Palestine Polytechnic University

College of Engineering and Technology

Mechanical Engineering Department

Graduation Project

Design of a Refrigerator that Operates on CO_2 as a Refrigerant

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جامعة بوليتيكنيك فلسطين

الخليل – فلسطين

كلية الهندسة والتكنولوجيا

دائرة الهندسة الميكانيكيه

اسم المشروع

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أسماء الطلبة

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بناء على نظام كلية الهندسة والتكنولوجيا وإشراف ومتابعة المشرف المباشر على المشروع وموافقة أعضاء اللجنة الممتحنة ، تم تقديم هذا المشروع إلى دائرة الهندسة الميكانيكية ذلك للوفاء بمتطلبات درجة البكالوريوس في الهندسة تخصص هندسة ميكانيكيه فرع هندسة تكييف وتبريد .

توقيع مشرف المشروع

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توقيع اللجنة الممتحنة

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توقيع رئيس الدائرة

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المشرف

د. اسحق سدر

تقرير مشروع التخرج

مقدم إلى دائرة الهندســـة الميكانيكيه في كلية الهندسة و التكنولوجيا

جامعة بوليتيكنيك فلسطين

للوفاء بجزء من متطلبات الحصول على

درجة البكالوريوس في الهندسة تخصص هندسة تكييف و تبريد

جامعة بوليتيكنيك فلسطين

الخليل – فلسطين

حزيران – ٢٠٠٨

Dedication

To our dear parents and families...

To whom who have added anything to the science........................

To whom who have taught us any letter, word or information...........

To our instructors and colleagues...

To whom we love...

Acknowledgment

Here as we finished our introduction of project we stop for amoment to thank every body who has helped us to complete this work.

First we want to thank our supervisor Dr. Ishaq Seder who gave us a lot of his time and experience in order to complete the project and gave us the opportunity to start scientific life and methodology in the real life by asking us to do this work

Abstract

In this project a tempt of designing a refrigerator which using carbon dioxide (CO_2) as aworking fluid is to be performed . Carbon dioxide properties and its effects as a refrigerant used in the refrigeration cycle , the environmental and economic effects , coefficient of performance of cycle using CO_2 refrigerant , power consumption ,design and selecting of the required components of the cycle are to be analyzed and studied .

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Chapter One

Introduction

This chapter talks about the project adjectives , the chemical and physical properties for the carbon dioxide refrigerant that used as a working fluid in the refrigeration cycle and its sources , and comparing that refrigerant with other refrigerants.
At the end of this chapter suggest the refrigeration cycle would be used .

1.1 Overview:

Carbon dioxide is a chemical compound composed of two oxygen atoms bonded to a single carbon atom. It is a gas at standard temperature and pressure and exists in earth's atmosphere in this state. It is currently at a globally averaged concentration of approximately 375 part per million (ppm) by volume in the earth's atmosphere, although this varies both by location and time. Carbon dioxide's chemical formula is CO_2.

R-744 is chemical reference for carbon dioxide (CO_2) used as refrigerant which will be used as working fluid in the transcritical refrigeration cycle that will be constructing it.

1.2 The Project Objectives :

- Design a 0.78 kW cooling capacity refrigerating system utilizing CO_2 as refrigerant.
- Design and selecting of the required components of the cycle are to be analyzed and studied.

1.3 Carbon Dioxide Gas As Refrigerant

1.3.1 Historical background

Carbon dioxide was one of the first gases to be described as a substance distinct from air which a small but important constituent of it with typical concentration about 0.038% and contains as much as 4% carbon dioxide. [1]

In general, it is exhaled by animals and utilized by plants during photosynthesis. Additional carbon dioxide is created by the combustion of fossil fuels or vegetable matter, among other chemical processes.

It was first liquefied (at elevated pressures) in 1823 by Humphry Davy and Michael Faraday, the earliest description of solid carbon dioxide was given by Charles Thilorier, who in 1834 opened a pressurized container of liquid carbon dioxide, only to find that the cooling produced by the rapid evaporation of the liquid yielded a "snow" of solid CO_2. [2]

It was already used as a refrigerant in the mid-nineteenth century and its use steadily increased, reaching a peak in the 1920s. After that, the natural refrigerant was displaced by chlorofluoro-carbons (CFCs) that operated at much lower pressures.

Due to the impact on the environment of CFCs, and later HFCs, CO_2 was re-evaluated as an alternative refrigerant in the late 1980s. [2]

Research activities for ozone depletion potential (ODP) and the global warming potential (GWP) of the synthetically refrigerants are initiated since the 1990s.[2]

At the beginning of 1982s to the end of 1993s many applications have been identified, where CO_2 is as good as or better than other refrigerants, e.g. as secondary refrigerant or as refrigerant in the lower stage of a cascade cycle.[3]

The first commercial use of transcritical CO_2 Systems, in hot water heat pumps starting from 1999s.

Producing Numeric simulation of an integrated CO_2 cooling system, modeling and simulation of refrigeration systems with the natural refrigerant CO_2 in technical university Hamburg–Harburg in 2000s.[5]

A new way to contain the high pressure was discovered by Professor Gustav Lorentz and his group of researchers from Norway, giving way to the application of transcritical R744 to different heating and cooling systems, however, for higher temperature applications, the major challenge with CO_2 is low energy efficiency of transcritical system due to the large throttling loss.

Recent research on carbon dioxides is pushed for mobile, in fuel cell electric vehicles, automotive air-conditioning and refrigeration and has focused on the development of a transcritical cycle.

1.3.2 Carbon dioxide sources:

- Naturally producing by all animals, plants.
- Generating as a byproduct of the combustion of fossil fuels or vegetable matter.
- Obtaining as output by volcanoes.
- Forming by combustion and biological processes including decomposition of organic material.
- Obtaining from air distillation but this yields only very small quantities of CO_2.
- Reaction between most acids and most metal carbonates:

  $H2SO4 + CaCO3 \rightarrow CaSO4 + H2CO3$

  The H2CO3 then decomposes to water and CO_2 .[6]

- Reaction between methane and oxygen.

  $$CH_4 + 2\ O_2 \rightarrow CO_2 + 2\ H_2O$$

- Forming by thermal decomposition of CaCO3.
- Obtaining directly from natural carbon dioxide gas wells.
- Coming from the boilers at the bottom of the tall chimney.
- Coming from houses and using electricity.

- Resulting from combustion processes include the reaction of a hydrocarbon with oxygen as by-product flue gas stream.

1.3.3 Reasons for selecting carbon dioxide refrigerant (CO_2, R-744)

2. Benefits for environmental:

- Preventing the air from pollution in the case of leakage of the refrigerant into the atmosphere.
- Preventing the ozone damaging effect and reducing emissions of greenhouse gases badly effect on environmental.
- Environmentally clean, friendly to use.

3. commercial Benefits:

- Little drawbacks include high cost of carbon dioxide (CO_2, R-744) refrigerant, which means very cheap when bought in industrial quantities, inexpensive compared to costly synthetics hydrocarbons.
- Space savings for piping arrangements (self-contained system), reducing costs of piping arrangements and insulation.
- Increasing Energy efficiency, i.e. offers up to 30% energy savings compared with traditional heat pump systems for the same heating and cooling.

4. **Benefits for technology:**

- It is a naturally occurring substance that can be applied as a working fluid in different heating and cooling applications, due to its excellent heat transfer properties and its high volumetric cooling capacity.
- Can be applied in most car manufacturers in Europe, Korea, Japan and the US All system and component suppliers, Institutes and universities all over the world, related industries are either developing or are already in production with CO_2.
- Reduced size and weight.

5. **Benefits for education and research:**

Recently the scientists tends to use carbon dioxide refrigerant widely in the refrigeration world, so there are several continuous education and researches deal with this subject to complying with new technology and eliminate using of other refrigerant gases because of their harmful to environmental

1.3.4 Physical and chemical properties of carbon dioxide as refrigerant

R744 is the chemical reference for carbon dioxide used as refrigerant has the following properties[7]:

- It freezes at -78.5 °C to form carbon dioxide snow at atmospheric pressure.
- Liquid density: 1032 kg/m^3.
- Colorless, Odorless, Taste is neutral, Versatile material.
- Liquefied, solidified Gases (can obtain dry ice).
- Has no liquid state at pressures below 5.1 atm.
- In an aqueous solution it forms carbonic acid, which is too unstable to be easily isolated.
- Latent heat of vaporization: 571.08 kJ/kg.
- Vapor pressure: 58.5 bar.
- Carbon dioxide is soluble in water.
- Extremely low global warming potential, the global warming of CO_2 is extremely low about 1/1300of HCF 134a.
- Reactivity does not react with oxygen, but will combine with other elements and compounds.
- Non-toxic :

  Moderate concentrations, given the small system charge used for heating and Cooling application ants .
- Non-flammable :

  Presents no risks in case of interaction with other substances or chemical blends. CO_2 is applied in many fire extinguishers.
- Non-ozone-depleting :

  Does not contribute in any sense to the depletion of the ozone layer.
- Environmentally friendly:

  With a Global Warming Potential = 1, which reduces significantly the amount of greenhouse gas (GHG) emissions released to the atmosphere, which are at the origin of climate change.

The physical properties are shown in the following (figure 1.1) of pressure. Temperature phase diagram for CO_2 [8].

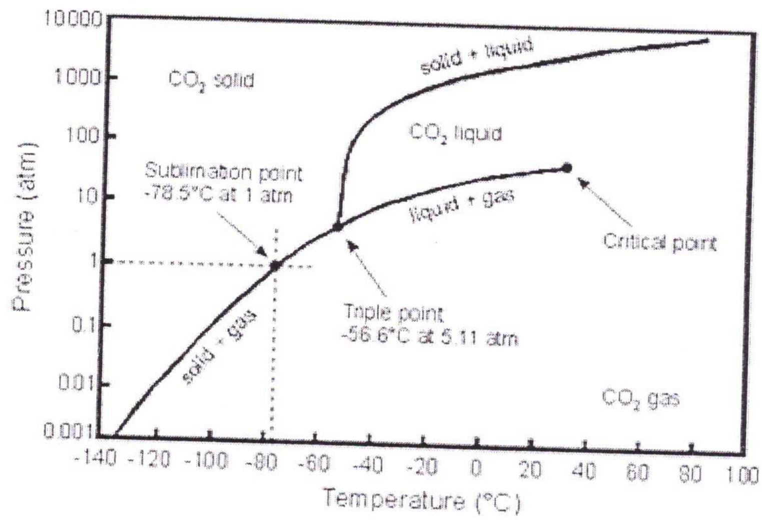


Figure (1.1): temperature pressure phases for CO_2

From (figure 1.1) specific characteristics of CO_2 are relatively high working pressures, low critical point (+31°C), relatively high triple point (-56.6°C) at 5.1 atm and high density compared to air. The latter may represent a certain danger to human health if CO_2 should leak uncontrolled into confined.

1.4 Comparison Between (R-744) Refrigerant And (R-134a) Refrigerant:

The following table (1.1) indicates some comparison issues between R744 and R134a refrigerants.

Table (1.1): comparison table issues between R744 & R134a refrigerants.

Refrigerant	**R134a**	**R744 / CO2**
Natural	no	yes
Flammable	no	no
Combustion Product	Toxic	Non toxic
Critical Temp.°C	101	31
ODP(Ozone Depletion Potential)	0	0
Pressure. at 21°C in bar	5.9	59
GWP (Global Warming Potential)	1300	1

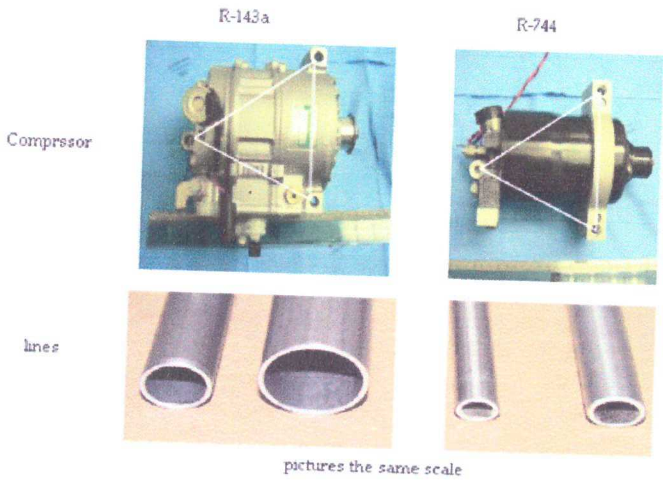
Specific Comparison for:

1- Packaging:

- Heat exchangers will be smaller at equal performance (capacity).
- Compressor and lines will be smaller (higher volumetric capacity).
- Integration of accumulator and suction line heat exchanger possible (either stands alone or in place of current Integrated Receiver Dryer).

- The packaging of R744 system into space used currently for HFC-134a Offers a wide range.
- The evaporator for CO_2 systems is 25% smaller than the one for R-134a systems.

The comparison packaging is shown in the following (figure 1.2).



Figure(1.2): comparison packaging

2- System cost:

As shown in the following (figure 1.3) as the time proceed the system cost for R-744 refrigerant becomes less than the cost of R-134a refrigerant, and on the contrary no Global Warming Potential (GWP) & Ozone Depletion Potential (ODP) issue for R744 for exist of GWP issue related to R-134a [9].

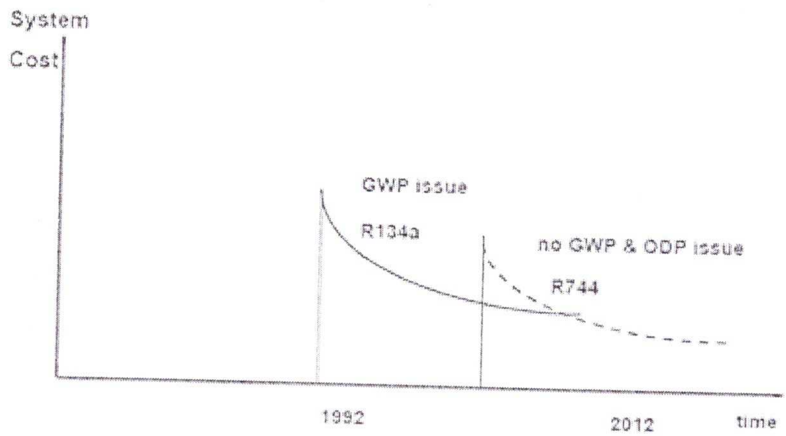


Figure (1.3): System cost

3- Technology

R-744 shows advantages in comparison with R-134a:

- Better cooling performance and efficiency.
- reduced emissions thanR-134a.
- Possibility of supplemental heating system

4- Energy demand:

Compared to R-134a systems, CO_2 unit needs 30%, less energy to provide the same amount of cooling, means reduced fuel consumption for R-744 system.
This comparison was taken from [10].

1.5 Suggested Refrigeration Cycle For Carbon Dioxide.

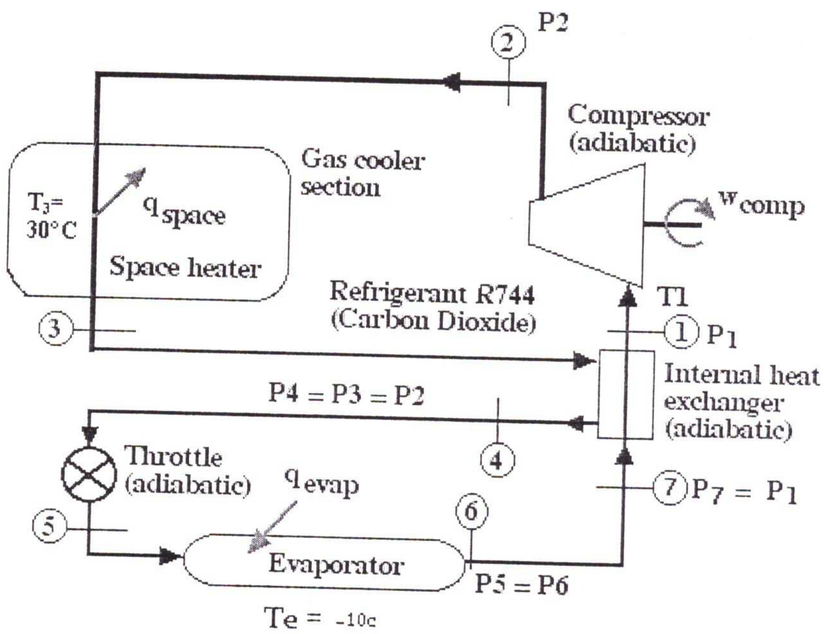


Figure (1.4): model refrigeration cycle for carbon dioxide

As shown in (figure 1.4) the main components of a CO_2-Refrigeration cycle are:

- Compressor.
- Space heater (gas cooler section).

- Internal heat exchanger.
- Throttle (expansion valve).
- Evaporator.

Principle operation for cycle

The path represented by 1-2-3-4-5-6-7-1in figure(1.4) shows compression (1-2), isobaric heat rejection at space heater (2-3) , isobaric cooling in the internal heat exchanger (3-4), adiabatic expansion (4-5), isobaric evaporation (5-6),in the suction line the gas may experience both a pressure drop and a temperature increase(6-7),and isobaric superheating at internal heat exchanger (7-1).

The analysis of this cycle in chapter two .

1.6 project Schedule

Table (1.2): Project plan for the first semester

Week / Activity	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Choosing Project Title																
Data Collection																
CO_2 Properties Studying																
Pervious Refrigeration Cycle																
Calculation & Design																
Writing& Printing																
Preparing The Presentation																

Table (1.3): Project plan for the second semester

Activity \ Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Data Collection																
Gas Cooler Design																
Internal Heat Exchanger Design																
Evaporator Design																
Electrical																
Writing& Printing																
Preparing The Presentation																

1.7 Project budget

Table(1.4): Estimated cost

Item #	Item	Price (NIS)
1	Internet	120
2	Journals and paper	420
3	Print	280
Grand Total		820NIS

Chapter Two

Analysis of carbon dioxide refrigeration cycle

2.1 Introduction

This chapter calculates cooling capacity (Q_e) for refrigeration chamber. That is to find the mass flow rate of refrigerant ($\dot{m}$) , and then analyze the cycle by calculating its coefficient of performances (COP) & power consuming (W) by depending on the given temperatures of evaporator and outlet gas cooler temperature (T_3), these calculations could be through thermodynamic and conservation of energy laws, using p-h diagram and tables of properties for R-744 refrigerant , and using software program (cool back) .

2.2 Cooling Load Calculations

Calculating the cooling load for refrigeration cubic chamber as shown in (figure 2.1) based on taking the refrigeration space temperature ($t_{R/S} = -10\ ^{O}C$) and the out side temperature ($t_{out} = 30\ ^{O}C$) , the area of refrigeration chamber sides are identical ($A = 0.5m \times 0.5m$).it contains 6 litters of water for freezing during 2 hours .

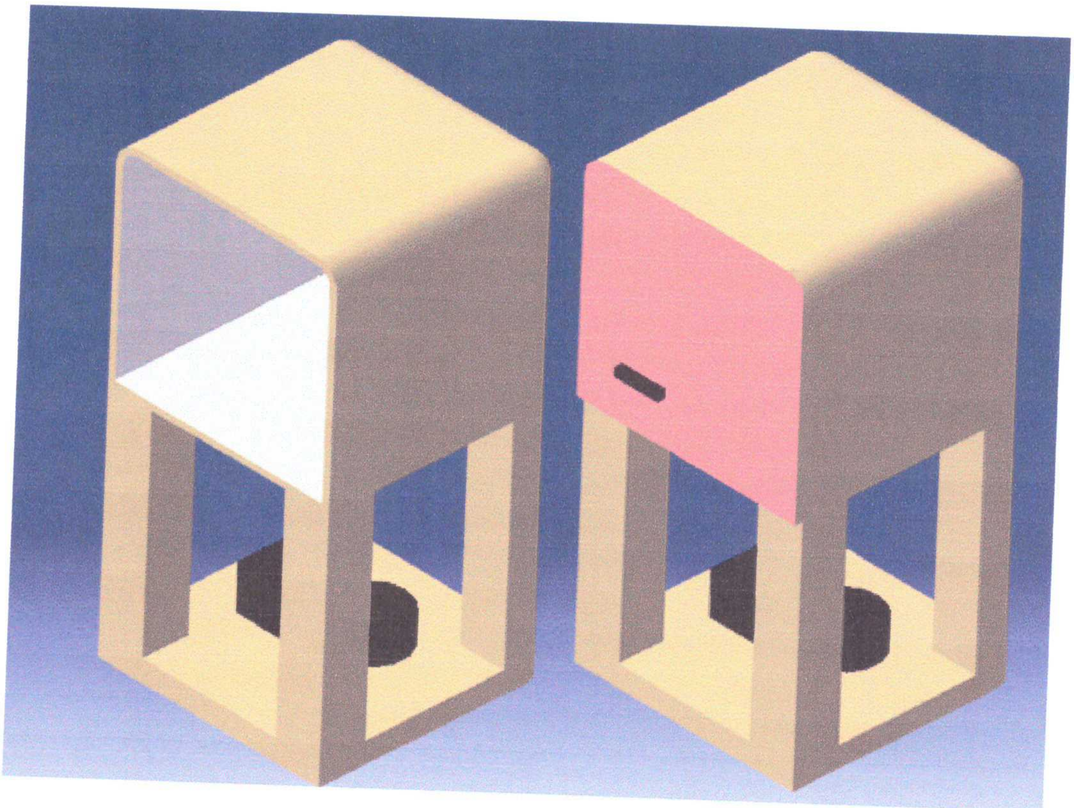


Figure 2.1: the suggested refrigeration cubic chamber

2.2.1 Walls, Ceiling, Floor & Door Heat Gains

The heat transfer rate through the walls, ceiling, floor and door is given by:

$$Q_{wall} = A_{wall} \cdot U_{wall} \cdot (t_{R/S} - t_{out})$$

$$Q_{ceiling} = A_{ceiling} \cdot U_{ceiling} \cdot (t_{R/S} - t_{out})$$

$$Q_{floor} = A_{floor} \cdot U_{floor} \cdot (t_{R/S} - t_{out})$$

$$Q_{door} = A_{door} \cdot U_{door} \cdot (t_{R/S} - t_{out})$$

Where:

t_{out} : Out side temperature [°C].

$t_{R/S}$: Refrigeration space temperature [°C].

Q_{wall} : Heat transfer rate through the wall [W] .

A_{wall} : The surface area of the wall [m^2].

U_{wall} : Overall heat transfer coefficient for wall [W/m^2. °C] .

$Q_{ceiling}$: Heat transfer rate through the ceiling [W]

$A_{ceiling}$: The surface area of the ceiling [m^2].

$U_{ceiling}$: Overall heat transfer coefficient for ceiling [W/m^2. °C] .

Q_{floor} : Heat transfer rate through the floor[W]

A_{floor} : The surface area of the floor [m^2].

U_{floor} : Overall heat transfer coefficient for floor [W/m^2. °C] .

Q_{door} : Heat transfer rate through the door [W]

A_{door}: The surface area of the door [m^2].

U_{door}: Overall heat transfer coefficient for door [W/m^2. °C] the walls, ceiling , floor and door have the same construction layers as shown in (figure 2.2).

