بسم الله الرحمن الرحيم

Palestine Polytechnic University

College of Engineering & Technology
Mechanical Engineering Department

Graduation Project

Design and building of refrigeration system for refrigerated chambers at three temperature levels

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جامعة بوليتكنيك فلسطين
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دائرة الهندسه الميكانيكية

اسم المشروع

Design and building of refrigeration system for refrigerated chambers at three temperature levels

أسماء الطلبة

حمزة "محمد اسماعيل" عرفه   رائد نهاد ابو منشار   ميره محمد دعدوع

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

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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….

Abstract

The project presents the design and implementation of three refrigerated chambers. The total volume of the chambers is about (200 L) for storing 30 kg meat, 30 kg cucumber, and 35 kg lemon, single stage vapor compression refrigeration cycle with multi evaporator swill be used for maintaining the required temperature in the champers, every chamber has a different temperature (0, 5, 8)C^0.

Full theoretical design and selection for each component, the design paid attention for ecological and environmental aspects in selecting the refrigerant, so R-134a presented the best refrigerant that is used.

The control of the system will be done by using traditional automation components with the possibility of improving the system by using direct digital controller (DDC) or programmable logic controller (PLC).

The system will be built and tested to make comparison between theoretical and actual results.

يعرض هذا المشروع تصميم وتنفيذ ثلاث غرف تبريد، يمثل الحجم الكلي للغرف ما يقارب (200 لتر) لتخزين ٣٠ كغ اللحوم، ٣٠ كغ الخيار، و٣٥ كغ الليمون، يستخدم النظام دورة تبريد ذات مرحله احادية الضغط متعدد المبخرات للوصول الى درجات الحرارة المطلوبه في الغرف ، كل غرفه لديها درجه حرارة مختلفه (٠،٥،٨) درجه مؤوية .

في هذا المشروع سوف يتم اختيار مكونات الدورة وعمل التصاميم النظرية الكامله لها، وفي هذا النظام تم اعطاء الاهمية الخاصة للجوانب الايكولوجية والبيئية في اختيار غاز التبريد، ولقد وجد ان غاز R143aهو افضل اختيار للاستخدام.

التحكم بالنظام باستخدام مكونات التحكم التقليدي مع امكانية تحسين التحكم باستخدام (DDC) او(PLC).

سيتم بناء النظام واختباره لعمل مقارنته بين القيم النظرية و القيم الحقيقية.

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16. Cool pack software.

17. Autocad software

18. Danfoss software

APPENDEX A

CHAPTER ONE

INTRODUCTION

CHAPTER ONE

INTRODUCTION

A vapor compression cooling cycles are very important, The technological advance change from better to the best, to achieve and maintain temperature below that of the surrounding .

1.1 Scope of the project

This project presents the design and implementation of three chambers at different temperatures. The system will include all mechanical and electrical parts to make it fully automation .It will be used as laboratory or workshop installation.

A single stage vapor compression refrigeration cycle is used. The volume of three chambers is (200L). These chambers will store cucumbers , lemons, and meat at three evaporators and one compressor. Three evaporators are designed to operate at three different temperature (8, 5, 0) C^0.Three temperature applications are normally accomplished with purely mechanical valve on a temperature controlled valve. This application is used in many applications that need multi temperatures .

1.2 History aboutmultiple evaporators system

Chambers are built for temperatures close to each other, so a single stage vapor compression refrigeration cycle is suitable for this project, but with wide range oftemperatures like (0,-35) C^0, other cycles like two stage or more should be used.

In 11-13/June/ 2008 , Matthew S. Elliott and Bryan P. Rasmussen produced paper about control architecture for multi-evaporator on vapor compression cooling cycle .The title of paper is Model-Based Predictive Control of a Multi-Evaporator Vapor Compression Cooling Cycle.[Reference12]

This paper studied the method of controlling multi-evaporator, the relationship between input and output parts, the relationship between evaporators, made liner equation, and found thatevaporator pressures and superheatseffected by Compressor speed. Compressor has a lower speed when evaporator pressure increases, Compressor speed increases when the condenser pressure increases, Relationship between first EEV and evaporator pressure, Relationship between discharge valve and second evaporator pressure.

In 10-12 / June / 2009, Matthew S. Elliott and Bryan P. Rasmussen wrote paper about control architecture for a multiple evaporators on HVAC , This paper entitled" A Model-Based Predictive Supervisory Controller for Multi- Evaporator HVAC Systems ". The paper presentscontrolarchitecture for system, and energy consumption studies. [Reference13]

In July /21/ 2003, Dr S. Forbes Pearson produced a paper entitled "Refrigerant Past, Present, and Future ", This paper presents improvement in refrigerants from first refrigerant to modern, and R134a properties.[Reference14]

Dr Forbes puts the (R 134a) properties and compares it with (R12), he has found that R134a and R12 properties are very similar so R134a is the best replacement for R12.Because it is less stable than R12, and has a lower critical temperature than R12.Since it is friendly for the environment and has ODP (ozone depletion potential) equal zero, R-134a is used in many applications like automotive, air conditioning, commercial refrigeration and air conditioning .[Reference13]

1.3 Backgrounds about a multiple evaporator system

The system consists of a single compressor and a single condenser but three evaporators. Evaporators-I, II, III operate at the same evaporator temperature (0C°) one evaporator (say Evaporator-I) caters to freezing while the other (Evaporator-II, III) caters to product cooling/space conditioning at 5C°, 8C°. [Reference 15]

1.4 The project cycle selection

In chapter six; page (33-34), shows a system schematic with all components and accessories of a multi-evaporator system that uses three evaporators at three different temperatures and a single compressor.

1.5 The budget for the project

Table (1.1) Actualbudget table for the project

Task	#	COST (NIS)	Total
Compressor	1	1000	1000
Condenser	1	500	500
pressure switches	1	200	200
Connecting pipes	30 m	1m=40	200
Expansion valve	3	250	750
Refrigerant R134a	2	600	1200
Heater	2	50	100
Fans	4	100	400
Building chambers	-	2000	1500
Structure of device			1500
Thermostat	3	200	600
Wires	-	600	600
Solenoid valves	3	150	450
Filter	1	100	50
Sight glass	1	60	60
contactors	3	100	300
Accessories		800	800
		Total	10210 NIS

1.6 The time planning for the project

The project plan follows the following time schedulewhich includes the related tasks of study and system analysis.The following time plansare for the first semester and second semester.

Table (1.2)thefirst semestertime plan

Task/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Choosing project title	▨	▨														
Collecting information	▨	▨	▨	▨	▨	▨	▨	▨	▨	▨	▨					
Reading	▨	▨	▨	▨	▨	▨	▨	▨	▨	▨						
Introduction			▨													
Load calculations			▨	▨	▨											
Cycle analysis					▨	▨										
Refrigerant choose					▨	▨										
Compressor Design							▨	▨								
Condenser Design										▨	▨	▨				
Evaporator Design									▨	▨						
Pipe Design							▨	▨								
Electric controlDesign				▨	▨	▨										
Preparing & printing			▨	▨	▨	▨	▨	▨	▨	▨	▨	▨	▨	▨	▨	▨
Project Documentation			▨	▨	▨	▨	▨	▨	▨	▨	▨	▨	▨	▨	▨	▨

Table (1.3)thesecondsemester time plan

Task/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Chambers building							▨	▨								
Compressorsselection		▨	▨													
Accessory Selection		▨	▨						▨	▨						
Cyclebuilding											▨	▨				
Electrical & Control building													▨			
Cycle test														▨		
Recommendations															▨	
Conclusions															▨	
Project Documentation				▨	▨	▨	▨	▨	▨	▨	▨	▨	▨	▨	▨	▨

CHAPTER TWO

COOLING LOAD

CHAPTER TWO

COOLING LOAD

2.1 Introduction

The total heat required to be removed from refrigerated space 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 its refrigeration equipment capacity to produce and maintain satisfactory inside conditions.[Reference 1]

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:-

- The wall heat gain.
- The product heat gain.
- Infiltration heat gain.
- Packing heat gain.
- Fan motor heat gain

Chambers are working at three temperature degrees (0, 5, 8)C^0, chambers are located in Hebron city , From Palestinian codes the Maximum surrounding temperature for Hebron city is 38◦C (TABLE A-1). The required mass product is30[kg] meat, 30[kg] cucumber ,35 [kg] lemon. Desired cooling time is 16[hr] From (TABLE A-14), chapter six explains the inner refrigerator size.

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 of the outside. The wall gain load is common to all refrigeration application and is ordinarily a considerable part of the total cooling load.

The quantity of heat transmitted through the wall is a function of three factors whose relationship is expressed in the following equation: [Reference2]

$$Q_{wall}=U\ A*\Delta T \tag{2.1}$$

Where A is the outside surface area of the wall [m^2], U is the overall heat transfer coefficient [W/m^2. ◦C], ΔT is the temperature difference across the walls [◦C].

$$\Delta T = T_{out} - T_{in} \tag{2.2}$$

Where T_{in} is the refrigeration space temperature, T_{out} is the outside temperature. Overall heat transfer coefficient is computed by the following:[Reference2]

$$U = \frac{1}{\frac{1}{hi} + \frac{\Delta x1}{k1} + \frac{\Delta x2}{k2} + \frac{\Delta x3}{k3} + \dots + \frac{1}{ho}} \tag{2.3}$$

Where U: is the overall heat transfer coefficient [W/m^2. C°],Δx: is the thickness of the layer of the wall [m] ,k: is the thermal conductivity of the material [W/m. °C] ,h_i: is the convection heat transfer coefficient of inside air film =9.37[W/m^2.°C] From (TABLE A-3), h_o: is the convection heat transfer coefficient of outside air film = 22.7[W/m^2.°C] From (TABLE A-3)All walls are constructed of three layers as shown in chapter six (22-34):

- Galvanized steel sheet 0.06 [cm];k=20[W/m. °C]. From (TABLE A-2)
- Polyurethane 4 [cm] k=0.025[W/m. °C]. From (TABLE A-2)
- Galvanized steel sheet 0.06 [cm];k=20[W/m. °C]. From (TABLE A-2)

Applying equation (2.3), the overall heat transfer coefficientis:

- $U = \frac{1}{\frac{1}{9.37} + \frac{0.0006}{20} + \frac{0.04}{0.025} + \frac{0.0006}{20} + \frac{1}{22.7}} = 0.57[W/m^2. C°]$

Applying equation (2.1), the wall load for first chamber (meat)is:

- $Q_{floor,\ roof}$= 2* {0.57*(0.6*0.5+0.4*0.05)*(38- 0)}= 13.8[W]
- Q_{sides}= 2*{0.57*(0.33*0.4+0.32*0.05)*(38- 0)}= 6.4[W]
- $Q_{front\ ,\ behind}$= 2*{0.57*(0.4*0.5)*(38- 0)}=8.6 [W]
- $Q_{all\ wall}$ =28 [W]

Applying equation (2.1) on second chamber (cucumber) is :

- $Q_{floor,\ roof}=2*\{0.57*(0.6*0.5+0.4*0.05)*(38-5)\}=12[W]$
- $Q_{sides}=2*\{0.57*(0.33*0.4+0.32*0.05)*(38-5)\}=5.56[W]$
- $Q_{front,\ behind}=2*\{0.57*(0.4*0.5)*(38-5)\}=7.5[W]$
- $Q_{all\ wall}=25\ [W]$

And applying equation (2.1) on third chamber (lemon) is:

- $Q_{floor,\ roof}=2*\{0.57*(0.6*0.5+0.4*0.05)*(38-8)\}=11[W]$
- $Q_{sides}=2*\{0.57*(0.33*0.4+0.32*0.05)*(38-8)\}=5.1\ [W]$
- $Q_{front,\ behind}=2*\{0.57*(0.4*0.5)*(38-8)\}=7.5\ [W]$
- $Q_{all\ wall}=23.6[W]$

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. 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. The space heat gain from product is computed by the following equation:[Reference1]

$$Q_{ch}=m.C_p.\ \Delta T/\tau \qquad (2.4)$$

Where Q_{ch} is the quantity of heat [W],m is the mass of the product [kg],$C_p$is the specific heat above freezing [kJ/kg.°C]τ: desired cooling time [seconds],ΔT is the change in product temperature [°C].

$$\Delta T = T_o - T_{ch} \tag{2.5}$$

Where T_o: is the entering product temperature [°C],T_{ch} is the chilling product temperature [°C].

The meat frozen at (- 0.5_ - 2.75°C) but in our project we store the meat at 0°C so the product load is only the above freezing load.

Applying equation (2.4) to find quantity of heat for the first product (meat):

- CP = 3.14 [kJ/kg.C°] (TABLE A-3) ,T_p=24[°C] , T_{ch} = 0 [C °] ,t=16hour From (TABLE A-14)
- Q_{ch} = 30*3.14*1000*(24-0) /(16*60*60)
- Q_{ch} =39.3[W]

Applying equation (2.4), for the second product (cucumber):

- C_P = 4.1 [kJ/kg.°C] (TABLE A-3) , T_p= [24°C] ,T_h = [5°C] ,t=16 hour
- Q_{ch} = 30*4.1*1000*(24-5) /(16*60*60)
- Q_{ch} =40.57 [W]

Applying equation (2.4), for the third product (lemon):

- C_P = 3.81 [kJ/kg.°C] (TABLE A-3) ,T_o = 24[°C] ,T_{ch} = [8°C] ,t=16 hour

- $Q_{ch} = 35*3.81*1000*(24-8) /(16*60*60)$
- $Q_{ch} = 37.04[W]$

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. [Reference1]

The heat gain resulting from air change can be determined by applying the following equation:

$$Q_{inf}= m^{\cdot}*C_p*(T_o-T_i) \tag{2.6}$$

$$Q_{inf}= \rho* V^{\cdot}_f*C_p*(T_o-T_i) \tag{2.7}$$

Where ρ:is the air density [1.25 kg/m^3] From (TABLE A-12),C_P is the specific heat of the air[1000J/kg.C°], $V^{\cdot}_f$ is the volumetric flow rate of infiltrated air[m³/s],T_o: is the outside temperature [C°],T_i : is the inside temperature [°C].

$V^{\cdot}_f$= number of air change * volume of room

Number of air change = 0.5 [times /h] (TABLE A-4)

Number of air change = 0.5 [times /h] (TABLE A-4)

- Volume of room = 0.5*0.33*0.4 =0.066 m^3.
- V'_f = 0.5*0.066 =0.033 m^3/hr.

Applying the equation (2.7), the **Q_{inf}** for the first product (meat):

- Q_{inf} = 1.25* (0.033/3600)*1000*(38- 0)
- Q_{inf} =0.44[W]

Applied the equation (2.7), the Qinf for the second product (cucumber):

- Q_{inf} = 1.25* (0.033/3600)*1000*(38- 5)
- Q_{inf} = 0.38[W].

Applied the equation (2.7), the **Q_{inf}** for the third product (lemon):

- Q_{inf} = 1.25* (0.033/3600)*1000*(38- 8)
- Q_{inf} = 0.343[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 has low specific heat.The cucumber and the lemon are stored in plastic boxes ,each chamber

Each chamber has 10boxes. Plastic box mass has 0.5 kg, each one can hold 3 kg of cucumber 3.5kg of lemon and 3 kg of meat.

The heat gain resulting from Packaging can be determined by applying the following equation:[Reference1]

$$Q_{pk} = \frac{mpk * Cpk * (Tpk - Ti)}{\tau} * 10^3 \tag{2.8}$$

Where $Q_{pk}$is the packaging heat load ,m_{pk} is the mass of packaging material [kg],$C_{pk}$is the packaging material specific heat [J/kg .C°],T_{pk} is the packaging material temperature [°C], T_i is the temperature of the refrigerant space [°C],τis the desired cooling time [seconds],$C_{pk\ plastic}$=1.6[kJ/kg. C°] (TABLE A-5), Applying equation (2.8), to find Q_{pk} for the first chamber (meat):

- $Q_{pk} = \frac{0.5*10*1.6*(24-0)}{10*3600} * 10^3$
- $Q_{pk\ plastic}$= 5.33[W]

Applying equation (2.8), to find $Q_{pk}$for the second chamber (cucumber):

- $Q_{pk} = \frac{0.5*10*1.6*(24-5)}{10*3600} * 10^3$
- $Q_{pk\ carton}$= 4.22[W]

Applying equation (2.8), to find $Q_{pk}$for the third chamber (lemon):

- $Q_{pk} = \frac{0.5*10*1.6*(24-8)}{10*3600} * 10^3$
- $Q_{pk\ carton}$= 3.55[W]

2.2.5 Miscellaneous gain

The miscellaneous load consists primary of the heat given by light and electrical motor operation in space. [Reference1].The following equation can be used to calculate the heat gain due to light

$$\mathbf{Q_{light}=P_{Lt}\,(F_uF_b)\,(CLF)_{Lt}} \tag{2.9}$$

Where:- $\mathbf{Q_{light}}$ the light heat load, $\mathbf{P_{Lt}}$ is the lamp rated power [W] ,$\mathbf{F_u}$ is the fraction of lamps that are in use ,$\mathbf{F_b}$ is the ballast factor that equal 1.2 for fluorescent lamp.$\mathbf{(CLF)_{Lt}}$ is the light cooling factor ,$\mathbf{P_{Lt}}$ =30[W], $\mathbf{(CLF)_{Lt}}$ =0.76 From (TABLE A-15). Applying equation (2.9), to find $\mathbf{Q_{light}}$:-

$$\mathbf{Q_{light}} = 30*(0.31*0.2)*(1.76) = 8.2\ [W]$$

Electrical motor heat gain are estimated according to their rated output power for motor rated power =25[W] the heat gain equal to 60[W] From (TABLE A-16)

2.2.6 Respiration gain

Fruits and vegetable are still alive after harvesting and continue to undergo changes while in storage; the more important of these changes are produced by respiration process during which oxygen from the air combines with the carbondrates in the plant tissue and results in the release of carbon dioxide and heat .The heat released is called respiration heat which is found by the following equation: - [Reference1]

Q_R=m*respiration rate **(2.10)**

Where:- **Q_R** the respiration heat load .m is the mass of product[Kg],respiration rate[W/Kg] which equal 0.02[W/Kg]for lemon at 8°C and 0.026[W/Kg]for cucumber at 5°C From (TABLE A-17).

Applying equation (2.10). to find Q_R for the second chamber (cucumber):

$Q_R = 30*0.026 = 0.88$ [W]

Applying equation (2.10). to find Q_R for the third chamber (lemon):

$Q_R = 30*0.02 = 0.6$ [W]

2.3 Total cooling load

The total cooling load is the summation of the heat gains:

$$Q_T = Q_w+Q_p+Q_{inf}+Q_{pk} + Q_{light} +Q_{motor}+\mathbf{Q_R}$$

Table 2.1 Chambers load

	Q_w[W]	Q_p[W]	Q_{inf}[W]	Q_{pk}[W]	Q_{light}[W]	Q_{motor}[W]	Q_R[W]	Q_T[W]
Chamber #1	28	39.3	0.44	5.33	8.2	60	–	141.3
Chamber #2	25	40.57	0.38	4.2	8.2	60	0.88	139.3
Chamber #3	23.6	37.04	0.343	3.55	8.2	60	0.6	133.3

- Q_T for the three champers=413.93[W]

It is common practice to add 5%-10%as factor of safety as **general rule** 10%is used:[Reference1]

- $Q_{T(1)}=141.3+(141.3*0.1)=155.5[W]$
- $Q_{T(2)}=139.3*(139.3*0.1)=153.23[W]$
- $Q_{T(3)}=133.3+(133.3*0.1)=146.63[W]$

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 to determine the average cooling load which is used as basis for equipment selection:[reference1]

$$\mathbf{Q_a}=\frac{TCL\times 24}{operatingtime} \tag{2.11}$$

Operating time = 16 hour From (TABLE A-14)

- $Q_{a(1)}=233.3[W]$
- $Q_{a(2)}=229.8[W]$
- $Q_{a(3)}=219.95[W]$

The calculated value is $Q_{a(1)}=233.3[W]$,$Q_{a(2)}=229.8[W]$,$Q_{a(3)}=219.95[W]$ but to be able to store any other kind of product we will use the following value$Q_{a(1)}=645[W]$, $Q_{a(2)}=626[W]$, $Q_{a(3)}=310[W]$.

CHAPTER THREE

REFRIGERANT SELECTION AND CYCLE CALCULATION

CHAPTER THREE

3.1 Refrigerant Selection

A refrigerant is a primary fluid .It absorbs and transports heat from a cooling space.

There are many refrigerants, they classified into five groups, one of these groups named HFC, one of its refrigerant R 134a (CH2CFC3), table (3.1) shows R134a properties and compare them with other refrigerants (R12, R22).

Chapter one presents some papers about R-134a, within the paper comparison between the refrigerant and other refrigerants.

3.1.1 Refrigerant properties

Refrigerants are used for the same purpose, but some give better results than others, so what is required of a good refrigerant? A good refrigerant should have the following properties:

- Economical properties

☐ High operating efficiency,
☐ low cost
☐ Availability.

- Thermodynamic properties

- Low boiling point,
- High critical point temperature,
- High latent heat of vaporization,
- Low specific heat of liquid,
- Mixes well with oil.

. Environment friendly

- Having zero ozone depletion potential(ODP),
- Having low global warming potential(GWP),
- Non corrosive to metal,
- Easily detectable in case of leakage,
- Non flammable and non explosive, and
- Non toxic.

Table (3.1): physical properties of refrigerants(the properties @1 atm)

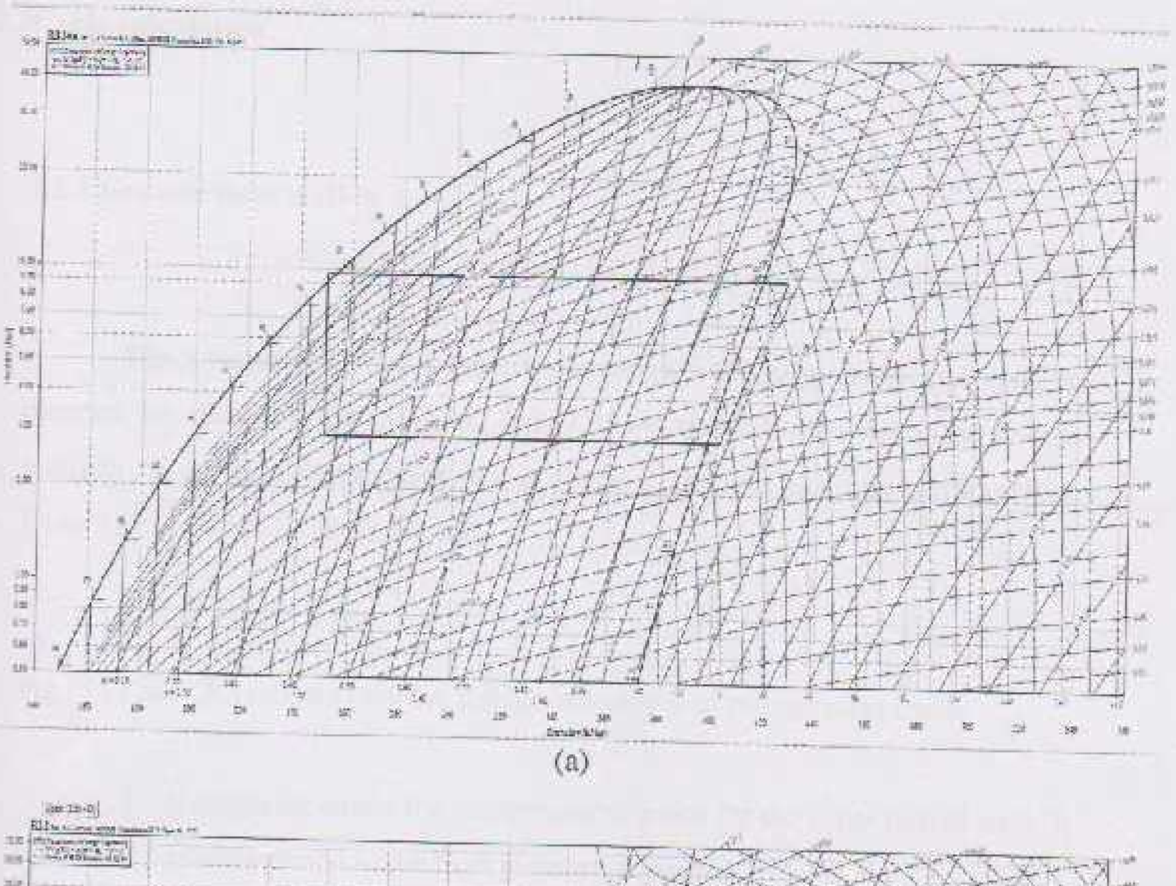
Properties	R 12	R 134a	R 22
Molecular formula	CCl_2F_2	$C_2H_2F_4$	$CHClF_2$
boiling point(°C)	29.8	-26.3	-40.8
Melting point (°C)	----	-101	-157
critical point (°C)	112	101.1	96.2
toxic	Non	Non	Non
flammable	Non	Non	Non
ODP	1.0	0.0	0.05
GWP	8100	1,300	1700

From table (3.1),boiling and melting temperature for R-134a is higher than R-12 and R-22 .All refrigerants are nontoxic, non flammable and low ODP "ozone

So R134a is best suited for refrigeration system because R-134a is morefriendly than R-12 and R-22.

3.1.2Relationship between Refrigerant and compressor

R-134a is selected for refrigeration system. We have to remember that the compression ratio of the R-134a system is higher than that of the R12 system even though both have the same condensing and evaporating temperature, because either an increase in head pressure or decrease in suction pressure will cause higher compression ratios. Fig(3.1) shows the properties of the R-134a and R-12 on P-h diagrams.



(a)

(b)

Figure (3.1): (a) P-h diagram for the R-134a (b) P-h diagram for the R-12

3.2Cycle calculation

3.2.1 General information

The schematic of such a system and corresponding operating cycle on P-h diagram are shown in Fig (3.1) and Fig (3.2) .As shown in the figure the system consists of a single compressor and a single condenser but three evaporators. Evaporators –I, II. III. [Reference 2]

Fig (3.2) and (3.3) shows how the refrigerant moves in refrigeration cycle:

1. Refrigerant enters the compressor at point 1at the super heated state. It is compressed to the high pressure at point 2.
2. Hot gas enters the condenser, in which it is de-superheated and sub cooled at point 3.
3. After the condensing process, the liquid refrigerant flows through a throttling device (thermostatic expansion valve).
4. Refrigerant is divided into three evaporators,which enters evaporators at low temperature and low pressure at points (4, 5, 6) The liquid refrigerant is vaporized at points(7, 8, 9) in the evaporator. The vapor then flows to the inlet of the compressor at point 1 and completes the cycle.

