EXPERIMENTAL ANALYSIS OF A REFRIGERANT AIR DRYER

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EXPERIMENTAL ANALYSIS OF A REFRIGERANT AIR DRYER

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ABSTRACT

EXPERIMENTAL ANALYSIS OF A REFRIGERANT AIR DRYER

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Compressed air is widely used particularly in industry. In order to increase the quality of the process and lifetime of the machine, the compressed air should be dried. Therefore the air is used after compression and drying processes. The most commonly used machines that do this process are called “Refrigerant Air Dryers”. These air dryers are designed to cool and dehumidify the moist air. The process of decreasing temperature is carried out by a refrigerant, R134a. Unlike design conditions, dryers are working in variable loads (variable compressed air flow rates). An experimental setup is prepared for analyzing the variance on the machine and the performance under these variable loads. This thesis includes the design, preparation and the modification of the refrigeration experimental setup for refrigerant air dryers. The setup is tested under three different conditions and the results are compared.

Keywords: Refrigerant air dryers, R134a, refrigeration cycle, experimental study

ÖZ

SOĞUTMALI HAVA KURUTUCUSUNUN DENEYSEL ANALİZİ

Uslu, Mustafa

Yüksek Lisans, Makine Mühendisliği Bölümü

Tez Yöneticisi: Doç. Dr. Cemil Yamalı

Kasım 2012, 109 Sayfa

Basınçlı hava sanayi uygulamalında yaygın bir şekilde kullanılmaktadır. Makinanin ömrünü ve prosesin kalitesini artırmak için basınçlı hava kurutulmalıdır. Bu nedenle hava sıkıştırıldıktan sonra kurutma işleminden geçirilerek kullanılmaktadır. Bunu yapan makineler arasında en yaygın olarak kullanılanlardan birisi soğutmalı tip basınçlı hava kurutucularıdır. Bu makineler basınçlı havayı soğutarak nemini alma işlemi için tasarlanmıştır. Sıcaklık düşürme işlemini bir soğutkan olan R134a yardımıyla yapmaktadır. Tasarımın aksine kurutucular farklı yüklerde çalışmaktadır. Bu değişken yüklerin kurutucu performansında olan değişimlerin tespiti için bir deney düzeneği hazırlanmıştır. Bu tez soğutmalı kurutucular için hazırlanmış olan soğutma deney düzeneğinin tasarımını ve hazırlanmasını ve soğutma devresinin analizini içermektedir. Hazırlanan düzenek üç ayrı yükte denenmiş olup sonuçlar karşılaştırılmıştır.

Anahtar Kelimeler: Soğutmalı hava kurutucusu, R134a, soğutma çevrimi, deneysel çalışma

To My Family

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P: Pressure, bar

T: Temperature , K, °C

$\dot{m}$: Mass flow rate, kg/s

$\dot{V}$: Volumetric flow rate, m³/s

Vo: Voltage, Volts

I: Current, Amps

ρ: Density, kg/m³

$\dot{Q}$: Heat transfer rate, W

$\dot{W}$:Work rate, W

η: Efficiency

$\emptyset$: Relative humidity

w: Specific humidity, kgw/kga

COP: Coefficient of performance

h: Enthalpy, kJ/kg

s: Entropy, kJ/kg-K

II: 2nd Law of Thermodynamics

Δ: Difference

Subscripts

0:Reference state

1: Air to air heat exchanger outer tube, compressor inlet

1e: Evaporator outlet

2: Air to air heat exchanger inner tube, compressor outlet

2c: Condenser inlet

2s: Isentropic compression state

3: Condenser outlet

4: Evaporator inlet

E: Electrical

t,: Total

c: Cold

h: Hot

m: Mean

s: Isentropic

R: Reservoir

Cond: Condenser

i,inlet: Inlet state

o, outlet: Outlet state, overall

condense: Condensed

Exp: Expansion valve

Evap: Evaporator

Comp: Compressor

Gen: Generation

Ref: Refrigerant

Water: Water

Air: Air

Dp: Dew point

comp-cond: Between compressor and condenser

Comp-electrical: Compressor electricity

Comp-actual: Actual compressor

Comp-exact: Required compressor

Ref-comp: Refrigerant flow through compressor

Ref-evap: Refrigerant flow through evaporator

Ref-HGBPV: Refrigerant flow through hot gas by pass valve

CHAPTER 1

INTRODUCTION

1.1. High Pressure Air Drying

Compressed air is commonly used for power in industrial, military and medical applications. In average, 10% of the electric power used in industry is consumed to service the air in compressed form in European Union countries [1; 2].

Since atmospheric air contains moisture, compressed air contains in higher/denser values. The equipments, which are working with compressed air, are affected adversely by this moisture. Corrosion and damage in equipment, resistance in flow, and higher energy consumptions are some of unfavorable consequences of the moisture in compressed air. Additionally, especially some progress that the direct usage compressed air is necessary; the moisture affects the progress directly. Spray painting with compressed air is commonly used process in industry. If there is even small amount of moisture inside the paint, the homogeneous distribution and the quality of the paint are affected indifferently. These factors cause low quality in application and product, and high cost in maintenance.

1.2. Air Drying Systems

In order to obtain optimum level in cost and quality of the product, the moisture in compressed air should be removed. The air drying systems are developed to reach this goal. There are several systems available in use [3].

Among these air drying systems, five of them are commonly used.

1. Deliquescent Air Dryers: Contains a desiccant (salt tablets mostly) which absorbs moisture
2. Regenerative Air Dryers: Uses microscopic porous desiccant like silica gel, active alumina etc.
3. Heatless Air Dryers: 2 chamber of desiccant dryers are placed on the compressed air line. Needs purge to regenerate which costs nearly 18% of compressed air to atmosphere.
4. Heat Regenerative: 2 chambers and additional heat is applied to 1st chamber to dry the desiccant before it reaches its max adsorption capacity
5. Refrigerant Air Dryers: Dries air by cooling down to below waters dew point at that pressure
    - Non Cycling Refrigerant Air Dryers
    - Cycling Refrigerant Air Dryers
    - Digital Cycling Refrigerant Air Dryers

First four items of these dryers use an additional material (salt tablets, silica gel, active alumina and/or other desiccants) to dry the air. These are "chemical" processes and with help of mass transfer, very dry air (up to 0.34 PPM) outlets could be obtained. Unlike refrigerant air dryers, desiccants have a lifetime and need to be changed. The refrigerant air dryers are known "physical" process dryers. These dryers dehumidify the air up to 973 PPM. In following paragraph,

the physics of these dryers are explained briefly. For long service and life time with highly reliability, refrigerant air dryers are commonly used in applications [3].

Refrigerant air dryers cool the compressed air to its dew-point at the relevant water vapor pressure. In order to cool the compressed air to dew-point temperature, a refrigeration system is used with a special designed heat exchanger. The “dew point” is the temperature of water vapor-gas mixture (moist air in this case) that water vapor could condense or solidify. A model that is called “Dalton Model” could be used to define this point [4]. Dalton Model defines partial pressures in the mixture. By using partial pressure of the water vapor and saturation tables of the water, saturation temperature could be found and assigned to be “dew point” temperature of the mixture.

A vapor compression cycle is generally used as a refrigeration unit in these dryers. Since the aim is to reach the dew point of air-water vapor mixture, the design of the air dryer is carried out accordingly. The calculation code has been written in Matlab ® and could be found in Appendix B.

In most cases in industry, refrigeration cycles are designed for steady state operations. However, the temperature and moisture of compressed air varies because of the climate where the air dryer is operated. On the other hand, the amount of compressed air varies depending on the application and hours of operation. Therefore designing a system for steady operation will yield either excess energy usage or undesired (wet) air. For this reason, a hot-gas by pass valve and alternative solutions are mentioned in this thesis and used in experimental setup.

The “Cycling” concept becomes important while selecting the right size of refrigerant air dryers for desired operation. If the operation requires constant load of compressed air (i.e. constant flow rate) non-cycling refrigerant air dryers are preferred. The design is based on only to a certain constant flow rate. If the amount of compressed air varies, the cycling refrigerant air dryers are better option. The refrigeration cycle cools down a water mixture (generally glycol and water) and a pump circulates this mixture to a heat exchanger to dry the compressed air. The mixture is known as “Thermal Mass”. The idea of using this concept is to cool down a certain amount of mixture and keep it in insulated vessel and after desired temperature is reached, the refrigeration cycle shuts down and drying process is continuing with water mixture. The main advantage of these dryers is to operate at design and lower flow rates without operating the refrigeration cycle all the time which increases the operation costs [3].

Following section gives main information about; vapor compression cycles, components that are commonly used, and previous studies about these cycles. The next chapter contains the experimental setup, design of the cycle, selection of the components, preparation, and calibration of the equipments. Chapter 2 and 3 cover preparation of the experimental setup and analysis of the refrigeration cycle. Chapter 4 involves the measurements and Chapter 5 presents the results of the analysis. Chapter 6 concludes by giving an overview and further studies that could be carried out.

1.3. Vapor Compression Cycles

Vapor compression cycles are the most commonly used refrigeration cycles in industrial and domestic heating and cooling applications. A basic vapor compression cycle includes four components which are a compressor, a condenser, a metering device (i.e. expansion valve) and an evaporator. This cycle is generated from “Reversed Vapor Power Cycle of Carnot” [5]. Figure 1 shows an ideal reversed vapor power cycle of Carnot and temperature-entropy diagram.

Basically in ideal case, the compressor pressurizes the saturated gas refrigerant to superheat. This superheated refrigerant is cooled down by a condenser under constant pressure till it reaches saturated liquid phase. An expansion device expands the liquid into liquid-vapor mixture isenthalpically to low pressure. In evaporator, the heat is transferred to the mixture which results in evaporation and cycle restarts.

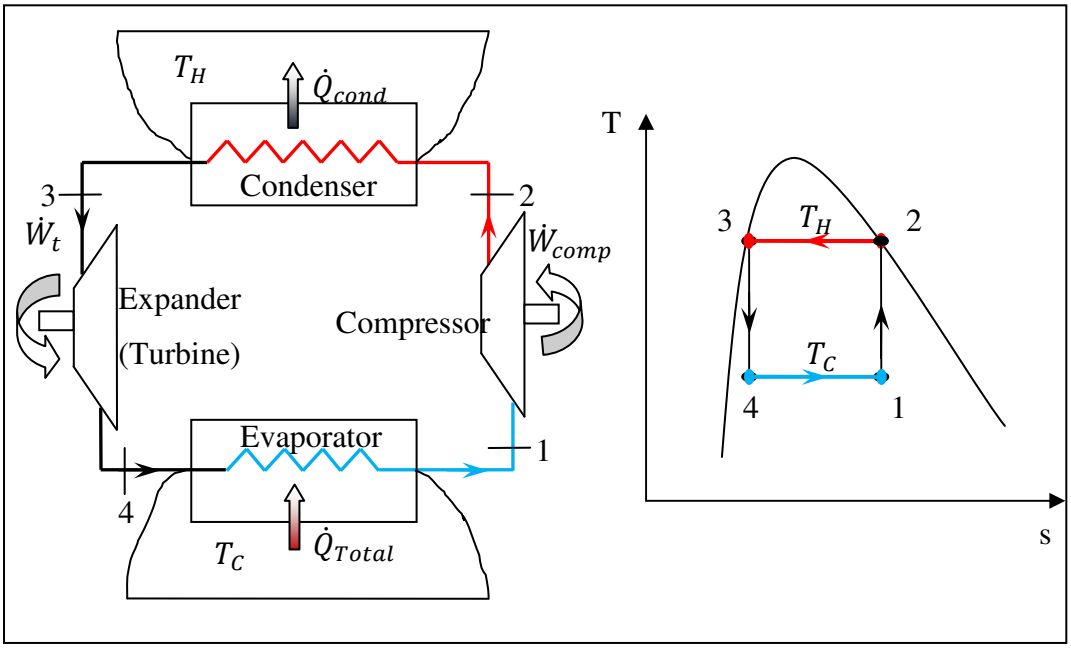

Figure 1. Reversed Carnot Cycle and its T-s diagram

In vapor compression cycle, the compressor pressurizes the saturated vapor refrigerant to superheat state. Condenser rejects the heat from the refrigerant to obtain saturated liquid at high pressure. An expansion valve or capillary tube expands the liquid into saturated liquid-vapor mixture at low pressure state which is ready to boil. The heat is transferred from the evaporator to this low temperature boiling fluid and this heat vaporizes the mixture until it reaches saturated vapor phase at the inlet of compressor. Figure 2 shows an ideal vapor compression cycle with its temperature versus entropy diagram [5].

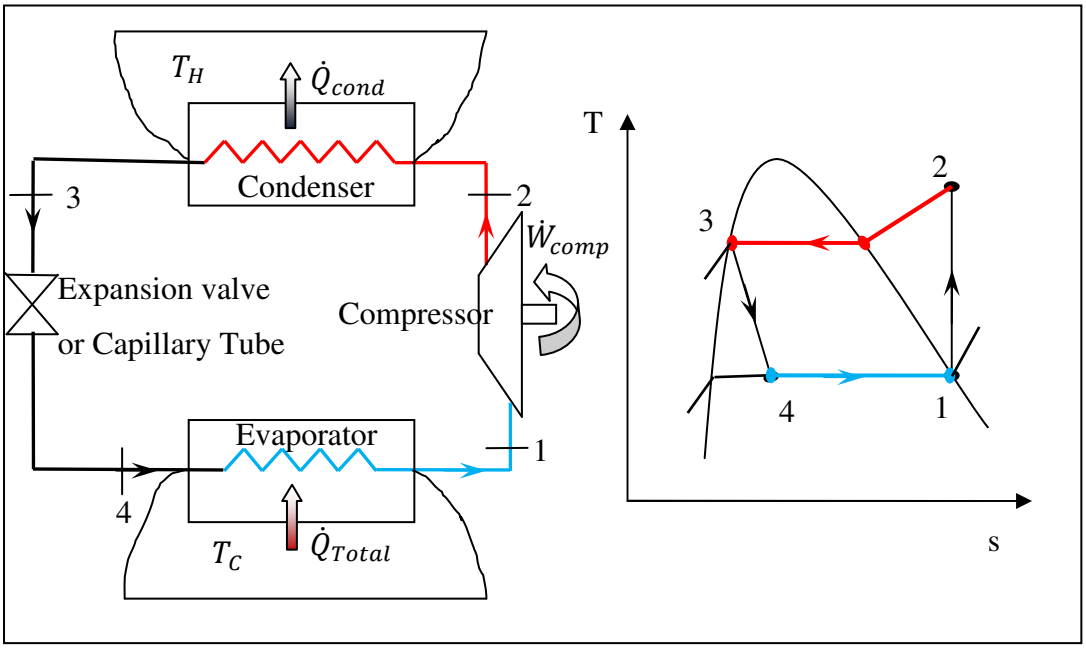

Figure 2. Ideal vapor compression cycle and its T-s diagram

In reality, this ideal cycle does not show up. There are some impacts to the cycle which force the cycle not to be ideal. Some of them are on purpose such as outlet of the condenser is tried to be in subcooled region to have liquid phase in expansion inlet or in outlet of evaporator is tried to hold in superheat phase in order to avoid liquid flood back to compressor which is not desired. There are some unintentional facts such as pressure drop during flow of refrigerant or heat gain or loss to the ambient etc. Because of these facts, the cycle deviates from ideal cycle [4]. Figure 3 illustrates the deviation of actual refrigeration cycle. The ideal cycle is shown with black straight line while the deviation is seen with red dashed line.

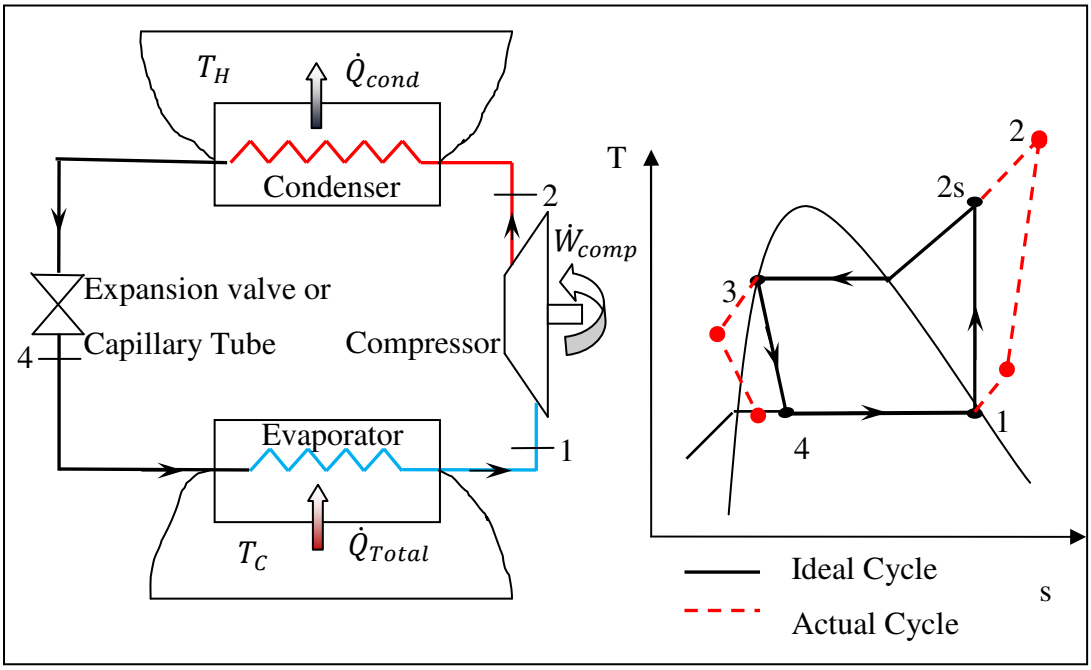

Figure 3. Actual vapor compression cycle and its T-s diagram

There are several models have been developed about the behavior of vapor compression cycles for their steady and unsteady operations. The transient model of liquid chillers with vapor compression cycle simulates the heat transfer coefficients till the cycle reaches its steady conditions. Validating with experiments showed that compressor reaches steady state as well as evaporator while condenser temperature profile oscillates because of "part-load" condition found by Browne and Bansal [6].

In order to simulate start up and shut down operations, in evaporator and condenser where two phase flow occur, "Switched Boundary Model" was used by Li and Alleyne [7]. The model was checked with experimental data. The results of model seem to be accurate but with an error which requires future work for dynamic operations.

