The A/C condenser, in many cases, will have much the same appearance as the radiator in car as the two have very similar functions.

The condenser is designed to radiate heat. Its location is usually in front of the radiator, but in some cases, due to aerodynamic improvements to the body of a vehicle, its location May differ. A/C Condensers must have good air flow anytime the system is in operation.

On rear wheel drive vehicles, this is usually accomplished by taking advantage of your existing engine's cooling fan. On front wheel drive vehicles, condenser air flow is supplemented with an electric cooling fan(s).As hot compressed gasses are introduced into the top of the condenser, they are cooled off. As the gas cools, it condenses and exits the bottom of the condenser as a high pressure liquid.

In CO_2 A/C in automobile there is no condensation occur as in vapor compression cycle, since the refrigerant come from compressor as superheated gas, and leave condenser (gas cooler) as a high temperature gas (there is no phase change of refrigerant occurs in condenser).

2.2.3 Evaporator

Located inside the vehicle, the evaporator serves as the heat absorption component. The evaporator provides several functions. Its primary duty is to remove heat from the inside of your vehicle. A secondary benefit is dehumidification. As warmer air travels through the aluminum fins of the cooler evaporator coil, the moisture contained in the air condenses on its surface. Dust and pollen passing through stick to its wet surfaces and drain off to the outside. On humid days you may have seen this as water dripping from the bottom of your vehicle. Rest assured this is perfectly normal.

2.2.4 A\C Receiver Drier or Accumulator

The receiver-drier or Accumulator is used on the high side of systems that use a thermal expansion valve. This type of metering valve requires liquid refrigerant. To ensure that the valve gets liquid refrigerant, a receiver is used. The primary function of the receiver-drier is to separate gas and liquid.

The secondary purpose is to remove moisture and filter out dirt. The receiver-drier or Accumulator usually has a sight glass in the top. This sight glass is often used to charge the system. Under normal operating conditions, vapor bubbles should not be visible in the sight glass. The use of the sight glass to charge the system is not recommended in R-134a systems as cloudiness and oil that has separated from the refrigerant can be mistaken for bubbles. This type of mistake can lead to a dangerous overcharged condition.

There are variations of receiver-driers / Accumulators and several different desiccant materials are in use. Some of the moisture removing desiccants found within are not compatible with R-134a. The desiccant type is usually identified on a sticker that is affixed to the receiver-drier. Newer receiver-driers use desiccant type XH-7 and are compatible with both R-12 and R-134a refrigerants.

The receiver-drier / Accumulator must be changed each time a system is empty regardless of the reason for loss of refrigerant. It should also be changed every three years, because the desiccant pellets will break down and clog the expansion valve. This will in turn cause the system to become inoperable and may damage the compressor.

The receiver-drier is strictly a disposable item and is thought of in the same terms as a fuel, oil, or air filter. In fact, if any component fails or is replaced for any reason, the receiver-drier must also be replaced to prevent corrosion and moisture in the system.

The receiver-drier performs three functions:

- It filters the system of non-condensable.
- It receives the liquid refrigerant and maintains a certain level of liquid at the bottom at all times in a properly charged system.
- It contains a stack of pellets called desiccant (drying agent) to trap and absorb moisture. NOTE that moisture is the most harmful enemy of the air conditioning system. If any moisture is in the system, it will combine with the refrigerant to form hydrochloric acid which is extremely corrosive to metal components.

Replacing the receiver-drier is essential when servicing the A/C system. Whenever you replace a component of the A/C system you must also replace the receiver-drier. If you do not change the receiver-drier there could be serious damage

to the other parts of the system, which could be very costly. You must also have proof of changing the receiver-drier in order to receive a compressor warranty.
Figure 2.10 shows how the vapor go to compressor from the accumulator , and vapor come from evaporator, if there is any liquid the accumulator trap it.

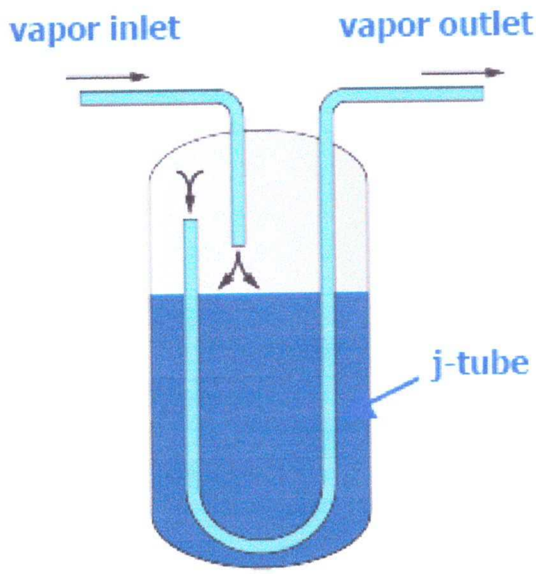


Figure 2.10 Accumulator

Drier is used before the evaporator and after the condenser. The Filter Drier is where the system will filter out small amounts of contamination and moisture from the system. ach time the system is opened or worked on the Drier should be replaced. The desiccant inside the dryer will absorb moisture. Also spelled drier.

2.2.5 Expansion Valve

The most common expansion valve used in automotive air conditioning using R744 is electrically controlled expansion valve see, figure 2.11, the pilot valve moving up and down by electrically control, and the main valve may open or close to according the pressure of refrigerant moving from pilot valve.

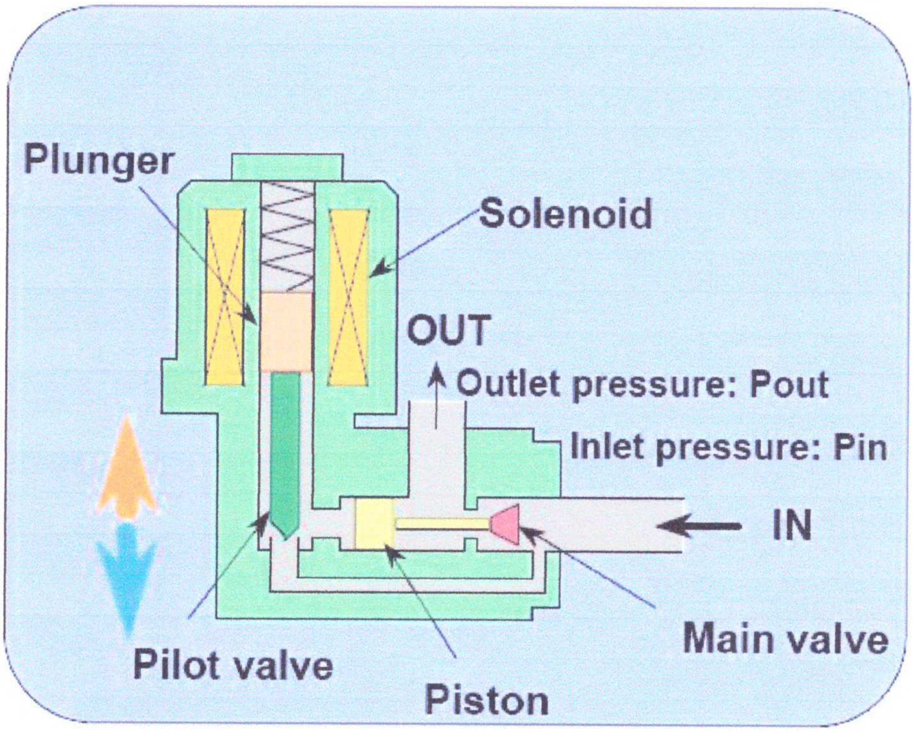


Figure 2.11 electrical expansion valve

2.2.6 Pipes

The most important properties of the pipes using in CO_2 air conditioning can hold high pressure, and the diameter is smaller and the thickness is bigger than other pipes used in normal air conditioning cycles. Figure 2.12 compares in size and thickness between pipes used in CO_2 and R134a A/C.

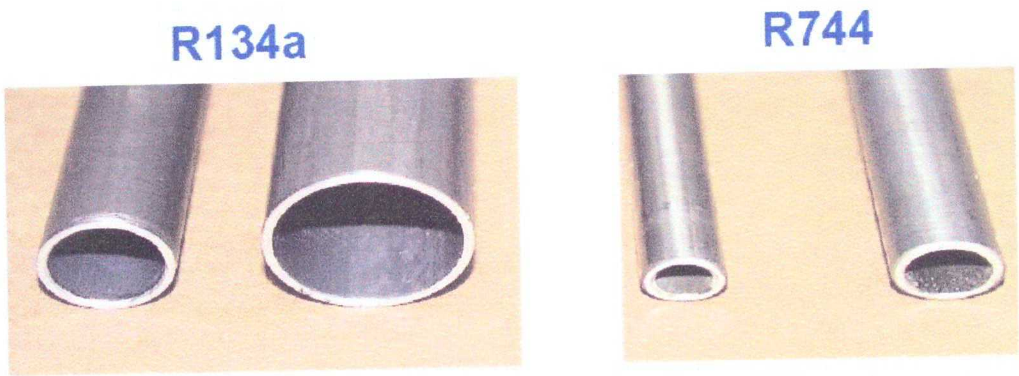


Figure 2.12 pipes in R744 and in R134a

2.2.7 Fan and blower

Fan located behind the gas cooler to produce force convection to increase heat transfer from gas cooler.

An important component in the cooling action of the evaporator is the blower motor/fan (usually the same one that blows air through the heater core), also located in the evaporator housing. The blower draws warm air from the passenger compartment over the evaporator and blows the "cooled" air out into the passenger area. The blower motor is controlled by a fan switch with settings from Low to High. High blower speed will provide the greatest volume of circulated air. A reduction in speed will decrease the air volume. But the slower speed of the circulated air will allow the air to remain in contact with the fins and coils of the evaporator for a longer period of time. The result is more heat transfer to the cooler refrigerant. Therefore, the coldest air temperature from the evaporator is obtained when the blower is operated at its slowest speed. Figure 2.13 represents evaporator and its fan blower.

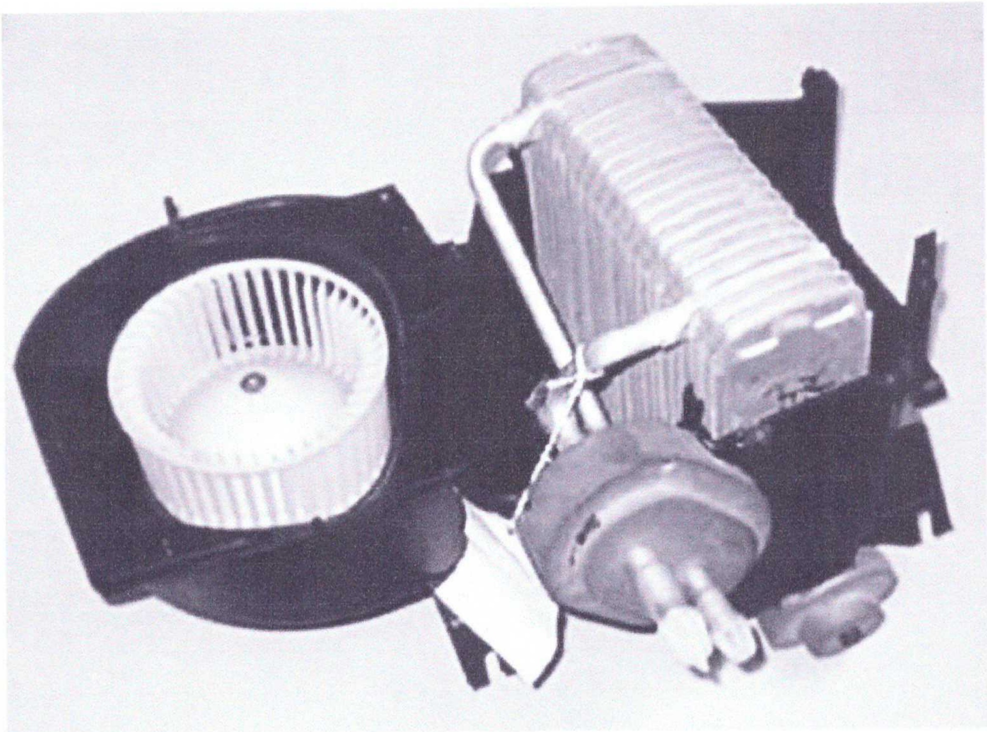


Figure 2.13 evaporator and its fan blower

2.2.8 Electrical motor

In normal case the compressor in automotive air conditioning system driven by the motor of the car, by connect it with the shaft of the motor.

In this project the compressor driven by electrical motor, this motor is selected in the next chapter.

CHAPTER THREE

CALCULATIONS of CYCLE AND COMPRESSOR DESIGN

Chapter Three

Calculations and Design

3.1 Introduction

The cycle of the project occurs below and above critical point of CO_2 , Refrigeration and air-conditioning systems where the cycle incurs temperatures and pressures both above and below the refrigerant's critical point are often called transcritical systems. Transcritical systems are somewhat similar to the subcritical systems.

Figure 3.1 illustrates the transcritical vapor compression process. It begins when the superheated refrigerant enters the compressor at point 1. Its pressure, temperature and enthalpy are increased until it leaves the compressor at point 2 located in the supercritical region. Next the refrigerant enters the gas cooler whose function is to transfer heat from the fluid to the environment. Unlike the condensing process in the subcritical system, the refrigerant has not undergone a distinct phase change when it leaves the gas cooler at point 3. Note that this gas cooling process does not occur at constant temperature. The cooled gas then enters an internal heat exchanger (sometimes called a 'suction line heat exchanger'), which transfers heat to that portion of the refrigerant that is just about to enter the compressor.

This results in additional cooling of the refrigerant to point 4 on the figure, improving performance at high ambient temperatures. From there, the flow undergoes a constant-enthalpy expansion process that decreases its temperature and pressure until it exits at point 5 in the mixed liquid/vapor region, at temperature and pressure well below the critical values. Next, the refrigerant enters an evaporator where it absorbs heat from the cooled space and its enthalpy and vapor fraction gradually increase until it exits at point 6. Finally, the flow enters the internal heat exchanger where it absorbs more heat, until it is ready to enter the compressor again at point 1 to repeat the cycle.

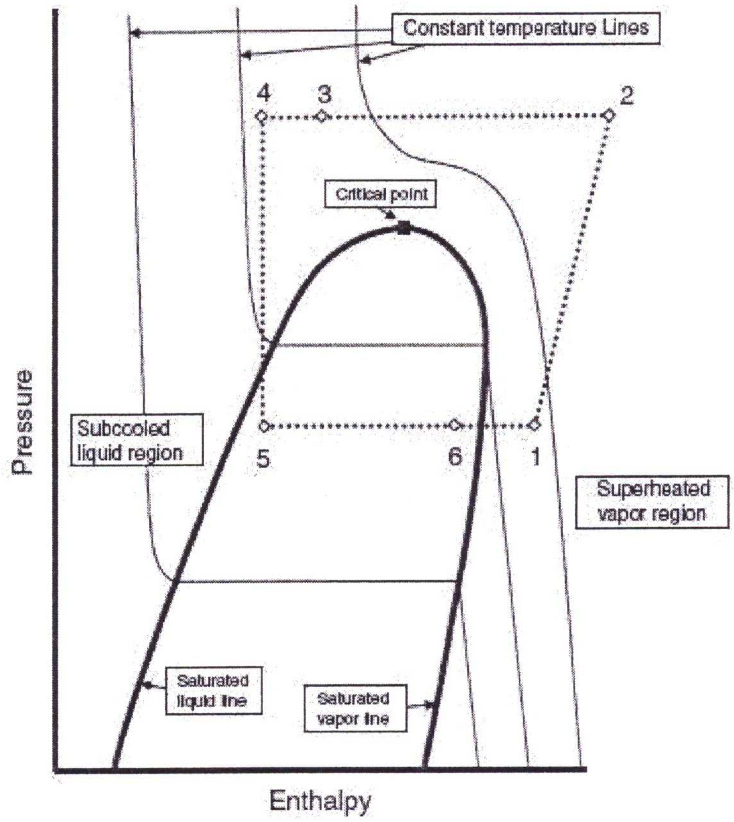


Figure 3.1 Pressure/enthalpy graph transcritical system using R744 (CO2)

Note – the R744 cycle will also work in the subcritical region (i.e. some condensation in the gas cooler will take place) in case the ambient temperature is considerably lower than the critical temperature of R744.

3.2 Cooling load calculation

Cooling load calculation need the dimensions of car, figure 3.2 and figure 3.3 explain dimensions of car, figure 3.2 is the side view of the car..

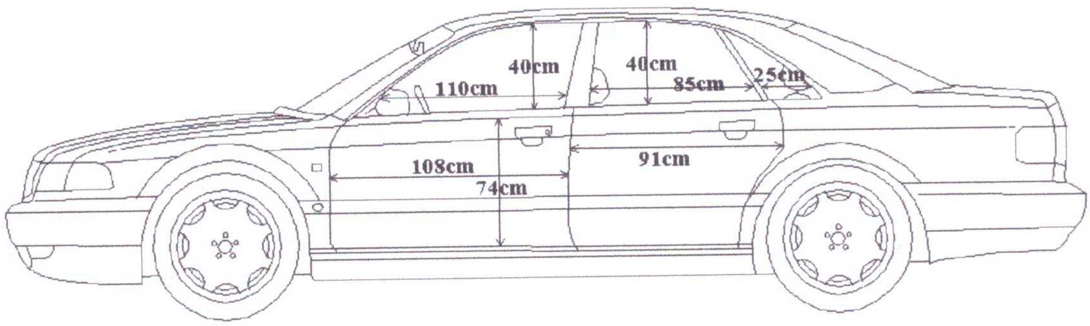


Figure 3.2 dimensions of car at side view

Figure 3.3 represent the top view of the car.

The load calculation of the car for the air condition system consists of two parameters the first one is the conduction load and the other is by solar heat gain ,and the calculation start with the first parameter.

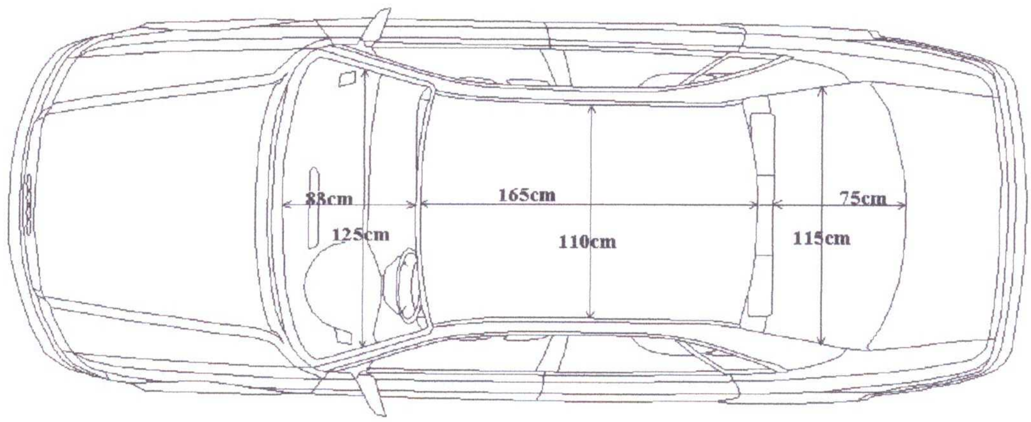


Figure 3.3 top of the car

3.2.1 Heat gain load

The conduction load caused by the heat transfer through the car body by conduction

Assumptions:

The car which chosen in this project is Volkswagen model Passat the deigns done for the hottest month the summer season which is August ,and the car's velocity is 80 km/h. The outside temperature is 40 ^{O}C ,the inside design temperature is 22 ^{O}C .

3.2.1.1 Heat gain from front doors

The figure 3.4 show the component materials of the car doors (cross section).

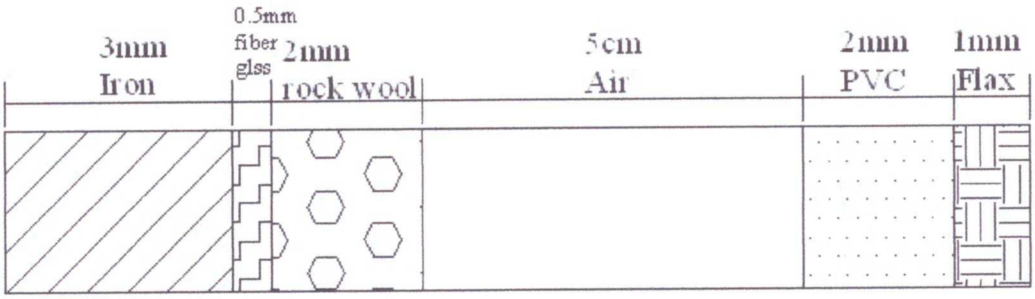


Figure 3.4 cross section in doors

From Material except glass :

$$Q = UA\Delta t \tag{3.1}$$

where :

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

A:aera of heat trensfer [m^2].

Δt: differance of temperature between inside car and outside car (°C).

$$U = \frac{1}{\frac{1}{h_{f,in}} + \frac{\Delta x_1}{k_1} + \frac{\Delta x_2}{k_2} + \frac{\Delta x_3}{k_3} + \frac{\Delta x_4}{k_4} + \frac{\Delta x_5}{k_5} + \frac{\Delta x_6}{k_6} + \frac{1}{h_{f,out}}} \tag{3.2}$$

where:

$h_{f,in}$: convection heat transfer coefficient inside the car (W/m^2.°C).

$h_{f,out}$: convection heat transfer coefficient outside the car (W/m^2.°C).

Δx_i: thickness of material (m).

k_i : thermal conductivity (W/m.°C) (table 3.1), (see Appendix A, table A.1).

$h_{f,out} = 4 + 6v$ [reference 4] (3.3)

where :

v : velocity in (m/s).

$$h_{f,out} = 4 + 6 \times \frac{80000}{3600} = 137.4 \text{ W/m}^2\text{K}$$

$h_{f,in}$= 9.37 W/m^{2}K for still air.

The table 3.1 thermal conductivity of doors materials . [reference 16]

Material	Thermal Conductivity (k) (W/m K)
Iron	79.5
Fiberglass	0.04
Rock wool	0.039
Air	0.025
Plastic (PVC)	0.147
Flax	0.039
Asphalt	36.91
Aluminum	200

$$U = \frac{1}{\frac{1}{9.37} + \frac{0.003}{79,5} + \frac{0.002}{0.147} + \frac{0.004}{0.039} + \frac{0.001}{0.039} + \frac{1}{137.4}}$$

$U = 3.91 \text{ W/m}^2\,°\text{C}$

$A_1 = 0.74 \times 1.08 = 0.7992 \text{ m}^2$

According to equation (3.1)

$Q_{material} = 0.451 \times .7992 \times (40 - 22) = 6.487 \text{ W}$

From glass:

$Q = UA\Delta t$

$$U = \frac{1}{\frac{1}{h_{f,in}} + \frac{\Delta x_G}{k_G} + \frac{1}{h_{f,out}}} \tag{3.4}$$

where:.

Δx_G: thickness of Glass=0.005 m.

k_G : therm al conductivity of glass=1.1 W/m^{2}k.

$$\text{Uglass}=\frac{1}{\frac{1}{9.37}+\frac{0.005}{1.1}+\frac{1}{137.4}} = 8.435 \text{ W/m}^{2\,o}\text{C}.$$

$A_2 = 0.4 \times 1.1 = 0.44 \text{ m}^2$

$Q_{Glass} = 8.435 \times 0.44 \times (40\text{-}22) = 66.8$ W

The total conduction load (Q_{total}) for the front doors can be obtain by the equation

$$Q_{total} = 2 \times Q_{material} + 2 \times Q_{Glass} \tag{3.5}$$

$$Q_{total} = 2 \times 6.487 + 2 \times 66.8 = 146.6 \text{ W}$$

3.2.1.2 Heat gain from Backdoors

The backdoors contain the same material which exist in the construction of front doors so they have the same over all heat transfer coefficient (U), but they have different in areas , Conduction load from material except glass :

According to equation (3.1),and(3.2)

$$Q_{material} = 0.451 \times 0.69 \times (40 - 22) = 5.44 \text{ W}$$

Conduction load from glass:

According to equation (3.1),and(3:4)

$$Q_{Glass} = 8.435 \times 0.34 \times (40\text{-}22) = 51.62 \text{ W}$$

The total conduction load (Q_{total}) for the backdoors can be obtain according equation (3.5)

$$Q_{total} = 2 \times 5.44 + 2 \times 51.62 = 114.12 \text{ W}$$

3.2.1.3 Heat gain from car ceiling

The figure 3.5 show the component materials of the car ceiling and by equation (3.1) can be calculate :

$$Q = UA\Delta t$$

over all heat transfer coefficient (U) can be calculate by equation(3.2),and The table 3.2 represent the thermal conductivity of car ceiling material

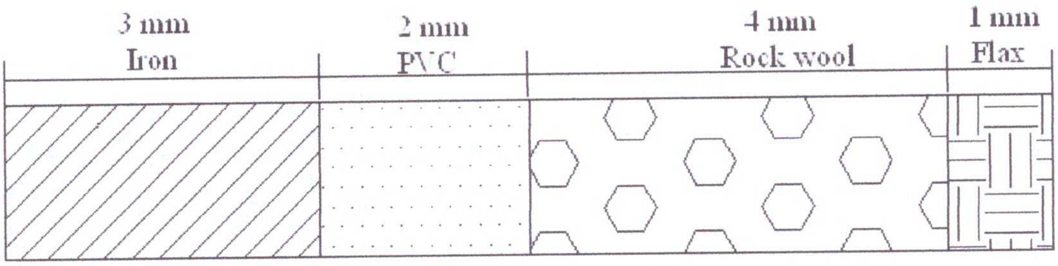


Figure 3.5 cross section in ceiling of the car

$$U = \frac{1}{\frac{1}{9.37}+\frac{0.003}{79,5}+\frac{0.004}{0.039}+\frac{0.002}{0.147}+\frac{0.001}{0.039}+\frac{1}{137.4}}$$

$U = 0.451\ W/m^2\ {}^oC$

$A_{ceiling} = 1.65 \times 1.1 = 1.815\ m^2$

$Q_{ceiling} = 3.91 \times 1.815 \times (40 - 22) = 128\ W$

3.2.1.4 Heat gain from front glass

According to equation (3.1) the Conduction load from front glass of the car can be calculated as :

$Q_{f\,ront\ glass} = 8.435 \times 1.0375 \times (40 - 22) = 158\ W$

3.2.1.5 Heat gain from back glass

According to equation (3.1) the Conduction load from back glass of the car can be calculate :

$$Q_{\text{back glass}} = 8.435 \times 0.86 \times (40 - 22) = 131 \text{ W}$$

3.2.1.6 Heat gain from the floor

The calculation due to floor heat transfer, the figure 3.6 show the component materials of the car.