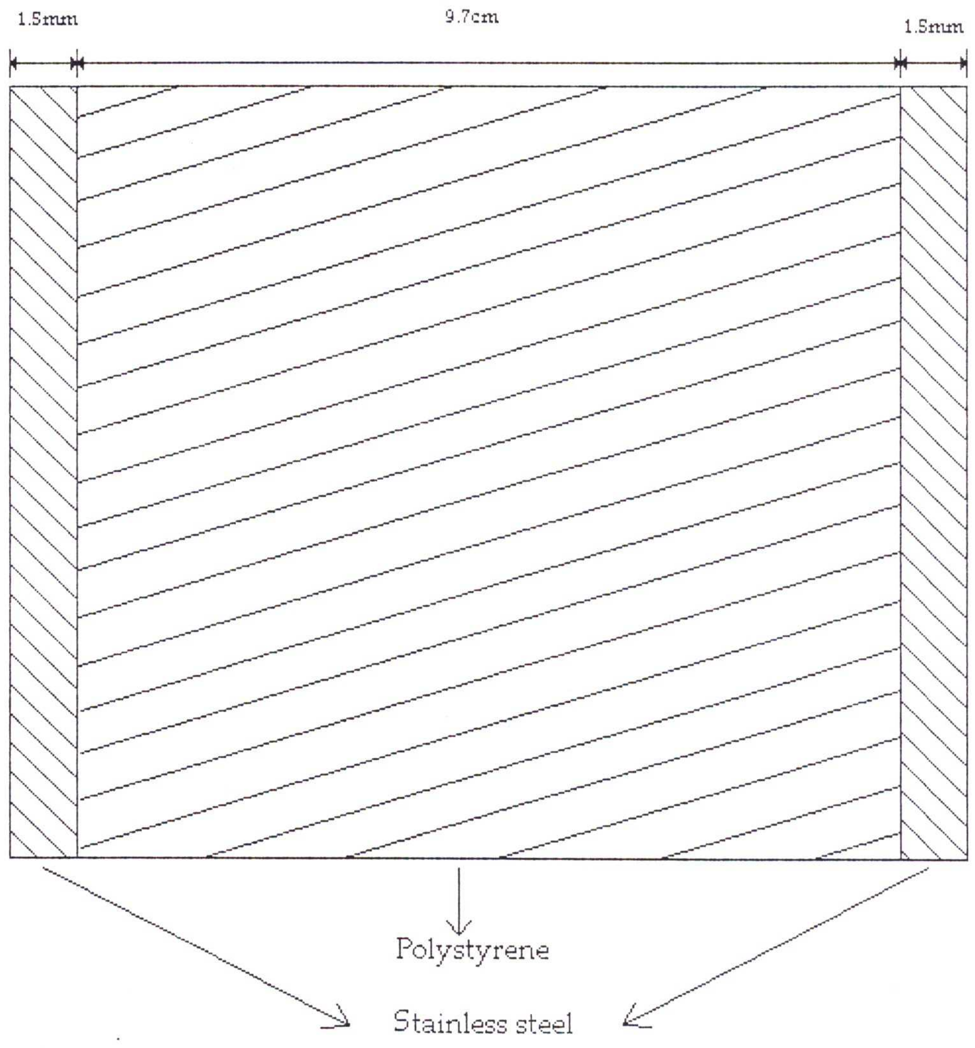


Figure 2.2: construction layers for the walls, ceiling, floor and doo

so the overall heat transfer coefficient (U) for walls, ceiling , floor and door have the same values given by :

$$U = \frac{1}{\frac{1}{h_{in}} + \frac{x_{s\tan.}}{K_{s\tan.}} + \frac{x_{poly.}}{K_{poly}} + \frac{x_{s\tan.}}{K_{s\tan.}} + \frac{1}{h_{out}}}$$

Where :

h_{in} : The convection heat transfer coefficient of inside air in refrigeration space [W/m^2. C].

h_{out} : The convection heat transfer coefficient of outside air [W/m^2. °C].

X_{stan} : The thickness of the layer of the Stainless steel[m].

K_{stan}: The thermal conductivity of the Stainless steel [W/m. °C].

X_{poly}: The thickness of the layer of the Polystyrene [m].

K_{poly}: The thermal conductivity of the Polystyrene [W/m. °C].

So the value of overall heat transfer coefficient is:

$$U = \frac{1}{\frac{1}{9.37} + \frac{0.0015}{15.6} + \frac{0.097}{0.029} + \frac{0.0015}{15.6} + \frac{1}{15}} = 0.285\ W/m^2.^oC$$

The values of heat transfer rate through the walls, ceiling , floor and door are:

$$Q_{wall} = 0.285 \times (0.5 \times .05) \times (30 - -10)$$
$$Q_{wall} = 0.00285\ [kW]$$

$$Q_{wall} = Q_{wall} \times number\ of\ wall$$
$$Q_{wall} = 0.00285 \times 4$$
$$Q_{wall} = 0.0114[kW].$$

$$Q_{floor} = 0.285 \times (0.5 \times 0.5) \times (30 - -10)$$

$$Q_{floor} = 0.00285\ [kW].$$

$$Q_{door} = 0.285 \times (0.5 \times 0.5) \times (30 - -10)$$
$$Q_{door} = 0.00285[kW]$$

The all heat transfer rate through the refrigeration chamber :

$$Q_{all} = Q_{wall} + Q_{floor} + Q_{door}$$
$$Q_{all} = 0.0114 + 0.00285 + 0.00285$$
$$Q_{all} = 0.0171\ [kW]$$

2.2.2 Product heat gain

The heat transfer rate gain from product is :

$$Q_{product} = \frac{Q_1 + Q_2 + Q_3}{\tau}$$

$$Q_1 = m \times c_p \times \Delta t$$

$$Q_2 = m \times \Delta h$$

$$Q_3 = m \times c_p \times \Delta t$$

where:

$Q_{product}$: packing heat load [kW]

$m_{product}$: mass of product [kg]

C_p : packing material specific heat [kJ/kg. °C]

Δt : difference temperature between out side & refrigerator space [°C].

τ : time of cooling load [sec]

$$Q_1 = 6 \times 4.18 \times (30 - 0)$$

$$Q_1 = 752.4[kJ]$$

$$Q_2 = 6 \times 333.7$$

$$Q_2 = 2002.[kJ]$$

$$Q_3 = 6 \times 2.05 \times (0 - -10)$$

$$Q_3 = 123[kJ]$$

$$Q_{product} = \frac{752.4 + 2002.2 + 123.}{7200}$$

$$Q_{product} = 0.39[kW]$$

2.2.3 Infiltration Heat Gain

The heat transfer rate gain from infiltration is:

$$Q_{\inf eltration} = \dot{V} \times \rho \times (h_{out} - h_{in})$$

Where :

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

ρ : density of air [kg /m^3]

h_{out}: enthalpy of outside air [kJ/kg] which equal (71.5[kJ/kg]).

h_{in}: enthalpy of air inside the refrigeration chamber[kJ/kg] which equal (-8.6[kJ/kg]).

By using Psychometrics Demo software, at T_{out}(dry bulb)=30^{0}C and relative humidity =55% [9].

$h_{out} = 71.5[kJ/kg]$.

By using Psychometrics Demo software, at T_{in}(dry bulb)=--10^{0}C and relative humidity =45% [9].

$h_{in} = -8.6[kJ/kg]$.

From table (A.1) Average air infiltration rates in L/s due to door opening, the value of $\dot{V} = 0.041 \times 10^{-3}$

$$Q_{\inf eltration} = 0.041 \times 10^{-3} \times 1.2 \times (71.5 - -8.6)$$

$$Q_{\inf eltration} = 3.9[W].$$

2.2.4 Door heater heat gains

$$Q_h = \eta_h \times p$$

Where:

p :power of heater, take as 60[W]

η_h : heater using factor

$Q_h = 1 \times 0.60 = 0.060\ [KW]$.

The total heat gain is :

$$Q_{total} = Q_{all} + Q_{product} + Q_{\inf eltration} + Q_h$$
$$Q_{total} = 0.0171 + 0.39 + 0.0039 + 0.06$$
$$Q_{total} = 0.47[kW]$$

The total cooling load IS:

Total cooling load = Q_{total} + (Q_{total} * SF)

Total cooling load = 0.47 + (0.47 * 0.1)

= 0.51 [kW]

The cooling capacity is:

$$Q_e = \frac{TCL \times 24}{Working\ time}$$
$$Q_e = \frac{0.51 \times 24}{16}$$
$$Q_e = 0.78[kW].$$

2.3 Cycle Analysis

This section analysis transcritical carbon dioxide refrigeration cycle as shown in (figure 1.4) operating between two pressure limits (high pressure P_H = 80[bar] = 8000 [kpa] & low pressure P_L = 22.91[bar] = 2291 [kpa]) ,with pressure drop in suction line (ΔP = 0.21 [bar] =210[kpa]),with refrigerant tempreture at outlet compressor (T_2 = 145.8[°C]) , cooling capacity (Q_e= 0.78[kW]) which found in previous section .

Analysis procedures :

- using p-h digram for carbon dioxide refrigerant (R-744)

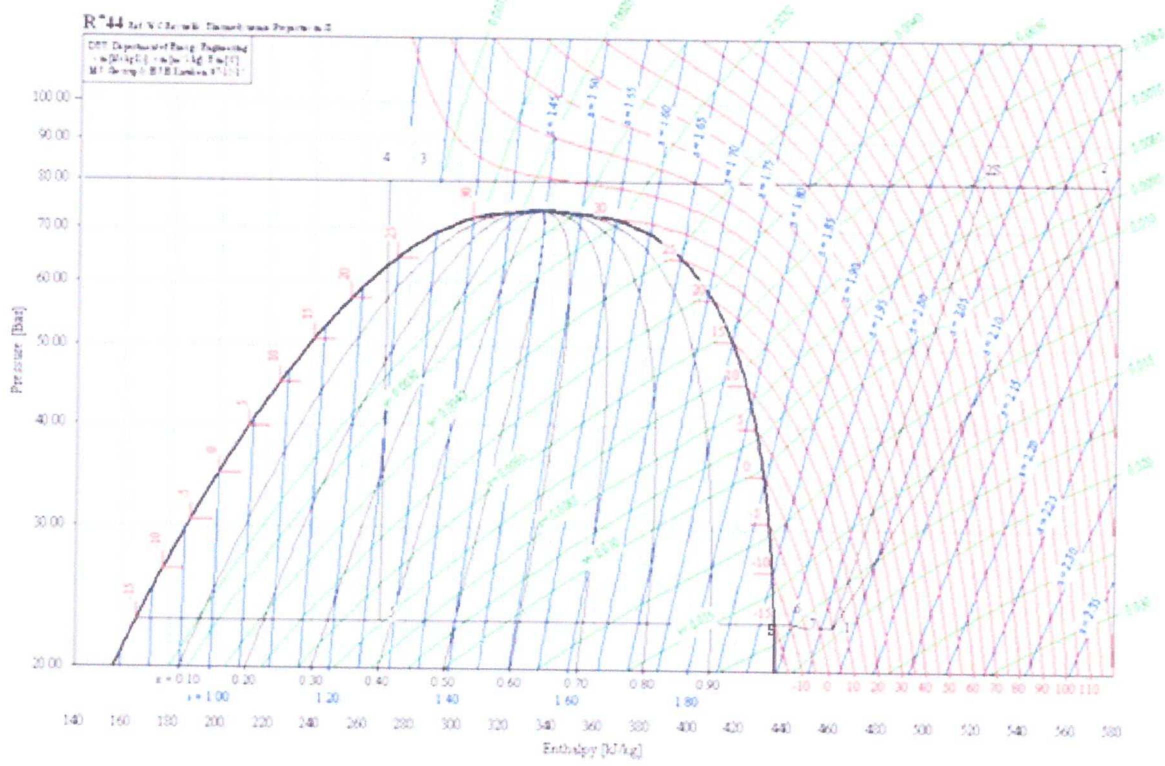


Figure 2.3: p-h digram for carbon dioxide refrigerant (R-744).

By operating conditions and using p-h digram of refrigerant(R-744) can be determined the physical properties for states of the refrigeration cycle which labeled in the fllowing Table 2.1 .

Table 2.1: physical properties for states of the refrigeration cycle

State point #	Temperature T (°C)	Enthalpy h (kJ/kg)	Pressure p (kpa)	Specific volume v (m^3/kg)	Density ρ (kg/m^3)
١	5.5	460	2270	0.019	51.4
٢	145.8	576	8000	0.0087	114.8
2S	103	523	8000	_	_
3	30	282	8000	0.00142	703
4	27.2	270	8000	0.00133	748
5	-15	270	2291	_	_
5	-15	433	2291	_	_
6	-10	442	2291	0.0172	58.1
7	-5	448	2270	0.0180	55.5

By using table 2.1 can be calculate the following :

- The refrigeration effect is :

$$q_e = h_6 - h_5$$

Where :

q_e: The refrigeration effect[kJ/kg].

h_6 : Specific enthalpy for refrigerant after evaporator [kJ/kg].

h_5 : Specific enthalpy for refrigerant before evaporator [kJ/kg].

$q_e = (442 - 270) = 172$ [kJ/kg].

- The mass flow rate is :

$$\dot{m} = \frac{Q_e}{q_e}$$

Where:

$\dot{m}$: mass flow rate of the refrigerant [kg/s].

Q_e: the cooling capacity [kW]

$$\dot{m} = \frac{0.78}{172} = 0.0045[Kg/s]$$

- the compressor power is :

$W_c = \dot{m}\,(h_2 - h_1)$

Where :

W_c: The power input of the compressor [kW].

h_1: Specific enthalpy for refrigerant before compressor [kJ/kg].

h_2: Specific enthalpy for refrigerant after compressor [kJ/kg].

$$\dot{W}_c = 0.0045(576 - 460) = 0.522 \text{ [kW]}$$

- the gas cooler heat transfer rate is :

$$Q_{out} = \dot{m}(h_2 - h_3)$$

Where:

Q_{out}: gas cooler heat transfer rate [kW].

h_2: Specific enthalpy for refrigerant before gas cooler [kJ/kg].

h_3: Specific enthalpy for refrigerant after gas cooler [kJ/kg].

We find h_3 from energy balance at heat exchanger, the energy balance at heat exchanger gives by:

$$h_4 - h_3 = h_7 - h_1$$

$$h_3 = h_1 + h_4 - h_7$$

where :

h_3: Specific enthalpy for refrigerant at low pressure before heat exchanger [kJ/kg].

h_4: Specific enthalpy for refrigerant at low pressure after heat exchanger [kJ/kg].

h_1: Specific enthalpy for refrigerant at high pressure after heat exchanger [kJ/kg].

h_7: Specific enthalpy for refrigerant at high pressure before heat exchanger [kJ/kg].

$h_3 = 270+460 - 448 = 282$ [kJ/kg]

so

$Q_{out} = 0.0045(576 - 282) = 1.32[kW]$

- the refrigeration coefficient of performances are :

$$COP_{cooling} = \frac{Q_e}{w_C} = \frac{h_5 - h_3}{h_2 - h_1}$$

$$COP_{cooling} = \frac{0.78}{4.64} = 1.46$$

$$COP_{heating} = \frac{Q_{out}}{w_C}$$

$$COP_{heating} = \frac{1.32}{0.552} = 2.53$$

- $$COP^{*} = \frac{\dot{m}.(h_3 - h_1)}{W_c}$$

Where:

COP* is the ratio between the supply of energy (mass flow multiplied by enthalpy difference) to the refrigerant from the outlet of the condenser to the inlet of the compressor[cool back].

$$= \frac{0.0045(460 - 282)}{1.32} = 0.61$$

- For the Carnot vapor refrigeration cycle, the coefficient of performance is:

$$COP_{carnot} = \frac{T_L}{T_H - T_L}$$

Where:

COP_{carnot}: Represents the maximum theoretical coefficient of performance of any refrigeration cycle operating between regions at T_L and T_H .

T_L: is the evaporating temperature [k].

T_H: is the temperature of the refrigerant leaving the gas cooler [k].

$$COP_{carnot} = \frac{(-15+273)}{(30+273)-(-15+273)} = \frac{258}{45} = 5.7$$

- the compressor isentropic efficiency is:

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1}$$

Where :

η_c: isentropic efficiency which accounts the effect of irreversible compression

h_{2s}: Specific enthalpy after compressor due to isentropic compression [KJ/Kg].

$$\eta_c = \frac{523-460}{576-460} = 0.69$$

- The Carnot efficiency is :

$$\eta_{carnot} = \frac{COP^*}{COP_{carnot}}$$

$$= \frac{0.61}{5.7} = 0.1$$

Chapter Three

main component Cycle Design

Introduction

The entire system (carbon dioxide refrigeration system) has been designed based on energy balance of individual components yielding conservation equations presented below. To consider the lengthwise property variation, all the three heat exchangers (gas cooler, evaporator, and internal heat exchanger) have been discredited and momentum and energy conservation equations have been applied to each one of them. The following assumptions have been made in the analysis:

1. Heat transfer with the ambient is negligible.

2. Only single-phase heat transfer occurs for water.

3. Compression process is adiabatic but not isentropic.

4. Pressure drop on waterside and in connecting pipes are negligible.

5. The inlet and outlet temperature for the water of the gas cooler are equal $25^{o}C$ and $81^{o}C$ respectively.

3.1 Compressor

The refrigerant mass flow rate through the compressor is given by ,

$$\dot{m}_{ref} = \rho_1 \times n_v \times v_s \times \frac{N}{60}$$

Where:

Vs: Swept volume of compressor [m^3].

ρ_1: Density [kg/m^3].

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

η_v: Volumetric efficiency dimensionless unit.

N: Compressor speed [rpm].

Where volumetric efficiency η_v for semi-hermetic compressor is estimated from [11]:

$$n_v = 0.9207 - 0.0756(\frac{P_{dis}}{P_{suc}}) + 0.0018(\frac{P_{dis}}{P_{suc}})^2$$

Where:

P_{dis} : Compressor discharge pressure [bar].

P_{suc}: Compressor suction pressure [bar].

$$n_v = 0.9207 - 0.0756(\frac{80}{22.7}) + 0.0018(\frac{80}{22.7})^2$$

$$n_v = 0.9207 - 0.0756(3.5) + 0.0018(3.5)^2$$

$$n_v = 0.68$$

The isentropic efficiency of the compressor is given by,

$$n_{is,c} = \frac{(h_{2s} - h_1)}{(h_2 - h_1)}$$

And is estimated by employing the following correlation for the semi_ hermetic compressor [8].

$$n_{is,c} = -0.26 + 0.7952(\frac{p_{dis}}{p_{suc}}) - 0.2803(\frac{p_{dis}}{p_{suc}})^2 + 0.0414(\frac{p_{dis}}{p_{suc}})^3 - 0.0022(\frac{p_{dis}}{p_{suc}})^4$$

$$n_{is,c} = -0.26 + 0.7952(\frac{80}{22.7}) - 0.2803(\frac{80}{22.7})^2 + 0.0414(\frac{80}{22.7})^3 - 0.0022(\frac{80}{22.7})^4$$

$$n_{is,c} = 0.46$$

- When we choose the swept volume for compressor is 90 ccm.

Then:

$$0.0045 = 51.4 \times 0.68 \times 150 \times 10^{-6} \times \frac{N}{60}$$

$$N = \frac{2.4}{0.00031} = 51.5\,rpm$$

3.2 Design Of Gas Cooler

One of the computational segment of gas cooler (double pipe counter flow heat exchanger) is shown in Fig(3.1).

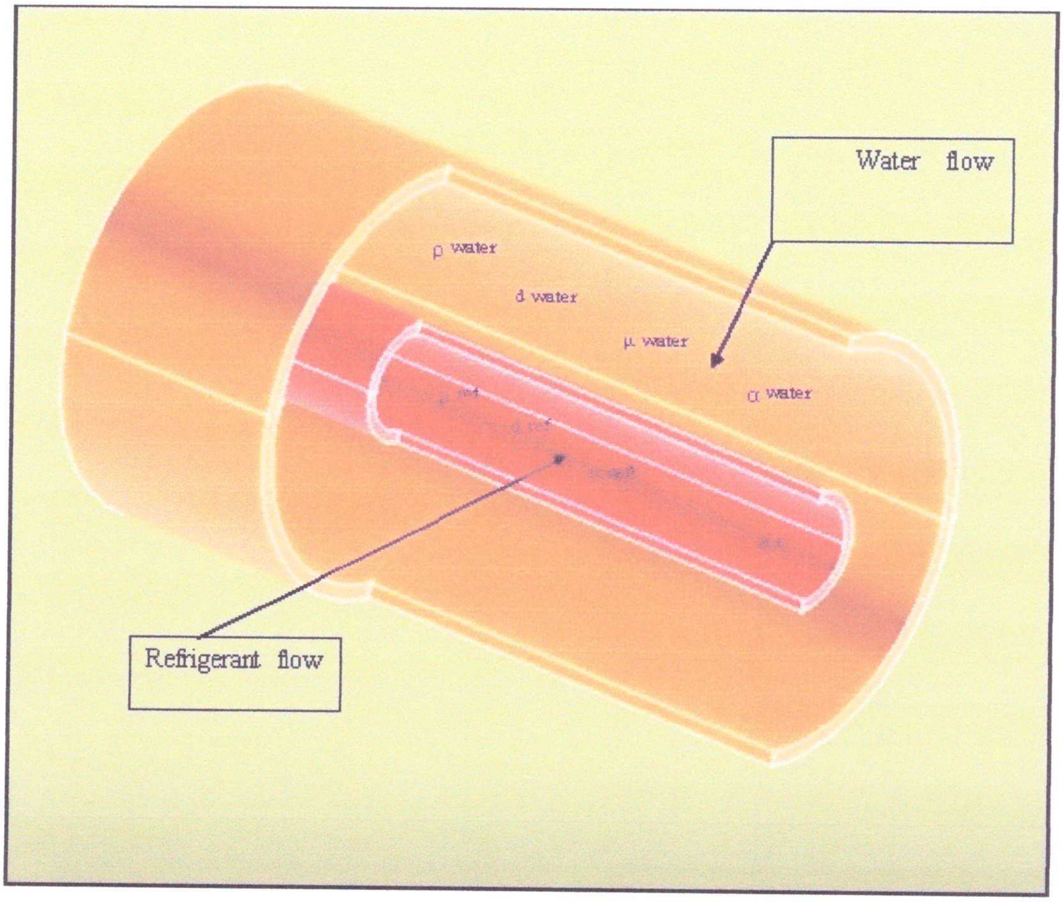


Figure (3.1) : A computational segments of segment of gas cooler

3.2. 1 Calculation of heat transfer rate for gas cooler

Employing energy balance, heat transfer in gas cooler is given by:

$$Q_{gc} = \dot{m}_{ref}(h_2 - h_3)$$

Where:

$\dot{Q}_{gc}$: Gas cooler Heat transfer rate [W].

$\dot{m}_{ref}$: Refrigerant mass flow rate [kg/s] which equals (0.0045m/s) , is found in chapter two.

h_2: Specific enthalpy for refrigerant before gas cooler [kJ/kg] which equals (576[kJ/kg]) , is found in chapter tow.

h_3: Specific enthalpy for refrigerant after gas cooler [kJ/kg] which equals (282[kJ/kg]) , is found in chapter two.

So the gas cooler heat transfer rate equal :

$$\dot{Q}_{gc} = \dot{m}_{ref}(h_2 - h_3)$$

$$Q_{gc} = 0.0045(576 - 282) = 1.32 \times 10^3 [W]$$

3.2.2 Calculation of over all heat transfer coefficient area of the gas cooler

Employing LMTD (Logarithmic Mean Temperature Difference) expression , heat transfer in gas cooler is given by :

$$Q_{gc} = (UA)_{gc} \frac{(T_{gci} - T_0) - (T_{gco} - T_i)}{\ln((T_{gci} - T_o)/(T_{gco} - T_i))}$$

Where:

$(UA)_{gc}$: Over all heat transfer coefficient area for the gas cooler [W/K].