Apart from unsteady modeling, experimental analysis has been carried out by Zeng and at al. [8] for HFC Refrigerants in vapor compression cycles. Depending on refrigerant, the operating and refrigerating efficiency could be different in the same experimental setup. Similarly, Cabello, Torrella and Navarro-Esbri prepared a study about refrigerants and their performance on vapor compression cycle in terms of evaporating pressure, superheating degree of vapor and condensing pressure. It is concluded that the mass flow rate is the most effective variable in refrigeration cycle on the capacity. On the other hand, for same compression ratios, COP differs from one fluid to another [9].

For unstable operation there are several applications and equipments in use for refrigeration cycles. Variable speed compressors and hot gas by pass valves are commonly used for adjusting the refrigeration capacity. Especially variable speed compressor, it was found beneficial that to have better stability, the speed reductions should be kept at smaller varying magnitudes [10]. Additionally, in order to increase the coefficient of performance (COP), ejectors were used as expander in the cycle by Nehdi, Kairouani and Bouzania. The results showed that the COP of modified cycle differs from one refrigerant to other while for all refrigerants, the COP increased [11].

CHAPTER 2

EXPERIMENTAL SETUP

The main goal of this thesis is to build an experimental setup that helps to analyze the behavior of a vapor compression cycle components of a compressed air dryer. The experimental setup, components, design and preparation of the setup, calibration-verification of the equipments, experiments that are carried out preliminary are described in this chapter of the thesis.

2.1. Components

The main equipments that are used in this experimental setup are: compressors, condensers, evaporators, expansion valves, liquid receivers and gas accumulators, dryers and hot gas bypass valves. Additionally, liquid and gas flow meters, pressure transmitters, manometers, RTDs, thermocouples, a dew-point temperature reading device and a data accusation device are used to measure all necessary readings to test and evaluate the cycle.

2.1.1. Compressors

There are two different types of compressors in the experimental setup. The first type is hermetic reciprocating compressor. These types of compressors are commonly used in domestic and industrial applications because of their lower purchase cost and repairable body. Figure 4 shows the reciprocating compressor that is used in experimental setup. The brand is Tecumseh and the model is

FH4518Y. The electrical power of this compressor is 1427 W and under experiment conditions at nominal capacity the COP should be about 3.04.

Figure 4. Reciprocating hermetic compressor[22]

The second type of compressor is called scroll compressor. The volumetric efficiency of these compressors is better than the reciprocating piston compressor since there is the clearance volume required very small in this type of compressor [12]. Another advantage of these compressors is small amount of liquid flood (liquid shock) in to compressor does not result in any kind of harm. Especially at the start-up operations of refrigeration cycles, some liquid refrigerant flood might be seen at the inlet of the compressors. Main disadvantage of these compressors is initial cost. Figure 5 shows the scroll type compressor that is used in experimental setup. Both of the compressor specifications are given in Appendix A.1.

Figure 5. Scroll type compressor[33]

2.1.2. Condensers

Four different sizes of condensers are used in the experimental setup. The purpose is to have a wide range of condensing capacity so the cycle could be experimented for smaller and larger capacity of the condenser and the effect on the cycle. The type of condensers in this setup is forced convection air-cooled condensers. Basically the copper pipes are surrounded by plate fins and a fan forces air to cross this section and cool down and change the phase of the refrigerant from superheated vapor to saturated or subcooled liquid. As a heat exchanger, cross-flow pattern is used. Figure 6 illustrates the condensers that are on the experimental setup. Specifications of the condensers are given in Appendix A.2.

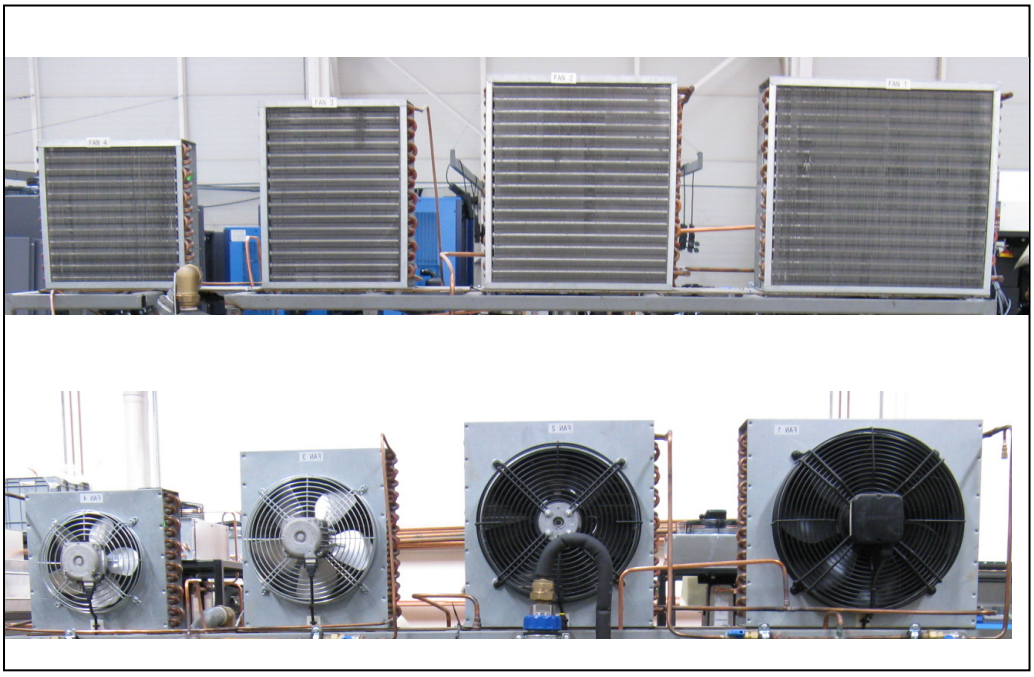

Figure 6. Air cooled condensers

2.1.3. Evaporators

There are many types of specific evaporators in use in industrial application. The main components such as compressors, expansion valves are similar to each other in refrigeration application. However depending on the application, the evaporators are designed specifically. In order to dry the compressed air, a unique design of annular tube heat exchanger is used in the experimental setup. The picture and technical drawing of the evaporator is seen in Figure 7. Specifications of the evaporators are given in Appendix A.3.

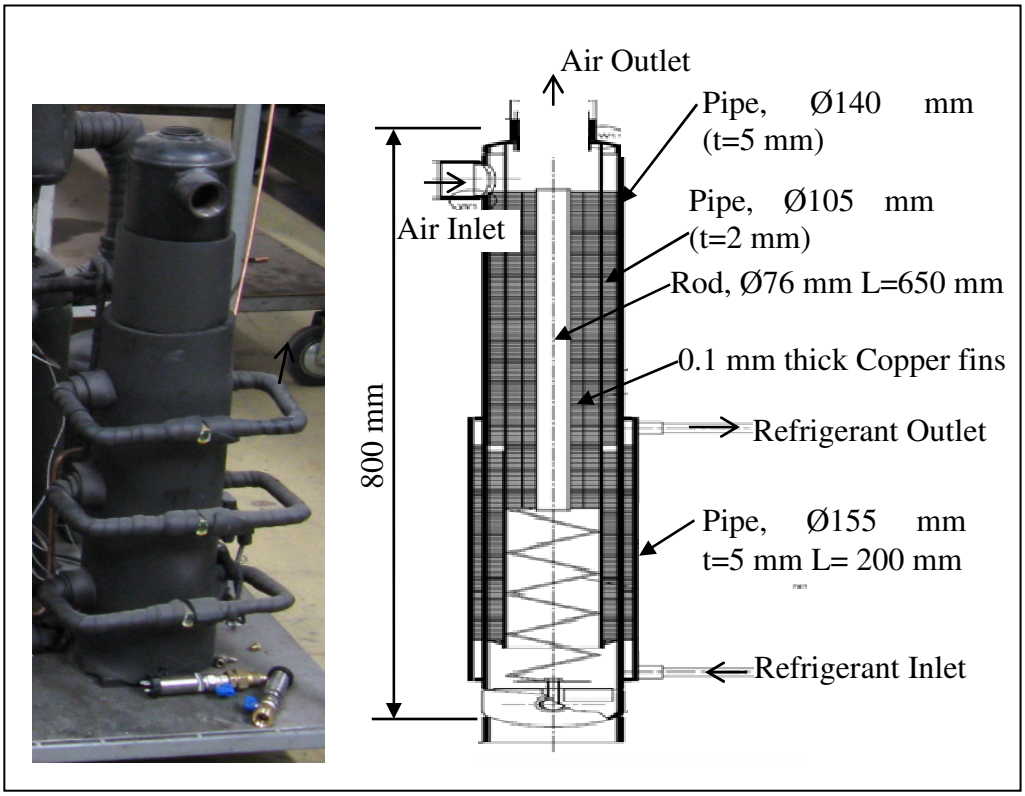

Figure 7. Evaporator and its technical drawing

Essentially there are two main parts of this design. First part is called "economizer". Since the outlet temperature of heat exchanger is not important for compressed dry air usage, this cold air is used to pre-cool the entering of moist compressed air. Basically this is a counter-flow air to air heat exchanger. There are fins placed between annular tubes to enhance heat transfer. The cross section of evaporator could be seen in Figure 8.

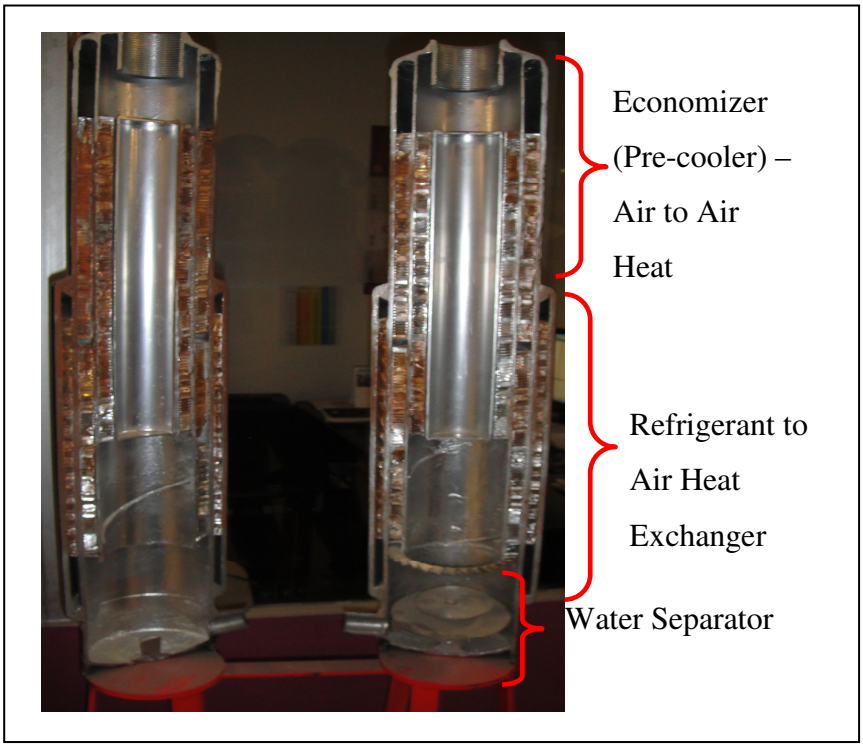

Figure 8. The cross section of evaporator with its internal sections

The second part of the heat exchanger is the place where refrigerant fluid is in use. A well insulated jacket-like tube welded at the outside of outer tube to hold refrigerant fluid which is seen in Figure 8 as well. In this part, condensation of water droplets occurs. Since the aim is cooling the compressed moist air down to dew point to condense the water, the minimum temperature should be at the bottom of the heat exchanger. Therefore an economizing system is in use by annular tube pattern to pre-cool incoming moist air in this section as well.

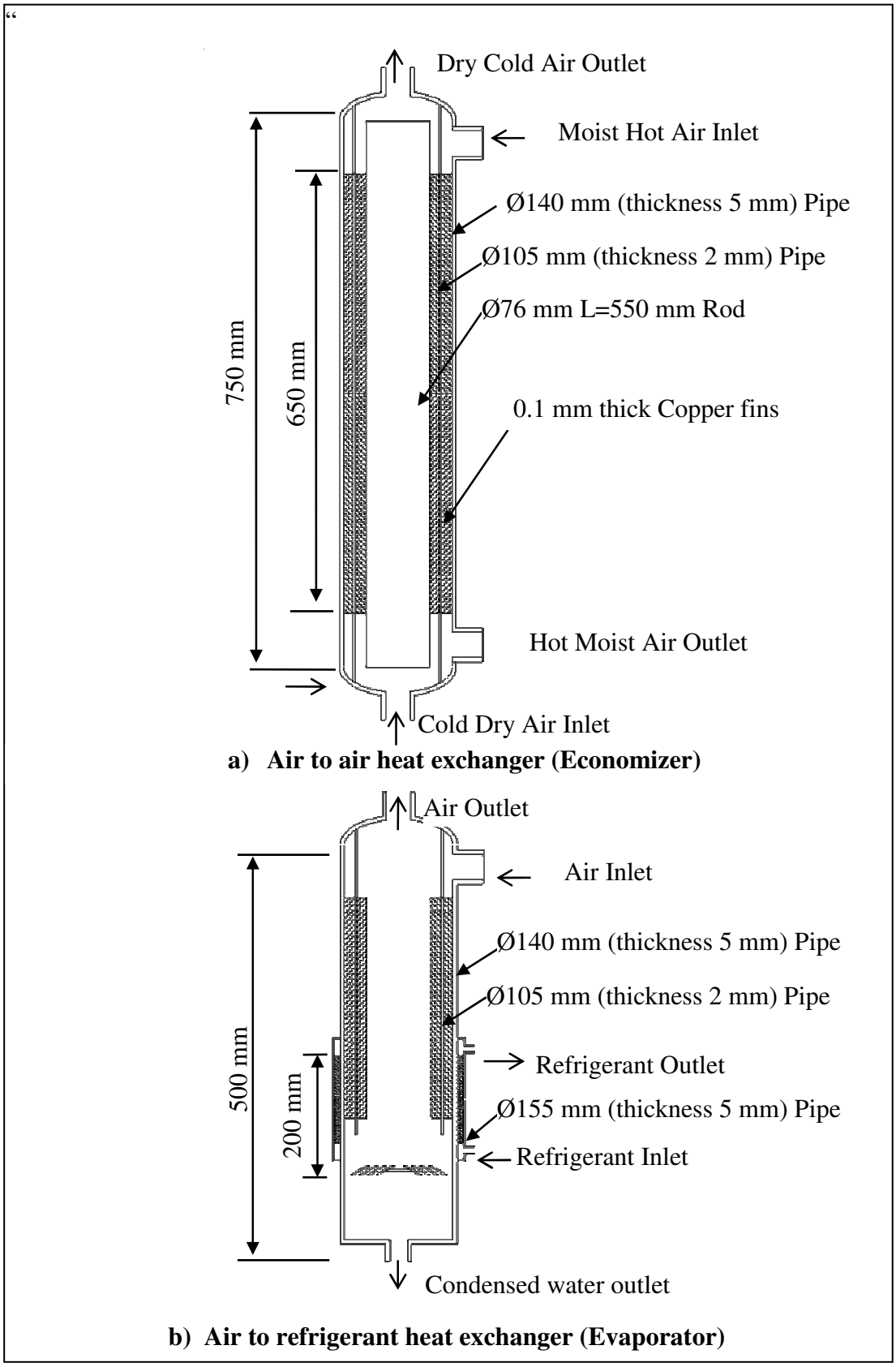

Figure 9. Detailed technical drawing of economizer and evaporator.

In Figure 9 the details of new construction of the heat exchangers could be found. All fins are made with copper with a thickness of 0.1 mm. Upper heat exchanger is called “Economizer” and bottom heat exchanger is called “Evaporator “ in this thesis. In economizer, there is a Ø76 mm rod in the middle of the heat exchanger. This purpose of this rod is to restrict the flow cross sectional area to increase the velocity and in the end obtain turbulent flow in the heat exchanger.

In Figure 8, it is also seen that there are fins between all sections. The fin has 0.1 mm thickness and the material is copper. The structure of the fins is called “Louvered Strip Fins”. There are several studies about this fins however all studies are carried out in single phase flow. It is not common to find these strip fins with their boiling and condensing characteristics in terms of heat transfer. One study about this matter, has been carried out by B.Kim and B.Sohn [13]. After processing experimental data with correlation, it was found that the heat transfer coefficients could be predicted with an accuracy of ±25%.

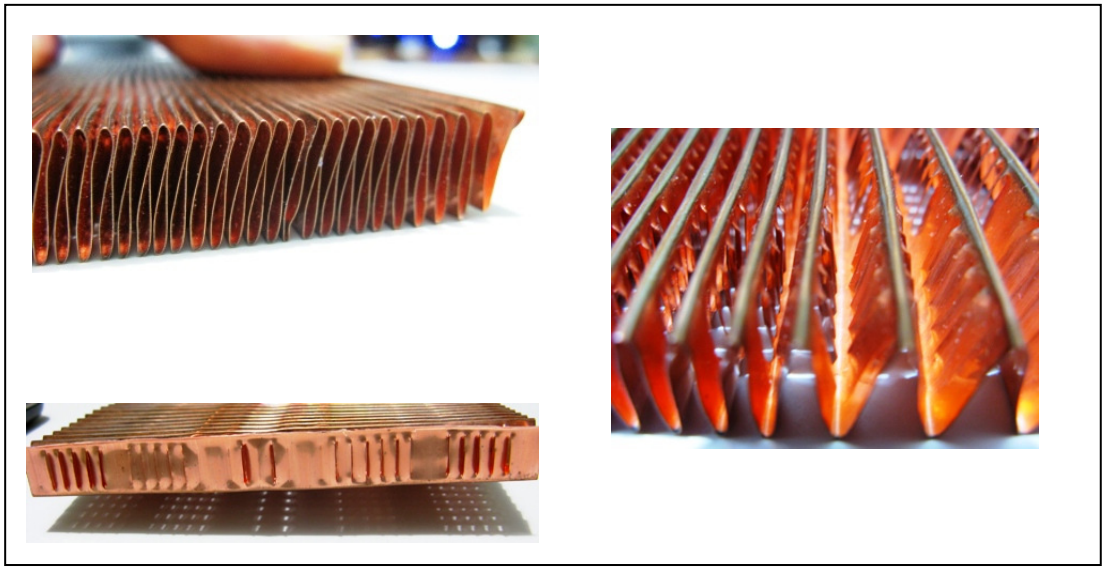

Figure 10. Pictures of the louvered fins in detail

The unique design of the louvered fins is illustrated in Figure 10. The louvers allow the fluid to flow all through the channel up and down which creates turbulence to enhance the heat transfer coefficient.

