بسم الله الرحمن الرحيم

Palestine Polytechnic University

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Mechanical Engineering Department

Graduate Project

Design and development of refrigerated chamber for blood plasma at-40 C

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دائرة الهندسة الميكانيكية

اسم المشروع

Design and development of refrigerated chamber for blood plasma at -40 C

اسماء الطلبة

عامر عبد القادر الزرو **يوسف مازن شويكي**

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

توقيع المشرف

...

توقيع اللجنة الممتحنة

....

توقيع رئيس الدائرة

...

Dedication …

To our parents and families ….

To our supervisor Dr. Ishaq Sider …

To our instructors and colleagues …

To our friends …

To whom their guidance and support made this work possible …

Acknowledgment (شكر و تقدير)

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

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

Abstract

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

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

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

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CHAPTER ONE

INTRODUCTION

CHAPTER ONE

INTRODUCTION

A two stage vapor compression cooling cycle is very important ,the technological advance change from better to the best , to achieve and maintain temperature below that of surrounding .

1.1 Scope of Project:

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

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

1.2 History of two stage refrigeration system

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

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

1.3 Back ground about two stage refrigeration system

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

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

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

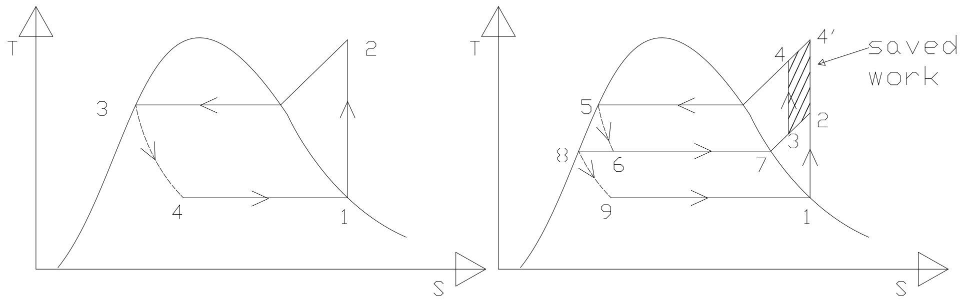


Figure 1.1. Saved work in two stage system

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

1.3 The project cycle selection

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

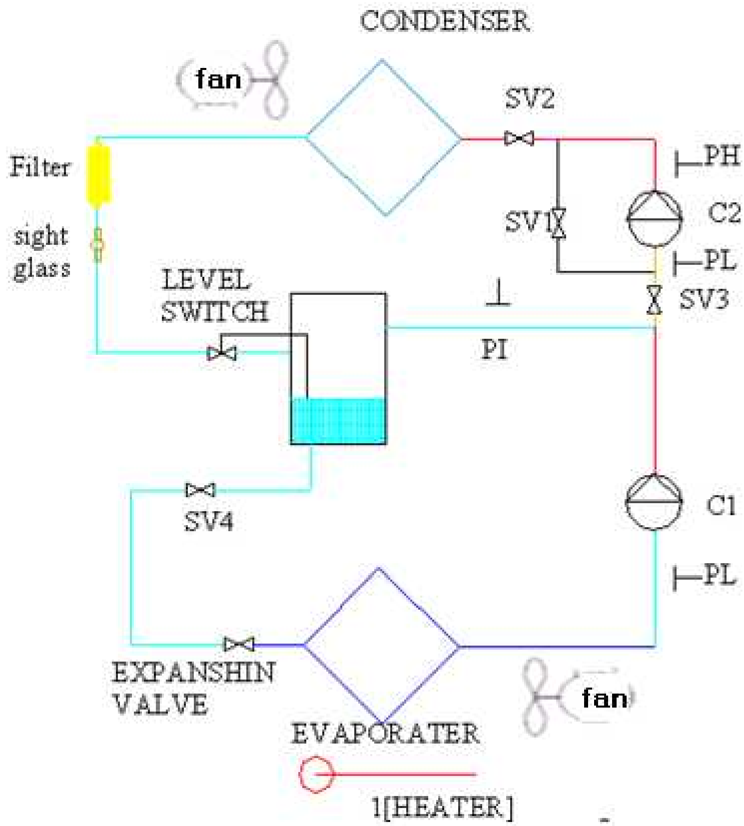


FIGURE 1.6 Tow stage cycle with all components and accessories

1.5 The budget for the project

Table 1.4 Actual budget table for the project

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

1.6 The time planning for the project

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

The following time plan is for the first semester

Table 1.5 The first semester time plan

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

CHAPTER TWO

COOLING LOAD

CHAPTER TWO

COOLING LOAD

2.1 Introduction

The total heat required to be removed from refrigerated space in order to bring it at the desired temperature and maintain it by the refrigeration equipment is known as cooling load. The purpose of load estimation is to determine the size of the refrigeration equipment that is required to maintain inside design conditions during periods of maximum outside temperatures. The design load is based on inside and outside design conditions and it's refrigeration equipment capacity to produce and maintain satisfactory inside conditions. [reference 13]

2.2 Load Sources

The cooling load on refrigerating equipment seldom results from any one single source of heat. Rather, it is the summation of the heat which usually evolves from several different sources. Some of the more common sources of heat that impose the load on refrigerating equipment are: [reference 5]

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

- Packing heat gain.
- Defrosts' heater heat gain
- Fan motor heat gain

2.2.1 The wall heat gain

The wall heat gain load, sometimes called the wall leakage load, is a measure of the heat flow rate by conduction through the walls of refrigerated space from the outside to the inside. Since there is no perfect insulation, there is always a certain amount of heat passing from the outside to the inside whenever the inside temperature is below that of the outside. The wall gain load is common to all refrigeration application and is ordinarily a considerable part of the total cooling load, commercial storage coolers and residential air conditioning applications are both examples of applications wherein the wall gain load often accounts for the greater portion of the total load. [reference 5]

$Qwall = U * A * \Delta T$……………………..…………………………………(2.1)

Where:-

A: out side surface area of the wall $[m2]$.

U: the overall heat transfer coefficient $[W/m2.C]$.

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

$\Delta T = Tout - Tin$

Where:

Tin: The refrigeration space temperature.

$Tout$: The outside temperature.

Overall heat transfer coefficient is computed by the following :

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

Where:

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

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

K: the thermal conductivity of the material [$W/m.\ ^{\circ}C$].

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

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

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

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

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

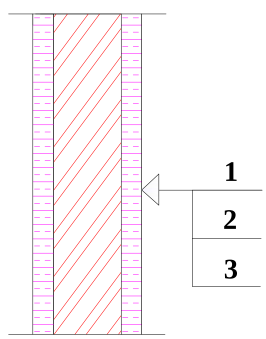


Figure 2.2 walls construction

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

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

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

3-foams 5 [cm]; K=0.036[W/m. °C]. From (Table A1)

$$U = \frac{}{\frac{1}{50} + \frac{0.001}{15.6} + 0}$$

$$U = 0.233 [W/m2\ °C]$$

Q_{wall}=U *A*ΔT

- Q_{floor}=U *A*ΔT =0.233*(0.754*0.604)*(38- -40).

  Q_{floor}=8.27 w.

- Q_{sides}=U *A*ΔT =0.233*(0.604*0.540)*(38- -40) .

  Q_{sides}=6 w.

- $Q_{front, behind}$=U *A*ΔT =0.233*(0.754*0.604)*(38- -40).

  $Q_{front, behind}$ =8.27 w.

  $Q_{for\ all\ walls}$= (Q_{floor} +Q_{sides}+$Q_{front, behind)}$*2

  $Q_{for\ all\ walls}$ =2 *(8.27+6+8.27) =45.08 w.