T_{gci} : The temperature of refrigerant at the inlet of the gas cooler [°C] which equals (148.8°C) , is found in chapter two .

T_{gco} : The temperature of refrigerant at the outlet of the gas cooler [°C] which equals (30°C) , is found in chapter two .

T_i: Inlet water temperature [°C].

T_o: Outlet water temperature [°C].

So the Over all heat transfer coefficient area of the heat transfers for gas cooler equal:

$$11760 = (UA)_{gc} \frac{(145.8 - 81) - (30 - 25)}{\ln((145.8 - 81)/(30 - 25)}$$

$$(UA)_{gc} = 56.6[W / K]$$

3.2.3 Calculation of the refrigerant pipe diameter

- The velocity of the refrigerant that enters the refrigerant pipe of the gas cooler at vapor phase , is supposed to equal :

  $u_{ref} = 15$ m/s.

- The wall between the refrigerant and water pipes in the gas cooler has a temperature which is supposed to equal:

  $T_w = 68$ °C .

- The reference temperature of the refrigerant equals :

$$T_{refernce}[°C] = \frac{T_w[°C] + T_{average}[°C]}{2}$$

To find the average temperature of the refrigerant that flows inside the refrigerant pipe , the following equation should be used :

$$T_{average} = (T_{gci} + T_{gco})/2$$

$$T_{average} = \frac{145.8+30}{2} = 85.4[°C].$$

Then,

$$T_{refernce} = \frac{68 + 85.4}{2} = 76.7[°C].$$

By the principle of the mass conservation (continuity equation) for the carbon dioxide refrigerant that flow through the refrigerant pipe with supposed velocity (u_{ref} =15 m/s),the diameter of the inside pipe (refrigerant pipe) is calculated by the following equation :

$$\dot{m}_{ref} = \rho_{ref} \times u_{ref} \times A_{ref}$$

Where :

ρ_{ref}: The density of the refrigerant at $T_{refernce}$[Kg/m^3].

u_{ref}: Refrigerant velocity [m/s].

A_{ref}: Cross sectional area of the refrigerant pipe [m^2].

By using Pace software, the value of the carbon dioxide refrigerant density at $T_{refernce}$and 81 bar equals [12]:

$$\rho_{ref} = 168 Kg/m^3$$

So the diameter of the refrigerant pipe equals:

$$d_{ref} = \sqrt{\frac{4 \times \dot{m}_{ref}}{\pi \times u_{ref} \times \rho_{ref}}}$$

Then,

$$d_{ref} = \sqrt{\frac{4 \times 0.04}{\pi \times 15 \times 168}} = 0.0015\ [m].$$

By using Copper Tube Hand Book from table A2, the selected pip is type K with standard size 1/4 (0.00635m) inch , and its wall thickness is 0.035 inch(0 .00008m)[13].

3.2.4 Calculation of Reynolds number for the refrigerant in the refrigerant pipe.

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

$$\mathrm{Re}_{refb} = \frac{\rho_{refb} \times u_{ref} \times d_{ref}}{\mu_{refb}}$$

Re_{refb}: Reynolds number at reference temperature for the refrigerant that flows inside the refrigerant pipe, is dimensionless unit.

d_{ref}: The diameter of the refrigerant pipe [m].

μ_{refb}: *Viscosity of the refrigerant* that flows inside the refrigerant pipe *at* $T_{refernce}$ $[\frac{kg}{(m.s)}]$

.

By using Peace Software , the viscosity value of the carbon dioxide refrigerant that flows inside the refrigerant pipe at $T_{refernce}$ and 81 bar equals [12]:

$$\mu_{refb} = 2.07 \times 10^{-5}\ Kg/(m.s)$$

So Reynolds number at reference temperature equals:

$$\mathrm{Re}_{refb} = \frac{168 \times 15 \times 0.0015}{2.07 \times 10^{-5}} = 182608$$

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

$$\mathrm{Re}_{refw} = \frac{\rho_{refw} \times u_{ref} \times d_{ref}}{\mu_{refw}}$$

Where,

Re_{refw}: Reynolds number at wall temperature for the refrigerant that flows inside the refrigerant pipe, is dimensionless unit.

d_{ref}: The diameter of the refrigerant pipe [m].

μ_{refw}: *Viscosity of the refrigerant* that flows inside the *refrigerant pipe at* $T_{wall}\ [\frac{kg}{(m.s)}]$

By using Peace Software, the viscosity value of the carbon dioxide refrigerant that flows inside the refrigerant pipe at T_{wall} and 81 bar equals [12]:

$\mu_{refw} = 2.08 \times 10^{-5} [Kg/(m.s)]$.

By using Peace Software, the density value of the carbon dioxide refrigerant that flows inside the refrigerant pipe at T_{wall} and 81 bar equals [12]:

$\rho_{refw} = 182[Kg/(m^3)]$.

So Reynolds number at wall temperature equals:

$$\mathrm{Re}_{refw} = \frac{182 \times 15 \times 0.0015}{2.08 \times 10^{-5}} = 196875$$

3.2.5 Calculation of Nusselt number for the refrigerant in the refrigerant pipe

To estimate Nusselt number for the refrigerant, Gnielinski equation is suitable for channel due to large variation of fluid properties in the radial direction. To alleviate this deficiency, Pitla et al. [14] proposed a modification for transcritical in-tube carbon dioxide cooling, incorporating both reference (bulk) and wall properties. This correlation, used for gas cooler model, is given by:

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

where ,

Nu_r :Nusselt number, for the refrigerant that flows inside the refrigerant pipe, is dimensionless number .

Nu_{rb} : Nusselt number at bulk temperature for the refrigerant that flows inside the refrigerant pipe , is dimensionless number.

Nu_{rw} : Nusselt number at wall temperature for the refrigerant that flows inside the refrigerant pipe, is dimensionless number.

k_{rb} : Thermal conductivity at reference (bulk) temperature for the refrigerant that flows inside the refrigerant pipe ,[W/(m.K)].

k_{rw} : Thermal conductivity at wall temperature for the refrigerant [W/(m.K)].

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

- To find Nusselt number at wall temperature for the refrigerant that flows inside the refrigerant pipe, the following equation should be used:

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

Where:

f_{rw}

: Friction factor at wall temperature for the refrigerant in the refrigerant pipe, is dimensionless number.

$\Pr_{refw}$: Prandtle number at wall temperature for the refrigerant in the refrigerant pipe, is dimensionless number.

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

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

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

$$f_{rw} = (0.79\ln(196875) - 1.64)^{-2} = 0.0156$$

By using Peace Software , the value of Prandtle number for the refrigerant that flows inside the refrigerant pipe at T_{wall} and 81 bar equals [12]:

$\Pr_{refw} = 1.19$ this number exists within Gnielinski range of Prandtle number.

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

$$Nu_{rw} = \frac{(0.0159/8)(196875 - 1000)1.19}{1.07 + 12.7(0.0159/8)^{1/2}(1.19^{2/3} - 1)} = 231$$

❖ To find Nusselt number at reference temperature for the refrigerant that flows inside the refrigerant pipe, the following equation should be used:

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

Where:

f_{rb} : Friction factor at reference temperature for the refrigerant that flows inside refrigerant pipe , is dimensionless number.

Pr_{refb} : Prandtle number at reference temperature for the refrigerant that flows inside refrigerant pipe, is dimensionless number.

To find Friction factor at reference temperature for the refrigerant that flows inside the refrigerant pipe, the following equation should be used:

$$f_{rb} = (0.79\ln(\mathrm{Re}_{refb}) - 1.64)^{-2}$$

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

$$f_{rb} = (0.79\ln(818260) - 1.64)^{-2} = 0.0158$$

By using Peace Software, the value of Prandtle number for the refrigerant that flows inside the refrigerant pipe at $T_{reference}$ and 81 bar equals [12]:

Pr_{refb} = 1.09 this number exists within Gnielinski range of Prandtle number.

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

$$Nu_{rb} = \frac{(0.0158/8)(773043-1000)1.09}{1.07+(0.0158/8)^{1/2}(1.09^{2/3}-1)} = 326$$

By using Peace Software, the value of the thermal conductivity at T_{wall} and 81 bar equals [12]:

K_{rw} =0.030 [W/m.°C] .

By using Peace Software, the value of the thermal conductivity at $T_{refernce}$ and 81bar equals [12]:

K_{rb} =0.029 [W/m.°C].

Then Nusselt number for the refrigerant that flows inside refrigerant pipe equals :

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

$$Nu_r = \left(\frac{231+236}{2}\right)\frac{0.030}{0.029} = 288$$

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

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

$$\alpha_r = \frac{Nu_r}{d_{ref}} K_{rb}$$

Where,

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

Then the convection heat transfer coefficient for the refrigerant in the refrigerant pipe equals :

$$\alpha_r = \frac{288}{0.0015} \times 0.030 = 5762[W/(m^2.c)]$$

3.2.7 Calculation of water mass flow rate in water pipe.

The water mass flow rate ($\dot{m}_{gcw}$) which is required for the gas cooler could be found as the following equation:

$$\dot{m}_{gcw} = \frac{Q_{gc}}{C_{pw} \times (T_o - T_i)}$$

Where;

$\dot{m}_{gcw}$: Water mass flow rate (external fluid) [kg/s].

C_{pw} : Water specific heat [kJ/kg.°C].

So the water mass flow rate equal:

$$\dot{m}_{gcw} = \frac{1.32}{4.18 \times (81 - 25)} = 0.0056[\ m/s].$$

3.2.8 Calculation of the water pipe diameter

- The velocity of the water enters the gas cooler at liquid phase, is supposed to be equal :

  u = 0.5 [m/s].

- The wall temperature of the gas cooler is supposed to be equal:
  T_w = 68 [°C] .

❖ The reference temperature of the water is to be equals :

$$T_{refernce}[°C] = \frac{T_w[°C] + T_{average}[°C]}{2}$$

To find the average temperature of the water that flows inside the water pipe, the following equation should be used:

$$T_{average} = \frac{(T_o + T_i)}{2}$$

$$T_{average} = \frac{25+81}{2} = 53\,[°C].$$

Then,

$$T_{refernce}[°C] = \frac{68+53}{2} = 60.5[°C]$$

By the principle of the mass conservation (continuity equation) for the water that flow through the outside pipe with supposed velocity (u_{water} =0.5 m/s),the diameter of the outside pipe (water pipe) is calculated as the following equation :

$$\dot{m}_{gcw} = \rho_{water} \times u_{water} \times A_{water}$$

ρ_{water}: The density of the refrigerant at $T_{refernce}$ [Kg/m^3].

u_{water}: Water velocity [m/s].

A_{water}: Cross sectional area of the water pipe [m^2].

By using Peace Software, the value of the water density at$T_{refernce}$ and 1 bar equals [12]:

$$\rho_{water} = 952\, kg/m^3$$

So the diameter of the water pipe equals :

$$d_{water} = \sqrt{\frac{4 \times \dot{m}_{water}}{\pi \times u_{water} \times \rho_{water}}}$$

Then,

$$d_{water} = \sqrt{\frac{4 \times 0.05}{\pi \times 0.5 \times 952}} = 0.0038\ [m].$$

By using Copper Tube Hand Book from table A2., the select pipe is type K with standard size 1 /2 inch(0.0127),and its wall thickness is 0.049 inch(0 .0012m)[13].

3.2.9 Calculation of Reynolds number for the water in the water pipe

* Reynolds number at reference temperature for the water that flows inside the water pipe is calculated by the following equation:

$$\mathrm{Re}_{water} = \frac{\rho_{water} \times u_{water} \times d_{water}}{\mu_{water}}$$

Where;

Re_{water}: *Reynolds number dimensionless unit.*

d_{water}: *Diamerter of water pipe [m]*

μ_{water}: *water viscosity [kg/m s].*

By using Peace Software, the value of the water viscosity at $T_{refernce}$ of the water and 1 bar equals [12]:

$$\mu_{water} = 0.00046\, Kg/(m.s).$$

So the Reynolds number for the water equals:

$$\mathrm{Re}_{water} = \frac{952 \times 0.5 \times 0.0038}{0.00046} = 4014.9$$

3.2.10 Calculation of Nusselt number for the water in the water pipe

❖ To find the Nusselt number of the water, the following equation is used:

$$Nu_{water} = 0.023 \times (\mathrm{Re}_{water})^{0.8} \times (pr_{water})^{0.4}$$

Where,

Pr_{water}: Prandtle number for the water dimensionless number.

By using Peace Software, the values of thermal conductivity and dynamic viscosity for the water at $T_{refernce}$ of the water and 1 bar equals [12]:

$$k_{water} = 0.06106[W/m.K]$$

$$\mu_{water} = 0.000452 \ [kg/(m.s)]$$

By using Peace Software, the value of Prandtle number at the reference temperature of the water and 1 bar equals [12]:

$$pr = 585$$

Then the Nusselt number for the water equal:

$$Nu_{water} = 0.023 \times (4014)^{0.8} \times (585)^{0.4} = 662$$

3.2.11 Calculation of Hydraulic diameter for the water in the water pipe

To find the Hydraulic diameter of the water the following equation is used:

$$DH_{water} = \frac{Re_{water} \times \mu_{water} \times A_{water}}{\dot{m}_{gcw}}$$

Then the Hydraulic diameter for the water equals :

$$DH_{water} = \frac{13141 \times 0.00046 \times (0.0015^2 \times \frac{3.14}{4})}{0.0056} = 0.0037[m]$$

3.2.12 Calculation of convection heat transfer coefficient for the water in the water pipe

The heat transfer coefficient (convection heat transfer coefficient) for the water is calculated by the following equation:

$$\alpha_{water} = \frac{k_{water}}{DH} Nu_{water}$$

α_{water} : Convection heat transfer coefficient of water [W/(m^2.K)].

k_{water}: Thermal conductivity of water at reference temperature of the water [W/(m.K)].

D_H: Hydraulic diameter [m].

Nu_{water} : Nusselt number for water dimensionless number.

Then the convection heat transfer coefficient for water equal:

$$\alpha_{water} = \frac{0.6106}{0.0037} 662 = 108016[W/(m^2.C)]$$

3.2.13 Calculation of gas cooler length.

Overall heat transfer coefficient for the gas cooler has been calculated by using fundamental equation for overall heat transfer coefficient yields:

$$\frac{1}{(UA)_{gc}} = \frac{1}{\alpha_r A_{ref}} + \frac{\ln(d_o / d_{ref})}{2\pi L_{gc} K_w} + \frac{1}{\alpha_{water} A_{water}}$$

Where:

d_o: The outer diameter of the refrigerant pipe [m].

L_{gc}: Gas cooler length [m].

K_w: Thermal conductivity for copper wall [W/(m.k)].

A_{ref}: Heat transfer area of the refrigerant pipe [m^2].

A_{water}: Heat transfer area of the water pipe [m^2].

To find heat transfer area of the refrigerant pipe, the following equation should be used:

$$A_{ref} = \pi d_{ref} L_{gc}$$

Where:

d_{ref}: The inner diameter of the refrigerant pipe[m].

To find the heat transfer area of the water pipe, the following equation is to be used:

$$A_{water} = \pi d_{water} L_{gc}$$

To find the gas cooler length, the following equation is to be used:

$$L_{gc} = ((UA)_{gc} \frac{(2K_w a_r d_{ref} + d_{ref} d_{water} a_r a_{water} \ln(d_o / d_{ref}) + 2 d_{water} a_{water} K_w)}{2\pi (d_{ref} a_r K_w a_{water} d_{water})}$$

$$.L_{gc} = \frac{503((2 \times 388 \times 0.0015 \times 5762 + 0.0015 \times 5762 \times 108016 \times 0.0038 \ln(0.0017/0.0015) + 2 \times 0.0038 \times 108016 \times 388)}{2\pi(0.0015 \times 5762 \times 388 \times 0.0038 \times 108016)}.$$

$$L_{gc} = 2.1m$$

3.2.14 Calculation of pressure drop in the gas cooler.

The pressure drop in the refrigerant pipe of the gas cooler is given by[15]:

$$\Delta p_{gc} = \frac{G^2{}_r}{2\rho_{refb}} (\xi_r \frac{L_{gc}}{d_{ref}} + 1.2)$$

Where:

Δp_{gc}: Pressure drop in refrigerant pipe [bar].

G_r^2: $Mass\ velocity\ of the\ \ refrigerant\ in\ the$ refrigerant $pipe[kg/(m^2.s)]$

ζ_r: $fricition\ factor\ for\ refrigerant\ \ in\ \ refigerant\ pipe\ dimensionless\ unit.$

The mass velocity of the refrigerant that flows inside the refrigerant pipe of the gas cooler is given by:

$$G_r = \frac{\dot{m}_{ref}}{A_{ref}}$$

Where:

A_{ref}: Cross sectional area of the refrigerant pipe [m^2].

Then ,the mass velocity of the refrigerant in the refrigerant pipe equals:

$$G_r = \frac{0.0045}{(\pi \times 0.0045^2)/4} = 2516\ [\frac{kg}{m^2}.s].$$

The friction factor for refrigerant in the refrigerant pipe of the gas cooler is given by:

$$\xi_r = (1.82\ln(\mathrm{Re}_{refw}) - 1.64)^{-2}\frac{\rho_{refw}}{\rho_{refb}}(\frac{\mu_{refw}}{\mu_{refb}})^s$$

Where;

$s: Exponent\ factor\ [(Wm^2.s)/kg]$

The exponent factor for refrigerant in the refrigerant pipe of the gas cooler is given by:

$$s = 0.023\left|\frac{Q_{gc}}{G_r}\right|^{0.42}$$

Then ,the exponent factor for the refrigerant in the refrigerant pipe equal:

$$s = 0.023\left|\frac{1320}{1263}\right|^{0.42} = 0.06 \ \ (W.m^2.s)/kg$$

Then, the friction factor for the refrigerant in the refrigerant pipe equals:

$$\xi_r = (1.82\ln(833437) - 1.64)^{-2}\frac{182}{168}(\frac{2.07\times10^{=5}}{2.08\times10^{-5}})^{0.06} = 2.01\times10^{-3} \ .$$

Then, the pressure drop for the refrigerant in the refrigerant pipe equals:

$$\Delta p_{gc} = \frac{G^2{}_r}{2\rho_{refb}}(\xi_r \frac{L_{gc}}{d_{ref}} + 1.2)$$

$$\Delta p_{gc} = \frac{1263^2}{2\times168}(2.01\times10^{-3}\ \frac{8.85}{0.00635} + 1.2) = 19046 pa = 0.19 bar$$

3.3 Internal Heat Exchanger Design

One of the computational segments of internal heat exchanger (double pipe counter flow heat exchanger) is shown in Fig 3.2.

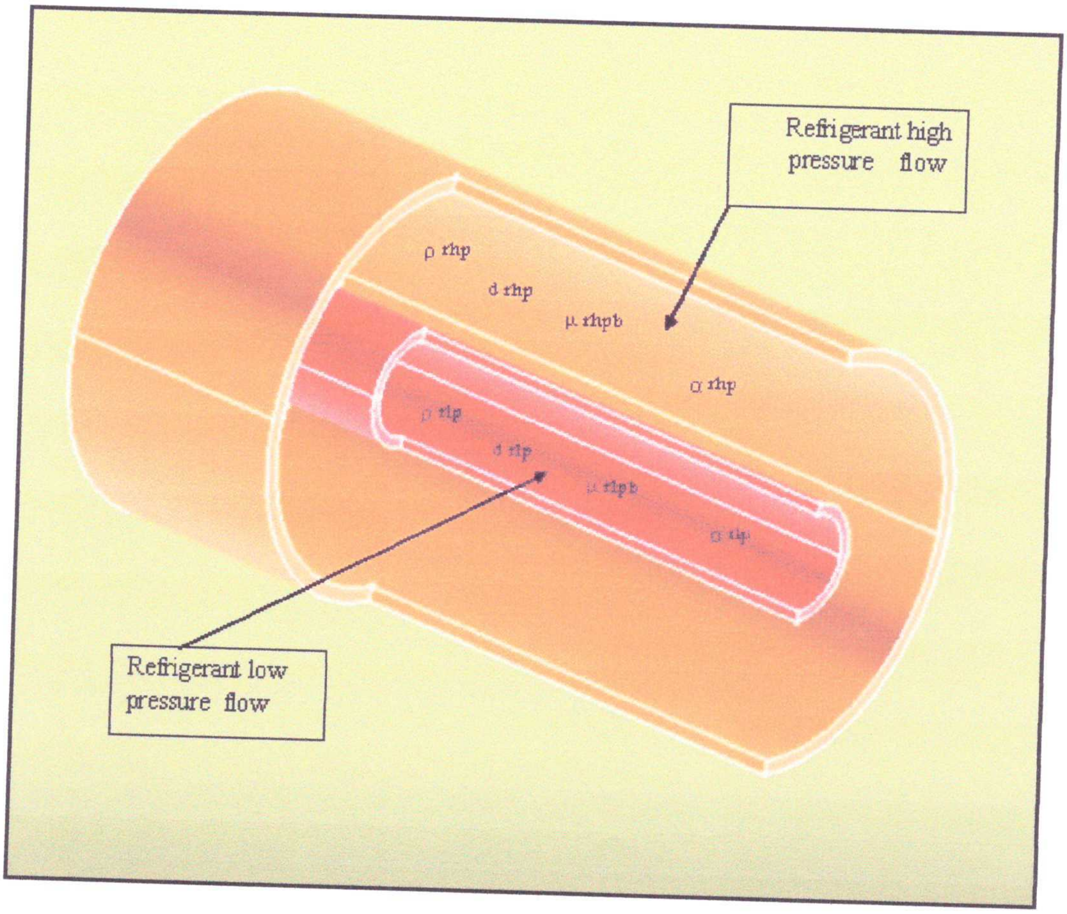


Figure 3.2: A computational segment of internal heat exchanger .

3.3.1 Calculation of heat transfer rate for internal heat exchanger

Employing energy balance, heat transfer in internal heat exchanger is given by:

$$\dot{Q}_{HE} = \dot{m}_{ref} \times (h_3 - h_4)$$

Where:

$\dot{Q}_{HE}$: Internal heat exchanger heat transfer rate [W].

$\dot{m}_{ref}$: Refrigerant mass flow rate [kg/s] which equal (0.16[m/s]), is found in chapter tow

h_4: Specific enthalpy for refrigerant before the internal heat exchanger [kJ/kg] which equals (282[kJ/kg]), is found in chapter tow

h_3: Specific enthalpy for refrigerant after the internal heat exchanger [kJ/kg] which equal (270 [kJ/kg]), is found in chapter tow

So the heat transfer rate for internal heat exchanger is equal:

$$\dot{Q}_{HE} = \dot{m}_{ref} \times (h_3 - h_4)$$

$$\dot{Q}_{HE} = 0.0045 \times (282 - 270) = 0.054\ [kW]$$

The mass flow rate ($\dot{m}_{ref}$) of the CO_2 refrigerant through the high pressure refrigerant pipe of the internal heat exchanger is the same for the low pressure refrigerant pipe of it.

3.3.2 Calculation Of over all heat transfer coefficient area of the heat transfer

Employing LMTD expression, heat transfer in internal heat exchanger is given by:

$$\dot{Q}_{HE} = (UA)_{HE} \frac{(T_3 - T_1) - (T_4 - T_7)}{\ln((T_3 - T_1)/(T_4 - T_7))}$$

Where:

$(UA)_{HE}$: Over all heat transfer coefficient area for the internal heat exchanger [W/K].