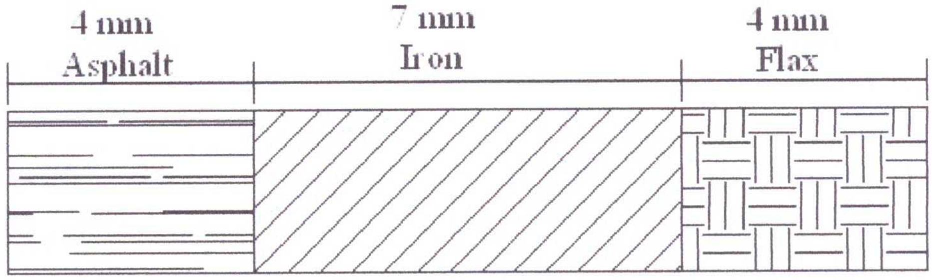


Figure 3.6 cross section in floor

floor and according to equation 3.1 the load can be calculate ,the table 3.3 represent the thermal conductivity material, and the thermal conductance of car floor material

over all heat transfer coefficient (U) can be calculate by the following equation:

$$U = \frac{1}{\frac{1}{h_{f,in}} + \frac{\Delta x_1}{k_1} + \frac{1}{C_1} + \frac{\Delta x_2}{k_2} + \frac{\Delta x_3}{k_3} + + \frac{1}{h_{f,out}}} \qquad (3.6)$$

where:

C_i: thermal conductance

$$U = \frac{1}{\frac{1}{9.37} + \frac{1}{36.91} + \frac{0.007}{79{,}5} + \frac{0.004}{0.039} + \frac{1}{137.4}}$$

$U = 4.1\ W/m^2\ {}^{o}C$

$A = 1.53 \times 1.25 = 1.9125\ m^2$

$Q_{floor} = 4.1 \times 1.9125 \times (40 - 22) = 141.14\ W$

3.2.1.7 Heat gain from the front of car (from engine)

This load is caused by the heat transfer from around the car engine ,the figure 3.7 show the component materials of the car barrier between the car engine and car cabin , table 3.1 represent the thermal conductivity of material that exist in this barrier , and the load can be calculated by equation 3.1 :

$Q = UA\Delta t$

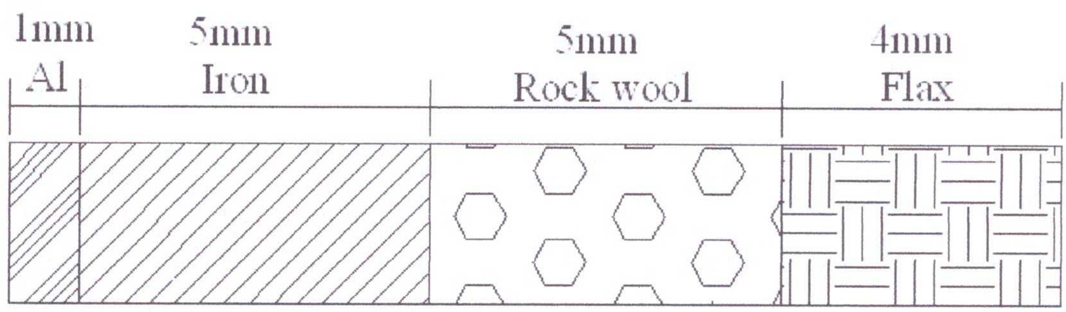


Figure 3.7 cross section in the car barrier between the engine & the cabin

over all heat transfer coefficient (U) can be calculated by equation 3.2:

$$U = \frac{1}{\frac{1}{9.37} + \frac{0.005}{79,5} + \frac{0.005}{0.039} + \frac{0.001}{200} + \frac{0.004}{0.039} + \frac{1}{137.4}}$$

$U = 2.9\ \mathrm{W/m^2\,°C}$

$A = 1.25 \times 0.74 = 0.925\ \mathrm{m}^2$

$\Delta t = t_{out} - t_{in}$

t_{out} : the temperature around the car engine (°C).

t_{in} : the temperature inside the car(°C).

$\Delta t = 60 - 22 = 38\ \ °\mathrm{C}$

$Q_{barrier} = 2.9 \times 0.925 \times 38 = 102\ \mathrm{W}$

3.2.2 load from the human in the car

the assumption that take for this load is for four person exist in the car, and there load equal to:

$Q_{human} = 375\ \mathrm{W}$ [reference 1]

$Q_{human} = 375 \times 4 = 1500\ \mathrm{W} = 1.5\,\mathrm{kW}$

3.2.3 Load from solar radiation

In order to calculate the load from solar radiation, we assumed that a factor; that every one square meter emits 450 W, and calculations will be as follows:

$$Q_{solar} = A_{glass} \times \text{Factor} \quad \text{[reference 12]} \tag{3.7}$$

Where:

A_{glass}: Aera of Glass which heat transfer through it.

Factor: 450 (W/m^2) [reference 12]

The solar radiation load calculation for the front glass can be calculate by equation (3.7):

$$A_{glass} = 1.0375\ m^2$$

$$Q_{solar\ f} = 1.0375 \times 450 = 466.875\ W$$

And for safety the previous value multiplied by 1.5 as safety factor so $Q_{solar\ f}$ became as the following:

$$Q_{solar\ f} = 1.5 \times 466.875 = 700.3125\ W$$

The solar load calculation for the front doors glass can be calculate by equation (3.7):

$$A_{glass} = 0.44$$

$$Q_{solar\ fd} = 0.44 \times 450 = 198\ W$$

Since the car have two front doors previous value multiplied by 2 so $Q_{solar\ fd}$ became as the following:

$$Q_{solar\ fd} = 2 \times 198 = 396\ W$$

The solar load calculation for the backdoors glass can be calculate by equation (3.7):

$A_{glass} = 0.415 \text{ m}^2$

$Q_{solar\ bd} = 0.415 \times 450 = \ 186.75 \text{ W}$

Since the car have two backdoors previous value multiplied by 2 so $Q_{solar\ bd}$ became as the following:

$Q_{solar\ fd} = 2 \times 186.75 = 373.5 \text{ W}$

The solar load calculation for the front glass can be calculated by equation (3.7):

$A_{glass} = 0.8625 \text{ m}^2$

$Q_{solar\ b} = 0.8625 \times 450 = \ 388.125 \text{ W}$

3.2.4 Ventilation and Infiltration

To determine the load from infiltration and ventilation the air change method used, any by using Psychometric chart to find the specific volume, enthalpy, and other properties.

Inside the car the dry bulb temperature 22 °C and 50% relative humidity, and outside the car dry bulb temperature 40 °C and relative humidity 65%. Figure 3.8 shows the cycle on Psychometric chart.

- **Ventilation:**

$$Q_{vent.} = \dot{m}_f \times cp_{air} \times (t_{out} - t_{in}) \qquad (3.8)$$

where :

$\dot{m}_f$: mass flow rate of ventilation air (kg/s).

cp_{air} : specific temprature at constant pressure (J/g.K). Appendix A, table A-2.

t_{out} = outside temprature (°C).

t_{out} = inside temprature(°C).

$$\text{The rate of ventilation air}(\dot{V}_f) = \text{No.of people} \times \text{outdoor air requiremet} \qquad (3.9)$$

where :

No.of people = 4

outdoor air requiremet = 8 L / S / Person

$\dot{V}_f = 8 \times 4 = 32\,L/s = 0.032\ m^3/s$

$$\text{Mass flow Rate of Ventilation Air}\ (\dot{m}_f) = \frac{\dot{V}_f}{\dot{\nu}} \qquad (3.10)$$

where :

$\dot{V}_f$ = The Rate of Ventilation Air = 0.032 m^3/s

$\dot{\nu}$ = Specific Volume @ t = 40 °C and ϕ = 65%

$\dot{\nu} = 0.935\,m^3/kg$

Then :

$$\dot{m}_f = \frac{0.032}{0.935} = 0.0342\,kg/s$$

By substitution in equation (3.8)

$$Q_{vent.} = \dot{m}_f \times cp_{air} \times (t_{out} - t_{in})$$

$$Q_{vent.} = [0.0342 \times 1.00658 \times (40 - 22)] \times 1000 = 620\,W$$

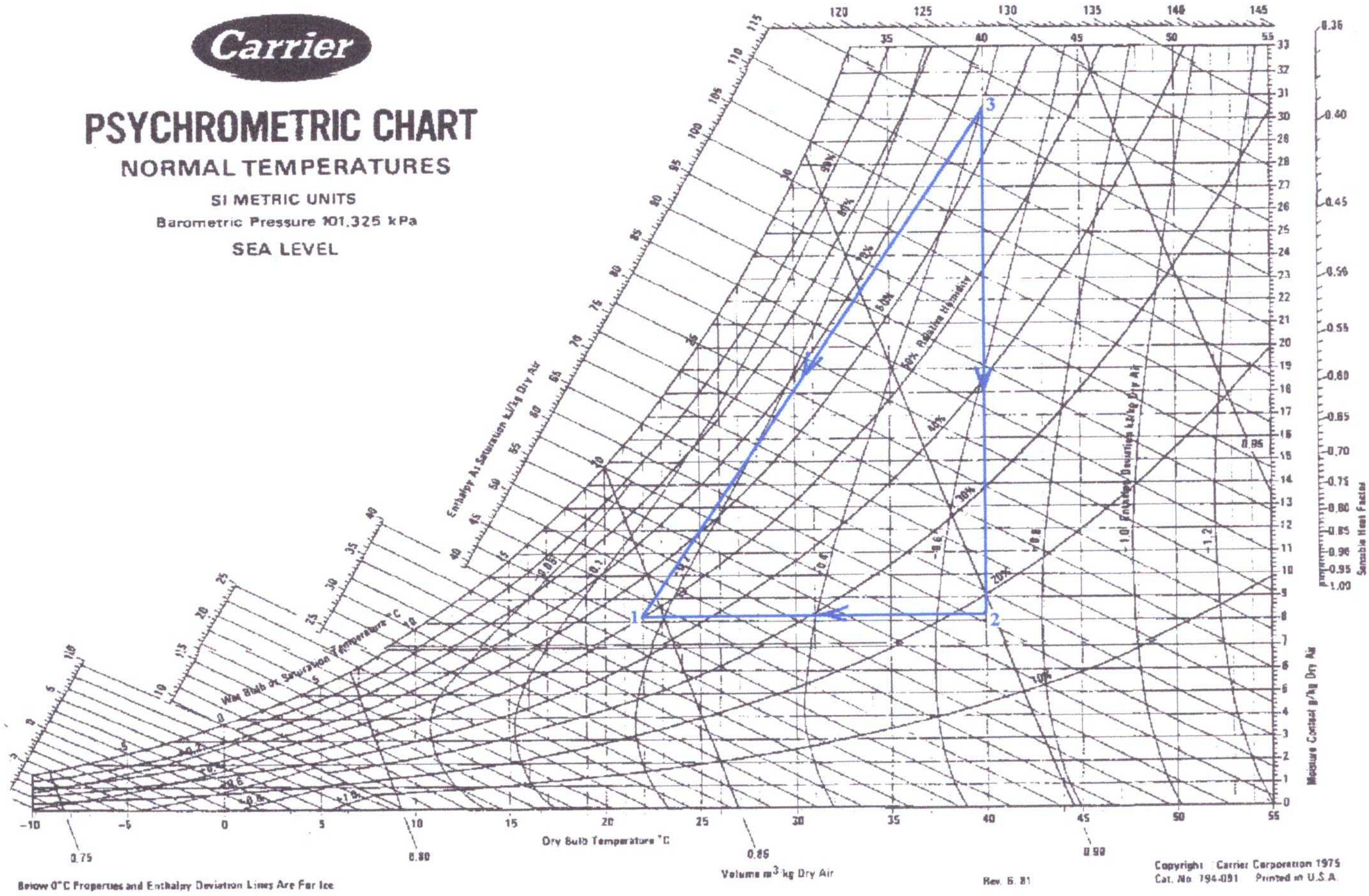


Figure 3.8 Psychometric chart.

- **Infiltration**

$$\dot{Q}_{inf.} = \dot{m}_f \times (h_{out} - h_{in}) \qquad (3.11)$$

where :

$\dot{m}_f$: is mass flow rate of infltrated outside air(kg/s).

h_{out} : outside enthalpy of infltrated air. ($h_{out} = 118\, kJ/kg\, dry\, air$)

h_{in} : inside enthalpy of infltrated air.($h_{in} = 43.5\, kJ/kg\, dry\, air$)

h_{out} and h_{in} from Psychometric chart.

$$\dot{m}_f = \rho_o \times \dot{V}_f$$

where :

ρ_o : density of infltrated air. (3.12)

$\dot{V}_f$: volumetric flow rate of infltrated air (m^3/s).

$\rho_o = \dfrac{1}{v_o}$, v_o : specific volume at 40°C and $\phi = 65\%$ from Psychometric chart.

$$\rho_o = \frac{1}{0.935} = 1.0695\, kg/m^3$$

$$\dot{V}_f = \text{No. of air change in car} \times \text{volume of car}$$

$$\dot{V}_f = 2\, \text{air change/hr} \times 2.85 = 5.7 m^3/hr = 0.00158 m^3/s$$

By substitution in equation (3.12)

$$\dot{m}_f = 1.0695 \times 0.00158 = 0.00169\, kg/s$$

By substitution in equation (3.11)

$$\dot{Q}_{inf.} = 0.00169 \times (118 - 43.5) \times 1000 = 126\, W$$

After this calculations the total load for the car can be determined by the following equation :

$$Q_{Total} = \sum Q_{con} + \sum Q_{so} + \sum Q_{huma.} + \sum Q_{vent.} + \sum Q_{inf.} \quad \text{[reference 14]} \qquad (3.8)$$

$\sum Q_{con}$: sumation of conduction heat teansfer to car

$\sum Q_{so}$: sumation of solar heat teansfer to car

$\sum Q_{huma.}$: sumation of human load.

$\sum Q_{vent.}$: sumation of ventilation load.

$\sum Q_{inf.}$: sumation of infiltration load.

$$Q_{Total} = 920.86 + 1857.94 + 1500 + 620 + 126 = 5025 \text{ W}$$

As factor of safety previous value assumed to be Q_{Total} **= 1.5*5025 =7538 W**

Q_{Total} **= 7.538 kW**

3.3 Cycle calculations:

Figure 3.8 represents the schematic drawing of the project, and show the phase change of refrigerant in points (1-7).

Figure 3.9 shows the refrigerant flow and its status. Where it leaves the compressor in gas phase and at high temperature degree , and leaves gas cooler in gaseous phase but at smaller temperature than temperature after compressor temperature, then it is becomes mixture after expansion valve, then it converts to gas again after evaporator, any liquid leave evaporator is trapped by accumulator.

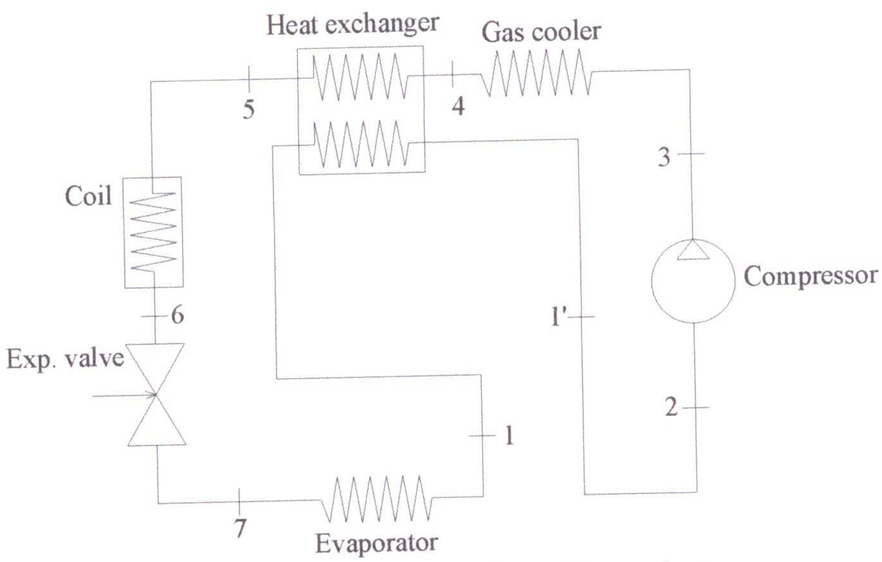


Figure 3.8 schematic drawing of the project.

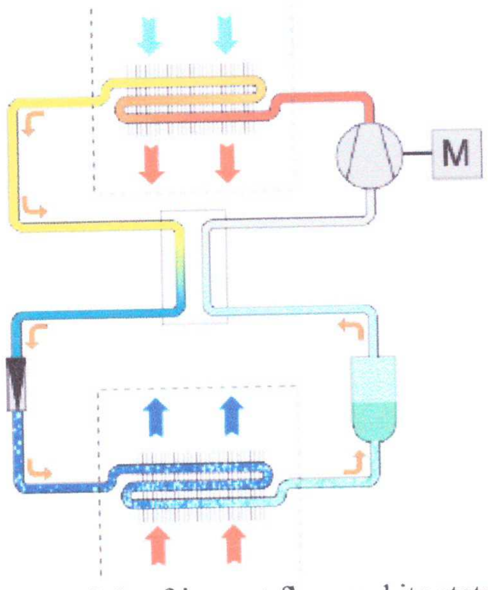


Figure 3.9 refrigerant flow and its status

After drawing the cycle of project on P-h (pressure – enthalpy) diagram of R744, the calculations starts by taking the values of enthalpy at each point to determine the cycle properties. In figure 3.10 explains the cycle on P-h diagram.

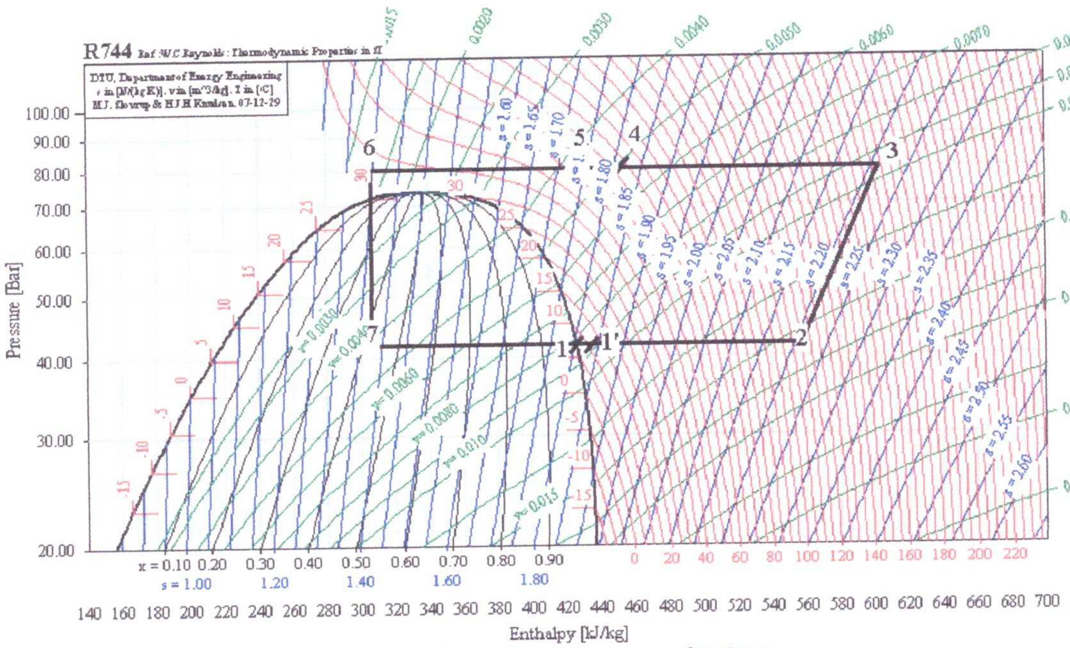


Figure 3.10 P-h diagram of R744

From the P-h diagram of the cycle some of properties of R744 can be obtained, these properties are shown in table 3.5, (for others properties see Appendix A, table 3) .

Table 3.5 some properties of R744 from P-h diagram.

Property / State	Enthalpy (h) kJ/kg	Pressure (P) bar	Temperature(T) °C
1	428	42	7
1'	433	42	12
2	554	42	110

3	604	80	167
4	452	80	57
5	428	80	47
6	308	80	30
7	308	42	7

Refrigeration effect can be determined by following equation:

$$q_e = h_1 - h_7$$ [reference 16] (3.9)

$$q_e = 428 - 308 = 120 kJ/kg$$

$$w_{net} = h_3 - h_2$$ [reference 16] (3.10)

where :

w_{net} : work of the compressor.

$$w_{net} = 604 - 554 = 50 kJ/kg$$

Coefficient of performance (COP) can be calculated from the equation:

$$COP = \frac{q_e}{w_{net}}$$ [reference 16] (3.11)

$$COP = \frac{120}{50} = 2.4$$

From cooling load calculation in the previous section the total load of the car equal 3.5 kW. And from total cooling load and refrigeration effect the mass flow rate it can be calculated by the following equation:

$$m^\circ = \frac{Q_e}{q_e}$$ [reference 16] (3.12)

$$m^\circ = \frac{7.538}{120} = 0.063 kg/s$$

And the next equation represent the load of the gas cooler (condenser), [reference 9]:

$$Q_{\text{gas cooler}} = q_{\text{gas cooler}} \times m^{o} \tag{3.13}$$

$$Q_{\text{gas co.}} = (h_3 - h_4) \times m^{o} = (604 - 452) \times 0.063 = 9.576 \text{ kW}$$

The amount of heat ejected from coil after gas cooler can be calculated by the following equation, [reference 9]:

$$Q_{coil} = m^{o} q_{coil} = m^{o}(h_4 - h_5) \tag{3.14}$$

$$Q_{coil} = 0.063 \times (452 - 428) = 1.512 \text{ kW}$$

The power of the compressor in kilo Watts is:

$$\text{Power of compressor } (P_{comp.}) = m^{o} \times w_{net} \tag{3.15}$$

$$P_{comp.} = 0.063 \times 50 = 3.15 \text{ kW}$$

Power of the compressor in horse power is:

$$P_{comp.} = \frac{3150 \text{ W}}{746} = 4.22 \text{ hp} \tag{3.16}$$

The coefficient of performance (COP) according to power of compressor is, [reference 16]:

$$COP = \frac{Q_e}{P_{comp.}} = \frac{7.538 \text{ kW}}{3.15 \text{ kW}} = 2.4 \tag{3.17}$$

3.3.1 compressor calculations:

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

$$V^{\circ}_{theo.} = m^{\circ}\upsilon_1 \tag{3.18}$$

where:

$V^{\circ}_{theo.}$: theoretical volume flow rate of the compressor (m^3/s).

m° : mass flow rate (kg/s).

υ_1 : specific volume at the inlet of compressor (m^3/kg) (Appendex A, table A-3).

$$V^{\circ}_{theo.} = 0.063 \times 0.0175 = 1.1025 \times 10^{-3}\ m^3/s$$

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

$$V^{\circ}_{act.} = \frac{V^{\circ}_{theo.}}{\eta_v} \tag{3.19}$$

where:

$V^{\circ}_{act.}$: actual volume flow rate (m^3/s).

$V^{\circ}_{theo.}$: theoretical volume flow rate (m^3/s).

η_v : volumetric efficiency.

$$\eta_v = \eta_{v,c} \times \eta_{v,h} \qquad \text{[reference 6]} \tag{3.20}$$

where:

$\eta_{v,c}$: volumetric efficiency due to clearance volume in compressor.

$\eta_{v,h}$: volumetric efficiency due to heating occures in compressor.

The volumetric efficiency due the clearance volume in compressor calculated by equation (3.21), [reference 6]:

$$\eta_{v,c} = 1 - c\left[\left(\frac{P_H}{P_L}\right)^{\frac{1}{n}} - 1\right] \qquad (3.21)$$

where:

n : exponential cofficient of rexpansion for CO_2.