2.1.4. Expansion Valves

Expansion valves are main metering devices that are used in refrigeration cycles. A metering device is designed to throttle the flow of refrigerant into the evaporator. By restricting the cross section of the flow, the velocity is increased and pressure and temperature are decreased. With the help of this operation, a low temperature saturated vapor-liquid mixture is obtained which is ready to boil. There are seven basic types of expansion valves in use. In this experimental setup, externally equalized thermal expansion valve is used in the inlet of both evaporators. This type of expansion device contains a diaphragm that is balanced by sensing bulb charge and spring pressure with external equalizer pressure of outlet of evaporator. The main advantage of this kind of valve is avoiding negative effect of pressure drop in heat exchanger which results better performance [12]. A schematic view of expansion valves and the one which are used in experimental setup are seen in Figure 11. Further information could be seen in Appendix A.4.

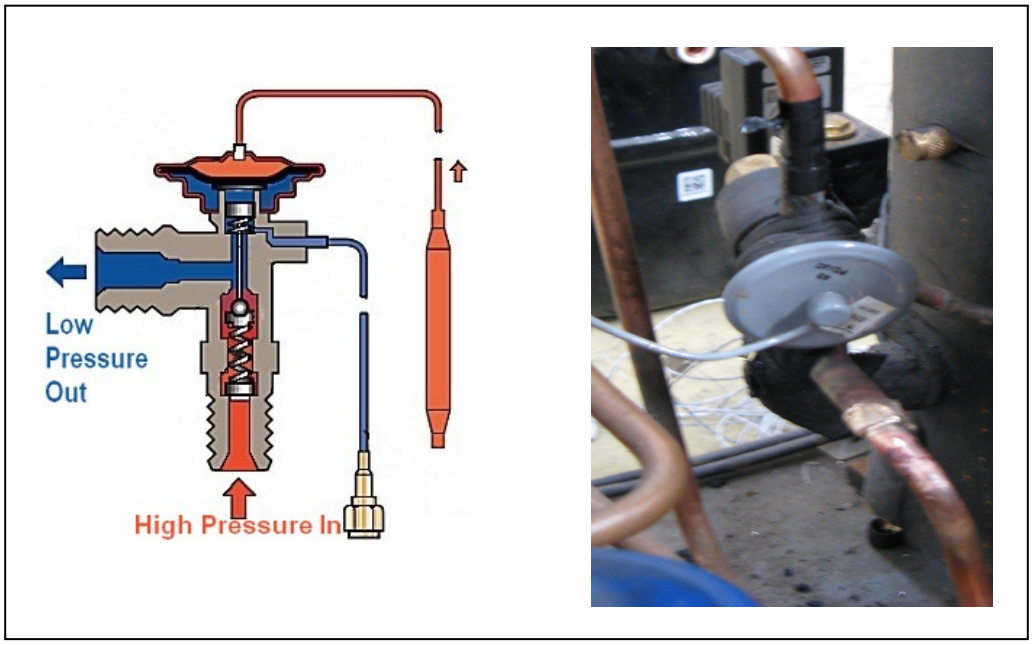

Figure 11. The schematic view of externally equalized thermostatic expansion valve and the one used in experimental setup [23].

2.1.5. Receivers and Accumulators

An accumulator is designed to contain the initial surge of liquid that flows from the evaporator when the liquid refrigerant or oil leaving the evaporator in order to reduce the possibility of liquid flood-back to the compressor. Figure 12 shows the accumulator and its inlet and outlet.

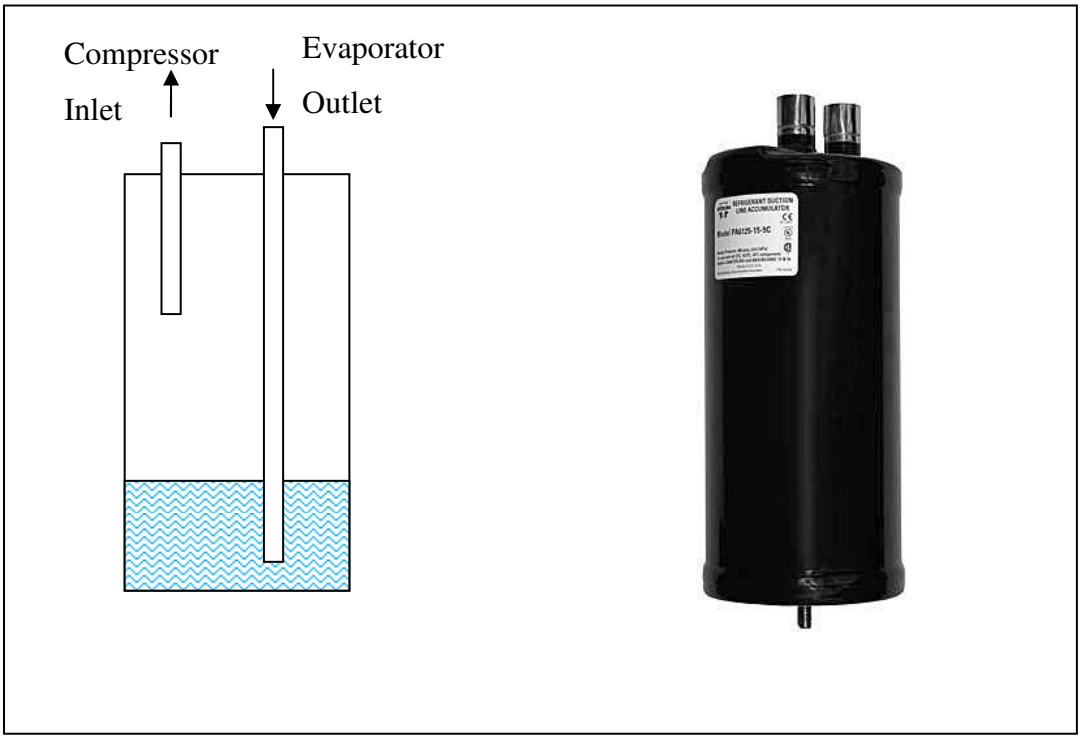

Figure 12. The schematic view and picture of an accumulator [34]

The piping coming from the outlet of evaporator goes down to the bottom of the accumulator to avoid any drizzling liquid suction to compressor and inlet pipe of the compressor is at the top of the accumulator.

When compressor is shut down, expansion valve does not allow refrigerant flow. However, if capillary tube is used as metering device, when compressor is shut down, the fluid in condenser flows through the capillary tube to evaporator. Since there is no load in evaporator, this fluid (generally liquid refrigerant) flows to the inlet of compressor. When the cycle is desired to start up, compressor sucks liquid because of its inlet filled with liquid, and this fact results in a failure in compressor. To avoid this, a receiver placed after condenser. The schematic view and photograph of the receiver and its inlet and outlet are seen in Figure 13.

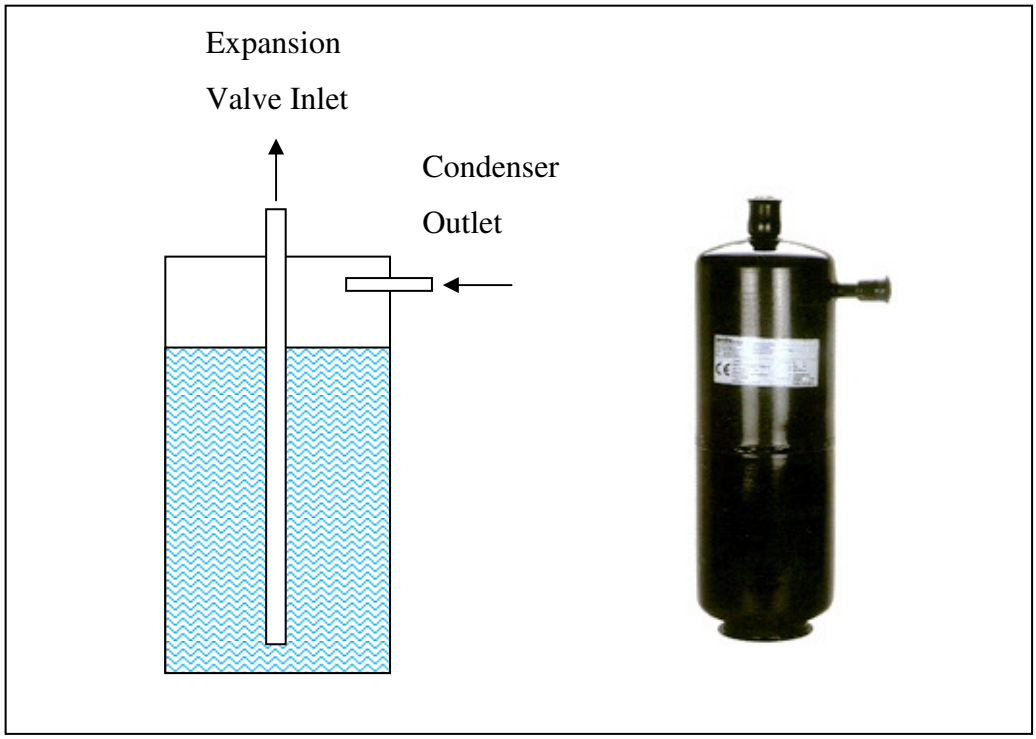

Figure 13. The schematic view and picture of liquid receiver [34]

In general, even with expansion valve used in the cycle, a receiver is placed to the refrigeration cycle to stabilize and operates like an ideal cycle. With this vessel, expansion valve or capillary tube inlet is nearly always in saturated liquid phase.

2.1.6. Hot Gas By-Pass Valve

In actual refrigeration cycles where variable loads are in charge, cycle is designed for optimum and generally for maximum load. Inherently, if the evaporator encounters with the load below its design capacity, the cycle starts to behave unsteady, such as frost formation in evaporation side, liquid suction at compressor, low temperature condensing and so on. In order to avoid such problems, there are several actions that can be taken. One of the most popular and practical action is carried out by using a valve called “hot gas by pass valve". This valve operates between outlet of compressor and inlet of the evaporator. Highly pressurized superheated vapor refrigerant crosses through the valve to low pressure and temperature inlet of evaporator and mixes before entering to the

evaporator. Therefore it may be told that this valve adds an artificial load on evaporator, reduce a little amount refrigerant in compressor inlet to avoid instabilities. Figure 14 shows this type of connection.

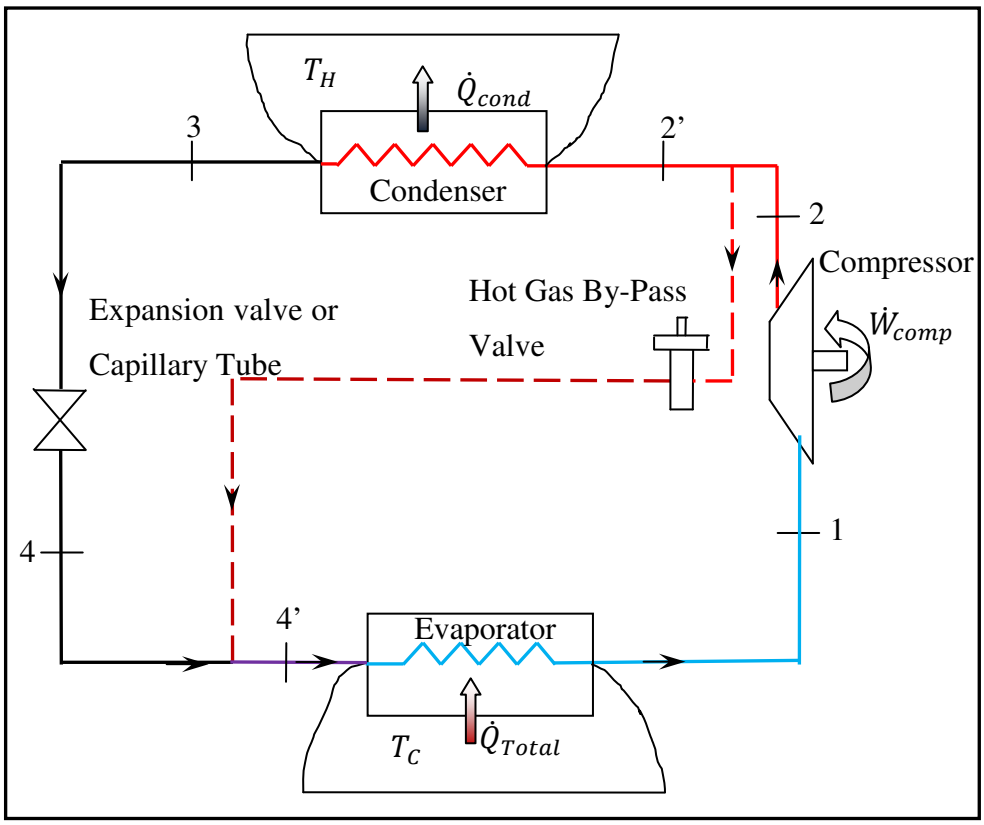

Figure 14. Hot gas by pass valve application from compressor outlet to evaporator inlet

Another connection in application is carried out by mixing high pressure hot gas at the outlet of compressor with low pressure low temperature gas at the outlet of evaporator / inlet of compressor. With this application, there is no artificial load on evaporator, however the amount of refrigerant that flows to evaporator inlet is reduced, and additionally mixing hot gas with cold outlet of evaporator increase the temperature of inlet of compressor to ensure only gas phase flows to the suction of compressor. The connection details are given in Figure 15.

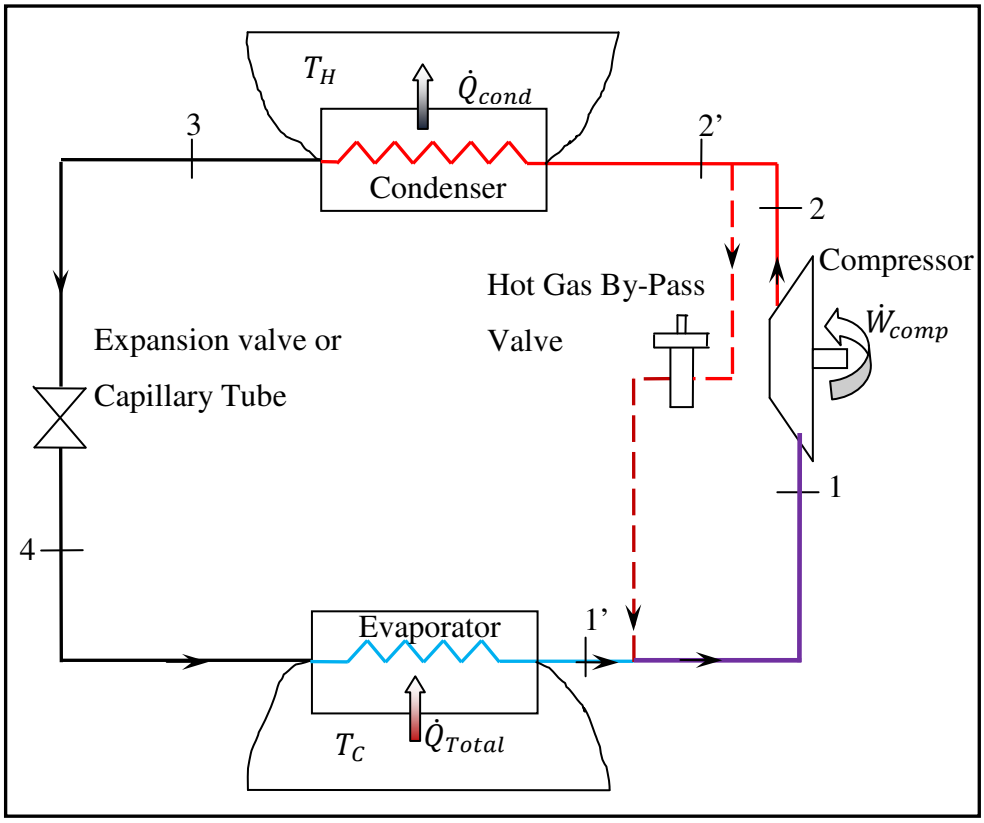

Figure 15. Hot gas by pass valve application from compressor outlet to the inlet

Second application of the hot gas by pass valve is not commonly used in industry since the concern is most of the time is response time of evaporator under variable load. In dryer case, where liquid shock to the inlet of the compressor is not desired and should be defeated quickly, by passing the hot gas to the inlet of the compressor is selected as a solution. This way the response time of the evaporator is not fast but this guarantees there is no liquid shock on the compressor. The specification of the valve is given in Appendix A.5.

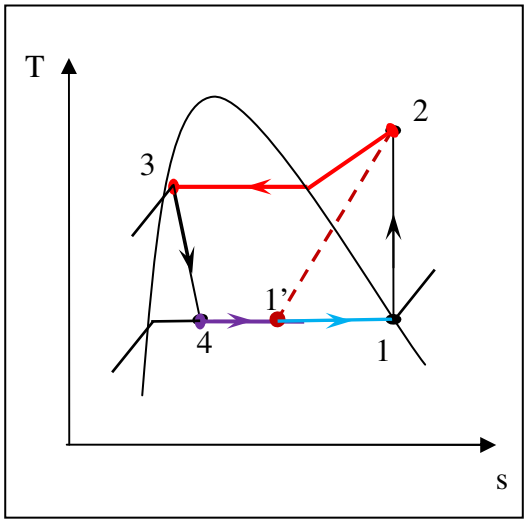

Figure 16. Temperature – Entropy diagram of the hot gas by pass valve application

Figure 16 indicates the effect of the hot gas by pass valve on the T-S diagram. The outlet of evaporator is shown as 1' and this position moves to the left if the load is increased and to the right if the load is decreased.