T_1: The temperature of the refrigerant at the outlet of low pressure refrigerant pipe in internal heat exchanger [°C] which equal (5.5[°C]), is found in chapter tow

T_7: The temperature of the refrigerant at the inlet of low pressure refrigerant pipe in internal heat exchanger [°C] which equal (-5[°C]), is found in chapter tow.

T_3: The temperature of the refrigerant at the inlet of high pressure refrigerant pipe in internal heat exchanger [°C] which equal (30[°C]), is found in chapter tow.

T_4: The temperature of the refrigerant at the outlet of high pressure refrigerant pipe in internal heat exchanger [°C] which equal (27.2[°C]), is found in previous chapter tow

So the Over all heat transfer coefficient area of the heat transfers for internal heat exchanger equal:

$$480 = (UA)_{HE} \frac{(30-5.5)-(27.2--5)}{\ln((30-5.5)/(27.2--5)}$$

$$(UA)_{HE} = 2.2[W/K].$$

3.3.3 Calculation of the high pressure refrigerant pipe diameter.

- The velocity of the refrigerant enters the internal heat exchanger at liquid phase, is supposed to equal :

u_{rhp} = 15 [m/s].

- The wall between the high pressure and low pressure pipes in the internal heat exchanger has a temperature which is supposed to equal:
  T_w = 11[°C].

- The reference temperature of the refrigerant is to be equals:

$$T_{refernce}[^\circ C] = \frac{T_w[^\circ C] + T_{average}[^\circ C]}{2}$$

To find the average temperature of the refrigerant that flows inside the high pressure refrigerant pipe, the following equation is to be used:

$$T_{average} = (T_3 + T_4)/2$$

$$T_{average} = \frac{30 + 27.5}{2} = 28.75[^\circ C].$$

Then,

$$T_{refernce} = \frac{28.75 + 11}{2} = 19.87[^\circ C].$$

By the principle of the mass conservation (continuity equation) for the carbon dioxide refrigerant that flow through the high pressure refrigerant pipe with supposed velocity (u_{rhp} =12 m/s), the diameter of it is calculated as the following:

$$\dot{m}_{rhp} = \rho_{rhp} \times u_{rhp} \times A_{rhp}$$

Where:

$\dot{m}_{rhp}$: Refrigerant mass flow rate through the high pressure refrigerant pipe [m/s] which equal (0.0045 m/s) ,is found in chapter tow.

ρ_{rhp}: The density of the refrigerant that flows through the high pressure refrigerant pipe at $T_{referncs}$[kg/m^3].

u_{rph}: Refrigerant velocity in the high pressure refrigerant pipe [m/s].

A_{rhp}: Cross sectional area of the high pressure refrigerant pipe [m^2].

By using Peace Software, the value of the carbon dioxide refrigerant density at $T_{referncs}$ and 81 bar equals[12]:

$$\rho_{rhp} = 830 kg/m^3$$

So the diameter of the high pressure refrigerant pipe equals :

$$d_{rhp} = \sqrt{\frac{4 \times \dot{m}_{rhp}}{\pi \times u_{rhp} \times \rho_{rhp}}}$$

$$d_{rhp} = \sqrt{\frac{4 \times 0.0045}{\pi \times 12 \times 830}} = 0.000073\ [m].$$

By using Copper Tube Hand Book from table A2., the select pip is type K with standard size 1/4 inch ,and its wall thickness is 0.035 inch(0 .000875m)[13].

3.3.4 Calculation of Reynolds number for the refrigerant in the high pressure refrigerant pipe.

- Reynolds number at reference temperature for the refrigerant that flows inside the high pressure refrigerant pipe is calculated by the following equation:

$$Re_{rhpb} = \frac{\rho_{rhpb} \times u_{rhp} \times d_{rhp}}{\mu_{rhpb}}$$

Re_{rhpb}: Reynolds number at reference temperature for the refrigerant in the high pressure refrigerant pipe dimensionless unit.

d_{rhp}: The diameter of the high pressure refrigerant pipe [m].

μ_{rhpb}: *Viscosity of the refrigerant in the high pressure refrigerant pipe at* $T_{refernce}[kg/(m.s)]$

By using Peace Software, the viscosity value of the carbon dioxide refrigerant that flows inside the high pressure refrigerant pipe at $T_{refernce}$ and 81 bar equals [12]:

$$\mu_{rhpb} = 7.88 \times 10^{-5} \, [kg/(m.s)]$$

So Reynolds number at reference temperature equal:

$$Re_{rhpb} = \frac{830 \times 12 \times 7.88 \times 10^{-4}}{7.88 \times 10^{-5}} = 95890$$

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

$$Re_{rhpw} = \frac{\rho_{rhpw} \times u_{rhp} \times d_{rhp}}{\mu_{rhpw}}$$

Re_{rhpw}: Reynolds number at wall temperature for the refrigerant in the high pressure refrigerant pipe dimensionless unit.

d_{rhp}: The diameter of the high pressure refrigerant pipe [m].

μ_{rhpw}: *Viscosity of the refrigerant in the high pressure refrigerant pipe at* T_{wall} $[kg/m.s]$

By using Peace Software, the viscosity value and density of the carbon dioxide refrigerant that flows inside the high pressure refrigerant pipe at T_{wall} and 81 bar equals[12]:

$$\mu_{rhpw} = 9.51 \times 10^{-5} \, [kg/m.s]$$

$$\rho_{rhp} = 896 [kg/m^3]$$

So Reynolds number at wall temperature equals:

$$Re_{rhpw} = \frac{896 \times 12 \times 7.88 \times 10^{-4}}{9.51 \times 10^{-5}} = 88186$$

3.3.5 Calculation of Nusselt number for the refrigerant in the high pressure refrigerant pipe.

Nusselt number for the refrigerant that flows inside the high pressure refrigerant pipe, is calculated by the following equation:

$$Nu_{rhp} = \left(\frac{Nu_{rhpw} + Nu_{rhpb}}{2}\right)\frac{K_{rhpw}}{K_{rhpb}} \quad ,$$

Where,

Nu_{rhp} :Nusselt number for the refrigerant in the high pressure refrigerant pipe dimensionless number .

Nu_{rhpb} : Nusselt number at reference temperature for the refrigerant in the high pressure refrigerant pipe dimensionless number.

Nu_{rhpw} : Nusselt number at wall temperature for the refrigerant in the high pressure refrigerant pipe dimensionless number.

k_{rhpb} : Thermal conductivity for the refrigerant in the high pressure refrigerant pipe at reference (bulk) temperature [W/m.K].

k_{rhpw} : Thermal conductivity for the refrigerant in the high pressure refrigerant pipe at wall temperature [W/ m.K].

- To find Nusselt number at wall temperature for the refrigerant that flows inside the high pressure refrigerant pipe, the following equation should be used:

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

Where:

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

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

To find Friction factor at wall temperature for the refrigerant that flows inside the high pressure refrigerant pipe, the following equation should be used:

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

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

$$f_{rhpw} = (0.79\ln(88186) - 1.64)^{-2} = 0.0184$$

By using Peace Software, the value of Prandtle number for the refrigerant that flows inside the high pressure refrigerant pipe at T_{wall} and 81 bar equals [12]:

$\mathrm{Pr}_{rhpw} = 2.27$ This number exists within Gnielinski range of Prandtle number.

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

$$Nu_{rhpw} = \frac{(0.0184/8)(717930 - 1000)2.27}{107 + 12.7(0.0184/8)^{1/2}(2.27^{2/3} - 1)} = 302$$

- To find Nusselt number at reference temperature for the refrigerant that flows inside the high pressure refrigerant pipe, the following equation should be used:

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

Where:

f_{rhpb} : Friction factor at reference temperature for the refrigerant in the high pressure refrigerant pipe dimensionless number.

$\Pr_{rhpb}$: Prandtle number at reference temperature for the refrigerant in the high pressure refrigerant pipe dimensionless number.

To find Friction factor at reference temperature for the refrigerant that flows inside the high pressure refrigerant pipe, the following equation should be used:

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

By using the value of Reynolds number at reference temperature for the refrigerant in the high pressure refrigerant pipe which is calculated previously, the friction factor will be equal:

$$f_{rhpb} = (0.79\ln(95890) - 1.64)^{-2} = 0.0181$$

By using Peace Software, the value of Prandtle number for the refrigerant that flows inside the high pressure refrigerant pipe at $T_{refernce}$ and 81 bar equals [12]:

$\Pr_{hpb} = 2.47$ This number exists within Gnielinski range of Prandtle number.

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

$$Nu_{rhpb} = \frac{(0.018/8)(802614-1000)2.47}{107+12.7(0.018/8)^{1/2}(2.47^{2/3}-1)} = 353.$$

By using Peace Software, the value of the thermal conductivity at T_{wall} and 81 bar equals [12]:

K_{rhpw} =0.105 [W/m.°C].

By using Peace Software, the value of the thermal conductivity at $T_{refernce}$ and 81bar equals [12]:

K_{rhpb} =0.094 [W/m.°C].

Then Nusselt number for the refrigerant in the high pressure refrigerant pipe equals:

$$Nu_{rhp} = \left(\frac{Nu_{rhpw} + Nu_{rhpb}}{2}\right)\frac{K_{rhpw}}{K_{hprb}} ,$$

$$Nu_{rhp} = \left(\frac{302+353}{2}\right)\frac{0.105}{0.094} = 365$$

3.3.6 Calculation of convection heat transfer coefficient for the refrigerant in the high pressure refrigerant pipe.

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

$$\alpha_{rhp} = \frac{Nu_{rhp}}{d_{rhp}} K_{rhpb}$$

Where,

α_{rhp} : Convection heat transfer coefficient for the refrigerant in the high pressure refrigerant pipe [W/m^2.K].

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

$$\alpha_{rhp} = \frac{365}{7.8 \times 10^{-4}} \times 0.094 = 44086[\ W/m^2.c]$$

3.3.7 Calculation of the low pressure refrigerant pipe diameter.

- The velocity of the refrigerant enters the internal heat exchanger at vapor phase, is supposed to equal :

  u_{rlp} = 15[m/s].

- The wall between the high pressure and low pressure pipes in the internal heat exchanger has a temperature which is supposed to equal:

  $T_w = 11\ ^{o}C$.

- The reference temperature of the refrigerant is to be equals:

$$T_{refernce}[^{\circ}C] = \frac{T_w[^{\circ}C] + T_{average}[^{\circ}C]}{2}$$

To find the average temperature of the refrigerant the following equation should be used:

$$T_{average} = (T_1 + T_7)/2$$

$$T_{average} = \frac{-5 + 5.5}{2} = 0.25[^{\circ}C]$$

Then,

$$T_{refernce} = \frac{11 + 0.25}{2} = 5.625[^{\circ}C]$$

By the principle of the mass conservation (continuity equation) for the carbon dioxide refrigerant that flow through the low pressure refrigerant pipe with supposed velocity (u_{rlp} =15 m/s), the diameter of it is calculated as the following:

$$\dot{m}_{rlp} = \rho_{rlp} \times u_{rlp} \times A_{rlp}$$

Where:

$\dot{m}_{rlp}$: Refrigerant mass flow rate through the low pressure refrigerant pipe [m/s] which equal (0.16 [m/s]), is found in chapter tow.

ρ_{rlp}: The density of the refrigerant that flows through the low pressure refrigerant pipe at $T_{refernce}$[kg/m^3].

u_{rlp}: Refrigerant velocity in the low pressure refrigerant pipe [m/s].

A_{rlp}: Cross sectional area of the low pressure refrigerant pipe [m^2].

By using Peace Software, the value of the carbon dioxide refrigerant density at $T_{refernce}$ and 22 bar equals [12]:

$$\rho_{rlp} = 50.2 kg/m^3$$

So the diameter of the low pressure refrigerant pipe equals:

$$d_{rlp} = \sqrt{\frac{4 \times \dot{m}_{rlp}}{\pi \times u_{rlp} \times \rho_{rlp}}}$$

$$d_{rlp} = \sqrt{\frac{4 \times 0.0045}{\pi \times 15 \times 50.2}} = 0.00027 [m].$$

By using Copper Tube Hand Book from table A2., the select pip is type K with standard size 3/8 inch ,and its wall thickness is 0.049 inch(0 .00124m)[13].

3.3.8 Calculation of Reynolds number for the refrigerant in the low pressure refrigerant pipe.

- Reynolds number at reference temperature for the refrigerant that flows inside the low pressure refrigerant pipe is calculated by the following equation:

$$\mathrm{Re}_{rlpb} = \frac{\rho_{rlpb} \times u_{rlp} \times d_{rip}}{\mu_{rlpb}}$$

Re_{rlpb}: Reynolds number at reference temperature for the refrigerant in the low pressure refrigerant pipe dimensionless unit.

d_{rlp}: The diameter of the low pressure refrigerant pipe [m].

μ_{rlpb}: *Viscosity of the refrigerant in the low pressure refrigerant pipe at* $T_{refernce}$ $[kg/(m.s)]$

By using Peace Software, the viscosity value of the carbon dioxide refrigerant that flows inside the low pressure refrigerant pipe at $T_{refernce}$ and 22 bar equals[13]:

$\mu_{rlpb} = 1.43 \times 10^{-5}\ [kg/m.s]$

So Reynolds number at reference temperature equals:

$$\mathrm{Re}_{rlpb} = \frac{50.2 \times 15 \times 0.0027}{1.43 \times 10^{-5}} = 145289.$$

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

$$\mathrm{Re}_{rlpw} = \frac{\rho_{rlpw} \times u_{rlp} \times d_{rlp}}{\mu_{rlpw}}$$

Re_{rlpw}: Reynolds number at wall temperature for the refrigerant in the low pressure refrigerant pipe dimensionless unit.

d_{rlp}: The diameter of the low pressure refrigerant pipe [m].

μ_{rlpw}: *Viscosity of the refrigerant in the low pressure refrigerant pipe at* T_{wall} *[kg /m.s]*

By using Peace Software, the viscosity value of the carbon dioxide refrigerant that flows inside the high pressure refrigerant pipe at T_{wall} and 22 bar equals[12]:

$\mu_{rlpw} = 1.45 \times 10^{-5}\ [kg/m.s]$

$\rho_{rlpw} = 48.3 kg/m^3$

So Reynolds number at wall temperature equals :

$$\mathrm{Re}_{rlpw} = \frac{48.3 \times 15 \times 0.0027}{1.45 \times 10^{-5}} = 114920.$$

3.3.9 Calculation of Nusselt number for the refrigerant in the low pressure refrigerant pipe.

Nusselt number for the refrigerant that flows inside the low pressure refrigerant pipe is calculated by the following equation:

$$Nu_{rlp} = \left(\frac{Nu_{rlpw} + Nu_{rlpb}}{2}\right)\frac{K_{rlpw}}{K_{rlpb}} \quad ,$$

Where,

Nu_{rlp} :Nusselt number for the refrigerant in the low pressure refrigerant pipe dimensionless number .

Nu_{rlpb} : Nusselt number at reference temperature for the refrigerant in the low pressure refrigerant pipe dimensionless number.

Nu_{rlpw} : Nusselt number at wall temperature for the refrigerant in the low pressure refrigerant pipe dimensionless number.

k_{rlpb} : Thermal conductivity for the refrigerant in the low pressure refrigerant pipe at reference (bulk) temperature [W/m.K].

k_{rlpw} : Thermal conductivity for the refrigerant in the low pressure refrigerant pipe at wall temperature [W/ m.K].

- To find Nusselt number at wall temperature for the refrigerant that flows inside the low pressure refrigerant pipe, the following equation should be used:

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

Where:

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

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

To find Friction factor at wall temperature for the refrigerant that flows inside the low pressure refrigerant pipe, the following equation should be used:

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

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

$$f_{rlpw} = (0.79\ln(114920) - 1.64)^{-2} = 0.0174$$

By using Peace Software, the value of Prandtle number for the refrigerant that flows inside the low pressure refrigerant pipe at T_{wall} and 22 bar equals [12]:

$Pr_{rlpw} = 0.93$ This number exists within Gnielinski range of Prandtle number.

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

$$Nu_{rlpw} = \frac{(0.0174/8)(114920 - 1000)0.93}{1.07 + 12.7(0.0174/8)^{1/2}(0.93^{2/3} - 1)} = 221$$

- To find Nusselt number at reference temperature for the refrigerant that flows inside the low pressure refrigerant pipe, the following equation should be used:

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

Where:

f_{rlpb}

: Friction factor at reference temperature for the refrigerant in the low pressure refrigerant pipe dimensionless number.

$\Pr_{rlpb}$: Prandtle number at reference temperature for the refrigerant in the low pressure refrigerant pipe dimensionless number.

To find Friction factor at reference temperature for the refrigerant that flows inside the low pressure refrigerant pipe, the following equation should be used:

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

By using the value of Reynolds number at reference temperature for the refrigerant in the low pressure refrigerant pipe which is calculated previously, the friction factor will be equal:

$$f_{rlpb} = (0.79\ln(145289) - 1.64)^{-2} = 0.0166$$

By using Peace Software, the value of Prandtle number for the refrigerant that flows inside the low pressure refrigerant pipe at $T_{refernce}$ and 22 bar equals [12]:

$\Pr_{rlpb} = 0.95$ This number exists within Gnielinski range of Prandtle number.

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

$$Nu_{rlpb} = \frac{(0.0166/8)(145289-1000)0.95}{1.07+12.7(0.0166/8)^{1/2}(0.95^{2/3}-1)} = 477.5$$

By using Peace Software, the value of the thermal conductivity at T_{wall} and 22 bar equals [12]:

K_{rlpw} =0.0172 [W/m.°C] .

By using Peace Software, the value of the thermal conductivity at $T_{refrence}$ and 22bar equals [12]:

K_{rlpb} =0.017 [W/m.°C] .

Then Nusselt number for the refrigerant in the low pressure refrigerant pipe

equal:

$$Nu_{rlp} = \left(\frac{Nu_{rlpw} + Nu_{rlpb}}{2}\right)\frac{K_{rlpw}}{K_{hlrb}} ,$$

$$Nu_{rlp} = \left(\frac{221+477}{2}\right)\frac{0.0172}{0.017} = 353$$

3.3.10 Calculation of convection heat transfer coefficient for the refrigerant in the low pressure refrigerant pipe.

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

$$\alpha_{rlp} = \frac{Nu_{rlp}}{d_{rlp}} K_{rlpb}$$

Where,

α_{rlp} : Convection heat transfer coefficient for the refrigerant in the low pressure refrigerant pipe [W/m^2.K].

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

$$\alpha_{rlp} = \frac{353}{0.00027} \times 0.017 = 2609[W / m^2.s].$$

3.3.11 Calculation of internal heat exchanger length.

Overall heat transfer coefficient for the internal heat exchanger has been calculated by using fundamental equation for overall heat transfer coefficient yields:

$$\frac{1}{(UA)_{he}} = \frac{1}{\alpha_{rlp} A_{rlp}} + \frac{\ln(d_0 / d_{rhp})}{2\pi L_{he} K_w} + \frac{1}{\alpha_{rhp} A_{rhp}}$$

Where:

d_{rhp}: The inner diameter of the high pressure refrigerant pipe[m].

d_o: The outer diameter of the high pressure refrigerant pipe[m].

K_w: Thermal conductivity for copper wall [W/m.k].

L_{he}: Internal heat exchanger length [m].

A_{rlp}: Heat transfer area of the low pressure refrigerant pipe [m^2].

A_{rhp}: Heat transfer area of the high pressure refrigerant pipe [m^2].

To find heat transfer area of the low pressure refrigerant pipe, the following equation is to be used:

$$A_{rlp} = \pi d_{rlp} L_{he}$$

Where:

d_{rlp}: The inner diameter of the low pressure refrigerant pipe[m].

To find the heat transfer area of the high pressure refrigerant pipe, the following equation should be used:

$$A_{rhp} = \pi d_{rhp} L_{he}$$

To find the internal heat exchanger length, the following equation should be used:

$$L_{he} = ((UA)_{he} \frac{(2K_w a_{rhp} d_{rhp} + d_{rhp} d_{rlp} a_{rhp} a_{rlp} \ln(d_o / d_{rhp}) + 2d_{rlp} a_{rlp} K_w)}{2\pi (d_{rlp} a_{rlp} K_w a_{rhp} d_{rhp})}$$

$$L_{hs} = \frac{27((2\times 388\times 44086\times 0.000078+0.000078\times 0.0023\times 44086\times 2609\ln(^{0.00275}/_{0.0027})+2\times 0.0027\times 2609\times 388)}{2\pi(0.0027\times 2609\times 388\times 0.000078\times 44086)}$$

$$L_{hs} = 0.14[m]$$

3.3.12 Calculation of pressure drop in the internal heat exchanger.

The pressure drop in the high pressure refrigerant pipe of the internal heat exchanger is given by:

$$\Delta p_{hp} = \frac{G^2{}_{hp}}{2\rho_{rhpb}}(\xi_{rhp}\frac{L_{he}}{d_{rhp}}+1.2)$$

Where:

Δp_{hp}: Pressure drop in high pressure refrigerant pipe [bar].

G_{hp}^2 :Mass velocity of the refrigerant in the higher pressure pip [k/m^2.s]

ζ_{rhp}: friction factor for the refrigerant in high pressure refrigerant pip dimensionless unit.

The mass velocity of the refrigerant that flows in the high pressure refrigerant pipe of the internal heat exchanger is given by:

$$G_{hp} = \frac{m^{\cdot}_{rhp}}{A_{rhp}}$$

Where:

A_{rhp}: Cross sectional area of the high pressure refrigerant pipe [m^2].

Then the mass velocity of the refrigerant in the high pressure refrigerant pipe equal:

$$G_{hp} = \frac{0.0045}{(\pi \times 0.00635^2)/4} = 1263[\ kg/m^2.s\ ].$$

The friction factor for refrigerant in the high pressure refrigerant pipe of the internal heat exchanger is given by:

$$\xi_{rhp} = (1.82\ln(\mathrm{Re}_{rhpw}) - 1.64)^{-2} \frac{\rho_{rhpw}}{\rho_{rhpb}} (\frac{\mu_{rhpw}}{\mu_{rhpb}})^{s}$$

Where;

s: Exponent Factor [Wm^2.s/kg].

The exponent factor for refrigerant in the high pressure refrigerant pipe of the internal heat exchanger is given by:

$$s = 0.023\left|\frac{Q_{he}}{G_{rhp}}\right|^{0.42}$$

Then ,the exponent factor for the refrigerant in the high pressure refrigerant pipe equals:

$$s = 0.023\left|\frac{480}{1263}\right|^{0.42} = 0.015\ [W.m^2.s/kg]$$

Then ,the friction factor for the refrigerant in the high pressure refrigerant pipe equals:

$$\xi_{rhp} = (1.82\ln(717930) - 1.64)^{-2}\frac{896}{830}(\frac{9.51\times10^{-5}}{7.88\times10^{-5}})^{0.0103} = 2.08\times10^{-3}\ .$$

Then, the pressure drop for the refrigerant in the high pressure refrigerant pipe equals:

$$\Delta p_{hp} = \frac{G^2{}_{hp}}{2\rho_{rhpb}}(\xi_{rhp}\frac{L_{he}}{d_{rhp}} + 1.2)$$

$$\Delta p_{he} = \frac{1263^2}{2\times830}(2.08\times10^{-3}\frac{0.95}{0.00635} + 1.2)\ = 0.014[bar]$$

The pressure drop in the low pressure refrigerant pipe of the internal heat exchanger is given by:

$$\Delta p_{lp} = \frac{G^2{}_{lp}}{2\rho_{rlpb}}(\xi_{rlp}\frac{L_{he}}{d_{rlp}} + 1.2)$$

Where:

Δp_{lp}: Pressure drop in low pressure refrigerant pipe [bar].