2.2.2 The product heat gain

The heat emitted from the product to be stored is very important in case of cold storages. The loads to be considered in the cold storages are divided into the following groups

1. Chilling load above freezing: The product chilling load above freezing depends upon the mass product, mean specific heat of the products above freezing, entering product temperature, final product temperature desired, and the chilling time. This heat gain can be calculated by the following equation: [reference 13]

$$Q\ ch = m . Cp . \Delta T \text{..} (2.3)$$

Where:

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

m :mass of the product in [kg]

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

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

Where:

T_o : entering product temperature [˚C]

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

C_P = 3.92 [kJ/kg.ºC] (table A2)

T_o = 38[ºC]

T_{ch} = -0.9 [ºC] (table A2)

$Qch = 10 * 3.92 * (38 - -0.9)$

Q_{ch} = 1525[kJ]

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

$Qc = m.CP'.\Delta T$……………………………………………………………(2.4)

Where :

Qc : cooling product load in [kJ]

m : mass of the product load in [kg]

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

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

Where:

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

T_{Rs} : refrigerated temperature [˚C]

$Cp' = 2.00\ [kJ/kg.\ ^{o}C]$ table A2

$Tch = -0.9\ [^{o}C]$ table A2

$TRS = -35\ [^{o}C]$

$Qc = m.CP'.\Delta T$

$Qc = 10 * 2 * (-0.9 - -35)$

$Qc = 682[\ kJ]$

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

$Qf = m.HL$ ………………………………………………………….……(2.5)

Where

Qf : freezing load[kJ]

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

$HL = 307[kJ/kg](\ table\ A2)$

$Qf = 10 * 307$

$Qf = 3070[\ kJ]$

Total product load

$Q\,p = \frac{Qch+Qc+Qf}{\tau}$ ……………………………………………………..……..(2.6)

Where

τ : desired cooling time in [seconds]

$$Q\,p = \frac{1525 + 682 + 3070}{10 * 3600} * 1000$$

$Qp = 146.6\ [W]$

2.2.3 Infiltration heat gain

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

$$Qinf = \dot{m} * Cp * (To - Ti) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2.7)$$

$$Qinf = \rho * \dot{V}f * Cp * (To - Ti)$$

Where

ρ : air density $[1.25\ kg/m3]$

Cp : the specific heat of the air $[1000 J/kg.{}^{\circ}C]$

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

To : the out side temperature $[{}^{\circ}C]$

Ti : the inside temperature $[{}^{\circ}C]$

$\dot{V}f$ = number of air change $*$ volume of room

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

volume of room $= 0.75 * 0.6 * 0.5 = 0.225\ m3$

$\dot{V}f = 0.5 * 0.225 = 0.1125\ m3/hr$

$Qinf = 1.25 * (0.1125/3600) * 1000 * (38 - -35)$

$Qinf = 3[W]$

2.2.4 Packaging Heat gain

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

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

$$\mathrm{Qpk} = \frac{\mathrm{mpk} * \mathrm{Cpk} * (\mathrm{To} - \mathrm{Ti})}{\tau} * 10^3 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2.8)$$

Where

Qpk : packaging heat load[W]

mpk : mass of product [kg]

Cpk : packaging material specific heat [J/kg.˚C]

To : out side temperature [˚C]

Ti : temperature of the refrigerant space [˚C]

τ : desired cooling time in [seconds]

For stainless steel crate

Cpk steel = 0.5[kJ/kgºC] (table A4)

$$Q_{pk} = \frac{5 * 0.5}{10}$$

Qpk steel = 5.41[W]

For plastic Packaging

Cpk plastic $= 1.6[\mathrm{kJ/kg^oC}]$ (table A4)

$$Q_{pk} = \frac{5 * 1.6 * (38—40}{10 * 3600} * 1000 = 17.3w$$

Qpk plastic $=$ 17.3[W]

Qpk total $=$ Qpk steel $+$ Qpk plastic

$$Q_{pk\ total} = 17.3 + 5.41 = 22.7w.$$

2.2.5 Defrosts' heater heat gain

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

Defrosts' heater heat gain of the evaporator

If the surface temperature of evaporator coil is (0°C) and lower, frost accumulates on the coil surface. Because frost impedes air passage and reduces the rate of heat transfer of the coil, it must be removed periodically. An electric heating element is used as a simple and effective way to defrost the coil. [reference 1]

Qh1 $= \eta * P$……………………………………..………………..…. (2.9)

Where

P: power of heater , taken 480[W]

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

Qh1 $= 0.2 * 480 = 96$ [W]

Defrosts' heater heat gain around the door

The function of the heater around the door is to prevent frost from forming around the door, making it difficult to open, and to prevent condensation phenomena around the door that happen because high temperature difference between the refrigeration space and the surrounding.

$Qh2 = \eta * P$

Where

P: power of heater , taken 80[W]

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

$Qh2 = 0.2 * 80 = 16[W]$

Total defrosts' heater load

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

2.2.6 Fan motor heat gain

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

$Qmotor = \text{power of motor} = 25\ W$

2.3 Total cooling load

The total cooling load is the summation of the heat gains

$$QT = Qw + Qp + Qinf + Qpk + Qh + Qmotor$$

$$Q_T = 45.08 + 146.6 + 3.04 + 22.77 + 112 + 25$$

$$Q_T = 354.49 \text{ W}$$

Add 20% upon QT as a safety factor

So

$$\text{Total cooling load} = QT * 1.2$$

$$Q_{Total,\ f.s} = Q_T * 1.2 = 425.388\text{w}$$

2.4 Required equipment capacity

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

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

$$Q_e = \frac{425.3 * 24}{16}$$

$$Qe = 638.1[W]$$

CHAPTER THREE

REFRIGERANT SELECTION AND CYCLE ANALYSIS

CHAPTER THREE

REFRIGERANT SELECTION
AND CYCLE ANALYSIS

3.1 Refrigerant Selection

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

Most of the early refrigerant materials have been discarded for safety reasons or for lack of chemical or thermal stability. In the present days many new refrigerants are used and can be classified into the following five main groups.

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

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

Thermodynamic properties

- Low boiling point.
- High critical point temperature.
- High latent heat of vaporization.
- Low specific heat of liquid.
- Positive pressure at evaporator temperature.
- Low specific volume at suction pressure and temperature.
- Mixes well with oil.

Thermophysical properties

- High thermal conductivity.
- High convection of heat transfer coefficient.
- Low dynamic viscosity.

Environment friendly

- Having zero ozone depletion potential (ODP).
- Having low global warming potential (GWP).
- Non contribution of the atmosphere’s heat retention.
- Non corrosive to metal.
- Nonacidic in case of a mixture with water or air.
- Chemically stable.
- Easily detectable in case of leakage,.
- Nonreactive with the lubricating oils of the compressor.
- Non flammable and non explosive.
- Non toxic.

Economics

- High operating efficiency.
- Low cost.
- Easy availability.

Table 3.1 Common refrigerants and some of its properties.

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

To select refrigerant successfully must be considered the above properties. A comparison between various refrigerants are made and found that, the best refrigerant could be used is R410A, but the small compressor that used this refrigerant are scarce, so will be avoid using this refrigerant. The second best refrigerant is R-404A from zeotropic mixtures family that consists from precise mixtures of substances that have different properties. If used R507 the COP of the cycle is less than the COP of R404A because the latent heat is less than it. If used R22 it is possible, but the environment harmed. So the best and suitable refrigerant could be used are R404A.