2.1.7. Oil Separator And Filter Dryer

In both reciprocating and scroll compressors, there is oil for smooth movement of the piston or scroll. Most of the case, fairly small amount of this oil leaks from the seals and pollutes the refrigerant. This fact results in decrease in heat transfer in evaporators and condensers and means a decrease in coefficient of performance [14; 15]. To avoid this effect, there are several materials in use in refrigeration applications such as oil separators, filters etc. The oil separators are used with compressors with low temperature discharges such as R404a or cryogenic applications. However in the experimental setup, where temperature of discharge is not low, it is not necessary to use. Therefore a filter dryer is used in the setup. This filter dryer contains solid based silica gel with molecular sieve and activated aluminum oxide. This equipment keeps any undesired materials such as oil, water, vapor etc away in the refrigeration cycle. The schematic cross section and the picture of the dryer is given in Figure 17.

Figure 17. The schematic cross section and the picture of the filter dryer [29]

2.1.8. Measurement Devices/Sensors

2.1.8.1. Flow Measurements

There are two flow meters purchased for this thesis for refrigeration cycle. They are both in 0.011-0.1 kg/s range. H250 type is designed for high pressure liquid flow of R134a and VA40 type is for gas flow in low pressures. To keep the equipment cost at a reasonable limit, the mass flow measurements are done manually. In Figure 18 both of the flow meters are seen. The vertical tube flow meter is for gas while the rectangular one is designed for high pressure liquid flows. The specification and detailed information for both flow meters are given in appendix A.6.

Figure 18. VA40 and H250 type flow meters [31]

2.1.8.2. Pressure Measurements

There are five pressure transmitters and five manometers are connected to the cycle in various parts of the refrigeration cycle to measure pressures in different regions. Pressure transmitters are 4-20 mA types and used in between 0-20 bars. The connection pins are numbered. Transmitters are connected to the data logger with a 24V DC power supply. The power supply input is 220 V AC and the output is 24 V DC. The picture of the transmitter and power supply are shown in Figure 19. In Appendix A.7 the specification of the pressure transmitter is describe in detail.

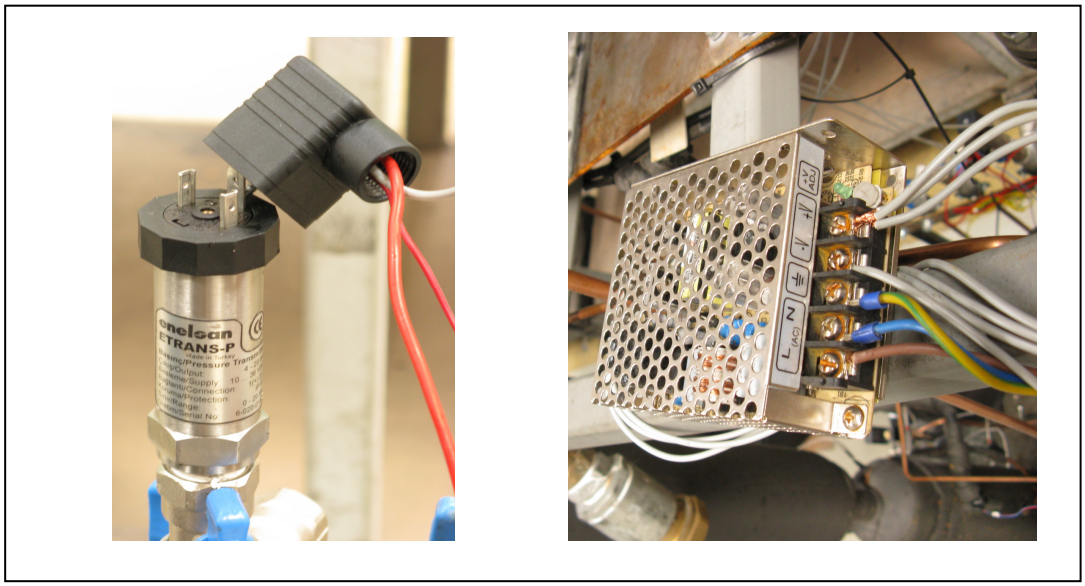

Figure 19. The pictures of pressure transmitter and DC power supply [32]

On the pressure transmitter, letter “E” represents the earth or ground, “1” is live and to be connected to “+” pole of DC power supply and 2 is return signal for data logger. “-” pole of DC power supply is connected directly to the data logger external “#” port.

2.1.8.3. Temperature Measurements

Two types of temperature devices are used in the experimental setup. The first type is copper – constantan (T type) thermocouples. These thermocouples are commonly used in industrial and laboratory applications. They are well known with their fast response time and accurate readings.

The other probe is called RTD which is also known “Resistance Temperature Detector”. They are both sensitive temperature measurement devices and commonly used in many applications. Basically RTDs are made with fine coiled wire, most of the time platinum, wrapped around a core which is ceramic or glass. The idea of RTD is determining the behavior of the wire resistance against the temperature difference and finding the relation between these two parameters.

The main advantages of using RTD in the experimental setup are high accuracy, precision measurement and with a special fitting the probe can be used to read directly from the flow.

Since the sensitivity is important for measurements, 1/16" RTD probes are selected for this setup. Both RTDs and thermocouples are connected to the data logger. The purchased RTDs are platinum type, three wired Class A (α=0.00385 $\Omega/\Omega/°C$) which can be used in the temperature range of -200°C to 260°C (the maximum allowed temperature for transition and cable). Further information could be found in Appendix A.8.

2.1.8.4. Dew Point Temperature Determination Device

A Dew-Point Temperature Device is connected at the outlet of air flow. This device has already been owned by the company where compressed air is provided as well. The device takes sample and finds the saturated dew point temperature according to the amount of water that is measured. The technical data is given in Appendix A.10.

2.1.8.5. Data Accusation Device

One of the most important parts of the setup is data accusation device. The brand and model of the logger are "Datataker ® DT85". This device converts all data signals to a readable document mostly in text or Microsoft Excel® file. The technical data is given in Appendix A.11. All pressure and temperature readings are carried out by this device. The device and its software recognize RTD and thermocouples directly. However, to clarify all readings and confirm their stability, thermocouples and RTDs are double-checked with calibrated ones. Connections and picture of the data logger could be seen in Figure 20.

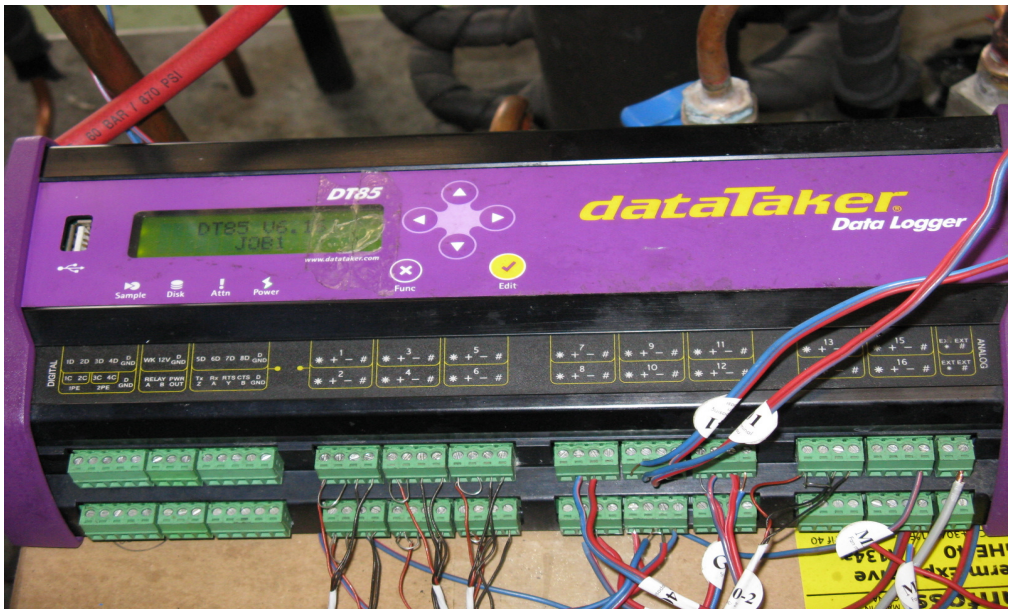

Figure 20. Data logger [21]

2.1.8.6. Installation And Preparation Of Experimental Setup

At the beginning of installation, a structural frame is designed to hold all devices and equipments that would be in experimental setup. For this purpose, 20 mm box profile steel was used.

The experimental setup was placed on a vertical patform. The reason of vertical structure is to see all components clearly and access any device and accessory easily. The height is determined by the height of highest structure which is the double deck heat exchanger. There are two shelves on structural frame to put heavy equipments on them. One is placed at the top for condensers and the bottom one is for heavy parts such as compressors, receivers-accumulators and evaporators. The schematic drawing is given in Figure 21.

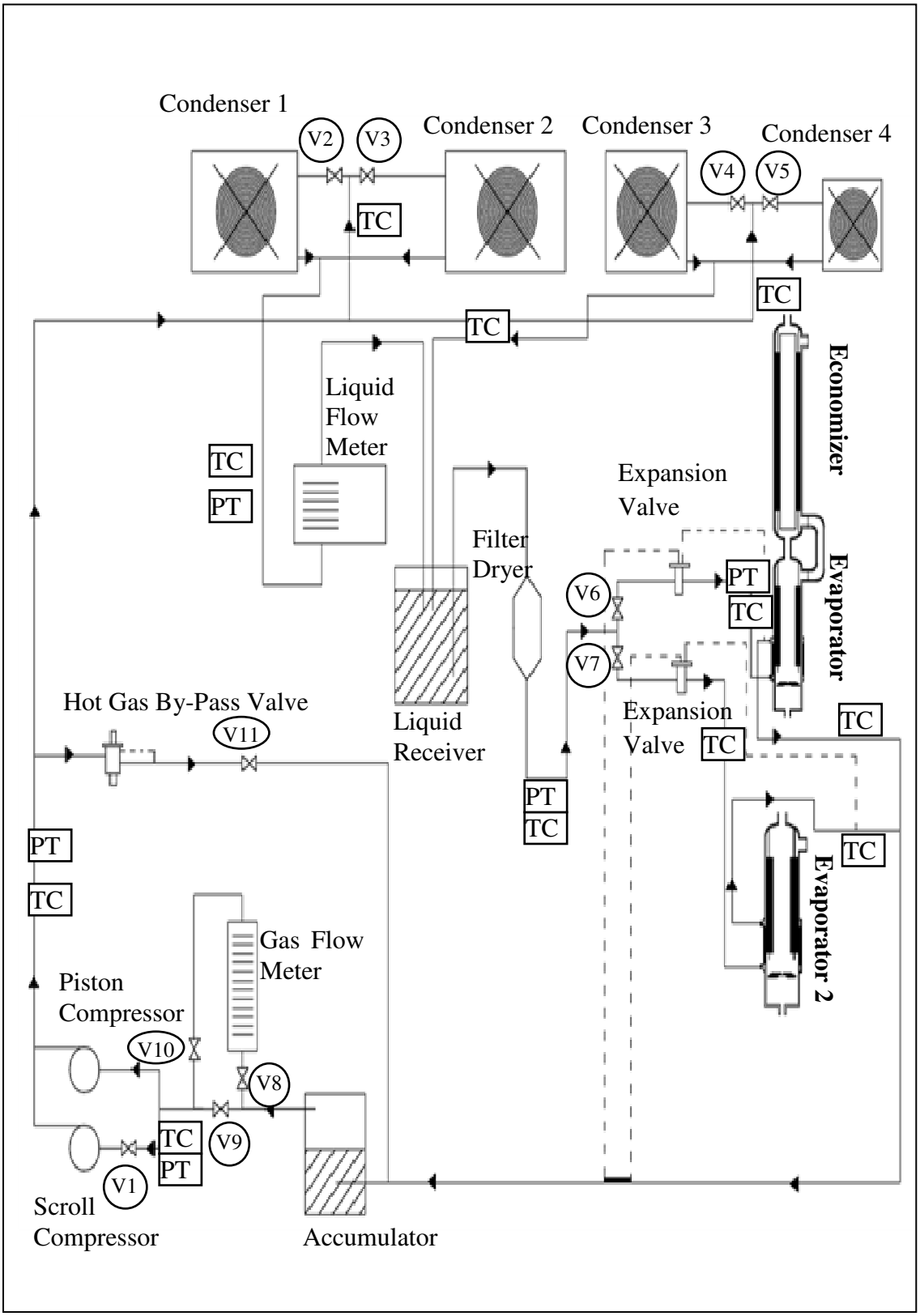

Figure 21. Schematic drawing of the experimental setup components

In the refrigeration cycle, compressor pressurizes the refrigerant and the line is split in two lines. The one that goes to hot gas by pass valve to avoids any liquid shock at the inlet of the compressor. Other line goes to condensers section. The selection of the condenser is carried out with relevant manual valves. After condensing the superheat vapor, the liquid refrigerant flows through the liquid flow meter then the liquid receiver. This component guaranties always liquid phase refrigerant staying at the inlet of the expansion valve. The refrigerant then flows through a filter dryer to leave all possible contaminants before reaching to expansion valve. The operation of each evaporator is controlled by manual valves (V6 and V7) that are placed just before the related expansion valves. After evaporation occurs in the evaporators, the refrigerant flows to accumulator which is a small vessel that holds the liquid refrigerant to avoid liquid flood in to the inlet of the compressor. In application this solution is not enough for various evaporation loads, therefore at lower capacity conditions, hot gas by pass valve injects hot gas to evaporate liquid on the compressor inlet line.

Table 1 - Valve list

Valve Number	**Description**
V1	Manual Valve for Scroll Compressor Activation
V2	Manual Valve for Condenser #1 Activation
V3	Manual Valve for Condenser #2 Activation
V4	Manual Valve for Condenser #3 Activation
V5	Manual Valve for Condenser #4 Activation
V6	Manual Valve for Expansion Valve #1 Activation
V7	Manual Valve for Expansion Valve #2 Activation
V8	Manual Valve for Gas Flow Meter Replacement
V9	Manual Valve for By-Passing Gas Flow Meter
V10	Manual Valve for Gas Flow Meter Replacement
V11	Manual Valve for Cancelling hot gas by pass valve

In Figure 21 “PT” represents the pressure transmitter while “TC” represents thermocouples. The thermocouples are mounted on the surface of the copper fins. The RTDs are located to the heat exchangers with a special union to have a direct flow read. The detailed RTD locations could be seen in Figure 22.

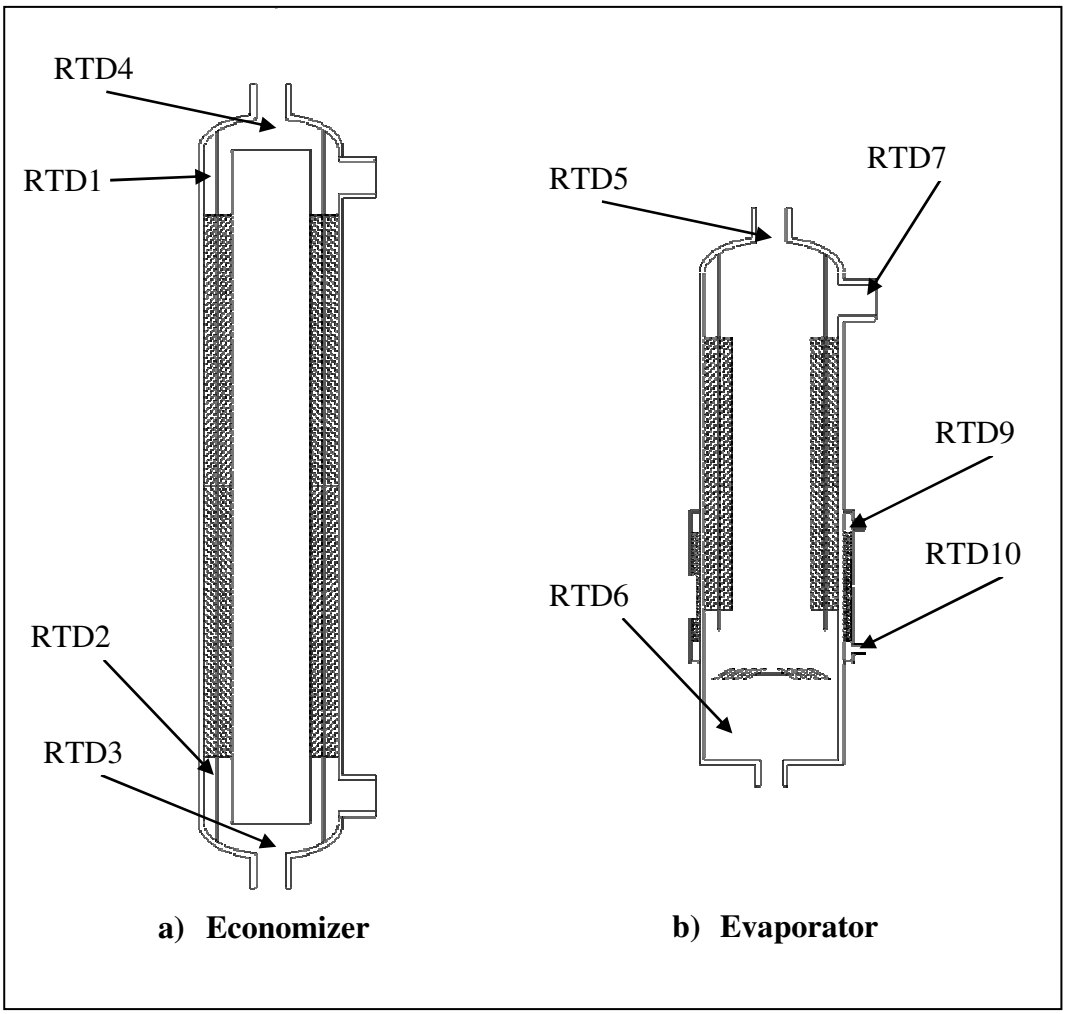

Figure 22. RTD locations on the heat exchangers

In the beginning, two reciprocating type compressor were used. After some time, the smaller capacity compressor was broken down; therefore the compressor was replaced with scroll type compressor. The compressors are placed at the bottom shelf of the frame since they are the heavy and in the future, they may be replaced by different types and capacities of compressors. The compressors are selected for

operation by a manual valve called “V1” in the diagram. Whenever the second compressor is desired to be used, V1 valve should be opened and from panel board, Compressor 1 (Piston Compressor) should be shut down and Compressor 2 (Scroll Compressor) should be switched on.

The picture of the panel board could be seen in Figure 23. The buttons from left to right represents “Piston Compressor”, “Scroll Compressor”, “1st Condenser’s Fan Motor”, “2nd Condenser’s Fan Motor”, “3rd Condenser’s Fan Motor” and “4th Condenser’s Fan Motor” respectively. The buttons are lighted up when they are switched on.