$n = 1.1$,[reference 4]

c:ratio between volumetric clearance and volume of cylinder of CO_2 compressor.

$c = 0.04$, [reference 4]

P_H : high pressure of the cycle (bar).

$P_H = 80$ bar

P_L : low pressure of the cycle (bar).

$P_L = 42$ bar

$$\eta_{v,c} = 1 - 0.04\left[\left(\frac{80}{42}\right)^{\frac{1}{1.1}} - 1\right] = 0.968 = 97\%$$

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

$$\eta_{v,h} = \frac{T_e}{T_m} \qquad (3.22)$$

where:

T_e : temperature of evaporator (°K).

$$T_m = \frac{\text{T at outlet of compressor} - \text{T at inlet of expa. valve}}{2} \quad (°K) \qquad (3.23)$$

$$T_m = \frac{167 - 30}{2} = 68.5\ °K$$

Because there are no temperature of condenser in the cycle of a high side is not available, T_m is taken as the mean value of the high side of cycle between outlet of compressor and inlet of expansion valve.

$$\eta_{v,h} = \frac{7+273}{68.5+273} = 82\%.$$

By substitution in equation (3.20) to calculate the total volumetric efficiency :

$$\eta_v = 0.97 \times 0.82 = 79.54\%$$

After that by substitution in equation (3.19), the actual volume flow rate is:

$$V^{\circ}_{act.} = \frac{1.1025 \times 10^{-3}}{0.795} = 1.387 \times 10^{-3}\ m^3/s$$

The actual power for compressor is can be determine from equation (3.24).

$$P_{act.} = P_{ind.} + P_{fri.} \qquad \text{[reference 12]} \qquad (3.24)$$

where :

$P_{act.}$: actual power of compressor (kW).

$P_{ind.}$: indicator power (kW).

$P_{fri.}$: power due friction (kW).

Indicator power can be calculate by the equation (3.25), [reference 12]

$$P_{ind.} = \frac{P_{theo.}}{\eta_{ind.}} \qquad (3.25)$$

The indicator efficiency can be calculated by the next emperical equation, [reference 6]

$$\eta_{ind.} = \eta_{v.h} + b_0 \times t_e \qquad (3.26)$$

where:

$\eta_{ind.}$: indicator efficiency.

$\eta_{v.h}$: volumetric efficiency due to heating.

b_0 : constan, in CO_2 compressor $b_0 = 0.001$

t_e : evaporater temperature. (7°C).

$$\eta_{ind.} = 0.82 + 0.001 \times 7$$
$$= 82.7\%$$

By the substitution in equation (3.25), [reference 12]:

$$P_{ind.} = \frac{P_{theo.}}{\eta_{ind.}}$$

$$P_{ind.} = \frac{3.15}{0.827} = 3.809 \text{ kW}$$

The next equation to determine frictional power, [reference 12]:

$$P_{fri.} = \overset{o}{V}_{cylinder} \times \text{pressure friction} \qquad (3.27)$$

where:

$\overset{o}{V}_{cylinder}$: volume flow rate in cylinder (m^3/s).

pressure friction : the pressure occures due friction (kPa).

$$\overset{o}{V}_{cylinder} = \frac{\overset{o}{V}}{\eta_v} \qquad (3.28)$$

$$= \frac{1.1025 \times 10^{-3}}{0.795} = 1.386 \times 10^{-3} \text{ m}^3/\text{s}$$

pressure friction $= 0.6 \times 10^2$ kPa $\rightarrow$ for CO_2

By substitution in equation (3.24)

$$P_{fri.} = 1.386 \times 10^{-3} \times 0.6 \times 10^{2} \times 1000$$
$$P_{fri.} = 83.16\,W$$

$$P_{act.} = 3809 + 83.16 = 3.89\ kW$$

The mechanical power that needed from the engine power to drive the air condition compressor can be calculate by the following equation:

$$P_{mech} = \frac{P_{act}}{\eta_{mech}} \tag{3.29}$$

Where:

P_{mech} : mechanical power from car engine of car (kW).
P_{act} : actual (shaft) power of the compressor (kW)
η_{mech} : efficiency of the belt which transmit the power from the engine to compressor shaft.

The actual (shaft) power had been determined in previous calculation , and it's value as the following:

$$P_{act} = 3.89\ kW$$

and value of the mechanical efficiency is alternate from 88 - 95% ,and it's value in this calculation chose as 90%, so the mechanical power now can be calculate

$$P_{mech} = \frac{3.89}{0.90} = 4.32\ kW$$

the machanical power in powerhorse can be calculate by the following equation :

$$P = \frac{P_{mech}}{746} \tag{3.30}$$

$$P=\frac{4320}{746}=5.8\ \text{hp}$$

Finally the power required to compressor (5.8 hp), its mean the output power (rated power) of the electrical motor to drive compressor (5.8 hp).

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 upon 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).

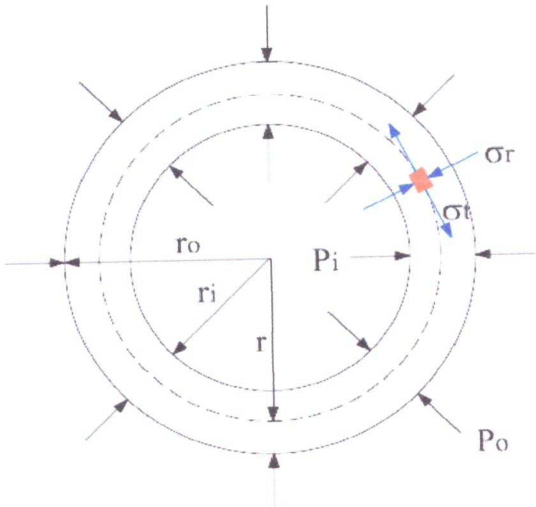


Figure 4.1 stresses, internal and external pressures

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

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

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 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:

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

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

$$\sigma_r = -P_i$$

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

$$\sigma' = \sqrt{\sigma_t^2 - \sigma_t \sigma_r + \sigma_r^2} \tag{4.5}$$

$$n = \frac{S_y}{\sigma'} \tag{4.6}$$

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 . (see appendix A, table A-4).

4.2 Pipe design in high pressure side

After compressor the pressure is 80 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

$$Q = \dot{m}\nu \tag{4.8}$$

Where :

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

ν : specific volume at compressor outlet (0.0095 m^3/kg) (see appendix A, table A-3).

$Q = 0.063 \times 0.0095 = 5.985 \times 10^{-4}\ m^3/s.$

By substitution in equation (5.7)

$$A = \frac{Q}{V} = \frac{5.985 \times 10^{-4}}{12} = 4.987 \times 10^{-5}\ m^2$$

$$A = \frac{\pi d^2}{4} \Rightarrow d = \sqrt{\frac{4 \times 4.987 \times 10^{-5}}{\pi}} = 8 \times 10^{-3}\, m$$

the interanal diameter of the pipe in high pressure side ($d_i = 8\, mm$).

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]

$$\sigma' = \sqrt{\sigma_t^{\,2} - \sigma_t\sigma_r + \sigma_r^{\,2}}\ ,\ \sigma_r = -P_i = -80\, bar.$$

$$350 = \sqrt{\sigma_t^{\,2} + 80\sigma_t + 6400}$$

$$\sigma_t^{\,2} + 80\sigma_t = 116100$$

$$\sigma_t = 606.1486\, bar$$

from equation (4.3)[reference 10]

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

$$r_i = \frac{8}{2} = 4\, mm$$

$$606.148 = \frac{(4 \times 10^{-3})^2 + r_o^{\,2}}{r_o^{\,2} - (4 \times 10^{-3})^2} \times 80$$

$r_o^{\,2} = 5 \times 10^{-3}\, m = 5\, mm$

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

$t = r_o - r_i = 5 - 4 = 1\, mm$

The internal and outer diameter in high pressure side (80 bar) in inch

$$d_{i,inch} = \frac{d_{i,mm}}{25.4} = \frac{8}{25.4} = 0.315\,\text{inch}$$

$$d_{o,inch} = \frac{d_{o,mm}}{25.4} = \frac{10}{25.4} = 0.39\,\text{inch}$$

4.3 Pipe design in low pressure side

Like equation used in pipe design in high pressure side , it used for low pressure side (42 bar) .

Using equation (4.7)

$$Q = VA$$

equation (4.8)

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

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

$\nu = 0.016\ \text{m}^3/\text{kg}$ (see appendix A, table A-3).

$$Q = 0.063 \times 0.016 = 1.008 \times 10^{-3}\,\text{m}^3/\text{s}$$

$$A = \frac{Q}{V} = \frac{1.008 \times 10^{-3}}{10} = 1.008 \times 10^{-4}\,\text{m}^2$$

$$d_i^2 = \frac{4 \times A}{\pi} \Rightarrow d_i = \sqrt{\frac{4 \times 1.008 \times 10^{-4}}{\pi}} = 11 \times 10^{-3}\,\text{m} = 11\text{mm}$$

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)

$$\sigma' = \sqrt{\sigma_t^2 - \sigma_t\sigma_r + \sigma_r^2}\ ,\ \sigma_r = -P_i = -42\,\text{bar}.$$

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

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

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

from equation (4.3)

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

$$r_i = \frac{11}{2} = 5.5\,\text{mm}$$

$$654.21 = \frac{(5.5\times10^{-3})^2 + r_o^2}{r_o^2 - (5.5\times10^{-3})^2} \times 42$$

$$r_o^2 = 6.3\times10^{-3}\,\text{m} = 6.3\,\text{mm}$$

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

$$t = r_o - r_i = 6.3 - 5.5 = 0.8\,\text{mm}$$

The internal and outer diameter in low pressure side (42 bar) in inch

$$d_{i,inch} = \frac{d_{i,mm}}{25.4} = \frac{11}{25.4} = 0.433\,\text{inch}$$

$$d_{o,inch} = \frac{d_{o,mm}}{25.4} = \frac{12.6}{25.4} = 0.5\,\text{inch}$$

4.4 Pipe Selection

By referring to copper hand book (appendix B, table B-1), 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 3/8 D which has outer diameter 0.375 inch, and inside diameter 0.315 inch, and wall thickness 0.030 inch. (appendix B, table B-2). 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 1/2 A which has outer diameter 0.5 inch, and inside diameter 0.436 inch, and wall thickness 0.032 inch. (appendix B, table B-2). According to this result the calculations of pipe in low pressure side is near to the this selection data.

CHAPTER FIVE

HEATEXCHANGERS DESIGN

5.1 The Gas Cooler Design :

5.1.1 Gas Cooler design without fins

The gas cooler location in this project is after the compressor outlet & its work in this cycle is cooling the refrigerant which comes from the compressor and in CO_2 A/C automobile there is no condensation occur as vapor compression cycle, since the refrigerant comes from compressor as superheated gas ,and leave gas cooler as a high temperature gas (there is no phase change of refrigerant occurs in gas cooler) .
Design the gas cooler required many calculations such as fluid mechanical calculation , thermal calculation and area calculation , the sequence of design start with fluid mechanical calculation until reaching area calculation , the calculation start according to the following equation:

$$Q = UA\,\Delta t \qquad (5.1)$$

Where:

Q: Heat transfered through the gas cooler (W).

A: surface area of heat transfer (m^2).

U: The overall heat transfer coefficient ($W/m^2.°C$).

Δt: Temperature difference (°C).

The heat that transferred through the gas cooler was determined in chapter three in section 3.3 (cycle calculations) page 56 and its value was $Q_{\substack{total \\ gas\ cooler}}$ =9.576 kW, and in order to determine (U) the following equation is used [reference 11] :

$$U_i = \frac{1}{\frac{1}{h_i} + \frac{r_i \ln(r_o/r_i)}{k} + \frac{r_i}{r_o} \times \frac{1}{h_o}} \qquad (5.2)$$

Where:

k:Thermal conductivity of tube (copper)(W/m.°C).

h_i : Convection heat transfer coefficient for refrigerant(CO_2) inside the tube (W/m^2.°C).

h_o :Convection heat transfer coefficient for air outside the tube (W/m^2.°C).

r_i : The internal raduis of the tube (m).

r_o: The outer raduis of the tube (m).

To determine the value of U there are many parameters should be determined before reaching U value, and it will be started with the flow rate of refrigerant, radiuses, then the heat transfer coefficients, and all calculation done at the entrance of the gas cooler, so the CO_2 properties were determined according to the state at entrance of the gas cooler.

The refrigerant temperature at entrance of the gas cooler $167\ °C$. And all properties of CO_2 mention in (appendix A, table A-3).

$$Q = AV = \dot{m}\nu \qquad (5.3)$$

Where :

Q: The flow rate of the refrigerant (CO_2) in tube (m^3/s).

A: The cross section area of tube (m^2).

V: The velocity of refrigerant (CO_2) in side tube (m/s).

$\dot{m}$: The mass flow rate of refrigerant (CO_2) (kg/s).

ν: The specific volume of refrigerant (CO_2) at the entrance of gas cooler (m^3/kg).

The value of $\dot{m}$ was determined in chapter three in section 3.3, $\dot{m} = 0.063\,kg/s$, and the velocity of refrigerant was given in (table A-4 in appendix A), And its value V=12 m/s so

$$Q = \dot{m}v = 0.063 \times 0.0095 = 5.985 \times 10^{-4} m^3/s$$

$$A = \frac{Q}{V} = \frac{5.985 \times 10^{-4}}{12} = 4.9875 \times 10^{-5}\ m^2$$

After determining the cross section area of the tube the diameter can be determined by the following equation :

$$A = \frac{\pi}{4} d_i^2 \qquad (5.4)$$

Where:

d_i: The internal diameter of tube (m).

$\therefore$

$$d_i = \sqrt{\frac{A \times 4}{\pi}} = \sqrt{\frac{4.9875 \times 10^{-5} \times 4}{\pi}} = 7.978 \times 10^{-3} m$$

$$d_i = 8mm$$

$$r_i = \frac{d_i}{2}$$

Where:

r_i = The internal radius of the tube (m).

The outer radius of the tube can be determined according to the following equation :

$$r_o = r_i + t \qquad (5.5)$$

Where:

r_o : The outer radius of the tube (m).

t: The thickness of tube (m) [chapter 4].

Where the thickness of the tube was determined in chapter four

$r_o = 4\times10^{-3} + 1\times10^{-3} = 5\times10^{-3}$ m=5mm

$\therefore$

$d_o = r_o \times 2$

Where:

d_o : The outer diameter of tube (m).

$d_o = 10\times10^{-3}$ m=10mm

After determining the radiuses of the tube that use for the gas cooler , then the heat transfer coefficients will be calculated by calculating the Reynolds ,Prandtls , and Nusselts numbers in order to calculate the inner convection heat transfer coefficient , by using the following equations, start with calculating Reynolds number by using the following equation [reference 11] :

$$Re=\frac{V\times\rho\times d_i}{\mu} \tag{5.6}$$

Where:

Re: Reynolds number

V: The velocity of refrigerant (CO_2) in side tube (m/s).

d_i : The internal diameter of tube (m).

ρ: The density of refrigerant(CO_2) at the entrance of gas cooler (kg/m^3).

μ : Dynamic viscosity of refrigerant(CO_2) at the entrance of gas cooler (Pa.s).

$\rho = \frac{1}{v} = \frac{1}{0.0095} = 105.26$ kg/m^3

$\mu = 22.679\times10^{-6}$ Pa.s

$d_i = 8$mm

$$Re=\frac{12\times105.26\times8\times10^{-3}}{22.679\times10^{-6}} = 445.564\times10^{3}$$

According to the value of Reynolds number the flow inside the tube is turbulent flow ,after calculating the value of Reynolds number so the value of Prandtls number can be calculated by the following equation [reference 11]:

$$Pr=\frac{c_p\mu}{k} \tag{5.7}$$

Where:

Pr: Prandtl number.

c_p : Specific heat at constant press ure of the refrigerant(CO_2) at the entrance of gas cooler (kJ/kg.°C).

k: Thermal conductivity of refrigerant (CO_2)(W/m.°C).

$$c_{p_{CO_2@160^\circ C\&80\,bar}}=1.16427kJ/kg=1164.27J/kg$$

$$k=32.1922\times10^{-3}\ W/m^{2\circ}C$$

$$\mu=22.679\times10^{-6}\ Pa.s$$

$$Pr=\frac{1164.27\times22.679\times10^{-6}}{32.1922\times10^{-3}}=0.82$$

According to the value of Reynolds number and Prandtls number which determine the equation that will be used to calculate the value of Nusselts number as

$0.5<Pr<1.5$ [reference 11]

$10^4<Re<5\times10^6$

Then the following equation can be used to calculate the Nusselts number(Nu):

$$Nu=0.0214(Re^{0.8}-100)Pr^{0.4} \tag{5.8}$$

$$Nu=0.0214\times((445.564\times10^3)^{0.8}-100)\times0.82^{0.4}=651.255$$

By determining the value of Nusselts number the internal convection heat transfer coefficient (h_i) inside the gas cooler tube can be determined using the following equation:

$$Nu=\frac{h_i \times d_i}{k} \tag{5.9}$$

$$651.255=\frac{h_i \times 8\times 10^{-3}}{32.1922\times 10^{-3}}$$

$$h_i=\frac{651.255\times 32.1922\times 10^{-3}}{8\times 10^{-3}}=2620.66\ W/m^2.°C$$

$$h_i=2620.66\ W/m^2.°C$$

After determining internal heat transfer coefficient (h_i) inside the gas cooler tube now the external convection heat transfer coefficient(h_o) for outside the gas cooler tube by using the following equation [reference 6]:

$$h_o=4+6\times V_{car} \tag{5.10}$$

V_{car}: The velocity of car (m\s).

The velocity of car was determined in chapter three section 3.2.1,so V_{car}=22.22m/s and:

$$h_o=4+6\times 22.22=137.32\ W/m^2.°C$$

After determining h_o, h_i ,inner radius and outer radius of gas cooler the value of overall heat transfer coefficient (U) by using equation (5.2):

$$U_i=\frac{1}{\frac{1}{2620.66}+\frac{4\times 10^{-3}\ \ln(5\times 10^{-3}/4\times 10^{-3})}{401}+\frac{4\times 10^{-3}}{5\times 10^{-3}}\times\frac{1}{137.32}}$$

$$U_i=161\ W/m^2.°C$$

After determining the value of (U) it can be calculated the area of gas cooler according to equation (5.1):

$$Q_{\substack{\text{total}\\ \text{gas cooler}}} = U_i A_i \ \Delta t$$

$$9576 = 161 \times A_i \times (167 - 40)$$

$$A_i = \frac{9576}{161 \times 127} = 0.5\text{m}^2$$

By determine the internal area (A_i) of gas cooler the total equivalent length of the gas cooler tube can be calculated by the following equation :

$$A_i = 2\pi r_i L \qquad (5.11)$$

$$0.5 = 2\pi \times 4 \times 10^{-3} \times L$$

$$L = \frac{0.5169}{2\pi \times 4 \times 10^{-3}} = 19.89\text{m}$$

Then now the external area of gas cooler can be determined by using the following equation:

$$A_o = 2\pi r_o L$$

A_o : The external surface area of gas cooler (m^2).

$$A_o = 2\pi \times 5 \times 10^{-3} \times 19.89 = 0.625\text{m}^2$$

5.1.2 Gas Cooler design with fins

All the previous calculations for gas cooler without using fins on the tube and the next calculations for gas cooler with fin and to do this calculations it need to applied on an element and figure (5.1) show the element (B-B) and its dimensions:

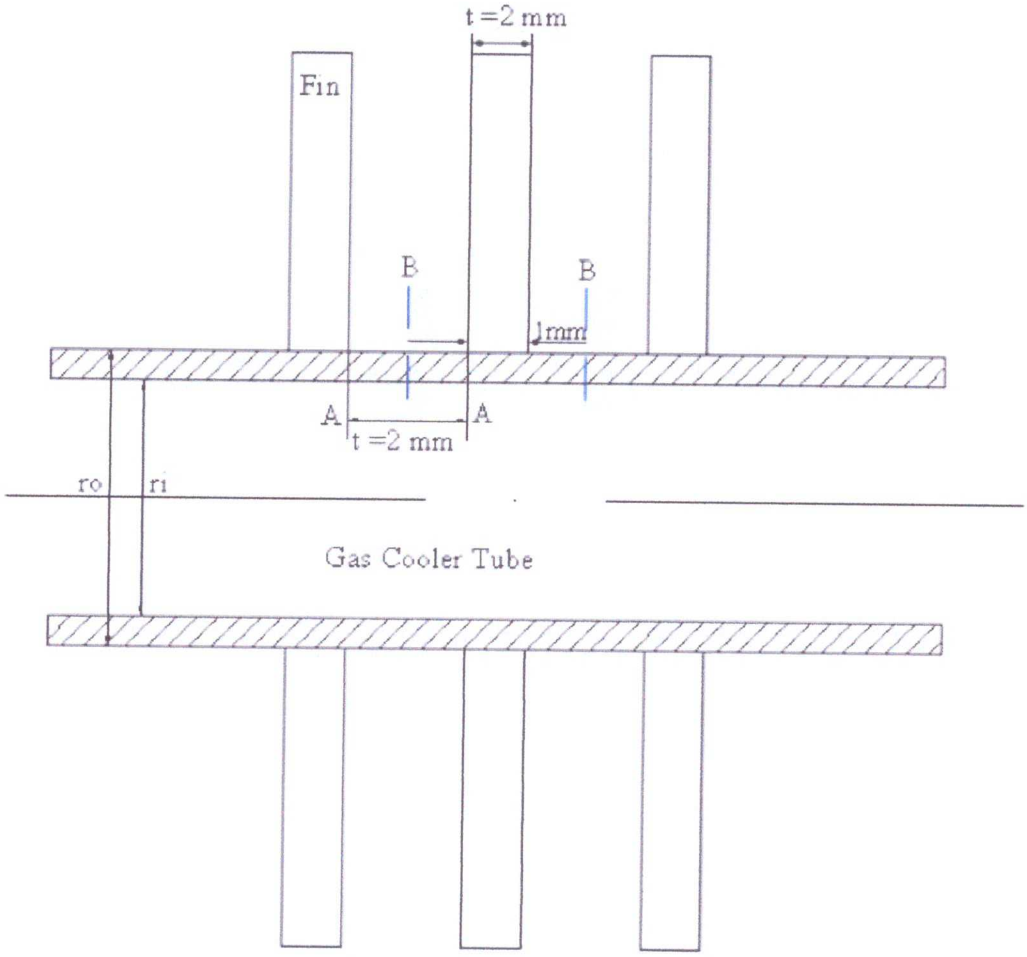


Figure 5.1 the element and its dimensions

In order to calculate the total equivalent length of finned tube and external area of gas cooler the following equation is used [reference 11] :

$$q_{\substack{total\\ elemant}} = q_f + q_{original} \tag{5.12}$$

Where:

$q_{\substack{total\\ elemant}}$: the total heat transfer from the element (W)

q_f : Heat transfer rate per fin (W).

$q_{original}$: Heat transfer rate from tube without fin (W).

The calculation will start with heat transfer rate per fin (q_f) and then for heat transfer from tube without fin ($q_{original}$), and the calculation of (q_f) depends on the geometry of the fin and figure (5.2) show the fin geometry and the dimensions of the fin:

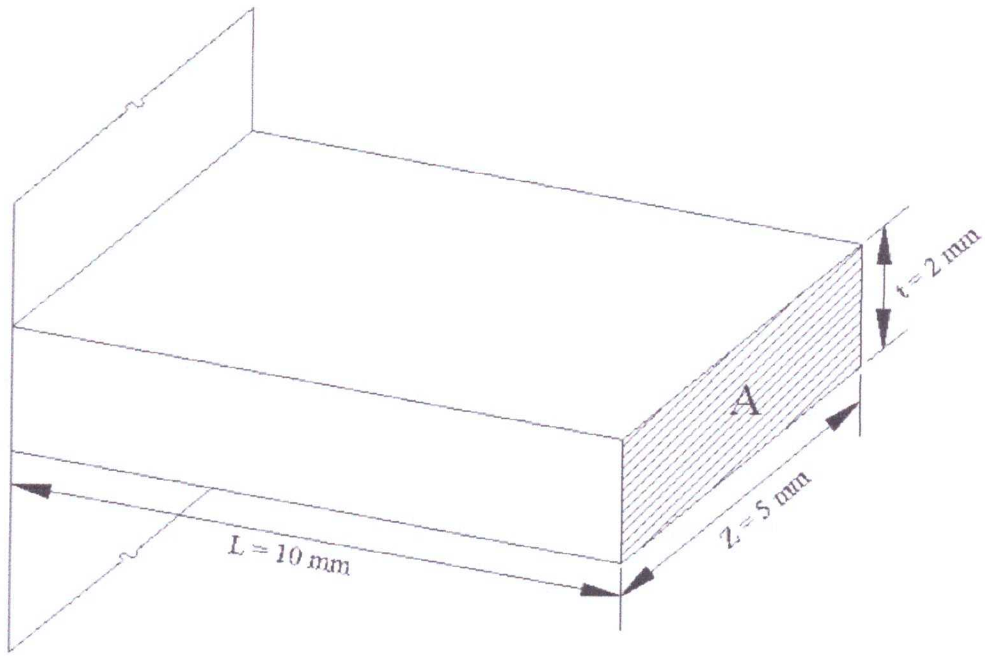


Figure 5.2 fin geometry and dimensions

and (q_f) can be calculated by the following equation [reference 11] :

$$q_f = \sqrt{hPkA} \times (T_o - T_\infty) \times \frac{\sinh(mL_c) + \frac{h}{mk}\cosh(mL_c)}{\cosh(mL_c) + \frac{h}{mk}\sinh(mL_c)} \tag{5.13}$$

Where:

h: Convection heat transfer coefficient of air which surrounding the fin (W/m^2.°C).