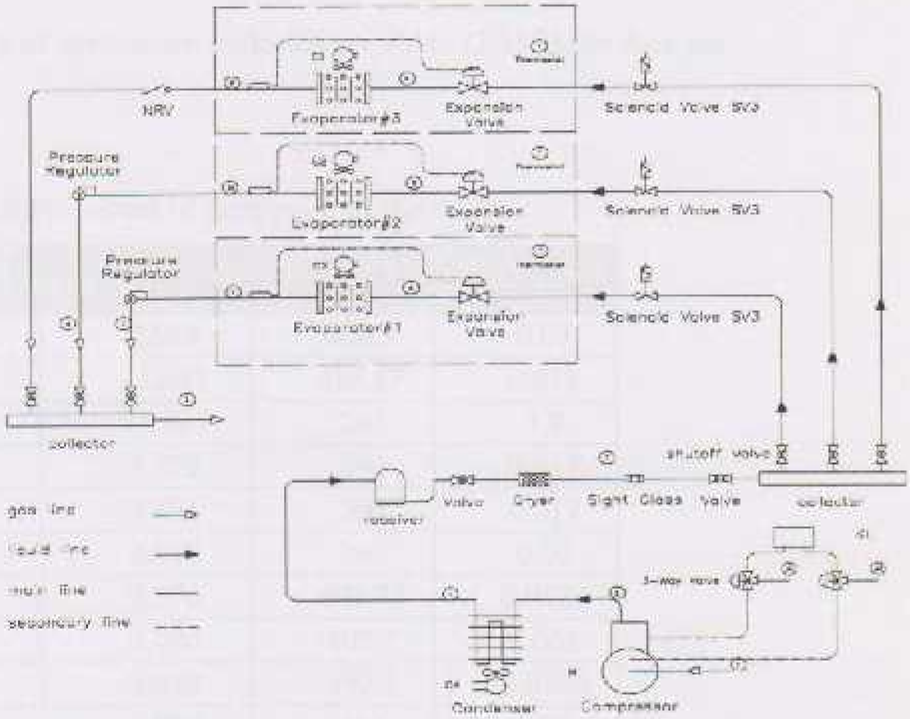


Figure (3.2) Schematic diagram for a multiple evaporator system

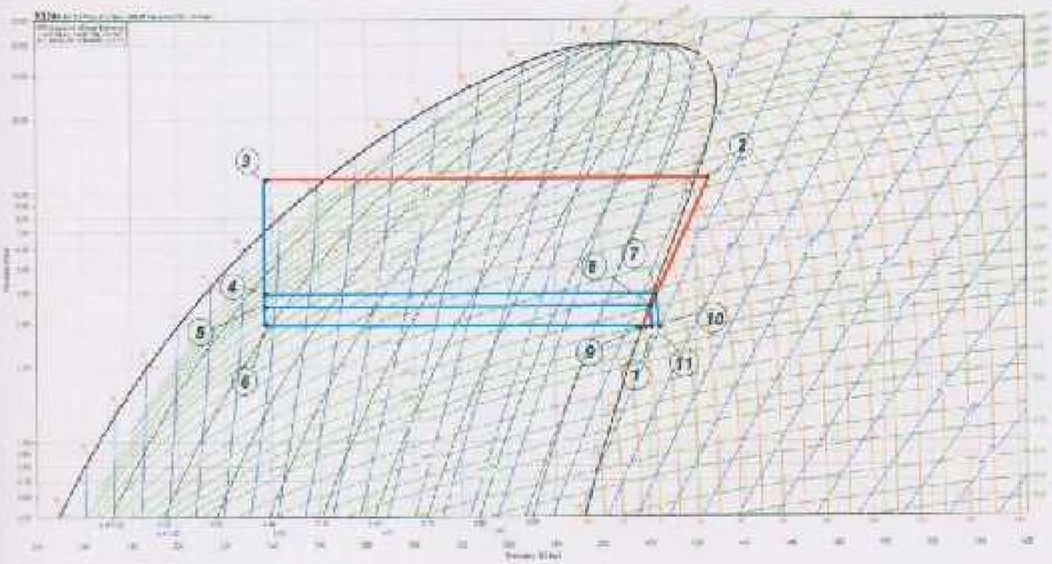


Figure (3-3). P-h diagram of a multiple evaporator system

From figure (3.3),data of system are collected on Table (3.3).These data are used to cycle calculation.

Table (3.2)Properties of each point on the cycle

#	T(°C)	P(bar)	h(kJ/kg)	υ(m³/kg)
1	3	2.928	400.3	0.07
2	52	11.597	428.87	0.018
3	42.5	11.597	260	7.9
4	8	3.876	260	0.015
5	5	3.496	260	0.017
6	0	2.928	260	0.02
7	8	3.876	401.77	0.0524
8	5	3.496	400.7	0.058
9	0	2.928	397.2	0.0866
10	8	2.928	404	0.07
11	5	2.928	401.5	0.07

3.2.2 Mass flow rate

In this cycle we have three evaporators; each one has amass of refrigerant that flows through it .To find the mass flow rate we use the following equation: [Reference 3]

$$m'_1 = \frac{Qe}{qe} \tag{3.1}$$

Where: $m'_1$is the mass flow rate in the evaporator, [kg/s] ,Q_e : refrigeration capacity, [kW] , q_e= refrigeration effect[reference 3].

$$q_e = \Delta h \tag{3.2}$$

WhereΔh is the difference between the enthalpy of saturated vapor leaving evaporator and the enthalpy of refrigerant entering evaporator: [k J/kg] .. Applying equation (3.2), the q_e for first chamber is:

- $q_e = h_9 - h_6 = 397.2 - 260 = 137.2$[kJ /kg]

Applying equation (3.1) the mass flow rate is :

- $m'_1 = \frac{0.6452}{137.2}$
- $m'_1 = 0.0047$[kg/s]

Applying equations (3.1) and (3.2), the q_e and m' for second chamber are :

- $q_e = h_8 - h_5 = 400.7 - 260 = 140.7$[kJ /kg]

- $m'_2 = \frac{0.626}{158.1}$
- $m'_2 = 0.00396$ [kg/s]

Applying equations (3.1) and (3.2), the q_e and $m^{\cdot}$ for third chamber are :

- $q_e = h_7 - h_4 = 401.77 - 260 = 141.77$[kJ /kg]
- $m'_3 = \frac{0.310}{159.17}$
- $m'_3 = 0.00195$[kg/s].

Finally, the total mass flow rate is:

- $m'_T = m'_1 + m'_2 + m'_3 = 0.0106$ [kg/s]

3.2.3 Condenser load

After calculating thetotal mass flow, condenser load is found by the following equation:[Reference 3]

$$\mathbf{Q_c = m'_T (h_2 - h_3)} \qquad \textbf{(3.3)}$$

Where:$Q_c$is the condenser load [kW], h_2 is the enthalpy of the hot gas discharged from the compressor [kJ /kg],h_3 is the enthalpy of the liquid discharged from the condenser [kJ /kg], applying equation (3.3) and table (3.1) the condenser load is :

- $Q_c = 0.0111\ (428.87 - 260)$

- $Q_c = 1.79$ [kW]

3.2.4 Power of compressors

To find the compressor power ,we use the following equation:[Reference 3]

$$\mathbf{P=m'_1 *(h_2-h_1)} \tag{3.4}$$

Where P: is the compressor power [kW],h_2: is the enthalpy of the hot gas discharged from the compressor [kJ /kg],h_1: is the enthalpy of the gas discharged from the evaporator [kJ /kg].

Enthalpy at point 1(inlet to compressor) is obtained by applying equation (3.5) massand energy balance to the mixing of three refrigerant streams.[reference15]

$$\mathbf{h1 = \frac{(m.1 * h9) + (m.2 * h11) + (m.3 * h10)}{m.1 + m.2 + m.3}} \tag{3.5}$$

- $h1 = \frac{(0.006*379.5)+(0.0055*401)+(0.0019*404)}{0.006+0.0055+0.0019} = 392.09 kj/kg$

Applied equation (3.4) the power of compressor is :

- $P = 0.0106(428.87 - 392.09)$
- $P = 0.389$[kW]

3.2.5 Coefficient of performance

$$COP = \frac{Qetotal}{P} \tag{3.6}$$

Where COP is the Coefficient of performance of the cycle, P is the compressor power [kW],Q_e total is the total cooling capacity [kW].[Reference1]

- $COP = \frac{1581.59}{309.4}$
- $COP = 4$

To ensure the accuracy of the calculation a software programme (Cool Pack) was used ,a single stage compression cycle at the lowest evaporating temperature and at the same condensing temperature give a cooling capacity(1.6 [kW]) ,compressor power (0.4[kW]), condensation capacity (2 [kW]) , refrigerant mass flow (0.012[kg/s]) ,COP(3.9) (TABLE A-6) and the Carnot efficiency(0.65) (TABLE A-7) close to the calculated value .

CHAPTER FOUR

COMPONENTS DESIGN AND SELECTION

CHAPTER FOUR

COMOPNENTS DESIGN AND SELECTION

4.1 Introduction

Mechanical components are required to work a multiple evaporatorsrefrigeration system. In this part, we discuss the main and auxiliary components of a system and make a calculation and selection of these components.

4.2 Cycle Components

The main components of the multiple evaporators refrigeration system are :

- Compressor.
- Evaporator.
- Condenser.
- Connecting pipes.
- Throttling device.

4.2.1 Compressor

The refrigeration compressor (vapor pump) is the heart of the refrigeration system. It moves the vapor to high pressure side of the system. It maintains the low side pressure at which the refrigerant evaporates, and the high side pressure at which it condenses, it supplies the pressure difference necessary to keep the system refrigerant flowing through the system. [Reference 4]

There are many types of compressors; one of them is called hermetic compressor.In this compressor, a motor and a compressor are connected together.The reason for choosing this compressor is that:

- It has a long life, so the life of the equipment should be even longer.

4.2.2 Evaporator

An evaporator isa heat exchanger, in which a volatile liquid is vaporized for the purpose of removing heat from a refrigerated space , evaporators are manufactured in wide variety of types, shapes, sizes, designs and may be classified in different ways according to the contraction liquid feed, operating condition, method of air (or liquid) circulation, control, and application . [Reference5]

There are many types of evaporators;a finned evaporator is used in this project, because of:

- Finned evaporator has fins, and fins increase the efficiency of evaporator.
- Finned evaporator will occupy less space than other types of evaporator.

- Air cooling application is used in this project, finned evaporator is the best performance at (0, 5, 8) C^0.

4.2.3Condenser

The condenser is heat exchanger. Heat from the hot refrigerant passes through the walls of the condenser to the condensing medium. The action in the condenser is just the opposite of the action in the evaporator. an air cooled condenser "forced convection ". Which employs air as the condensing medium by action of a fan is used because of. [Reference5]

- Air quantity circulation over the condenser is higher than the free convention condenser
- Has fins, that are widely spaced so that little or no resistance is offered to the free circulation of air.

4.2.4Connecting pipes

The piping system provides passage for the refrigerant to the evaporator, the compressor, the condenser and the other parts .It also provides the way for oil to drain back to the compressor. Tubing, piping, and fitting are used in numerous applications. Copper tubing is suitable with R-134a; copper tubing is used with domestic and small systems. [Reference6]

4.2.5 Throttling devices

The basic functions of an expansion device used in refrigeration systems areto reduce pressure from condenser pressure to evaporator pressure, and regulate the refrigerant flow from the high-pressure liquid line into theevaporator at a rate equal to the evaporation rate in the evaporator. [Reference7]

There are basically seven types of refrigerant expansion devices. "Thermostatic Expansion Valve (TEV) ", used before the evaporator.

4.3 Components calculation

4.3.1 Compressor

The actual volumetric flow rate for the compressor must be calculated to determine the type of compressor. To find this value;volumetric efficiency and the theoretical volume flow rate of the compressor must be calculate.

4.3.1.1The volumetric efficiency

First, the volumetric efficiency must be determined using equation (4.1) is [reference 8] :

$$\eta_v=\eta_c*\eta_h \qquad (4.1)$$

Where: η_v is the volumetric efficiency,η_c is the volumetric efficiency due to clearance volume in compressor,η_h is the volumetric efficiency due to heating occurring in compressor. But the volumetric efficiency due to the clearance volume in compressor is calculated by equation (4.2), [Reference 8]:

$$\eta_c = 1 - c[(\frac{PH}{PL})^{1/n} - 1] \tag{4.2}$$

Where c: is the 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 8], n is the exponential coefficient of expansion for refrigerant, n= 1. [Reference8], PH is the high pressure of the cycle; PL is the low pressure of the cycle. Applied equation (4.2) the volumetric efficiency is:

- $\eta_c = 1 - 0.02[(\frac{11.597}{2.928})^{1/1} - 1] = 94\%$

Equation (4.3) is used to find the volumetric efficiency due to the heating in compressor [Reference 9]:

$$\eta_h = \frac{Tevap}{Tcond} \tag{4.3}$$

Where T_{evap} : is the evaporator temperature [°K] , T_{cond} is the condenser temperature [°K] :

- $\eta_h = \frac{273}{318} = 85.8\%$

Applying equation (4.1), the volumetric efficiency is :

- $\eta_v = 94\%* 85.8\% = 80.69\%$

4.3.1.2 The theoretical volume flow rate

Second, The theoretical volume flow rate ($V^{\cdot}$) of the compressor can be calculated byusing equation (4.4),[Reference 10]:

$$V_{theo}= m^{\cdot}*\upsilon \qquad (4.4)$$

Where: - $V^{\cdot}_{theo}$ is the theoretical volume flow rate of the compressor [m^3/s], $m^{\cdot}$ is the mass flow rate of refrigerant [kg/s], υ is the specific volume at the inlet of compressor [m^3/s] which =0.07 from the PH diagram. Applied equation (4.4):

- $V^{\cdot}_{theo}= 0.0106*0.07 = 7.42*10^{-4}$ [m^3/s]

To determine the actual volume flow rate by the equation (4.5), [Reference 9]:

$$V^{\cdot}_{act} = \frac{V.theo}{\eta v} \qquad (4.5)$$

Where $V^{\cdot}_{act}$: is the actual volumetric flow rate[m^3/s], applied equation (4.5) the actual volumetric flow rate is :

- $V^{\cdot}_{act}=\frac{7.42*10-4}{0.8069} = 0.919*10^{-3}[m^3/s]$

4.3.1.3 The compressor selection

The main consideration to select the compressor is the actual volumetric flow rate, so we chose a compressor that satisfies it, Danfoss compressor is selected for this project from Danfoss catalog, we select a compressor typeSC12/12G with code number104G8280.Which has displacement of 26.3 cm3 per revolution and 2900 RPM .

The actual flow rate for the compressor can be calculated as the following [Reference 9]:

$$V'_{act} = V'_{theo} * \eta_v \tag{4.6}$$

Where: -V'_{act} is the actual volumetric flow rate for the compressor $[m^3/s]$, V'_{theo} is the theoretical volumetric flow rate for the compressor $[m^3/s]$,η_v: is the volumetric efficiency.

- $V'_{theo} = 26.3 * 10^{-6} * (2900/60) = 1.27 * 10^{-3}\ [m^3/s]$
- $V'_{act} = 1.27 * 10^{-3} * 0.8069 = 1.026 * 10^{-3}\ [m^3/s]$

4.3.2 Evaporator

An evaporator design was manufactured. The evaporator was made from a Aluminum tube; the dimension of evaporator is shown in chapter (6), page (24-34),So:

- S_n: Transverse tube spacing [m]

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

- S_p: Longitudinal tube spacing [m]

$$S_p = \frac{\text{evaporator width}}{\text{number of column}} = \frac{0.2}{7} = 0.0286[\text{m}]$$

- $fin\ length(L_f) = (S_n - D_0)$

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

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

$$W_f = (0.0286 - 0.01) = 0.0186[\text{m}]$$

- number of fins in row =70 fins
- fin pitch $=P_f=\frac{40}{70}=0.571$ [cm]
- $\text{Finethickness} = t_f = 0.03[\text{cm}]$
- $\text{bare tube thickness} = t_b = P_f - t_f$

$$t_b = 0.571 - 0.03 = 0.54[\text{cm}]$$

Design the evaporator required many calculations such as fluid mechanical calculation, thermal calculation and area calculation. The sequence of design starts 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 [Reference7].

$$Q= h_o *A*(T_w-T_\infty) \qquad (4.7)$$

Where Q: heat transfer through the evaporator [W], h_o: external convection heat transfer coefficient [$W/m^2\,^\circ C$], A: surface area of heat transfer [m^2], T_w: outer evaporator wall temperature [°C], T_∞: free air temperature.

To determine the external convection heat transfer coefficient can be expressed via the following equations,In order to calculate the total The heat that transferred through the evaporator was determined in chapter three and its value was $Q_c= 645$ [W], so:

[Reference7]

$$h_o= \frac{Nu*k}{D} \qquad (4.8)$$

Where Nu: Nusselts number ,k: thermal conductivity of air at entrance of evaporator [W/m °C], D: outer diameter of evaporator [m]. The Nusselts number can be calculated by the equation [Reference7]:

$$Nu=C(Re)^N\,Pr^{1/3} \qquad (4.9)$$

Where Re: Reynolds number, Pr: Prandtl number of air at filmtemperature,(C,N):constants can be obtained from table (A-11) according the following considerations.

- $S_p/d = 28.6/10 = 2.86$.
- $S_n/d = 25/10 = 2.5$.

From In line arrangement tube banks table(C =0.366, N=0.595). [Reference7]

$$\mathrm{Re} = \frac{\rho \cdot Vmax \cdot D}{\mu} \qquad \textbf{(4.10)}$$

Where ρ: density of air at film temperature [kg/m^3], V_{max}: maximum velocity of air between the evaporator tubes [m/s], D :outer diameter of the evaporator tubes [m], μ :dynamic viscosity of air at film temperature [Pa.s].

For flows normal to in -line tube banks the maximum flow velocity can be calculated as follows.[Reference7]

$$V_{max} = V_\infty \frac{Sn}{Sn-D} \qquad \textbf{(4.11)}$$

Where V_∞ is the free air velocity entering the evaporator [m/s], can be calculated by the following equation:

$$V_\infty = \frac{V'}{A} \qquad \textbf{(4.12)}$$

Where v. : flow rate of air through the evaporator [m3/s],V. =50[cfm]= 0.02359 [m3/s] from fan manufacturer company, A: cross sectional area of evaporator [m^2].

- $A = 0.4*0.05 = 0.02\ [m^2]$
- $V\infty = \frac{0.02359}{0.02} = 1.18$ m/s
- $V_{max} = 1.048\,\frac{0.025}{0.025-0.01}. = 1.966[m/s]$

4.3.2.1 First evaporator

The properties of air are evaluated at the film temperature, which at entrance to the tube bank is [Reference7]:

$$\mathbf{Tf} = \frac{\mathbf{Tw + T\infty}}{\mathbf{2}} \qquad \textbf{(4.13)}$$

Where T_f : film temperature[°C], T_w : wall surface temperature[°C], assume that it equals the refrigerant temperature, T_∞:free air temperature[°C].

- $T\,f = \frac{-10+0}{2} = -5$ [°C] =268 [°K]

Then from (table A-12) :

- ρ =1.292 [kg/m^3]
- μ= 1.725 *10^{-5} [kg/m.s]
- K =0.0243 [W /m.°C]
- Pr =0.715
- Re $=\frac{1.295*1.967*0.01}{1.725*10^{5}}$=1476.7
- Nu=0.366*(1476.7)$^{0.595}$ * (0.715)$^{1/3}$ = 25.2
- h_o= $\frac{25.2*0.0243}{10*10^{-3}}$ = 61.24 [W/m^2°C]

Heat transfer from one element (one fin and one bare tube) the following equation is used [Reference 11]:

$$Q_{total} = Q_{fin\ act} + Q_{original} \tag{4.14}$$

Where Q_{total} : the total heat transfer from the element[W], $Q_{fin\ act}$: actual heat transfer rate per fin[W], $Q_{original}$: heat transfer rate from tube without fin[W], $Q_{original}$ can be calculated by the equation:

$$Q_{original} = h_o * A_{original} * (T_w - T_\infty) \tag{4.15}$$

Whereh_o: external convection heat transfer coefficient [W/m2°C], $A_{original}$:the outer surface area of bare tube [m^2], T_w: outer evaporator wall temperature [°C], T_∞: free air temperature:

- $A_{original}$,= π*D*L =π *10*10^{-3}*5.4*10^{-3}=1.696 *10^{-4} [m^2]

- $Q_{original}$ =61.24*1.696*10^{-4} *(0- -10) =0.104 [W]

Q_{fin} can be calculated by the equation:

$Q_{fin} = h_o A_{fin}(T_w - T_\infty)$ **(4.16)**

WhereQ_{fin}:theoretical heat transfer rate per fin [W], A_{fin}`:surface area for fin [m2]:

- $A_{fin} = 2(S_n * S_P - A_{pip}) = 2[0.025*0.0286-\frac{\pi}{4}(0.01)^2] = 12.73*10^{-4}$ [m^2]
- Q_{fin} = 61.24*12.73*10^{-4}*(0 - - 10) = 0.778 [W]

Now, Fin efficiency iscalculated by the following equations[Reference 2]:

$$\eta_f = \frac{\tanh(m\dot{L}_f)}{m\dot{L}_f} \tag{4.17}$$

$$\dot{L}_f = \left(\frac{L_f}{2}\right)\left[1 + 0.35\ln\frac{\left(\frac{D_0}{2} + \frac{L_f}{2}\right)}{\frac{D_0}{2}}\right] \tag{4.18}$$

- $\dot{L}_f = \left(\frac{0.015}{2}\right)\left[1 + 0.35\ln\frac{\left(\frac{0.01}{2} + \frac{0.015}{2}\right)}{\frac{0.01}{2}}\right] = 0.01[m]$

$$m = \sqrt{\frac{h * P}{kA}} \tag{4.19}$$

Where h:external convection heat transfer coefficient [W/m^2.°C], k:thermal conductivity of aluminum fin ,[W/m. °C], P :perimeter of the fin[m], A :surface area for convection of fin[m^2]:

- $P = 2*t+2*L = (2*0.3*10^{\wedge -3})+(2*0.02) = 0.0406$ [m]
- $A = t*L = 0.3*10^{\wedge -3}*0.02 = 6*10^{\wedge -6}$[m2].

Applying equations (4.17) and (4.19), m,$\eta_f$equal:

$$m = \sqrt{\frac{61.24 * 0.0406}{202 \times 6 \times 10^{\wedge -6}}} = 45.25$$

$$\eta_f = \frac{\tanh(45.25 * 0.01)}{45.25 * 0.01} = 0.94$$

So, The heat transfer flow from the fin is:

$$Q_{\text{fin net}} = Q_{\text{fin}} * \eta_f \qquad (4.20)$$

- $Q_{\text{fin net}} = 0.776 * 0.94 = 0.729$ [W]

Now the total heat transfer from the element is:

- $Q_{\text{total}} = 0.729 + 0.104 = 0.833$ [W]

Now the number of elements needed to perform the evaporator load can be determined by dividing the total heat transfer through the evaporator by the element total heat transfer, by using the following equation:

$$n = \frac{Qe}{q\ total} \qquad \textbf{(4.21)}$$

- $n = \frac{645.2}{0.833} = 774.5$ elements

Number of elements in available evaporator = (N*R), where N: number of elements in rows, R: number of rows, number of elements in available evaporator =70* 14= 980elements .

4.3.2.2SecondEvaporator

Applying equation (4.13) the t film is:

- $T\,f = \frac{-5+5}{2} = 0$ [°C] =273 [°K]

Then from table A-9 :

- ρ =1.28 [kg/m^3]
- μ= 1.7 *10^{-5} [kg/m.s]
- K =0.024 [W /m.°C]
- Pr =0.715

- $Re = \frac{1.28*1.967*0.01}{1.7*10^{6}}$=1475
- $Nu=0.366*(1475)^{0.595}*(0.715)^{1/3}=25.17$
- $h_o=\frac{25.17*0.024}{10*10^{-3}}=60.4$ [W/m²°C]

Applyingequation (4.15) :

- $A_{original,}=\pi*D*L=\pi*10*10^{-3}*5.4*10^{-3}=1.696*10^{-4}$ [m²]
- $Q_{original}=60.4*1.696*10^{-4}*(5--5)=0.1024$ [W]

Applying equation (4.16) :

- $A_{fin}=2(S_n*S_P-A_{pip})=2[0.025*0.0286-\frac{\pi}{4}(0.01)^2]=12.73*10^{-4}$ [m²]
- $Q_{fin}=60.6*12.73*10^{-4}*(5--5)=0.769$ [W]

Applying equation (4.18) :

- $\dot{L}_f=\left(\frac{0.015}{2}\right)\left[1+0.35\ln\frac{\left(\frac{0.01}{2}+\frac{0.015}{2}\right)}{\frac{0.01}{2}}\right]=0.01[m]$

Applying equations (4.17) and (4.19), m,η_f equal :

- $P=2*t+2*L=(2*0.3*10^{\wedge-3})+(2*0.02)=0.0406$ [m]
- $A=t*L=0.3*10^{\wedge-3}*0.02=6*10^{\wedge-5}$[m2].

- $m = \sqrt{\frac{60.6*0.0406}{202\times6\times10^{\wedge-6}}} = 45.1$
- $\eta_f = \frac{\tanh(45.1*0.01)}{45.1*0.01} = 0.94$

Applying equation (4.20) :

- $Q_{\text{fin act}} = 0.772 *0.94 = 0.723$ [W]

Applying equation (5.8) :

- $Q_{\text{total}} = 0.723 + 0.102 = 0.825$ [W]

Applying equation (4.21) :

- $n = \frac{626.39}{0.825} =$ **759** elements

Number of elements in available evaporator =70* 14= **980** elements.

4.3.2.3ThirdEvaporator

Applying equation (4.13) the t film is :

- $T f = \frac{3+8}{2} = 5.5$ [°C] =278.5 [°K]

Then from table A-9:

- ρ =1.22 [kg/m^3]
- μ= 1.85 *10^{-5} [kg/m.s]
- K =0.0261 [W /m.°C]
- Pr =0.715
- Re = $\frac{1.22*1.967*0.01}{1.85*10^{-5}}$=1303.1
- Nu=0.366*(1303.1)$^{0.595}$ * (0.715)$^{1/3}$ =23.35
- h$_o$=$\frac{23.35*0.0261}{10*10^{-3}}$ = 60.95 [W/m^2°C]

Applyingequation (4.15) :

- A$_{original,}$= π*D*L =π *10*10^{-3}*5.4*10^{-3}=1.696 *10^{-4} [m^2]
- Q $_{original}$ =60.95*1.696*10^{-4} *(8- 3) =0.052 [W]

Applying equation (4.16) :

- A$_{fin}$ =2(S$_n$ *S$_P$– A$_{pip}$) = 2[0.025*0.0286-$\frac{\pi}{4}$ (0.01)^2] =12.73*10^{-4} [m^2]
- Q $_{fin}$ = 60.95*12.73*10^{-4}*(8 - 3) = 0.388 [W]

Applying equation (4.18) :

- $\hat{L}_f = \left(\frac{0.015}{2}\right)\left[1 + 0.35\ln\frac{\left(\frac{0.01}{2}+\frac{0.015}{2}\right)}{\frac{0.01}{2}}\right] = 0.01[m]$

Applying equations (4.11) and (4.13), m,$\eta_f$equal:

- $P= 2*t+2*L =(2*0.3*10^{\wedge-3})+(2*0.02) =0.0406$ [m]
- $A = t*L =0.3*10^{\wedge-3}*0.02 = 6*10^{\wedge-6}$[m2].
- $m = \sqrt{\frac{60.95*0.0406}{202\times 6\times 10^{\wedge-6}}} = 45.2$
- $\eta_f = \frac{\tanh(45.2*0.01)}{45.2*0.01} = 0.94$

Applying equation (4.20) :

- $Q_{\text{fin act}} = 0.388 * 0.94 = 0.365$ [W]

Applying equation (4.14) :

- $Q_{\text{total}} = 0.365 + 0.052 = 0.417$ [W]

Applying equation (4.21) :

- n= $\frac{310}{0.417}$ =743elements

Number of elements in available evaporator =70*14= 980 elements.

4.3.3Condenser

A condenser design was manufactured. The condenser was made from a (10mm) diameter, coppertube, the following is geometrical data, condenser:

- condenser length L_c =40 [cm]
- condenser height H_c =40 [cm]
- condenser width W_c =10[cm]

S_n: Transverse tube spacing [m]

- $S_n = \frac{\text{evaporatorheight}}{\text{numberofrows}} = \frac{0.4}{10}$ =0.04[m]

S_p: Longitudinal tube spacing [m]

- $S_p = \frac{\text{evaporator width}}{\text{numberofcolumn}} = \frac{0.1}{3} = 0.033[\text{m}]$

$\text{finlength} = L_f = (S_n - D_0)$

- $L_f = (0.04 - 0.01) = 0.03[\text{m}]$

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

- $W_f = (0.033 - 0.01) = 0.023[\text{m}]$

- number of fins in a row =190 fins
- fin pitch $=P_f=\frac{60}{190}=$ 0.316 [cm]
- $\text{Finthickness} = t_f = 0.02[\text{cm}]$

$$\text{baretubethickness} = t_b = P_f - t_f$$

- $t_b = 0.315 - 0.02 = 0.295[\text{cm}]$

Condenser has two regions; first region is (desuperheating) region then is (mixture) region, now desuperheating must be calculated:

4.3.3.1 Desuperheating region

Applying equation (4.22) the $Q_{desuper}$ is [Reference7]:

$$Q_{desuper} = \quad *(h_{in} - h_{sat,vap}) \qquad (4.22)$$

Where $Q_{desuper}$: condenser load from desuperheating region [W], : mass flow rate in the condenser and calculated in chapter three , [kg/s], h_{in} :enthalpy of refrigerant entering the condenser, [kJ /kg], $h_{sat\,vap}$: enthalpy of saturated vapor leaving the desuperheating region in condenser [kJ /kg]so:

- $Q_{desuper} == 0.0106(428.87 - 417.7) = 118.4[W]$

From (table A-11) value of C, N are determined by S_p/D and S_r/D, so the value $S_p/D = 19/10 = 3.3$, $S_r/D = 26/10 = 2.6$:

- C =0.446
- N=0.586

For flows normal to staggered arrangement the maximum flow velocity can be calculated as the follows[Reference7]:

$$V_{max} = \frac{V\infty\left(\frac{Sn}{2}\right)}{\sqrt{\left[\left(\frac{Sn}{2}\right)^2 + Spl^2\right]} - D} \qquad (4.23)$$

Applying equation (4.12), the free air velocity is determined by a known value. V = 700 [m^3/h]=0.19444 [m^3/s]from fan manufacturer company.