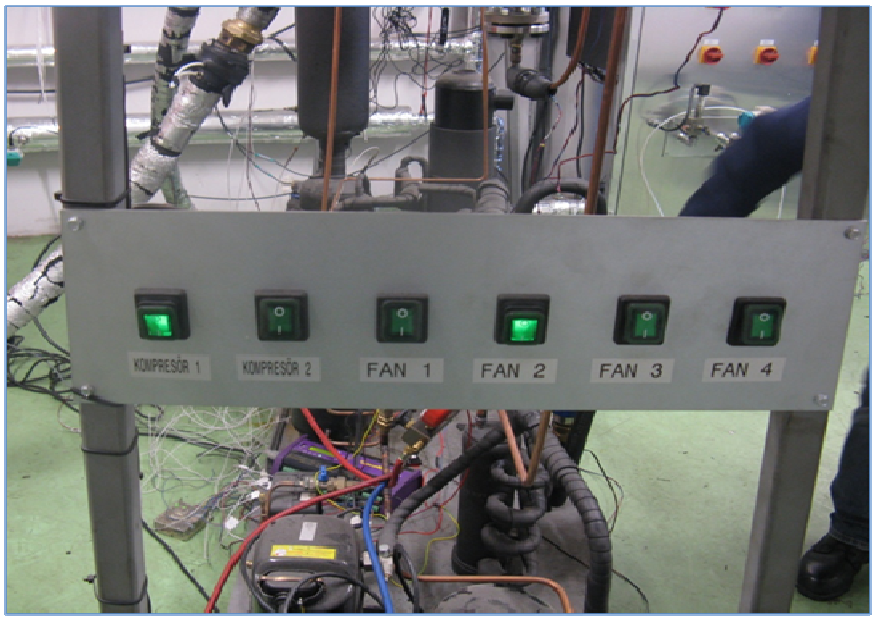

Figure 23. Panel board

Four condenser units are placed at the top shelf. The reason of placing condensers at the top shelf is to provide continuous and stable air flow. Additionally, heated air could affect the refrigeration cycle components, but with this application it is avoided. Switching to the desired condenser for operation is provided by using

relevant manual valves (V2, V3, V4 or V5) and the buttons called “Fan 1”, “Fan 2”, “Fan 3” and “Fan 4” in panel board.

A liquid receiver is placed after the condensers. The purpose of the receiver is to provide saturated liquid to expansion valve’s inlet. On the other hand, avoiding defrost and liquid flow to the compressor inlet while compressor is shut down could be the receiver’s second task. There is also an accumulator placed after the evaporators. There are two identical expansion valves used in experimental setup for both evaporators.

There are ten RTDs, eleven thermocouples, five pressure transmitters and two mass flow meters/rotameters placed on the experimental setup to have all necessary measurements. Apart from mass flow meters, all measurement components are connected to data logger. The picture of the completed experimental setup could be seen in Figure 24 and Figure 25.

Figure 24. Frontal view of the experimental setup

Figure 25. Rear view of the experimental setup

2.2.Leakage Test

After installing all components, the experimental setup should be submitted to a leakage test. There are several ways to determine the leakage in the experimental setup pipes and components. During the preparation of the experimental setup, several leakage tests were carried out. Initially, after assembling all components, a bubble test is performed with high pressure Nitrogen purged to the refrigeration cycle. The special liquid, which results in bubbles if there is any leakage, is used to find major leaks.

After bubble testing, a mass spectrometer is used for minor leaks. The mass spectrometer is basically sniffs the air in control volume, takes samples and searches for identified/selected element. The main advantage of this equipment is

searching the leakage in atomic scale which gives higher precision in sealed cycles. The picture of the mass spectrometer is seen in Figure 21.

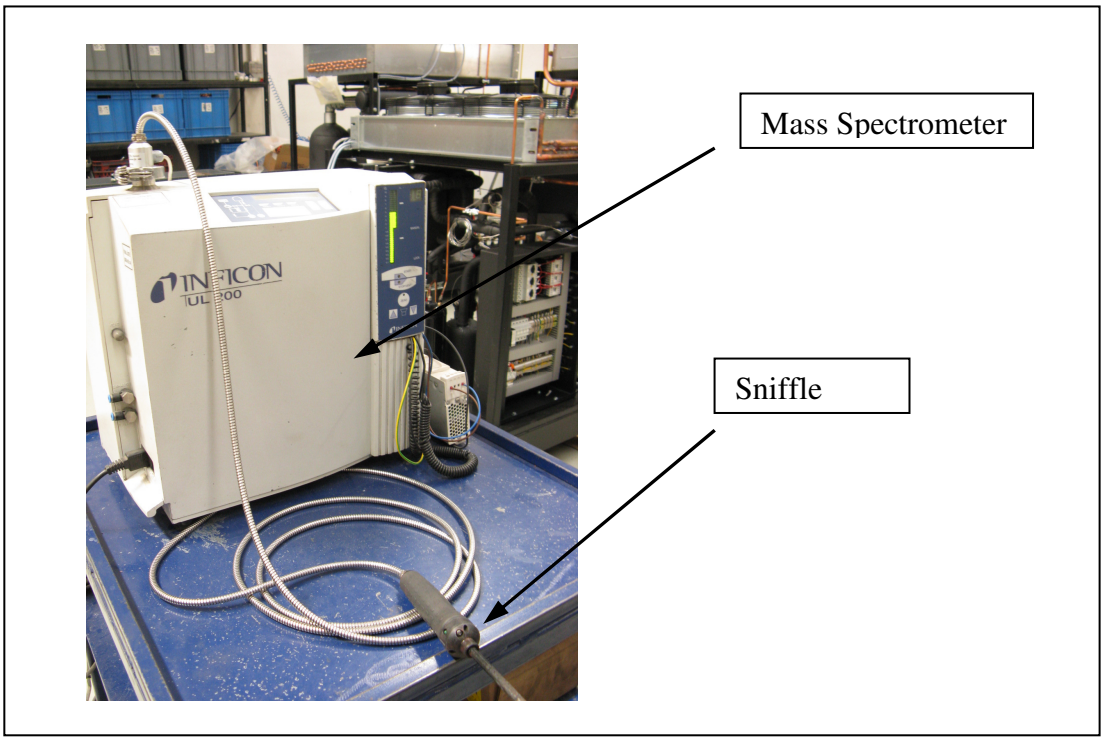

Figure 26. Mass Spectrometer (Helium Detector) with its sensor/sniffle

Since Helium has the smallest atomic size, leakage tests are carried out by using this element. If there is no leakage of Helium from the cycle, other gases (such as R134a in our case) cannot leave the cycle, and then the cycle is assumed to be sealed. There is an average amount of Helium stored in ambient air, in spectrometer, that value is corrected before the test. The sensor/sniffle of the spectrometer is opened to air and let it have the sample. The amount of Helium is recorded and set to be as the default value.

After initializing the mass spectrometer, the refrigeration cycle is filled with 1 barg of Helium and 24 barg amount of Nitrogen. Since Helium is relatively expensive gas, an inert gas is used to help to fill the refrigeration cycle. Nitrogen

is selected for this purpose because of its non-active/inert behavior, containing no moisture, and cost-effective price. After filling the refrigeration cycle with Helium and Nitrogen, the mass spectrometer is started to have samples from connections, joints and all components that are in use.

Since there are a lot of connections and joints in experimental setup, it is difficult to keep leakage under desired states ($6x10^{-2}$ PPM). Specifically the problematic parts like screwed joints such as elbows, manual valves and "Tee" junctions for pressure transmitters and manometers. Following pictures in Figure 27 show how sniffle and spectrometer acts under a leakage at the joints.

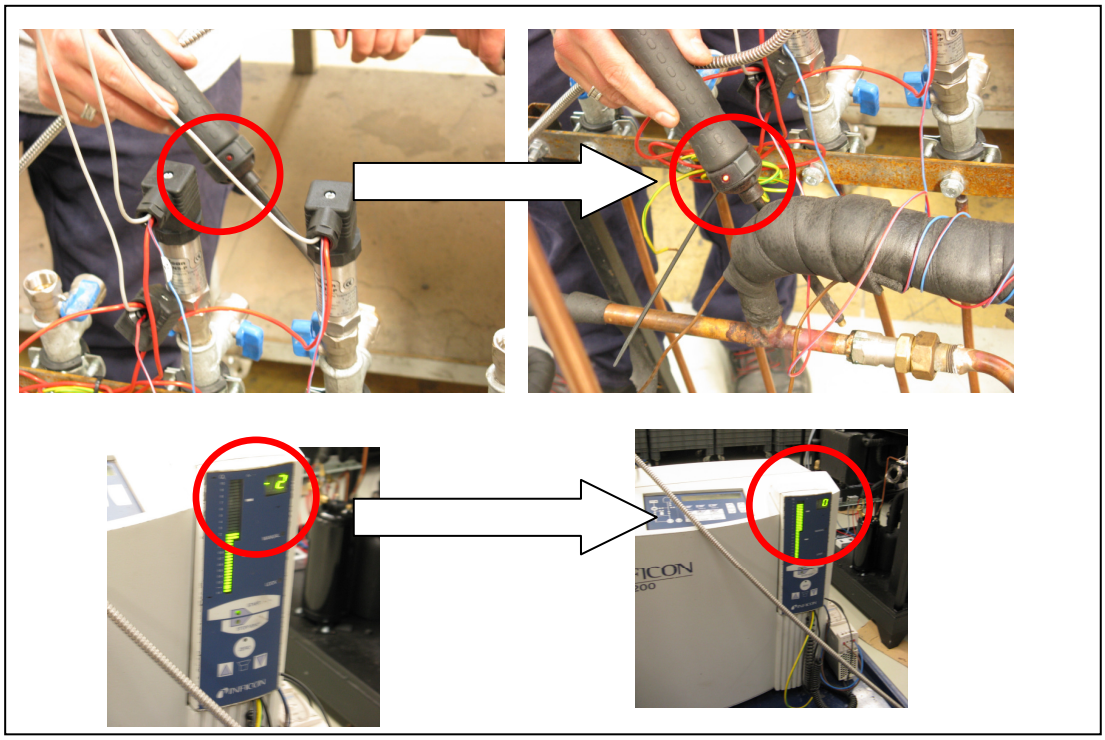

Figure 27. The leakage test with Helium detector

The most difficult part of this procedure is the moment when a leakage of He occurs. The sniffle senses the leak, after sensing high leakage joints, He and Nitrogen are emptied from the refrigeration cycle and rejected gas is displaced to

the roof of the building. The reason behind this is to keep away high amount of He around the refrigeration cycle to redo the leakage test accurately after fixing. All these procedures should be followed until there is no leak detected by the mass spectrometer, which results in spending lots of time.

After leakage test, the refrigeration cycle is vacuumed before filling the refrigerant gas R134a. The main reason behind this is to keep air, moisture and other gases away from the refrigeration cycle since they result in fouling effects and reducing heat transfer rates. The refrigeration cycle is vacuumed to about 100 mbars which is acceptable for filling the refrigeration cycle with R134a. The procedure of leakage tests are given in Figure 28.

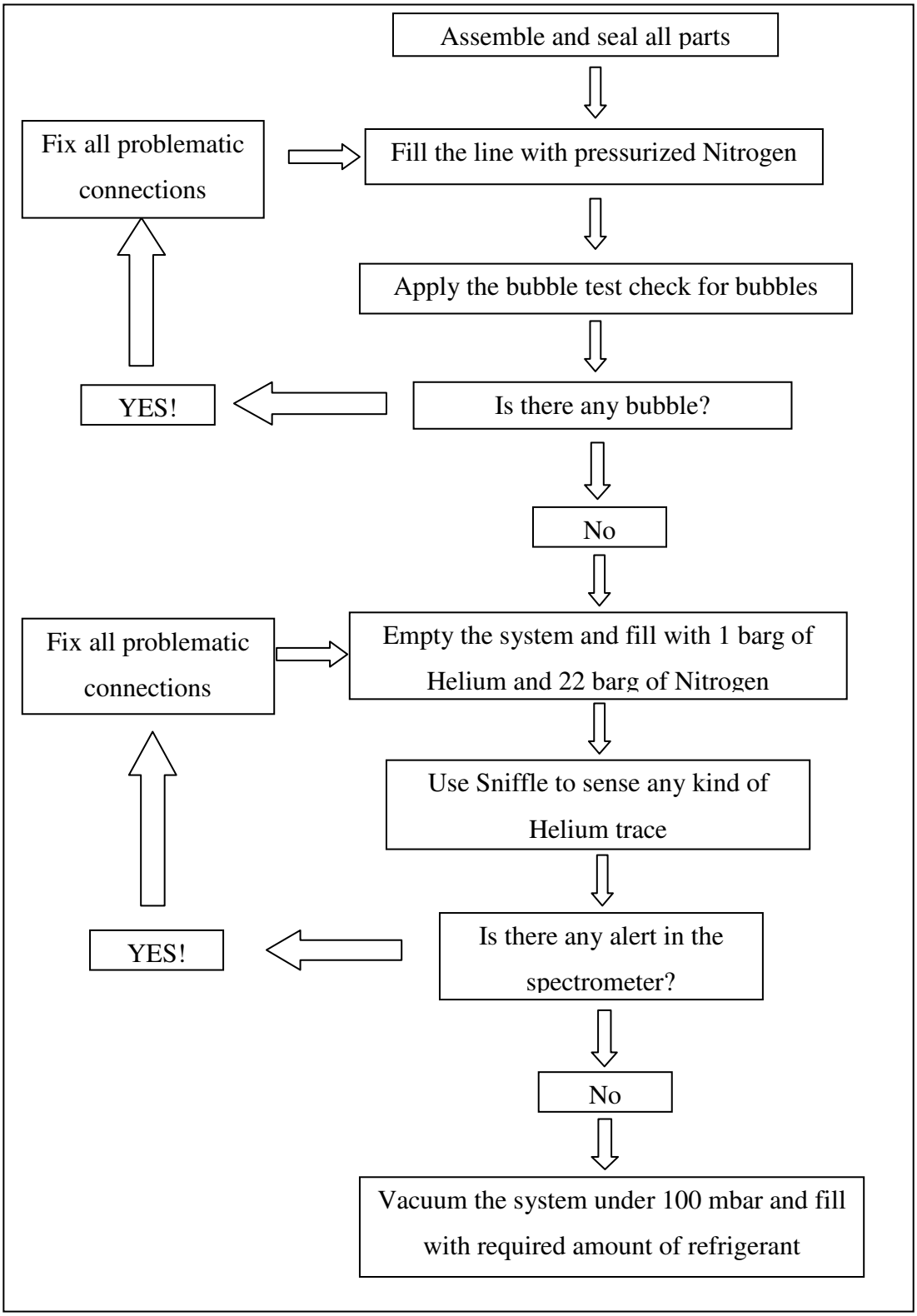

Figure 28. Procedure of leakage test

2.3. Calibration of The Components

The calibration of RTDs has been carried out by using a reference RTD which was calibrated previously in the company. Both the calibrated/reference RTD and non-calibrated RTD were put in a bath and several temperature measurements have been carried out. The data logger recognizes most of the measurement components (including RTDs, thermocouples etc.). Each RTD probes were confirmed that the temperature readings were similar to calibrated thermocouple and RTD.

The pressure transmitter's calibration was carried out by using a calibrated manometer and pressure vessel (Liquid Receiver tank in the refrigeration cycle). The transmitters are 4-20 mA ranged and used in between 0-20 bars. They are both connected to the same fitting. The pressure of the vessel is increased by heating and the measurements were taken simultaneously. The results and the graph could be seen in Appendix C. The curve fitting method is used and following relation is obtained to convert data in mA to barg.

$$P(\text{barg}) = 1.224 \times (\text{mA}) + 4.967 \qquad (1)$$

To read directly, the formulation given above is applied on relevant channels on the software of the data logger.

CHAPTER 3

ANALYSIS OF THE REFRIGERATION CYCLE

In this section, the analysis of the refrigeration cycle in terms of COP, 2nd Law Efficiency and Irreversibility are explained briefly. A Matlab® code is written to carry out relevant calculations and to obtain thermodynamic properties of relevant states that are measured. The properties of the fluids are taken from highly reliable and commonly used software Refprop V8 ® [16]. Additionally its function that is written for Matlab called "refpropm" is used to have all properties on the code directly. The use and more information are mentioned in the code that is also added to Appendix B of this thesis.

The experiments are carried out by using the compressed air that has all the time 100% relative humidity on purpose. To ensure that, at the inlet there is a water spray, which sprays excessive amount of water to the flow and a liquid water separator to separate the liquid droplets before the experimental setup. A calibrated dew point meter is placed at the outlet. The amount of water at the inlet of the dryer is found by help of the inlet temperature with 100% relative humidity while it is found by dew point meter placed at the outlet. The details of the experimenting section could be found in Appendix G.

The mass flow rate of the compressed air ($\dot{m}_{air}$) could be obtained by help of measured volumetric flow rate of the compressed air ($\dot{V}_{air}$) in Nm³/h. The density of air under normal/standard conditions of air is taken with help of Refprop

software for air at 15°C of temperature and 101.325 kPa of pressure according to ISO standards [17].

$$\dot{m}_{air} = \rho \, \dot{V}_{air} \tag{2}$$

The inlet and outlet conditions of the compressed air that flows through the dryer such as the pressure, temperature and relative humidity values are measured and fixed. The amount of water at the inlet of the dryer ($\dot{m}_{water-inlet}$) could be found with help of the relative humidity in the compressed air. During experiments, the relative humidity at the inlet of the dryer ($\emptyset_{inlet}$) is kept always at 100%. The inlet temperature in all experiments is about 35°C and the compressed air pressure is 7 barg. Therefore the partial pressure ($P_{water-inlet}$) and enthalpy ($h_{water-inlet}$) of the water at the inlet could be found at relevant temperature by help of Thermodynamic Tables for saturated water at measured temperature. The specific humidity at the inlet (w_{inlet}) could be found by following formula [5; 18]

$$w = \frac{0.622}{\frac{P}{\emptyset \, P_{water}} - 1} \tag{3}$$

The mass flow rate of water at the inlet could be obtained by multiplying the specific humidity at the inlet with the total mass flow rate of the compressed air.

$$\dot{m}_{water-inlet} = \dot{m}_{air} \; w_{inlet} \tag{4}$$

By applying the same procedures, since the outlet humidity and temperature is known, the mass flow rate ($\dot{m}_{water-outlet}$) and the enthalpy ($h_{water-inlet}$) of the water at the outlet state could be obtained.