P: Perimeter of fin (m).

k: Conduction heat transfer coefficient for fin of material(Al) (W/m.°C).

A: Surface area for convection of fin (m^2).

T_o: Temperature surrounding air of fin(°C).

T_∞ : Temperature of the base of fin(°C).

L_c: Corrected fin length (m).

m: factor.

In this equation there are many parameters need to be determined such as perimeter, area of fin, L_c ,and m and the value of (P) can be obtained by the following equation [reference 11]:

$$P = 2t + 2z \tag{5.14}$$

Where:

t: Thickness of fin(m).

z: Depth of fin(m).

$P = (2 \times 5 \times 10^{-3}) + (2 \times 2 \times 10^{-3}) = 14 \times 10^{-3} = 14\text{mm}.$

$A = t \times z = 5 \times 10^{-3} \times 2 \times 10^{-3} = 10 \times 10^{-6}\text{m} = 10\text{mm}$

$k_{Al} = 200\,\text{W/m.°C}$

And the value factor (m) can be determined by the following equation and its dimensionless factor :

$$m = \sqrt{\frac{hP}{kA}} \tag{5.15}$$

$$m = \sqrt{\frac{137.32 \times 14 \times 10^{-3}}{200 \times 10 \times 10^{-6}}} = 31$$

and the value of (L_c) can be calculated by the following equation:

$$L_c = L + \frac{d_o}{4} \tag{5.16}$$

L: The length (m)

d_o : The outer diameter of tube (m).

$$L_c = 10 \times 10^{-3} + \frac{10 \times 10^{-3}}{4} = 12.5 \times 10^{-3} m = 12.5mm$$

After determining all the parameters the heat transfer rate per fin (q_f) can be calculated by the equation (5.13) [reference 11]:

$$q_f = \sqrt{137.32 \times 14 \times 10^{-3} \times 200 \times 10 \times 10^{-6}} \times (167 - 40) \times$$
$$\times \frac{\sinh(31 \times 12.5 \times 10^{-3}) + \frac{137.32}{31 \times 200} \cosh(31 \times 12.5 \times 10^{-3})}{\cosh(31 \times 12.5 \times 10^{-3}) + \frac{137.32}{31 \times 200} \sinh(31 \times 12.5 \times 10^{-3})}$$

$$q_f = 0.95 \text{ W}$$

To calculate the efficiency of the fin(η_f) the following equation is used [reference 11]:

$$\eta_f = \frac{\tanh(mL_c)}{mL_c} \tag{5.17}$$

$$\eta_f = \frac{\tanh(31 \times 12.5 \times 10^{-3})}{31 \times 12.5 \times 10^{-3}} = 0.95 = 95\%$$

This result of (η_f) shows the amount heat transfer from the fin is 95% from the total of heat that fin is exposed .

After determining the amount of q_f then the amount of heat transfer rate from tube without fin $q_{original}$ will be determined by the following equation:

$$q_{original} = q_{A_1-A_2} = UA\Delta t \qquad (5.18)$$

The section $A_1 - A_2$ shown in figure 5.1 and its dimensions, the surface area of heat transfer is obtain from the following:

$$A = 2\pi r_o L \qquad (5.19)$$

Where:

L: Length of tube without fin(m).

The length of tube without fins can be obtained from figure 5.1 , or be calculated by the following equation:

$$L = L_{B_1-B_2} - t_f \qquad (5.20)$$

$$L = 4 \times 10^{-3} - 2 \times 10^{-3} = 2 \times 10^{-3} \text{m} = 2\text{mm}$$

Then :

$$A = 2\pi \times 5 \times 10^{-3} \times 2 \times 10^{-3} = 6.283 \times 10^{-5} \text{m}^2$$

By determining the area of heat transfer surface the amount of heat transfer rate from tube without fin ($q_{original}$) can be determined according to equation(5.18):

$$q_{original} = 6.283 \times 10^{-5} \times 161 \times 127 = 1.285\text{W}$$

Now the total heat transfer from the element ($q_{\substack{total\\elemant}}$) that assumed to carry out the calculations ,and according to equation (5.12) the ($q_{\substack{total\\elemant}}$) can be calculated:

$$q_{\substack{total\\elemant}} = 2\times 0.95 + 1.285 = 3.185\,W$$

Now the number of elements that needed to perform the gas cooler can be determined by dividing the total heat transfer through the gas cooler which determined in chapter three in section 3.3 (cycle calculations) by the element total heat transfer , by using the following equation:

$$Q_{\substack{total\,gas\\cooler}} = n\times q_{\substack{total\\elemant}} \tag{5.21}$$

Where:

n: Number of element

$9576 = n\times 3.185$

$n = 3007$ element.

After determining the number of elements the total equivalent length of the gas cooler tube can be calculated by the following equation [reference 11]:

$$L_{\substack{total\\gas\ cooler}} = n\times L_{elem.} \tag{5.22}$$

Where:

$L_{elem.}$: the element length (m).

The length of element can be determined by the following equation:

$$L_{elem.} = L_{B_1-B_2} + t_f \tag{5.23}$$

$$L_{elem.} = 2\times 10^{-3} + 2\times 10^{-3} = 4\times 10^{-3}\,m = 4mm$$

Then the total equivalent length can be determined according to equation (5.22):

$$L_{\substack{total \\ gas\ cooler}} = 3007 \times 4 \times 10^{-3} = 12m$$

Now the external area of gas cooler can be calculated according to the following equation [reference 11]:

$$A_o = 2\pi r_o (L-(t_f \times n)) + n \times A_{fin} \qquad (5.24)$$

$$A_o = \{2\pi \times 5 \times 10^{-3} \times (12 - (3007 \times 2 \times 10^{-3}))\} + \{3007 \times 10 \times 10^{-6}\} = 0.218m^2$$

By comparing the calculations of gas cooler without fin with the calculations of gas cooler with fin its more economic to execute the gas cooler with fins and its more effective in heat transfer, so figure 5.3.a shows the suggested (according to calculations) construction of the gas cooler. let it contains 7 rows of pipes and 3 columns, figure 5.3.b shows side view of the gas cooler, and figure 5.3.c shows 3-dimensions of the gas cooler.

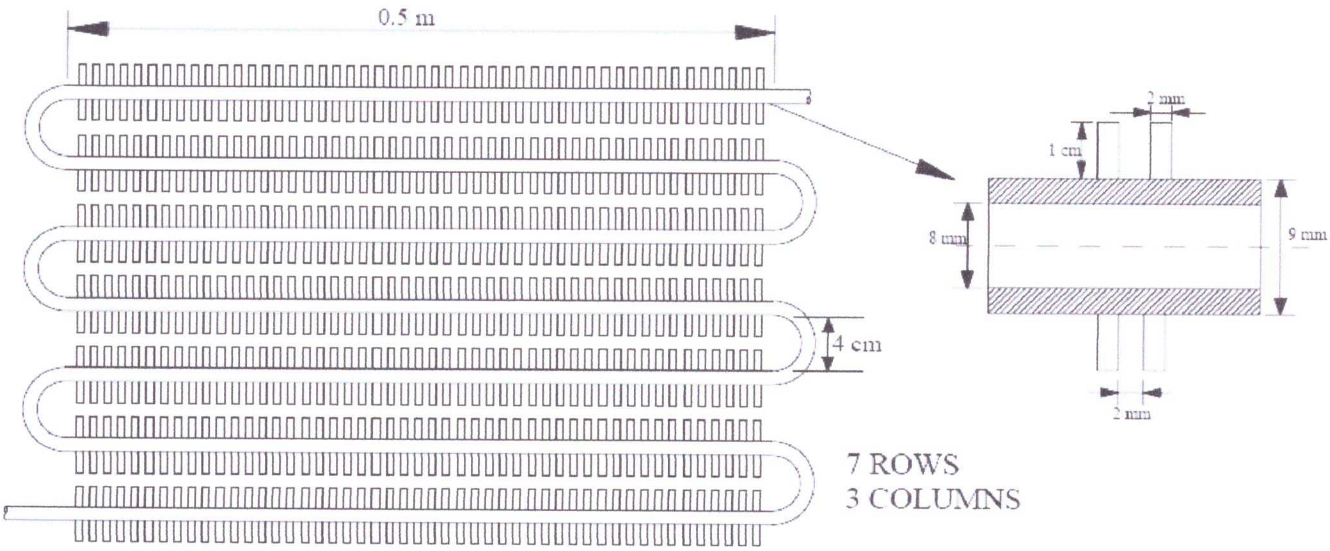


Figure 5.3.a gas cooler dimensions.

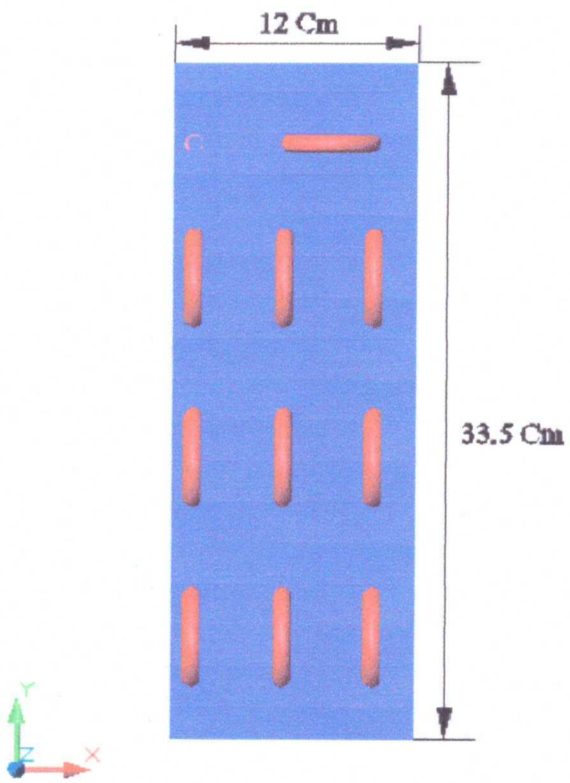


Figure 5.3.b side view of gas cooler

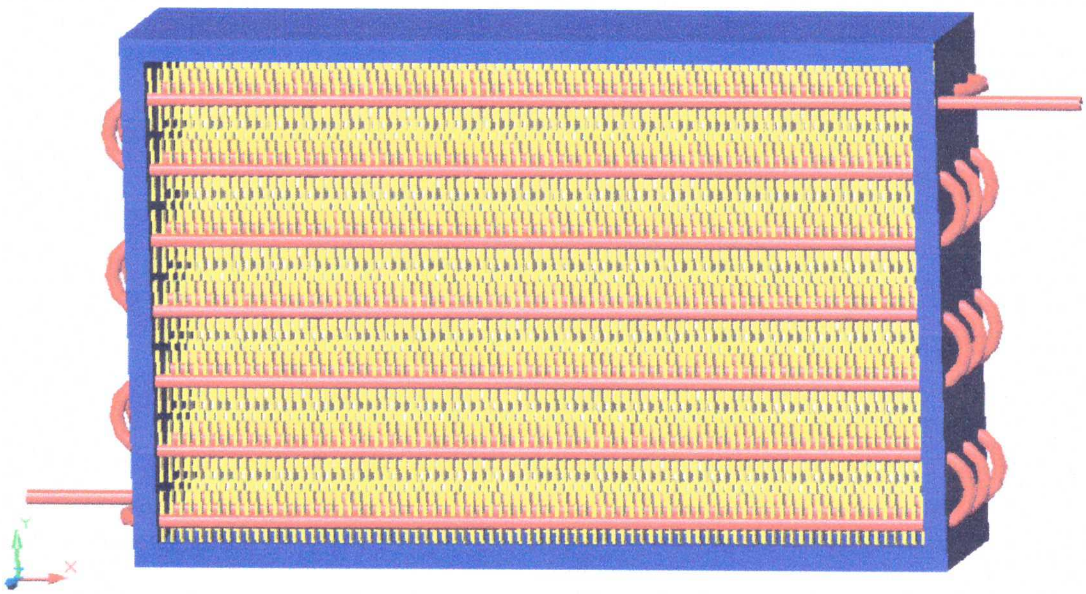


Figure 5.3.c gas cooler in 3- dimensions.

5.2 Coil Design

5.2.1 Coil design without fins

The coil location in this project between the heat exchanger and the expansion valve ,and its work to reduce the temperature of refrigerant which comes from the heat exchanger, and the heat transfer occurs between the refrigerant inside the coil tube and the surrounding medium which is air at atmospheric pressure.

The design of the coil required many calculations such as fluid mechanical calculation , thermal calculation and area calculation, the sequence of design starts with fluid mechanical calculation until reaching area calculation , the calculation start according to equation (5.1).

$$Q = UA\,\Delta t$$

The heat that transferred through the coil was determined in chapter three in section 3.3 (cycle calculations) and its value was $Q_{\text{total coil}} = 1.512\ \text{kW}$, and in order to determine (U) the equation (5.2) is used:

$$U_i = \frac{1}{\frac{1}{h_i} + \frac{r_i \ln(r_o/r_i)}{k} + \frac{r_i}{r_o} \times \frac{1}{h_o}}$$

To determine the value of U there are many parameters should be determined before reaches U value , and it will be started with the flow rate of refrigerant , radiuses ,then the heat transfer coefficients , and all calculations done at the entrance of the coil, so the CO2 properties were determined according to the state at entrance of the coil.

The refrigerant temperature at entrance of the coil 47 °C. And all properties of CO_2 mention in (appendix A, table A-3) .

The volumetric flow rate (Q)is determined by using equation (5.3) and after seeing

The value of $\dot{m}$ which determined in chapter three in section 3.3, $\dot{m} = 0.063\,\text{kg/s}$ and the velocity of refrigerant table (A-4 in appendix A) V=14 m/s so:

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

$$Q = \dot{m}v = 0.063 \times 0.00422 = 2.6586 \times 10^{-4}\,\text{m}^3/\text{s}$$

After determining the value of Q the value of cross sectional area of coil tube can be calculated by using the same equation before:

$$A = \frac{Q}{V} = \frac{2.6586 \times 10^{-4}}{14} = 1.899 \times 10^{-5}\ \text{m}^2$$

After determining the cross section area of the tube the diameter can be determined by using equation (5.4) :

$$A = \frac{\pi}{4} d_i^2$$

$$d_i = \sqrt{\frac{1.899 \times 10^{-5} \times 4}{\pi}} = 4.91 \times 10^{-3}\,\text{m}$$

$$d_i = 5\text{mm}$$

$$r_i = \frac{d_i}{2} = \frac{5\times10^{-3}}{2} = 2.5\times10^{-3}\,m = 2.5mm$$

The outer radius of the tube can be determined according to the equation (5.5):

$$r_o = r_i + t$$

$$r_o = 2.5\times10^{-3} + 1\times10^{-3} = 3.5\times10^{-3}\,m = 3.5mm$$

$\therefore$

$$d_o = r_o \times 2$$

$$d_o = 3.5\times10^{-3}\times2 = 7\times10^{-3}\,m = 7mm$$

After determining the radiuses of the tube that used for the coil, the heat transfer coefficients will be calculated by calculating the Reynolds ,Prandtls , and Nusselts numbers in order to calculate the inner convection heat transfer coefficient , by using the following equations, starting with calculating Reynolds number according to equation (5.6):

$$Re = \frac{V\times\rho\times d_i}{\mu}$$

$$\rho = \frac{1}{v} = \frac{1}{0.00422} = 236.81\ kg/m^3$$

$$\mu = 20.97\times10^{-6}\ Pa.s$$

$$d_i = 5mm$$

$$Re = \frac{14\times236.81\times5\times10^{-3}}{20.97\times10^{-6}} = 790.496\times10^3$$

According to the value of Reynolds number the flow inside the tube is turbulent flow ,after calculating the value of Reynolds number the value of Prandtls number can be calculated according to equation (5.7):

$$Pr=\frac{c_p\mu}{k}$$

$$c_{p_{CO_2 @ 47^o C\&80\,bar}} = 3.2443kJ/kg = 3244.3J/kg$$

$$k=36.109\times10^{-3}\ W/m^{2o}C$$

$$\mu = 20.97\times10^{-6}\ Pa.s$$

$$Pr=\frac{3244.3\times20.97\times10^{-6}}{36.109\times10^{-3}}=1.884$$

According to the value of Reynolds number and Prandtls number the equation that will be used to calculate the value of Nusselts number determined as

$$1.5 < Pr < 500$$

$$3000 < Re < 10^6$$

Then the Nusselts number(Nu) can be calculated according to the following equation:

$$Nu=0.012(Re^{0.87}-280)Pr^{0.4} \qquad (5.25)$$

$$Nu= 0.012\times((790.496\times10^3)^{0.87}-280)\times1.884^{0.4}=2086.8$$

By determining the value of Nusselts number the value of internal convection heat transfer coefficient (h_i) inside the coil tube can be determined according to equation (5.9) :

$$Nu=\frac{h_i\times d_i}{k}$$

$$2054.04=\frac{h_i\times4.91\times10^{-3}}{36.109\times10^{-3}}$$

$$h_i=\frac{2086.8\times36.109\times10^{-3}}{5\times10^{-3}}=15070.5\ W/m^2.^oC$$

After determining internal heat transfer coefficient (h_i) inside the coil tube now the outer convection heat transfer coefficient(h_o) for outside coil tube can be calculated according to equation (5.10):

$$h_o = 4 + 6 \times V_{car}$$

The velocity of car was determined in chapter three section3.2.1,so V_{car}=22.22m/s and:

$$h_o = 4 + 6 \times 22.22 = 137.32 \text{ W/m}^2.°\text{C}$$

After determining h_o, h_i ,inner radius and outer radius of coil the value of overall heat transfer coefficient (U) is calculated by using equation (5.2):

$$U_i = \frac{1}{\frac{1}{h_i} + \frac{r_i \ln(r_o/r_i)}{k} + \frac{r_i}{r_o} \times \frac{1}{h_o}}$$

$$U_i = \frac{1}{\frac{1}{15105.76} + \frac{2.5\times10^{-3} \ln(3.5\times10^{-3}/2.5\times10^{-3})}{401} + \frac{2.5\times10^{-3}}{3.5\times10^{-3}} \times \frac{1}{137.32}}$$

$$U_i = 189 \text{ W/m}^2.°\text{C}$$

After determining the value of (U) it can be calculated the area of coil can be calculated according to equation (5.1):

$$Q_{\substack{total\\coil}} = U_i A_i \ \Delta t$$

$$1512 = 189 \times A_i \times (47 - 40) \quad A_i = \frac{1512}{189 \times 7} = 1.143\text{m}^2$$

By determine the internal area (A_i) of coil the total equivalent length of the coil tube can be calculated according to equation (5.11) :

$$A_i = 2\pi r_i L$$

$$1.143 = 2\pi \times 2.5 \times 10^{-3} \times L$$

$$L = \frac{1.143}{2\pi \times 2.5 \times 10^{-3}} = 72.76\text{m}$$

Then now the external area of coil can be determined by using the following equation:

$$A_o = 2\pi r_o L$$

$$A_o = 2\pi \times 3.5 \times 10^{-3} \times 72.76 = 1.6\text{m}^2$$

5.2.2 Coil design with fins

All the previous calculations for coil without using fins on the tube and the next calculations for coil with fin and to do this calculations it needs to be applied on an element and figure 5.4 show the element and its dimensions.

In order to calculate the total equivalent length of finned tube and external area of coil according to equation (5.12):

$$q_{total} = q_f + q_{original}$$

The calculation will start with heat transfer rate per fin (q_f) and then for heat transfer from tube without fin ($q_{original}$) , and the calculation of (q_f) depends on the geometric of the fin and figure 5.5 show the fin geometer and the dimensions of the fin.

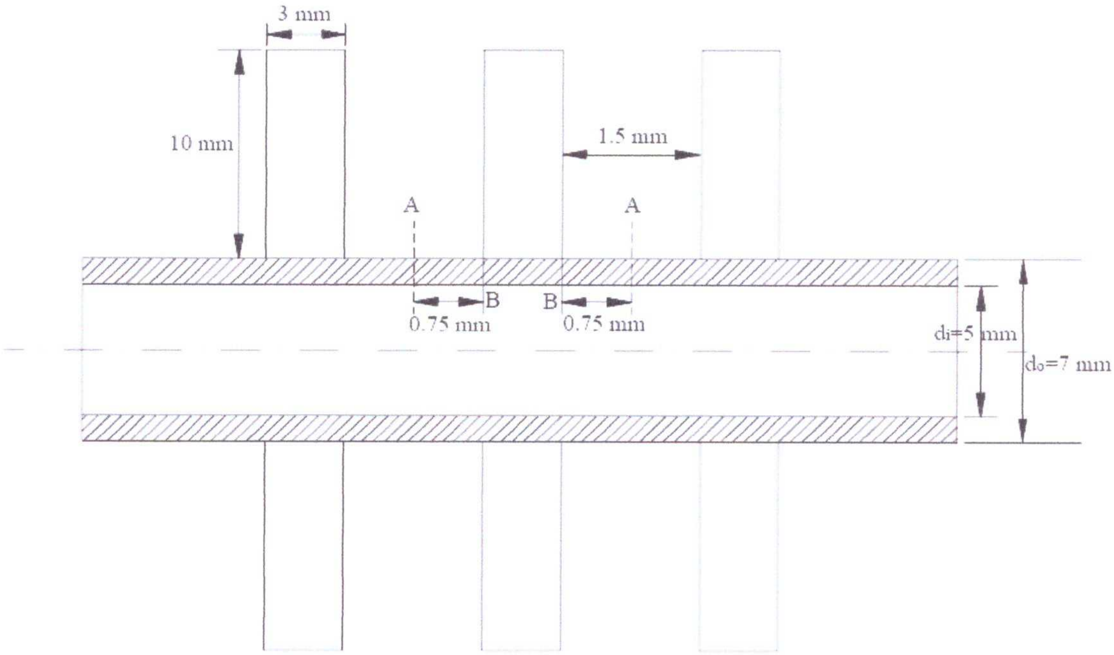


Figure 5.4 element and its dimensions

and (q_f) can be calculated according to equation (5.13) :

$$q_f = \sqrt{hPkA} \times (T_o - T_\infty) \times \frac{\sinh(mL_c) + \frac{h}{mk}\cosh(mL_c)}{\cosh(mL_c) + \frac{h}{mk}\sinh(mL_c)}$$

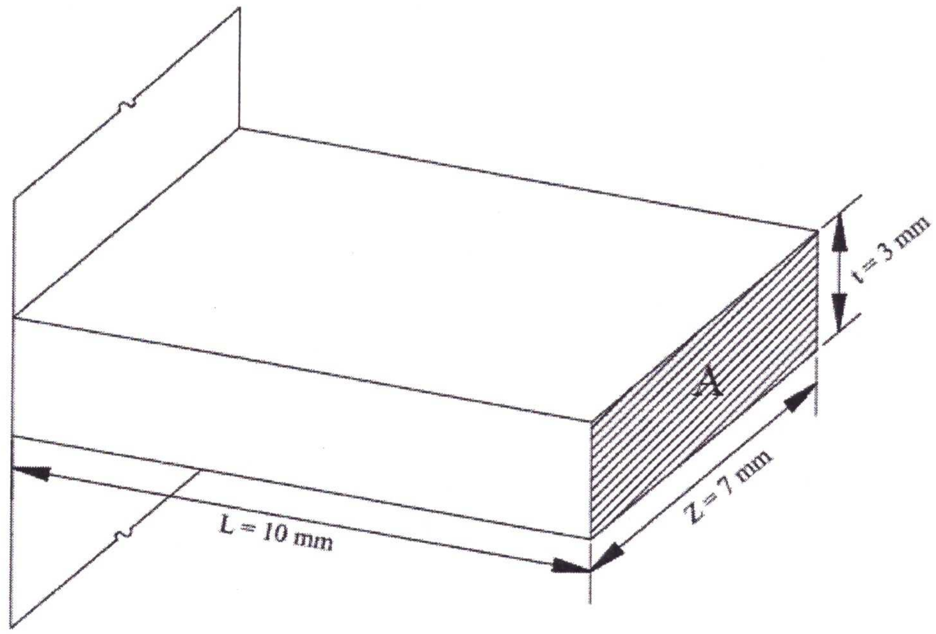


Figure 5.5 fin geometer and dimensions

In this equation there are many parameters need to be determined such as perimeter ,area of fin, L_c ,and m and the value of (P) can be obtained according to equation (5.14):

$$P = 2t + 2z$$

$$P = (2 \times 3 \times 10^{-3}) + (2 \times 7 \times 10^{-3}) = 20 \times 10^{-3} = 20\,\text{mm}.$$

$$A = t \times z = 3 \times 10^{-3} \times 7 \times 10^{-3} = 21 \times 10^{-6}\,\text{m} = 21\,\text{mm}$$

$$k_{Al} = 200\ \text{W/m.°C}$$

And the value of factor (m) can be determined according to equation (5.15) and its dimensionless factor :

$$m = \sqrt{\frac{hP}{kA}}$$

$$m=\sqrt{\frac{137.32\times 20\times 10^{-3}}{200\times 21\times 10^{-6}}}=21.046$$

and the value of (L_c) can be calculated according to equation (5.16) [reference 11]:

$$L_c = L+\frac{d_o}{4}$$

$$L_c = 10\times 10^{-3}+\frac{7\times 10^{-3}}{4}=11.75\times 10^{-3}\,\text{m}=11.75\,\text{mm}$$

After determining all the parameters the heat transfer rate per fin (q_f) can be calculated by the equation (5.13):

$$q_f=\sqrt{137.32\times 20\times 10^{-3}\times 200\times 21\times 10^{-6}}\times(47-40)\times$$
$$\times\frac{\sinh(21.046\times 11.75\times 10^{-3})+\dfrac{137.32}{21.046\times 200}\cosh(21.046\times 11.75\times 10^{-3})}{\cosh(21.046\times 11.75\times 10^{-3})+\dfrac{137.32}{21.046\times 200}\sinh(21.046\times 11.75\times 10^{-3})}$$

$$q_f=0.205\text{ W}$$

To calculate the efficiency of the fin(η_f) the equation is used (5.17):

$$\eta_f=\frac{\tanh(mL_c)}{mL_c}$$

$$\eta_f=\frac{\tanh(21.046\times 11.75\times 10^{-3})}{21.046\times 11.75\times 10^{-3}}=0.98=98\%$$

This result of η_f shows the amount of heat transfer from the fin is 98% from the total of heat that fin is exposed.