3.2 Cycle Analysis

3.2.1 Flow Processes

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

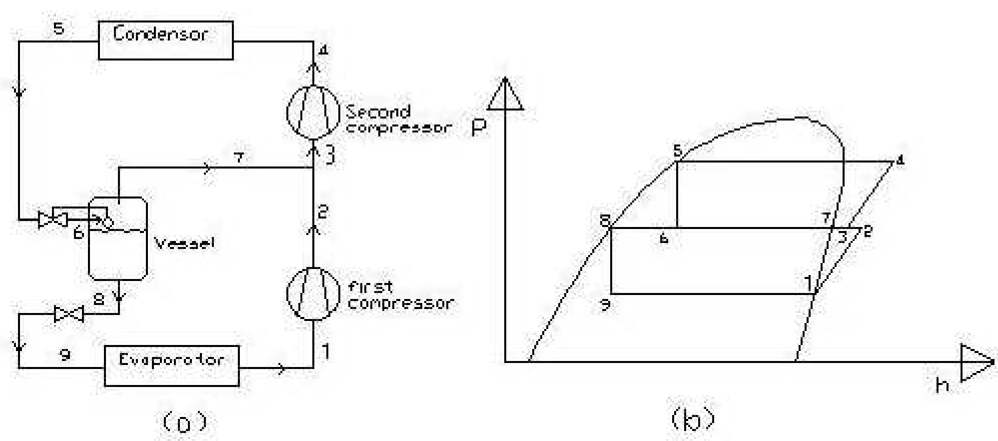


Figure 3.1. Two-stage compound system with a vessel: (a) schematic diagram; (b) Refrigeration cycle.

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

3.2.2 Intermediate Pressure

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

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

Where

Pi : intermediate pressure[bar]

pc : condensing pressure, [bar]

pe : evaporating pressure, [bar]

$pc = 20.4$ [bar]

$pe = 1.055$[bar]

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

$Pi = 4.64$ [bar]

The compression ratio in each stage can be calculated as,

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

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

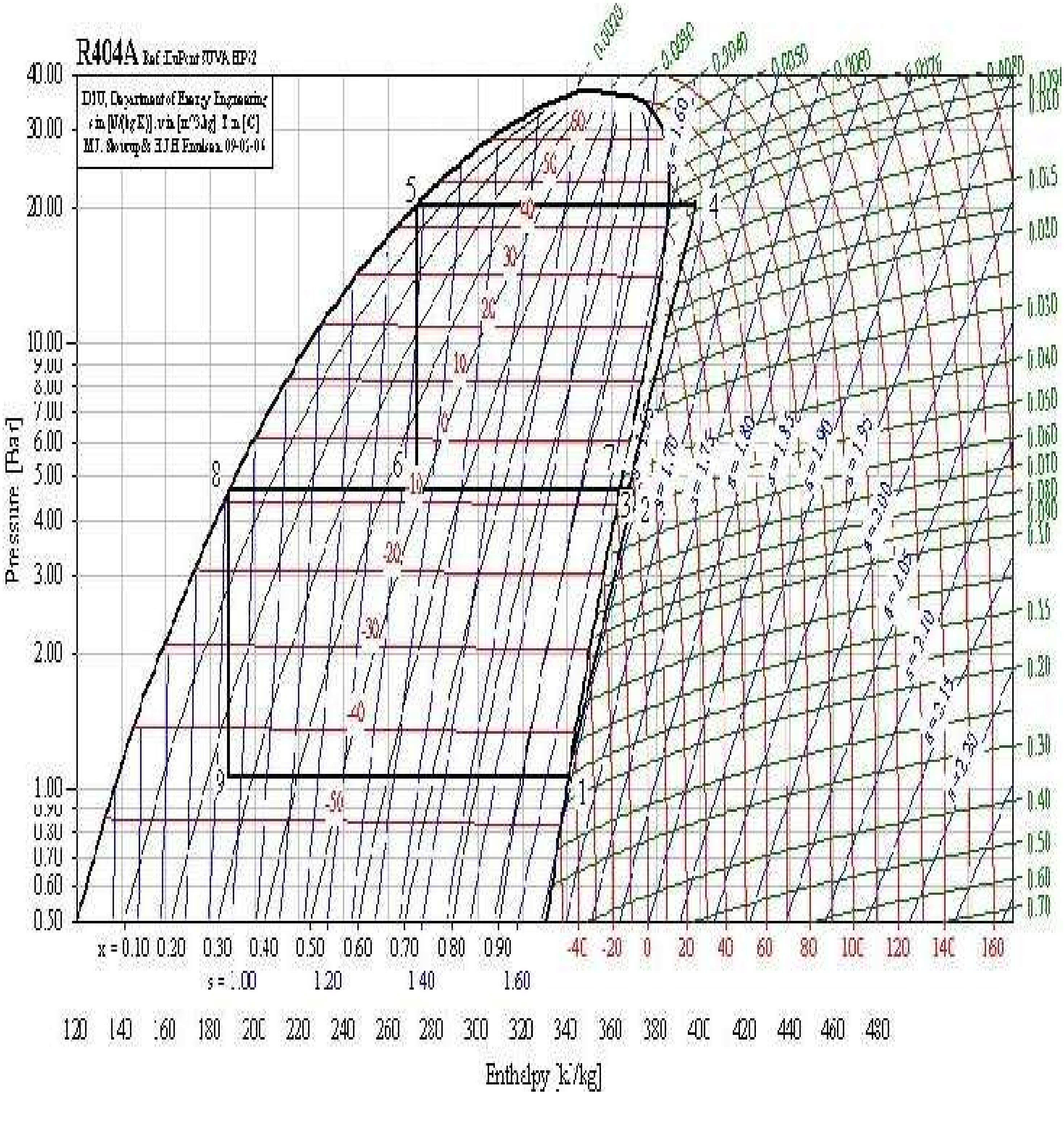


Figure 3.2. P-h diagram for R-404A

Table 3.2 Properties of each point on the cycle

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

3.2.2 Mass flow rate

In the vessel, out of $m.2$ of refrigerant flowing through the condenser, $m^{\cdot}$ of it enter the intermediate line and cools down the remaining portion of liquid refrigerant $m^{\cdot}_{1}$ to saturated temperature T8 at intermediate pressure that enter to the evaporator. The mass flow rate entering the evaporator can be calculated as, [reference 1]

$$m.1 = \frac{Qe}{qe} \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots . (3.3)$$

Where :

$m.1$: mass flow rate in the evaporator, [kg/s].

Qe : refrigeration capacity, [kW]

$$qe = (h1 - h9) \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots . (3.4)$$

Where:

qe : refrigeration effect, [kJ/kg]

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

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

$$qe = 340.6 - 188 \quad = 152.6 [kJ/kg]$$

$$m.1 = \frac{0.693}{152.6}$$

$$m.1 = 0.00454\ [kg/s]$$

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

$$\dot{m}_2 = \dot{m}_1 + \dot{m} \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \quad (3.5)$$

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

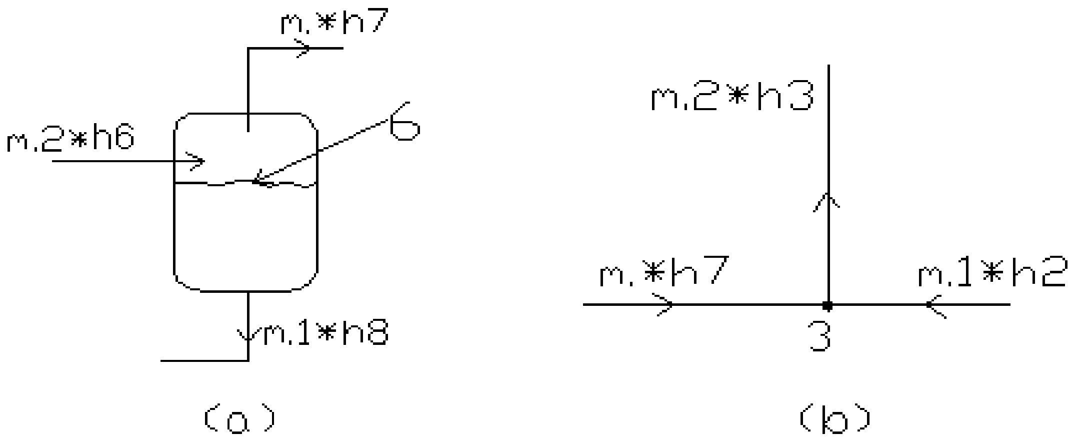


Figure 3.3. Heat balance of entering and leaving refrigerants: (a) in the vessel; (b) at the mixing point 3 before entering the second compressor.