Applying the 1st Law of Thermodynamics, total heat that is transferred from the compressed air ($\dot{Q}_{Total}$) could be obtained. Figure 29 illustrates the energy balance on the economizer and evaporator. In here extracted water is at dew point temperature that is measured (mostly observed around 3°C in experiments) and the mass flow rate ($\dot{m}_{water-condense}$) could be obtained by applying the conservation of mass.

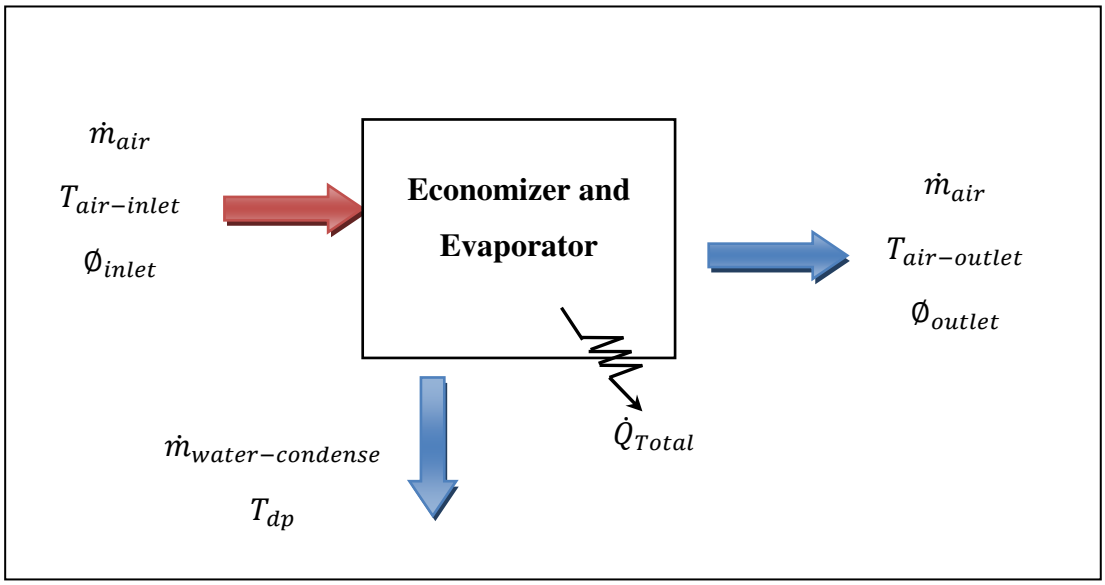

Figure 29. The schematic illustration of the economizer and evaporator as a control volume

$$\dot{m}_{water-condense} = \dot{m}_{water-inlet} - \dot{m}_{water-outlet} \tag{5}$$

$$\dot{Q}_{Total} = \dot{m}_{air}(h_{air-inlet} - h_{air-outlet}) + \dot{m}_{water-condense}\left(h_{water-inlet} - h_{water-dp}\right) \tag{6}$$

It is difficult to determine the liquid flow rate after the condenser. During condensation, the fan is on and off and condensed refrigerant flows unsteadily the liquid receiver. Not only the mass flow meters but also the hot gas by pass valve (HGBPV) avoided the measurements on the liquid flow meter since all the time

the valve injects hot gas to the inlet of the compressor. The solution is relying on the compressed air flow rate which is read and controlled well. The compressor was experimented by disabling HGBPV manually to determine the electrical convergence efficiency that is converted to work. The cycle was connected to data logger to read voltage and amp values precisely. Mass flow rate of the refrigerant at the compressor inlet was stable and 0.02 kg/s. The average current value was all the time close to I = 6.3 amps while the voltage was stable and about Vo=224V and by help of temperature and pressure readings at the inlet and outlet with mass flow rate reading (at the inlet of the compressor), the convergence efficiency was found η_{ee}= 71%. The code is given in Appendix B. Figure 30 illustrates the schematic view of the process.

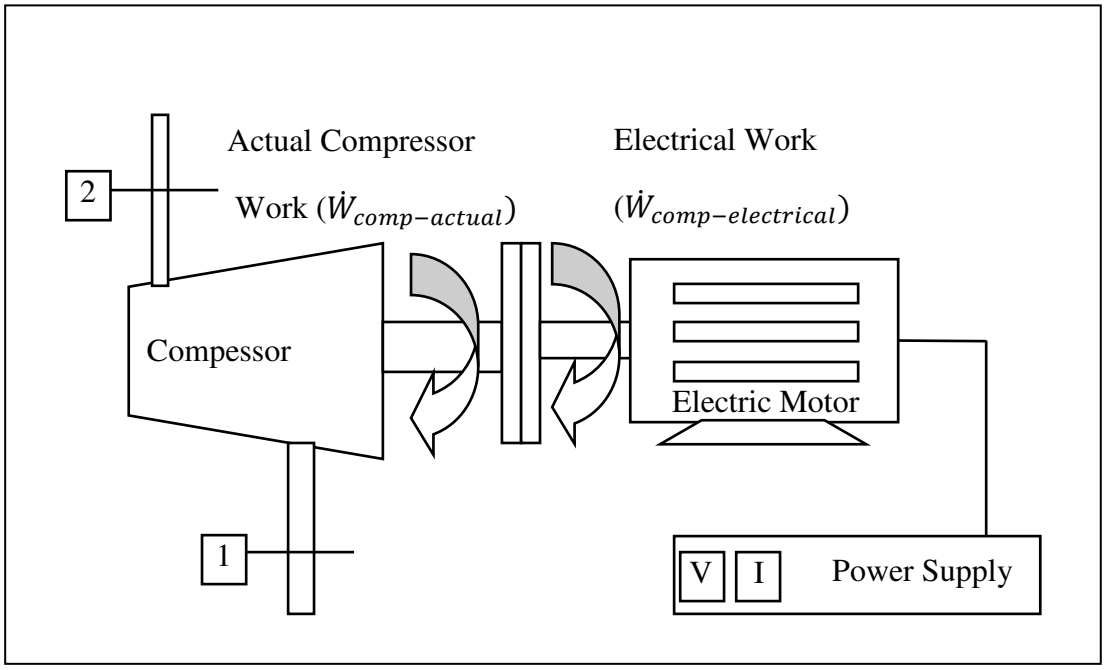

Figure 30. Schematic view of electrical and actual work of the compressor

In such case, the total power of electrical motor that is used to turn the compressor shaft could be obtained by:

$$\dot{W}_{comp-electrical} = V\ I \tag{7}$$

The electrical motor will rotate the shaft of the piston compressor with some losses (friction loss etc.). The electrical convergence efficiency of the compressor package could be obtained by the mass flow rate that is read from the gas flow meter, the enthalpy values of inlet (h_1) and outlet (h_2) state of the compressor are used. The schematic of these states could be found in Figure 3.

$$\eta_e = \frac{\dot{m}_{ref-comp}(h_2 - h_1)}{\dot{W}_{comp-electrical}} \tag{8}$$

The mass flow rate of the refrigerant that flows through the heat exchanger, condenser and expansion valve can be found by Equation (9)

$$\dot{m}_{ref-evap} = \frac{\dot{Q}_{total}}{(h_{1`} - h_4)} \tag{9}$$

Where Q_{total} is total heat that is transferred during drying process was found before, h_{1e} is the enthalpy of the refrigerant at the outlet of the evaporator and h_4 is the enthalpy of the refrigerant at the outlet of the expansion valve so the inlet of the evaporator. This mass flow rate can be provided alone with a variable speed compressor instead of using HGBPV to create artificial load on the refrigeration cycle or by-passing the actual compressor. Therefore, the exact required compressor, or if there was a variable speed compressor, the work can be calculated with Equation (10)

$$\dot{W}_{comp-exact} = \dot{m}_{ref-evap}(h_2 - h_1) \tag{10}$$

The heat that is rejected from condenser to the ambient can be found by Equation (11).

$$\dot{Q}_{cond} = \dot{m}_{ref-evap}(h_3 - h_{2`}) \tag{11}$$

The heat transfer from the pipe which connects compressor to the condenser can be calculated by Equation (12).

$$\dot{Q}_{comp-cond} = \dot{m}_{ref-evap}(h_{2`} - h_2) \tag{12}$$

The refrigerant mass flow rate circulating inside the evaporator is found by help of the total heat transfer. The total mass flow rate, including HGBPV action, is found by help of compressor efficiency and power with inlet and outlet conditions. Subtracting the mass flow rate of the compressor (total mass flow rate) from the mass flow rate that is flowing through the evaporator resulted in the amount of mass that the HGBPV is injected to the inlet of the compressor.

$$\dot{m}_{ref-HGBPV} = \dot{m}_{ref-evap} - \dot{m}_{ref-comp} \tag{13}$$

It is expected that the effect of the mass flow rate difference would affect not only the refrigeration cycle capacity but also the Coefficient of Performance (COP).

The formulation of coefficient of performance (COP) is given in Equation (14).

$$COP = \left|\frac{\dot{Q}_{total}}{\dot{W}_{comp}}\right| \tag{14}$$

In this study, there are two COP values are found for both compressor work with HGBPV and variable compressor. For actual case, $W_{comp\text{-}actual}$ is used while $W_{comp\text{-}exact}$ is used for variable speed/required case. Isentropic compression efficiency can be found by Equation (15). The schematic view could be found in Figure 3.

$$\eta_s = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{15}$$

The irreversibility can be found by using the entropy generation (S_{gen}) by multiplying by reference temperature (T_0) which is considered the reference as ambient temperature at that moment (T_{amb}). The irreversibility values are calculated by using Equations (16, 17, 18, 19, 20) for each component and total irreversibility is found by Equation (21).

$$Compressor: \quad I_{Comp} = T_0\left(\dot{m}_{ref}(s_2 - s_1) - \cancelto{0 \text{ (Adiabatic Compression)}}{\frac{\dot{Q}}{T_0}}\right) \tag{16}$$

$$Condenser: \quad I_{Cond} = T_0\left(\dot{m}_{ref-evap}(s_3 - s_{2c}) - \frac{\dot{Q}_{Cond}}{T_0}\right) \tag{17}$$

$$Expansion\ Valve: \quad I_{Exp} = T_0(\dot{m}_{ref-evap}(s_4 - s_3) - \cancelto{0 \text{ (No heat transfer)}}{\frac{\dot{Q}}{T_0}}) \tag{18}$$

$$Evaporator: \quad I_{Evap} = T_0 \left(\dot{m}_{ref-evap}(s_{1e} - s_4) - \frac{\dot{Q}_{Total}}{T_{Evap}} \right) \tag{19}$$

$The\ Pipe\ Between\ Compressor\ and\ Condenser$:

$$I_{Comp-Cond} = T_0 \left(\dot{m}_{ref-evap}(s_2 - s_{2c}) - \frac{\dot{Q}_{Comp-Cond}}{T_0} \right) \tag{20}$$

$Total\ Irreversibility$:

$$I_{Total} = I_{Comp} + I_{Cond} + I_{Exp} + I_{Evap} + I_{Comp-Cond} \tag{21}$$

Two irreversibility values are calculated for both actual case with HGBPV and variable speed compressor without HGBPV case.

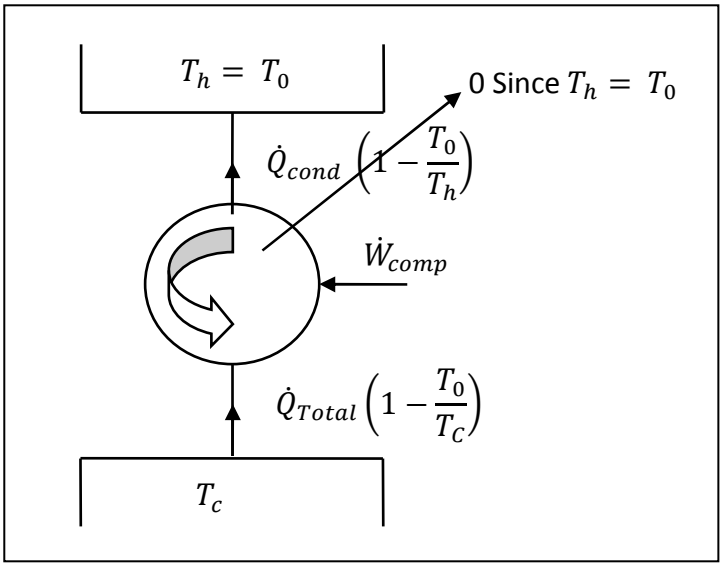

Figure 31. Schematic view of Exergy Transfer for the refrigeration cycle

The 2nd Law Efficiency of the cycle could be found by Equation (22) [18].

$$\eta_{II} = \frac{Desired\ result\ in\ exergy}{Destructed\ source\ in\ exergy} = \frac{\dot{Q}_{Total}\left(1 - \frac{T_0}{T_C}\right)}{\dot{W}_{comp}} \tag{22}$$

T_h and T_C represents hot and cold reservoir temperatures respectively. Hot reservoir is the ambient temperature (T_{amb}) and cold reservoir is evaporator and the average of evaporator inlet and outlet temperature of the refrigerant ((T_4+T_{1e})/2) is assumed to be cold reservoir temperature. Reference temperature is to be $T_0 = T_{amb} = T_h$. Therefore the exergy accompanying heat transfer to hot reservoir is zero. Work of the compressor is considered for both variable speed ($\dot{W}_{comp-exact}$) and actual compressor ($\dot{W}_{comp-actual}$) cases, therefore two 2nd law efficiencies are found.

CHAPTER 4

EXPERIMENTS

After completing the construction of the experimental setup and connecting all measurement devices to the data logger, the experiments were performed.

Since the main goal of this study is to build an experimental setup for a refrigerant compressed air dryer and experiment the setup for various load conditions, several experiments have been carried out and logged simultaneously. In the air dryers, most of the time, the air usage varies depending on the application. Selection of components and design is based on maximum air usage per day. Therefore, the dryers do not work under design conditions. During the experiments, different air flow rates used to determine the behavior of the refrigeration cycle. The cycle is experimented at 250 Nm^3/h, 400 Nm^3/h and 500 Nm^3/h. The letter "N" represents air at normal conditions which is defined by DIN ISO standards and these conditions are at 101.325 kPa pressure and 15°C temperature. [17]

4.1. Experimental Conditions

The experiments are carried out for various conditions. The main parameters that are considered are ambient air temperature, flow rate of air and its inlet temperature. Depending on these terms, the evaporator load varies. Operating above or below the design conditions results in unsteady behavior like frost in evaporator (if hot gas by pass valve is not active), liquid shock at the compressor

or increase in inlet and outlet temperature of compressor which forces the refrigeration cycle work in nearly always in superheat which is not desired. On the other hand ambient temperature affects condenser outlet temperature which also causes chain effect on the cycle to work in super heat zone.

Compressed air volume flow rate adjustment was carried out manually in the test room. The refrigeration cycle was kept working for 15 minutes to reach steady operating conditions. On the other hand, data logger was online and recording all the time from beginning of the experiments to record the unsteady and steady behavior. The actions that had been taken were recorded to the computer (i.e. at what time the flow rate changed etc.) and a notebook.

Experiments were carried out with the second condenser, piston compressor and the evaporator with economizer separated. After selecting configuration related valves should be opened, the rest should be kept closed. Since during start-up operation, there is no load on evaporator, the hot gas by-pass valve should be opened in order to avoid defrosting and liquid flow to compressor suction.

Selected compressor and condenser buttons should be switched on to start refrigeration cycle. If hot gas by pass valve is opened, the values of flow meters could be different from one to another.

After reaching steady operation, by pass valve could be kept opened if the air side load is lower than design conditions otherwise should be closed by relevant valves. Deciding the amount that flow through by – pass valve is done by checking the low pressure zone. It is desired to be kept about 2 barg (3 bara) according to the saturation point of R134a (~0.7°C) which is above defrost point of water. The amount is adjusted by a screw which is integrated on the hot gas by

pass valve. Tightening the screw (turning the screw in clock-wise) results in increase the amount of hot gas flow to the low pressure zone while loosening does the opposite. The picture of the adjustment screw is shown below in Figure 32.

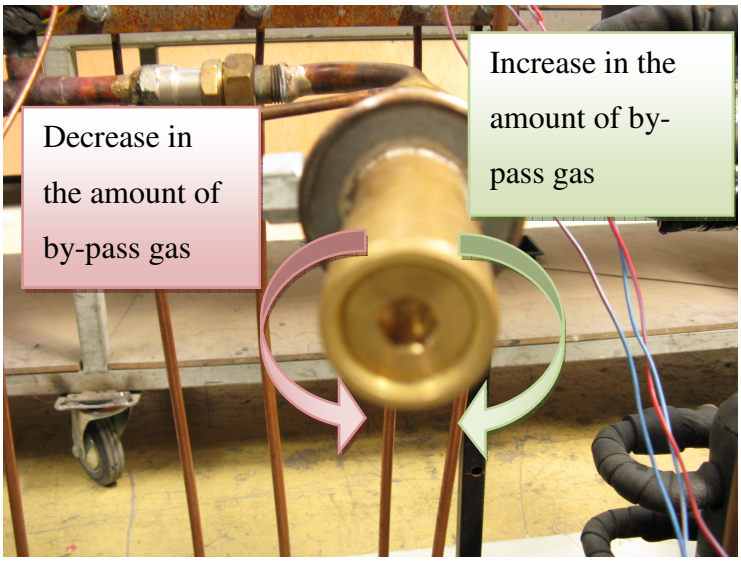

Figure 32. Hot gas by-pass valve operation

Adjustment of the compressed air flow is carried out manually by using a globe valve in the test room. The flow rate could be read from the monitor placed on the test room electric and control panel and adjusted to desired value. The inlet temperature of compressed air is controlled by using the electrical heater and water cooler placed on the compressed air supply pipe line. The valves and temperature controls of compressed air supply are given in Appendix G.

Before supplying the compressed air to the dryer, it should be confirmed that data logging is on progress. After checking the logging progress, the synchronized camera should record the flow rates before supplying air to the experimental

setup. After all these preparation the experiments could be carried out at desired condition.

4.2. Experiments

4.2.1. Start – Up Experiments

Before supplying the compressed air to drying process, it is necessary to wait until the experimental setup reaches its operating temperature (~3 °C at evaporator). Figure 33 shows how the refrigeration cycle behaves during start – up.