After determining the amount of q_f the amount of heat transfer rate from tube without fin ($q_{original}$)will be determined according to equation (5.18):

$$q_{original} = q_{A_1-A_2} = UA\Delta t$$

The section $A-A$ is shown in figure 5.4 and its dimensions, the surface area of heat transfer is obtain according to equation (5.19):

$$A = 2\pi r_o L$$

The length of tube without fin can be obtained from figure 5.4 , or be calculated according to equation (5.20) :

$$L = L_{B_1-B_2} - t_f$$

$$L = 4.5\times10^{-3} - 3\times10^{-3} = 1.5\times10^{-3}\,m = 1.5mm$$

Then :

$$A = 2\pi\times3.5\times10^{-3}\times1.5\times10^{-3} = 3.299\times10^{-5}\,m^2$$

By determining the area of heat transfer surface the amount of heat transfer rate from tube without fin ($q_{original}$) can be calculated according to equation(5.18):

$$q_{original} = 3.299\times10^{-5}\times189\times7 = 0.0436\,W$$

Now the total heat transfer from the element ($q_{\substack{total\\elemant}}$)that assumed to carry out the calculations ,and according to equation (5.12) the ($q_{\substack{total\\elemant}}$) can be calculated :

$$q_{\substack{total\\elemant}} = (2\times0.205) + 0.0436 = 0.454W$$

Now the number of elements that needed to perform the coil can be determined by dividing the total heat transfer through the coil which was determined in chapter three in section 3.3 (cycle calculations) on the element total heat transfer , according to equation (5.21):

$$Q_{\substack{total\\coil}} = n \times q_{total}$$

$$1512 = n \times 0.454$$

$$n = 3331 \text{element.}$$

After determining the number of elements the total equivalent length of the coil tube can be calculated by the following equation(5.22) :

$$L_{\substack{total\\coil}} = n \times L_{elem.}$$

The length of element can be determined according to (5.23) :

$$L_{elem.} = L_{B_1-B_2} + t_f$$

$$L_{elem.} = 1.5 \times 10^{-3} + 3 \times 10^{-3} = 4.5 \times 10^{-3} m = 4.5 mm$$

Then the total equivalent length can be determined according to equation (5.22):

$$L_{\substack{total\\coil}} = 3331 \times 4.5 \times 10^{-3} = 15 m$$

Now the external area of coil can be calculated according to equation (5.24):

$$A_o = 2\pi r_o (L\text{-}(t_f \times n)) + n \times A_{fin}$$

$$A_o = \{2\pi \times 3.5 \times 10^{-3} \times (15 - (3331 \times 3 \times 10^{-3}))\} + \{3331 \times 21 \times 10^{-6}\} = 0.18 m^2$$

By comparing the calculations of coil without fin with the calculations of coil with fins its more economic to execute the coil with fin and its more effective in heat transfer ,so the figure 5.6 shows the suggested construction of the coil according to calculation .

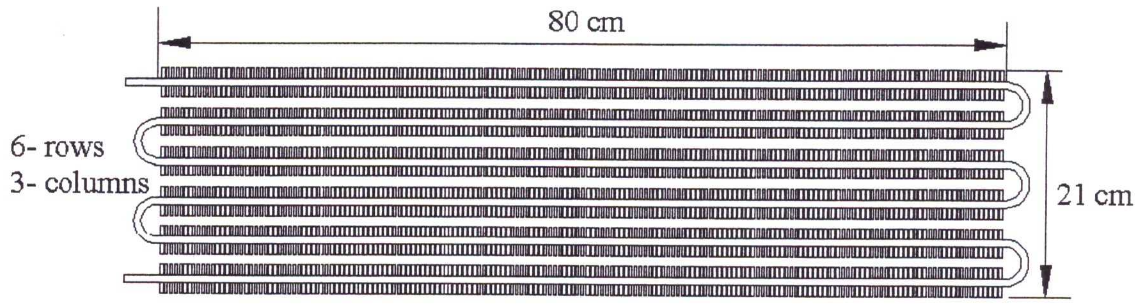


Figure 5.6 construction of the coil

5.3 Heat Exchanger Design

The heat exchanger in this cycle is used to reducing the temperature of the refrigerant (CO_2) that comes from gas cooler outlet to coil inlet and superheating the refrigerant (CO_2) that comes from evaporator outlet to compressor inlet , and the type of the heat exchanger which is used in this project is double pipe counter flow, and in this heat exchanger the heat transfer occur between the same refrigerant in both sides of heat exchanger and since this cycle the same flow rate ($\dot{m}$) of refrigerant is in both sides of heat exchanger. The figure 5.7 schematic drawing for double heat exchanger explaining the flow in the heat exchanger.

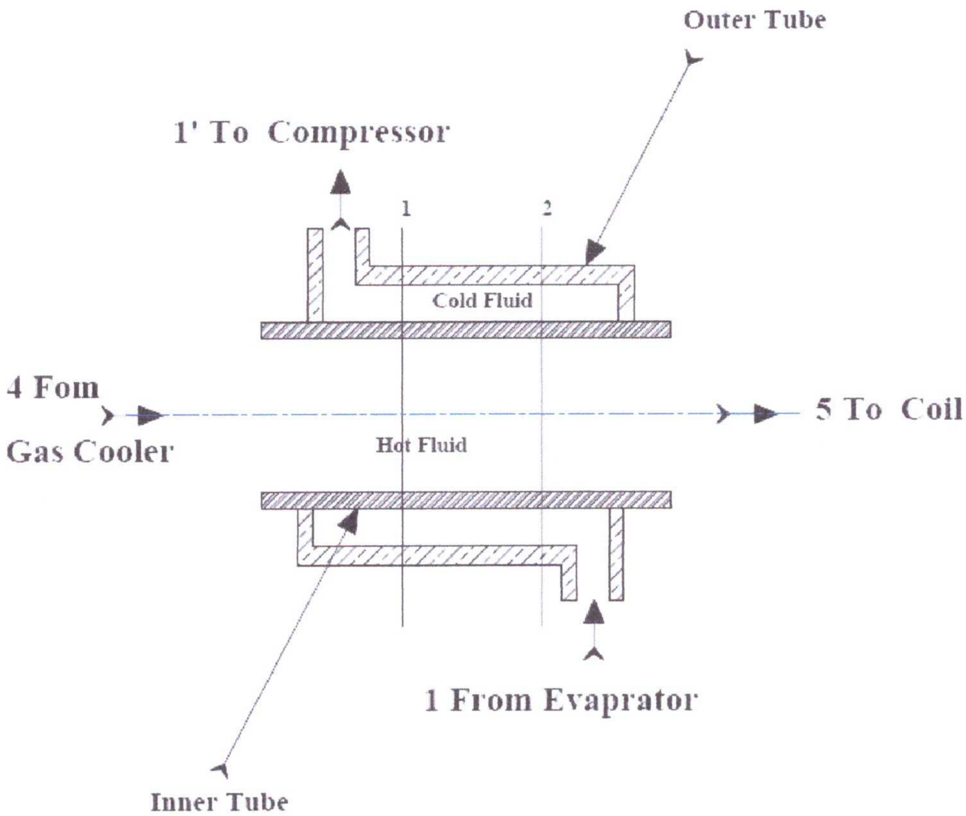


Figure 5.7 schematic drawing for double heat exchanger

The calculations of heat exchanger are similar to gas cooler and the main different between them is the heat transfer occurs between the same fluid , so the diameters of pipes will be calculated, convection heat transfer coefficient ,overall heat transfer coefficient, external area , and the total equivalent length of the heat exchanger, the calculations will starts with assuming the temperature of pipe wall as $t_{wall} = 31°C$, and the first calculation for the inner pipe (tube) in the heat exchanger start with determining reference temperature ($t_{ref.}$) for the inner fluid by the following equation:

$$t_{ref.h} = \frac{\bar{t}_m + t_{wall}}{2} \tag{5.26}$$

Where :

$t_{ref.h}$: The reference tempertaure for the hot fluid (°C).

$\bar{t}_m$: The mean average tempertaure for the fluid in the tube (°C).

And the mean average temperature ($\bar{t}_m$) can be obtained by the following equation:

$$\bar{t}_m = \frac{t_i + t_o}{2} \tag{5.27}$$

Where:

t_i : The tempertaure at the inlet of the tube (°C).

t_o : The tempertaure at the outlet of the tube (°C).

And from P-h diagram chapter three section 3.3 the vales of t_i and t_o for the hot fluid was as the following:

$t_i = t_4 = 57°C$

$t_o = t_5 = 47°C$

For the hot fluid and ($\bar{t}_m$) can be calculated according to equation (5.27):

$$\bar{t}_m = \frac{57+47}{2} = 52°C$$

And so ($t_{ref.h}$) can be calculated according to equation (5.26):

$$t_{ref.h} = \frac{52+31}{2} = 41.5°C$$

Now all the properties for the inner refrigerant(CO_2) will be taken at $t_{ref.h}$ and those properties are mentioned (appendix A, table A-3).

The volumetric flow rate (Q)is determine by using equation (5.3) and after seeing The value of $\dot{m}$ which was determined in chapter three in section 3.3, $\dot{m} = 0.063\,kg/s$, and the velocity of refrigerant was given in table 4 in appendix A And its value V=12m/s so

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

$$Q = \dot{m}\,v = 0.063 \times 0.00372 = 2.344 \times 10^{-4}\,m^3/s$$

After determining the value of Q the value of cross sectional area of coil tube can be calculated by using the same equation before:

$$A = \frac{Q}{V} = \frac{2.344 \times 10^{-4}}{12} = 1.95 \times 10^{-5}\,m^2$$

After determining the cross sectional area of the tube then the diameter can be determined by using equation (5.4) :

$$A = \frac{\pi}{4} d_i^2$$

$$d_i = \sqrt{\frac{4 \times 1.95 \times 10^{-5}}{\pi}} = 5mm$$

$$r_i = \frac{d_i}{2} = \frac{5\times10^{-3}}{2} = 2.5\times10^{-3}\,m = 2.5mm$$

The outer radius of the tube can be determined according to the equation (5.5):

$$r_o = r_i + t$$

$$r_o = 2.5\times10^{-3} + 1\times10^{-3} = 3.5\times10^{-3}\,m = 3.5mm$$

$\therefore$

$$d_o = r_o \times 2$$

$$d_o = 3.5\times10^{-3}\times2 = 7\times10^{-3}\,m = 7mm$$

After determining the radiuses of the inner tube that used for the heat exchanger , the heat transfer coefficients will be calculated by calculating the Reynolds ,Prandtls , and Nusselts numbers in order to calculate the inner convection heat transfer coefficient , by using the following equations, start with calculating Reynolds number for inner tube according to equation (5.6):

$$Re = \frac{V\times\rho\times d_i}{\mu}$$

$\rho = 269.095\ kg/m^3$

$\mu = 20.015\times10^{-6}\ Pa.s$

$d_i = 5mm$

$$Re_i = \frac{12\times5\times10^{-3}\times269.095}{20.015\times10^{-6}} = 968016$$

According to the value of Reynolds number the flow inside the tube is turbulent flow ,after calculating the value of Reynolds number the value of Prandtls number for inner tube can be calculated according to equation (5.7):

$$Pr = \frac{c_p\mu}{k}$$

$$c_{p_{CO_2 @ 41.5^\circ C \& 80\,bar}} = 4.58435 kJ/kg = 4581.35 J/kg$$

$$k_i = 41.185 \times 10^{-3}\, W/m.^\circ C$$

$$\mu_i = 20.0154 \times 10^{-6} m^2/s$$

$$Pr = \frac{4581.35 \times 20.015 \times 10^{-6}}{41.187 \times 10^{-3}} = 2.226$$

According to the value of Reynolds number and Prandtls number the equation which will be used to calculate the value of Nusselts number as

$$1.5 < Pr < 500$$

$$3000 < Re < 10^6$$

Then the internal Nusselts number(Nu) for the inner tube can be calculated according to (5.25):

$$Nu_i = 0.012(Re^{0.87} - 280)\, Pr^{0.4}$$

$$Nu_i = 0.012((968016)^{0.87} - 280)(2.226)^{0.4} = 2662$$

By determining the value of Nusselts number the value of internal convection heat transfer coefficient (h_i) inside the inner tube of the heat exchanger can be determined according to equation (5.9) :

$$Nu = \frac{h_i \times d_i}{k}$$

$$2662 = \frac{h_i \times 5 \times 10^{-3}}{41.185 \times 10^{-3}}$$

$$h_i = \frac{2662 \times 41.185 \times 10^{-3}}{5 \times 10^{-3}} = 21927\, W/m^2.^\circ C$$

After determining internal heat transfer coefficient (h_i) inside the inner tube of the heat exchanger the outer convection heat transfer coefficient(h_o) for outer tube of heat exchanger will be calculated by the same method that used to find(h_i), so now is required to determine ($t_{ref.}$) for refrigerant which flows in the outer tube by the following equation:

$$t_{ref.c} = \frac{\bar{t}_m + t_{wall}}{2} \tag{5.28}$$

Where :

$t_{ref.c}$: The reference tempertaure for the cold fluid (°C).

$\bar{t}_m$: The mean average tempertaure for the fluid in the tube (°C).

And the mean average temperature ($\bar{t}_m$) can be obtained according to equation (5.27):

$$\bar{t}_m = \frac{t_i + t_o}{2}$$

And from P-h diagram chapter three section 3.3 the vales of t_i and t_o for the cold fluid was as the following:

$$t_i = t_1 = 7°C$$
$$t_o = t_{1'} = 12°C$$

For the cold fluid and ($\bar{t}_m$) can be calculated according to equation (5.27):

$$\bar{t}_m = \frac{7+12}{2} = 9.5°C$$

And so ($t_{ref.c}$) can be calculated according to equation (5.28):

$$t_{ref.c} = \frac{9.5+31}{2} = 20.25°C$$

Now all the properties for the inner and outer refrigerant(CO_2) will be taken at $t_{ref.c}$ and those properties are mentioned (appendix A, table A-3).

The volumetric flow rate (Q)is determined by using equation (5.3):

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

And after seeing The value of $\dot{m}$ which was determined in chapter three in section 3.3, $\dot{m} = 0.063\,kg/s$, and the inner diameter for the outer tube was assumed as

$d_{i_o} = 15\times10^{-3}m = 15mm$ so

$$r_i = \frac{d_{i_o}}{2} = \frac{15\times10^{-3}}{2} = 7.5\times10^{-3}m = 7.5mm$$

The outer radius of the outer tube can be determined according to the equation (5.5):

$$r_o = r_i + t$$

$$r_o = 7.5\times10^{-3} + 1\times10^{-3} = 8.5\times10^{-3}m = 8.5mm$$

$\therefore$

$$d_o = r_o \times 2$$

$$d_o = 8.5\times10^{-3}\times2 = 17\times10^{-3}m = 17mm$$

and in order to calculate the internal cross sectional area of outer tube (A_{i_o}) the following equation:

$$A_{i_o} = \frac{\pi}{4}\left(d_{i_o}^{\ 2} - d_{o_i}^{\ 2}\right) \tag{5.29}$$

Where:

d_{i_o} : The inner diameter of the outer tube for the heat exchanger (m).

d_{o_i} : the outer diameter of the inner tube for the heat exchanger (m).

$$A_{i_o} = \frac{\pi}{4}\left(\left(15\times10^{-3}\right)^2 - \left(7\times10^{-3}\right)^2\right)$$

$$A_{i_o} = 1.3823\times10^{-4}m^2$$

Then

$$Q = \dot{m}\,v = 0.063 \times 9.4451 \times 10^{-3} = 5.95045 \times 10^{-4}\,m^{-3}/s$$

After determining the value of Q the value of the velocity of refrigerant which flows in side the outer tube of heat exchanger can be calculated by using the same equation before (5.3):

$$V = \frac{Q}{A} = \frac{5.95045 \times 10^{-4}}{1.3823 \times 10^{-4}} = 4.305\,m/s$$

After determining the radiuses of the outer tube that will be use for heat exchanger and velocity of refrigerant which flows in side the outer tube , then the heat transfer coefficients will be calculated by calculating the Reynolds ,Prandtls , and Nusselts numbers in order to calculate the inner convection heat transfer coefficient , by using the following equations, start with calculating Reynolds number according to equation (5.6):

$$Re = \frac{V \times \rho \times d_i}{\mu}$$

$\rho = 105.8744\ kg/m^3$

$\mu = 15.9095 \times 10^{-6}$ Pa.s

$d_e = d_{i_o} - d_{o_i}$

Where :

d_e : The hydraulic diameter (m)

$d_e = 15 \times 10^{-3} - 7 \times 10^{-3} = 8 \times 10^{-3}$

$$Re = \frac{105.8744 \times 4.305 \times 8 \times 10^{-3}}{15.9095 \times 10^{-6}} = 229191$$

According to the value of Reynolds number the flow inside the tube is turbulent flow ,after calculating the value of Reynolds number the value of Prandtls number can be calculated according to equation (5.7):

$$Pr = \frac{c_p \mu}{k}$$

$$c_{p_{CO_2 @ 20.25^\circ C\&80bar}} = 1.63682 kJ/kg = 1636.28 J/kg$$

$$k = 21.8949 \times 10^{-3} \ W/m^{2\circ}C$$

$$\mu = 15.9095 \times 10^{-6} \ Pa.s$$

$$Pr = \frac{16636.28 \times 15.9095 \times 10^{-6}}{21.8949 \times 10^{-3}} = 1.1889$$

According to the value of Reynolds number and Prandtls number the equation will be used to calculate the value of Nusselts number can be determined as

$$0.5 < Pr < 1.5$$

$$10^4 < Re < 5 \times 10^6$$

Then the equation (5.8) can be used to calculate the Nusselts number(Nu):

$$Nu = 0.0214\left(Re^{0.8} - 100\right)Pr^{0.4}$$

$$Nu = 0.0214\left((229191)^{0.8} - 100\right)(1.1889)^{0.4} = 442.98$$

By determining the value of Nusselts number the value of external convection heat transfer coefficient (h_o) inside the outer tube of the heat exchanger can be determined according to equation (5.9) :

$$Nu = \frac{h_o \times D_i}{K}$$

$$442.98 = \frac{h_\circ \times 8 \times 10^{-3}}{21.8949 \times 10^{-3}}$$

$$h_\circ = \frac{442.98 \times 21.8949 \times 10^{-3}}{8 \times 10^{-3}} = 1212.4\,\mathrm{W/m^2.°C}$$

After determining h_o, h_i ,inner radius and outer radius of the heat exchanger the value of overall heat transfer coefficient (U) is calculated by using equation (5.2):

$$U_i = \frac{1}{\frac{1}{h_i} + \frac{r_i \ln(r_o/r_i)}{k} + \frac{r_i}{r_o} \times \frac{1}{h_o}}$$

$$U_i = \frac{1}{\frac{1}{21927} + \frac{2.5 \times 10^{-3} \ln\left(3.5 \times 10^{-3}/2.5 \times 10^{-3}\right)}{401} + \frac{1}{1212.4}}$$

$$U_i = 1570.22\,\mathrm{W/m^2.°C}$$

After determining the value of (U) the area of heat exchanger can be calculated by using the log mean temperature difference (LMTD) method according to the following equation:

$$Q_{\substack{\text{total} \\ \text{heat exch.}}} = U_i \times A_i \times \Delta T_m \qquad (5.30)$$

Where:

U_i : The overall heat transfre coefficient (W/m^2.°C).

A_i : Internal surface area for heat transfer (m^2).

ΔT_m : Mean temperature difference across heat exchanger(°C).

The $Q_{\substack{\text{total} \\ \text{heat exch.}}}$ can be calculated from hot fluid side by using the following equation:

$$Q_{\substack{total \\ heat\ exch.}} = Q_{hot} = \dot{m}_h \, c_{p_h} \Delta T_h \tag{5.31}$$

Where:

$\dot{m}_h$ = The mass flow rate for the hot fluid (kg/s).

c_{p_h} = The specific heat at constant pressuer for the hot fluid (kJ/kg.°C).

ΔT_h = The temperature difference beteween the inlet and outlet temperatures in the hot fluid tube (°C).

And :

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

$$c_{p_{h@57°C\&82bar}} = 2.1062 \text{kJ/kg.°C} = 2106.2 \text{J/kg.°C}$$

$$\Delta T_h = t_i - t_o = t_4 - t_5 = 57 - 47 = 10°$$

$$Q_{\substack{total \\ heat\ exch.}} = 0.063 \times 2106.2 \times 10 = 1327 \text{W}$$

After determining $Q_{\substack{total \\ heat\ exch.}}$ the value of mean temperature difference (ΔT_m) by the

following equation that use for counter flow in double pipe heat exchanger :

$$\Delta T_m = \frac{(T_{h2} - T_{c2}) - (T_{h1} - T_{c1})}{\ln[T_{h2} - T_{c2} / T_{h1} - T_{c1}]} \tag{5.32}$$

Where:

T_{h1}: The temperature at the inlet of hot fluid pipe (°C).

T_{h2}: The temperature at the outlet of hot fluid pipe (°C).

T_{c1}: The temperature at the outlet of cold fluid pipe(°C).

T_{c2}: The temperature at the inlet of cold fluid pipe (°C).

All these temperatures can be obtained from the figure (5.8) which shows the temperatures distribution in the heat exchanger.

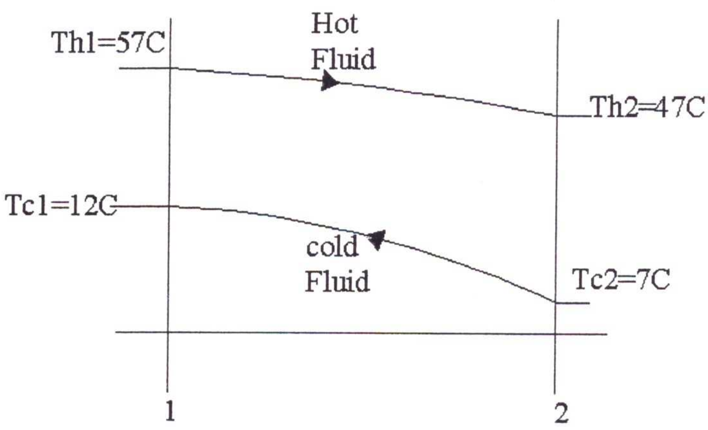


Figure 5.8 temperatures distribution

and the value of (ΔT_m) can be calculated according to equation (5.32):

$$\Delta T_m = \frac{(47-7)-(57-12)}{\ln[47-7/57-12]}$$

$$\Delta T_m = 42.45°C$$

After determining the value of $Q_{\substack{total \\ heat\ exch.}}$, U, and ΔT_m ,then the internal surface area of the heat exchanger can be calculated according to equation(5.30):

$$1327 = 1570.22 \times A_i \times 42.45$$

$$A_i = \frac{1327}{1570.22 \times 42.45} = 0.01991 m^2$$

By determine the internal area (A_i) of the heat exchanger , the total equivalent length of the heat exchanger tube can be calculated according to equation (5.11) :

$$A_i = 2\pi r_i L$$

$$0.01991 = 2\pi \times 2.5 \times 10^{-3} \times L$$

$$L = \frac{0.01991}{2\pi \times 2.5 \times 10^{-3}} = 1.2675\text{m} \simeq 1.27\text{m}$$

Then now the external area(A_o) of the heat exchanger can be determined by using the equation (5.19):

$$A_o = 2\pi r_{o_o} L$$

$$A_o = 2\pi \times 8.5 \times 10^{-3} \times 1.27 = 0.0678\text{m}^2$$

After calculating the total equivalent length of the heat exchanger the figure 5.9.a shows the dimensions of pipes, and figure 5.9.b shows the suggested construction of double pipe heat exchanger in three dimensions according to calculation.