- A = 0.4*0.4 =0.16 [m^2]
- $V_\infty = \frac{0.19444}{0.16} = 1.215$m/s

Applying equation (4.23) the max velocity is :

$$V_{max} = \frac{1.215\left(\frac{0.04}{2}\right)}{\sqrt{\left[\left(\frac{0.04}{2}\right)^2 + 0.033^2\right]} - 0.01} = 0.85 \text{ [m/s]}$$

The properties of air are evaluated at the film temperature, which can calculatedby equation (4.13), applying this equation:

- Tav. w = $\frac{45+52}{2}$ = 48.5 [°C]
- Tf = $\frac{48.5+38}{2}$ =43.25 [°C] =316.25 [°K]

Using table A-12, different properties can be determined:

- $\rho = 1.1176$ [kg/m^3]
- $\mu = 1.85 * 10^{-5}$ [kg/m.s]
- $K = 0.0275$ [W /m.°C]
- $Pr = 0.704$

Applying equation (4.10), Renaldo number is :

- $Re = \frac{1.1176 * 0.85 * 0.01}{1.85 * 10^{5}} = 513.49$

Applying equation (4.9), Nusselts numberis :

- $Nu = 0.458(513.49)^{0.598} * (0.704)^{1/3} = 16.7$

Applying equation (4.8), external convection heat transfer coefficientis :

- $h_o = \frac{16.7 * 0.0275}{10 * 10^{-3}} = 45.9$ [W/m^2°C]

The heat transfer coefficient that would be obtained if there are 10 rows of tubes in the direction of the flow,because there are only 3rows ,this value must be multiplied by the factor 0.83 ,as determined from (table A-10) [Reference 7]

- $h_o = 42.3 * 0.83 = 35.11$[W/m^2°C]

In order to calculate the total heat transfer from one element (one fin and one bare tube) the equation (4.15) is used [Reference 11]:

- $A_{original} = \pi * D * L = \pi * 10 * 10^{-3} * 4 * 10^{-3} = 125.7 * 10^{-6}$ [m^2]
- $Q_{original} = 45.99 * 125.7 * 10^{-6} * (48.5-38) = 0.0607$ [W]

Q_{fin} can be calculated by the equation (4.16):

- $A_{fin} = 2(S_n * S_P - A_{pipe}) = 2[0.04 * 0.033 - \frac{\pi}{4}(0.01)^2] = 2.483 * 10^{-3}$ [m^2]
- $Q_{fin} = 45.99 * 2.483 * 10^{-3}\ (48.5-38) = 1.19$ [W]

Fin efficiencycan be calculated by equation (4.18):

- $L_f = \left(\frac{0.015}{2}\right)\left[1 + 0.35 \ln \frac{\left(\frac{0.01}{2} + \frac{0.015}{2}\right)}{\frac{0.01}{2}}\right] = 0.01[\text{m}]$

- perimeter of the fin = $2 * t + 2 * L = 2 * 0.2 * 10^{-3} + 2 * 0.033 = 0.0664$[m]
- Area of fin = $t * L = 0.2 * 10^{-3} * 0.033 = 6.6 * 10^{-6}$[m^2]

Applying equations (4.17) and (4.19):

- $m = \sqrt{\frac{29.17 * 0.0664}{202 \times 6.6 \times 10^{-6}}} = 38.12$
- $\eta_f = \frac{\tanh(38.12 * 0.01)}{38.12 * 0.01} = 0.95$

So the actual heat transfer flow through the fin is

- $Q_{fin\,act} = 1.19 * 0.95 = 1.139$[W]

Now the total heat transfer from the element is:

- $Q_{total} = 1.139 + 0.0607 = 1.199$ [W]

Now the number of elements needed to perform the condenser load to desuperheating process can be determined by the following equation:

$$n = \frac{Qdesuper}{Qtotal} \tag{4.24}$$

- $n = \frac{118.4}{1.199} = 98.75$ elements

4.3.3.2 Mixture region

After finishing the desuperheating region, and finding the element number, the mixture region will be calculated and find the number of element, so applying equation (4.25) , Q_{mix} is determined :

$$\mathbf{Q_{mix} = m^{\cdot} (h_{sat.vap} - h_{sat.\,liq.})} \tag{4.25}$$

Where Q_{mix} : condenser load from mixture region [W], : mass flow rate in the condenser, [kg/s], $h_{sat.vap.}$: enthalpy of saturated vapor entering the mixture region in condenser, [kJ /kg], $h_{sat.liq.}$: enthalpy of saturated liquid leaving the mixture region incondenser [kJ /kg].

- Q_{mix} =0.0106(417.7-260) = 1671.6[W]

The properties of air are evaluated at the film temperature .It can calculated by equation (4.13), applied this equation :

- $Tf = \frac{45+38}{2}$ =41.5 [°C] =314.5 [°K]

Then from table A-9

- ρ =1.13 [kg/m^3]
- μ= 1.848 $*10^{-5}$ [kg/m.s]
- K =0.0275 [W /m.°C]
- Pr =0.704

Applying equation (4.10), Renaldo number is :

- $Re = \frac{1.13*0.85*0.01}{1.848*10^{5}}$ =519.75

Applyingequation (4.9), Nusselts numberis :

- $Nu=0.458(519.75)^{0.598} * (0.704)^{1/3}$ =16.9

Applying equation (4.8), external convection heat transfer coefficientis :

- $h_o = \frac{16.9*0.0275}{10*10^{-3}}$ = 46.5 [W/m^2°C]

This is the heat transfer coefficient that would be obtained if there are 10 rows of tubes in the direction of the flow .Because there are only 3rows ,this value must be multiplied by the factor 0.83 ,as determined from table A-13[reference 8],so:

- $h_o = 46.5 * 0.83 = 38.6 [W/m^{2\circ}C]$

In order to calculate the total heat transfer from one element (one fin and one bare tube) the equation (4.15) is used [Reference 11]:

- $A_{original} = \pi * D * L = \pi * 10 * 10^{-3} * 4 * 10^{-3} = 125.48 * 10^{-6}\ [m^2]$
- $Q_{original} = 38.6 * 125.48 * 10^{-6} * (45-38) = 0.0339\ [W]$

Q_{fin} can be calculated by the equation (4.16):

- $A_{fin} = 2(S_n * S_P - A_{pip}) = 2[0.05 * 0.033 - \frac{\pi}{4}(0.01)^2] = 3.143 * 10^{-3}\ [m^2]$
- $Q_{fin} = 38.6 * 3.143 * 10^{-3}(45-38) = 0.849\ [W]$

Fin efficiencycan be calculated by equation (4.18):

- $\hat{L}_f = \left(\frac{0.015}{2}\right)\left[1 + 0.35 \ln \frac{\left(\frac{0.01}{2} + \frac{0.015}{2}\right)}{\frac{0.01}{2}}\right] = 0.01[m]$

- perimeter of the fin $= 2*t + 2*L = 2*0.2*10^{-3} + 2*0.033 = 0.0664[m]$
- Area of fin $= t*L = 0.2*10^{-3}*0.033 = 6.6*10^{-6}[m^2]$

Applying equations (4.17) and (4.19):

- $m = \sqrt{\frac{29.17 * 0.0664}{202 \times 6.6 \times 10^{-6}}} = 38.12$
- $\eta_f = \frac{\tanh(38.12 * 0.01)}{38.12 * 0.01} = 0.95$

So the actual heat transfer flow through the fin is

- $Q_{fin\ act} = 0.849 * 0.95 = 0.806$[W]

Now the total heat transfer from the element is:

- $Q_{total} = 0.806 + 0.0339 = 0.84$ [W]

Now the number of elements needed to perform the condenser load from mixture region can be determined by the following equation

$n = \frac{1671.65}{0.84} = 1988.3$ elements

Now the number of elements needed to perform the total condenser load is:

- $N_{total} = 98.75 + 1988.3 = 2087$ elements, Number of elements in available condenser= 190*30=5700 elements.

4.3.4 Pipe Design and Selection

To calculate the inner diameter for the pipe the following stepscan be used: [Reference2]

$$Q=\dot{m}*\upsilon \qquad (4.26)$$

Where Q: is the flow rate [m^3/s], $\dot{m}$ is the mass flow rate of refrigerant [kg/s],υ is the specific volume [m^3/kg] ,[table 3.1] [reference 8]:

$$Q=V*A \qquad (4.27)$$

Where V: is the velocity of refrigerant[m/s]. [TABLE A-10] ,A is the cross sectional area [m^2]

$$A= \pi d_i^2 /4 \qquad (4.28)$$

Where: d_i is the inner diameter [m]:

$$d_i = \sqrt{\frac{A*4}{\pi}} \qquad (4.29)$$

To calculate the outer diameter for the pipe can be as follows[Reference10]

$$\sigma_t =\frac{Pin(ro^2+ri^2)}{(ro^2-ri^2)} \qquad (4.30)$$

Where σ_t: is the tangential stress [Mpa], P_{in}: is the inner pressure [Mpa], r_o: is the outer radius [m], r_i: is the inner radius [m]. σ_t: can be calculated from the following equation:[reference 10]:

$$\frac{SY}{n} = \sqrt{\sigma t^2 + Pin\sigma t + Pin^2} \qquad (4.12)$$

Where S_Y is the yield strength [MPa],[70 MPa for copper],[reference 110], n is the factor of safety, taken 8 [recommended from cooper hand book]. Now by using equation (4.11) the outer radius of the pipe can be calculated:

$t = r_o - r_i$ **(4.13)**

Where: - t is the thickness of the pipe [mm]

4.3.4.1 Suction line pipe

Inlet pipe for compressor named suction line.Last equation is used to determine the suction line diameter:

- $Q = 0.0134*0.07 = 9.38*10^{-4}$ [m^3/s]
- $A = \frac{Q}{V} = \frac{0.000938}{10} = 9.38*10^{-5}$[m^2]
- $d_i = \sqrt{\frac{0.0000938*4}{\pi}} = 10.92*10^{-3}$ [m]
- $r_i = 5.46*10^{-3}$ [m] = 5.46 [mm]
- $\frac{700}{8} = \sqrt{бt^2 + 2.928\ бt + (2.928)^2}$
- $б_t = 84.57$ [bar]
- $84.57 = \frac{2.928\,(ro^2 + 0.00546^2)}{(ro^2 - 0.00546^2)}$
- $r_o = 5.52$ [mm]
- $t = 5.52 - 5.46 = 0.064$[mm]

The inner and outer diameter in inch is :

- $d_{i,\ inch} = d_{i,\ mm}/25.4 = 10.92/25.4 = 0.423$ [inch]
- $d_{o,\ inch} = d_{o,\ mm}/25.4 = 11.04/25.4 = 0.435$ [inch]

By referring to copper hand book (TABLE A-11), 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 0.43 inch .

4.3.4.2 Discharge line pipe

Discharge line is the outlet pipe from compressor.To determine the diameter of discharge line, last equation used :

- $Q = 0.0134*0.018 = 2.4*10^{-4}$ [m^3/s]
- $A = \frac{Q}{V} = \frac{0.00024}{14} = 1.72*10^{-5}$[m^2]
- $d_i = \sqrt{\frac{0.0000172*4}{\pi}} = 4.68*10^{-3}$ [m]
- $r_i = 2.34*10^{-3}$ [m] = 2.34 [mm]
-
- $\frac{700}{8} = \sqrt{6t^2 + 11.5976t + (11.597)^2}$
- $\text{б}_t = 75.9$ [bar]

The inner and outer diameter in inch is

- $d_{i,\ inch}$ =4.68/25.4=0.184 [inch]
- $d_{o,\ inch}$ = 4.8/25.4=0.189 [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 .

4.3.5 Capillary tube Design and Selection

By using the software Danfoss capillary tube selector under the following conditions the condenser temperature 45C°, the evaporator's temperature (0, 5, 8C°), the cooling load (645, 626 ,310 W), and R143a is the refrigerant used we found that the length and the diameter of the tube as in the table:-

Table 4.1 Capillary tube Selection

	L[m]	Φ[m]
Chamber #1	1.94	**0.042**
Chamber #2	1.94	**0.042**
Chamber #3	2.11	**0.042**

CHAPTER FIVE

ELECTRICAL DESIGN AND SELECTION

CHAPTER FIVE

ELECTRICAL DESIGN AND SELECTION

5.1 Introduction

This chapter presents the protection of refrigeration cycle.The refrigeration cycle is keeping food at a suitable temperature.If a motor stops at several times due to power failure or a single breakdown, food will be destroyed. So the electric motor must be protected.

5.2 Types of Electrical Circuits

There are two types of electrical circuits in general. There arepower circuits and control circuits. Capacities for small units are usually the control and power of one; either for units with high capacities controlling circuit is controlled separately from the power circuit.[Reference7]

5.2.1 Control Circuit

This circuit is working to influence the controls to follow up the implementation of required control program as defined by introducing elements operating according to the requirements of control thermostat and unequivocal pressure and break convection. Also working to introduce elements of the capacity as the exact timing advance . Often control circuit is working with single phase. And potential voltages in control circuit are less or equal in power circuit. The energy consumed to control much less of the energy power circuit. [Reference 7]

5.2.2 Power Circuit

Power circuit is working to operate or stop power elements such as motors depending on the signal of the control circuit. The potential voltage and the electric power consumed in the power circuit equal to or greater than what is used in the control circuit. The power circuit is working in one or three phases.[Reference7]

5.3 Components of Electrical Circuits

The electrical circuit is built with more conditions.First the requirement is economy and this is important in the installation of any equipment. Second cost is an impotent part of total cost; repair and maintenance costs are also part of total cost. Third the requirement for driving equipment is simplicity, simple mean can be understood, simple to operate, maintain and repair.There are many components which are used in this project to protect a refrigeration cycle.The components are:

5.3.1 Highpressure and low pressure control

Figure (5.1)shows the a schematic diagram of a high and low pressure control ,the devise parts located on discharge line and suction line of compressor , high pressure shut off compressor when the pressure rises on high pressure region and same on the low pressure region .

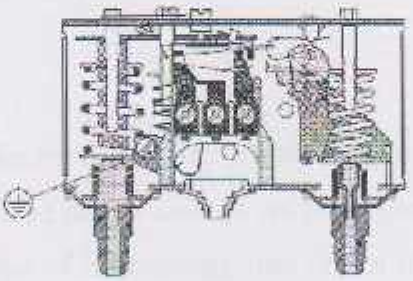


Figure (5.1)a schematic diagram high and Low pressure control

5.3.2 Solenoid valve

Figure (5.2)shows the a schematic diagram of a solenoid valve , solenoid valve is switch , may be normally closes or normally opens and is selected depending on desire function .

This valve is operating by a thermostat control, so open and close valve depends on room temperature, when the room temperature reaches a set point the valve closes.

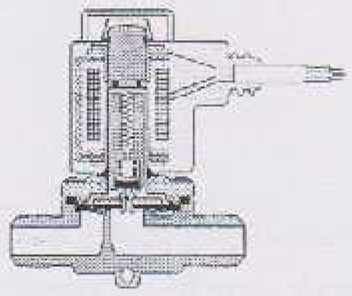


Figure (5.2)a schematic diagram of a solenoid valve

5.3.3 Thermostat

A thermostat is a device for regulating the temperature of a system so that the system's temperature is maintained near a desired set point temperature. The thermostat does this by controlling the flow of heat energy into or out of the system. That is, the thermostat switches heating or cooling devices on or off as needed to maintain the correct temperature. Figure 5.3 shows thermostat. [Reference 7]

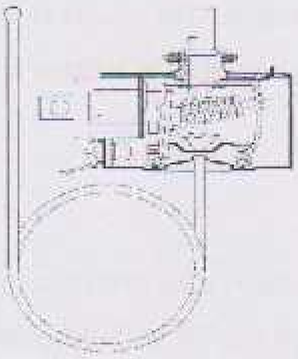


Figure (5.3) Thermostat

5.3.4Potential relay

Potential relay, also known as voltage relay, is used on single phase motors that require large starting torque. During the off cycle, the contacts are closed to the start winding. This prevents arcing and possible burning of the contacts. When the thermostat contacts close, electricity is applied to the start and run winding in the motor. As the motor picks up speed, voltage is actually generated in the winding and causes a small amount of current to flow. As the motor approaches operating speed this counter electromotive force becomes strong enough to open the start winding relay and power is no longer applied to the start winding. Power remains on the run winding and the motor continues to run. [Reference 4]

5.3.5 Contactor

Figure (5.4)shows the schematic diagram of contactor and contactor device,contactor is electric switch.It is similar to relay but it used at higher amperage. One of the contactor components is magnet, which is used to move switch open and close.

Motor starters are basically contactors with overload protection built in. This overload protection supplements the protection offered by breakers. Breakers protect the entire circuit, while the protection built into the stator protects a specific. [Reference 4]

(a) (b)

Figure (5.4) (a) a schematic diagram of contactor and (b) contactor device

5.3.6 Overload:

The most common cause of motor failure is overheating. The condition is created when a motor exceeds its normal operating current flow. The result can be either a breakdown of the motor winding insulation and a short circuit, or a winding burn-out. For this reason overload protection is provided in the form of a current and temperature sensitive control which will open the circuit before any damage can occur. [Reference 7]

5.4 Control circuit design

5.4.1 Principle Control circuit

In page (32/34), chapter six ,there is control circuit plane. There are notes to explain the control circuit work:

- A manual switches S1, S2, and S3 must be pressed at the beginning S1control the motor evaporator fan K0, S2control the motor evaporator fan K1, S3control the motor evaporator fan K2,and S1,S2 and S3 control the running compressorK3.

- The thermostat in the +8°C room controls solenoid valve SV1 in the liquid line while the two other thermostats in the +5°C and 0°C rooms respectively controlSV2 andSV3.When K0,K1 and K2 is closed, the evaporator fans motor work, then solenoid valves of SV1, SV2 andSV3.
- Combined high and low pressure startthe compressor motor K3.
- The compressor motor is thus only indirectly controlled by the room thermostats and is able; for example, to run for some time after all the thermostats have cut out.
- If high pressure or low pressure or over load has happened the cycle will be shut down immediately.
- As monitoring system for the cycle there are a lamps for the compressor lights when it shuts down and offwhen it runs,and there are also lamps for evaporator fanlights iftheyʼrerunning .

5.4.2Principle of work

- Pressing the start button ,the compressor works if there is no low or high pressure
- Thermostat measures the temperature if it is higherthan 8 degrees Celsius, and Switch S1 on, thermostat gives signal to the motor fan (K0)to run then the fan contactor close gives signal to the solenoid valve (SV1) which will open and pass the flow of the refrigerant to the first chamber .

- The other thermostat measures the temperature if it is higherthan 5 degrees Celsius,and Switch S2 on, thermostat gives signal to the motor fan (K1)to run then the fan contactor closes and the solenoid valve (SV2) opens and passes the refrigerant to the second chamber .
- The third thermostat measures the temperature if it is higher than 0 degrees Celsiusand Switch S3 on, thermostat gives signal to the fansmotor (K2)to run then the fan contactor closes and the solenoid valve (SV3) opens and passes the refrigerant to the thirdchamber.
- If thermostat of the three evaporators reach the temperature or one of the switchesmakesoff,the solenoid valvewill close and prevent the pass of the flow the pressure decreases and the low pressure stops the compressor.

5.5 Power Circuit

Power circuit hasfive motors that work according to the instructions receivedfrom the control circuit, In page (2/54), chapter six depicts powercircuit.For power cycle The evaporator fan runs immediately with the compressor,and also the condenser fan runs immediately with the compressor.

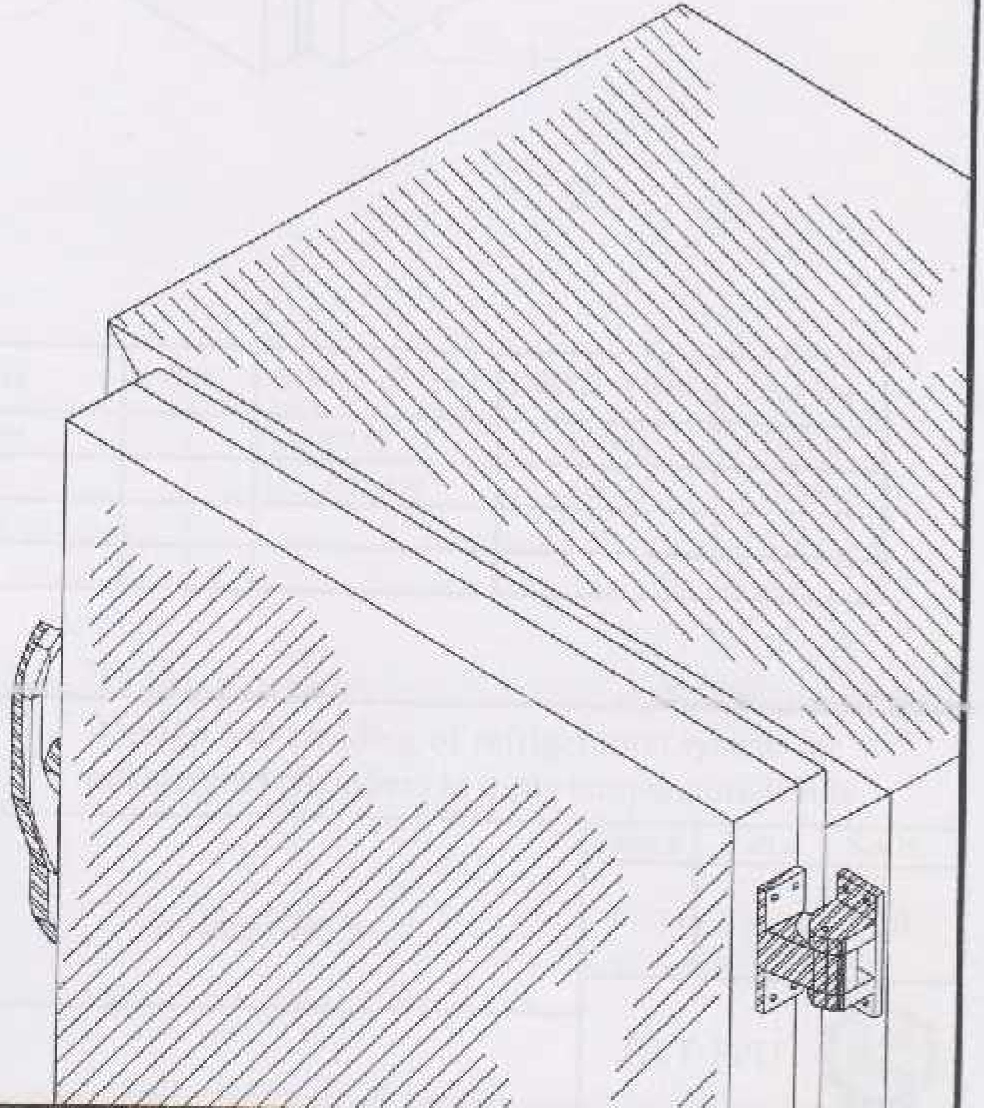
CHAPTER SIX

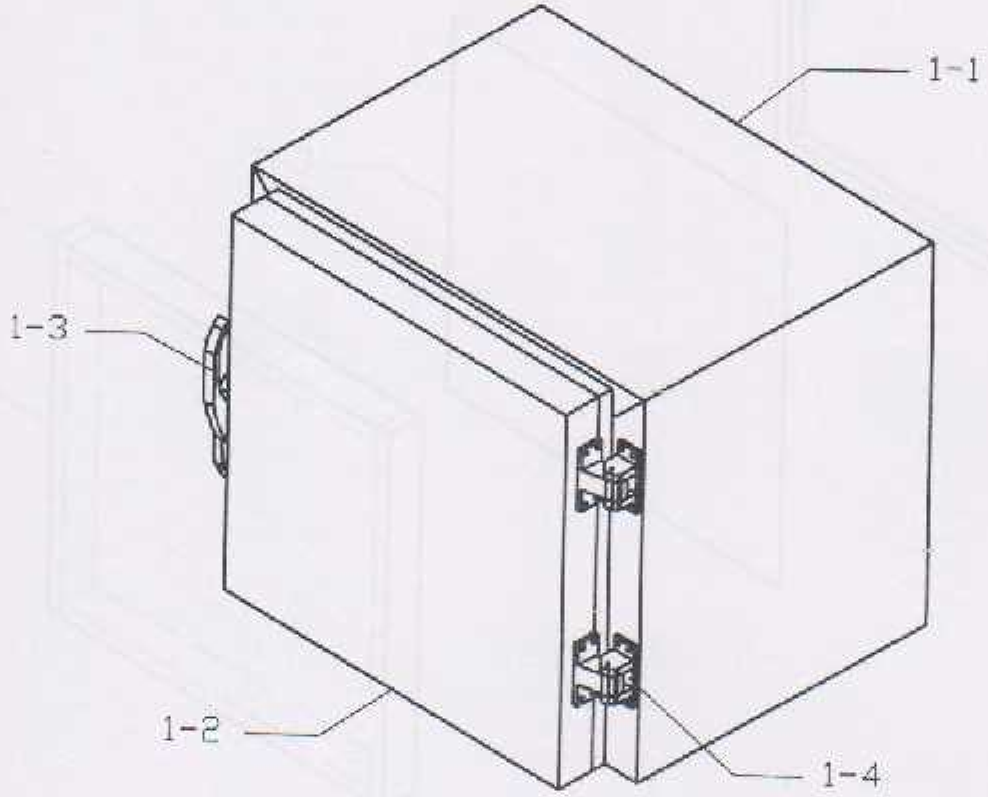
Design

DESIGN OF CHAMBER

Drawing by: Raed Nihad Abu Munshar
Mera Dadou
Hamzeh Arafeh

Check by: Dr . Ishaq Sider





Sub	Part Name	Qun	Stock Dim (cm)	Thickness (mm)	Material
1-1	External Body	1	60 * 43 * 50	0.07 mm	CR Steel
1-2	Door	1	51 * 5 * 49	0.07 mm	CR Steel
1-3	Hand Door	1	--	--	--
1-4	Hinge	2	--	--	--