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

That is,

$$\dot{m}_2 h_6 = \dot{m} h_7 + \dot{m}_1 h_8 \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \quad (3.6)$$

where

- $\dot{m}_1$: mass flow rate in the evaporator, [kg/s].
- $\dot{m}_2$: mass flow rate in the condenser, [kg/s].
- $\dot{m}$: mass flow rate in the intermediate line, [kg/s].
- h_6 : enthalpy of the saturated liquid refrigerant entering the vessel at point 6, [kJ/kg]
- h_7 : enthalpy of saturated vapor refrigerant from vessel at point 7, [kJ/kg]

$h8$: enthalpy of saturated liquid from vessel at point 8, [kJ / kg]

When $m.1 = 0.00454 \frac{kg}{s}$, and substitute equation 3.5 in equation 3.6 , resulted:

$m. = 0.0042$ [kg/s]

$m.2 = 0.00454 + 0.0042$

$m.2 = 0.00874$ [kg/s]

3.2.2 Enthalpy of vapor mixture entering second compressor

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

$$m.2\,h3 = m.1\,h2 + m.\,h7 \quad \text{.......................................}(3.7)$$

where:

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

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

$0.00874 * h3 = 0.00454 * 370 + 0.0042 * 363.8$

$h3 = 367$ [kJ/kg]

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

3.2.2 Condenser load

The condenser load can be calculated as

$$Qc = m.2\,(h4 - h5) \quad \text{.......................................}(3.8)$$

Where

Qc : condenser load [kW]

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

$Qc = 0.00874\ 397 - 272.6 = 1.087$ [kW]

3.2.2Work of compressors

The work of the compressors can be calculated as

$$WL = m.1(h2 - h1) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.9)$$

Where

WL : low cycle compressor work [kW]

$$WL = 0.00454(370 - 340.6)$$

$$WL = 0.133[kW]$$

$$WH = m.2(h4 - h3) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.10)$$

Where

WH : high stage compressor work [kW]

$$WH = 0.00874(397 - 367)$$

$$WH = 0.262[kW]$$

3.2.2Coefficient of performance

$$COP = Qe/Wtotal \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.11)$$

Where

COP : Coefficient of performance of the cycle

Wtotal : Total work input to the compressors , [kW]

$$Wtotal = WL + WH \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.12)$$

$$Wtotal = 0.133 + 0.262$$

$$Wtotal = 0.395 \ [kW]$$

$$COP = 0.693/0.395$$

$$COP = 1.75$$

CHAPTER FOUR

COMPONENTS SELECTION AND PROJECT DESIGN

CHAPTER FIVE

COMPONENTS SELECTION

4.1 Compressors Calculations And Selection

4.1.1 for low compressor

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

$$\eta v = \eta c * \eta h \ldots\ldots (5.1)$$

Where:

ηv: Volumetric efficiency

ηc: volumetric efficiency due to clearance volume in compressor

ηh: volumetric efficiency due to heating occurs in compressor

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

$$\eta c = 1 - c\left[\left(\frac{PH}{PL}\right)^{1/n} - 1\right] \ldots\ldots (5.2)$$

Where:

c : clearance volume (ratio between volumetric clearance and volume of cylinder of the compressor , c = 0.04 for low pressure different, c =0.02 for high pressure different [reference 7]:

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

PH: High pressure of the cycle

PL : Low pressure of the cycle

$$\eta c = 1 - 0.02 [(\frac{4.64}{1.055})1/1 - 1] = 93.2\,\%$$

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

$$\eta h = \frac{Tevap}{Tcond} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots . . (5.3)$$

Where:

Tevap. : Evaporator temperature $[\,^{o}K]$

Tcond. : Condenser temperature $[\,^{o}K]$

$$\eta h = \frac{228}{265} = 86\%$$

$$\eta v = 93.2\% * 86\% = 0.8\%$$

The theoretical volume flow rate ($V^{\cdot}$) of the compressor can be calculated in equation, [reference 6]:

$$V \cdot theo = m . * v \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots . . \ldots\ldots\ldots (5.4)$$

Where:

$V \cdot$ Theo: Theoretical volume flow rate of the compressor [m^3/s]

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

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

$$V.\,theo = 0.00454 * 0.17758 = 8 * 10^{-4}\ [m3/s]$$

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

$$V \cdot act = \frac{V.\,theo}{\eta v} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots . \ldots\ldots\ldots (5.5)$$

Where:

$V \cdot act$: Actual volumetric flow rate [m^3/s]

$$V.\,act = \frac{0.0008}{0.8} = 1 * 10^{-3} [m3/s]$$

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

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

So

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

$$V.act = V \cdot theo * \eta v \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (5.6)$$

Where:

$V \cdot act$: Actual volumetric flow rate for the compressor [m^3/s]

$V \cdot theo$: Theoretical volumetric flow rate for the compressor [m^3/s]

ηv: Volumetric efficiency

$$V \cdot theo = 26.2 * 10^{-6} * (3000/60) = 1.31 * 10^{-3} \, [m3/s]$$

$$V.act = 1.27 * 10^{-3} * 0.8 = 1.048 * 10^{-3} \, [m3/s]$$

4.1.2 for high compressor

$$\eta v = \eta c * \eta h$$

$$\eta c = 1 - 0.02\left[\left(\frac{20.4}{4.64}\right)1/1 - 1\right] = 93\,\%$$

$$\eta h = \frac{265}{318} = 83\%$$

$$\eta v = 93\% * 83\% = 77\%$$

$$V.theo = m.2 * \upsilon$$

$$V.theo = 0.00874 * 0.043 = 3.7 * 10^{-4} [m3/s]$$

$$V.act = \frac{0.00037}{0.77} = 4.8 * 10^{-4} [m3/s]$$

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

So

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

$$V.act = V \cdot theo * \eta v$$

$$V \cdot theo = 12.6 * 10^{-6} * (3000/60) = 6.3 * 10^{-4} [m3/s]$$

$$V.act = 6.3 * 10^{-4} * 0.77 = 4.851 * 10^{-4} [m3/s]$$

4.2 Pipe Design and Selection

4.2.1 Introduction

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

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

$$Q = m * \upsilon \ldots \ldots (5.7)$$

Where:

Q: Flow rate [m^3/s]

m : Mass flow rate of refrigerant [kg/s]

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

$$Q = V * A \ldots \ldots (5.8)$$

Where:

V: Velocity of refrigerant [m/s]. [Table A-6]

A: cross sectional area [m^2]

$$A = \pi di2 /4 \ldots \ldots (5.9)$$

Where:

di : inner diameter [m]

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

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

$$\text{б}t = \frac{Pin(ro^2 + ri^2)}{(ro^2 - ri^2)} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (5.10)$$

Where:

$\text{б}t$: tangential stress [Mpa]

Pin : inner pressure [Mpa]

ro : outer radius [m]

ri : inner radius [m]

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

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

Where:

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

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

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

$$t = ro - ri \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (5.12)$$

Where:

t : Thickness of the pipe [mm]