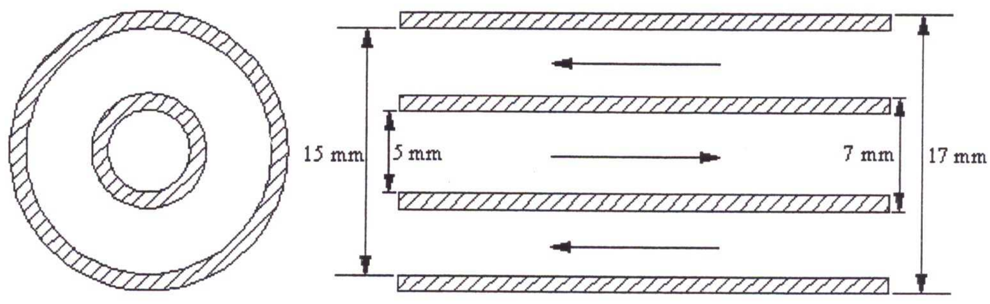


figure 5.9.a dimensions of double pipes

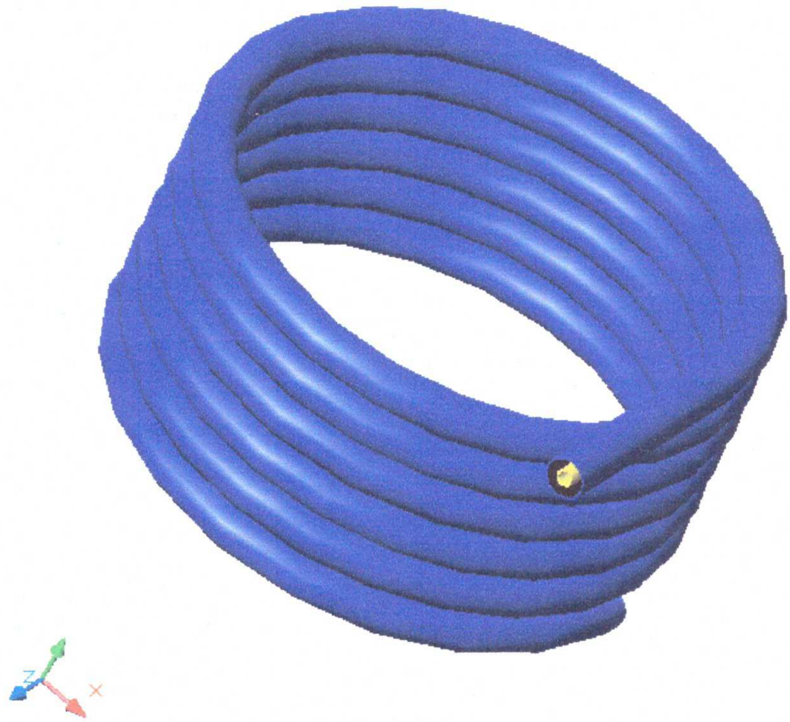


Figure5.9.b double pipe heat exchanger in three dimensions.

5.4 Evaporator Design

In order to design an effective evaporator ,several specific data must be known such as geometry, some physical properties and others will be mentioned later.

As it is mentioned in the previous paragraph that geometrical data must be known ; it is not only length ,high, and width of evaporator or diameters of is pipes, but there will be also what is called fins which is added over tubes in order to increase the heat transfer area directly increasing the evaporator efficiency.

Here are basic geometrical data:

L:length of evaporator =300mm

W:width of evaporator=256 mm

$\dot{V}$:flow rate of air through evaporator.

n_1:number of rows of evaporator=8rows

n_2:number of columns of evaporator=5columns

d_{in}:the inner diameter of evaporater tubes=10mm

δ_{tube}:thickness of tubes =1mm

d_{out} : the outer diameter of evaporater tubes=12mm

S_1:step of pipe in width=50mm

S_2:step of pipe in length=50mm

S_{fin}:step of fin=3mm

D_{fin}:total length of fin=32mm

h_{fin}:height of fin=10mm

δ_{fin}:thickness of fin=1mm

n:number of finned elment

Volumetric flow rate of air through evaporator can be calculated by referring to psychometric chart. Figure 5.10 shows the cycle on psychometric chart.

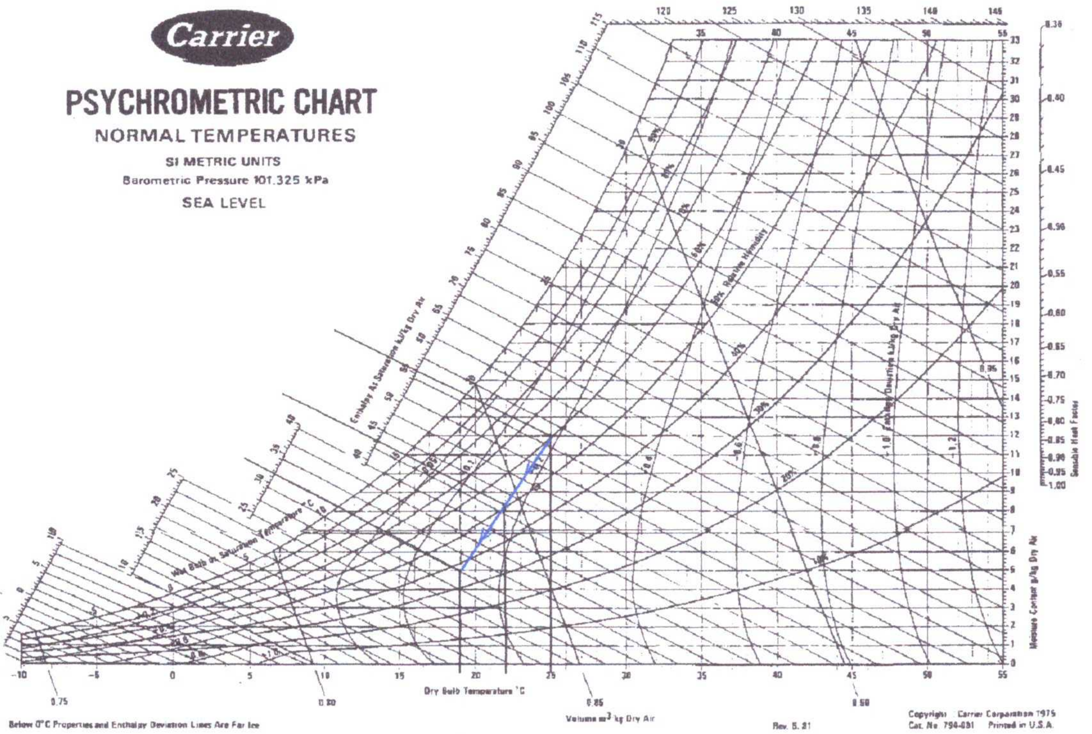


Figure 5.10 the cycle on psychometric chart

Let:

Temperature of evaporator surface $t_s = 19\,^{\circ}\mathrm{C}$

Temprature inside car $t_{in} = 22\,^{\circ}\mathrm{C}$

Temprature of return air $t_{return} = 25\,^{\circ}\mathrm{C}$

According to these temperature and by refering to psychometric chart:

$h_s = 31.5\,\text{kJ/kg dry air.}$

$h_{in} = 43.5\,\text{kJ / kg dry air.}$

$h_{return} = 56\,\text{kJ / kg dry air.}$

$$\dot{Q}_{total} = \dot{m}_{air} \times (h_{in} - h_s)$$

$$\dot{m}_{air} = \frac{7.538}{(43.5 - 31.5)} = 0.628\,\text{kg/s}$$

specific volume at evaporator surface $(v_s) = 0.835\,\text{m}^3/\text{kg}$

$$\dot{V}_{air} = \dot{m}_{air} \times v_s = 0.628 \times 0.835 = 0.5244\,\text{m}^3/\text{s}$$

According to complex structure of this evaporator the area of every small different part will be calculated accurately as follows the area which indicates spaces between fins and cross sectional area of tubes; it should be mentioned that fins have two faces so one face are must be multiplied be "2" as the following equation [reference 6]:

$$A_{\substack{\text{finned}\\ \text{element}}} = 2(D_{fin})^2 - \frac{\pi}{4} d_{out}^2 + A_{\substack{b/n\\ \text{fins}}} \tag{5.33}$$

where :

$A_{\substack{b/n\\ \text{fins}}}$: the area of the surface between two fins.

$$A_{\substack{\text{finned}\\ \text{element}}} = 2(0.032)^2 - \frac{\pi}{4}(0.012)^2 + \pi \times 0.012 \times 0.003$$

$$A_{\substack{\text{finned}\\ \text{element}}} = 2.05 \times 10^{-3}\,\text{m}^2$$

Inner area of finned element that indicates the area of the direct internal contact between fins and tubes can be calculated by the following equation [reference 6]:

$$A_{inner} = \pi d_{in} S_{fin} \tag{5.34}$$

$$A_{inner} = \pi \times 0.01 \times 0.003 = 9.42 \times 10^{-5}\,m^2$$

The maximum cross sectional area of evaporator can be calculated by the following equation:

$$A_{max} = W \times L \tag{5.35}$$

$$A_{max} = 0.256 \times 0.3 = 0.0768 m^2$$

Minimum cross sectional area of evaporator which allows air to pass through it can be calculated by the following equation [reference 6]:

$$A_{min} = A_{max} - n \times \delta_{fin} \times W - n_1 \times d_{out} \times L \tag{5.36}$$

Where the number of fined elements (n) can be determined by the following equation [reference 6]:

$$n = \frac{L}{\delta_{fin} + S_{fin}} \tag{5.37}$$

$$n = \frac{0.3}{0.001 \times 0.003} = 75 \text{finned element}$$

So:

$$A_{min} = 0.0768 - (75 \times 0.001 \times 0.256) - (8 \times 0.012 \times 0.3)$$

$$A_{min} = 0.0288 m^2$$

Minimum velocity of air while entering the cross sectional area of evaporator can be calculated by the following equation:

$$V_{min} = \frac{\dot{V}}{A_{max}} \tag{5.38}$$

$$V_{min} = \frac{0.5244}{0.0768} = 6.83 m/s$$

And maximum velocity of air while entering the cross sectional area of evaporator can be calculated by the following equation:

$$V_{max} = \frac{\dot{V}}{A_{min}} \tag{5.39}$$

$$V_{max} = \frac{0.5244}{0.0288} = 18.2 m/s$$

The calculation of convection heat transfer coefficient of air outside tubes of evaporator($h'_{f_{air\,out}}$) can be done by the following equation [reference 7]:

$$h'_{f_{air\,out}} = 0.117 (Pr_{air})^{0.35} \left(\frac{K_{air}}{S_{fin}^{0.28}}\right) \left(\frac{d_{out}}{d_{in}}\right)^{-0.54} \left(\frac{h_{fin}}{S_{fin}}\right)^{-0.14} \left(\frac{V_{max}}{\upsilon_{air}}\right)^{0.72} \tag{5.40}$$

where:

K_{air} : Thermal conductivity of air at inlet temperature of evaporator

$K_{air} = 0.02624 W/m^2.°C$

Pr_{air} : Prandtle number of air at inlet temperature of evaporator

$Pr_{air} = 0.708$

υ_{air} : kinematic viscosity of of air at inlet temperature of evaporator

$\upsilon_{air} = 15.63 \times 10^{-6} m^2/s$

$$h'_{f_{air\,out}} = 0.117 (0.708)^{0.35} \left(\frac{0.02624}{0.003^{0.28}}\right) \left(\frac{0.012}{0.01}\right)^{-0.54} \left(\frac{0.01}{0.003}\right)^{-0.14} \left(\frac{18.2}{15.63 \times 10^{-6}}\right)^{0.72}$$

$$h'_{f_{air\,out}} = 128.9 W/m^2.°C$$

see appendix A, table A-5.

But this value can effected by some factors, so the final value for convection heat transfer coefficient of air outside tubes of evaporator($h_{f_{air}}$)can be calculated by the following equation[reference 7]:

$$h_{f_{air}} = h'_{f_{air_{out}}} \times \xi \times \varepsilon \tag{5.41}$$

where:

ξ:moisture drop coefficient

ε:Coefficient that take into account the bad contact
between tubes and fins

$\varepsilon = 0.95$

$\xi = 1.1$

$h_{f_{air}} = 128.9 \times 1.1 \times 0.95$

$h_{f_{air}} = 134.7 W / m^2.°C$

The calculation of convection heat transfer coefficient inside tubes of evaporator can be calculated by the following equation [reference 7]:

$$h_{f_{in_L}} = 0.023(\frac{k_L}{d_{in}})(Re_L)^{0.8}(Pr_L)^{0.4} \tag{5.42}$$

Where:

k_L : Thermal conductivity of liquid refrigerant (CO_2) at inlet
temperature of evaporator($W / m.°C$).

Re_L : Reynolds numberof liquid refrigerant (CO_2) at inlet
temperature of evaporator.

Pr_L : Prandtle number of liquid refrigerant (CO_2) at inlet
temperature of evaporator.

$k_{L_{co_2 @ t=7^O C \& P=42 bar}} = 0.1019 W / m.°C$

$Pr_{co_2 @ t=7^O C \& P=42 bar} = 1.13$

$d_{in} = 10mm$

And to calculate the value of (Re_L) the following equation can be used [reference11]:

$$Re_L = \frac{V \times \rho_L \times d_i}{\mu_L} \qquad (5.43)$$

Where:

V: The velocity of refrigerant (CO_2) in side tube (m/s).

d_i : The internal diameter of tube (m).

ρ_L : The density of liquid refrigerant(CO_2) at the entrance of evaporator (kg/m^3).

μ_L : Dynamic viscosity of liquid refrigerant(CO_2) at the entrance of evaporator (Pa.s).

$$\rho_{L_{co_2 @ t=7^o C \& P=42 bar}} = 185 kg/s$$

$$\mu_{L_{co_2 @ t=7^o C \& P=42 bar}} = 9.11 \times 10^{-5} \text{ Pa.s}$$

$$Re_L = \frac{14 \times 185 \times 10 \times 10^{-3}}{9.11 \times 10^{-5}} = 284.3 \times 10^3$$

From the value of Reynolds number the flow inside the tube is turbulent flow, Now the value of($h_{f_{in_L}}$) can be determined according to equation (5.42):

$$h_{f_{in_L}} = 0.023(\frac{0.1019}{0.01})(284.3 \times 10^3)^{0.8}(1.13)^{0.4}$$

$$h_{f_{in_L}} = 5677.5 W/m^2.^\circ C$$

But this value can be effected by the formation of impurities ,lime and oil layers on the inner surface of tubes of the evaporator ,and final value of convection heat transfer coefficient inside tubes of evaporator ($h_{f_{in}}$) can be calculated by the following equation [reference 7]:

$$h_{f_{in}} = F \times h_{f_{inL}} \tag{5.44}$$

Where:

F: Dittus boelter factor.

Calculation of (F) can be done by use the following equation [reference 7]:

$$F = 1 + 1.925 X_{tt}^{-0.83} \tag{5.45}$$

Where:

X_{tt}:lockhart-Martinelli two phase flow parametr with turbulent flow.

The vale of (X_{tt}) is given by [reference 7]:

$$X_{tt} = \left[\frac{1-x}{x}\right]^{0.9} \left[\frac{\rho_v}{\rho_L}\right]^{0.5} \left[\frac{\mu_L}{\mu_v}\right]^{0.1} \tag{5.46}$$

Where:

x : vapour quality, the mass fraciton of vapour.

ρ_v : Density of vapour refrigerant (CO_2)

at the entrance of evaporator (kg/m^3).

μ_v: Kinematic viscosity of vapour refrigerant(CO_2)

at the entrance of evaporator (Pa.s).

(see Appendix A, table A-5)

$x = 0.53$

$\rho_v = 123.5 kg/m^3$

$\rho_L = 885 kg/m^3$

$\mu_v = 1.67 \times 10^{-5} Pa.s$

$\mu_L = 9.11 \times 10^{-5} Pa.s$

$$X_{tt} = \left[\frac{1-0.53}{0.53}\right]^{0.9} \left[\frac{123.5}{885}\right]^{0.5} \left[\frac{9.11 \times 10^{-5}}{1.67 \times 10^{-5}}\right]^{0.1}$$

$X_{tt} = 1.83$

After determine the value of (X_{tt}), the value of (F)can be calculated according to equation (5.45):

$$F = 1 + 1.925(1.83)^{-0.83}$$
$$F = 2.166$$

And the value of ($h_{f_{in}}$) can be calculated according to equation(5.44):

$$h_{f_{in}} = 2.166 \times 5677.5 = 12297.5\,W / m^2.°C$$

After calculating the values of convection heat transfer coefficient inside tubes of evaporator ($h_{f_{in}}$)and convection coefficient heat transfer of air outside tubes of evaporator($h_{f_{air}}$), the value of overall heat transfer coefficient (U) can be calculated by given equation [reference 6]:

$$U = \frac{1}{\frac{\beta}{h_{f_{in}}} + \frac{d_i \ln(d_o/d_i)}{2k_{metal}} + \frac{d_i}{d_o} \times \frac{1}{h_{f_{air}}}} \qquad (5.47)$$

Where:

$k_{k_{metal}}$:Thermal conductivity of tube (copper)(W/m.°C).

$h_{f_{in}}$: Convection heat transfer coefficient for refrigerant(CO_2) inside the tube (W/m^2.°C).

$h_{f_{air}}$:Convection heat transfer coefficient for air outside the tube (W/m^2.°C).

d_i : The internal diameter of the tube (m).

d_o: The outer diameter of the tube (m).

β: fining coefficient.

The fining coefficient (β) can be calculated by the following equation [reference 6]:

$$\beta = \frac{A_{\text{finned element}}}{A_{\text{inner}}} \tag{5.48}$$

$$\beta = \frac{2.05 \times 10^{-3}}{9.42 \times 10^{-6}} \simeq 21$$

So

$$U = \frac{1}{\frac{21}{12297.5} + \frac{0.01\ \ln(0.012/0.01)}{2 \times 401} + \frac{0.01}{0.012} \times \frac{1}{134.7}}$$

$$U = 109.5\ \text{W/m}^2.°\text{C}$$

Calculation of overall heat transfer area (A_{oa}) can be done by using the following equation:

$$A_{oa} = n_1 n_2 n A_{\text{finned element}} \tag{5.49}$$

Where:

n_1 : Number of rows of evaporator.

n_2:Number of columns of evaporator.

n: Number of finned elements.

$A_{\text{finned element}}$: Finned element aera (m^2).

$n_1 = 8$ rows.

$n_2 = 5$ columns.

$n = 75$ element.

$$A_{oa} = 8 \times 5 \times 75 \times 2.05 \times 10^{-3}$$

$$A_{oa} = 6.15 m^2$$

According to previous calculations the heat capacity for this evaporator (Q_e) can be calculated be the following equation:

$$Q_e = UA\,\Delta t \tag{5.50}$$

Where:

A: surface area of heat transfer (m^2).

U: The overall heat transfer coefficient ($W/m^2.°C$).

Δt: Temperature difference (°C).

All the parameters in previous were calculated except (Δt),and it is given by:

$$\Delta t = t_a - t_r \tag{5.51}$$

t_a : Average temperature of air entering and leaving evaporator (°C).

t_r : Average temperature of refrigerant entering and leaving evaporator (°C).

And they can be calculated by the following equations :

$$t_a = \frac{t_{air_{in}} + t_{air_{out}}}{2} \tag{5.52}$$

$$t_a = \frac{25+22}{2} = 23.5°C$$

$$t_r = \frac{t_{ref_{in}} + t_{ref_{out}}}{2} \tag{5.53}$$

$$t_r = \frac{7+7}{2} = 7°C$$

And now according to equation (5.51) the value of (Δt) canbe calculated :

$$\Delta t = 23.5 - 7$$

$$\Delta t = 16.5°C$$

And now calculating the value of (Q_e) according to equation (5.50):

$$Q_e = 109.5 \times 6.154 \times 16.5$$

$$Q_e = 11.11kW$$

Since the actual heat capacity of evaporator ($Q_{e\ actual}$) in this project determined in chapter three section 3.3 and it value was $Q_{e\ actual} = 7.56kW$, so the required area ($A_{req.}$)for this value of $Q_{e\ actual}$ can be calculated by the following [reference 6]:

$$\frac{Q_{e\ actual}}{Q_e} \times A_{oa} = A_{req.} \tag{5.54}$$

$$\frac{7560}{11110} \times 6.15 = A_{req.}$$

$$A_{req.} = 0.68 \times 6.15$$

$$A_{req.} = 4.18m^2$$

Since 3000 finned elements gave $Q_e = 11.11kW$ and the actual heat capacity of evaporator ($Q_{e\ actual}$) is less than that value. So the required number of finned element will be [reference 6]:

$$n_{req.} = \frac{n_{all\ finned\ element} \times Q_{e\ actual}}{Q_e} \tag{5.55}$$

$$n_{req.} = \frac{3000 \times 7560}{11110}$$

$$n_{req.} = 2041 \text{finned element}$$

In each row in the evaporator there is M=600 finned element , so the number of rows is given by the following equation [reference 6]:

$$N_{row} = \frac{n_{req.}}{M} \tag{5.56}$$

$$N_{row} = \frac{2041}{600}$$

$$N_{row} = 3.56 \text{row}$$

So the maximum value of rows number (N_{row}) is 4 row .So that to have an extra area for transfer as for as safety factor .

And the figure 5.10 shows the construction of evaporator.

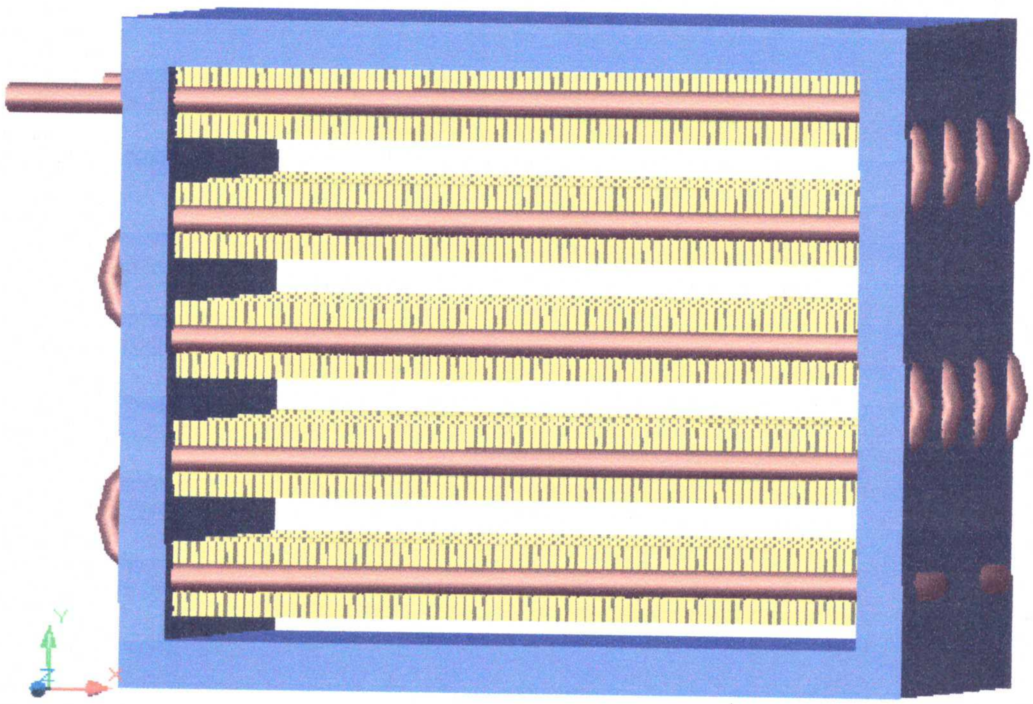


Figure 5.11 construction of evaporator

CHAPTER SIX

ELECTRICAL AND CONTROL DESIGN

Chapter Six

Electrical and Control Design

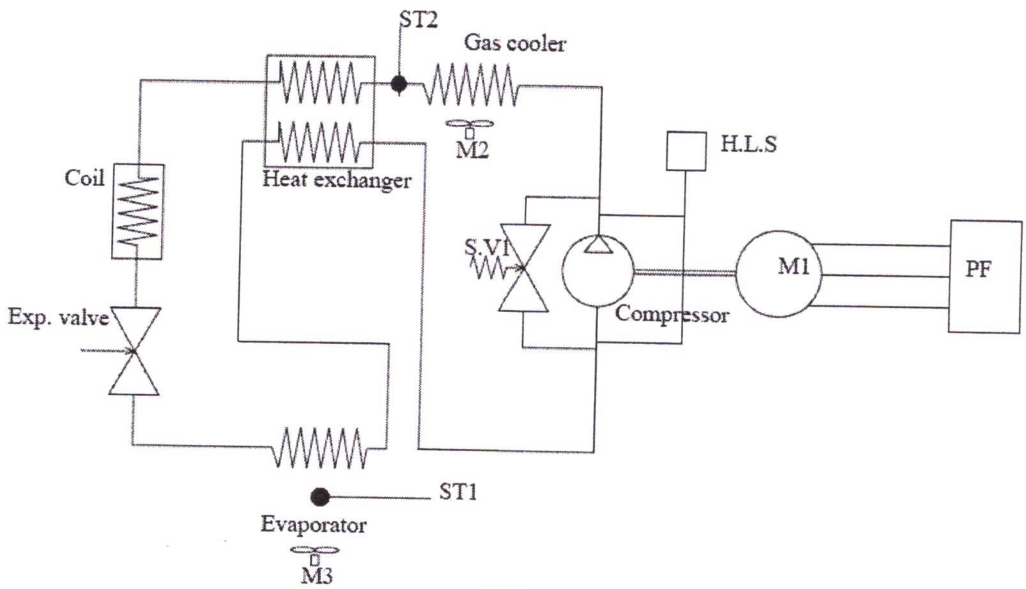
6.1 Introduction

In chapter six the electrical and control design will be performed for this project, and the electrical design talks about power circuit and the control design talks about the control circuit and programmable logic control (PLC) design.