G_{lp}^2: mass velocity of the refrigerant in the low pressure refrigerant pipe [kg/m^2.s].

ζ_{rlp}: friction factor for the refrigerant in high pressure refrigerant pip dimensionless unit.

The mass velocity of the refrigerant that flows in the low pressure refrigerant pipe of the internal heat exchanger is given by:

$$G_{lp} = \frac{m^{\cdot}_{rlp}}{A_{rlp}}$$

Where:

A_{rlp}: Cross sectional area of the low pressure refrigerant pipe [m^2].

Then the mass velocity of the refrigerant in the low pressure refrigerant pipe equal:

$$G_{lp} = \frac{0.04}{(\pi \times 0.0095^2)/4} = 564\ [kg/m^2.s].$$

The friction factor for refrigerant in the low pressure refrigerant pipe of the internal heat exchanger is given by:

$$\xi_{rlp} = (1.82\ln(\mathrm{Re}_{rlpw}) - 1.64)^{-2} \frac{\rho_{rlpw}}{\rho_{rlpb}} \left(\frac{\mu_{rlpw}}{\mu_{rlpb}}\right)^{s}$$

Where,

s: Exponent factor $[Wm^2.s/kg]$

The exponent factor for refrigerant in the low pressure refrigerant pipe of the internal heat exchanger is given by:

$$s = 0.023\left|\frac{Q_{le}}{G_{rlp}}\right|^{0.42}$$

Then ,the exponent factor for the refrigerant in the low pressure refrigerant pipe equals:

$$s = 0.023\left|\frac{54}{564}\right|^{0.42} = 0.021\ (W.m^2.s)/kg$$

Then, the friction factor for the refrigerant in the low pressure refrigerant pipe equals:

$$\xi_{rlp} = (1.82\ln(475921) - 1.64)^{-2}\frac{48.3}{50}\left(\frac{1.45\times10^{=5}}{1.43\times10^{-5}}\right)^{0.021} = 2.017\times10^{-3}\ .$$

Then, the pressure drop for the refrigerant in the low pressure refrigerant pipe equals:

$$\Delta p_{lp} = \frac{G^2{}_{hp}}{2\rho_{rlpb}}\left(\xi_{rlp}\frac{L_{he}}{d_{rlp}} + 1.2\right)$$

$$\Delta p_{lp} = \frac{564^2}{2\times50}\left(2.017\times10^{-3}\frac{0.95}{0.0095} + 1.2\right) = 0.044bar$$

3.4 Evaporator Design

3.4.1 General Assumptions

The following assumptions were used in the calculation procedure of the mathematical model to predict the performance of flat-finned-tube heat exchanger (carbon dioxide refrigeration evaporator) which is applied in the low-temperature refrigeration systems.

1. Steady state operation.

2.Constant air-side convective heat transfer coefficient over the evaporator element/region.

3. Negligible refrigerant pressure drops in the evaporator.

4. Negligible heat loss/gain from the evaporator.

5. Same number of tubes in each circuit, each with the same fraction of the total mass flow rate. also constant mass flux of air, Gmax, along the length of the coil.

6. State of the refrigerant in one element/region being either two-phase or superheated.

7.Dividing evaporator in two regions(two-phase region, superheated region) .

8. Frosting process being quasi-steady, where parameters calculated at time t are used for calculations at time $t + \Delta t$.

9.Frosting and condensation cannot simultaneously occur on evaporator surface.

10. The superheated region being dry, whereas the two-phase region being frosted.

3.4.2 Evaporator Details

Details of evaporator used in carbon dioxide refrigerator which are shown in (figure 4.3)

Fin material : aluminum plate .

Tube material : aluminum .

Fin thickness(t_F) : 0.2 [mm].

Fin pitch (P_F) : 8 [mm].

No. of traverse tube rows : 5 .

No. of refrigerant tubes : 30 .

No. of longitudinal tube rows : 6 .

No. of circuit : 1 .

Evaporator length (L_e) : 250 [mm]

Evaporator high (H_e) :195 [mm]

Evaporator depth (D_e) :19 [mm]

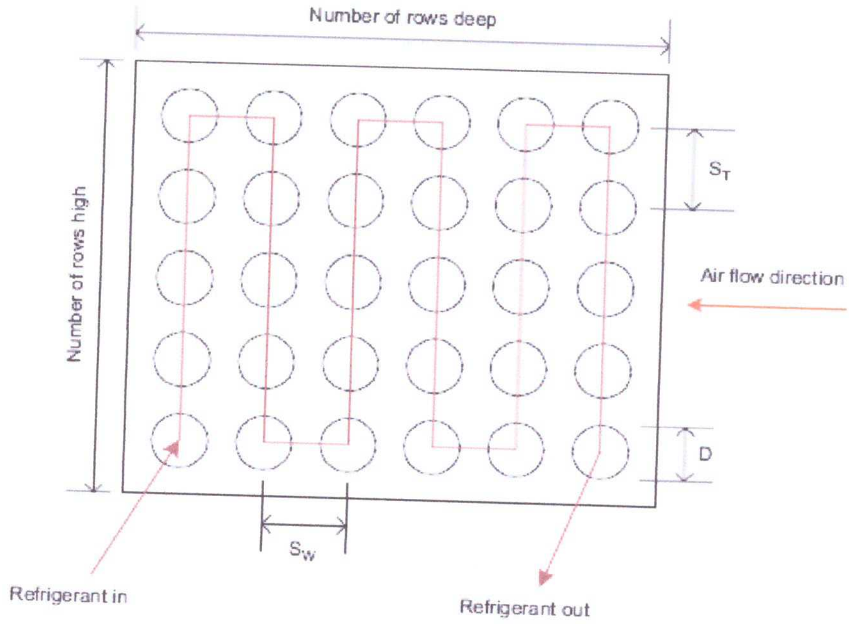


Figure 3.3: Schematic of evaporator used in CO2 refrigerator

From (figure 3.3) the traverse tube spacing (S_T) , and longitudinal tube spacing (S_w) are :

$$S_T = \frac{H_e}{number\ of\ traverse\ tube\ rows} = \frac{195}{5} = 39[mm]$$

$$S_W = \frac{D_e}{number\ of\ longitudinal\ \ tube\ rows} = \frac{19}{6} = 31.666\ [mm]$$

3.4.3 Calculation of refrigerant tube diameter

- The velocity of the refrigerant enters the refrigerant tube of the evaporator at one phase, is supposed to equal :

u_{ref} = 15 [m/s].

- The wall between the refrigerant and air in the evaporator has a temperature which is supposed to equal:

  T_w = -13 [°C] .

- The reference temperature of the refrigerant is to be equals:

$$T_{refernce}[°C] = \frac{T_w[°C] + T_{average}[°C]}{2}$$

To find the average temperature of the refrigerant that flows inside the refrigerant tube, the following equation should be used:

$$T_{average} = (T_{evi} + T_{evo})/2$$

$$T_{average} = \frac{-10 - 15}{2} = -12.5°C$$

Then,

$$T_{refernce} = \frac{-13 + -12.5}{2} = -12.75°C$$

By the principle of the mass conservation (continuity equation) for the carbon dioxide refrigerant that flows through the refrigerant tube with supposed velocity (u_{ref} =15 m/s),the diameter of the inside tube (refrigerant tube) is calculated as the following:

$$\dot{m}_{ref} = \rho_{ref} \times u_{ref} \times A_{ref}$$

Where :

ρ_{ref}: The density of the refrigerant at $T_{refernce}$[Kg/m^3].

u_{ref}: Refrigerant velocity [m/s].

A_{ref}: Cross sectional area of the refrigerant tube [m^2].

By using Pace software, the value of the carbon dioxide refrigerant density at $T_{refernce}$ and 22 bar equals[12]:

$$\rho_{ref} = 56.4\ Kg/m^3$$

So the inner diameter of the refrigerant tube equals:

$$d_{ref} = \sqrt{\frac{4 \times \dot{m}_{ref}}{\pi \times u_{ref} \times \rho_{ref}}}$$

Then,

$$d_{ref} = \sqrt{\frac{4 \times 0.0045}{\pi \times 15 \times 56.4}} = 0.0026\ [m].$$

By using Standard Aluminum Tubing Size, table (A.2), the selected pipe has standard size 1/4 inch ,and its wall thickness is 0.0245inch(0 .0006125m).

So,

Inner refrigerant tube diameter (D_I) :2.6[mm].

Outer refrigerant tube diameter (D_O): 3.22[mm].

3.4.4 Calculation of frosted hydraulic diameter and fin efficiency.

Each element of the evaporator, (figure 3-4) and (figure 3-5) illustrate the fin and unfin (bare tubes) parts for it.

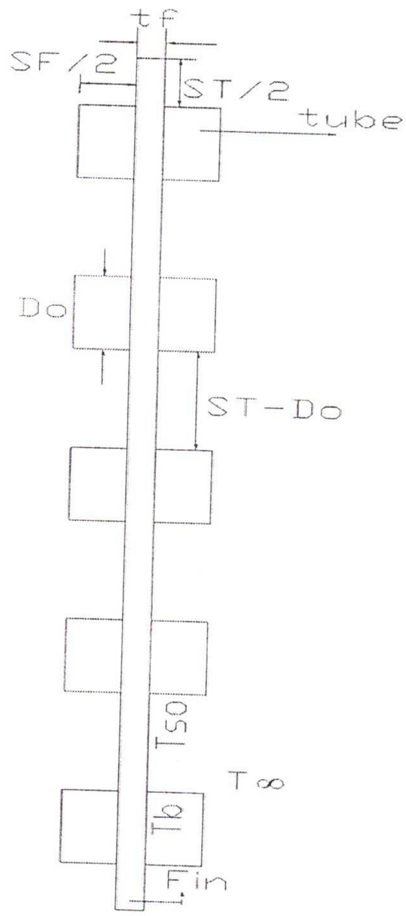


Figure (3-4): Finned and unfinned (bare tubes) parts for evaporator element

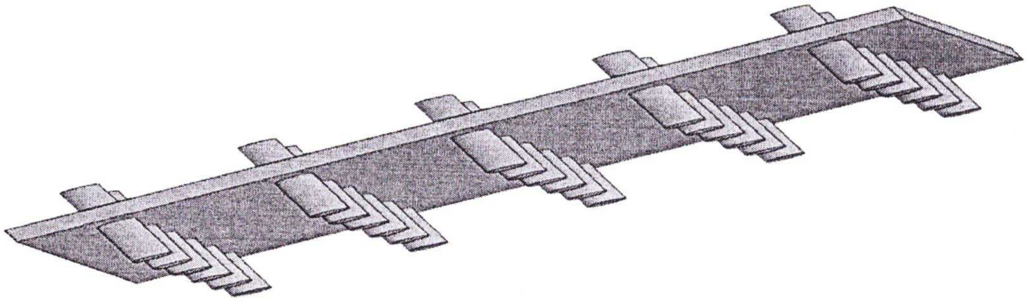


Figure (3-5): Finned and unfinned (bare tubes) parts for evaporator element

1. To find frosted hydraulic diameter, the following equation should be used:

$$DH = \frac{2 \times L_f \times S_F}{L_f + S_F}$$

Where,

DH: Frosted hydraulic diameter [m].

L_f: Fine high[m] .

S_F: Fin space [m] .

So ,

$$L_F = H_s - 5D_0 = 0.195 - (5 \times 0.00831) = 0.156[m]$$

$$S_F = P_F - t_F = 0.008 - 0.0002 = 0.0078\ [m]$$

$$DH = \frac{2(0.156 \times 0.0078)}{0.156 + 0.0078} = 0.0149[m]$$

2. To find fin efficiency, the following equation should be used:

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

Where :

η_f: Fin efficiency[dimensionless number].

m : mean flow conditions[dimensionless number].

$\hat{L}_f$: *corrected fin high[m]*

Where ,

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

Then ,

$$\acute{L}_f = \left(\frac{0.156}{2}\right)\left[1 + 0.35\ln\frac{\left(\frac{0.0032}{2} + \frac{0.156}{2}\right)}{\frac{0.0032}{2}}\right] = 0.1846\,[m]$$

Where ,

$$m = \sqrt{(2h_a/(k_{AL}t_F))}$$

k_{AL} : *Thermal conductivity for almumume fin plat*$[\frac{W}{m.\,°C}]$

Then,

$$m = \sqrt{\frac{2 \times 17}{202 \times 0.2 \times 10^{-3}}} = 0.029$$

So,

$$\eta_f = \frac{\tanh(0.029 \times 0.184)}{0.029 \times 0.184} = 0.98$$

3.4.5 Calculation of air side heat transfer coefficient

To estimate air side heat transfer coefficient, the following correlation for evaporator model should be used[14]:

$$h_a = \frac{j \times C_p \times G_a}{Pr_{DH}^{2/3}}$$

Where ,

h_a: Convection heat transfer coefficient for air side [W/(m^2.K)].

j : Colburn factor [dimensionless number].

C_p: Air specific heat at constant pressure [kJ/ kg. K] which is equal (1.0064kJ/kg).

G_a: Air mass flux [kg/m^2.s]

Pr_{DH}: Reynolds number based on frosted surface [dimensionless number].

By using Peace Software, at air temperature T_a= -7[^{0}C] and air pressure p_a= 1 [bar] the value of Pr_{DH} = 0.719.

- To find air mass flux that flows inside the evaporator , the following equation should be used:

$$G_a = \frac{\dot{m}_a}{A_{flux}}$$

Where,

A_{flux}: Area side of the evaporator at which the air flows through it [m^2].

Based on the measured mass flow rates of the refrigerant and assuming negligible difference between the heat transfer rates on the air- and refrigerant-side, the mass flow rate of air over the evaporator was calculated as:

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

Where,

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

$\dot{m}_r$: Refrigerant mass flow rate [kg/ s] which equal (0.04 kg/s), is found in chapter two.

$h_{r,out}$: Specific enthalpy for refrigerant at evaporator tube outlet [kJ/kg] which is equal (442 kJ/kg), is found in chapter two.

$h_{r,in}$: Specific enthalpy for refrigerant at evaporator tube inlet [kJ/kg] which is equal (270kJ/kg), is found in chapter two.

$h_{s,a,in}$: Air enthalpy at evaporator inlet [kJ/ kg].

$h_{s,a,out}$: Air enthalpy at evaporator outlet [kJ/ kg].

By using Psychometrics Demo software at air temperature $T_a(_{dry\ bulb}) = -3[^0C]$ and T_a $(_{wet\ bulb}) = -7[^0C]$,so the air enthalpy at evaporator inlet equals[9]:

$h_{s,a,in} = -6.038$ [kJ/kg].

By using Psychometrics Demo software, at air temperature $T_a(_{dry\ bulb}) = -10[^0C]$ and $T_a(_{wet\ bulb}) = -13[^0C]$,so the air enthalpy at the evaporator outlet equals[9]:

$$h_{s,a,out} = -9.984[\frac{\text{kJ}}{\text{kg}}].$$

So,

$$\dot{m}_a = \frac{m_r(h_{r,out}-h_{r,in})}{h_{s,a,in}-h_{a,out}} = \frac{0.0045\,(442-270)}{-6.038-9.984} = 0.196[\text{kg/s}]$$

$$A_{flux} = L_s \times H_s = 0.195 \times 0.25 = 0.0488[m^2].$$

$$G_a = \frac{\dot{m}_a}{A_{flux}} = \frac{0.196}{0.0487} = 4.026\left[\frac{\text{kg}}{\text{m}^2}.s\right].$$

❖ To find Colburn factor for the air that flows over the evaporator which has three or more than three tubes , the following equation should be used:

$$j = 0.24Re_{DH}^{-0.409}\left({}^{S_T}/_{S_L}\right)^{0.425}\left({}^{S_F}/_{D_o}\right)^{-0.035} \qquad if\ N_t \geq 3$$

Where,

Re_{DH}: Reynolds number based on frosted surface [dimensionless number].

S_T: traverse tube spacing [m]

S_w: longitudinal tube spacing [m]

D_o: The outer diameter of the refrigerant tube [m]

To find Reynolds number based on frosted surface for the air, the following equation should be used:

$$Re_{DH} = \frac{\rho_a \times v_a \times DH}{\mu_a}$$

Where,

ρ_a: The density of the air [Kg/m^3].

v_a: The velocity of the air[m/s].

μ_a:Viscosity of air$[\frac{kg}{(m.s)}]$

Where,

$$v_a = \frac{\dot{m}_a}{\rho_a \times A_{flux}}$$

Where,

$\dot{V}_a$: The volume flow rate of the air$[m^3/s]$

By using Peace Software , at air temperature $T_{a(dry\ bulb)}$ = -3[^{0}C] and $T_{a\ (wet\ bulb)}$ = -7[^{0}C] , so the value of air density ρ_a =1.34 [kg/m^3]and the value of air dynamic viscosity $\mu_a = 16.7 \times 10^{-6}[\frac{kg}{(m.s)}]$.

Then,

$$\dot{V}_a = \frac{\dot{m}_a}{\rho_a} = \frac{1.742}{1.34} = 1.3[\frac{m^3}{s}]$$

Then,

$$v_a = \frac{\dot{m}_a}{\rho_a \times \dot{V}_a \times A_{flux}} = \frac{1.742}{1.34 \times 0.0488} = 20.49[m/s]$$

Then,

$$Re_{DH} = \frac{\rho_a \times v_a \times DH}{\mu_a} = \frac{1.34 \times 20.49 \times 0.0149}{16.779 \times 10^{-6}} = 24426.744[m]$$

$$j = 0.24 \times 24426.744^{(-0.409)} (3.9/_{3.16})^{0.425} (7.8/_{3.2})^{-0.035} = 4.2112 \times 10^{-3}$$

So ,

$$h_a = \frac{4.2112 \times 10^{-3} \times 1.0064 \times 10^3 \times 35.37}{0.71^{2/3}} = 17\ [W/m^2 c]$$

3.4.6 Calculation for number of elements of the evaporator

- For one element fine part[14].

$$A_{fin} = [(D_e \times L_s) - (A_{tube} \times N_{tube})]$$

Where,

A_{tube}:tube cross sectional area[m^2].

D_e: Evaporator depth[m].

L_e: Evaporator length[m].

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

N_{tube}: number of tube.

Then,

$$A_{fin} = 2[(0.19 \times 0.195) - ((\pi(0.00831)^2/4) \times 30)\ ] = 0.071233m^2$$

To find heat transfer rate per flat fin, the following equation is to be used:

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

Where,

q_{fin}: heat transfer rate per flat fin[W]

T_∞: surrounding air temperature at evaporator inlet [^{0}C]which is equal (-3[^{0}C])which equal (-3[^{0}C])

T_b: tube surface temperature at fin base [^{0}C] which is equal (-13[^{0}C])

Then,

$q_{fin} = 17 \times 0.071 \times (-3 - -13) \times 0.98 = 11.82\ W$per fin element

❖ For bare tubes (unfin tubes) of one element of the evaporator.

$$A_{bare\ tubes} = N_{bare\ tubes} \times D_0 \times \pi \times s_F$$

Where,

$A_{bare\ tubes}$: bare tubes(unfin tubes) surface area for one element[m].

$N_{bare\ tubes}$: number of bare tubes for one element.

Then,

$$A_{bare\ tube} = 30 \times 0.00831 \times \pi \times 0.0078 = 6.1089 \times 10^{-3} m^2$$

To find heat transfer rate for bare tubes of one element , the following equation should be used:

$$q_{un\ fin} = h_a \times A_{bare\ tubes} \times (T_\infty - T_{so})$$

Where,

q_{unfin}: heat transfer rate for bare tubes[W].

T_{so}: air temperature at bare tubes surface [^{0}C]which equal (-10[^{0}C])

$$q_{un\ fin} = 17 \times 2.058 \times 10^{-3} \times (-3 - -10) = 0.349[W]$$

$$q_{total\ for\ one\ element} = q_{fin} + q_{un\ fin} = 11.82 + 0.34 = 12.17[W]$$

So,

The number of elements of the evaporator $= \frac{Q_s}{q_{total}} = \frac{780}{12.7} = 64\ element$

3.4.7 calculation of Refrigerant-side heat transfer coefficient

1- For tow phase

❖ Calculation of Reynolds number for tow phase refrigerant.

Reynolds number at reference temperature for the refrigerant that flows inside the tow phase refrigerant tube is calculated by the following equation:

$$\mathrm{Re}_D = \frac{\rho_{refb} \times u_{ref} \times D_I}{\mu_{refb}}$$

Re_D: Reynolds number at reference temperature for the refrigerant in the refrigerant pipe dimensionless unit.

μ_{refb}: $Viscosity\ of\ tow\ phase\ refrigerant\ in\ the\ refrigerant\ tube\ at\ T_{refernce}[\frac{kg}{(m.s)}]$

.

By using Peace Software, the viscosity value of the carbon dioxide refrigerant that flows inside the tow phase refrigerant tube at $T_{refernce} = -12°C$ and 22 bar is equals :

$\rho_{ref} = 56.7[kg/m^3]$

$\mu_{ref} = 1.38 \times 10^{-5} \, [Kg/m.s]$

So Reynolds number at reference temperature equals :

$$\mathrm{Re}_D = \frac{56.7 \times 15 \times 0.0026}{1.38 \times 10^{-5}} = 160239$$

- Calculation of convection heat transfer coefficient for the refrigerant in the refrigerant tube

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

$$h_{r,tp} = F \times h_L$$

Where,

h_L : Convection heat transfer coefficient for the refrigerant at liquid phase [W/(m^2.K)].

F : Dittus –Boelter factor [dimensinless number]

Then the convection heat transfer coefficient for the liquid refrigerant in the tow phase refrigerant tube equals:

$$h_L = 0.023 \frac{K_L}{D_l} \mathrm{Re}_l^{0.8} \mathrm{Pr}_l^{0.4}$$

Where,

k_L:thermal conductivity for liquid refrigerant[W/m.°C]

Re_l: Reynolds number for liquid refrigerant[dimensinless number]

Pr_L: Prandtle number for liquid refrigerant[dimensinless number]

By using Peace Software[4] the physical properties of the carbon dioxide refrigerant at $T_{refernce} = -12°C$ and 22 bar is equal:

k_L=0.112 [W/m.°C]

$Pr_L = 1.13$

Then the convection heat transfer coefficient for the liquid refrigerant in the refrigerant tube equals:

$$h_L = 0.023 \frac{0.112}{0.0062} 160239^{0.8} 1.04^{0.4} = 615.43 [W/m^2.s]$$

Dittus –Boelter factor is calculated as:

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

Where,

X_{tt}: Lockhart_Martinelli[dimensinless number]

Where, Lockhart_Martinelli factor is calculated as:

$$X_{u} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_{v}}{\rho_{1}}\right)^{0.5} \left(\frac{\mu_{1}}{\mu_{v}}\right)^{0.1}$$

Where,

X:quality of saturated carbon dioxide liquid vapor mixture [dimensinless number]

ρ_v: refrigerant density at vapor phase [kg/m^3].

ρ_L: refrigerant density at liquid phase [kg/m^3].

μ_v: refrigerant dynamic viscosity at vapor phase [kg/m^3].

μ_L: refrigerant dynamic viscosity at liquid phase [kg/m^3].

By using Cool Back, the quality value of the carbon dioxide refrigerant= 0.64.

By using Peace Software, the physical properties of the carbon dioxide refrigerant at $T_{refernce} = -12°C$ and 22 bar is equals:

$\rho_v = 58.8$ [kg/m^3].