4.2.2 Low Cycle Pipes

- Suction line pipe

$$Q = \dot{m}_1 * \upsilon_1$$

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

$$A = \frac{Q}{V} = \frac{0.0008}{10} = 8 * 10^{-5} [m2]$$

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

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

$$\sigma t = 87\ [bar]$$

$$87 = \frac{4.64(ro^2 + 0.005^2)}{(ro^2 - 0.005^2)}$$

$$ro = 5.06\ [mm]$$

$$t = 5.06 - 5 = 0.06\ [mm]$$

The inner and outer diameter in inch is

$$di, inch = di, mm\ /25.4 = 10/25.4 = 0.393\ [inch]$$

$$do, inch = do, mm\ /25.4 = 10.12/25.4 = 0.398\ [inch]$$

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

- discharge line pipe

$$Q = m.1 * \upsilon 2$$

$$Q = 0.00454 * 0.0448 = 2 * 10^{-4} \text{ [m3/s]}$$

$$A = \frac{Q}{V} = \frac{0.0002}{12} = 1.66 * 10^{-5} \text{[m2]}$$

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

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

$$\text{б}t = 85 \text{ [bar]}$$

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

$$ro = 2.43 \text{ [mm]}$$

$$t = 2.43 - 2.3 = 0.13 \text{ [mm]}$$

The inner and outer diameter in inch is

$$di, inch = 4.6/25.4 = 0.181 \text{ [inch]}$$

$$do, inch = 4.86/25.4 = 0.191 \text{ [inch]}$$

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

- liquid line pipe

$$Q = m.1 * \upsilon 8$$

$$Q = 0.00454 * 0.001 = 4.54 * 10^{-5} \text{ [m3/s]}$$

$$A = \frac{Q}{V} = \frac{0.0000454}{1} = 4.54 * 10^{-5} \text{[m2]}$$

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

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

$бt = 85\ [bar]$

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

$ro = 4.0134\ [mm]$

$t = 4.0134 - 3.8 = 0.2134\ [mm]$

The inner and outer diameter in inch is

$di, inch = 7.6/25.4 = 0.299\ [inch]$

$do, inch = 8.068/25.4 = 0.316\ [inch]$

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

4.2.3 High Cycle Pipes

- ❖ Suction line pipe

$Q = m.2 * υ3$

$Q = 0.00874 * 0.0435 = 3.8 * 10 - 4\ [m3/s]$

$$A = \frac{Q}{V} = \frac{0.00038}{12} = 3.16 * 10 - 5[m2]$$

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

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

$бt = 85\ [bar]$

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

$ro = 3.36\ [mm]$

$t = 3.36 - 3.17 = 0.19$ [mm]

The inner and outer diameter in inch is

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

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

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

- ❖ discharge line pipe

$Q = m.2 * \upsilon 4$

$Q = 0.00874 * 0.01 = 0.874 * 10^{-4}$ [m3/s]

$A = \frac{Q}{V} = \frac{0.0000874}{13} = 6.72 * 10^{-6}$ [m2]

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

$\frac{700}{8} = \sqrt{6t^2 + 20.4\,6t + (20.4)^2}$

$6t = 75.5$ [bar]

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

$ro = 1.76$ [mm]

$t = 1.76 - 1.5 = 0.26$ [mm]

The inner and outer diameter in inch is

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

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

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

❖ liquid line pipe

$Q = m.2 * \upsilon 5$

$Q = 0.00874 * 0.001 = 8.74 * 10^{-6}$ [m3/s]

$$A = \frac{Q}{V} = \frac{8.74 * 10 - 6}{1.2} = 7.283 * 10^{-6} [m2]$$

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

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

$\text{б}t = 75.5$ [bar]

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

$ro = 1.76$ [mm]

$t = 1.76 - 1.5 = 0.26$ [mm]

The inner and outer diameter in inch is

$di, inch = 3/25.4 = 0.118$ [inch]

$do, inch = do, mm / 25.4 = 3.52/25.4 = 0.138$ [inch]

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

4.2.4 Intermediate Line Pipe

$$Q = m. * \upsilon 7$$

$$Q = 0.0042 * 0.0431 = 1.81 * 10^{-4} [m3/s]$$

$$A = \frac{Q}{V} = \frac{0.000181}{12} = 1.5 * 10^{-5} [m2]$$

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

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

$$бt = 85 [bar]$$

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

$$ro = 2.27 [mm]$$

$$t = 2.27 - 2.15 = 0.12 [mm]$$

The inner and outer diameter in inch is

$$di, inch = 4.3/25.4 = 0.172 [inch]$$

$$do, inch = do, mm /25.4 = 4.54/25.4 = 0.178 [inch]$$

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

5.3 Evaporator Calculations and selection

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

evaporator length $Lc = 53$ [cm]

evaporator height $Hc = 5$ [cm]

evaporator width $Wc = 25$ [cm]

S_n: Transverse tube spacing [m]

S_p: Longitudinal tube spacing [m]