6.2 Electrical Design

In order to obtain the power circuit it must form an explanatory diagram for the project most formed , and figure (6.1) contains an schematic drawing for the project.

And the schematic drawing shows that the project contains one electric motor of three phase (3ph) type , two electrical motor of single phase (1ph) type , two temperature sensors (thermostat) ,phase filer ,high-low pressure switch ,solenoid valve, and overload .



M1:Compressor motor
M2:Fan motor
M3:Blower motor
ST1:R/S temp. sensor
ST2:Cond. temp. sensor
H.L.S:High-Low pressur switch (NC)

PF:Phase filer (NC)
S1: start the cycle (NO)
S0: Emergency (NC)
OL: Over load (NC)
S.V1: Solenoid valve

Figure 6.1 schematic drawing for the project

The work of (3ph) motor(M1) is to drive the compressor through shaft connecting between them, and the function of (1 ph) motor is one to drive the fan of gas cooler (M2)and the other is to drive the blower of evaporator (M3) , and the work of temperature sensors (thermostat) is to observe the temperature change around the evaporator (ST1) and the other is for refrigerant temperature at the outlet of the gas cooler (ST2) , phase failure (PF) its work is to protect the power circuit specially the (3ph) when any error occurs in the phases arrangement or when losing one phase of the three phases ,and the work of high-low pressure switch (H.L.S) is to check the pressure difference between the inlet and the outlet of the compressor at beginning of running the compressor and when there is no pressure difference the (H.L.S) prevents the compressor from working by stopping the (3ph) motor , solenoid valve (S.V1) its work to lower the starting load on the compressor , overload(OL) its work to protect the (3ph) motor from the increasing of the load, which causing damaging in motor winding.

The figure(6.2) shows the power circuit diagram for this project according to the components which appear in figure (6.1).

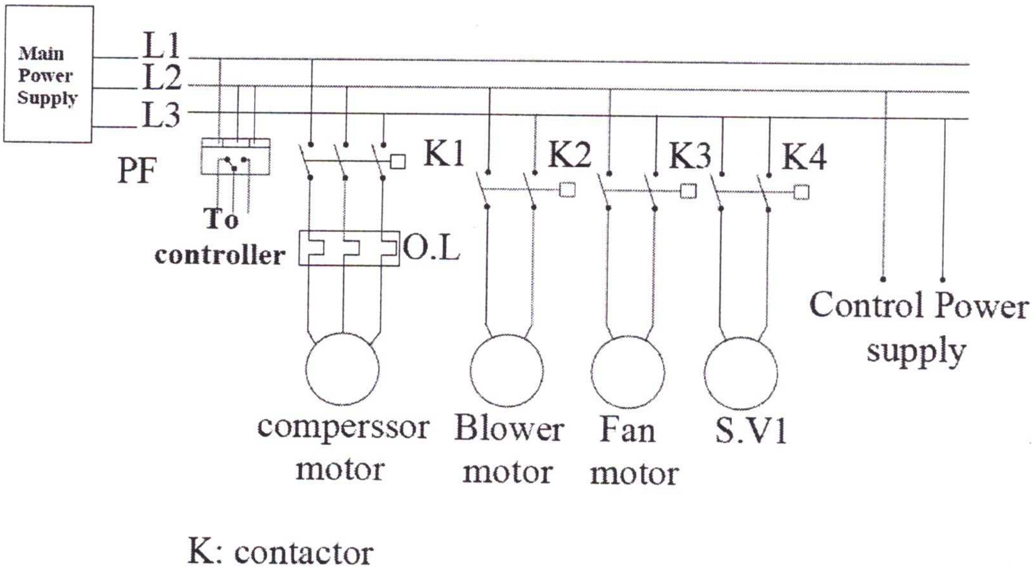


Figure 6.2 power circuit diagram

As shown in figure (6.2) the main power supply for this project is (3ph) supply , the (3ph) supply is used to operate the compressor motor (M1) ,and power circuit contain four contactors (K) where (K1) operates the (3ph) motor which is compressor motor (M1) ,(K2) is to operate the blower motor (M3), (K3) is to operate fan motor (M2), and (K4) is to operate the solenoid valve (S.V1).

The phase failure (PF) checks the connection the lines of (3ph),and if there is any problem, (PF) stops the power circuit by cutting off the power from main supply, then sends a signal to the controller (PLC), controller power supply is from power circuit .

This project works in determined sequence , and this sequence needs control to be executed in right way, so in next section this sequence will be designed .

6.3 Control Design

6.3.1 Control circuit Design

In order to turn-on the project it needs to press starting switch (S1), after pressing (S1) the compressor motor works if the following conditions are tested :

1. PF state must be normally closed (NC),that mean there is no problems in (3ph) lines connection .

2. High-Low pressure switch (H.L.S) state must be normally closed (NC), that means there is pressure difference between the inlet and the outlet of the compressor ,when their is no pressure difference the (H.L.S) prevent the compressor from working by stopping the (3ph) motor.

3. temperature sensor (ST1) state must be normally closed (NC), that mean the temperature around the evaporator (inside car cabin) is higher than adjusted

temperature (designed temperature 22 ^{o}C),if the temperature inside car cabin equal to designed temperature then (ST1) became normally opened (NO).

After testing these conditions the compressor motor (M1)starts running through (K1) and immediately the blower motor (M3) run through (K2) ,and solenoid valve (S.V1) is open the refrigerant pass through (S.V1) coming from compressor outlet to the compressor ,this happens during a specific period of time through a timer type off-delay (T1) which begins counting down until the end of this period (T1) become (NO) so the (S.V1) is closed .and immediately a signal lamp work to point out that (M1) is working and running lamp happen by (K1).

During the work of the cycle if the refrigerant temperature at the outlet of the gas cooler is vary high the temperature sensor(ST2) becomes (NC),so the gas cooler fan(M2) through (K3) until the refrigerant temperature is reduce at outlet of gas cooler ,then (ST2) becomes (NO) so (M2) is stopped.

If any emergency occurs then all the cycle will be stopped by pressing on the emergency switch (S0).

When an error occurs in (PF) or (H.L.S) the compressor motor (M1)will stopped and the signal lamp will be turned-on to point out where the error is in (PF) or (H.L.S).

Based on the previous sequence of operating and protecting the project equipments . It is conclusion figure (6.3) which is shows the control circuit diagram.

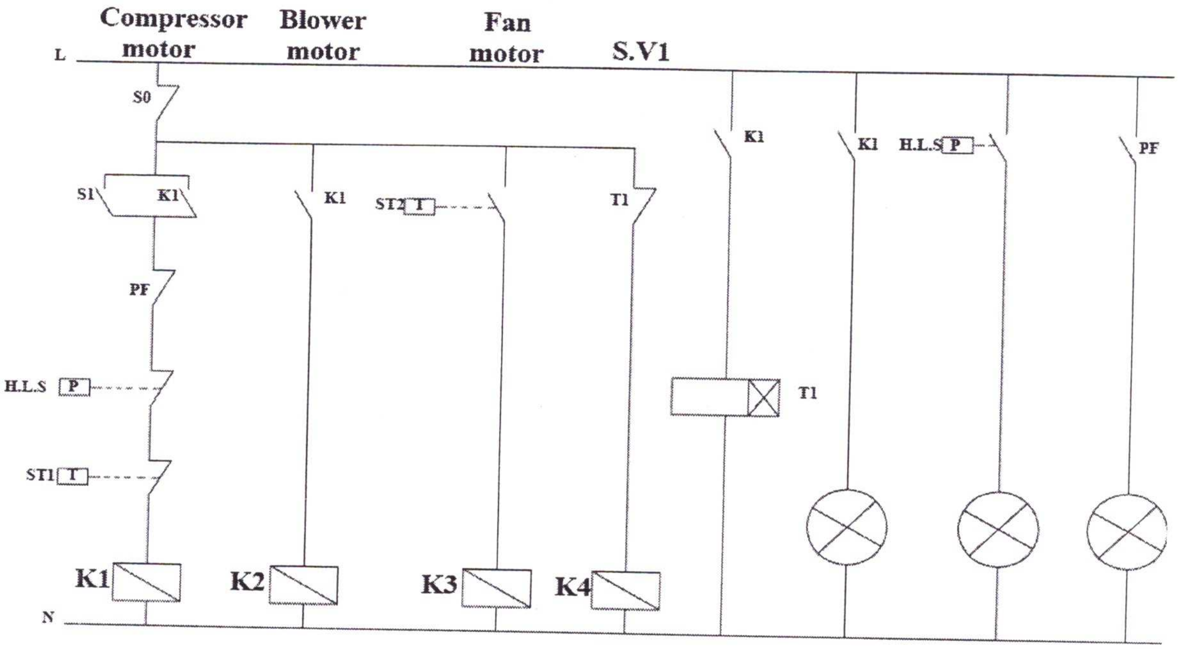


Figure 6.3 control circuit diagram

6.3.2 Programmable Logic Control (PLC) Design

From the sequence was explained in previous section the logically operating of the project is obtained, and according to this the programmable logic control (PLC) will be designed, and figure (6.4) shows (PLC) block diagram for this project.

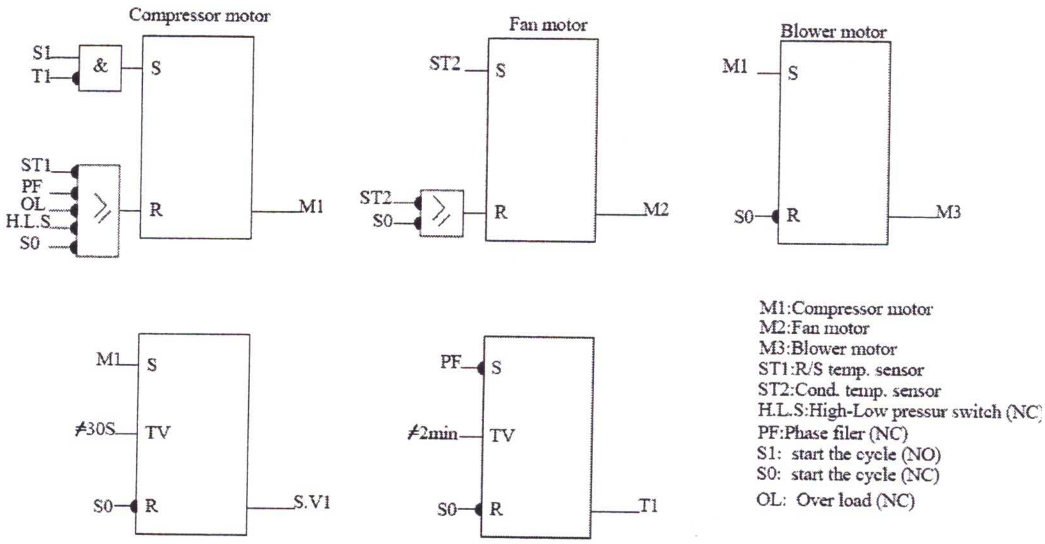


Figure 6.4 (PLC) block diagram

6.4 Electrical Motor Calculations

In normal case the compressor in automotive air conditioning system driven by the motor of the car, by connecting it with the shaft of the motor. In this case the automotive motor replaced by (3ph) electrical motor(M1), and in order to select it there are main parameters must be calculated such as power (P), run current (I), torque(T) ,and speed(n).

The car engine speed is between 1000-2500 rpm and in this project will be assumed as n=1500 rpm.

Net mechanical power that the car engine must be delivers to transmission belt was calculated in chapter three section (3.3.1) and its value is $P = 4.32\ \text{kW} = = 5.8\ \text{hp}$, this the output of (M1)

Running current (I) of (M1) can be calculated by the following equation:

$$P = \sqrt{3}\,VI \tag{6.1}$$

Where:

V : Voltage (V).

I : Current (A).

$4320 = \sqrt{3} \times 380 \times I$

$I = \dfrac{4320}{\sqrt{3} \times 380} = 6.65\text{A}$

To determine (T) of (M1) the following equation is used :

$$P = T\omega \tag{6.2}$$

Where:

T: Torque (N.m).

ω: Angular velocity (rad/s).

And in order to calculate the value of (ω)the following equation is used:

$$\omega = \frac{2\pi}{60} n \tag{6.3}$$

$\omega = \dfrac{2\pi}{60} \times 1500 = 157.079\text{rad/s}$

So the value of (T) is calculated according to equation(6.2):

$4320 = T \times 157.079$

$T = \dfrac{4320}{157.079} = 27.5\ \text{N.m}$

CHAPTER SEVEN

ACCESSORY SELECTION

Chapter Seven

Accessory Selection

7.1 Fan selection:

In order to select the right fan for the application the volumetric air flow rate must be determined and it can be calculated according to the following equation [reference 14]:

$$\dot{V} = \dot{m}\, v_{air} \qquad (7.1)$$

Where:

$\dot{m}$: Mass flow rate of air (kg/s).

v_{air} : Specific volume of air (m^3/kg).

and the specific volume can be calculated by the following equation:

$$v_{air} = \frac{1}{\rho_{air}} \qquad (7.2)$$

ρ_{air} : Density for air at inlet of gas cooler (kg/s).

$$\rho_{air_{@t=40^{o}C\&P=1\ atm.}} = 1.1774 kg/s$$

$$v_{air} = \frac{1}{1.1774}$$

$$v_{air} = 0.849 m^3/kg$$

The value of mass flow rate can be calculated be the following equation :

$$Q_{\substack{gas\\cooler}} = c_{p_{air}} \dot{m}(t_{in} - t_{air}) \qquad (7.3)$$

Where:

$Q_{\substack{gas\\cooler}}$: Heat transfered from gas cooler (W).

$c_{p_{air}}$: Specific heat for air (kJ/kg.°C).

t_{air} : Air temperatur at inlet gas cooler (°C).

t_{in} : temperature inisde gas cooler(°C).

$c_{p_{air}} = 1.006$kJ/kg.°C(see appendix A, table A-5)

$t_{air} = 40°C$

$t_{in} = 167°C$

The value of ($Q_{\substack{gas\\cooler}}$) determined in chapter three section 3.3 and it was

$Q_{\substack{gas\\cooler}} = 9576W$, and the value of $\dot{m}$ will be:

$$9576 = 1006 \times \dot{m} \times (167 - 40)$$

$$\dot{m} = \frac{9576}{1006 \times 127}$$

$$\dot{m} = 0.075 kg/s$$

And according to equation (7.1) the value of volumetric flow rate will be:

$$\dot{V} = 0.075 \times 0.849$$

$$\dot{V} = 0.06 m^3/s$$

To convert the value of ($\dot{V}$) from m^3/s unit to cfm unit it will be multiplied by convention factor (Z) according to following equation:

$$\dot{V}_{cfm} = Z\dot{V} \qquad (7.4)$$

$$\dot{V}_{cfm} = 2119 \times 0.06$$

$$\dot{V}_{cfm} = 127 cfm$$

After determining the value of volumetric air flow rate the pressure loss(ΔP) must be determined and it can be determined in two methods by calculations or by charts , here it will be determined by chart ,and figure 7.1 shows $\dot{V}$ vs. ΔP chart that was used to determined the value of ΔP [reference 37]:

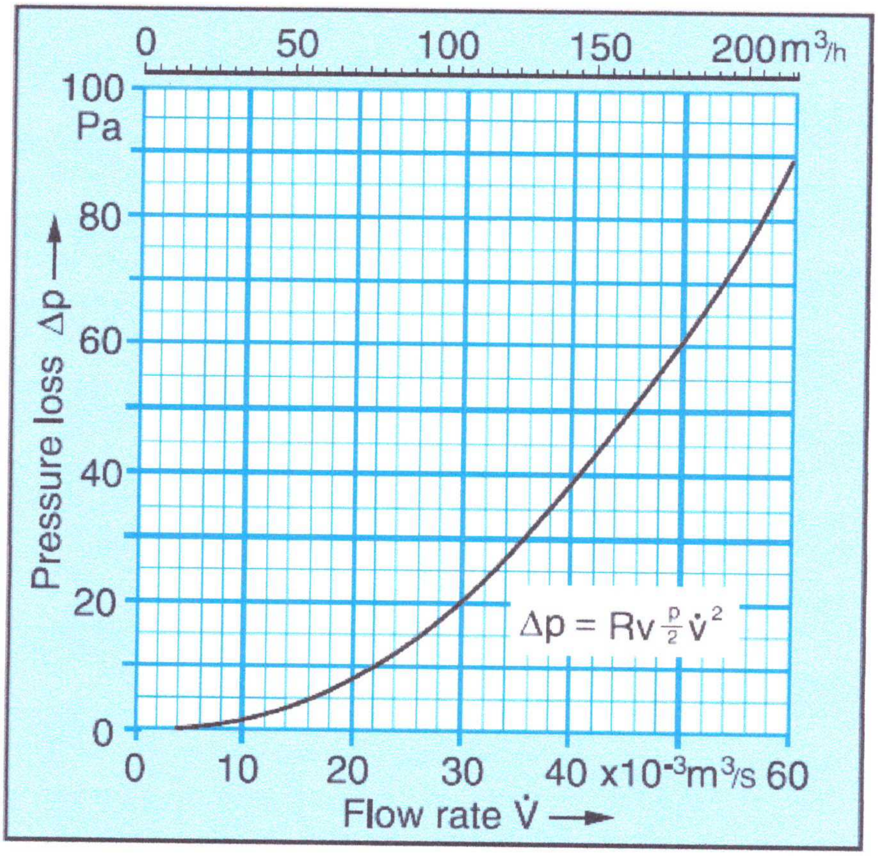


Figure 7.1 $\dot{V}$ vs. ΔP chart

and from chart in previous figure the value of ΔP can be determined and its value ΔP =90 Pa. and according to previous calculations sand determination the fan that has the following data:

$\dot{V} = 0.06m^3 / s = 127cfm$
$\Delta P = 90Pa$

And figure 7.2 shows the chosen fan which is axial fan and its model is AMR motorized axial fan produced by (CFM) Continental Fan Co. ,and it will be used in this project.

Figure 7.2 AMR motorized axial fan

7.2 Blower selection:

As in fan selection the value of volumetric air flow ($\dot{V}$) must be determined, and pressure loss(ΔP) for the blower ,the calculations were done in chapter five section 5.4 and it value was $\dot{V} = 0.5244 m^3/s = 1112 cfm$.

Evaporator load Qe = 7.538 kW = 2.154 ton. (see appendix B, table B-4).

Figure 7.3 shows the pressure loss of the blower, at various air flow rate.

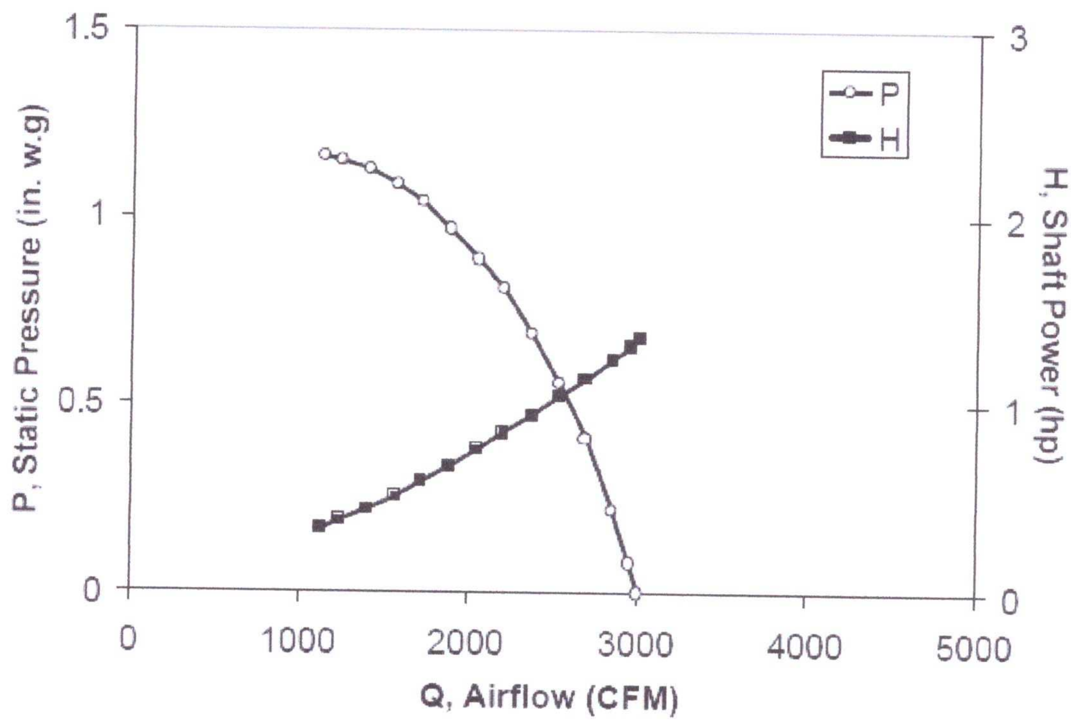


Figure 7.3 pressure loss of the blower, at various air flow rate

$$\text{Pressure loss} = 0.3\,\text{w.g (water gauge)}$$

$$101.325 \times 10^3\ \text{Pa} = 407.2\,\text{w.g}$$

$$\text{Pressure loss} = 74.7\,\text{Pa}$$

7.3 Accumulator Selection:

The accumulator location in this project is before the compressor inlet (in suction line), the accumulator selection depends on two main parameters and they are:

- The operation pressure , which is in this project 42bar
- The refrigerant which the accumulator can deal with it ,and in this project the refrigerant is R744(CO_2).

and the accumulator model which will be used in this project is CBRIST, and figure 7.4 shows the R744 accumulator, which is produced by Burkon Co. in Germany.

Figure 7.4 R744 accumulator

Basic specification of accumulator as the following:

- Accumulator Volume:335 ccm and 516 ccm (Sizes in between on request)
- Diameter: 64mm

- Height: 195mm - 260mm (Including burst plate)
- Weight: 550g - 670g (Including burst plate)
- Mounting Angle: +/- 15°
- Operating pressure: Max 132bar at 90°C
- Burst Pressure: 264bar
- Fitting Technology: VisCO2nnect Double Block Fitting
- Test pressure: 198bar
- Material: Aluminum.
- Burst Plate: 120bar +/-10bar at 90°C.

And its main features are:

- Vapor quality management for high system COP.
- Internal oil management for high system reliability.
- Lowest weight and packaging.
- Welding-free design.
- Option to use internal dryer material (40g dryer x H7l).
- Available with external insulation cover.
- Validation testing completed.
- Option active release device (instead of burst plate).

7.4 Solenoid Valve Selection:

The solenoid valve location is between inlet and outlet of compressor , solenoid valve operation conditions in this project are listed in table 7.1.

Table 7.1 operation conditions for solenoid valve.

Condition	Value
Inlet pressure	42 bar
Outlet pressure	80 bar
Inlet temperature	7
Outlet temperature	167
Mass flow rate	0.063 kg/s
Type of control	Automatically control.

Figure 7.5 shows a schematic diagram for solenoid valve.

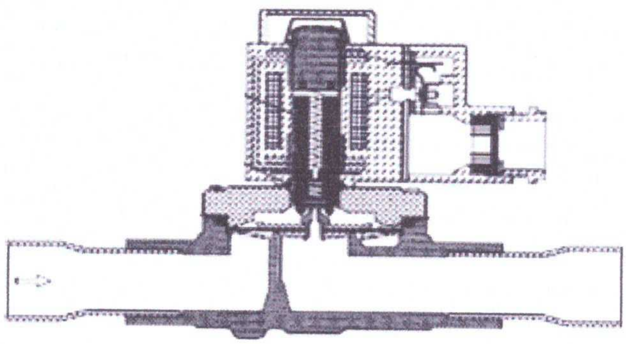


Figure 7.5 schematic diagram of solenoid valve

Since the equipments that used in transcritical refrigeration system using R744 as refrigeration fluid are under development so it was too difficult to find them in internet website, or in the market, so an emails has been sent to many companies which working in this subject to ask about their operation conditions for solenoid valve, and what is the name of their product, until now there is no reply on this emails.

7.5 Expansion valve selection :

To choose the perfect expansion valve for any application many thighs are kept in mind such as:

- Refrigerant
- Evaporator pressure (low pressure side)
- condensing pressure (high pressure side)

Table 7.2 contains the operation condition for expansion valve.

Table 7.2 operation conditions for expansion valve.

Condition	Value
Inlet pressure	42 bar
Outlet pressure	80 bar
Inlet temperature	7 °C
Outlet temperature	167 °C
Mass flow rate	0.063 kg/s
Type of control	Automatically control.

As was mentioned before about transcritical refrigeration system using R744 equipments the figure 7.6 shows an schematic diagram for expansion valve ,which is electrical operated expansion valve.