$\rho_L = 1004$ [kg/m^3]

$\mu_v = 1.38 \times 10^{-5}$ [kg/m.s].

$\mu_L = 0.000115$[kg/m. s].

Then,

$$X_u = (\frac{1-0.64}{0.64})^{0.9} (\frac{58.8}{1004})^{0.5} (\frac{0.000115}{1.38\times10^{-5}})^{0.1} = 0.178$$

Then,

$$F = 1 + 1.925 \times 0.17^{-0.83} = 9.25$$

Then ,

$$h_{r,tp} = 9.25 \times 615.43 = 5692 \; [W/m^2.s]$$

2-For superheated phase

The convection heat transfer coefficient for the refrigerant that flows inside the superheated region of the evaporator has been calculated in the same way as the convection heat transfer coefficient for the refrigerant that flows inside the low pressure pipe of the internal heat exchanger, and it equals:

$h_{sp,r}$ =1765 [W/m^2.°C]

3.4.8 calculation of overall evaporator efficiency

The air side total surface area for the evaporator is found by the following equation :

$$A_{total} = (A_{fin} + A_{bare\ tubes}) \times \#\ of\ elements\ + (29 \times A_{filet})$$

Where,

A_{filet}:filet heat transfer surface area[m^2]

And,

$$A_{filet} = \pi\left(\frac{D_o + s_T}{2}\right) + 0.02(\pi D_o)$$

Then,

$$A_{filet} = \pi\left(\frac{0.0083 + 0.039}{2}\right) + 0.02(\pi \times 0.0083) = 0.0936[m^2]$$

So,

$$A_{total} = (0.0733) \times\ 64 + (29 \times 0.0936) = 7.4[m^2]$$

- The total heat transfer surface area for the fins of the evaporator is found by the following equation:

$$A_{fins} = A_{fin} \times number\ of\ fins$$

Where,

Number of plat fins = Number of elements of the evaporator .

$$A_{fins} = 0.071233 \times 64 = 6.07[m^2]$$

- The total heat transfer surface area for the bare tubes of the evaporator is found by the following equation:

$$A_b = A_{total} - A_{fins}$$

Where,

A_b :Total heat transfer surface area for evaporator bare tubes[m^2] .

$$A_b = 7.4 - 4.55 = 2.73[m^2].$$

- The overall evaporator efficiency is found by the following equation:

$$\eta_o = 1 - \frac{A_{fins}}{A_{total}}(1 - \eta_f)$$

So,

$$\eta_o = 1 - \frac{4.55}{7.4}(1 - 0.60) = 0.98$$

3.4.9 Calculation of superheated tubes length of the evaporator .

Employing energy balance, The heat transfer rate of the superheated region of the evaporator, fig.(3-6) Illustrates it, is given by:

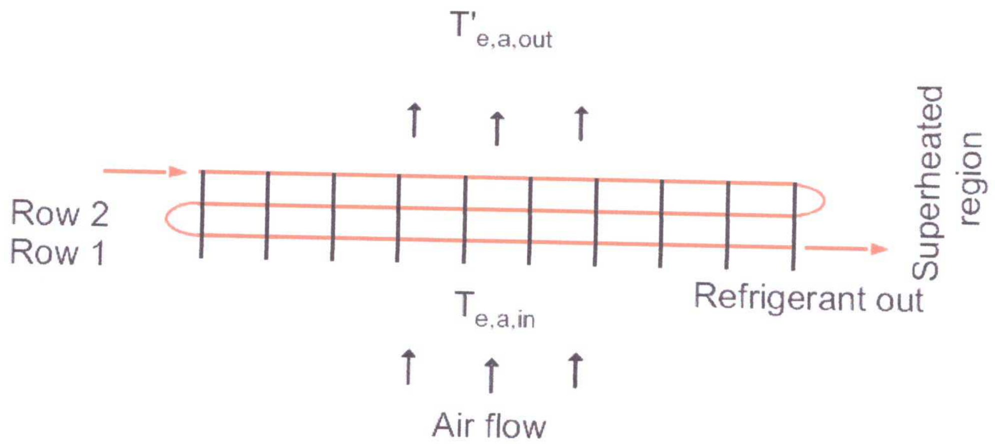


Figure 3-6: superheated region of the evaporator

$$\dot{Q}_{sp} = \dot{m}_{ref}(h_{5'} - h_6)$$

Where:

$Q^{\cdot}_{sp}$: Heat transfer rate for superheated region of the evaporator [W].

$m^{\cdot}_{ref}$: Refrigerant mass flow rate [kg/s] is found in previous chapter tow which is equal (0.0045 kg/s).

h_6: Specific enthalpy for refrigerant after evaporator [kJ/kg] is found in previous chapter tow which is equal (442 kJ/kg).

$h_{s'}$:Specific enthalpy for CO_2 refrigerant at saturation vapor point [kJ/kg] is found in previous chapter tow which is equal (433 kJ/kg).
So the heat transfer rate is equal:

$$\dot{Q}_{sp^-} = 0.0045(442 - 433) = 0.0405[KW].$$

The air side total surface area in superheated region of the evaporator is found by the following equation:

$$A_{sp} = (\frac{\dot{Q}_{sp}}{\dot{Q}_s})A_{total}$$

Where,

A_{sp}: overall surface area in superheated region [m^2]

$\dot{Q}_{sp}$: heat transfer rate in superheat region[kW]

Then

$$A_{sp} = \left(\frac{40.5}{780}\right) 7.41 = 0.364[m^2]$$

*Calculation of air temperature at outlet of the the superheated region is given by

By using Number of Transfer Unit (NTU) method ,the heat transfer rate of the superheated region of the evaporator is given by:

$$\dot{Q}_{sp} = (\dot{m}_r c_r)(T_{ro} - T'_{ro}) = (\dot{m}_a c_a)(T_{s,a,in} - T_{s,a,out})$$

Where ,

c_r: Refrigerant specific heat [kJ/kg.°C]

c_a: Air specific heat [kJ/kg.°C]

T'_{ro}: Refrigerant temperature at inlet tube of the superheated region of the evaporator [°C]which is equal (-15[°C]).

T_{ro}: Refrigerant temperature at outlet tube of the superheated region of the evaporator [°C]which is equal (-10[°C]).

So the refrigerant heat concentration($C_r[kW/°C]$) is equal:

$$C_r = \dot{m}_r c_r = \frac{Q_{sp}}{(T_{ro} - T'_{ro})} = \frac{40.5}{(-10 - -15)} = 8.1[W]$$

So the air heat concentration($C_a[kW/°C]$) is equal:

$$C_a = \dot{m}_a c_a = \frac{Q_{sp}}{(T_{a,in} - T_{a,out})} = \frac{40.5}{(-3 - -10)} = 5.8[W]$$

Therefore,

$$C_{min} = C_a$$

$$C_{max} = C_r$$

By using Logarithmic Mean Temperature Difference (LMTD),the overall heat transfer coefficient area of the superheated region of the evaporator is given by:

$$U_{sp} A_{SP} = \frac{Q_{sp}}{LMTD_{sp}}$$

Where ,

$$LMTD_{sp} = \frac{(T_{s,a,in} - T_{ro}) - (T_{s,a,out} - T'_{ro})}{\ln \frac{(T_{s,a,in} - T_{ro})}{(T_{s,a,out} - T'_{ro})}}$$

Then,

$$LMTD_{sp} = \frac{(-3 - -10) - (-10 - -15)}{\ln \frac{(-3 - -10)}{(-10 - -15)}} = 5.94[\ °C]$$

Then ,

$$U_{sp}A_{sp} = \frac{40.5}{5.94} = 6.82[W/^{o}\mathrm{C}]$$

The Number of Transfer Unit of the superheated region of the evaporator is given by

$$\mathrm{NTU}_{sp} = \frac{U_{sp}A_{sp}}{c_{min}}$$

Then,

$$NTU_{sp} = \frac{6.82}{5.8} = 1.179$$

The superheated region of the evaporator effectiveness is given by:

$$\varepsilon_{sp} = 1 - \exp\{(\frac{NTU_{sp}{}^{0.22}}{c})(\exp(-cNTU_{sp}{}^{0.78}) - 1\}$$

Where,

$$C = \frac{c_{min}}{c_{max}} = \frac{5.8}{8.1} = 0.714$$

Then ,

$$\varepsilon_{sp} = 1 - \exp\{\left(\frac{1.17^{0.22}}{0.714}\right)(\exp(-0.714 \times 1.17^{0.78}) - 1\} = 0.458$$

The air temperature at outlet of the superheated region is given by:

$$T''_{s,a,out} = T_{s,a,in} - \varepsilon_{sp} \frac{c_{min}}{c_{max}} \left(T_{s,a,in} - T_{r,avs}\right)$$

Where,

$T_{r,av}$: average temperature of the refrigerant [°C].

Which is calculated by :

$$T_{r,av} = \frac{T'_{ro} + T_{ro}}{2} = \frac{-15 - 10}{2} = -12.5[°C]$$

Then,

$$T''_{s,a,out} = -3\text{-}\left(0.458 \times 0.714(-3 - -12.5)\right) = -7.3557[°C]$$

The total heat transfer surface area of the unfin tubes of the superheated region of the evaporator is found by the following equation:

$$A_{b,sp} = (A_{total} - A_{fins})\left(\frac{\dot{Q}_{sp}}{\dot{Q}_s}\right)$$

Then,

$$A_{b,sp} = (2.847)\left(\frac{40.5}{780}\right) = 0.147[m^2]$$

The rate of heat gain due to the mass transfer, which is associated with the dehumidification of the air, is calculated based on a latent convective heat transfer coefficient of the air which flows over the superheated phase of the evaporator is proposed by Thrilled, Its general equation is:

$$(h_{lat})_{sp} = \frac{h_a (h_{sg})_{sp} (\omega_{in} - \omega'_{out})}{Le \times C_p (T_{a,in} - T'_{a,out})}$$

Where ,

$(h_{lat})_{sp}$: latent convective heat transfer coefficient of the air in superheated region[W/m^2.°C].

$(h_{sg})_{sp}$: sublimation latent heat in superheated region [J/kg].

ω_{in}: specific humidity of the air at inlet superheated region of the evaporator [kg moisture/kg dry air]

ω'_{out}: specific humidity of the air at outlet superheated region of the evaporator [kg moisture/kg dry air]

Le: Lewis number, is assumed to be 1.

By using Psychometrics Demo Software, at $T_{s,a,in}$ (dry bulb)= -3^{0}C and $T_{s,a,in}$ (wet bulb)= -7^{0}C , humidity ratio of the air is equals:

$\omega_{in} = 4.84 \times 10^{-4}$ kg moisture/kg dry air

By using Psychometrics Demo Software, at $T'_{a,o}$(dry bulb) $= -7.3557$[^{0}C]and $T'_{a,o}$(wet bulb)= -11[^{0}C] , humidity ratio of the air is equals:

$\omega'_{out} = 1.32 \times 10^{-5}$ [kg moisture/kg dry air].

the sublimation latent heat is calculated from the correlation that's reported by Ismail and Salinas and it is :

$$(h_{sg})_{sp} = 2322(-0.04667(1.8T_{r,av} + 32) + 1220.1)$$

Then,

$$(h_{sg})_{sp} = 2322(-0.04667(1.8 \times -12.5 + 32) + 1220.1) = 28332622 J/kg$$

Then,

$$(h_{lat})_{sp} = \frac{17 \times 28332622(4.84 \times 10^{-4} - 1.32 \times 10^{-5})}{1 \times 1.0064 \times 10^{3}(-3 - -7.3557)} = 51.73\ W/m^2.c$$

- overall thermal resistance of the superheated region of the evaporator ,figure (3-7) Illustrates that, is calculated as following[14]:

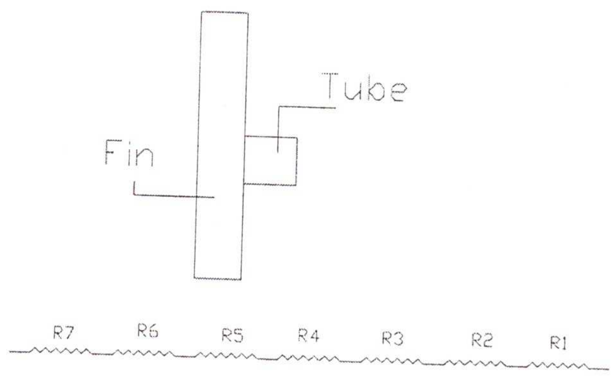


Figure (3.7): overall thermal resistance of the superheated region

$$R_{sp,o} = R_1 + R_2 + R_3 + R_4 + R_5 + R_6 + R_7$$

Where ,

R_1: External resistance of the superheated region of the evaporator [m^2.°C/W].

R_2: Frost resistance of the superheated region of the evaporator [m^2.°C/W].

R_3: Air side fouling resistance of the superheated region of the evaporator [m^2.°C/W].

R_4: Wall resistance of the superheated region of the evaporator [m^2.°C/W].

R_5: Fin tube constant resistance of the superheated region of the evaporator [m^2.°C/W].

R_6: Refrigerant side fouling resistance of the superheated region of the evaporator [m^2.°C/W].

R_7: Internal resistance of the superheated region of the evaporator [m^2.°C/W].

- External resistance

$$R_1 = \frac{1}{(h_a + (h_{lat})_{sp})\eta_o} = \frac{1}{(17 + 51.73)0.98} = 0.0165\ [m^2.°C/W]$$

- Frost resistance

$R_2 = 0$, no frost in superheated region.

- Air side fouling resistance

$$R_3 = \frac{1}{h_{f,a}\eta_o}$$

Where,

$h_{f,a}$: Air-side fouling conductance[$W/m^2.°C$] which is taken from Chemical Engineering Hand Book and it is equal(6000 [$W/m^2.°C$)].

Then,

$$R_3 = \frac{1}{6000 \times 0.98} = 1.894[\,m^2.°C/W].$$

- Wall resistance

$$R_4 = \frac{A_{sp}\,(\ln\frac{D_o}{D_i})}{2\pi K_w.L_{sp}}$$

Where,

K_w: thermal conductivity of aluminum tube wall[W/m.°C]

By using table (A.3) Properties of metal, the value of K_w= 202[W/m.°C]

L_{sp}: tube length of the superheated region of the evaporator [m]

Then,

$$R_4 = \frac{0.38(\ln\frac{3.2}{2.6})}{2\pi \times 202 \times L_{sp}} = \frac{6.28}{L_{sp}} [m^2.\ °C/W]$$

❖ Fin tube constant resistance

$$R_5 = \frac{1}{h_c(\frac{A_{b,sp}}{A_{sp}})}$$

Where,

h_c: fin-tube contact conductance [W /m^2 K] which is taken from Chemical Engineering Hand Book and it is equal(4×10^5 $[W/m^2.°C]$).

Then,

$$R_5 = \frac{1}{4 \times 10^5(\frac{0.1456}{0.384})} = 6.5 \times 10^{-4}\ [m^2.°C/W]$$

❖ Refrigerant side fouling resistance

$$R_6 = \frac{1}{h_{f,ref}\left(\frac{A_{sp,i}}{A_{sp}}\right)}$$

Where,

$h_{f,ref}$: refrigerant-side fouling conductance [W /m^2 K] which is taken from Chemical Engineering Hand Book. and it is equal(6000[$W/m^2.°C$]).

$A_{sp,i}$: internal tube surface area of superheated region [m^2].

Then,

$$R_6 = \frac{1}{5000\left(\frac{\pi \times 2.6 \times 10^{-3} \times L_{sp}}{0.384}\right)} = \frac{9.4 \times 10^{-3}}{L_{sp}} \quad [m^2.\,^{\circ}C/W]$$

❖ Internal resistance

$$R_7 = \frac{1}{h_{sp,r}\left(\frac{A_{sp,i}}{A_{sp}}\right)}$$

Where,

$h_{sp,r}$: Convective heat transfer coefficient for superheated phase refrigerant [W/m.°C]

Then,

$$R_7 = \frac{1}{1765\left(\frac{\pi \times 2.6 \times 10^{-3} \times L_{reg}}{0.384}\right)} = \frac{0.02662}{\mathrm{L}_{sp}} \quad [m.\,^{\circ}C/W].$$

By using Logarithmic Mean Temperature Difference (LMTD),the overall heat transfer coefficient of the superheated region of the evaporator is given by

$$U_{sp} = \frac{\dot{Q}_{sp}}{LMTD_{sp} \times A_{SP}}$$

Where,

$$LMTD_{sp} = \frac{(T_{a,in} - T_{ro}) - (T'_{a,out} - T'_{ro})}{\ln \frac{(T_{a,in} - T_{ro})}{(T'_{a,out} - T'_{ro})}}$$

Then,

$$LMTD_{sp} = \frac{(-3--10)-(-7.35--15)}{\ln\frac{(-3--10)}{(-7.35--15)}} = 7.317[^{\circ}\mathrm{C}]$$

Then,

$$U_{sp}A_{sp} = \frac{40.5}{7.317} = 5.53[W/^{\circ}\mathrm{C}]$$

Where,

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

Then,

$$U_{sp} = \frac{5.53}{0.384} = 14.4[W/m^2 c]$$

Where,

$$U_{sp} = \frac{1}{R_1 + R_2 + R_3 + R_4 + R_5 + R_6 + R_7}$$

Then,

$$107 = \frac{1}{0.0165 + 6.5\times10^{-4} + \frac{6.28\times10^{-5}}{L_{sp}} + 1.89\times10^{-4} + \frac{9.4\times10^{-3}}{L_{sp}} + \frac{0.0266}{L_{sp}}}$$

Then,

$$14.4 = \frac{1}{0.17 + \frac{0.3608}{L_{sp}}}$$

So , superheated region tube length of the evaporator is equal:

$$L_{sp} = \frac{0.5198}{0.749} = 0.7m$$

4.4.10 Calculation of length of the tow phase tubes of the evaporator

Employing energy balance, The heat transfer rate of the tow- phase region of the evaporator, fig.(3-8) Illustrates it, is given by:

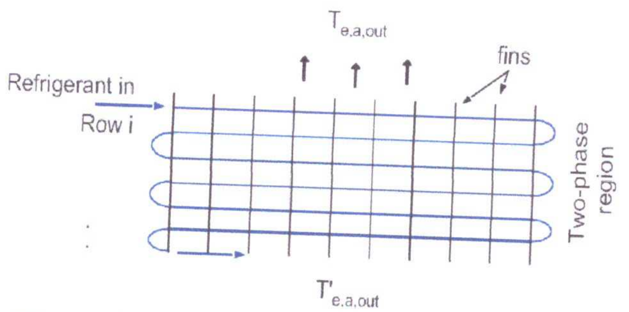


FIgure (3-8): two- phase region of the evaporator

$$\dot{Q}_{tp} = \dot{m}_{ref}(h_{s'} - h_s)$$

Where:

$\dot{Q}_{tp}$: Tow phase Heat transfer rate of the evaporator [W].

$\dot{m}_{ref}$: Refrigerant mass flow rate [kg/s] which is equal (0.0045 kg/s), is found in chapter two.

h_s: Specific enthalpy for refrigerant before evaporator [kJ/kg] which is equal (270 kJ/kg),is found in chapter two.

$h_{s'}$:Specific enthalpy for CO_2 refrigerant at saturation vapor point [kJ/kg] is found in previous chapter tow which is equal (433 kJ/kg).

So the tow phase heat transfer rate is equal:

$$\dot{Q}_{tp} = 0.0045(433 - 270) = 0.7KW$$

The air side total surface area in two phase region of the evaporator is found by the following equation:

$$A_{tp} = (\frac{\dot{Q}_{tp}}{\dot{Q}_s})A_{total}$$

Where,

A_{tp}: overall surface area in tow phase region [m^2]

$\dot{Q}_{tp}$: heat transfer rate in tow phase region[kW]

Then

$$A_{tp} = \left(\frac{700}{780}\right)7.4 = 6.641m^2$$

By using Number of Transfer Unit (NTU) method ,the heat transfer rate of the tow-phase region of the evaporator is given by:

$$\dot{Q}_{tp} = (\dot{m}_r c_r)(T'_{ro} - T_{ri}) = (\dot{m}_a c_a)(T'_{s,a,out} - T_{s,a,out})$$

Where ,

c_r: Refrigerant specific heat [kJ/kg.°C]

c_a: Air specific heat [kJ/kg.°C]

T'_{ro}: Refrigerant temperature at outlet tube of two phase region of the evaporator [°C]which is equal (-15 °C).

T_{ri}: Refrigerant temperature at inlet tube of two phase region of the evaporator [°C]which is equal (-15 °C).

$T'_{s,a,out}$:air temperature at inlet of two phase region of the evaporator [°C]which is equal (-7.33 °C).

$T_{s,a,out}$: air temperature at outlet of two phase region of the evaporator [°C]which is equal (-10 °C).

By using Psychometrics Demo Software at $T_{a,o}$(dry bulb)= -10^{0}C and $T_{a,o}$(wet bulb)=-13^{0}C , humidity ratio of the air is equal:

$\omega_{out} = 2.98 \times 10^{-6}$ kg moisture/kg dry air

By using Psychometrics Demo Software at $T'_{a,o}$(dry bulb) = -7.33^{0}C and $T'_{a,o}$(wet bulb)= -11^{0}C , humidity ratio of the air is equal:

$\omega'_{out} = 2.93 \times 10^{-5}$ kg moisture/kg dry air.

So the air heat concentration($C_a\,[kW/°C]$) is equal:

$$C_a = \dot{m}_a c_a = \frac{Q_{tp}}{(T'_{s,a,out} - T_{s,a,out})} = \frac{0.7}{(-7.33 - -10)} = 0.262 kW/°C$$

Then

$$C_{min} = C_a$$

By using Logarithmic Mean Temperature Difference (LMTD),the overall heat transfer coefficient of the tow- phase region of the evaporator is given by:

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

Where ,

$$LMTD_{tp} = \frac{\left(T'_{s,a,out} - T_{ri}\right) - \left(T_{s,a,out} - T'_{ro}\right)}{\ln \frac{\left(T'_{s,a,out} - T_{ri}\right)}{\left(T_{s,a,out} - T'_{ro}\right)}}$$

Then,

$$LMTD_{tp} = \frac{(-7.33 - -15) - (-10 - -15)}{\ln\frac{(-7.33 - -15)}{(-10 - -15)}} = 6.24\ ^{o}\mathrm{C}$$

Then ,

$$U_{tp}A_{tp} = \frac{0.7}{6.24} = 0.1122\ kW/^{o}\mathrm{C}$$

So,

$$U_{tp} = \frac{0.7}{6.24 \times 6.641} = 16.89\ W/m^2.^{o}\mathrm{C}$$

The Number of Transfer Unit of the tow- phase region of the evaporator is given by

$$\mathrm{NTU}_{tp} = \frac{U_{tp}A_{tp}}{c_{min}}$$

Then,

$$NTU_{tp} = \frac{0.1122}{0.262} = 0.428$$

The tow-phase heat exchanger effectiveness is given by:

$$\varepsilon_{tp} = 1 - \exp(-NTU_{tp}) = 1 - \exp(-0.428) = 0.35$$

The total heat transfer surface area of the bare tubes of the tow- phase region of the evaporator is found by the following equation:

$$A_{b,tp} = (A_{total} - A_{fins})\left(\frac{\dot{Q}_{tp}}{\dot{Q}_{e}}\right)$$

Then,

$$A_{b.tp} = (2.847)\left(\frac{700}{780}\right) = 2.55m^2$$

The rate of heat gain due to the mass transfer, which is associated with the dehumidification of the air, is calculated based on a latent convective heat transfer coefficient of the air which flows over the tow- phase of the evaporator proposed by Thrilled, Its general equation is:

$$(h_{lat})_{tp} = \frac{h_a (h_{sg})_{tp} (\omega'_{out} - \omega_{out})}{Le \times C_p (T'_{s,a,out} - T_{s,a,out})}$$

Where ,

$(h_{lat})_{tp}$: latent convective heat transfer coefficient of the air in tow phase region[W/m^2.°C].