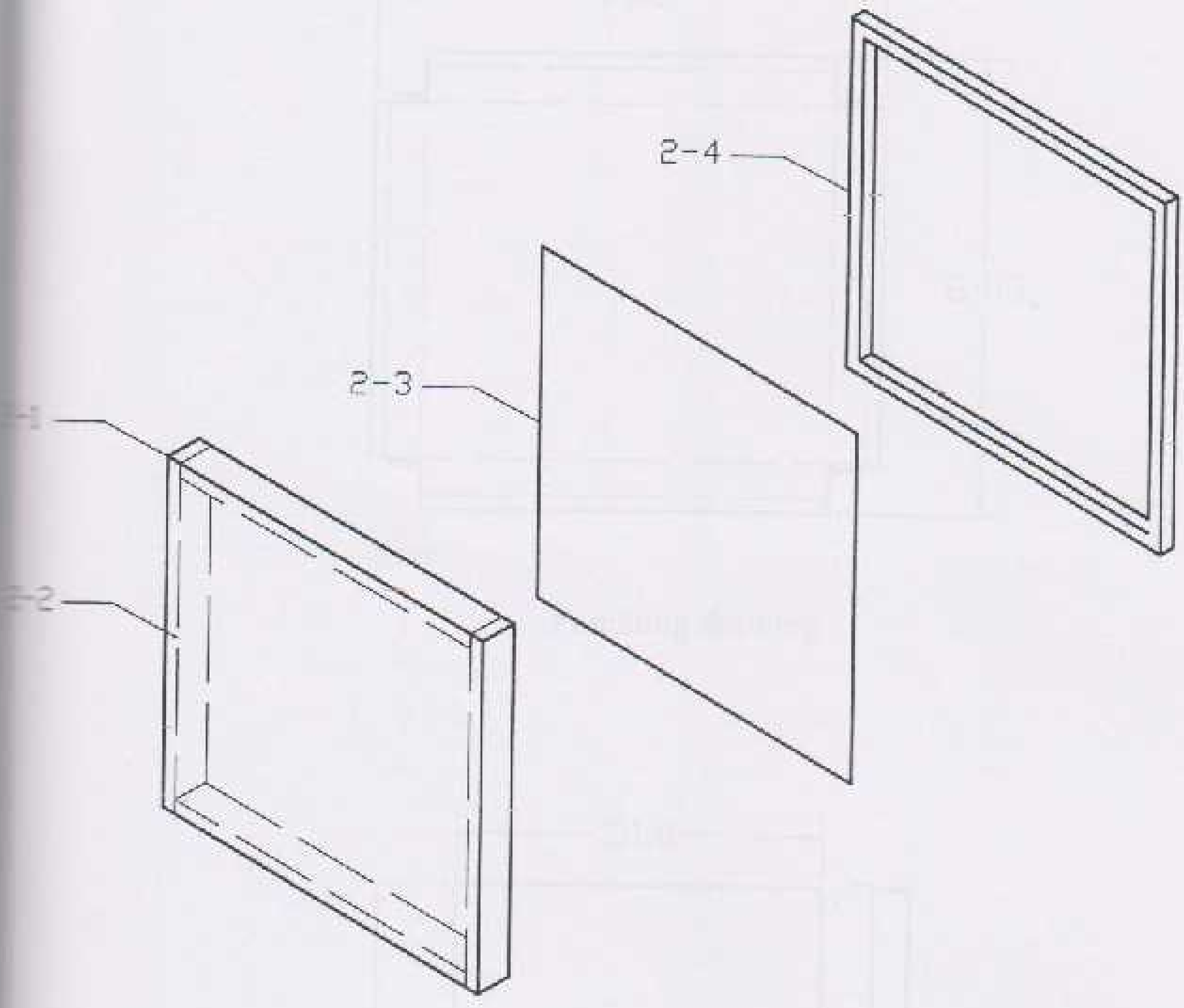
Design and building of refrigeration system for refrigerated chambers at three temperature levels

Name	Sign
Raed Nihad Abu Munshur	
Hamza Arafeh	
Mera Dadou	
Dr. Ishaq Sider	

Ordinary Body

Page #	Part #	Scale
1 - 34	A	1:10

CET-PPU



Sub	Part Name	Qun	Stock Dim (cm)	Thickness (mm)	Material
2-1	Front Side	1	51 * 5 *49	0.07 mm	CR Steel
2-2	White Wood	4	47 * 5	20	Wood
2-3	Back Side	1	51 * 49	0.07 mm	CR Steel
2-4	Magnetic Rubber	1	51 * 49 * 2	--	Plastic

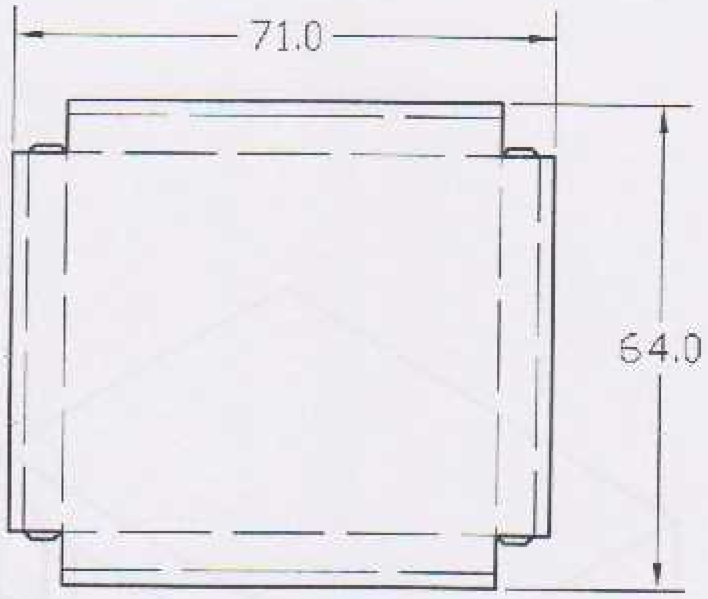
Design and building of refrigeration system for refrigerated chambers at three temperature levels

Name	Sign
Raed Nihad Abu Munshar	
Hamza Arafeh	
Mera Dadou	
Dr. Ishaq Sider	

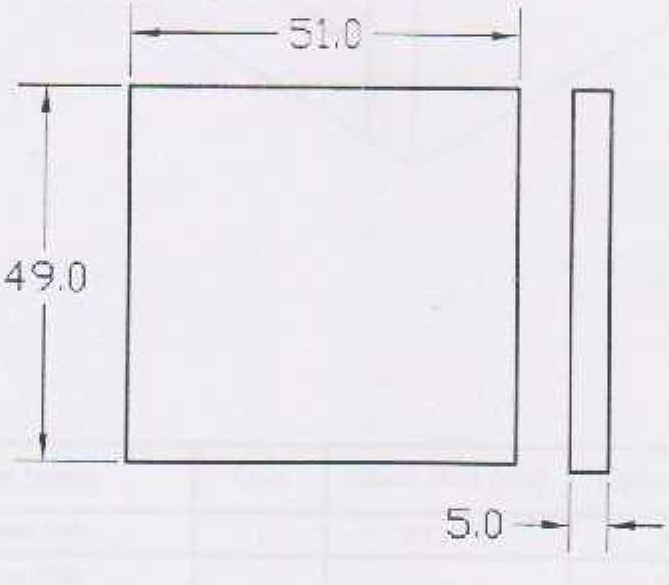
Door

Page #	Part #	Scale
2 - 34	A-1-2	1:10

CET-PPU



Punching drawing



Bending Drawing

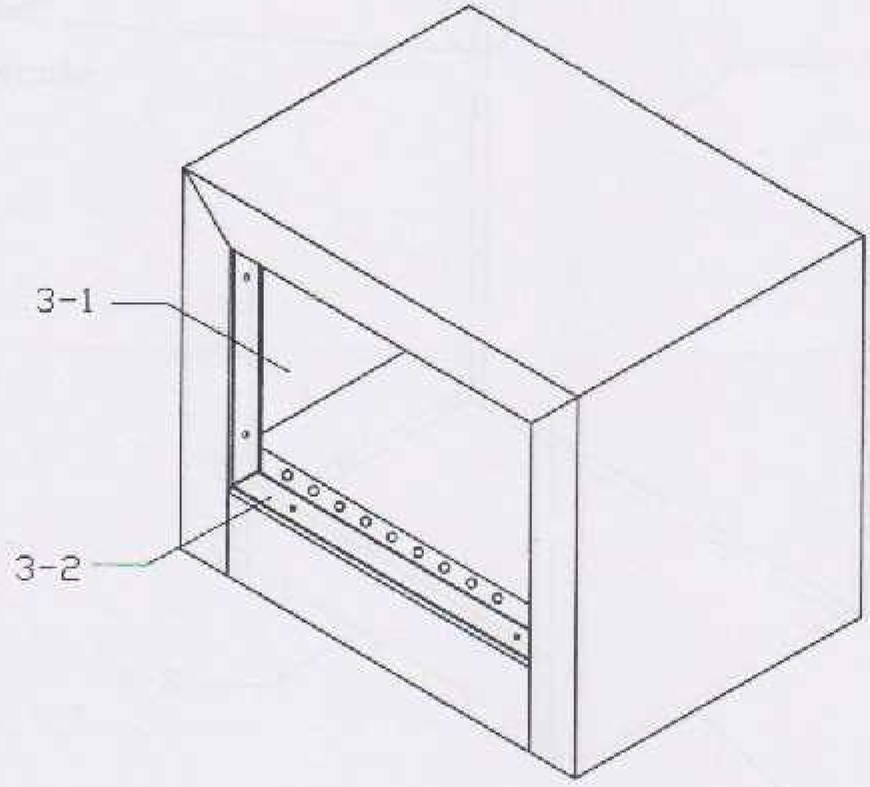
Name	Sign
Raed Nihad Abu Munshar	
Hamza Arafeh	
Mera Dadou	
Dr. Ishaq Sider	

Design and building of refrigeration system for refrigerated chambers at three temperature levels

Front side - Door

Page #	Part #	Scale
3 - 34	A-2-1	1:10

CET-PPU



Sub	Part Name	Qun	Stock Dim (cm)	Thickness (mm)	Material
3-1	Internal Body	1	50 * 35 *41	0.07	CR Steel
3-2	Cover Guide	4	–	0.07	CR Steel

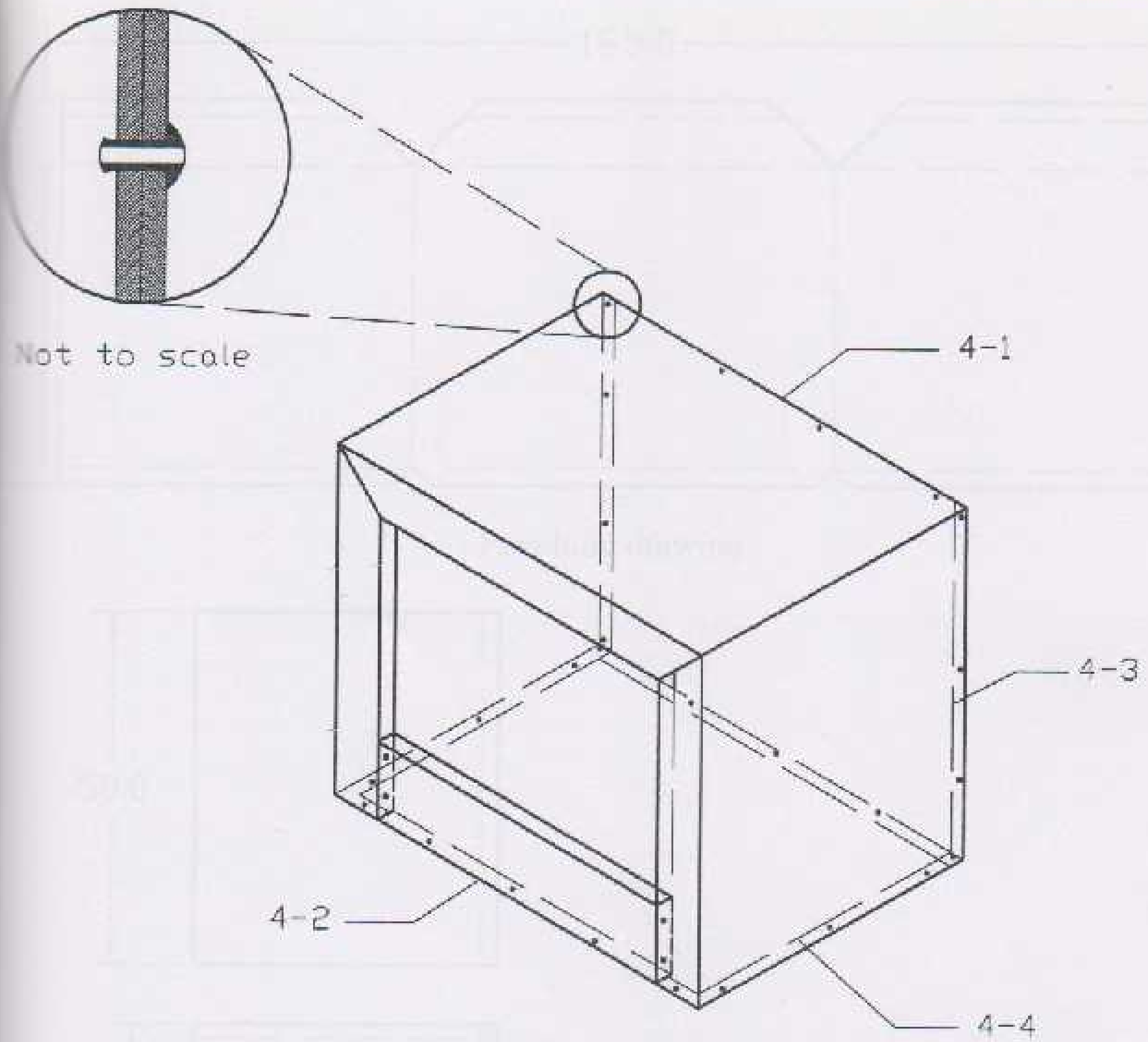
Design and building of refrigeration system for refrigerated chambers at three temperature levels

Name	Sign
Raed Nihad Abu Munshar	
Hamza Arafeh	
Mera Dadou	
Dr. Ishaq Sider	

External Body

Page #	Part #	Scale
4- 34	A-1-1	1:10

CET-PPU



Sub	Part Name	Qun	Stock Dim (cm)	Thickness (mm)	Material
4-1	External Side	1	--	0.07	CR Steel
4-2	Front Guide	1	--	0.07	CR Steel
4-3	Back side	1	60 * 52	0.07	CR Steel
4-4	Lower side	1	50 * 43	0.07	CR Steel

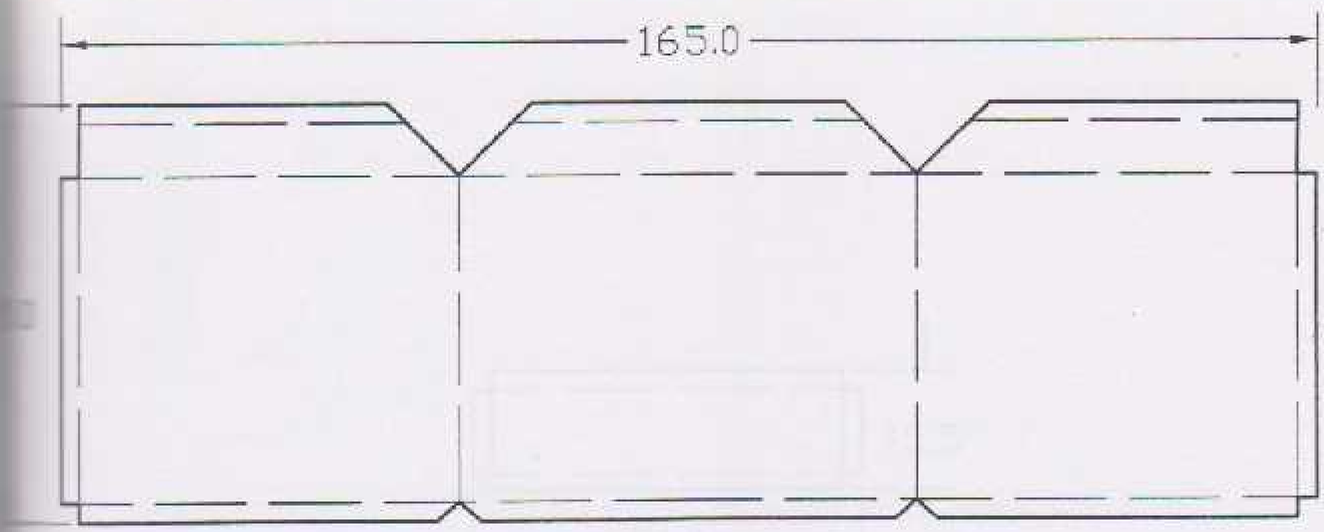
Design and building of refrigeration system for refrigerated chambers at three temperature levels

Name	Sign
Raed Nihad Abu Munshar	
Hamza Arafeh	
Mera Dadou	
Dr. Ishaq Sider	

External Body

Page #	Part #	Scale
5 - 34	A-1-1	1:10

CET-PPU



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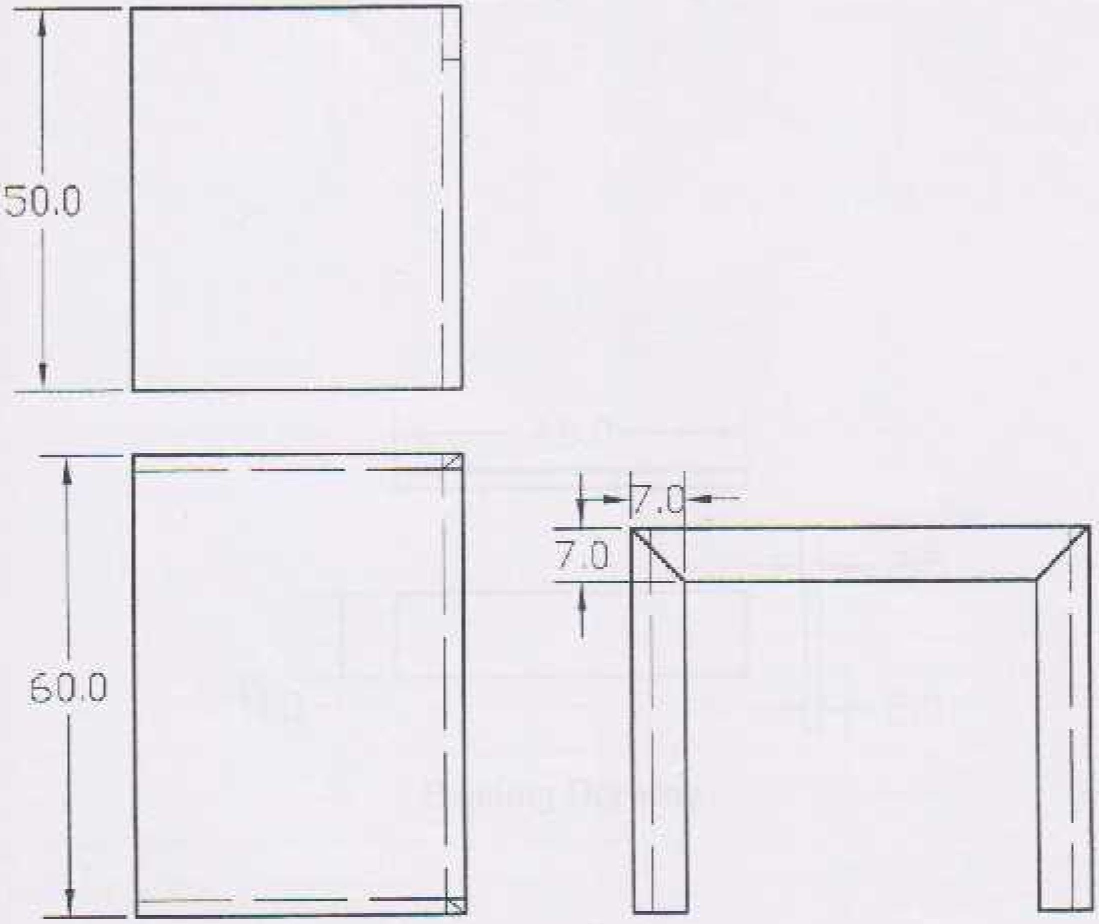


Bending Drawing

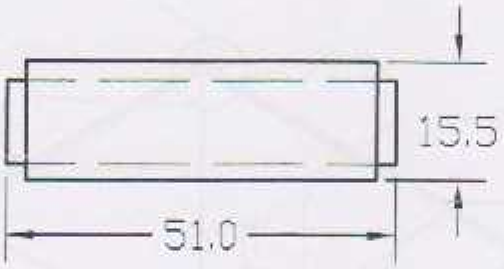
Design and building of refrigeration system for refrigerated chambers at three temperature levels

Name	Sign
Raed Nihad Abu Munshar	
Hamza Arafeh	
Mera Dadou	
Dr. Ishaq Sider	

External Side

Page #	Part #	Scale
6 - 34	A-4-1	1:10

CET-PPU



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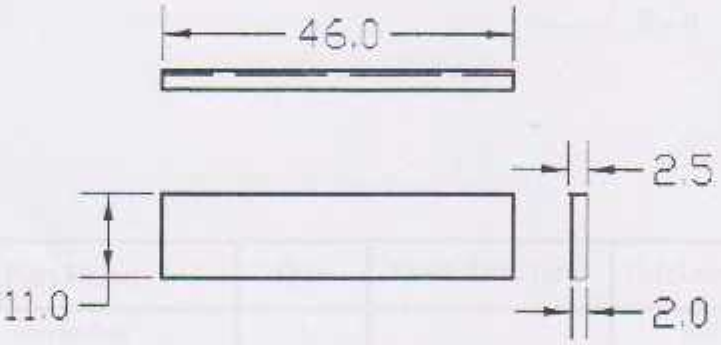


Bending Drawing

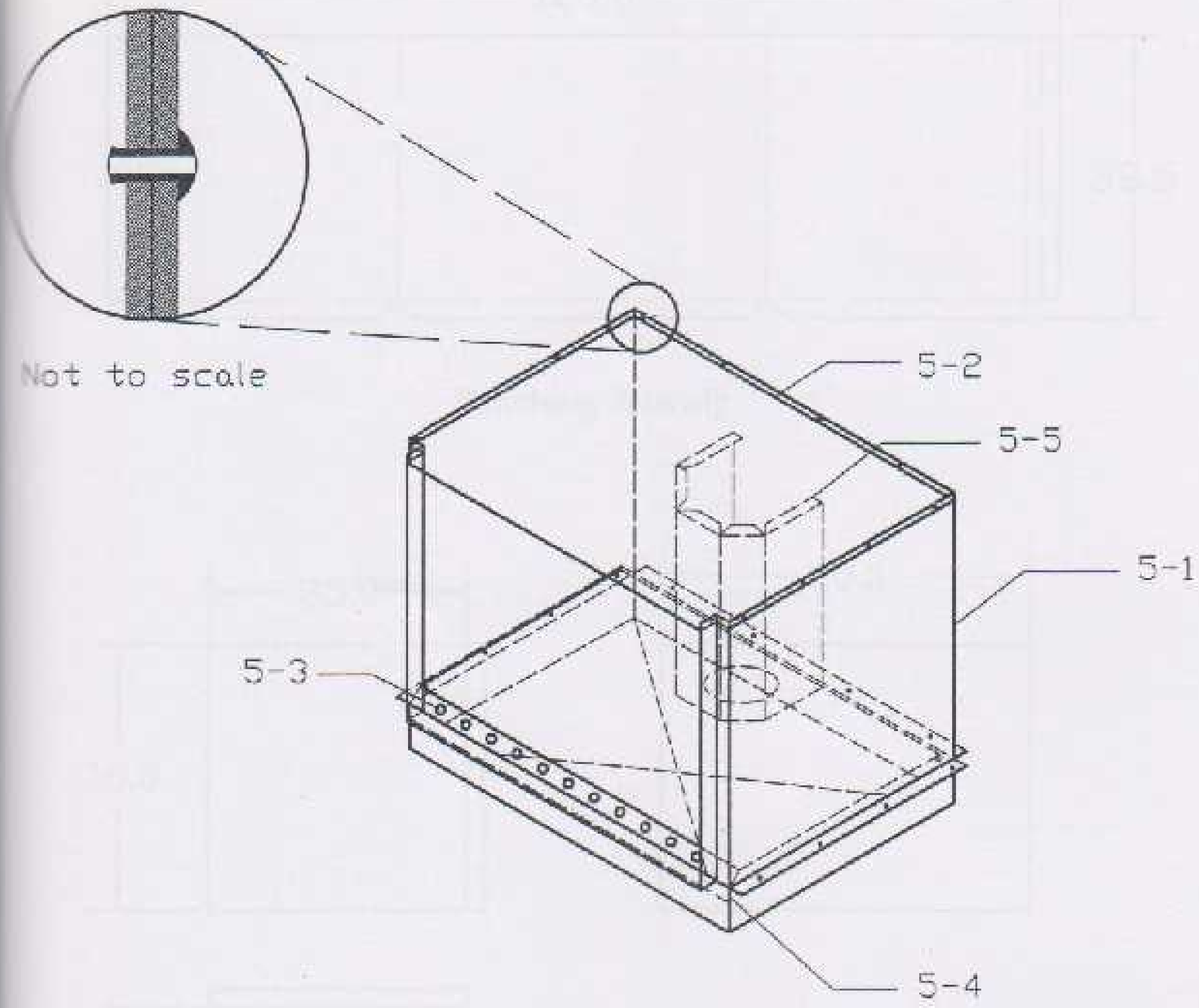
Design and building of refrigeration system for refrigerated chambers at three temperature levels

Name	Sign
Raed Nihad Abu Murshar	
Hamza Arafeh	
Mera Dadou	
Dr. Ishaq Sider	

Front Guide

Page #	Part #	Scale
7 - 34	A-4-2	1:10

CET-PPU



Sub	Part Name	Qun	Stock Dim (cm)	Thickness (mm)	Material
5-1	Internal Side	1	--	0.07 mm	CR Steel
5-2	Upper Internal side	1	--	0.07 mm	CR Steel
5-3	Cover	1	--	0.07 mm	CR Steel
5-4	Lower Internal side	1	--	0.07 mm	CR Steel
5-5	Cannel	1	--	0.07 mm	CR Steel

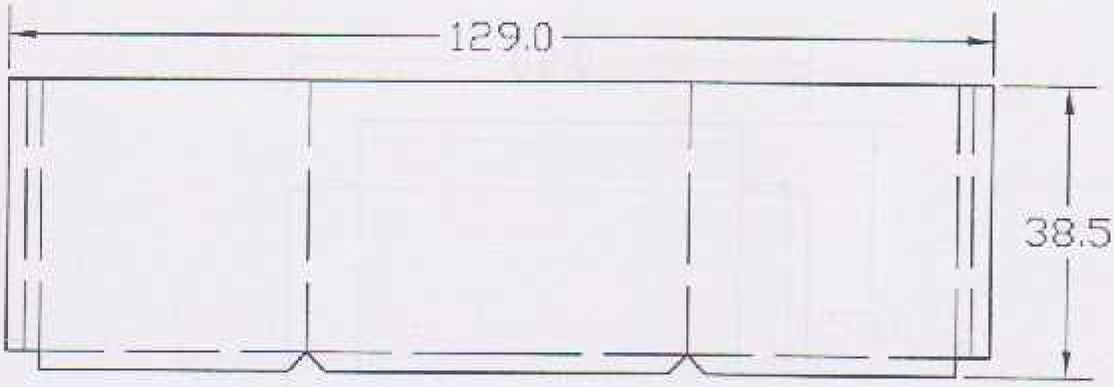
Design and building of refrigeration system for refrigerated chambers at three temperature levels

Internal Body

Page #	Part #	Scale
8 - 34	A-3-1	1:10

CET-PPU

Name	Sign
Raed Nihad Abu Munshar	
Hamza Arafeh	
Mera Dadou	
Dr. Ishaq Sider	



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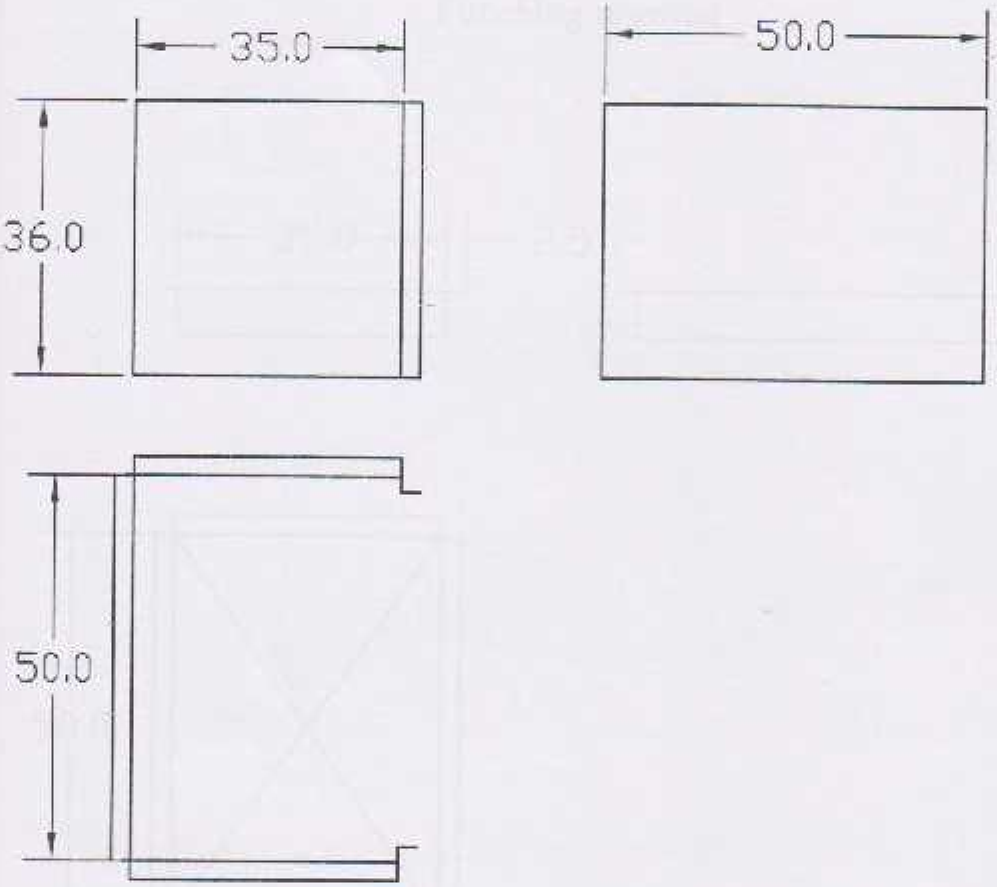