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

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

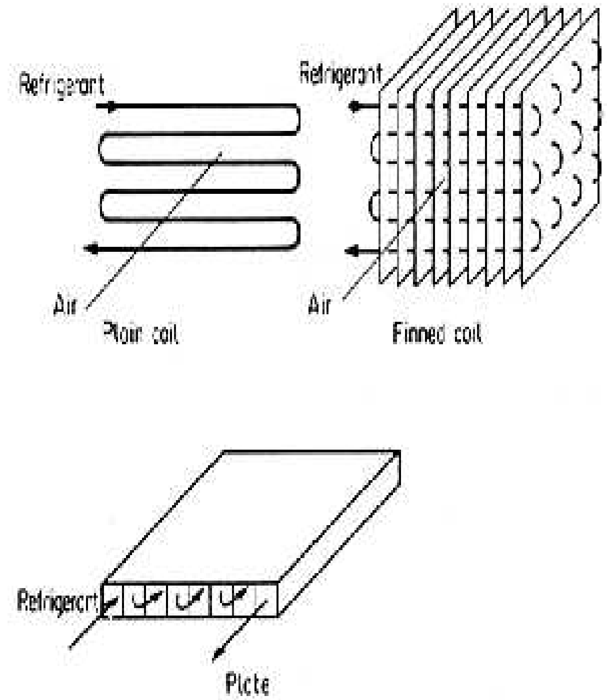


Figure 5.1 evaporator side view

$$fin\ length = L_f = S_n - D_o$$

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

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

$$W_f = 0.025 - 0.01 = 0.015 [m]$$

Figure (5.2) show the fin elements

number of fins in row $= 89$ fins

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

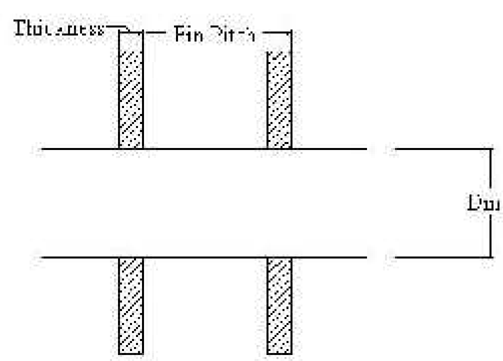


Figure 5.2 fin elements for evaporator

Fine thickness $= t_f = 0.03[cm]$

bare tube thickness $= t_b = P_f - t_f$

$t_b = 0.59 - 0.03 = 0.56[cm]$

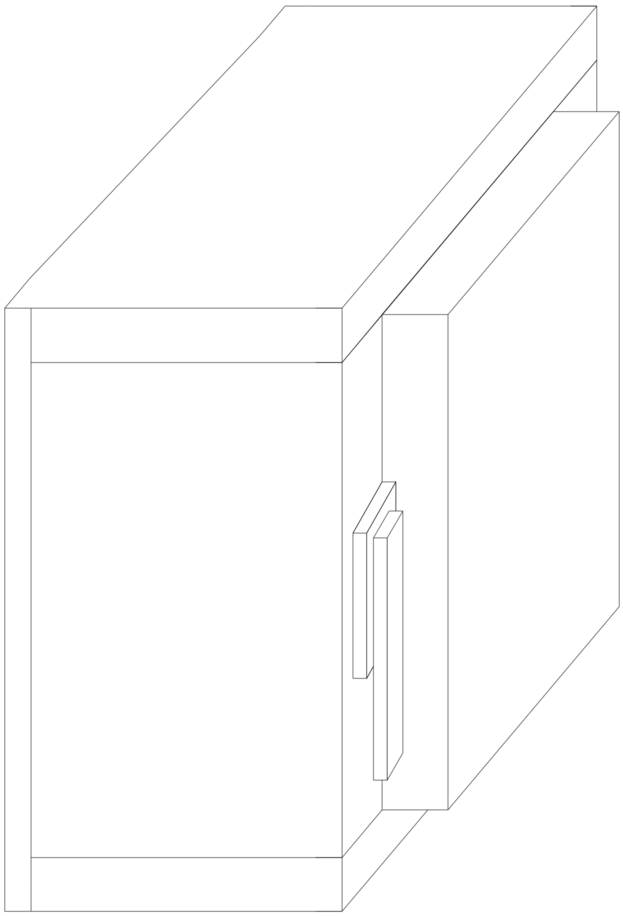
Design the evaporator required many calculations such as fluid mechanical calculation , thermal calculation and area calculation , the sequence of design start with fluid mechanical calculation ,in thermal calculation will be used the convection heat transfer equations for outer surface neglected the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

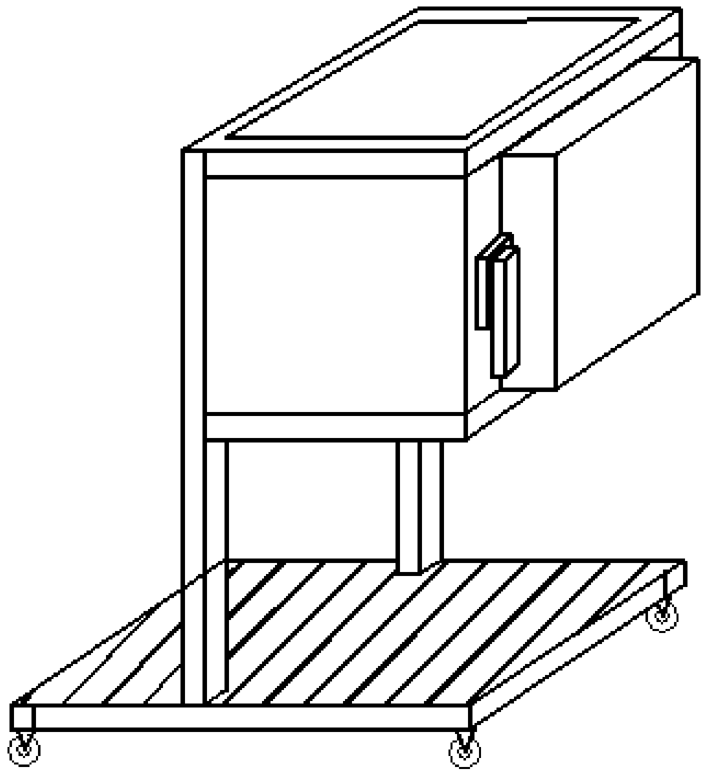
DESIGN OF STEEL STRUCTURE

Drawing by: AmerZaro

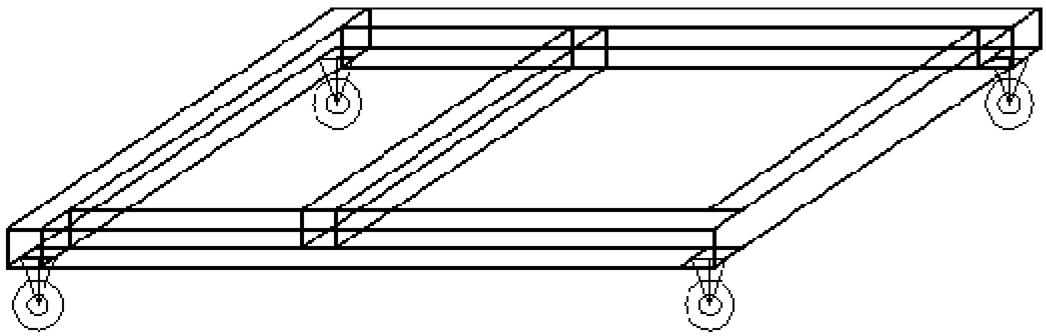
YousefShweiki

Check by: Dr. Ishaq Sider



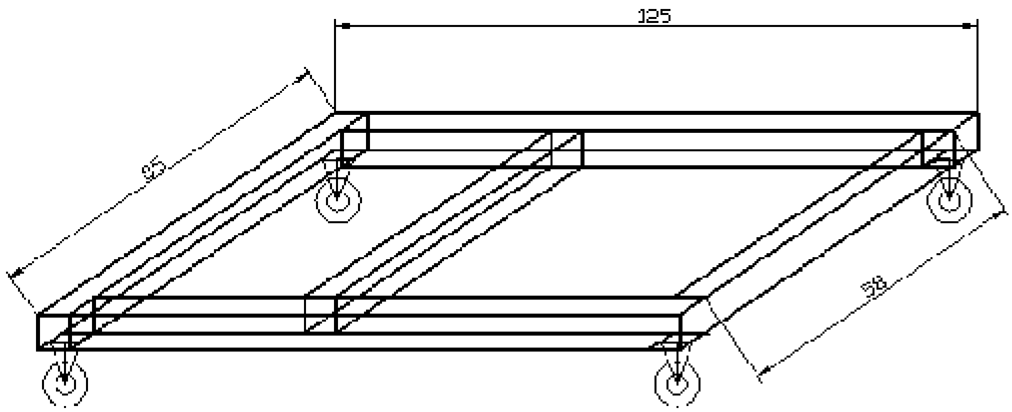


Palestine Polytechnic University		
Draw Number : **6 – 2 –1**	Descriptions and details : General shape of the refrigerator that will be established made by steel SE-66, connection using wielding different methods.	**Students names :** AmerZaro YousefShweiki
Title : The Refrigerator		**Project Supervisor :** Dr .Ishaq Sider
Scale : 1 – 100		

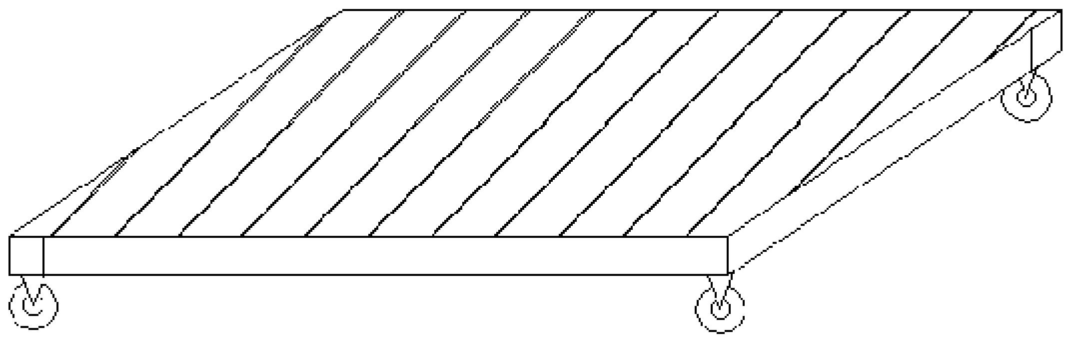


Palestine Polytechnic University

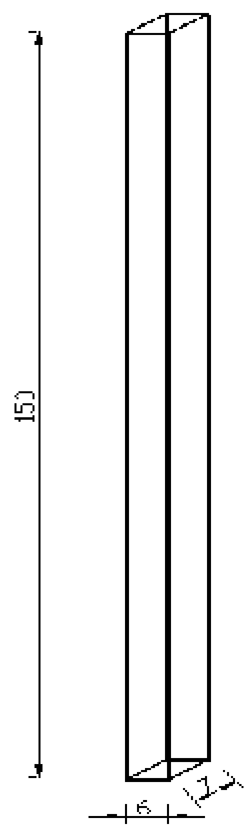
Draw Number : **6 – 2 –2**	Descriptions and details : This base is made to carry load about 150 kilograms, and the truss is steel SE-66 connected using wielding different methods.	**Students names :** AmerZaro YousefShweiki
Title : Basic base		**Project Supervisor :** Dr .Ishaq Sider
Scale : 1 – 100		