$(h_{sg})_{tp}$: sublimation latent heat in tow phase region [J/kg].

ω'_{out}: specific humidity of the air at inlet of two phase region of the evaporator [kg moisture/kg dry air]

ω_{out}: specific humidity of the air at outlet of two phase region of the evaporator [kg moisture/kg dry air]

Le: Lewis number, is assumed to be 1.

Where the sublimation latent heat is calculated from the correlation reported by Ismail and Salinas which is:

$$(h_{sg})_{tp} = 2322(-0.04667(1.8T_{r,av} + 32) + 1220.1)$$

Where,

$T_{r,av}$: average temperature of the refrigerant [°C].

Which is calculated by :

$$T_{r,av} = \frac{T'_{ro} + T_{ri}}{2} = \frac{-15 - 15}{2} = -15°C$$

Then,

$$(h_{sg})_{tp} = 2322(-0.04667(1.8 \times (-15) + 32) + 1220.1) = 28332622 J/kg$$

And,

$$(h_{lat})_{tp} = \frac{17 \times 28332622(2.98 \times 10^{-5} - 2.93 \times 10^{-6})}{1 \times 1.0064 \times 10^{3}(-7.355 - -10)} = 55.61\ W/m^{2}.c$$

❖ overall thermal resistance of the two –phase region of the evaporator , fig.(3-9) Illustrates that, is calculated as following[14]:

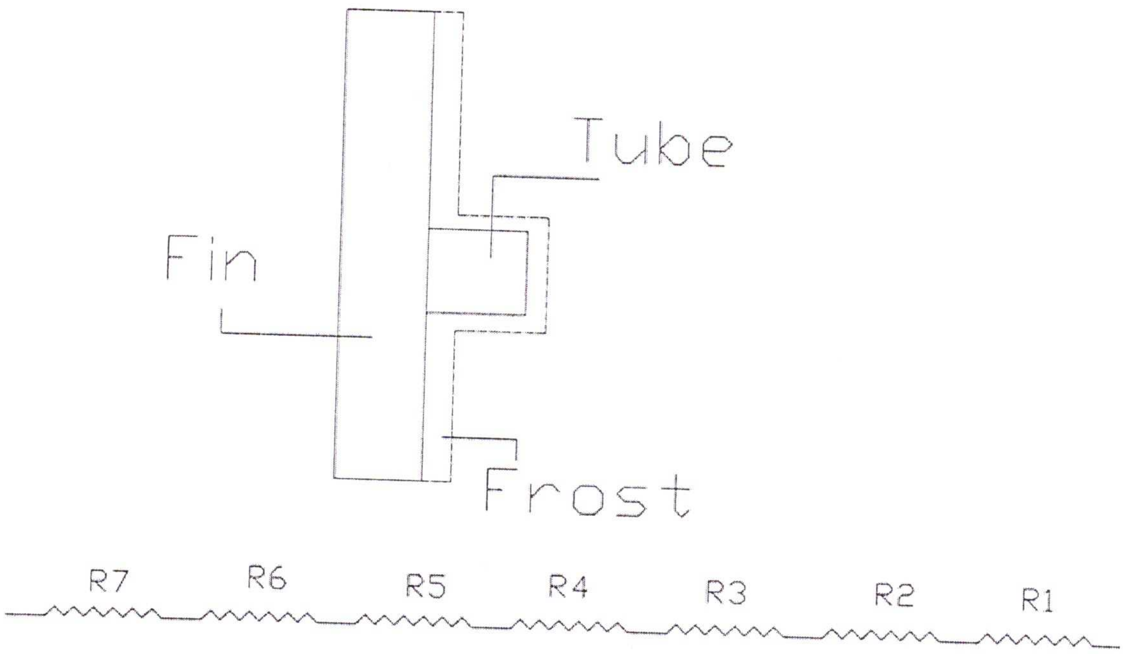


Figure (3.9): overall thermal resistance of the two –phase region

$$R_{tp,o} = R_1 + R_2 + R_3 + R_4 + R_5 + R_6 + R_7$$

Where ,

R_1: External resistance of the two phase region of the evaporator [m^2.°C/W].

R_2: Frost resistance of the two phase region of the evaporator [m^2.°C/W].

R_3: Air side fouling resistance of the two phase region of the evaporator [m^2.°C/W].

R_4: Wall resistance of the two phase region of the evaporator [m^2.°C/W].

R_5: Fin tube constant resistance of the two phase region of the evaporator [m^2.°C/W].

R_6: Refrigerant side fouling resistance of the two phase region of the evaporator [m^2.°C/W].

R_7: Internal resistance of the two phase region of the evaporator [m^2.°C/W].

- External resistance

$$R_1 = \frac{1}{(h_a + (h_{lat})_{tp})\eta_o} = \frac{1}{(17 + 55.61)0.88} = 0.01565\ m^2.°C/W$$

- Frost resistance

$$R_2 = \frac{\Delta t_{fst}}{K_{fst}\eta_o}$$

Where,

K_{fst}: Thermal conductivity of the frost [W/m.°C].

t_{fst}: Frost thickness[m]

The rate of frost accumulation is calculated as:

$$\dot{m}_{fst} = \dot{m}_a(\omega'_{out} - \omega_{out})$$

Where,

$\dot{m}_{fst}$: frost mass accumulation [kg /s]

Then,

$$\dot{m}_{fst} = 0.196(2.98 \times 10^{-5} - 2.93 \times 10^{-6}) = 3.3 \times 10^{-6}\ kg/s$$

So, the change in frost mass accumulation is calculated as:

$$\Delta m_{fst} = \frac{\dot{m}_{fst}}{A_{tp}}$$

Where,

Δm_{fst}:change in frost mass accumulation [kg /m^2 .s]

Then,

$$\Delta m_{fst} = \frac{3.3 \times 10^{-6}}{8.327} = 4.98 \times 10^{-7}\ kg/m^2 .s$$

The frost density is calculated as:

$$\rho_{fst} = 650e^{0.277T_{so}}$$

If T_{so} : -3^{0}C.

Then,

$$\rho_{fst} = 650e^{0.277\times-3} = \frac{283.148kg}{m^3}$$

The change in frost thickness is calculated as:

$$\Delta t_{fst} = \frac{\Delta m_{fst}}{\rho_{fst}} = \frac{4.98 \times 10^{-7}}{283.148} = 1.75 \times 10^{-9}\ kg/m^2.s$$

Then, the frost thickness which is accumulated on the two –phase region of the evaporator after two hours equals:

$$t_{fst\ after\ 2hours} = 1.75 \times 10^{-9} \times 2 \times 3600 = 1.26 \times 10^{-5} m$$

So ,adjusting the control for dealing with the melting of the frost thickness by operating the heater .

❖ Thermal conductivity of the frost is calculated as:

$$K_{fr} = \frac{(0.02422 + 7.214 \times 10^{-4}\rho_{fst} + 1.1797 \times 10^{-6}\rho_{fst}^{\ 2})}{1000}$$

Then,

$$K_{fr} = \frac{(0.02422+7.214\times10^{-4}\times283.148+1.1797\times10^{-6}\times283.148^2)}{1000} = 3.23 \times 10^{-4} W/m.^0c$$

Then,

$$R_2 = \frac{1.75 \times 10^{-9}}{3.23 \times 10^{-4} \times 0.88} = 5.28 \times 10^{-5}\ m^2\,{}^oC/W$$

❖ Air side fouling resistance

$$R_3 = \frac{1}{h_{f,a}\eta_o}$$

Where,

$h_{f,a}$: Air-side fouling conductance[$W/m^2.°C$] which is taken from Chemical Engineering Hand Book and it is equal(6000 $W/m^2.°C$).

Then,

$$R_3 = \frac{1}{6000 \times 0.88} = 1.89 \times 10^{-4}\ m^2.°C/W$$

❖ Wall resistance

$$R_4 = \frac{A_{tp}(\ln\frac{D_o}{D_i})}{2\pi K_w L_{tp}}$$

Where,

K_w: thermal conductivity of aluminum tube wall[W/m.°C] which is taken from Chemical Engineering Hand Book and it is equal(202 W/m.°C)

L_{tp}: tube length of the two phase region of the evaporator[m]

Then,

$$R_4 = \frac{6.641(\ln\frac{3.2}{2.6})}{2\pi \times 202 \times L_{tp}} = \frac{3.4 \times 10^{-3}}{L_{tp}} m^2.°C/W$$

- Fin tube constant resistance

$$R_5 = \frac{1}{h_c\left(\frac{A_{b,tp}}{A_{tp}}\right)}$$

Where,

h_c: fin-tube contact conductance [W /m^2 K] which is taken from Chemical Engineering Hand Book and it is equal($4 \times 10^5\ W/m^2.°C$).

Then,

$$R_5 = \frac{1}{4 \times 10^5\left(\frac{2.55}{6.641}\right)} = 6.49 \times 10^{-6}\ m^2.°C/W$$

- Refrigerant side fouling resistance

$$R_6 = \frac{1}{h_{f,ref}\left(\frac{A_{tp,i}}{A_{tp}}\right)}$$

Where,

$h_{f,ref}$: refrigerant-side fouling conductance [W /m^2 K] which is taken from Chemical Engineering Hand Book and it is equal($6000\ W/m^2.°C$).

$A_{tp,i}$: internal tube surface area of two phase region[m^2].

Then,

$$R_6 = \frac{1}{5000\left(\frac{\pi \times 0.0078 \times L_{tp}}{8.327}\right)} = \frac{0.0679}{L_{tp}} = \frac{0.1626}{L_{tp}}$$

❖ Internal resistance

$$R_7 = \frac{1}{h_{tp,r}\left(\frac{A_{tp,i}}{A_{tp}}\right)}$$

Where,

$h_{tp,r}$: Convective heat transfer coefficient for two phase refrigerant [W/m.°C]

Then,

$$R_7 = \frac{1}{5692\left(\frac{\pi \times 0.0078 \times L_{tp}}{6.641}\right)} = \frac{0.1428}{L_{tp}}\ m.°C/W$$

Then,

$$R_{tb,o} = R_1 + R_2 + R_3 + R_4 + R_5 + R_6 + R_7 = 0.0159 + \frac{0.3088}{L_{tp}}$$

Where,

$$U_{tp,o} = \frac{1}{R_{tb,o}}$$

Then,

$$16.89 = \frac{1}{0.0159 + \frac{0.3088}{L_{tp}}}$$

So length of the tube of two phase region of the evaporator is equal:

$$L_{tp} = \frac{5.21}{0.732} = 7.122m$$

So , total tube length of the evaporator is equal:

$$L_{total} = L_{sp} + L_{tp} = 0.7 + 7.122 = 7.82 \text{ m}$$

Evaporator details

(figure 3-10) and (figure 3-11) show the evaporator that is designed, and it's details is summarized in table(3-1).

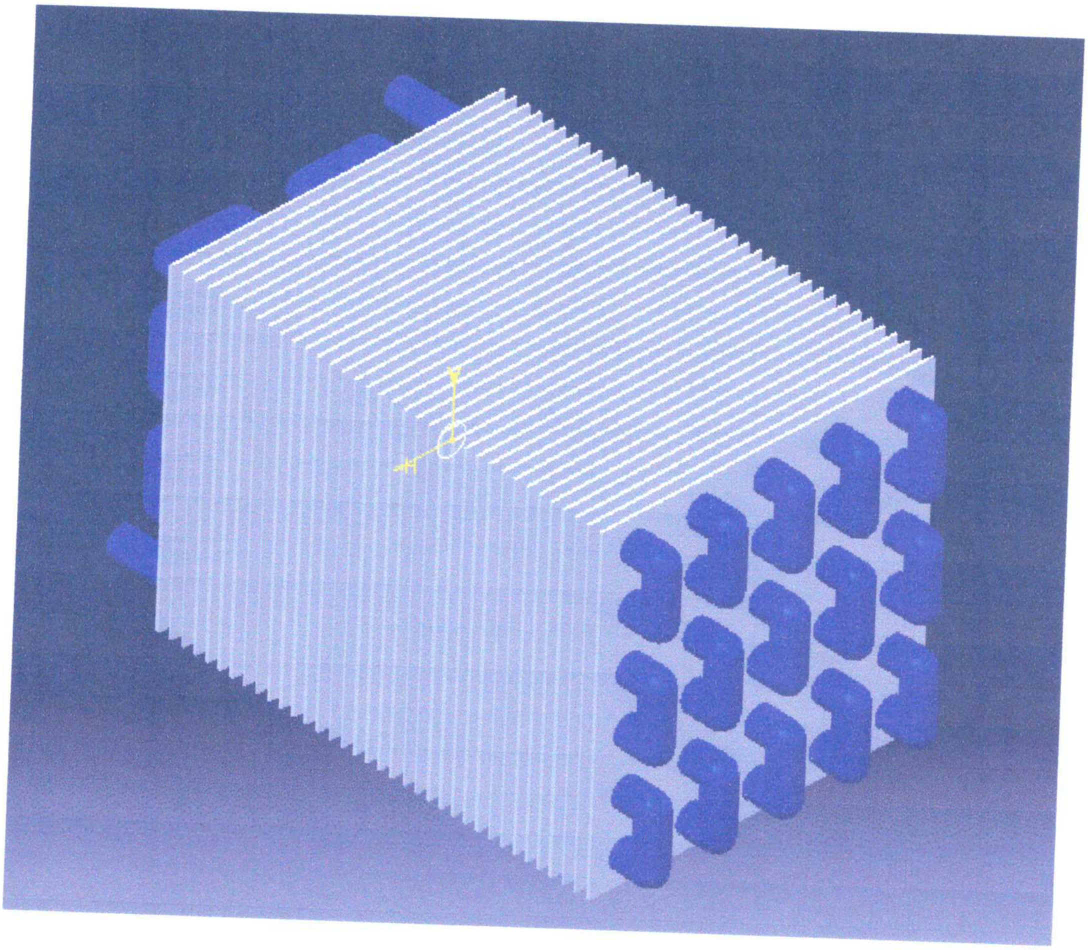


Figure (3-10): the evaporator that is designed

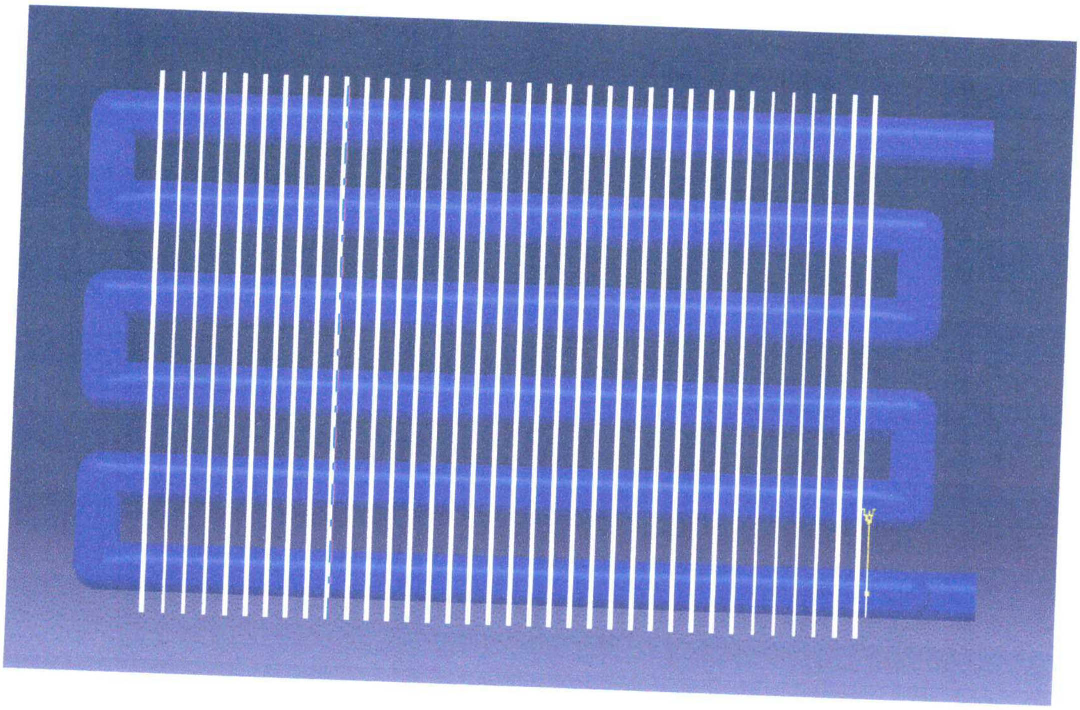


Figure (3-11): side view of the evaporator that is designed

Table (3.1): Details of evaporator that is designed

Geometrical parameter	
Fin material	Aluminum
Tube material	Aluminum
Fin thickness	0.2mm
Fin pitch	4mm
Tube I.D	2.6mm
Tube O.D	3.2mm
No .of fins	64
Coil height	195mm
Coil length	250mm
Coil depth	190mm
Tube length	7822mm

3.4.11 Air side pressure drop Calculation of

The friction factor contributions due to the fins and tube bank of the evaporator are used following Kimet al [14].

$$f_f = 1.435 Re_d^{-0.562} \left(\frac{ST}{Sw}\right)^{-0.365} + \left(\frac{SF}{D_0}\right)^{-0.131} \left(\frac{ST}{D_0}\right)^{1.22}$$

$$f_t = \frac{4}{\pi}\left(0.25 + \frac{0.118}{\left(\left({}^{ST}/_{D_0}\right)-1\right)} Re_D^{-0.16}\left(\frac{ST}{D_0} - 1\right)\right)$$

Then,

$$f_f = 1.435 \times 160239^{-0.562} \left(\frac{0.039}{0.03166}\right)^{-0.365} + \left(\frac{0.0078}{0.0032}\right)^{-0.131} \left(\frac{0.039}{0.0032}\right)^{1.22} = 4.73$$

$$f_t = \frac{4}{\pi}\left(0.25 + \frac{0.118}{\left(\left({}^{0.039}/_{0.0032}\right) - 1\right)} (196239)^{-0.16} \left(\frac{0.039}{0.0032} - 1\right)\right) = 0.32$$

The total friction factor (the combination of tube bank and fins friction factors) for the air-side pressure drop is calculated by using the following equation[14]:

$$f = f_f \left(\frac{A_f}{A_{total}}\right) + f_t\left(1 - \left(\frac{A_f}{A_{total}}\right)\left(1 - \frac{t_f}{s_F}\right)\right)$$

then,

$$f = 4.73\left(\frac{4.55}{7.4}\right) + 0.32\left(1 - \left(\frac{4.55}{7.4}\right)\left(1 - \frac{0.0002}{0.0038}\right)\right) = 3.367$$

Chapter Four

Cycle components

To construct carbon dioxide refrigeration cycle, select their proper components by basing on cycle calculations in chapter tow and operating them within transcritical condition, pressures and temperatures ranges, the selecting components are:

4.1 Compressor

The compressor raises the pressure of the refrigerant vapor so that the refrigerant saturation temperature is slightly above the temperature of the cooling medium used in the gas cooler, which has several functions as follows:

- Oil free lubrication.
- Efficient de-oiling filter.
- Recovery plants for breweries and soft drink industry.
- Helps prevent carbonation of the beer.
- Largely used in urea plant.

The selecting compressor is:

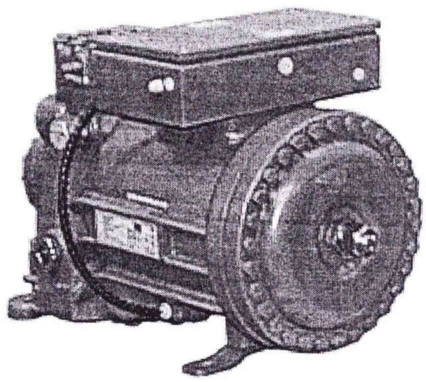


Figure 4.1: HGX2 - CO_2 Compressor

4.1.1 Description:

Complete semihermetic, electrical driven compressor-series for transcritical R-744 applications like industrial and commercial refrigeration.

4.1.2 Basic specifications:

- Type of Construction: semihermetic piston compressor with fixed displacement
- Actuator: integrated gas cooled electric motor 4- pole version (1.500 rpm)
- Swept volume: 90 ccm - 190 ccm(cubic centimeter)
- 4- Displacement volume: 7,70 m^3/h - 20,00 m^3/h
- Number of cylinders: 2
- Maximum speed of rotation: 1800 rpm (revolution per minute)

- Compressor Dimensions (L/W/H): 650 x 300 x 400 mm .
- Overal Weight: 160 - 170 kg.
- Oil: Bock C55E (Synthetic Oil) .
- HP and LP savety valves .
- Motor winding protection with PTC sensors.

4.1.3 Advantages:

- Simple and reliable mechanism, designed for transcritical R-744 applications.
- Housing made of spheroidal graphite Iron for maximum pressure resistance.
- Reliable oil management by oil-pump lubrication.

4.2 Evaporator

The evaporator absorbs heat from the surrounding air or liquid and moves it outside the refrigerated area by means of a refrigerant. It is also known as a cooling coil, blower coil, chilling unit or indoor coil.

The selecting evaporator is:

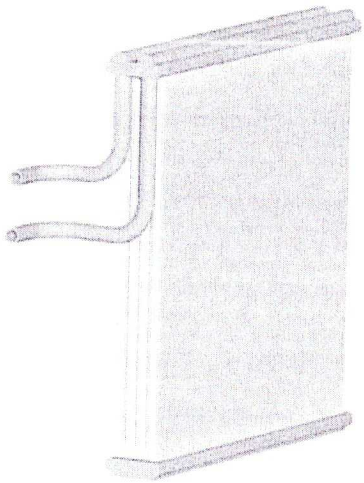


Figure: 4.2 evaporator

4.2.1 Description:

The R-744 micro channel evaporator which an effective and compact finless heat exchanger with dramatically reduced water storage capacity for use in commercial refrigeration installations.

4.2.2 Basic specifications:

- Core size: on request.
- Block size: on request.
- Block depth restriction: 38-50mm.
- Overall weight: depending on size.
- Surface Area (Air side / Refrigerant side): depending on size..

- Material: Aluminum.
- Coating Material: Hydrophilic.

4.2.3 Advantages:

- Compact design.
- Minimum depth for optimal packaging.
- Minimum water storage; prevents moist and odour problems.
- Width and height can be system optimized.
- Ideal for reversible cooling systems.

4.3 Internal Heat Exchanger :

The heat exchanger used to transfer the heat from an area of higher temperature to one of lower temperature. Heat transfer changes the internal energy of the system and surroundings. Heat transfer changes the internal energy of the system. Heat is transferred by conduction, convection and radiation.

The selecting Heat Exchanger is:

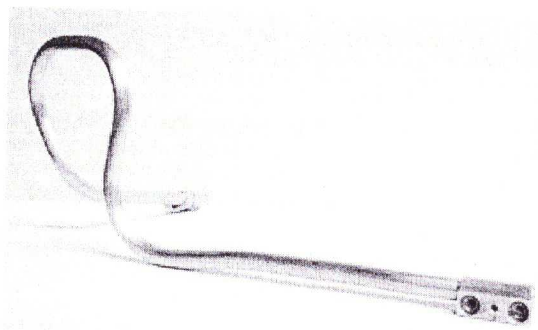


Figure4.3: Coaxial R-744 Internal Heat Exchange

4.3.1 Description

The coaxial R-744 internal heat exchanger (IHX) for use in commercial refrigeration installations is an implemented part of the line set with low pressure drop design for optimized system efficiency.