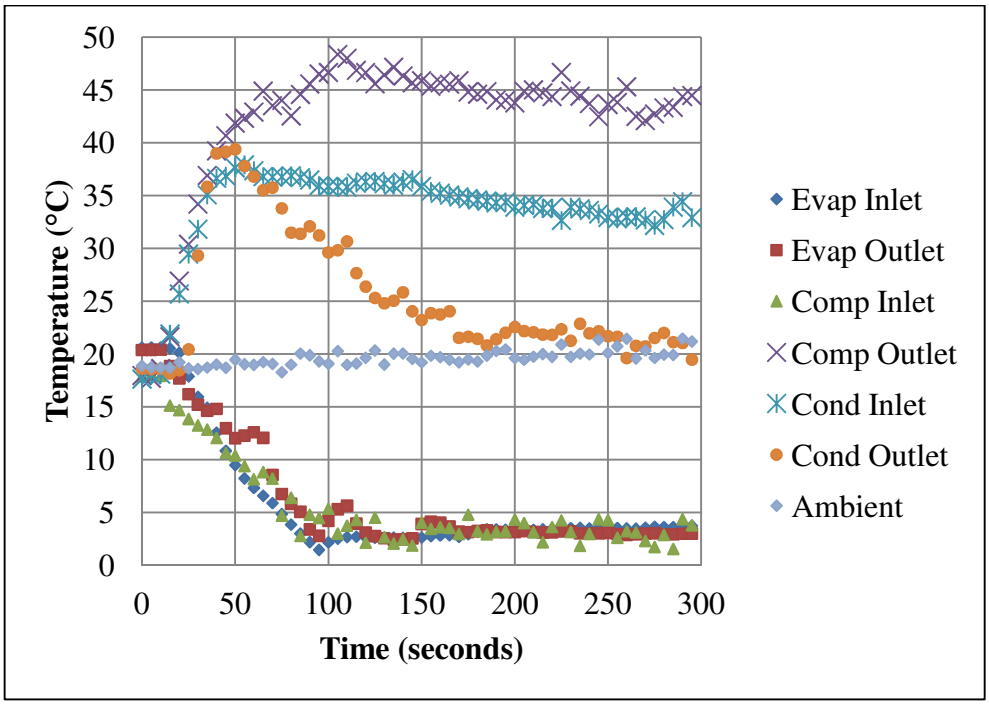

Figure 33. Initial (start-up) temperature measurements

In Figure 33 there is about 10°C drop between compressor outlet temperature and condenser inlet temperature because the pipe is not insulated. Condenser outlet temperature has the slowest behavior to reach steady state operation. The main

reason behind this could be the phase change, and by checking Figure 34, the pressure went up to 13 barg while the fans are starting at 12 barg. This means that the fan late response may cause this delay.

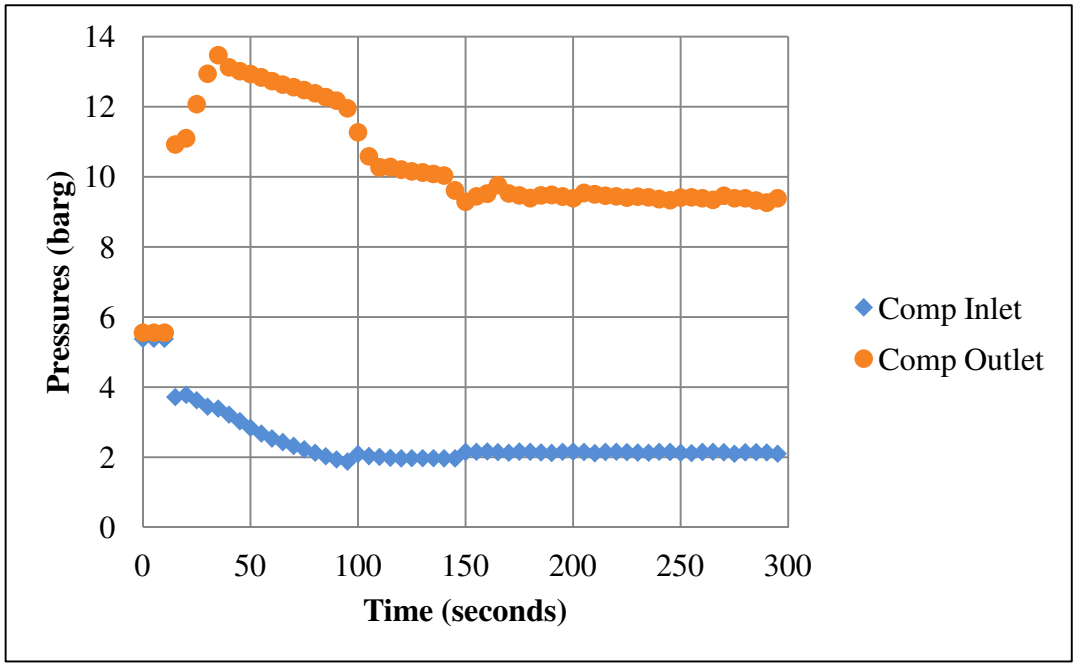

Figure 34. Initial (start-up) measurements of compressor inlet and outlet pressures

During start up experiments, the components reach to a stable point in various time depending on the component. As long as the concern is the temperature, the components reach their steady state no more than 120 seconds apart from the condenser outlet which is controlled by start-stop fan switches. Browne and Bansal stated that the compressor reaches its steady point faster than other components since the compressor reaches to its nominal operation speed very rapidly, steady state operation. During modeling, it is assumed that the dynamics of the compressor are neglected and assumed to be operating always at steady state [6]. This argument is observed during the experiments as well.

4.2.2. Steady State Experiments

It is difficult to adjust hot gas by pass valve for lower loads in evaporator which triggers unsteady operation. Even with hot gas by-pass valve, after some time, the refrigeration cycle operates at steady state. This section deals with steady state operations and results. The experiments are tried to carry out under similar conditions. The inlet compressed air is always kept at fixed condition at an inlet temperature of 35°C by using the heater and cooler, at all the time relative humidity of 100% by using water spray injection and liquid water separator and at a pressure of 7 barg with pressure regulator.

4.2.2.1. Experiments at 250 Nm^3/h Compressed Air Flow Rate

The condensers are usually selected oversized for operating at extreme conditions such as higher ambient temperature or the larger evaporation loads. For this reason, there is one fan switch placed on the experimental setup which allows the fan to work only necessary span. The parameter is the pressure in our experimental setup and switch is adjusted for 12 – 9 bars. When the pressure rises above 12 barg the switch sends signal to the panel to activate the fan, and whenever the pressure drops below 9 barg the pressure switch turns the fan off. This happens when the load (amount or inlet temperature of compressed air) is lower than the design or lower ambient temperatures.

For the flow rate around 250 Nm^3/h, which is lower than the design load 400 Nm^3/h, the experiment was carried out for 3 hours and 40 minutes. (~2700 x 18 data). The reason for such a long period for the experiment is to understand the experimental setup and all components are working very well, all fine adjustments have been carried out. After ensuring the refrigeration cycle is working smoothly and steady, all data is plotted on Microsoft Excel® to see how the amount of data could be reduced. Figure 35 illustrates the behavior of the main components clearly.

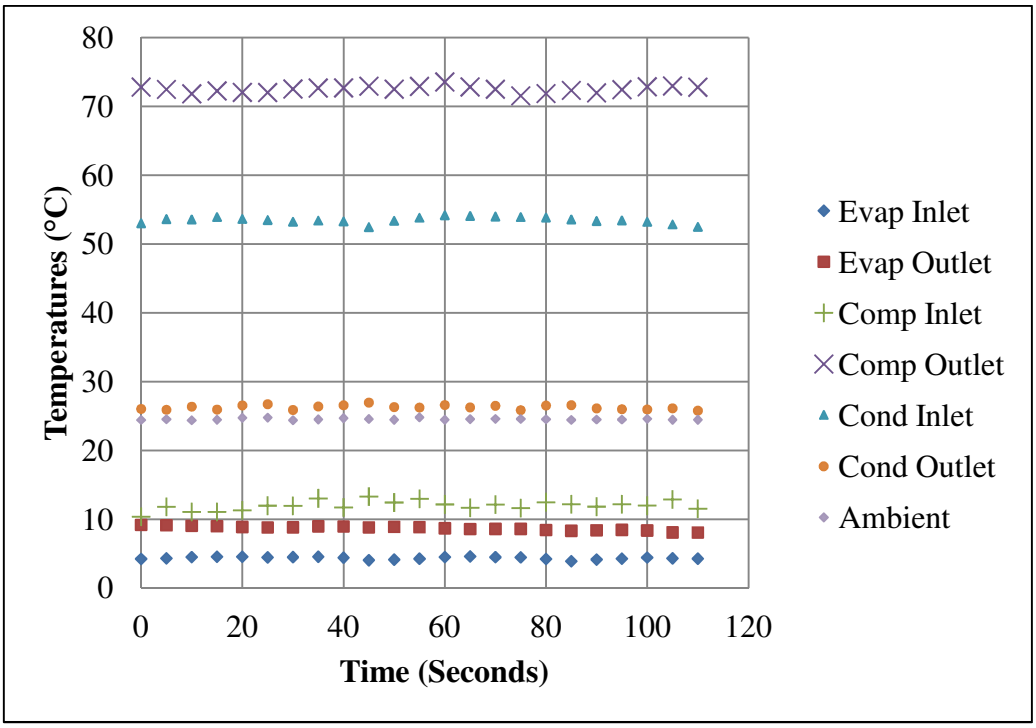

Figure 35. Temperature measurements for 250 Nm³/h air flow rate experiments

In Figure 35, the temperature profiles are to be stable. The suction temperature of the compressor is about 10-12°C which guaranties that there is no liquid flows through. The temperature difference between compressor outlet and condenser inlet is about 15°C because of the heat loss to the surroundings. Addition to that, because of the physics of evaporation, condensation and the effect of the hot gas by-pass on the refrigeration cycle creates a smooth oscillation in temperature profiles.

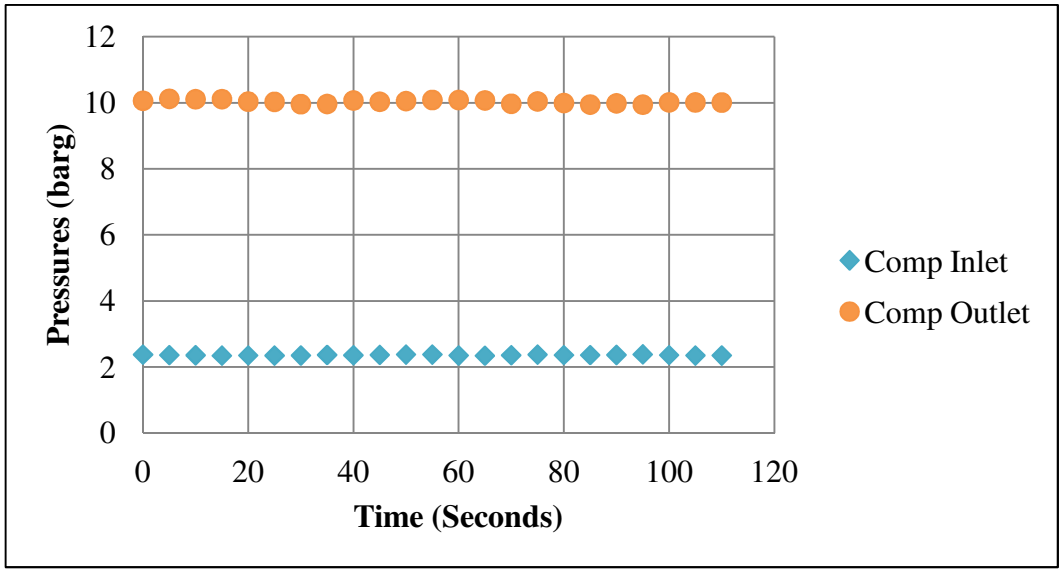

Figure 36. Pressure measurements for 250 Nm³/h air flow rate

The profile of the pressure values versus time in Figure 36 indicates that the cycle is working smoothly and in steady operation.

4.2.2.2. Experiments at 400 Nm³/h Compressed Air Flow Rate

The fan motor never stopped during the experiment since the discharge pressure of compressor is about 12 barg and switch shuts the fan motor when the pressure drops down to 9 barg. The experiment duration was about 3 hours with approximately 2300 x 18 data. After reaching steady state operation, all data was plotted on Microsoft Excel® to see how the amount of data could be reduced. The cycle worked smoothly and unlike low capacity experiments, there was no periodic/cyclic behavior since the fan motor was always on. However, the experiment shows that when the results are magnified, the condensation and evaporation have a smooth wavy behavior. This wavy behavior results in an oscillation and could be seen clearly while checking the pressures rather than the temperatures. The period seems to be ~50 seconds. The pressure oscillation for

discharge pressure is maximum ±0.2 bars while for suction pressure maximum ±0.05 bars. Figure 37 shows the behavior of the main components for design load conditions clearly.

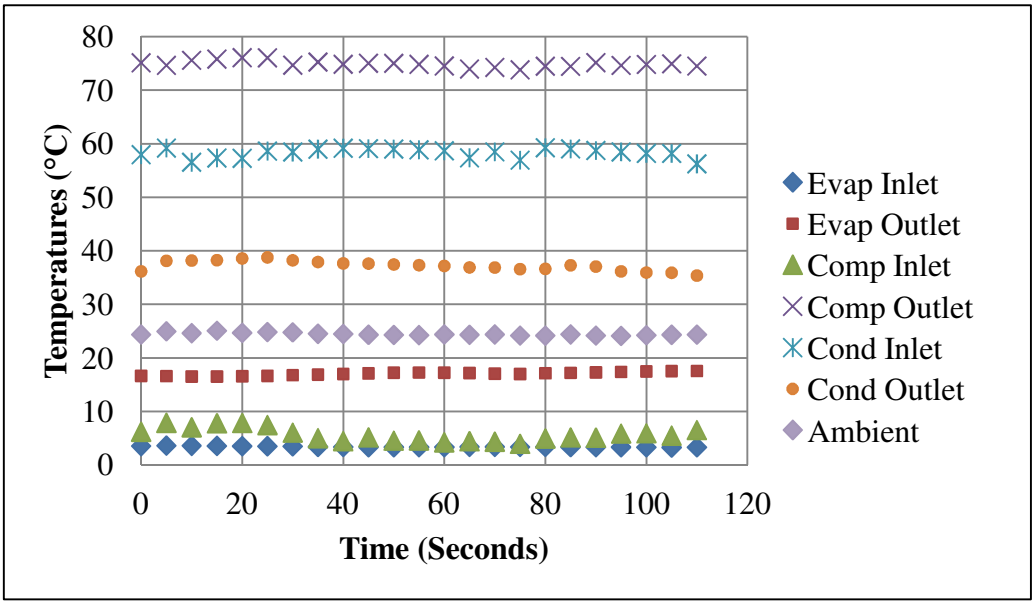

Figure 37. Temperature measurements for 400 Nm³/h air flow rate

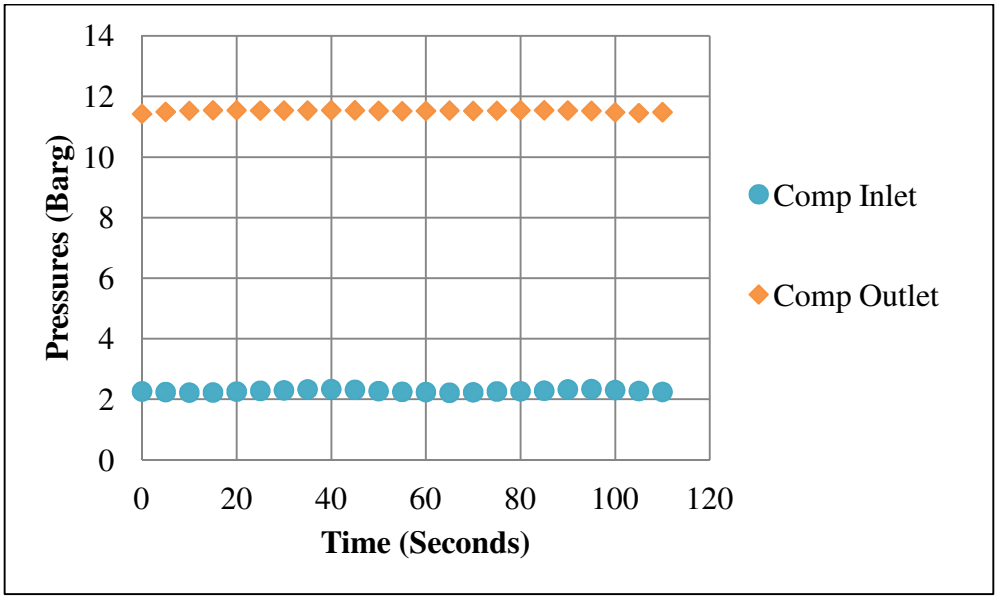

Figure 38. Pressure measurements for 400 Nm³/h air flow rate

4.2.2.3. Experiments at 500 Nm³/h Compressed Air Flow Rate

This time the experimental setup was used for 500 Nm³/h air flow. The condenser fan was working all the time during this experiment. After reaching steady state conditions the data was logged and sorted from ~400 x 18 data. Still the oscillation behavior of evaporation and condensation seemed on the graph that was plotted for overall data. By magnifying the data, the oscillation could be observed in Figure 39 and Figure 40.