Bending Drawing

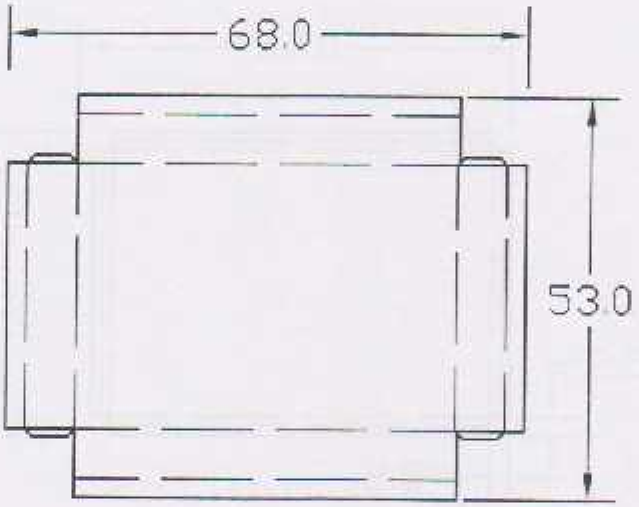
Name	Sign
Raed Nihad Abu Munshar	
Hamza Arafeh	
Mera Dadou	
Dr. Ishaq Sider	

Design and building of refrigeration system for refrigerated chambers at three temperature levels

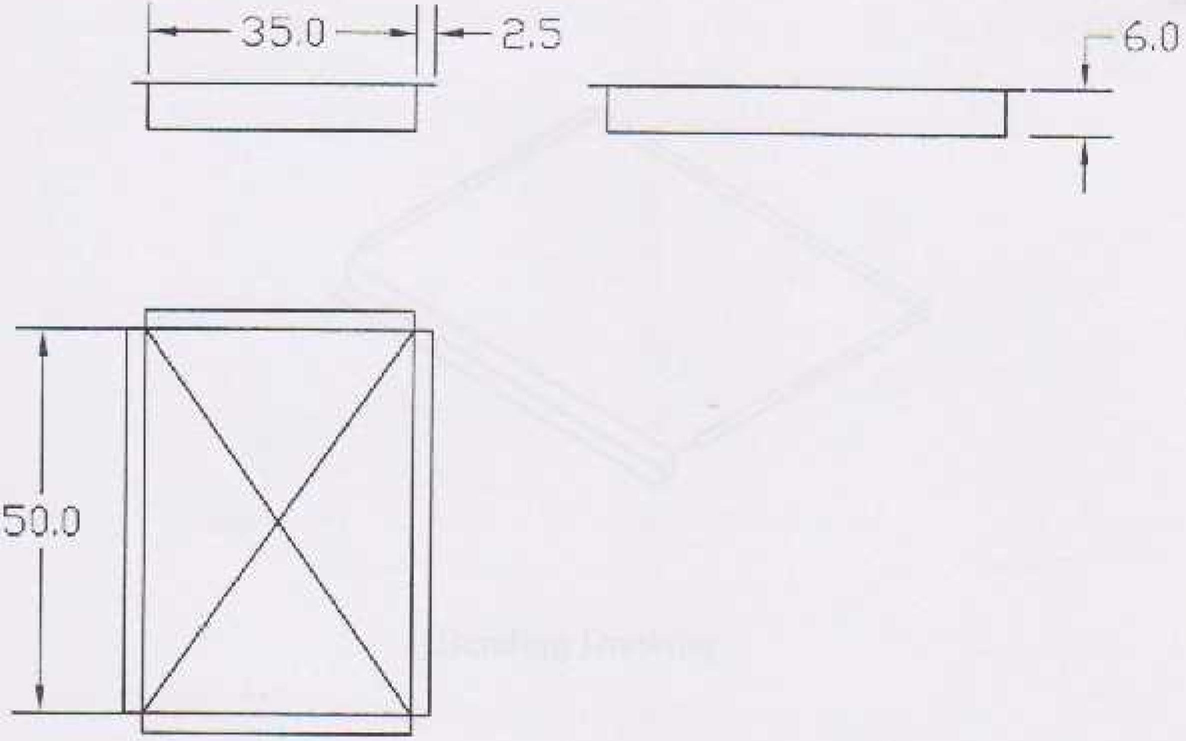
Internal side

Page #	Part #	Scale
9 - 34	A-5-1	1:10

CET-PPU



Punching drawing



Bending Drawing

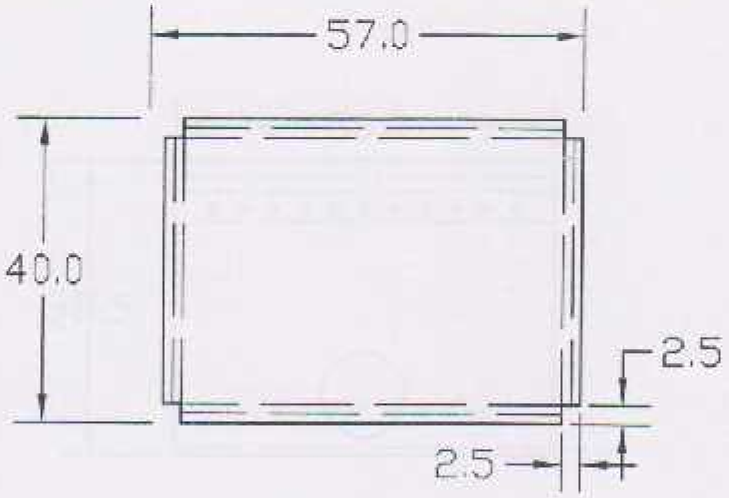
Name	Sign
Raed Nihad Abu Munshar	
Hamza Arafeh	
Mera Dadou	
Dr. Ishaq Sider	

Design and building of refrigeration system for
refrigerated chambers at three temperature levels

Lower internal side

Page #	Part #	Scale
10 - 34	A-5-3	1:10

CET-PPU



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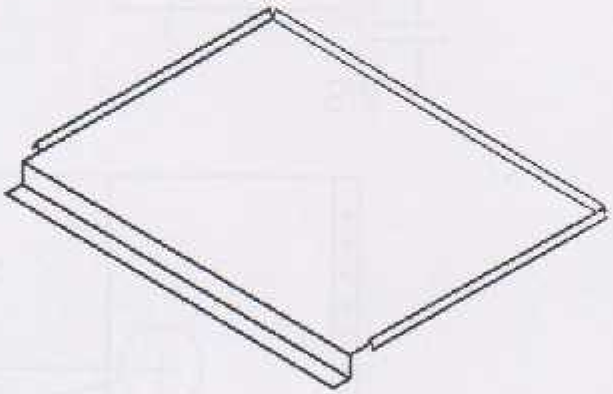


Bending Drawing

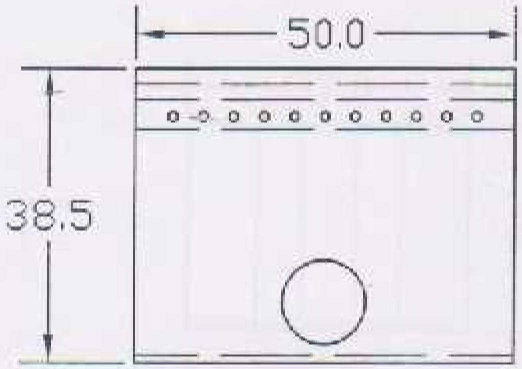
Design and building of refrigeration system for refrigerated chambers at three temperature levels

Name	Sign
Raed Nihad Abu Munshar	
Hamza Arafeh	
Mera Dadou	
Dr. Ishaq Sider	

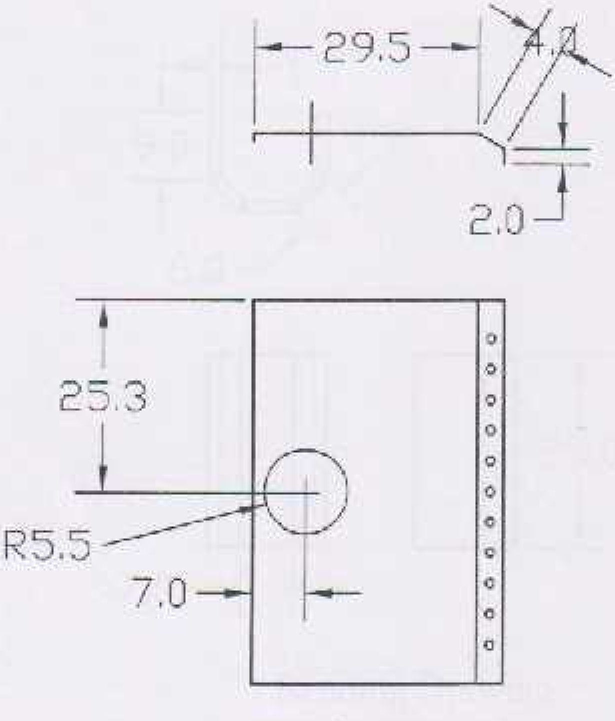
Upper internal side

Page #	Part #	Scale
11 - 34	A-5-2	1:10

CET-PPU



Punching drawing



Bending Drawing

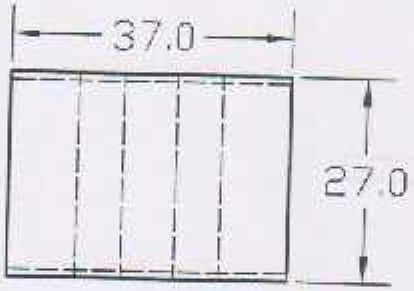
Design and building of refrigeration system for refrigerated chambers at three temperature levels

Name	Sign
Raed Nihad Abu Munshar	
Hamza Arafeh	
Mera Dadou	
Dr. Ishaq Sider	

Cover

Page #	Part #	Scale
12-34	A-5-4	1:10

CET-PPU



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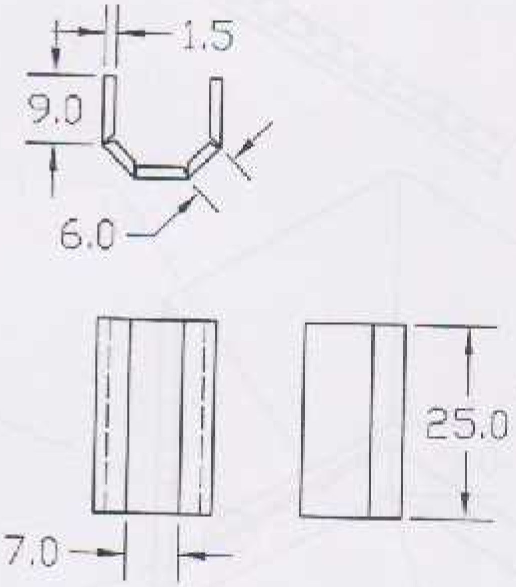


Bending Drawing

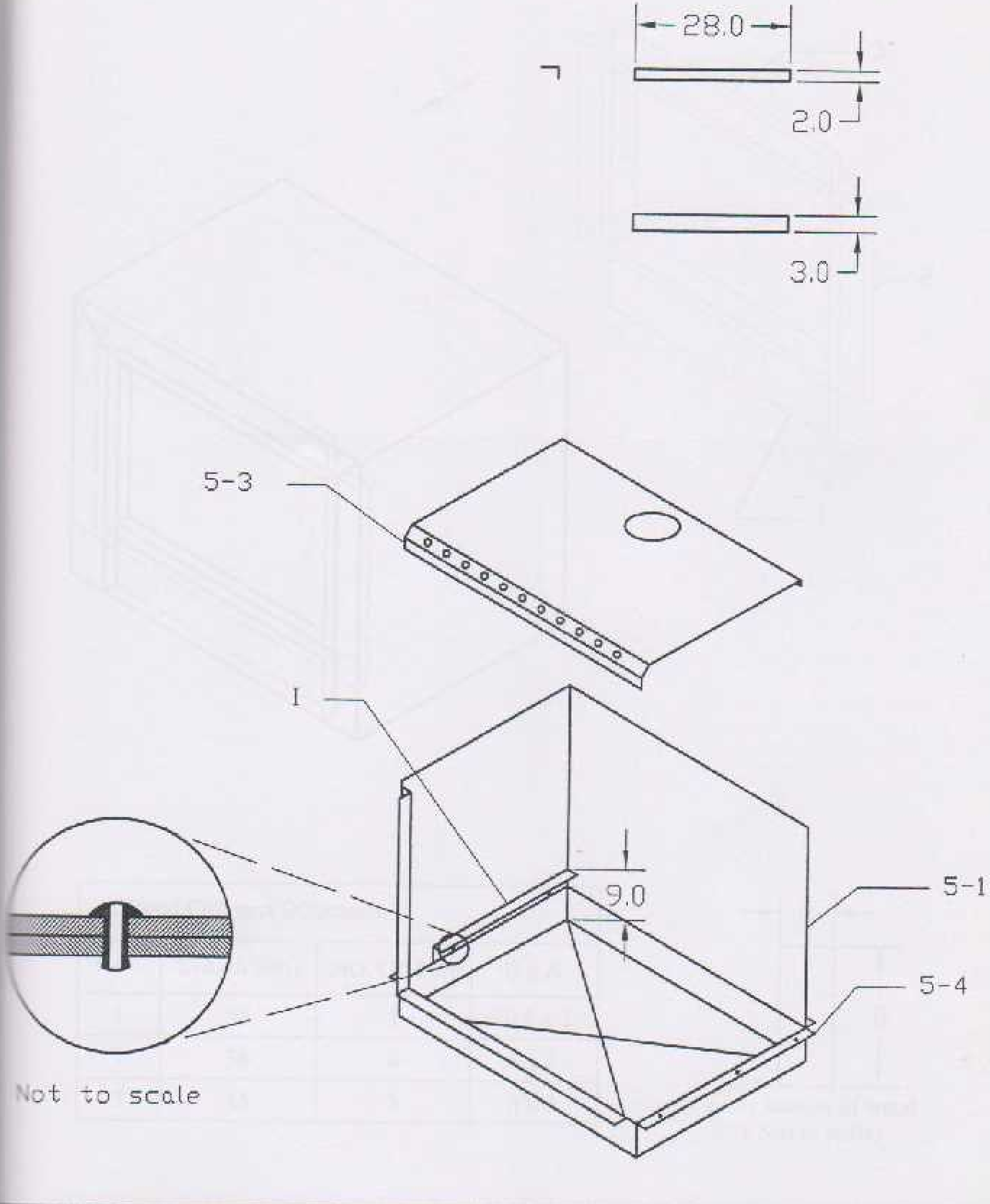
Design and building of refrigeration system for refrigerated chambers at three temperature levels

Name	Sign
Raed Nihad Abu Munshar	
Hamza Arafeh	
Mera Dadou	
Dr. Ishaq Sider	

Cannel

Page #	Part #	Scale
13 -34	A-5-5	1:10

CET-PPU



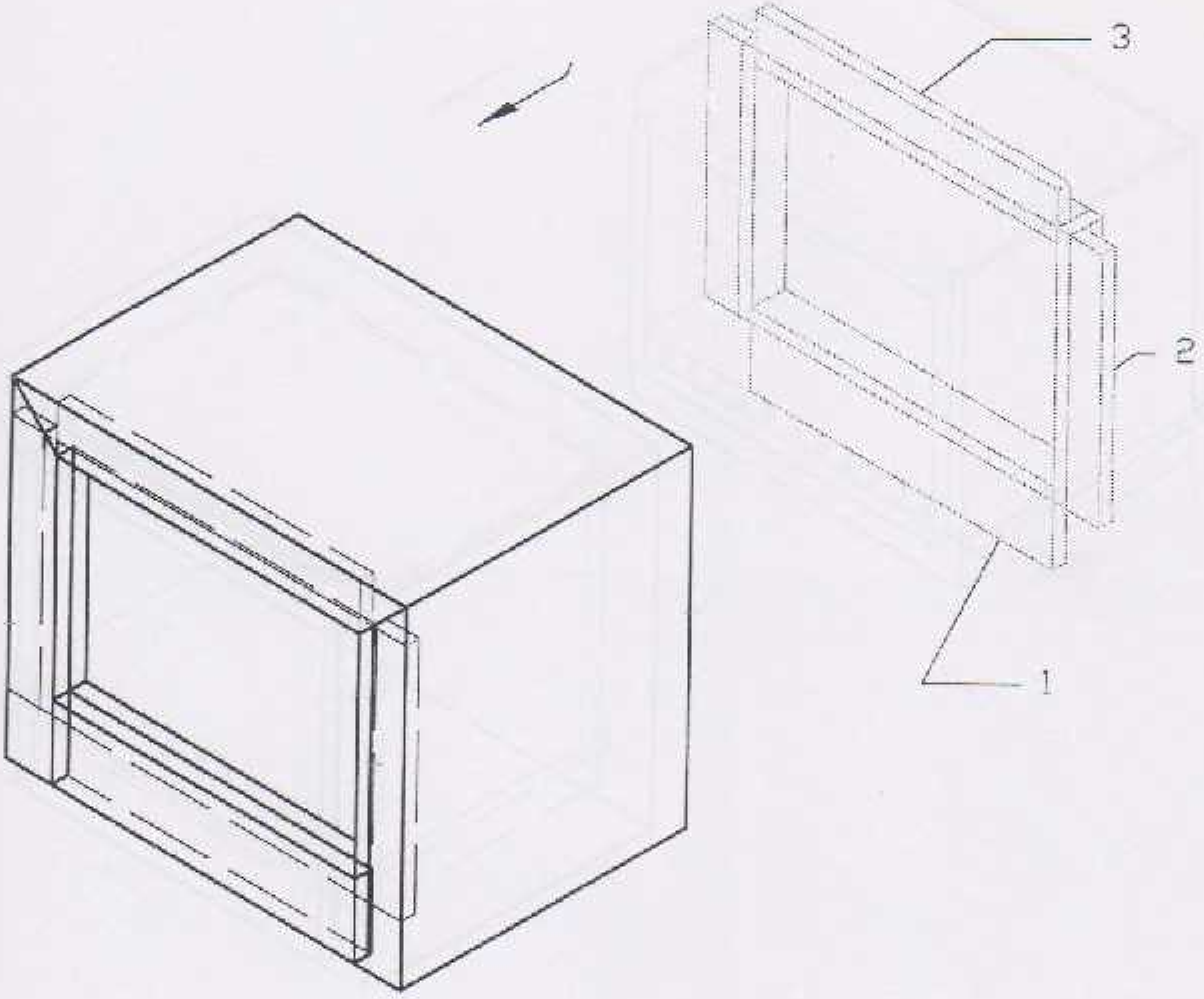
Design and building of refrigeration system for refrigerated chambers at three temperature levels

Name	Sign
Raed Nihad Abu Munshar	
Hamza Arafeh	
Meru Dudou	
Dr. Ishaq Sider	

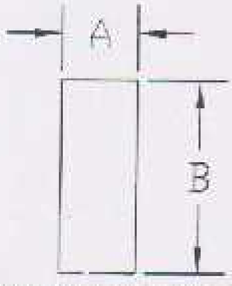
Side Guide

Page #	Part #	Scale
14 -34	I	1:10

CET-PPU



Wood Columns Dimension			
#	Length (cm)	NO. Of Parts	B x A
1	45	1	10.5 x 2
2	36	4	5 x 2
3	45	3	5 x 2



cross section of wood
(Not to scale)

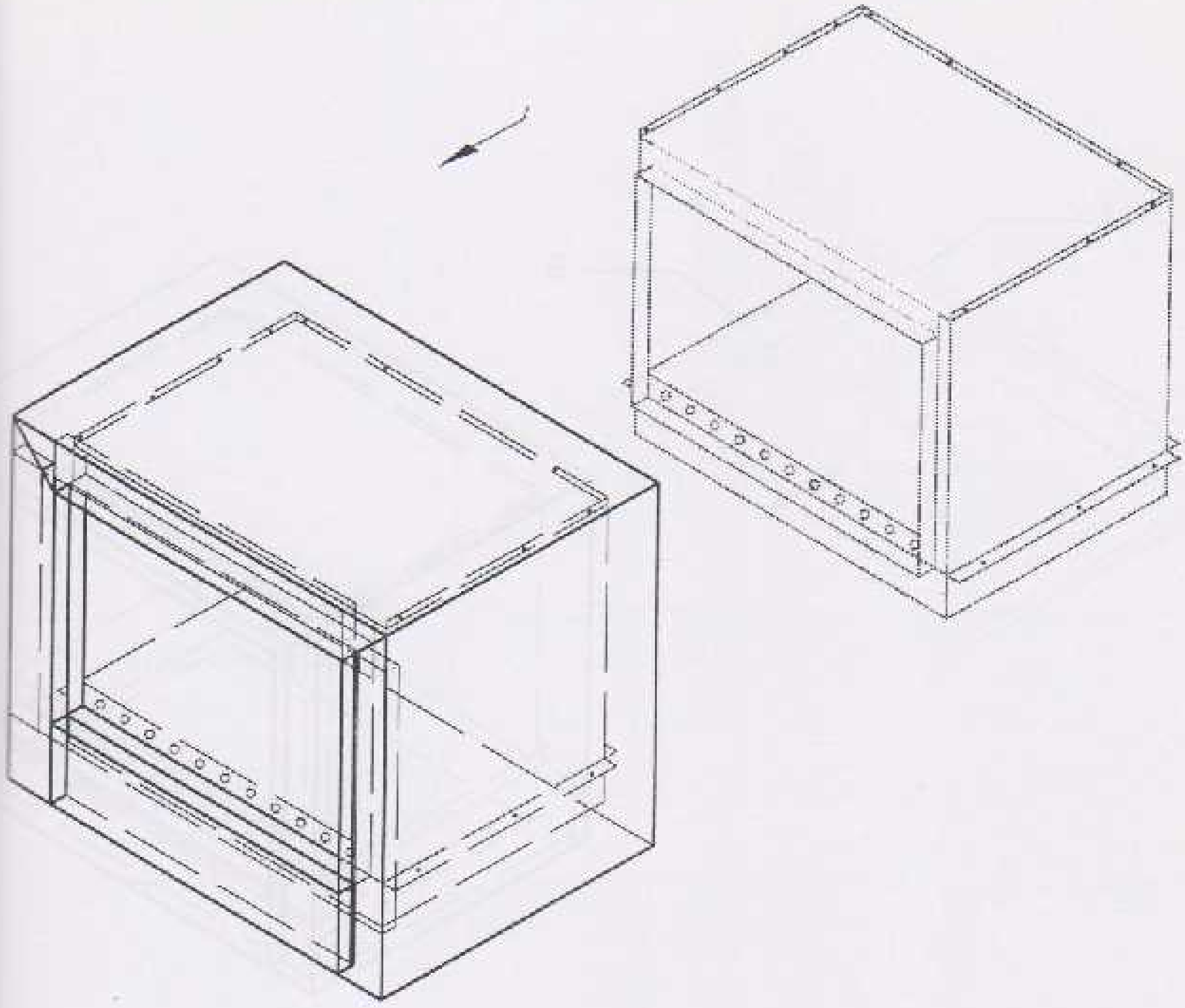
Design and building of refrigeration system for refrigerated chambers at three temperature levels

Name	Sign
Raed Nihad Abu Munshar	
Hamza Arafeh	
Mera Dadou	
Dr. Ishaq Sider	

Front Wood

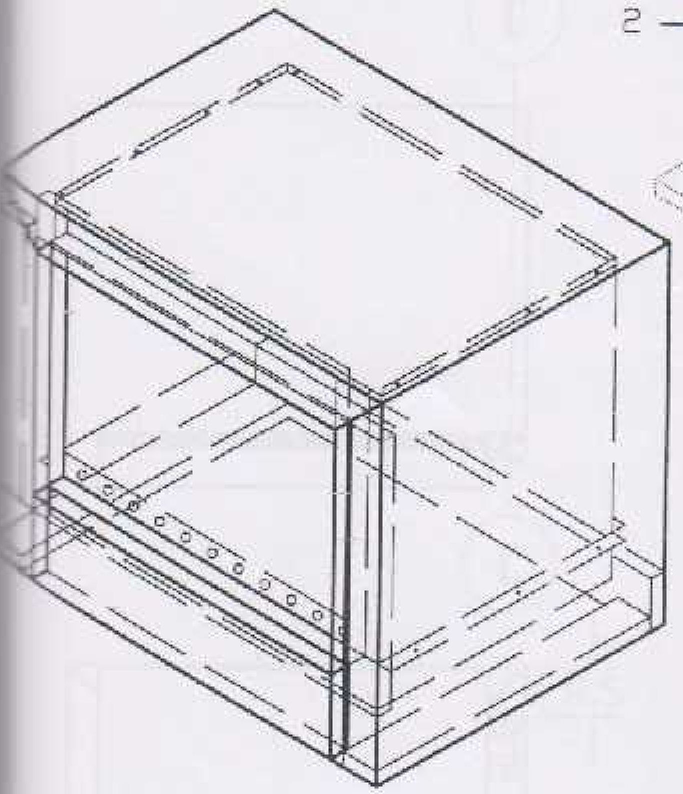
Page #	Part #	Scale
15 - 34		1:10

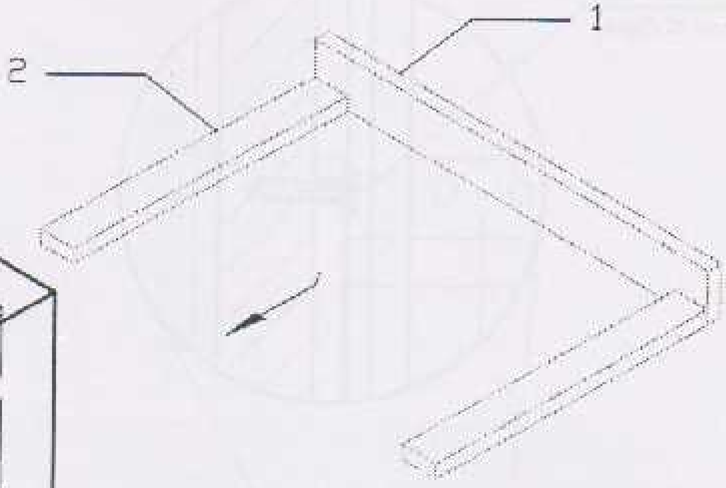
CET-PPU



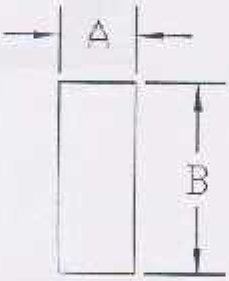
* External and internal body are connected by Staples Dim (16 X 7.3) mm , on wood guide .