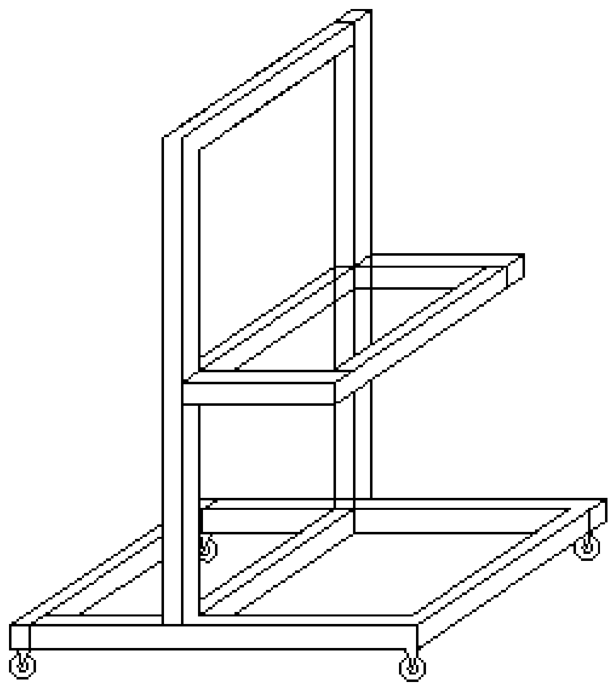
Palestine Polytechnic University		
Draw Number : **6 – 2 –3**	Descriptions and details : This base is made to carry load about 150 kilograms, and the truss is steel SE-66 connected using wielding different methods.	**Students names :** AmerZaro YousefShweiki
Title : Basic base dimensions		**Project Supervisor :** Dr .Ishaq Sider
Scale : 1 – 100		



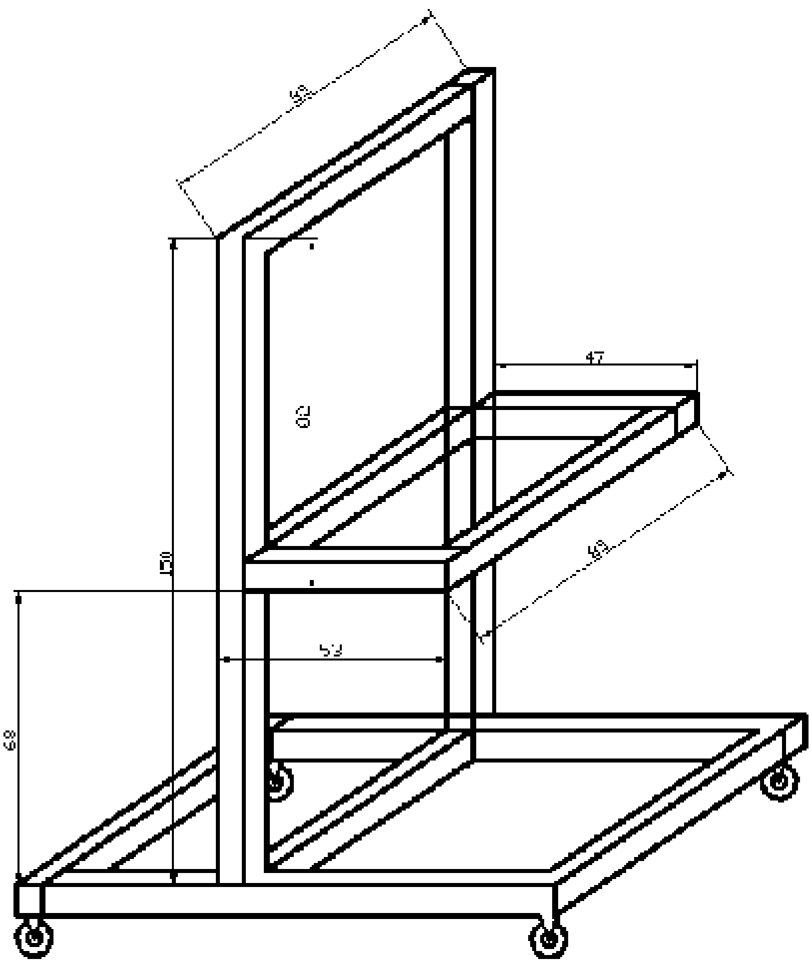
Palestine Polytechnic University		
Draw Number : **6 – 2 – 4**	Descriptions and details : The base covered by steals steel sheet specific as it's area .	**Students names :** AmerZaro YousefShweiki
Title : Basic base cover		**Project Supervisor :** Dr .Ishaq Sider
Scale : 1 – 100		



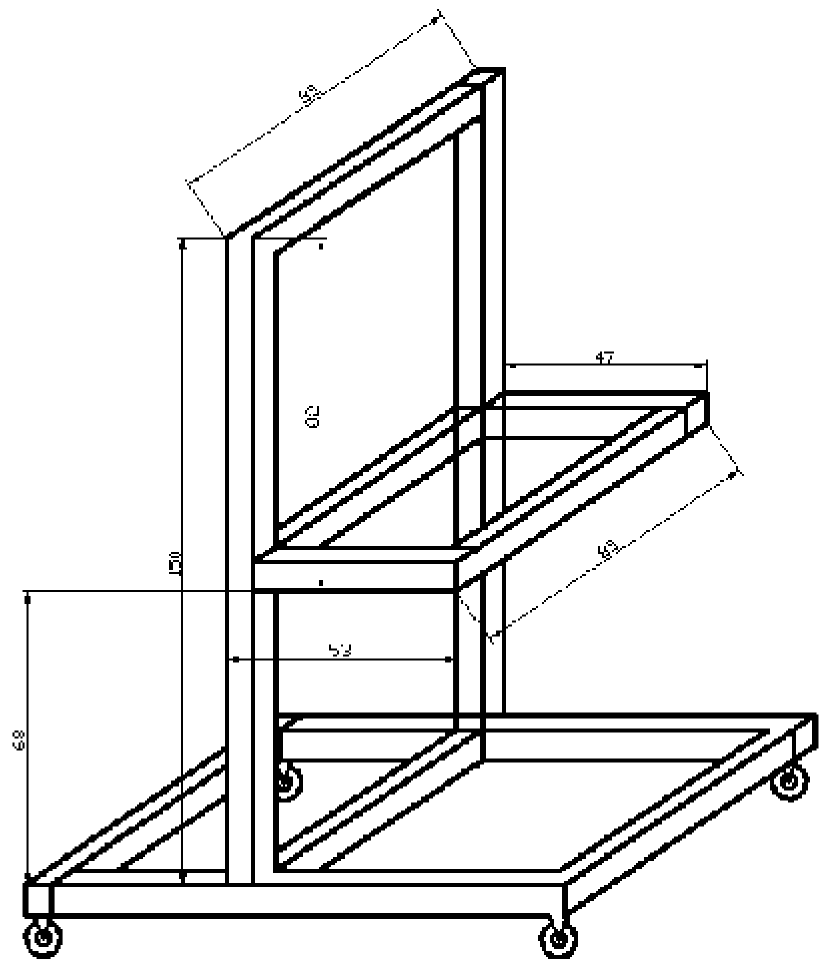
Palestine Polytechnic University		
Draw Number : **6 – 2 – 5**	Descriptions and details :	**Students names :** AmerZaro YousefShweiki
Title : Carrier column		**Project Supervisor :** Dr .Ishaq Sider
Scale : 1 – 100		