4.3.2 Basic specifications:

- Total length: depending on requested temperature efficiency.
- Total weight: depending on length.
- Outer diameter: 18mm.
- Shape/Design: custom design (bending radius restrictions).
- Profile: standard profile available (optimized profile on request).
- Material: AIMg2Mn0,8.
- Fitting technology: VisCO_2nnect double fitting blocks on both sides (others on request).

4.3.3 Advantages:

- Low pressure drop design.
- System optimized length.
- Flexible bending shape.

4.4 Gas Cooler

The gas cooler's purpose is to remove the heat of compression from the process gas which has just been discharged from a compressor.

The selecting gas cooler is:

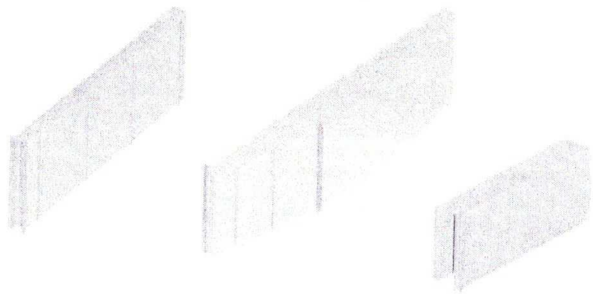


Figure 4.4:R744 Gas cooler Capillary Design

4.4.1 Description:

Cross counter flow gas cooler in finless capillary design with high design and application flexibility.

4.4.2 Basic specifications:

- Block Size: on request.
- Surface area: on request.
- Overall Weight: depending on size.
- Fin Density: no fins.
- Tube Profile: capillaries.

- Material: copper alloy .
- Fittings: VisCO_2nnect single fittings are standard, others on request.

4.4.2 Advantages:

- Finless design.
- Flexible block design.
- Few brazing points.
- Flexible design in all directions (LxWxH).

4.5 Expansion valve

The expansion valve liquid refrigerant expands from higher pressure to a lower pressure in it.

The selecting expansion valve is:

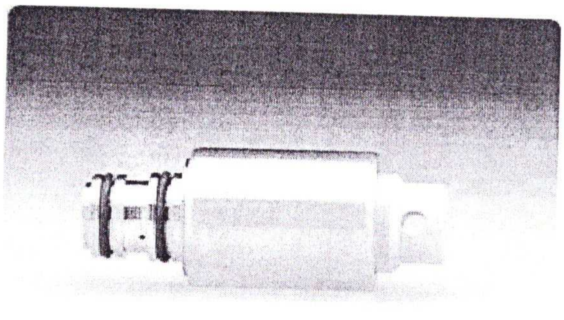


Figure 4.5: R744 Valve

4.5.1 Description

Highly efficient valves for R744. The modular concept (valve family) offers high potential for reducing system costs by using common parts.

4.5.2 Basic specifications

- 2/2-Way Proportional Valve, current controlled: PWM 400 Hz.
- Nominal voltage: 12V.
- Weight: ca. 250 g.
- Length: < 45 mm (excluding connector).
- Diameter: ca. 25 mm.
- Material: non-corrosive.
- Operational pressure: up to 15 MPa.
- Temperature range: -40°C to 180°C.

4.5.3 Advantages:

- Shows excellent performance in R744-systems.
- Increases the comfort attributes.
- noiseless, exact controllability with high precision.

4.6 Refrigeration pipes:

The pipes used for transfer the refrigerant which exists at liquid,vapour,and mixture phases in the cycle.

The selecting refrigeration pipe is:

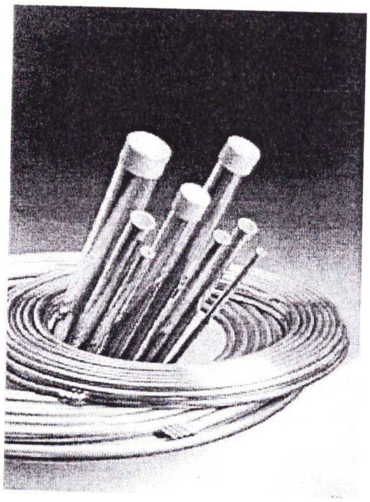


Figure 4.6: Copper alloy tubes for Commercial Refrigeration

4.4.3 Description:

Plain tubes and innergrooved tubes for an increased tubeside heat transfer in heat exchangers such as CO_2 gas coolers and internal heat exchangers and connection lines.

4.6.2 Basic specifications:

- Material: special high-tensile copper alloy.
- Dimensions: customized, according to the needs of the application.
- High surface cleanliness for Refrigeration applications.
- Available in straight length and in level-wound coils.

4.6.3 Advantages:

- High tensile strength and yield strength even after brazing.
- Significant reduction of wall thickness and material costs.
- Processible on existing machinery: wieldable, brazeable, bendable and expandable.
- Excellent corrosion resistance.

4.7 Sensors

Sensors used to sense and measure pressure and temperature, which indicates the operation behavior for each components in the cycle.

The selecting sensor is:

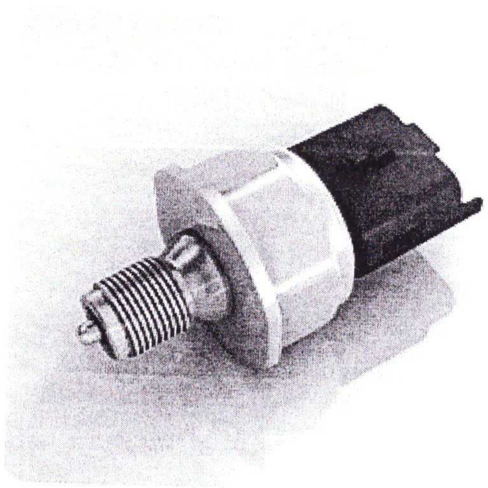


Figure 4.7: Piezo Pressure & Temperature sensors (PPT-series)

4.7.1 Description

Accurately measures both pressure and temperature in R744 / CO_2 refrigeration systems.

4.7.2 Basic specifications

- Pressure range: 0 - 176bar.
- Fluid temperature measurement range: -40°C - 180°C.
- Fast pressure and temperature response time.
- High pressure and temperature accuracy.

4.7.3 Advantages:

- Hermetic sensor; minimizes risks for leakages.
- Provides accurate measurement of both pressure and temperature.

- Protects the compressor against high temperatures.
- Pressure and Temperature data can be used for signaling leakages.

4.8 Seals:

The selecting seal is:

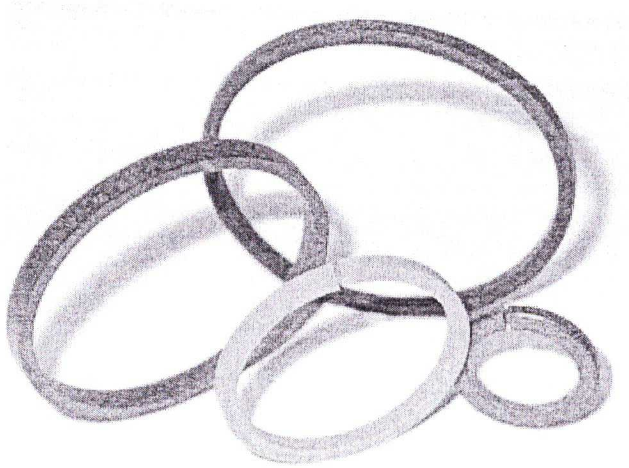


Figure 4.8: PTFE Piston Rings R744

4.8.1 Description:

High-resistant PTFE piston rings for the sealing of compressor pistons under high-dynamic translator load in refrigeration applications.

4 .8.2 Basic specifications:

Materials:

- Compound PTFE/F18237: for hard counterpart surfaces (steel).
- Compound PTFE/F52902: for soft counterpart surfaces.
- Compound PTFE/F18225: optimized against wear for soft counterpart surfaces.

Design:

- Customized design for each application.

4.8.2 Advantages:

- High resistance against explosive decompression.
- Very low friction.
- Minimal swelling.
- No aging.
- Very high medium resistance.

4.9 Charge & Service Valves

The charge and service valves using for charging the refrigerant in the cycle and get out it in the case excess it .

The selecting Charge & Service Valve is:

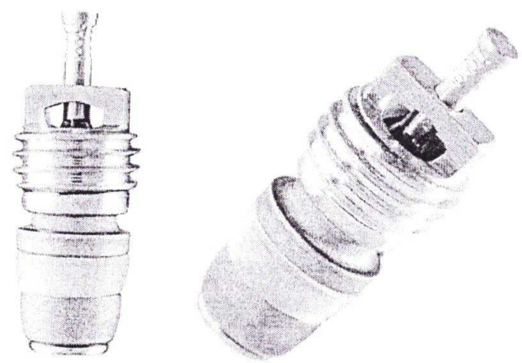


Figure 4.9: Charge & Service Valve for CO2 refrigeration systems.

4.9.1 Description:

The charge and service valve consists of a valve body (housing), a valve core, and a cap. It is suitable for the high- and low-pressure side of the system.

4.9.2 Basic specifications:

- Type of construction: Push pin type valve core.
- Same valve core for HP and LP side.
- Dimensions: M8x1 / length: 20mm.

- Weight: 3g.
- Pin travel: 3mm.
- Operating pressure: 0 - 14 MPa.
- Maximum pressure: 17 Mpa.
- Burst pressure: >34 Mpa.
- Braze-in valve bodies in aluminium, stainless steel or brass, shape according to SAE J639.
- Non sealing protection cap.
- Position in a/c loop: downstream of the gas cooler outlet according to SAE J639.

4.9.3 Advantages:

- Push pin concept for quick access to a/c system.
- Robust design with internal spring and short, strong pin.
- Permanent fixed micro sealing lip, vulcanized on brass part.

Chapter Five

Electrical Design

5.1 Introduction

This chapter tests the basic equipment for cooling cycles as well as organizations and accessories, and enables the control over the operation and the characteristics of CO2 refrigeration cycle. In order to obtain the highest efficiency and the safe operating conditions for the CO2 refrigeration cycle and people, and to achieve the necessary control requirements.

The requirements for controlling logical sequence to operate the refrigeration cycle , to reach the required cooling and to complete the controlling of operation processes in adding an electrical components on the refrigeration cycle.

5.2 Components Of Electrical Circuits

5.2.1 Current relay

This can be described as a magnetic switch. It comprises a small solenoid coil around a sleeve and an iron core. Inside the sleeve is a plunger to which the switch contact bridge is attached. The contacts are normally open. When the coil is energized, a strong magnetic field of force is created because the current will be high during the starting phase. The magnetic force will move the plunger upward and bridge the switch contacts, completing the circuit to the start winding. The run winding is wired through the relay so that it is always in circuit. A high starting current is drawn when the compressor motor starts. The current reduces as the motor gathers speed, the magnetic filed through the relay then becomes weaker so that it can no longer hold the contact bridge on to the switch. The plunger then drops down by gravity to open the circuit to the start winding.

It is not uncommon for a start capacitor to be fitted when a current ready is employed. This is wired in series with the start winding. (Figure 5.1) shows Current relay

Figure (5. 1): Current relay

5.2.2 Overload

The most common cause of motor failure is overheating. The condition is created when a motor exceeds it is normal operating current flow. The result can be either a breakdown of the motor winding insulation and a short circuit or a winding burn-out. For this reason overload protection is provided in the form of a current and temperature sensitive control which will open the circuit before any damage can occur. (Figure 5.2) shows overload.

Figure (5.2): overload

5.2.3 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.

Figure (5.3) : Thermostat

5.2.4 Contactors

A contactor is an electrical switch that opens and closes under the control of another electrical circuit. In the original form, the switch is operated by an electromagnet to open or close one or many sets of contacts.

When a current flows through the coil, the resulting magnetic field attracts an armature that is mechanically linked to a moving contact. The movement either makes or breaks a connection with a fixed contact. When the current to the coil is switched off, the armature is returned to a normal position

Contactors are used to control electric motors, lighting, heating, capacitor banks, and other electrical loads. (Figure 5. 4) shows contactor.

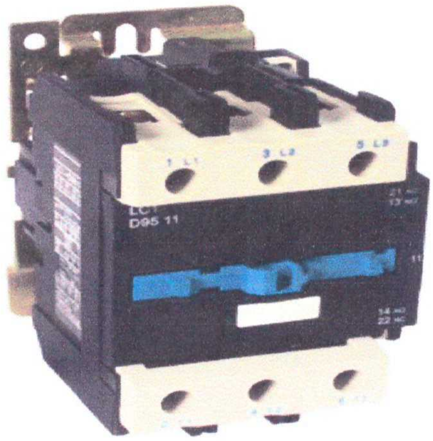


Figure (5.4): Contactor

5.3 Control And Power Circuits

As shown in the following (figure 5-5) the control and the power circuit where current relay is used to separate and connect starting coil.

When the temperature rises inside the refrigerator, the contact point of the thermostat is closed then the current passes from L1 to the emergency keys (overload OL, , low pressure LP) after that it passes through the current relay (CR) , then to the run coil (R), then to the common (C) and then to neutral (N).when the compressor withdraws high current (starting current), the magnetic field rises in the relay coil and attracts magnetic mold and that is to reach current to the starting coil, and to operate the starting coil with the run coil in parallel and then the motor of compressor is rotated.

The speed of rotation of the compressor's motor increases gradually, Thereby the consuming current reduces in relay coil then starting coil gets cut and the motor remains rotate by run coil effect. With operating the motor ,the compressor provides the refrigerant to cool the refrigerated space.

When the temperature reaches required thermostat temperature, the current gets separated from the compressor and it is stopped, then the temperature is raised again to operate the compressor again.

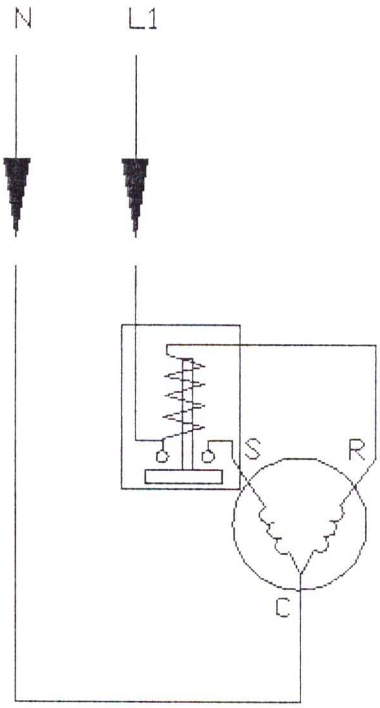


Figure (5.5): Current relay and motor circuit

5.3.1 Components of Control

1. Thermostate with range less than -20°C .

2. Current relay and overload.

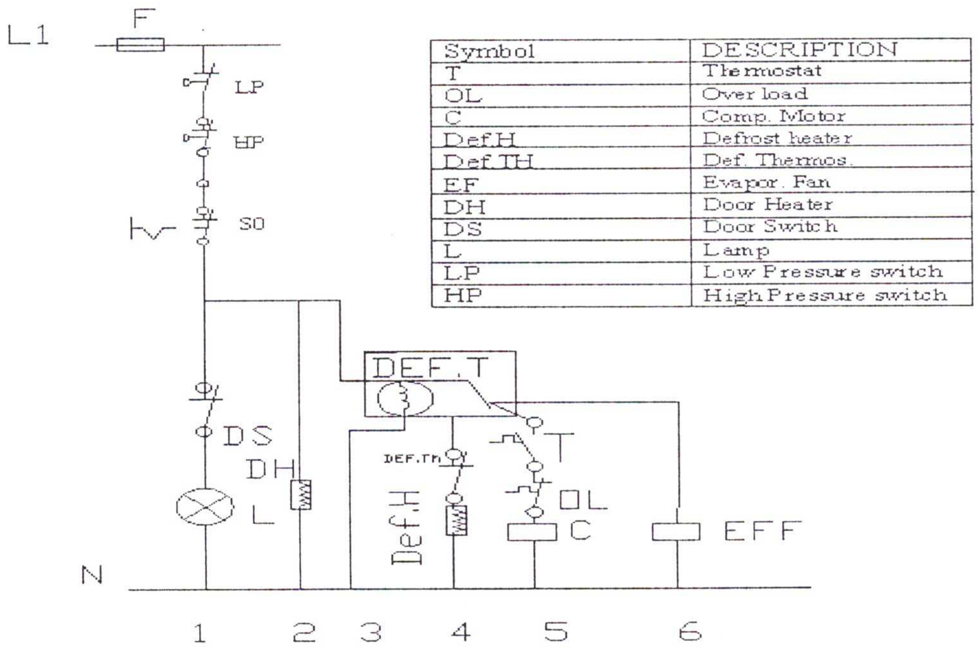
3. Electrical heater to melt the frost.

4. Timer to control the periods of freezing and melting frost.

5. Door heater.

6. Door lamp.

The following (figure 5.6) shows the control circuit:



Symbol	DESCRIPTION
T	Thermostat
OL	Over load
C	Comp. Motor
Def.H	Defrost heater
Def.TH	Def. Thermos.
EF	Evapor. Fan
DH	Door Heater
DS	Door Switch
L	Lamp
LP	Low Pressure switch
HP	High Pressure switch

Figure (5.6): Control circuit

5.3.2 Operation stages

1. Lamp lights when the door opens.

2. Heater in the front edge of the refrigerator to prevent the door from sticking.

3. Timer.

4. Defrost heater and defrost thermostat.

5. Compressor motor, its operation is affected with timer ,thermostat and overload.

6. Evaporator fan works during cycle operation .

5.3.3 CO_2 Refrigeration cycle performance

During refrigeration period, the following elements are being in operation state:

1. Doors heater.

2. Contactor of timer.

3. Compressor.

4. Evaporator's fan.

The timer controls the operation period of the refrigeration cycle for awhile (two hours) ,thermostat works during cycle operation in controlling of operating and stopping the compressor according to the change in the refrigerated space temperature while the evaporator fan keeping works during refrigeration period .

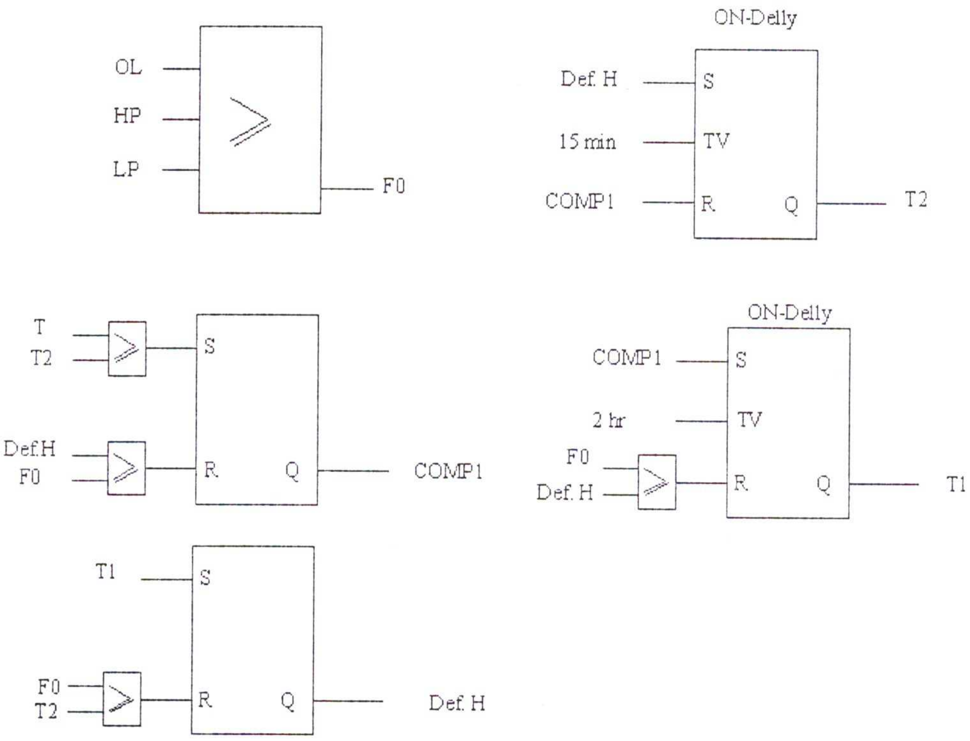
5.3.4 Defrost period

After finishing the refrigeration period, the timer disconnect evaporator fan and compressor motor while it connecting defrost heater with circuit (4) and if the frost has dissolved before finishing the limited time for this operation, the temperature of the evaporator surface will rise to -2°C ,therefore the defrost thermostat disconnect defrost heater and stay so until the finishing of defrost time. Then the timer disconnect heater circuit and connect refrigeration cycle ,then evaporator fan works directly with compressor motor which is affected by the timer.

Water is produced by the defrost which is collected in the basin and then it drains by through a pipe .

5.4 Cycle Control By Using PLC

The following block diagram language, (figure 5.7) Illustrates the controlling cycle in PLC.



Symbol	Discrbtion
OL	Overload
HP	High pressure
LP	Low pressure
F0	Flag
T	Thermostat
T2 & T1	Timers
Def.H	Defrost Heater
COMP1	Compressor

Figure (5.7):PLC block diagram

5.6 Project problem

- The carbon dioxide R-744 refrigerant needs for high pressure to obtain on suitable refrigeration and this cause difficulty in choosing the compressor with little available chooses types.

- The refrigerant temperatures at out let compressor relatively vary high with other refrigerant, so choosing the compressor based on temperature and pressure resulting from this temperature.

- Difficult dealing with R-744 refrigerant because of their transcritcal pressur and temperature comparison with other refrigerant.

- Difficult dealing with transcritical refrigeration cycle operating within available technology so obtaining on the cycle components is difficult and cost because of choosing them from the different companies in several countries ,and difficult obtaining on catalogue selection for compressor and evaporate and expansion valve.

5.7 Recommendation

1-One of the main problems encountered through the project is the lack of scientific resources, books, journals and scientific papers required for the researching process , so we recommend that the university takes the trouble of foundation and most modern journals in all field of knowledge and science.

2-when trying to begin in practical implementation .it has been founded that the most of the required equipment and device are not available within local or neiboring market , so we recommended that the university does its best to create others opportunities for supplying students with all what these need .

3- since this project is considered a very important topic of research according to industrial technological requirements ,on the other hand i5t is considered as environment friendly in the field of cooling , so we recommended that in the future other students to try to accomplish this project practically.

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Appendex A

Table(A.1): Average air infiltration rates in L/s due to door opening.

Room volume (m^3)	Infiltration rate (L/s) for room blow zero°c	Infiltration rate (L/s) for room above zero°c
0.125	0.041	0.9
1	0.45	1.3
3	2.01	2.85
7	2.3	3.1
8.5	2.6	3.4
10	2.8	3.7

Table(A.2): Standard aluminum tubing size.

Normal size (inches)	Out side diameter(inches)	Inside diameter (inches)	Wall thicness
3/16	0.281	0.117	0.035
1/4	0.375	0.305	0.035
3/8	0.500	0.402	0.049
5/8	0.750	0.652	0.049
3/4	0.875	0.745	0.065
1	1.125	0.995	0.065

Table(A.3): Properties of metal

Material	**Conductivity (W/m.°C)**	**Density (Kg/m^3)**	**Specific Heat (J/kg.°C)**
Aluminum	202	2800	795
Cupper	388	8933	385
Gold	318	18900	130
Iron	55	7920	456
Lead	35	11373	130
Nickel	99	8906	445.9