Name	Sign	Design and building of refrigeration system for refrigerated chambers at three temperature levels			
			Page #	Part #	Scale
Raed Nihad Abu Munshar		Connect External and internal body	16 - 34		1:10
Hamza Arafeh					
Mera Dadon			CET-PPU		
Dr. Ishaq Sider					





Wood Columns Dimension			
#	Length (cm)	NO. Of Parts	B x A
1	60	1	5 x 2
2	48	2	5 x 2



cross section of wood
(Not to scale)

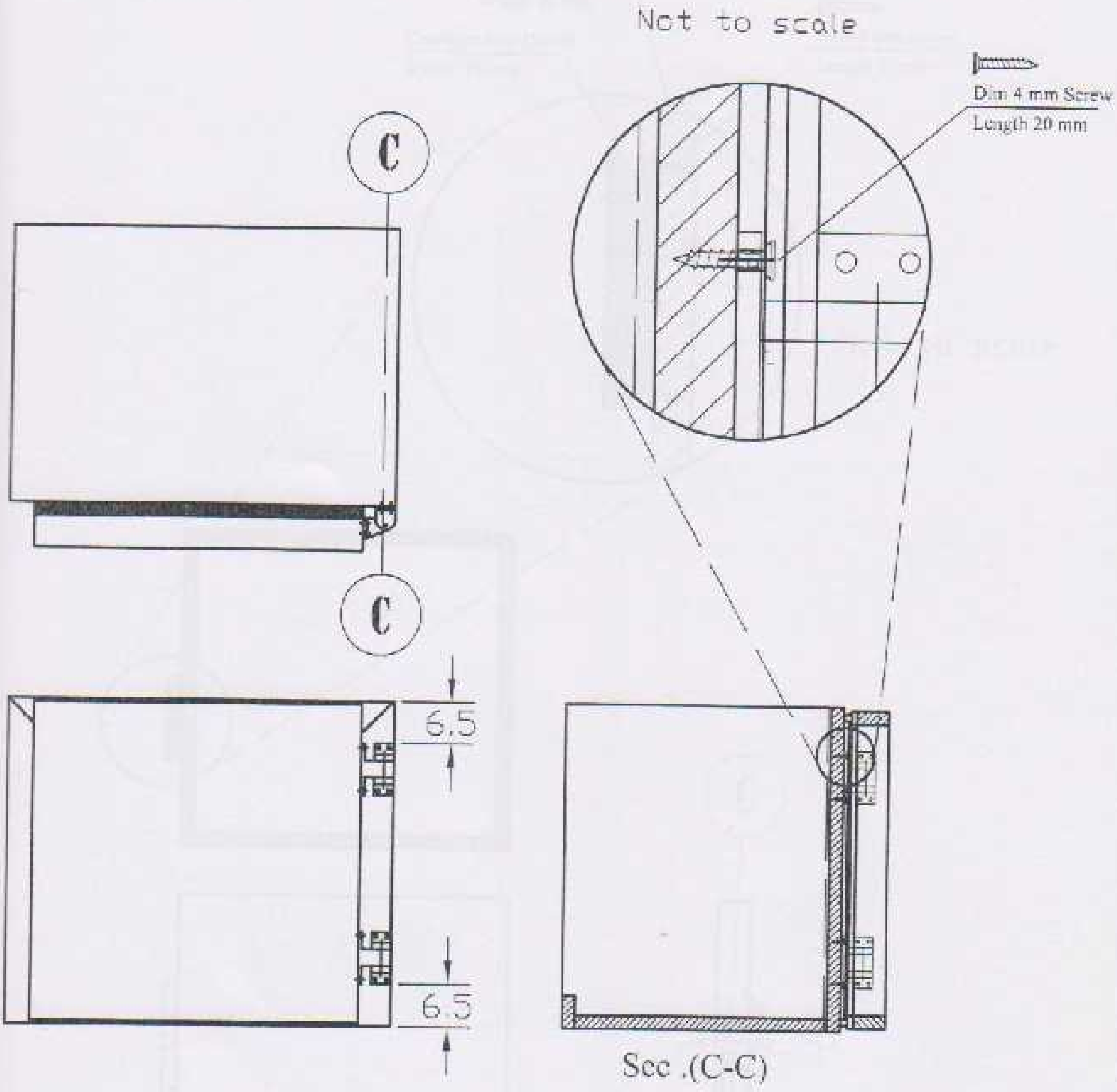
Name	Sign	
Raed Nihad Abu Munshar		
Hamza Arafeh		
Mera Dadou		
Dr. Ishaq Sider		

Design and building of refrigeration system for refrigerated chambers at three temperature levels

Back Wood

Page #	Part #	Scale
17 - 34		1:10

CET-PPU



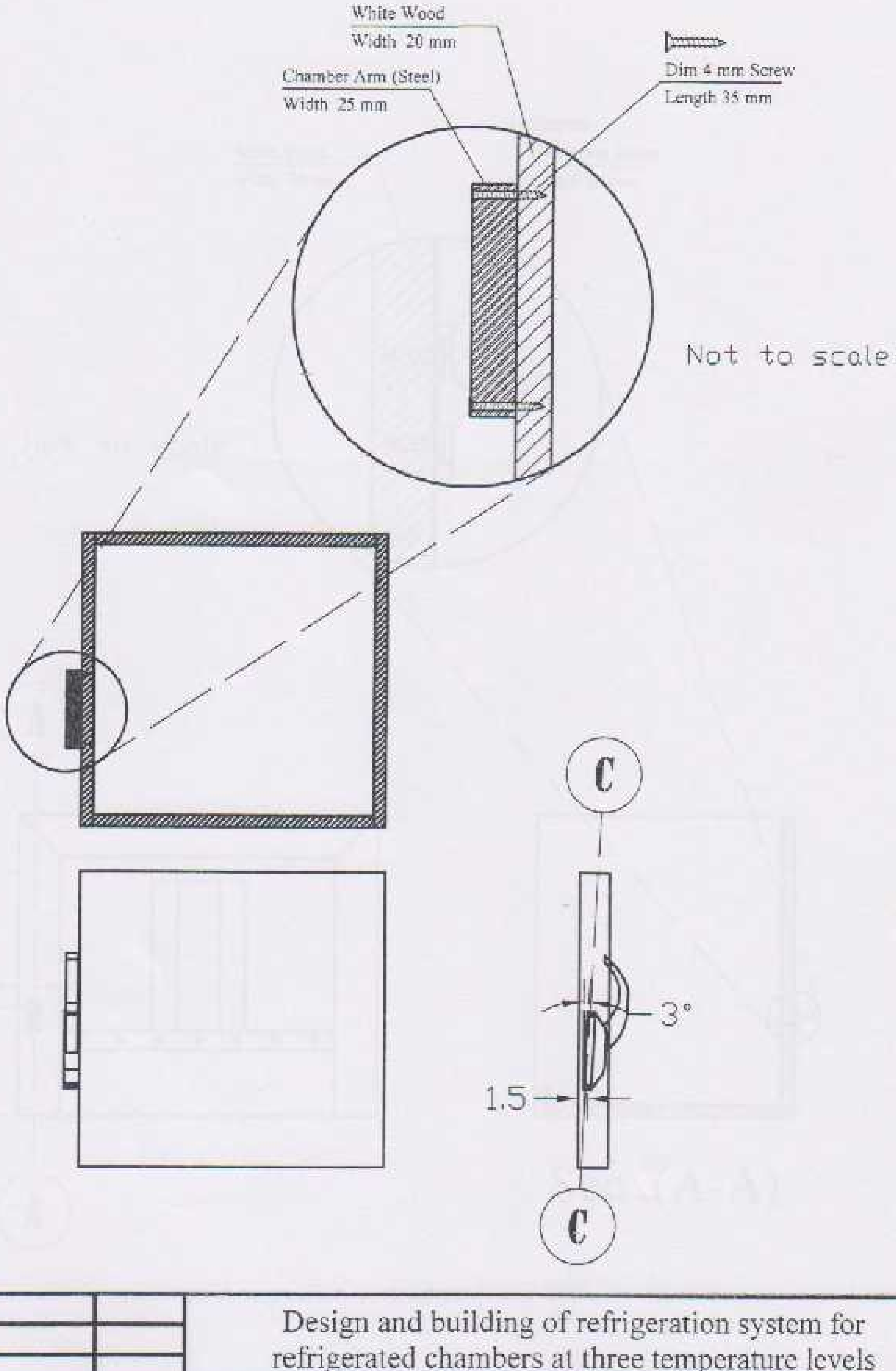
Sec .(C-C)

Name	Sign
Raed Nihad Abu Munshar	
Hamza Arafeh	
Mera Dadou	
Dr. Ishaq Sider	

Design and building of refrigeration system for refrigerated chambers at three temperature levels

	Page #	Part #	Scale
Hinge	18 - 34	A-1-4	1:10

CET-PPU



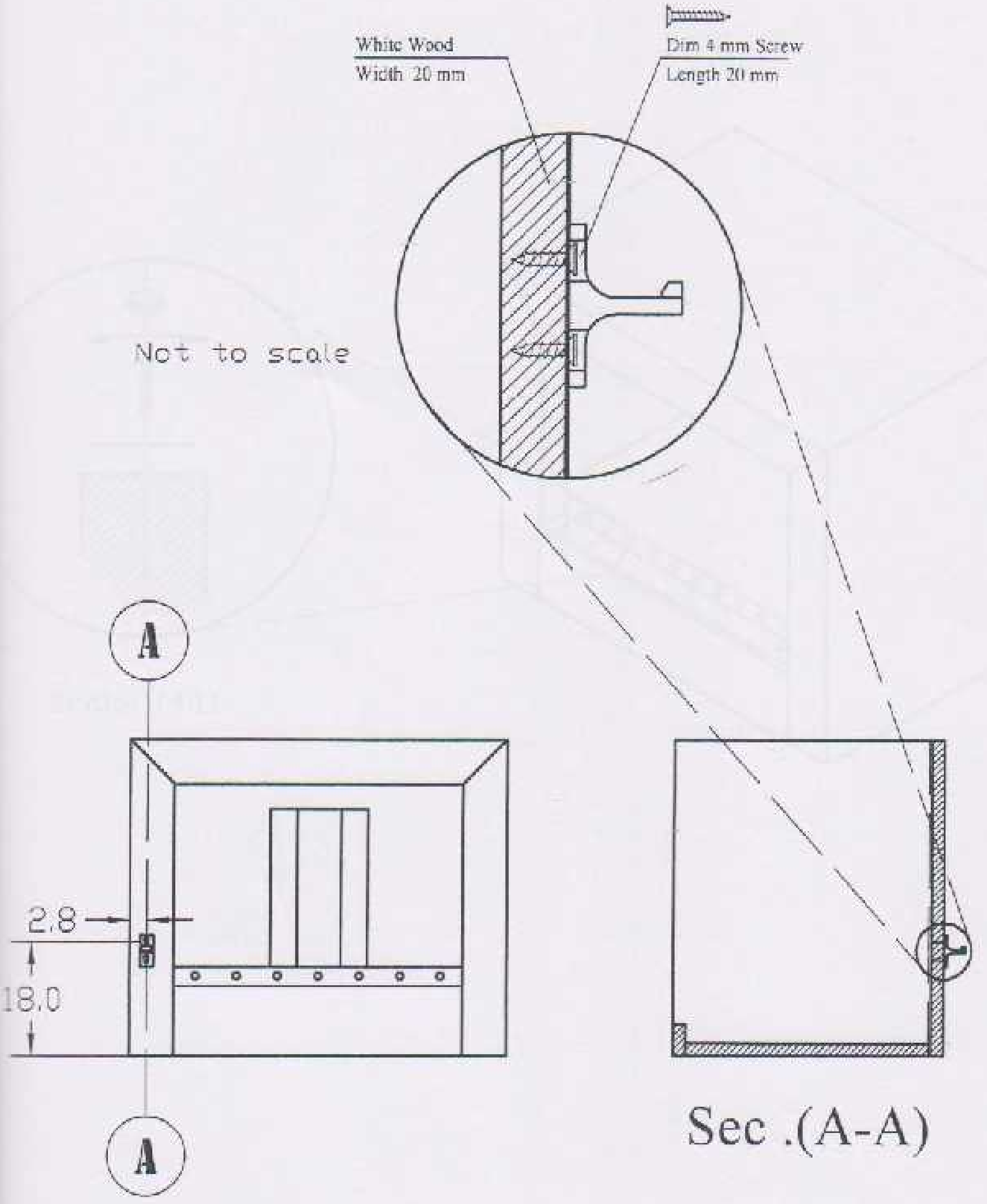
Design and building of refrigeration system for refrigerated chambers at three temperature levels

Name	Sign
Raed Nihad Abu Munshar	
Hamza Arafeh	
Mera Dadou	
Dr. Ishaq Sider	

Hand Door - 1

Page #	Part #	Scale
19 - 34	A-1-5	1:10

CET-PPU



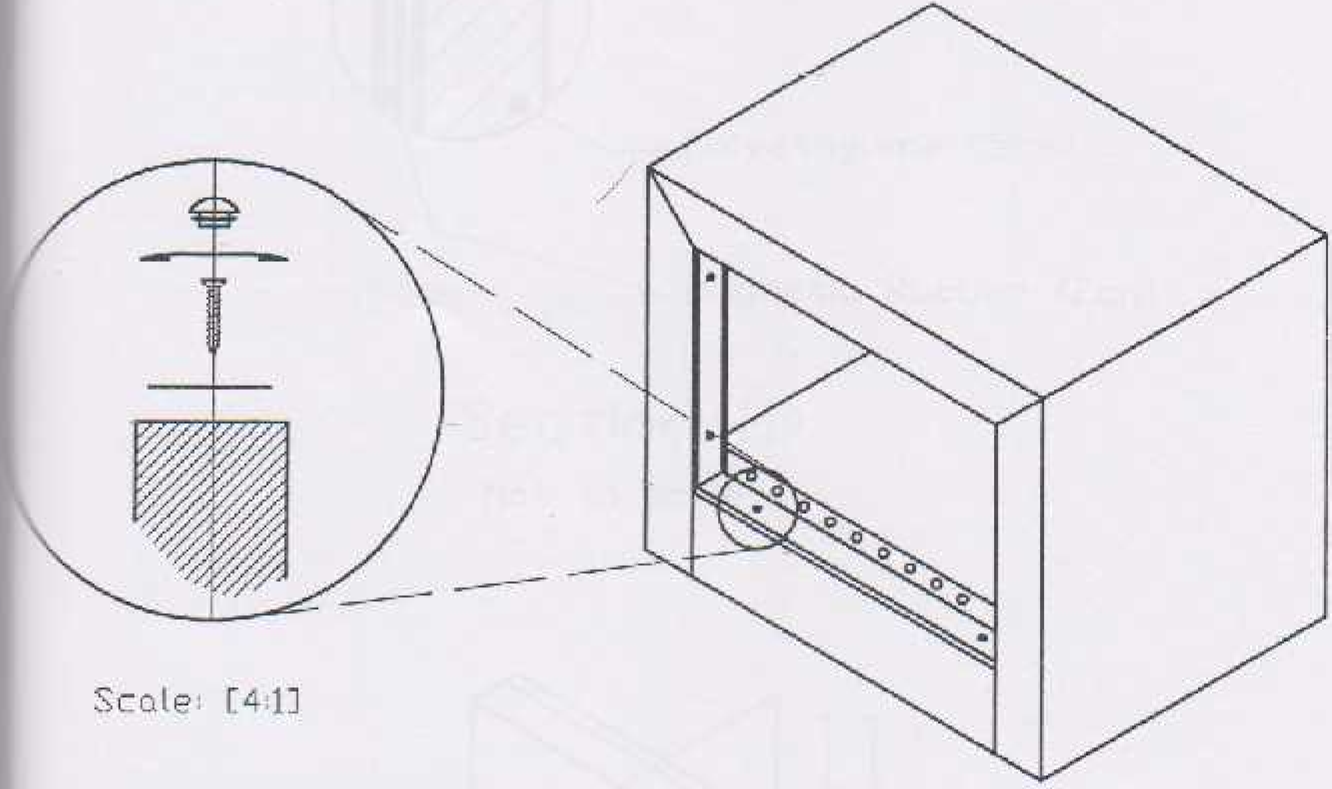
Design and building of refrigeration system for refrigerated chambers at three temperature levels

Name	Sign
Raed Nihad Abu Munshar	
Hamza Arafeh	
Mera Dadou	
Dr. Ishaq Sider	

Hand Door -2

Page #	Part #	Scale
20 - 34	A-1-5	1:10

CET-PPU



Scale: [4:1]

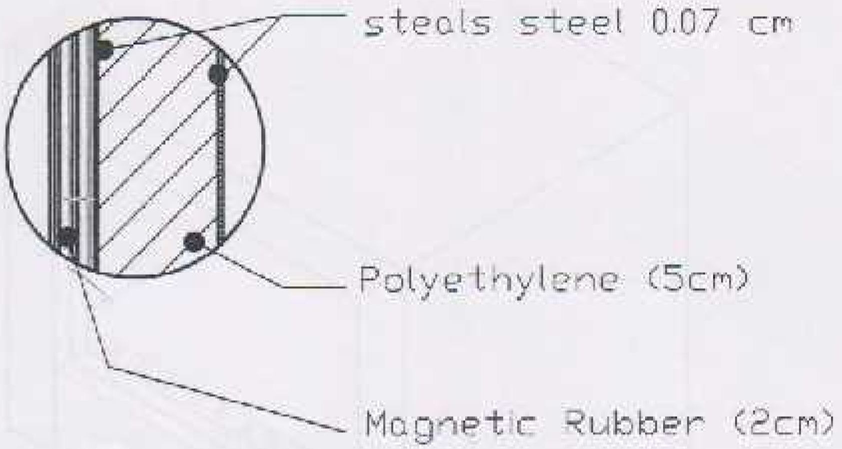
	Name	Sign
by	Raed Nihad Abu Munshar	
	Hamza Arafeh	
	Mera Dadou	
by	Dr. Ishaq Sider	

Design and building of refrigeration system for
refrigerated chambers at three temperature levels

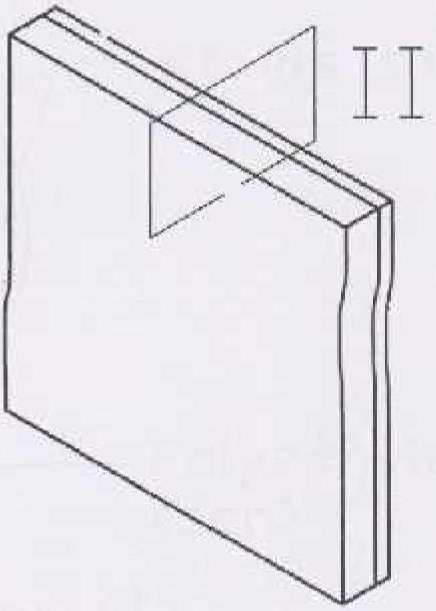
Cover Guide

Page #	Part #	Scale
21 - 34	A-3-2	1:10

CET-PPU



Section:(II)
Not to scale



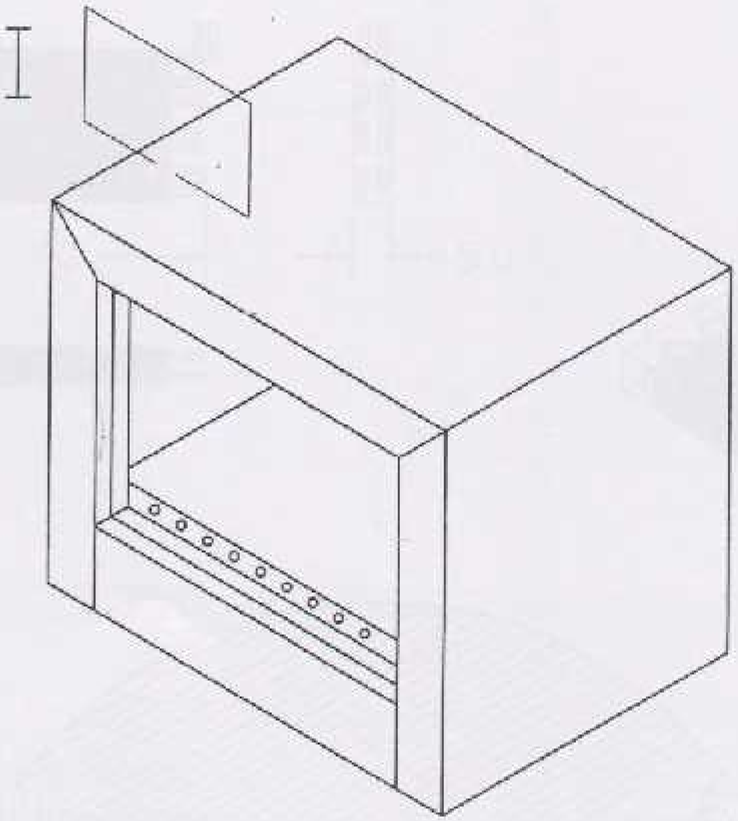
Name	Sign
Raed Nihad Abu Munshar	
Hamza Arafeh	
Mera Dadou	
Dr. Ishaq Sider	

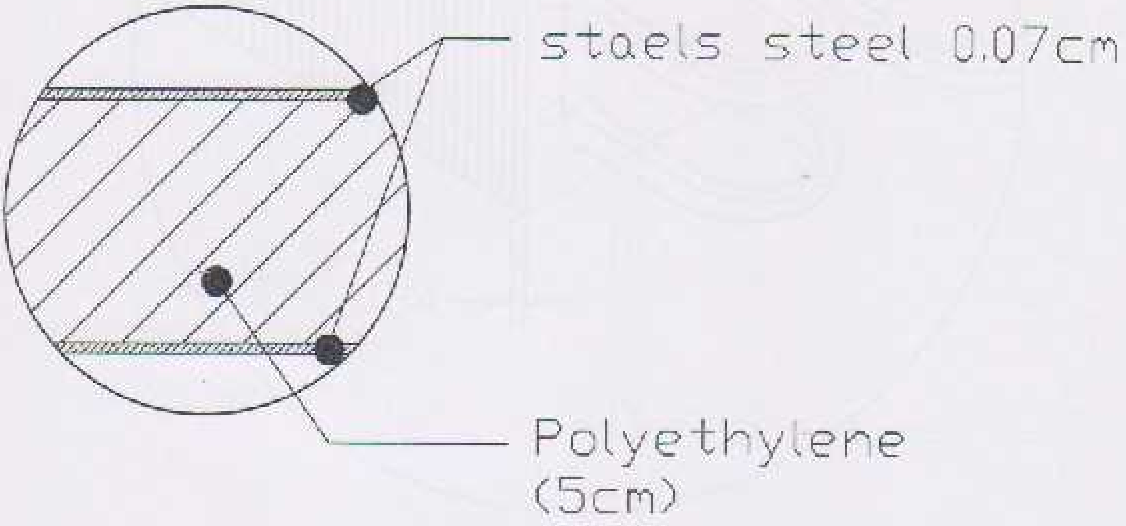
Design and building of refrigeration system for refrigerated chambers at three temperature levels

Door Insulation

Page #	Part #	Scale
22 - 34		1:10

CET-PPU





Section: (I)
Not to scale

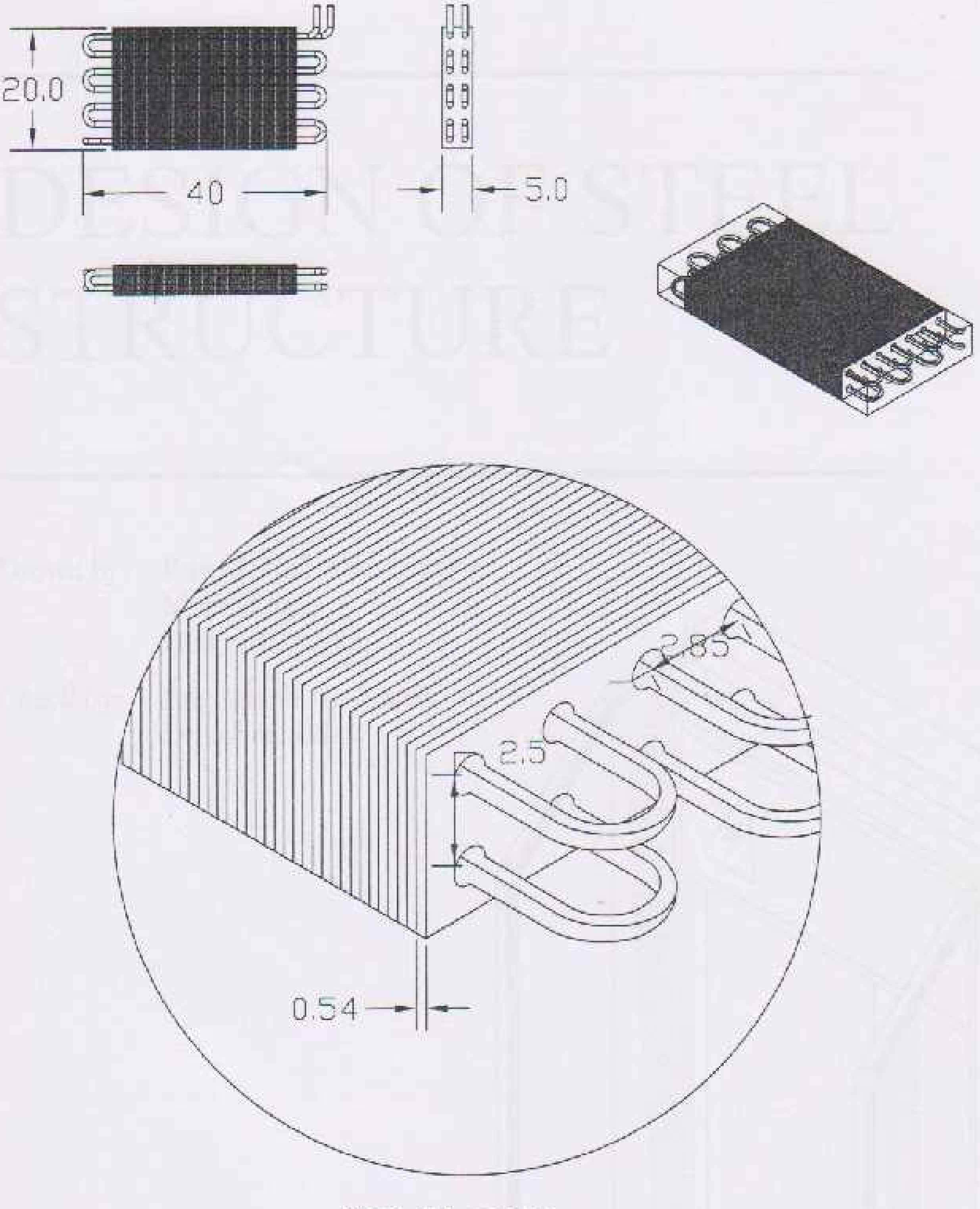
Design and building of refrigeration system for refrigerated chambers at three temperature levels

Name	Sign
Raed Nihad Abu Marshid	
Hamza Arafeh	
Mera Dadon	
Dr. Ishaq Sider	

Body insulation

Page #	Part #	Scale
23- 34		1:10

CET-PPU



Not to scale

Design and building of refrigeration system for
refrigerated chambers at three temperature levels

	Name	Sign
by	Raed Nbeel Abu Munshar	
	Hamza Arafeh	
	Mera Dadou	
by	Dr. Ishaq Sider	