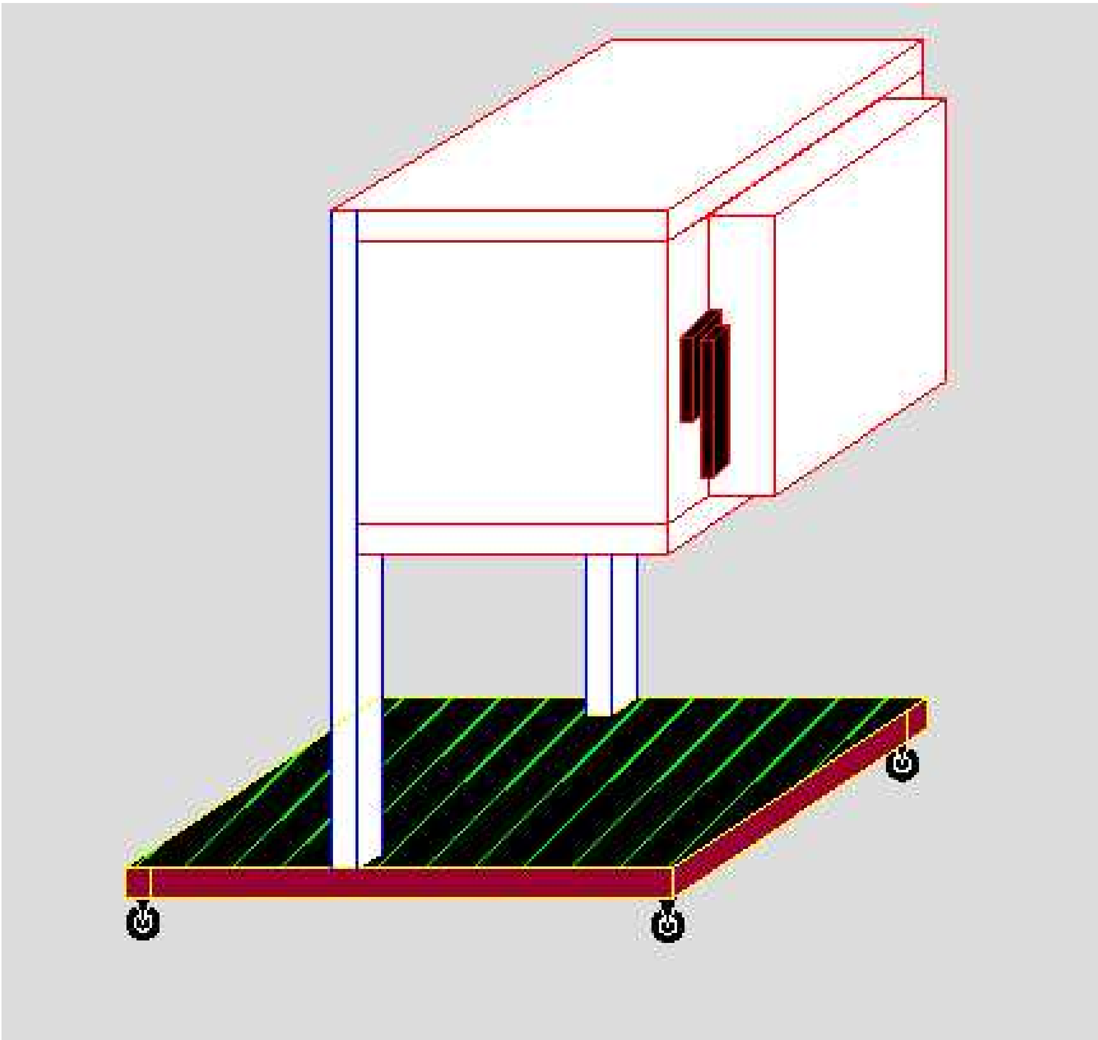
Palestine Polytechnic University		
Draw Number : **6 – 2 – 6**	Descriptions and details :	**Students names :** AmerZaro YousefShweiki
Title : Parts details		**Project Supervisor :** Dr .Ishaq Sider
Scale : 1 – 100		



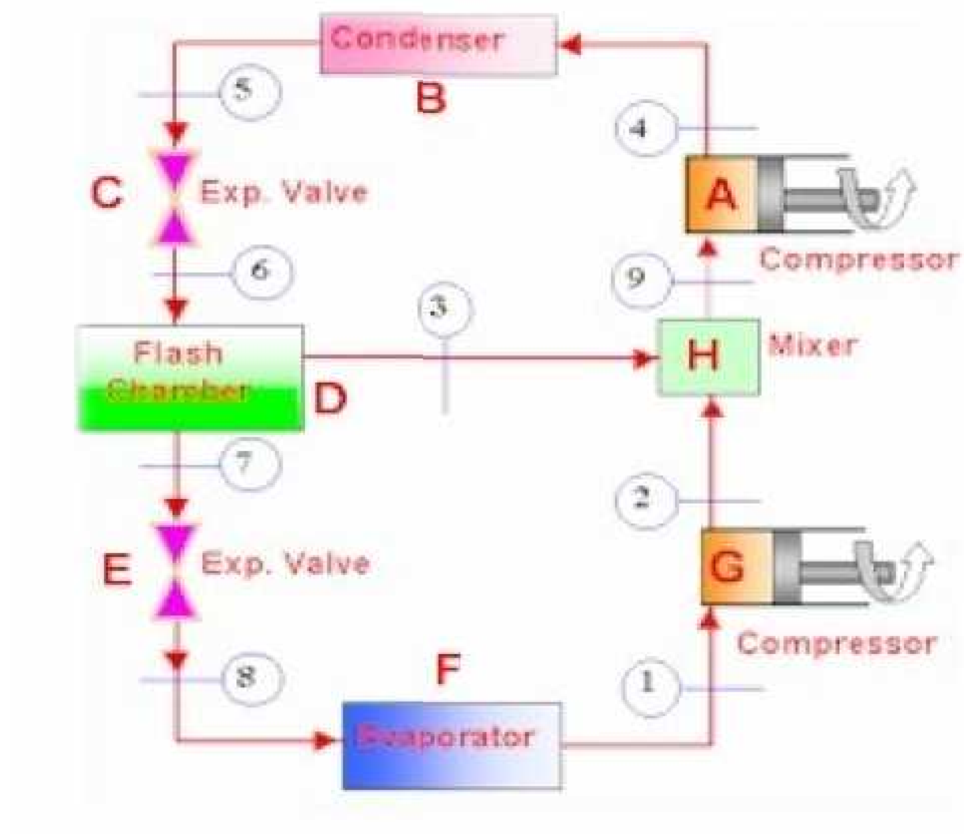
Palestine Polytechnic University		
Draw Number : **6 – 2 –7**	Descriptions and details :	**Students names :** AmerZaro YousefShweiki
Title : Parts details dimensions		**Project Supervisor :** Dr .Ishaq Sider
Scale : 1 – 100		



Palestine Polytechnic University		
Draw Number : **6 – 2 – 8**	Descriptions and details :	**Students names :** AmerZaro YousefShweiki
Title : refrigerator dimensions		**Project Supervisor :** Dr .Ishaq Sider
Scale : 1 – 100		



Palestine Polytechnic University		
Draw Number : **6 – 2 – 9**	Descriptions and details :	**Students names :** AmerZaro YousefShweiki
Title : The Refrigerator 3D		**Project Supervisor :** Dr .Ishaq Sider
Scale : 1 – 100		


Condenser
B
5
4
C
Exp. Valve
A
Compressor
6
9
3
Flash
Chamber
D
H
Mixer
7
2
E
Exp. Valve
G
Compressor
8
F
1
Evaporator