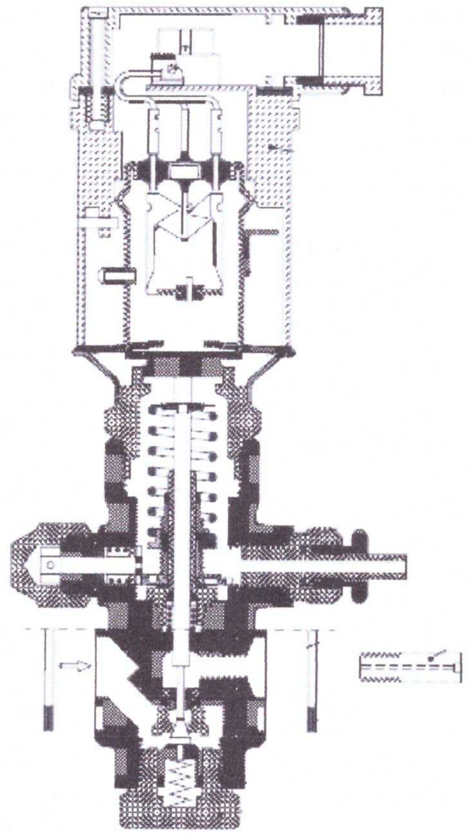


Figure 7.6 schematic diagram for electrical operated expansion valve.

7.6 High –Low pressure switch (H.L.S) selection

The high-low pressure switch location is between the inlet of compressor and the outlet of the compressor ,and the main characteristics that were used to determined the (H.L.S) are the following:

- inlet pressure of compressor
- outlet pressure of compressor
- the pressure deferent between inlet and outlet.

By searching on internet it was found that in Danfoss Co. the max working pressure is 20 bar and this value is less than the inlet pressure 42 bar, so the operation conditions for (H.L.S) in this project are list in table 7.3.

Table 7.3 the operation conditions for (H.L.S)

Condition	Value
Inlet pressure	42 bar
Outlet pressure	80 bar
Inlet temperature	7 °C
Outlet temperature	167 °C
pressure deferent	38 bar

And the figure 7.7 shows (H.L.S) that use in usual refrigeration systems.

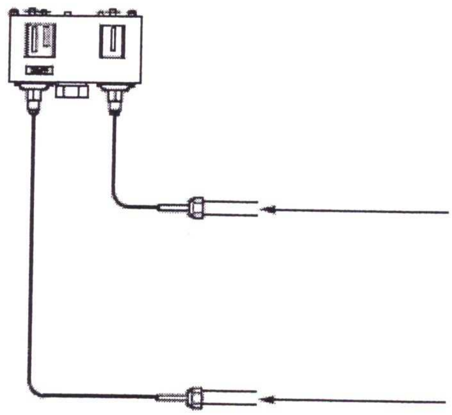


Figure 7.7 (H.L.S)

7.7 Temperature Sensor (Thermostat) selection

In this project there are two temperature sensors and the locations are as the following:

1. Between the evaporator and its blower to measure the temperature of returns air from the car cabin and then determine to stop the cycle or to start according to adjust temperature .

2. At the outlet of gas cooler to measure the temperature of refrigerant that leaves gas cooler and determine to stop fan of gas cooler or to start it .

And in this project the thermostat model KP will be used and its specifications are as the following:

1. Ambient temperature : -40 °C to +65°C (+80 °C for max. 2 hours).

2. Switch: single-pole, double- throw changeover switch.

3. Contact load , AC:
   AC1: 16A,400V.
   AC3: 16A, 400 V.
   AC15: 16A,400V.

4. Max. starting current (L.R):112A,400V.

5. Contact load, DC:
   DC13:12W,220V control current.

6. Cable connection : cable entry for cables 6-14mm diameter.

Figure 7.8 shows KP thermostat.

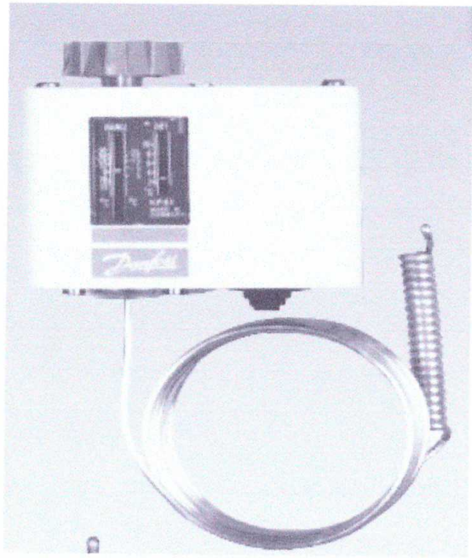


Figure 7.8 KP thermostat.

CHAPTER EIGHT

RECOMMENDATIONS & CONCLUSIONS

Chapter Eight

8.1 Recommendations

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

1. The project has been designed theoretically and all parts have been selected or operating conditions have been specified so it is recommend to continue the project.

2. The suggested project has a useful applications and the procedures will be useful for students and engineers.

3. The compressor used with the suggested refrigerant (R744) is at development stage and it is highly recommended to follow and observe its development and implementation and using the compressor when its available to construct the project practically.

4. The future of most refrigeration and air conditioning applications are depend on the suggested refrigerant (R744), so it is recommend to observing the refrigerant development.

5. the final recommendation is directed to Mechanical Engineering Department Council to founding a special computer lab for internet researching in

Mechanical Engineering Department, and this lab must be connected with data base websites to use in graduation projects researches in the department.

8.2 Conclusions :

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

1. This project is friendly to environment, specially the suggested refrigerant (R744) comparing with usually used refrigerants.

2. The COP of cycle this project with the suggested refrigerant (R77) is higher than COP for the some cycle using any one of common refrigerants.

3. The size of project cycle parts is smaller than that use with common refrigerants cycles.

4. The power that used to operate the R744 compressor is less than the power that use to operate common compressor .

APPENDIX (A)

Table A-1 Thermal Conductivity of some materials.

Material	Thermal Conductivity (k) (W/m K)
Iron	79.5
Fiberglass	0.04
Rock wool	0.039
Air	0.025
Plastic (PVC)	0.147
Flax	0.039
Asphalt	36.91
Aluminum	200

Table A-2 Properties of air.

Properties of air in the gaseous state:

P bar	*T* K	ρ g/L	*H* J/g	*S* J/g K	C_p J/g K
1	100	3.556	98.3	5.759	1.032
1	200	1.746	199.7	6.463	1.007
1	300	1.161	300.3	6.871	1.007
1	500	0.696	503.4	7.389	1.030
1	1000	0.348	1046.6	8.138	1.141
10	200	17.835	195.2	5.766	1.049
10	300	11.643	298.3	6.204	1.021
10	500	6.944	502.9	6.727	1.034
10	1000	3.471	1047.2	7.477	1.142
100	200	213.950	148.8	4.949	1.650
100	300	116.945	279.9	5.486	1.158
100	500	66.934	499.0	6.048	1.073
100	1000	33.613	1052.4	6.812	1.151

Table A-3 R744 Properties at various pressure and temperature .

State (1)

No.	Property	Value	Unit
1.	Medium :	**carbon dioxide**	
2.	state of aggregation :	**fluid**	
3.	Pressure :	42	[bar]
4.	Temperature :	7	[Celsius]
5.	Density :	123.3	[kg / m^3]
6.	Specific Enthalpy :	427.514	[kJ / kg]
7.	Specific Entropy :	1.0604943649907	[kJ / kg K]
8.	Specific isobar heat capacity : cp	2.152	[kJ / kg K]
9.	Specific volume	0.00811	[m^3/ kg]
10.	Heat conductance	65.95	[10^{-3} (W / m * K)]
11.	Dynamic viscosity :	16.70307	[10^{-6} (Pa s)]
12.	Kinematic viscosity :	10.051195225326	[10^{-6} m^2 / s]
13.	Prandtl-Number :	1.183	

State (1')

No.	Property	Value	Unit
1.	Medium :	**carbon dioxide**	
2.	state of aggregation :	**gas**	
3.	Pressure :	42	[bar]
4.	Temperature :	12	[Celsius]
5.	Density :	111.1676592157	[kg / m^3]
6.	Specific Enthalpy :	440.15438134444	[kJ / kg]
7.	Specific Entropy :	1.8546293924132	[kJ / kg K]
8.	Specific isobar heat capacity : cp	1.7441355521267	[kJ / kg K]
9.	Specific volume	0.008995	[m^3/ kg]
10.	Heat conductance	1.2812108183644	[10^{-3} (W / m * K)]
11.	Dynamic viscosity :	14.343795657515	[10^{-6} (Pa s)]
12.	Kinematic viscosity :	-0.045812268887013	[10^{-6} m^2 / s]
13.	Prandtl-Number :	1.4071277951381	

State (2)

No.	Property	Value	Unit
1.	Medium :	**carbon dioxide**	
2.	state of aggregation :	**overcritical fluid**	
3.	Pressure :	42	[bar]
4.	Temperature :	110	[Celsius]
5.	Density :	64.2636	[kg / m^3]
6.	Specific Enthalpy :	557.556	[kJ / kg]
7.	Specific Entropy :	2.2096	[kJ / kg K]
8.	Specific isobar heat capacity : cp	1.0806	[kJ / kg K]
9.	Specific volume	0.01556	[m^3/ kg]
10.	Heat conductance	25.7524	[10^{-3} (W / m * K)]
11.	Dynamic viscosity :	19.4552	[10^{-6} (Pa s)]
12.	Kinematic viscosity :	0.307976	[10^{-6} m^2 / s]
13.	Prandtl-Number :	0.817272	

State (3)

No.	Property	Value	Unit
1.	Medium :	**carbon dioxide**	
2.	state of aggregation :	**gas**	
3.	Pressure :	80	[bar]
4.	Temperature :	167	[Celsius]
5.	Density :	105.26	[kg / m^3]
6.	Specific Enthalpy :	601.484	[kJ / kg]
7.	Specific Entropy :	2.20486	[kJ / kg K]
8.	Specific isobar heat capacity : cp	1.16472	[kJ / kg K]
9.	Specific volume	0.0095	[m^3/ kg]
10.	Heat conductance	32.1922	[10^{-3} (W / m * K)]
11.	Dynamic viscosity :	22.679	[10^{-6} (Pa s)]
12.	Kinematic viscosity :	0.213356	[10^{-6} m^2 / s]
13.	Prandtl-Number :	0.821286	

State (4)

No.	Property	Value	Unit
1.	Medium :	**carbon dioxide**	
2.	state of aggregation :	**gas**	
3.	Pressure :	80	[bar]
4.	Temperature :	57	[Celsius]
5.	Density :	199.88	[kg / m^3]
6.	Specific Enthalpy :	451.56	[kJ / kg]
7.	Specific Entropy :	1.8092	[kJ / kg K]
8.	Specific isobar heat capacity : cp	2.1062	[kJ / kg K]
9.	Specific volume	0.005003	[m^3/ kg]
10.	Heat conductance	31.373	[10^{-3} (W / m * K)]
11.	Dynamic viscosity :	20.085	[10^{-6} (Pa s)]
12.	Kinematic viscosity :	0.100785	[10^{-6} m^2 / s]
13.	Prandtl-Number :	1.3448	

State (5)

No.	Property	Value	Unit
1.	Medium :	**carbon dioxide**	
2.	state of aggregation :	**gas**	
3.	Pressure :	80	[bar]
4.	Temperature :	47	[Celsius]
5.	Density :	236.81	[kg / m^3]
6.	Specific Enthalpy :	426.25	[kJ / kg]
7.	Specific Entropy :	1.7315	[kJ / kg K]
8.	Specific isobar heat capacity : cp	3.2443	[kJ / kg K]
9.	Specific volume	0.00422	[m^3/ kg]
10.	Heat conductance	36.109	[10^{-3} (W / m * K)]
11.	Dynamic viscosity :	20.97	[10^{-6} (Pa s)]
12.	Kinematic viscosity :	0.089204	[10^{-6} m^2 / s]
13.	Prandtl-Number :	1.8536	

State (6)

No.	Property	Value	Unit
1.	Medium :	**carbon dioxide**	
2.	state of aggregation :	**fluid**	
3.	Pressure :	80	[bar]
4.	Temperature :	30	[Celsius]
5.	Density :	701.7	[kg / m^3]
6.	Specific Enthalpy :	284.1	[kJ / kg]
7.	Specific Entropy :	1.272	[kJ / kg K]
8.	Specific isobar heat capacity : cp	5.229	[kJ / kg K]
9.	Specific volume	0.001425	[m^3/ kg]
10.	Heat conductance	80.14	[10^{-3} (W / m * K)]
11.	Dynamic viscosity :	55.98	[10^{-6} (Pa s)]
12.	Kinematic viscosity :	0.07979	[10^{-6} m^2 / s]
13.	Prandtl-Number :	3.652	

State (7)

No.	Property	Value	Unit
1.	Medium :	**carbon dioxide**	
2.	state of aggregation :	**fluid**	
3.	Pressure :	42	[bar]
4.	Temperature :	7	[Celsius]
5.	Density :	238.095	[kg / m^3]
6.	Specific Enthalpy :	217.514	[kJ / kg]
7.	Specific Entropy :	1.0604943649907	[kJ / kg K]
8.	Specific volume	0.00422	[m^3/ kg]
9.	Heat conductance	509.64013780261	[10^{-3} (W / m * K)]
10.	Dynamic viscosity :	102.1446122905	[10^{-6} (Pa s)]
11.	Kinematic viscosity :	10.051195225326	[10^{-6} m^2 / s]
12.	Thermal diffusivity :	0.88851606331471	[10^{-7} m^2 / s]
13.	Prandtl-Number :	0.33509035381751	

No.	Property	Value	Unit
1.	Medium :	**carbon dioxide**	
2.	state of aggregation :	**gas**	
3.	Pressure :	**42**	[bar]
4.	Temperature :	**20.25**	[Celsius]
5.	Density :	105.8744	[kg / m^3]
6.	Specific Enthalpy :	449.257	[kJ / kg]
7.	Specific Entropy :	1.88626	[kJ / kg K]
8.	Specific isobar heat capacity : cp	1.63628	[kJ / kg K]
9.	Specific volume	0.00945	[m^3/ kg]
10.	Heat conductance	21.8949	[10^{-3} (W / m * K)]
11.	Dynamic viscosity :	15.9095	[10^{-6} (Pa s)]
12.	Kinematic viscosity :	0.1531195	[10^{-6} m^2 / s]
13.	Prandtl-Number :	1.18348	

No.	Property	Value	Unit
1.	Medium :	**carbon dioxide**	
2.	state of aggregation :	**gas**	
3.	Pressure :	**80**	[bar]
4.	Temperature :	**41.5**	[Celsius]
5.	Density :	269.095	[kg / m^3]
6.	Specific Enthalpy :	407.825	[kJ / kg]
7.	Specific Entropy :	1.67375	[kJ / kg K]
8.	Specific isobar heat capacity : cp	4.58135	[kJ / kg K]
9.	Specific volume	0.00372	[m^3/ kg]
10.	Heat conductance	41.1855	[10^{-3} (W / m * K)]
11.	Dynamic viscosity :	22.015	[10^{-6} (Pa s)]
12.	Kinematic viscosity :	0.082153	[10^{-6} m^2 / s]
13.	Prandtl-Number :	2.4322	

Table A-4 Recommended Velocities

Line	Refrigerant		Recommended Velocity (m/s)
Suction	R12	R22	8-12
	CO_2		10-20
Discharge	R12	R22	10-18
	CO_2		12-25
Liquid Between Condenser and Receiver	R12	R22	1-1.25
	CO_2		0.5-0.7
Discharge and Suction with Pump in System	R12	R22	0.3-0.5
	CO_2		0.6-1.2

Table A-5 Liquid CO_2 properties

No.	Property	Value	Unit
1.	Medium :	**carbon dioxide**	
2.	state of aggregation :	**Liquid**	
3.	Pressure :	42	[bar]
4.	Temperature :	7	[Celsius]
5.	Density :	885	[kg / m^3]
6.	Specific Enthalpy :	217.481	[kJ / kg]
7.	Specific Entropy :	1.0604	[kJ / kg K]
8.	Specific isobar heat capacity : cp	2.6047	[kJ / kg K]
9.	Specific volume	0.0013	[m^3/ kg]
10.	Heat conductance	0.101938	[(W / m * K)]
11.	Dynamic viscosity :	9.11	[10^{-5}(Pa s)]
12.	Kinematic viscosity :	10.294	[10^{-6} m^2 / s]

APPENDIX (B)

Table B-1 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:**		
				1/4-inch to 8-inch	20 ft	20 ft
				10-inch	18 ft	18 ft
				12-inch	12 ft	12 ft
				COILS:		
				1/4-inch to 1-inch	—	60 ft
					—	100 ft
				1 1/4 inch and 1 1/2-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:**		
				1/4-inch to 10-inch	20 ft	20 ft
				12-inch	18 ft	18 ft
				COILS:		
				1/4-inch to 1-inch	—	60 ft
					—	100 ft
				1 1/4 inch and 1 1/2-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:**		
				1/4-inch to 12-inch	20 ft	N/A
DWV	Yellow	ASTM B 306	Drain, Waste, Vent, HVAC, Solar	**STRAIGHT LENGTHS:**		
				1 1/4-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:**		
				3/8-inch to 4 1/8-inch	20 ft	[4]
				COILS:		
				1/8-inch to 1 5/8-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:**		
				1/4-inch to 8-inch	20 ft	N/A

Table B-2 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	.363	.196	.178	.305	.00252
	A	.750	.666	.042	.348	.196	.174	.362	.00242
	D	.750	.666	.042	.348	.196	.174	.362	.00242
7/8	A	.875	.785	.045	.484	.229	.206	.455	.00336
	D	.875	.785	.045	.484	.229	.206	.455	.00336
1 1/8	A	1.125	1.025	.050	.825	.294	.268	.655	.00573
	D	1.125	1.025	.050	.825	.294	.268	.655	.00573
1 3/8	A	1.375	1.265	.055	1.26	.360	.331	.884	.00875
	D	1.375	1.265	.055	1.26	.360	.331	.884	.00875
1 5/8	A	1.625	1.505	.060	1.78	.425	.394	1.14	.0124
	D	1.625	1.505	.060	1.78	.425	.394	1.14	.0124
2 1/8	D	2.125	1.985	.070	3.09	.556	.520	1.75	.0215
2 5/8	D	2.625	2.465	.080	4.77	.687	.645	2.48	.0331
3 1/8	D	3.125	2.945	.090	6.81	.818	.771	3.33	.0473
3 5/8	D	3.625	3.425	.100	9.21	.949	.897	4.29	.0640
4 1/8	D	4.125	3.905	.110	12.0	1.08	1.02	5.38	.0833

Table B-3 Rated Internal Working Pressure for Copper Tube: ACR.

Tube Size (OD), in	Annealed							Drawn**						
	COILS													
	S= 6000 psi 100 F	S= 5100 psi 150 F	S= 4900 psi 200 F	S= 4800 psi 250 F	S= 4700 psi 300 F	S= 4000 psi 350 F	S= 3000 psi 400 F	S= 10,300 psi 100 F	S= 10,300 psi 150 F	S= 10,300 psi 200 F	S= 10,300 psi 250 F	S= 10,000 psi 300 F	S= 9,700 psi 350 F	S= 9,400 psi 400 F
1/8	3074	2613	2510	2459	2408	2049	1537	—	—	—	—	—	—	—
3/16	1935	1645	1581	1548	1516	1290	968	—	—	—	—	—	—	—
1/4	1406	1195	1148	1125	1102	938	703	—	—	—	—	—	—	—
5/16	1197	1017	977	957	937	798	598	—	—	—	—	—	—	—
3/8	984	836	803	787	770	656	492	—	—	—	—	—	—	—
1/2	727	618	594	581	569	485	363	—	—	—	—	—	—	—
5/8	618	525	504	494	484	412	309	—	—	—	—	—	—	—
3/4	511	435	417	409	400	341	256	—	—	—	—	—	—	—
3/4	631	537	516	505	495	421	316	—	—	—	—	—	—	—
7/8	582	495	475	466	456	388	291	—	—	—	—	—	—	—
1 1/8	494	420	404	395	387	330	247	—	—	—	—	—	—	—
1 3/8	439	373	358	351	344	293	219	—	—	—	—	—	—	—
1 5/8	408	347	334	327	320	272	204	—	—	—	—	—	—	—
	STRAIGHT LENGTHS													
	S= 6000 psi 100 F	S= 5100 psi 150 F	S= 4900 psi 200 F	S= 4800 psi 250 F	S= 4700 psi 300 F	S= 4000 psi 350 F	S= 3000 psi 400 F	S= 10,300 psi 100 F	S= 10,300 psi 150 F	S= 10,300 psi 200 F	S= 10,300 psi 250 F	S= 10,000 psi 300 F	S= 9,700 psi 350 F	S= 9,400 psi 400 F
3/8	914	777	747	731	716	609	457	1569	1569	1569	1569	1524	1478	1432
1/2	781	664	638	625	612	521	391	1341	1341	1341	1341	1302	1263	1224
5/8	723	615	591	579	567	482	362	1242	1242	1242	1242	1206	1169	1133
3/4	633	538	517	506	496	422	316	1086	1086	1086	1086	1055	1023	991
7/8	583	496	477	467	457	389	292	1002	1002	1002	1002	972	943	914
1 1/8	495	421	404	396	388	330	248	850	850	850	850	825	801	776
1 3/8	440	374	359	352	344	293	220	755	755	755	755	733	711	689
1 5/8	409	348	334	327	320	273	205	702	702	702	702	682	661	641
2 1/8	364	309	297	291	285	243	182	625	625	625	625	607	589	570
2 5/8	336	286	275	269	263	224	168	577	577	577	577	560	544	527
3 1/8	317	270	259	254	249	212	159	545	545	545	545	529	513	497
3 5/8	304	258	248	243	238	203	152	522	522	522	522	506	491	476
4 1/8	293	249	240	235	230	196	147	504	504	504	504	489	474	460

NOT MANUFACTURED

Table B-4 Nominal size according to air flow capacity.

Airflow Capacity	**Blower Size**
800 cfm (2 ton)	9 X 8
1200 cfm (3 ton)	10 X 8
1600 cfm (4 ton)	10 X 10
2000 cfm (5 ton)	11 X 10

REFERENCES

BOOKS

1- ASHRAE Handbook 1993, Fundamentals, SI edition

2- ASHRAE Handbook, HVAC Systems and Equipment, 1992.

3- Andrey Rozhentsev, Chi-Chuan Wang, Some design features of CO2 air conditioner, Energy and resources laboratories, Ghung Hsing road, Chutung, Hsinchu 310, Taiwan, 2000.

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

5- Chang, H., 2002, Development of a Supercritical Carbon Dioxide Brayton Cycle: Improving PBR Efficiency and Testing Material Compatibility, Idaho National Engineering and Environmental Laboratory (INEEL)

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

7- Coulson & Richardson's, Chemical Engineering Design, Forth Edition, Volume 6, ELSEVIR.

8- Ferdinand P. Beer, E. Russell Jonnston, JR., Mechnics of Materials, Second Edition (SI), McGRAW-HILL BOOK COMPANY.

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

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

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

12- Lwbedov V.F, and Others, Refrigeration Technology, Agroigdat, Moscow, 1986.

13- Mike Stubblefield, John H Haynes, The Haynes Repair Manual for automotive heating and air conditioning systems, Sparkford Nr Yeovil Somerset BA22 7JJ England, 2000.

14- Mohammed A. Alsaad, Heating and Air Conditioning, Third edition, University of Jordan, Amman, Jordan, 1995.

15- Ramesh K. Shah, and Dusan P. Sekalic, Fundamentals of Heat Exchanger Design, Jonn Wiley & Sons, Inc.

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

17- Yunis A. Gengel, Heat Transfer A Practical Approach, Second Edition.

PAPERS

18- Boiling Heat Transfer Inside Tubes, Wolverine Tubes, Inc.

19- Comport Enhancement Compressors, R744 Compressor, next generation compressor for environmentally friendly CO_2 refrigerant, VALEO.

20- Denso, CO_2 Air Conditioning Systems, March 4, 2006.

21- Ixetic, R744 Compressor Development, 2006.

22- "Scroll Compressors For CO2 Refrigeration Cooling with Carbon Dioxide" , 28th March 2007, London

23- Tadano, M., T. Ebara and A. Oda, 2000, "Development of the CO2 hermetic compressor", Proc. of the 4th IIR□Gustav Lorentzen Conference on Natural Working Fluids at Purdue, pp. 323–330.

24- Lau Industries, for fans and blowers.

HYPER LINKS

25- http://www.r744.com/news/news.ida221/php.

26- http://www.peacosoftware.de/einigewerte/co2-e.html.

27- http://membership.ashrae.org/template/AssetDetail?asseetid=42175.

28- www.aircondition.com/ co2/cycle/compressor.htm.

29- http://www.wou.edu/avorder/refrigeration.htm.

30- httm://www.wikipedia.org/wki/global_worming_potential.htm.

31- http://www.ent.ohiou.edu/thermo/me328/potter/CO2/CO2-tempSAT.htm.

32- http://www.visteon.com/products/automotive/CO2-system.shtm.

33- http://search.globalspec.com/Search?query=co2%20compressor&show=total.

34- http://www.eibis.com/

35- http://www.r744.com/knowledge/list_papers.php.

36- http://www.guentner.de/en/products/refrigerant-co2/

37- http://www.danfoss.com.

38- http://www.xpplc.com/fanselection.htm.

39- cool pack software.