Evaporator

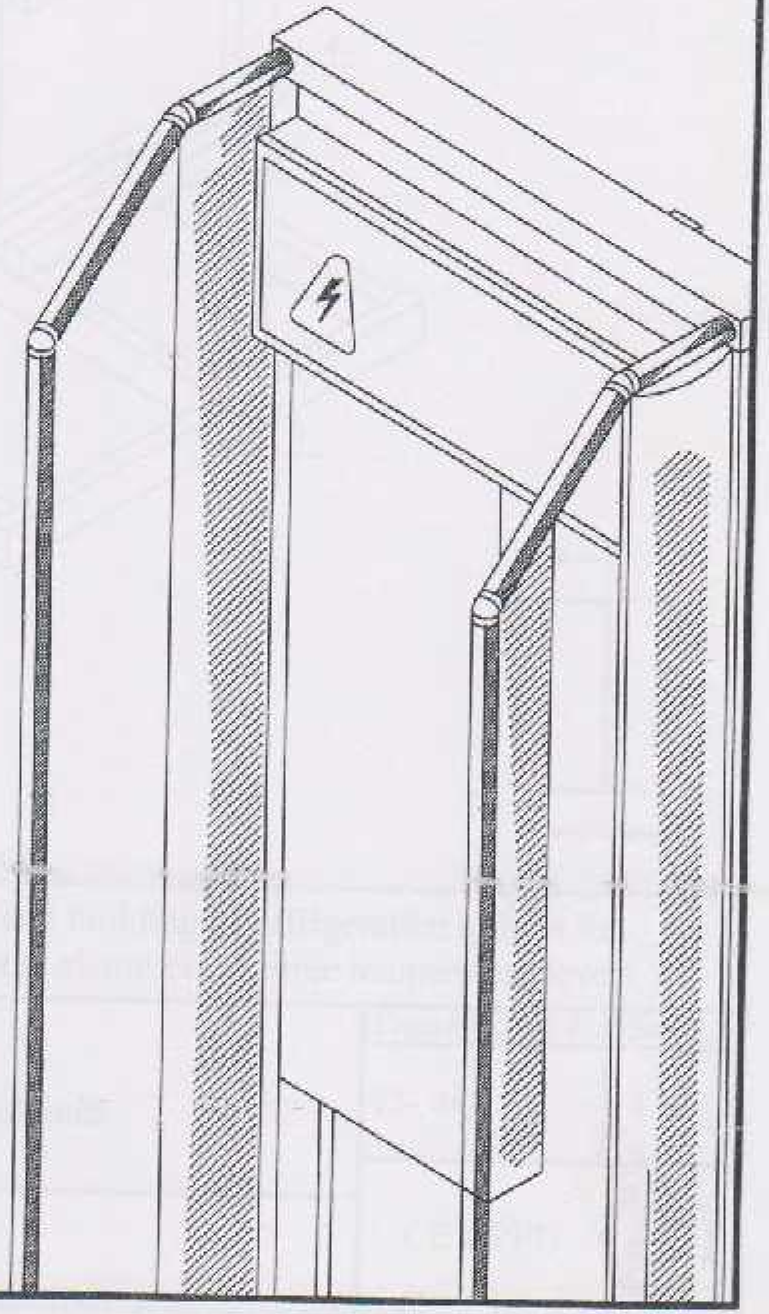
Page #	Part #	Scale
24- 34		1:10

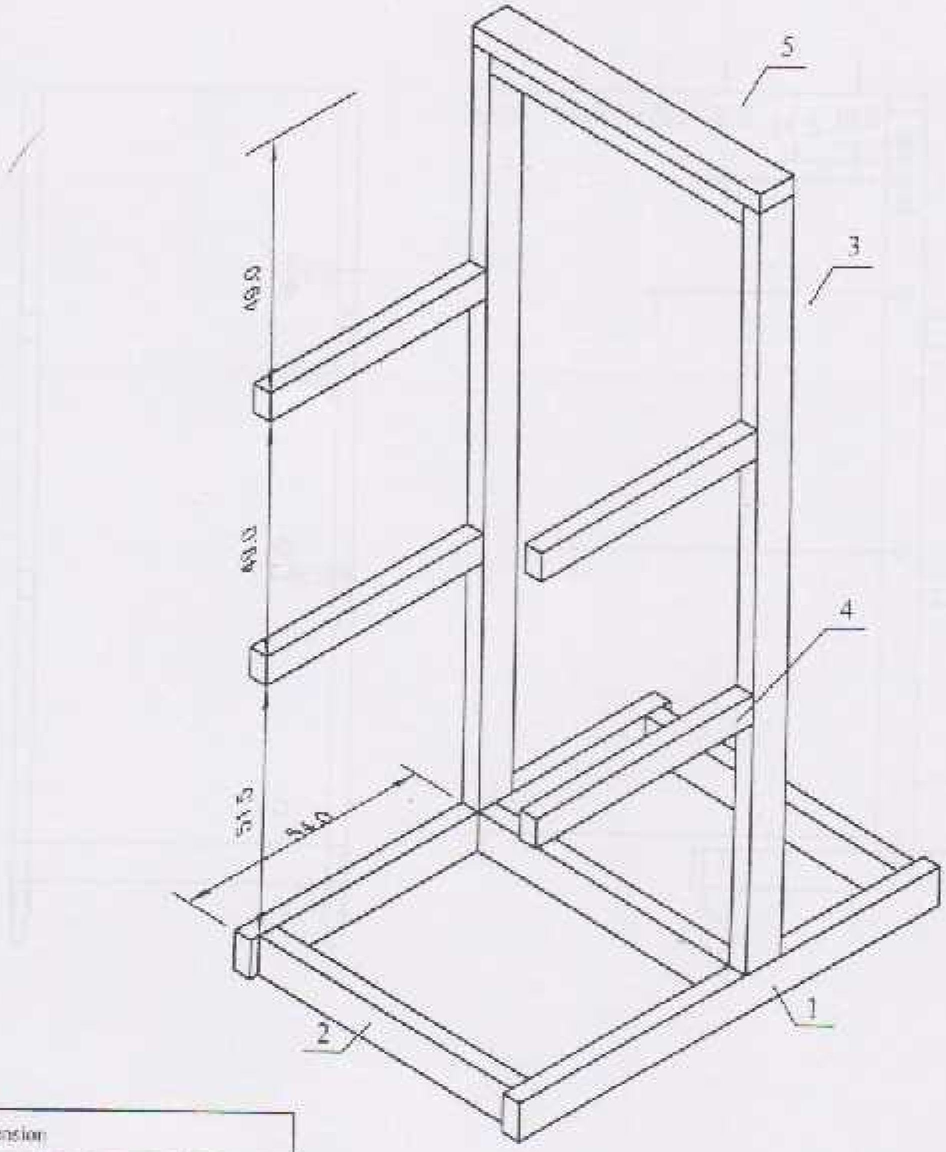
CET-PPU

DESIGN OF STEEL STRUCTURE

Drown by: Raed Nihad Abu Munshar
Mera Dadou
Hamzeh Arafeh

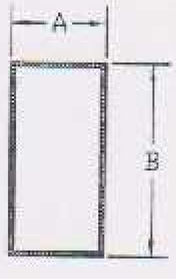
Check by: Eng.Ismail Al-Amleh
Dr. Ishaq Sider





Columns Dimension

#	Length (cm)	NO. Of Parts	B x A
1	100	2	8 x 4
2	62	4	8 x 4
3	158	2	8 x 4
4	52	4	6 x 4
5	70	1	8 x 4



SEC (A-A) : SC Steel

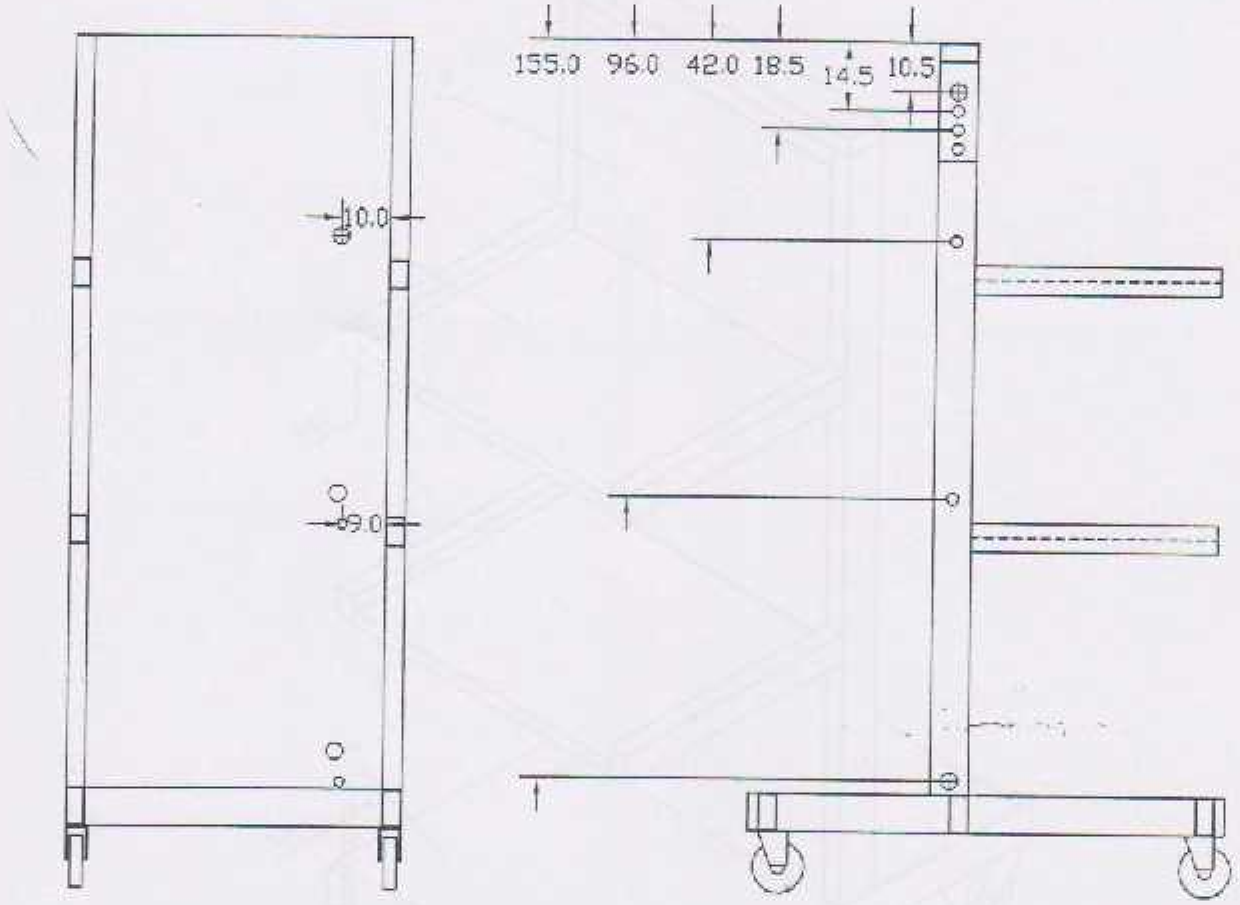
	Name	Sign
by	Raed Nihad Abu Munshar	
	Hamza Arafeh	
	Mara Dadna	
by	Dr. Ishaq Sider	

Design and building of refrigeration system for refrigerated chambers at three temperature levels

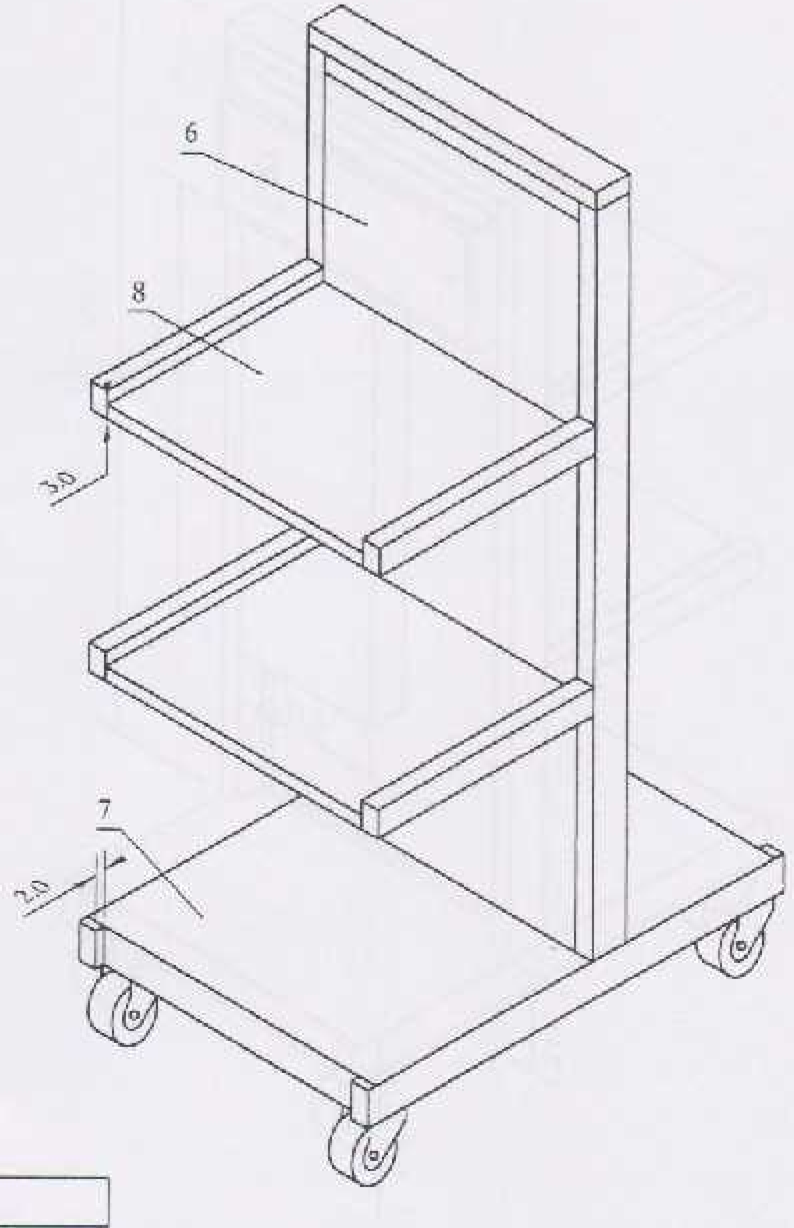
Columns

Page #	Part #	Scale
25- 34		1:16

CET-PPU



		Design and building of refrigeration system for refrigerated chambers at three temperature levels			
			Page #	Part #	Scale
Name	Sign	Location of electric holes	26- 34		1:16
Raed Nihad Abu Munshar					
Hamza Arafeh					
Mera Dadou			CET-PPU		
Dr. Ishaq Sider					



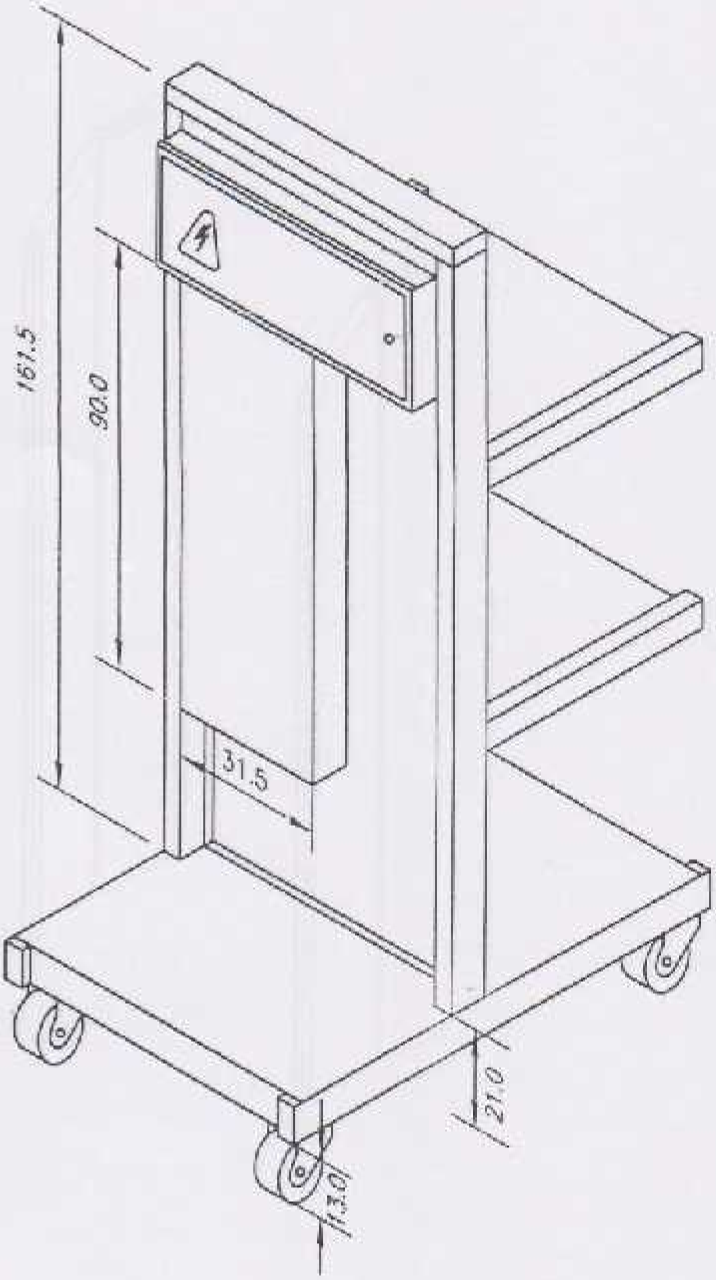
Plane Dimension			
	(L x W) cm	No. of Parts	Thickness (mm)
	160 x 64	1	1
	94 x 70	1	3
	65 x 55	2	2.5

	Name	Sign
by	Raed Nihad Abu Munshar	
	Hamza Arafeh	
	Mera Dadou	
by	Dr. Ishaq Sider	

Design and building of refrigeration system for refrigerated chambers at three temperature levels

	Page #	Part #	Scale
Shelves	27- 34		1:16

CET-PPU



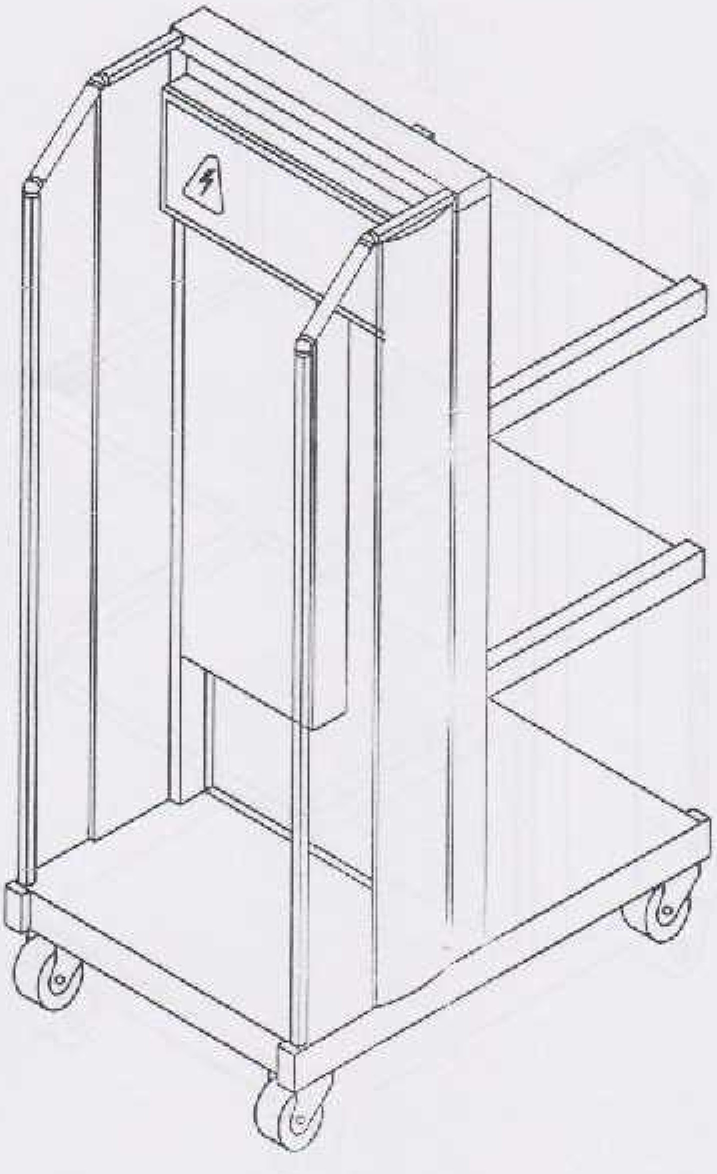
Name	Sign
Raed Nihad Abu Munshar	
Hamza Arafeh	
Mera Dadou	
Dr. Ishaq Sider	

Design and building of refrigeration system for refrigerated chambers at three temperature levels

Electric Box & Drown Plane

Page #	Part #	Scale
28- 34		1:16

CET-PPU



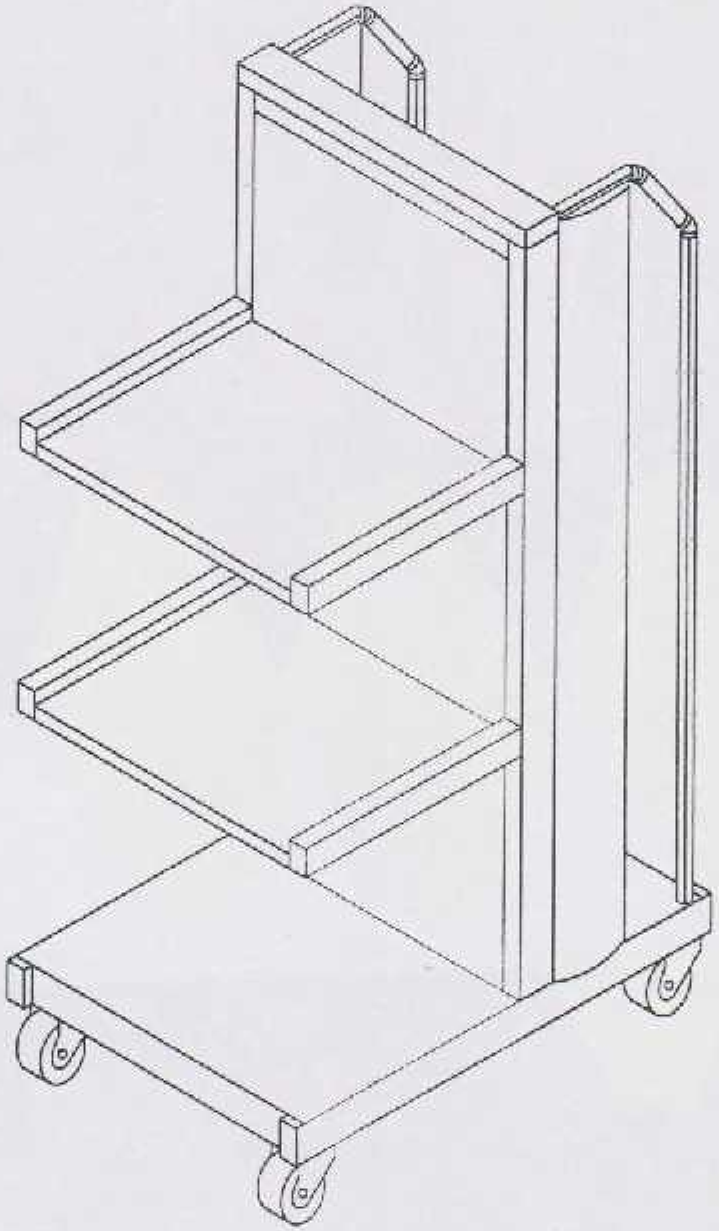
Design and building of refrigeration system for refrigerated chambers at three temperature levels

	Name	Sign
by	Raed Nihad Abu Munshar	
	Hamza Arafeh	
	Mera Dadou	
by	Dr. Ishaq Sider	

Back Side - Structure

Page #	Part #	Scale
29- 34		1:16

CET-PPU



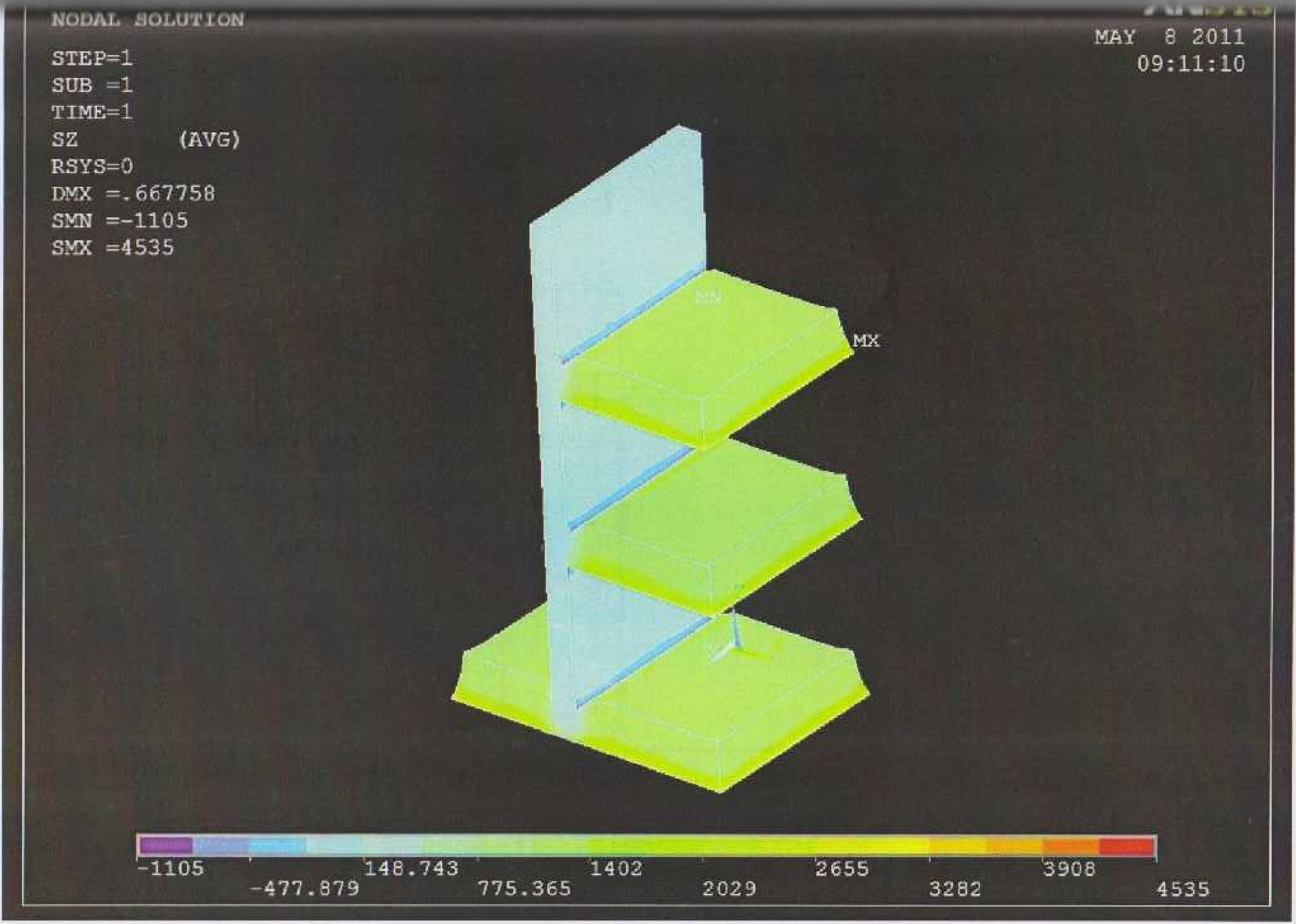
Design and building of refrigeration system for
refrigerated chambers at three temperature levels

	Name	Sign
by	Raed Nihad Abu Munshar	
	Hamza Arafeh	
	Mera Dadou	
by	Dr. Ishaq Sider	

Front Side - Structure

Page #	Part #	Scale
30 - 34		1:16

CET-PPU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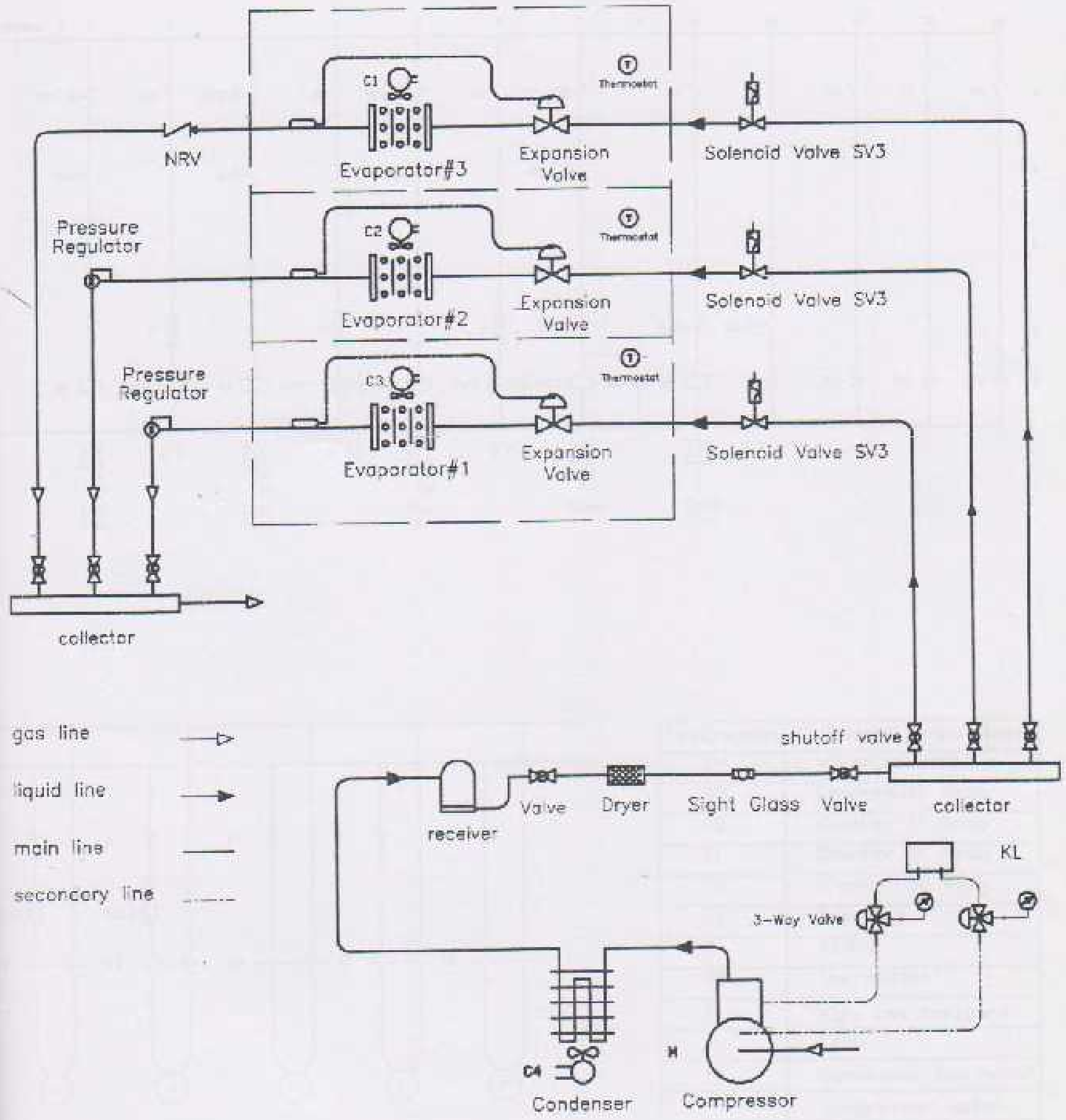
SUB =1
TIME=1
SZ (AVG)
RSYS=0
DMX =.667758
SMN =-1105
SMX =4535
MAY 8 2011
09:11:10
MN
MX
-1105
-477.879
148.743
775.365
1402
2029
2655
3282
3908
4535



Name	Sign	
Raed Nihad Abu Munshar		
Hamza Arafeh		
Mera Dadou		
Dr. Ishaq Sider		

Design and building of refrigeration system for refrigerated chambers at three temperature levels

AUTOMATION OF COMMERCIAL REFRIGERATION PLANT

Page #	Part #	Scale
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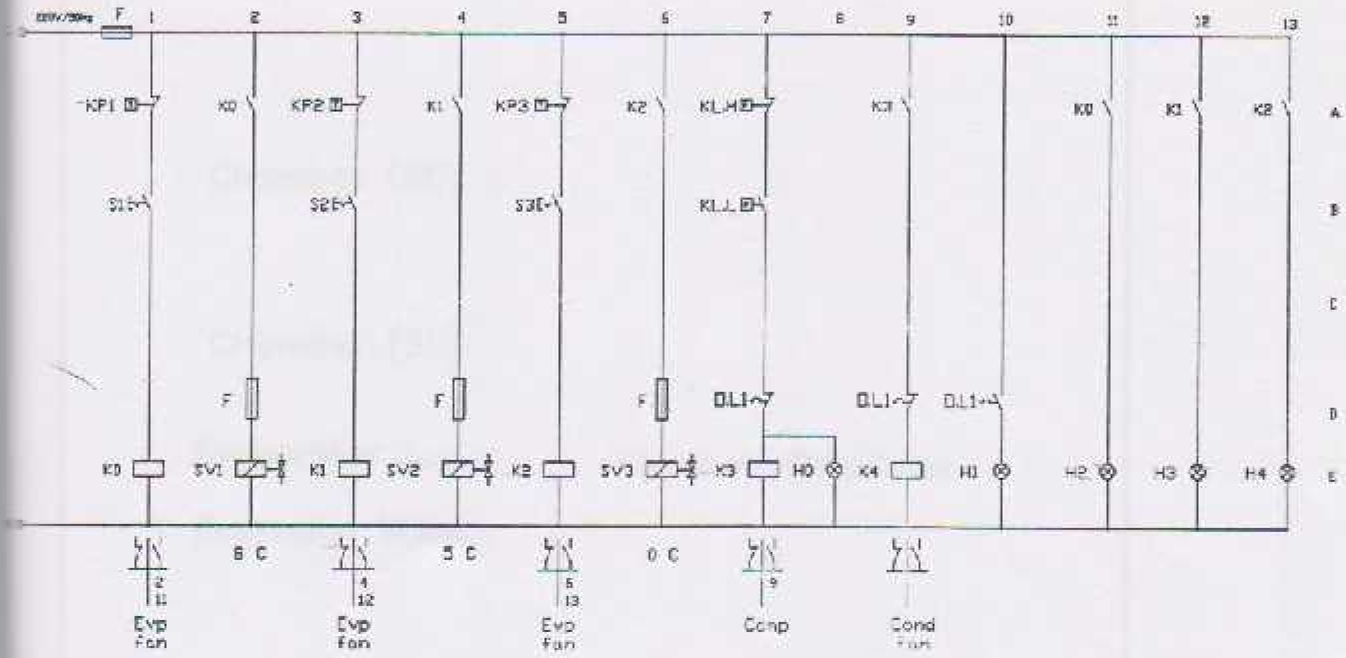

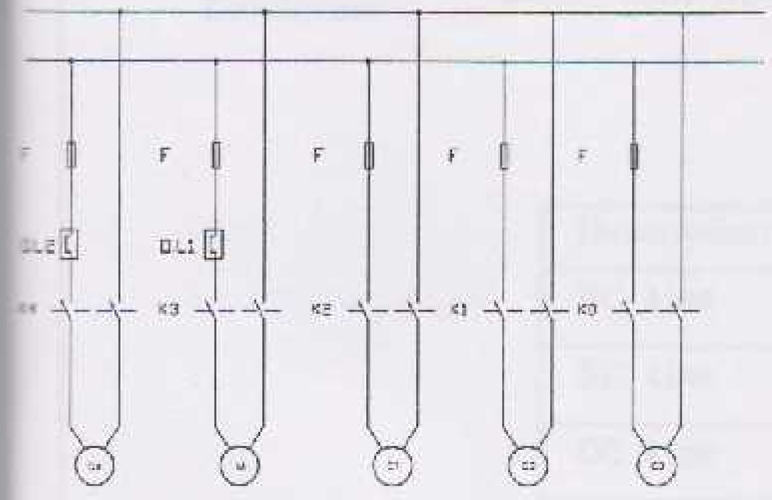

Designation	Component Description
SV	Volve solenoid
K3	Compressor Realy
K2	Chamber 0C Realy
K1	Chamber 5C Realy
K0	Chamber 8C Realy
OL	Overload
S	Switch
KP	Thermostat
KL	High, Low Pressure
H	Lamp
C	Condensor fan motor
M	Compressor motor
F	fuse

Design and building of refrigeration system for refrigerated chambers at three temperature levels

Name	Sign
by Raed Nihad Abu Munshar	
Hamza Arafeh	
Mera Dadou	
by Dr. Ishaq Sider	

Wiring Diagram

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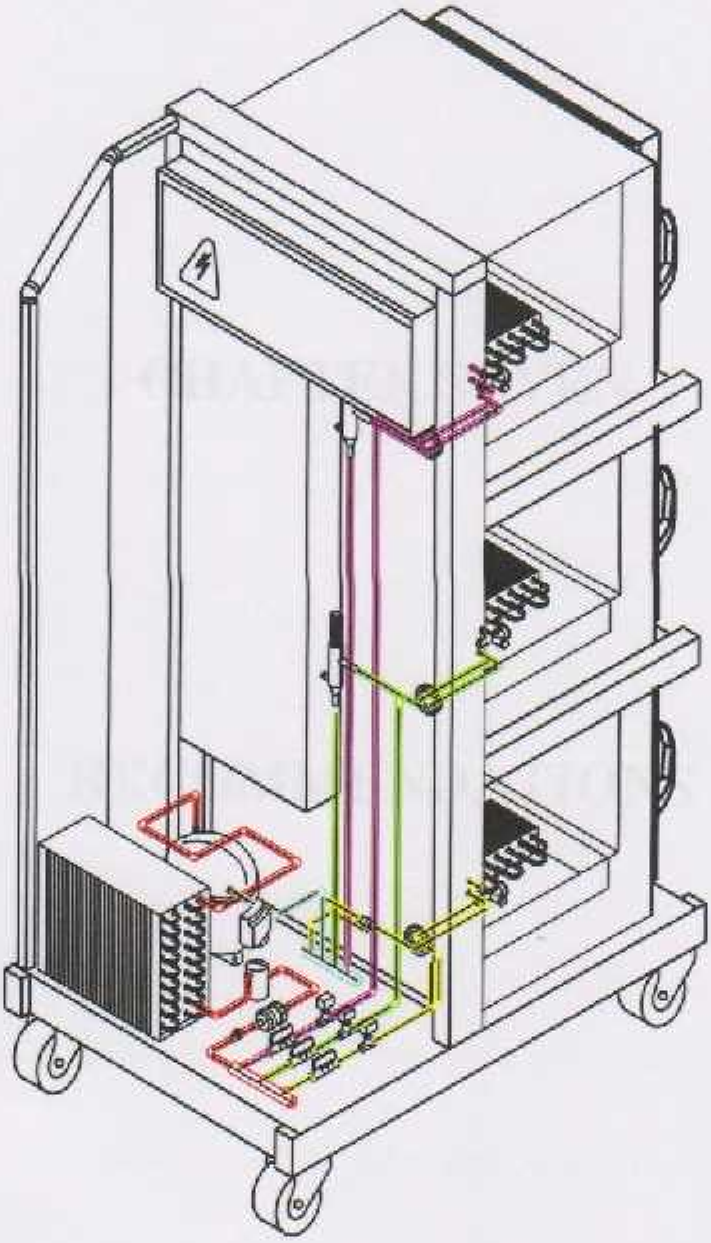

Design and building of refrigeration system for refrigerated chambers at three temperature levels

	Name	Sign
ing by	Raed Nihad Abu Munshar	
	Hamza Arafeh	
	Mera Dadou	
ised by	Dr. Ishaq Sider	

Device

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CHAPTER SEVEN

RECOMMENDATIONS

CHAPTER SEVEN

7.1 Recommendations

The working team has the following recommendations:

1. For such practical project the required equipment must be available at the first quarter of the semester.
2. It is an important issue to use more program and software during the courses and to pay attention to use them in calculation and design.
3. The device has been tested and it can be used as an educational tool for the laboratory and performs many experiments.
4. The device can be improved in the future by other students specially in the control of refrigeration system side.
5. It is an important issue to pay attention to the refrigerant development.
6. The suggested project has useful applications and the procedures will be useful for students.

APPENDEX A

APPENDEX A

TABLE A-1 Maximum and minimum temperature for Hebron city

Month	Max. Temp. C°	Min. Temp. C°
Jan	10.3	3
Fep	11.5	4.7
Mar	14.6	6.5
Apri	19.6	9.9
May	25.6	13.2
Jun	26	15.8
Jul	28	17
Aug	38	18
Sep	29	15
Oct	28	14
Nov	22	9.9
Dec	12	5.6

TABLE A-2 Thermal Conductivity of Materials

Material	Description	Thermal Conductivity (*k*) W/m K	Thermal Conductance (C) W/m^2 K
Masonry	Brick, common	0.72	
	Brick, face	1.30	
	Concrete, mortar or plaster	0.72	
	Concrete, sand aggregate	1.73	
	Concrete block		
	Sand aggregate 100 mm		7.95
	Sand aggregate 200 mm		5.11
	Sand aggregate 300 mm		4.43
Woods			
	Maple, oak, similar hardwoods	0.16	
	Fir, pine, similar softwoods	0.12	
	Plywood 13 mm		9.09
	Plywood 1.9mm		6.08
Roofing	Asphalt roll roofing		36.91
	Built-up roofing 9 mm		17.03
Insulating materials	Blanket or batt, mineral or glass fiber	0.039	
	Board or slab		
	Cellular glass	0.058	
	Corkboard	0.043	
	Glass fiber	0.036	
	Expanded polystyrene (smooth)	0.029	
	Expanded polystyrene (cut cell)	0.036	
	Expanded polyurethane	0.025	
Loose fill			
	Milled paper or wood pulp	0.039	
	Sawdust or shavings	0.065	
	Mineral wool (rock, glass, slag)	0.039	
Glass	Single pane		6.42
	Two pane		2.61
	Three pane		1.65
	Four pane		1.19
Metal	Stainless steel	18	
	Aluminum	202	
	Cooper	386	
	Galvanized steel	20	

TABLE A-3 Properties of common foods and substances

Food	Specific Heat Capacity above Freezing			Specific Heat Capacity below Freezing		
	(Btu/lb°F)	(KJ/kg°C)	(Kcal/kg°C)	(Btu/lb°F)	(KJ/kg°C)	(Kcal/kg°C)
Apples	0.87	3.64	0.87	0.42	1.76	0.42
Apricots. fresh	0.88	3.68	0.88	0.43	1.8	0.43
Avocados	0.72	3.01	0.72	0.37	1.55	0.37
Bananas	0.8	3.35	0.8	0.4	1.67	0.4
Beef, carcass	0.68	2.85	0.68	0.48	2.01	0.48
Beef, flank	0.56	2.34	0.56	0.32	1.34	0.32
Beef, round	0.74	3.1	0.74	0.38	1.59	0.38
Meat	0.76	3.14	0.76	0.39	1.6	0.39
Chicken, hens	0.65	2.72	0.65	0.44	1.84	0.44
Coconut, meat and milk	0.68	2.85	0.68	0.45	1.88	0.45
Cucumber	0.98	4.1	0.98	0.45	1.88	0.45
Lemons	0.91	3.81	0.91	0.44	1.84	0.44
Lemon juice	0.92	3.85	0.92	0.44	1.84	0.44

h_o [W/m^2.°C]	22.7[W/m^2.°C]	@38C & 1 atm
h_i [W/m^2.°C]	9.37[W/m^2.°C]	@38C & 1 atm

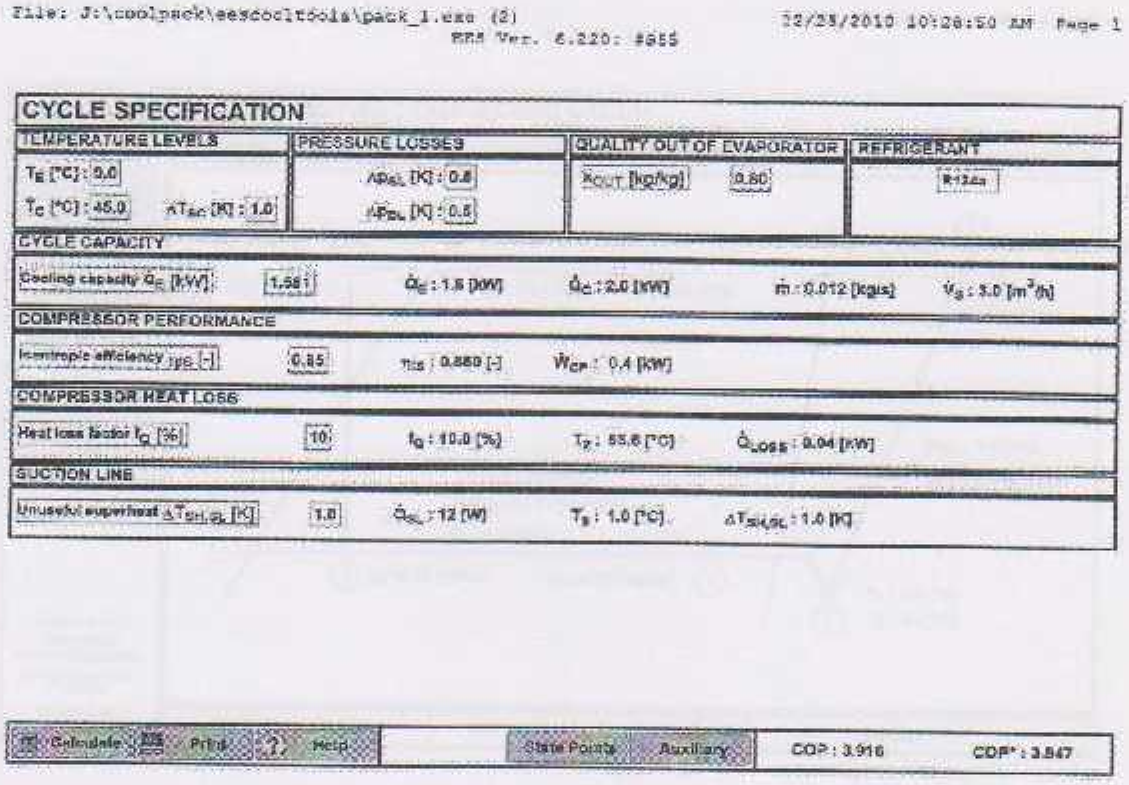
TABLE A-4 Air change per hour

Kind of room or building	Air Change [times/hr]
Room with no windows or exterior door	0.5
Room with windows or exterior door on one side only	1.0
Room with windows or exterior door on two side only	1.5
Room with windows or exterior door on three side only	2.0
Entrance halls	2.0
Factories, machine shops	1.0-1.5
Recreation room, assembly rooms, gymnasium	1.5
Home, apartment, offices	1.0-2.0
Class rooms, dining room, lounges, toilets, hospital room, kitchen, laundries, ballrooms, bathrooms	1.0-2.0
Stores, public buildings	2.0-3.0
Toilets, auditorium	3.0

TABLE A-5 Specific heat of packaging material

Packaging Material	Specific Heat [kJ/kg.°C]
Wood	2.3
Stainless steel	0.5
carton	0.34
Plastic	1.6
Aluminum	0.85

TABLE A-6 Cycle specification (Cool pack)



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CYCLE SPECIFICATION

TEMPERATURE LEVELS

T_E [°C]: 0.0

T_C [°C]: 45.0

ΔT_{SC} [K]: 1.0

PRESSURE LOSSES

ΔP_{SL} [K]: 0.8

ΔP_{DL} [K]: 0.5

QUALITY OUT OF EVAPORATOR

x_{OUT} [kg/kg]: 0.80

REFRIGERANT

R134a

CYCLE CAPACITY

Cooling capacity Q_E [kW]: 1.561

$\dot{Q}_E$: 1.6 [kW]

$\dot{Q}_C$: 2.0 [kW]

$\dot{m}$: 0.012 [kg/s]

$\dot{V}_S$: 5.0 [m^3/h]

COMPRESSOR PERFORMANCE

Isentropic efficiency η_{IS} [-]: 0.85

η_{IS} : 0.850 [-]

$\dot{W}_{CP}$: 0.4 [kW]

COMPRESSOR HEAT LOSS

Heat loss factor f_Q [%]: 10

f_Q : 10.0 [%]

T_2 : 55.6 [°C]

$\dot{Q}_{LOSS}$: 0.04 [kW]

SUCTION LINE

Unuseful superheat $\Delta T_{SH,SL}$ [K]: 1.0

$\dot{Q}_{SL}$: 12 [W]

T_1 : 1.0 [°C]

$\Delta T_{SH,SL}$: 1.0 [K]

COP : 3.916

COP* : 3.847

TABLE A-7 Cycle analysis (Cool pack)

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                                     EES Ver. 8.220: #955
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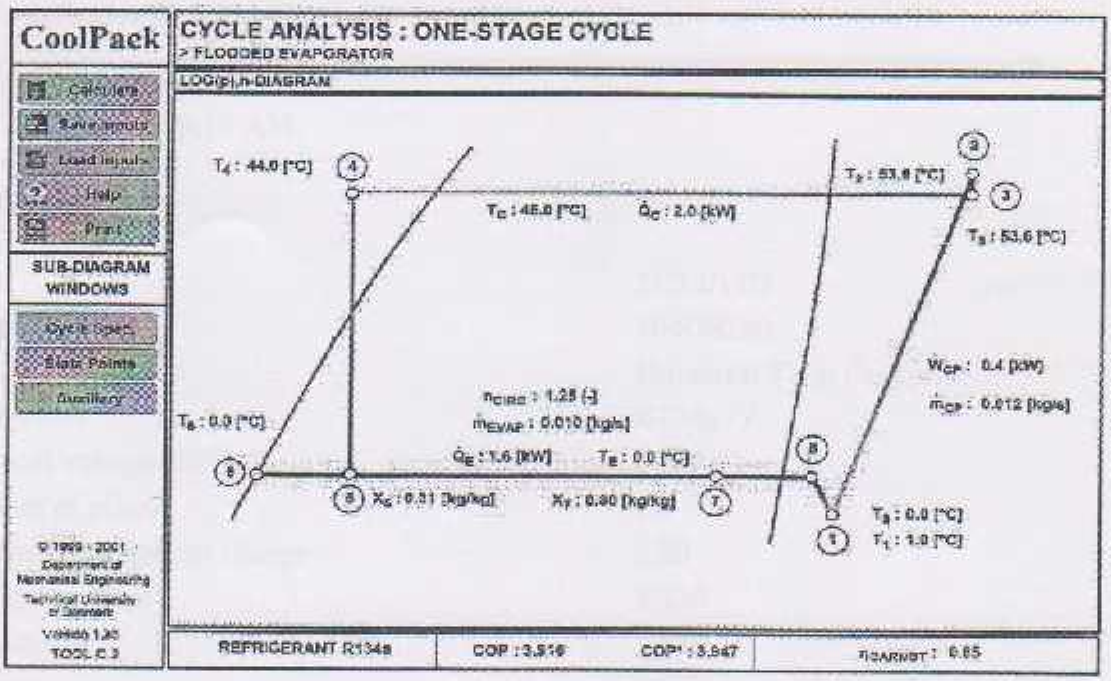


TABLE A-8 RS+3 Selection Report

Danfoss

RS+3 Selection Report

Dec 28, 2010 11:09:14 AM

General data

Type	SC12/12G
Code no.	104G8280
Compressor design	Universal Twin Compressor
Refrigerant	R134a / /
Nominal voltage (50 Hz)	220 240
Number of phases	1
Maximum refrigerant charge	2.20
Free gas volume	3,020
Oil quantity	1,120
Oil type	POE
Approvals	EN60335-2-34

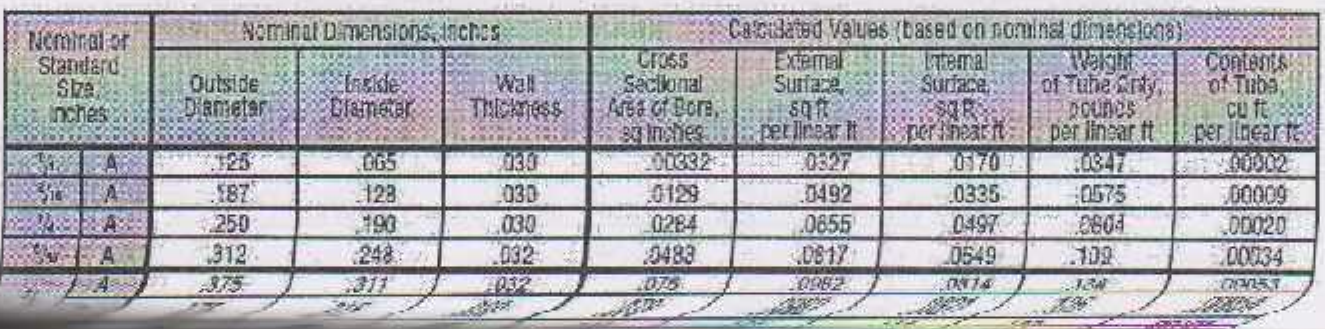
Operating Conditions

Evaporating temp. (dew)	0.0 °C
Condensing temp. (dew)	45.0 °C
Evap superheat:	5.0 K
Total superheat:	5.0 °C
Subcooling:	0.0 °C
Refrigerant:	R134a
Mains Voltage:	220-240, 1ph.
Cooling Capacity	1,740 W
Power Input	833 W

Table A-9 Recommended Refrigerant Velocities

Line	Refrigerant		Recommended Velocity (m/s)
Suction	R12	R22	8-12
	R404A	R410A	
Discharge	R12	R22	10-18
	R404A	R410A	

Table A-10 Dimensions and Physical Characteristics of Copper Tube ACR (Air - Conditioning and Refrigeration Fields Service).



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
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

TABLE A-11 modified correlation of grimson for heat transfer in tube banks

$\frac{S_p}{d}$	$\frac{S_n}{d}$ 1.25		1.5		2.0		3.0	
	C	*n*	*C*	*n*	*C*	*n*	*C*	*n*
				In line				
1.25	0.386	0.592	0.305	0.608	0.111	0.704	0.0703	0.752
1.5	0.407	0.586	0.278	0.620	0.112	0.702	0.0753	0.744
2.0	0.464	0.570	0.332	0.602	0.254	0.632	0.220	0.648
3.0	0.322	0.601	0.396	0.584	0.415	0.581	0.317	0.608
				Staggered				
0.6	—	—	—	—	—	—	0.236	0.636
0.9	—	—	—	—	0.495	0.571	0.445	0.581
1.0	—	—	0.552	0.558	—	—	—	—
1.125	—	—	—	—	0.531	0.565	0.575	0.560
1.25	0.575	0.556	0.561	0.554	0.576	0.556	0.579	0.562
1.5	0.501	0.568	0.511	0.562	0.502	0.568	0.542	0.568
2.0	0.448	0.572	0.462	0.568	0.535	0.556	0.498	0.570
3.0	0.344	0.592	0.395	0.580	0.488	0.562	0.467	0.574

TABLE A-12 Properties of air at atmospheric pressure

T (K)	ρ (kg/m³)	c_p (J/kg·K)	μ (kg/m·s)	ν (m²/s)	k (W/m·K)	α (m²/s)	Pr
				Air			
100	3.6010	1.0266×10^3	0.6924×10^{-5}	1.923×10^{-6}	0.009 246	0.0250×10^{-4}	0.768
150	2.3675	1.0099	1.0283	4.343	0.013 735	0.0574	0.756
200	1.7684	1.0061	1.3289	7.490	0.018 09	0.1016	0.739
250	1.4128	1.0053	1.5990	11.310	0.022 27	0.1568	0.722
300	1.1774	1.0057	1.8462	15.690	0.026 24	0.2216	0.708
350	0.9980	1.0090	2.075	20.76	0.030 03	0.2983	0.697
400	0.8826	1.0140	2.286	25.90	0.033 65	0.3760	0.689
450	0.7833	1.0207	2.484	31.71	0.037 07	0.4636	0.683
500	0.7048	1.0295	2.671	37.90	0.040 38	0.5564	0.680
550	0.6423	1.0392	2.848	44.27	0.043 60	0.6532	0.680
600	0.5879	1.0551	3.018	51.34	0.046 59	0.7512	0.682
650	0.5430	1.0635	3.177	58.51	0.049 53	0.8578	0.682
700	0.5030	1.0752	3.332	66.25	0.052 30	0.9672	0.684
750	0.4709	1.0856	3.481	73.91	0.055 09	1.0774	0.686
800	0.4405	1.0978	3.625	82.29	0.057 79	1.1951	0.689
850	0.4149	1.1095	3.765	90.75	0.060 28	1.3097	0.692
900	0.3925	1.1212	3.899	99.3	0.062 79	1.4271	0.696
950	0.3716	1.1321	4.023	108.2	0.065 25	1.5510	0.699
1000	0.3524	1.1417	4.152	117.8	0.067 52	1.6779	0.702
1100	0.3204	1.160	4.44	138.6	0.073 2	1.969	0.704
1200	0.2947	1.179	4.69	159.1	0.078 2	2.251	0.707
1300	0.2707	1.197	4.93	182.1	0.083 7	2.583	0.705
1400	0.2515	1.214	5.17	205.5	0.089 1	2.920	0.705
1500	0.2355	1.230	5.40	229.1	0.094 6	3.266	0.705
1600	0.2211	1.248	5.63	254.5	0.100	3.624	0.705
1700	0.2082	1.267	5.85	280.5	0.105	3.977	0.705
1800	0.1970	1.287	6.07	308.1	0.111	4.379	0.704
1900	0.1858	1.309	6.29	338.5	0.117	4.811	0.704
2000	0.1762	1.338	6.50	369.0	0.124	5.260	0.702
2100	0.1682	1.372	6.72	399.6	0.131	5.680	0.703
2200	0.1602	1.419	6.93	432.6	0.139	6.115	0.707
2300	0.1538	1.482	7.14	464.0	0.149	6.537	0.710
2400	0.1458	1.574	7.35	504.0	0.161	7.016	0.718
2500	0.1394	1.688	7.57	543.0	0.175	7.437	0.730

TABLE A-13 Ratio of N rows deep to that for 10rows deep

N	1	2	3	4	5	6	7	8	9	10
Ratio for staggered tubes	0.68	0.75	0.83	0.89	0.92	0.95	0.97	0.98	0.99	1.0
Ratio for in-line tubes	0.64	0.8	0.87	0.9	0.92	0.94	0.96	0.98	0.99	1.0

TABLE A-14 Desired cooling time