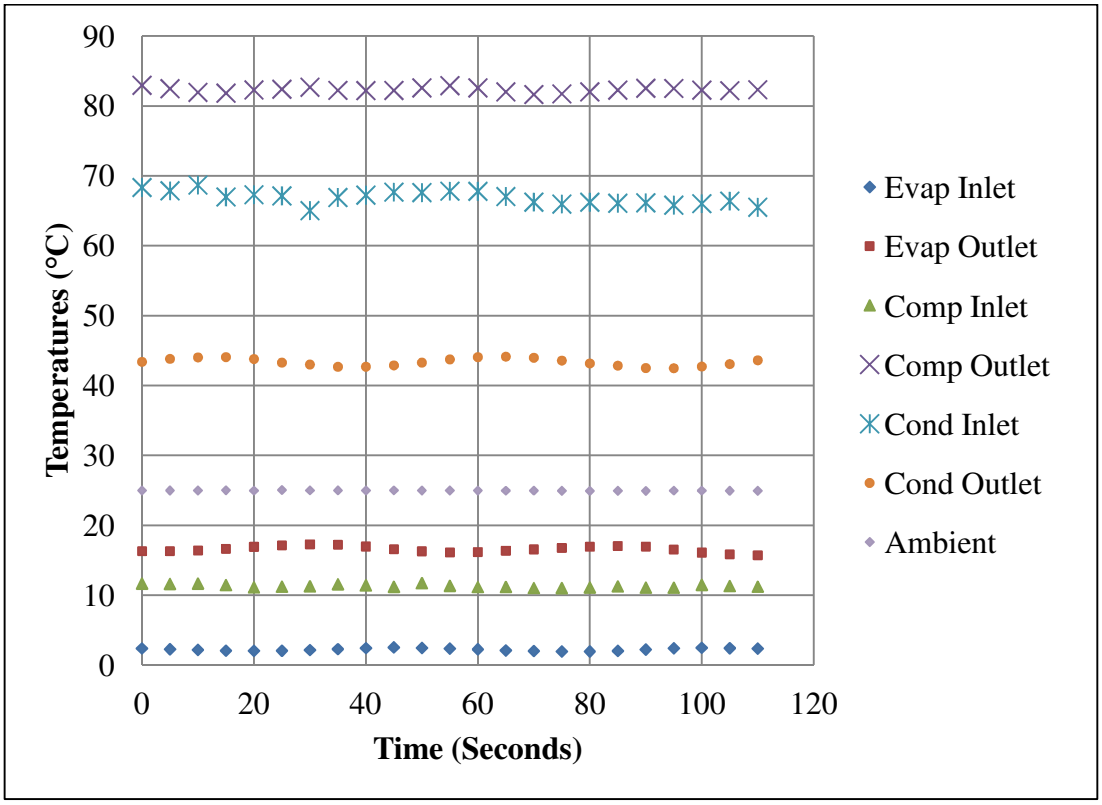

Figure 39. Temperature measurements for 500 Nm³/h air flow rate

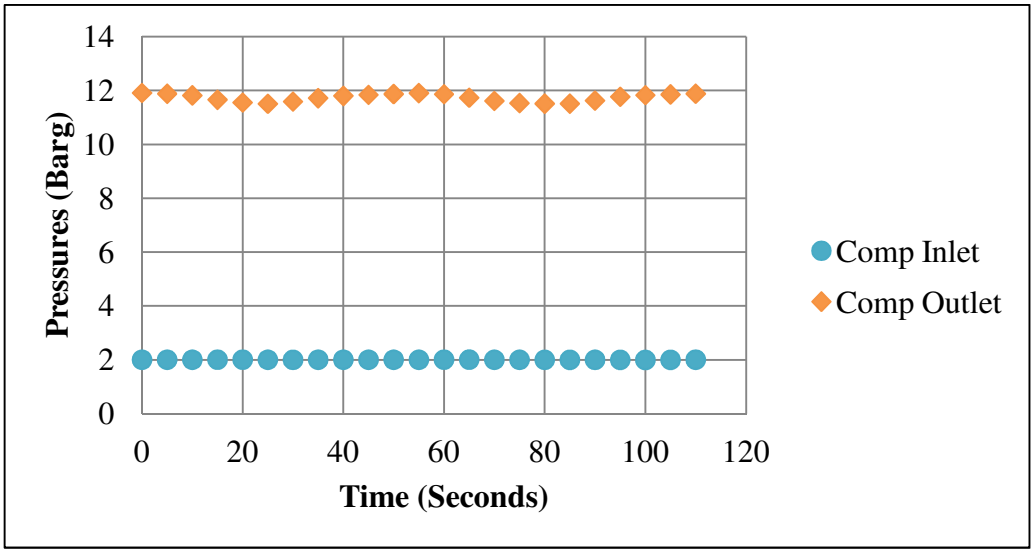

Figure 40. Pressure measurements for 500 Nm³/h air flow rate

4.2.2.4. Comments on Pressure Oscillations

In Figure 36, Figure 38 and Figure 40 the pressure profiles are seem to be stable and linear. However while the discharge and suction pressure values are separated, (0.1 bar max in suction pressures and 0.5 bars in discharge pressures) oscillation has been observed.

In steady state experiments the effect of hot gas by-pass valve is observed. There is always an oscillation at the inlet of the compressor where hot gas is injected after evaporator outlet. The valve is piloted by the pressure at the inlet of the compressor and the main goal of the valve is to keep the pressure always at set value (which is about 2.25 barg). In Figure 41, for a compressed air flow rate at 500 Nm³/h, the valve is not injecting the hot gas so often since the evaporation load possibly coming closer to the limit capacity of the cycle. Even though it may not necessary, by checking the sound, it is observed that the valve leaks some hot gas to the cycle which is just characteristic of the valve. These oscillations in

pressures and temperatures may be occurred because of unsteady behavior of condensation and evaporation, hot gas injection and response of the expansion valve.

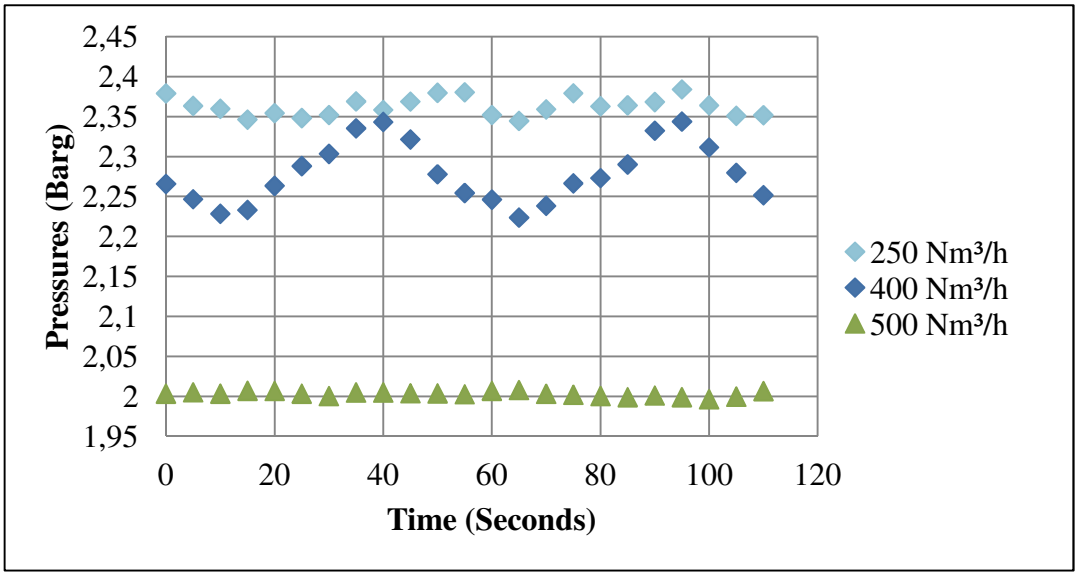

Figure 41. Comparison of suction pressure of the refrigerant compressor for relevant compressed air flow rates

In Figure 41 the pressure profiles in suction line of the compressor varies depending on the load on the evaporator and hot gas by pass activity. Since the boiling is unstable it is difficult to obtain a stable profile in pressures just like condensation. This instability is also provoked by hot gas injection by the hot gas by pass valve. In 250 Nm³/h experiment, the oscillation is very small compared to 400 and 500 Nm³/h. The possible reason behind this could be the evaporation rate of the refrigerant is very low therefore the unstable behavior of the phase change is not encountered like in larger evaporator loads.

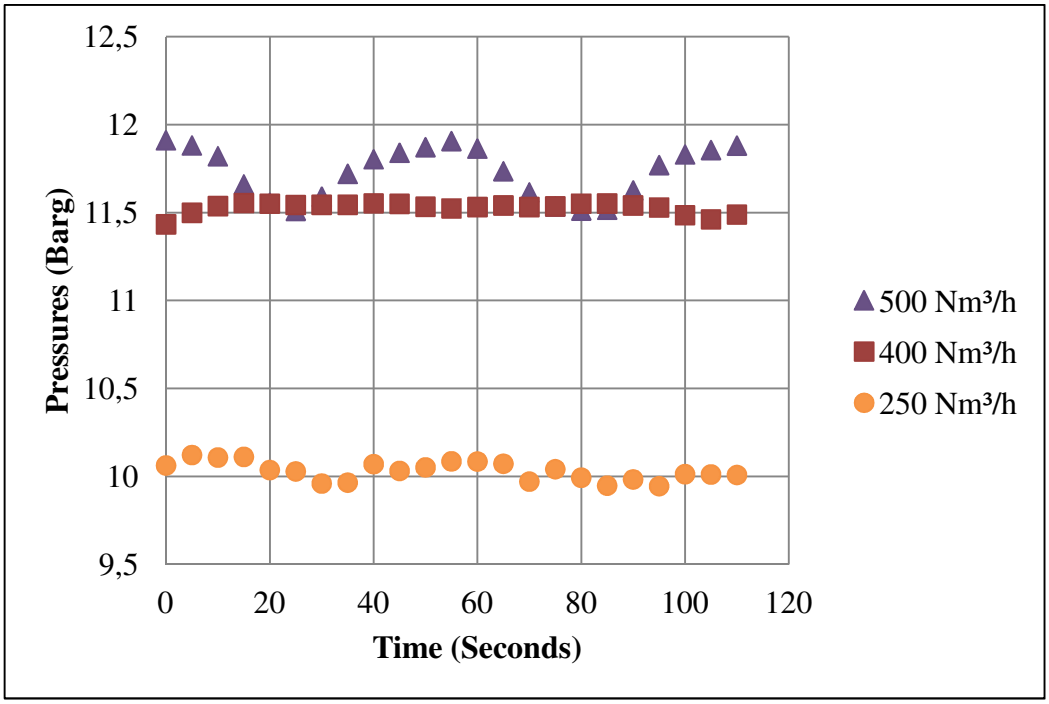

Figure 42. Comparison of discharge pressure of the refrigerant compressor for relevant compressed air flow rates

The pressure at the outlet of the compressor for each case seen in Figure 42. In here it is seen that condenser fan never lets the cycle to reach over 12 barg. The pressure switches are adjusted to run the fan at 12 barg and close at 10 barg. For 500 and 400 Nm³/h experiments, the fan is running all the time, the oscillation at 500 Nm³/h the oscillation could be seen because of the mass fraction between liquid and gas in condenser. When the gas phase is increasing, because of the friction, the pressure is increasing and when the gas change the phase to liquid the pressure drops and this results in this oscillation. This effect is seen in 400 Nm³/h and 250 Nm³/h experiments as well but in longer length. The reason is hot gas by-pass valve takes some amount of gas from the compressor outlet to inject at the inlet of the compressor. So the amount of gas is decreased in condenser and this fact enlarges the length of the oscillation.

CHAPTER 5

RESULTS

The Matlab ® code given in Appendix B used during calculation. The results are recorded to a Microsoft Excel ® file via code and this Excel file is used to obtain plots. In this chapter, the results for heat transfer coefficient calculation and thermodynamic calculation of the refrigeration cycle are given.

The code speeded up the solutions for all flow cases. The detailed table is given in the

APPENDIX E. Coefficient of performance (COP), 2nd law efficiency of the compressor and Irreversibility are considered as performance characteristics of the cycle for various flow rates.

Total heat transfer rates from compressed air to the refrigerant in the evaporator ($\dot{Q}_{Total}$) and from for all experimental conditions are given in Table 2.

Table 2. Total heat transfer rates versus compressed air flow rates

Air Flow Rate (Nm³/h)	250	400	500
$\dot{Q}_{Total}$ (W)	1150.65	2452.45	2923.71

To analyze the refrigeration cycle firstly the mass flow rates of the cycle are measured and obtained. The measured mass flow rate at the inlet of the compressor ($\dot{m}_{ref-comp}$), the calculated the mass flow rate that passes through hot gas by-pass valve ($\dot{m}_{ref-HGBPV}$), and calculate mass flow rate that passes through evaporator ($\dot{m}_{ref-evap}$) plotted on Figure 43.

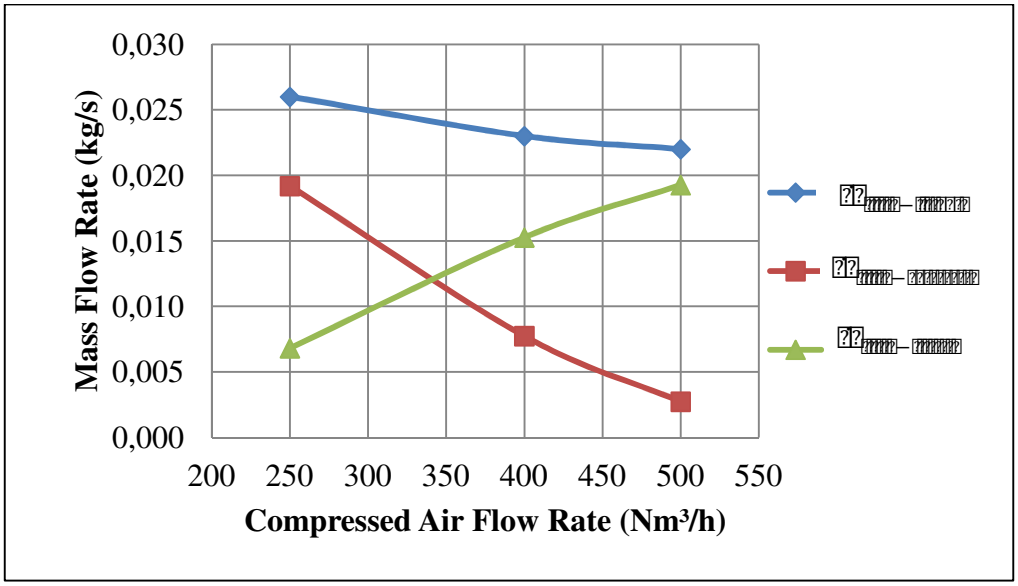

Figure 43. Refrigerant mass flow rates

When the evaporator load is low, the phase change could not complete in the evaporator then the refrigerant leaves the evaporator as a liquid-vapor mixture. It is not desired to have liquid at the inlet port of the compressor which harms the compressor directly. To avoid this defect, like told in 2.1.6. a hot gas by pass valve injects the hot gas to the outlet of the evaporator. This hot gas heats up the liquid refrigerant and vaporize it. This process may be expressed as "artificial load" on the evaporator. This process protects the compressor from liquid shock however cycle, especially compressor works like under full load conditions. Increasing compressed air flow rate, so increasing in evaporator load, results in reduction of the amount of the liquid refrigerant that leaves the evaporator.

Related to this fact, the required amount of hot gas injection is reduced as well as seen in Figure 43.

After finding the mass flow rates, the actual compressor work($\dot{W}_{comp-actual}$), required/exact compressor ($\dot{W}_{comp-exact}$) work and reversible compressor work($\dot{W}_{Rev}$) have been obtained.

Figure 44 illustrates the effect of hot gas by pass valve and "artificial load" concept on the compressor work. In all compressed air flow rates, the work of actual compressor($\dot{W}_{comp-actual}$) on stays at 1600 W while the required work to have same evaporation in the same cycle is lower than this value, i.e. for 250 Nm³/h compressed air flow rate, required work is about 424 W.

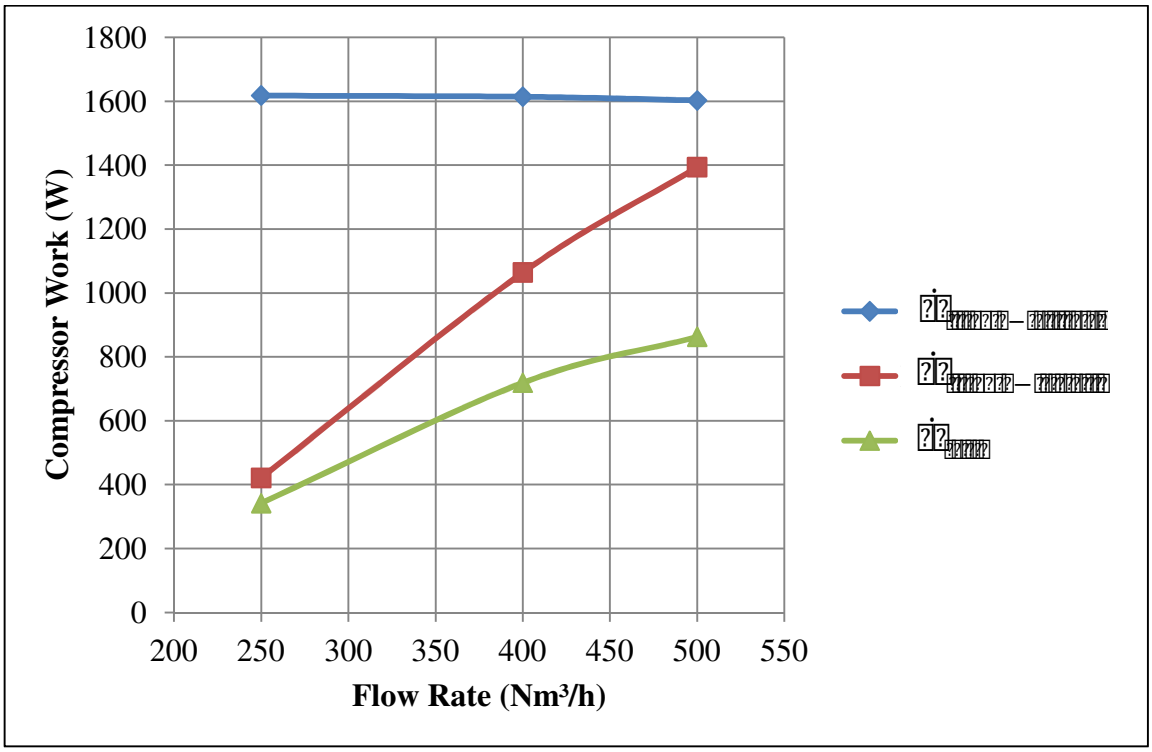

Figure 44. The amount of work that is applied on the cycle by compressor for actual and required and reversible work

It is expected that, this excessive work results in lower performance characteristics on the refrigeration cycle. The calculation results of the coefficient of performance prove this expectation. Figure 45 represents the results that are obtained.

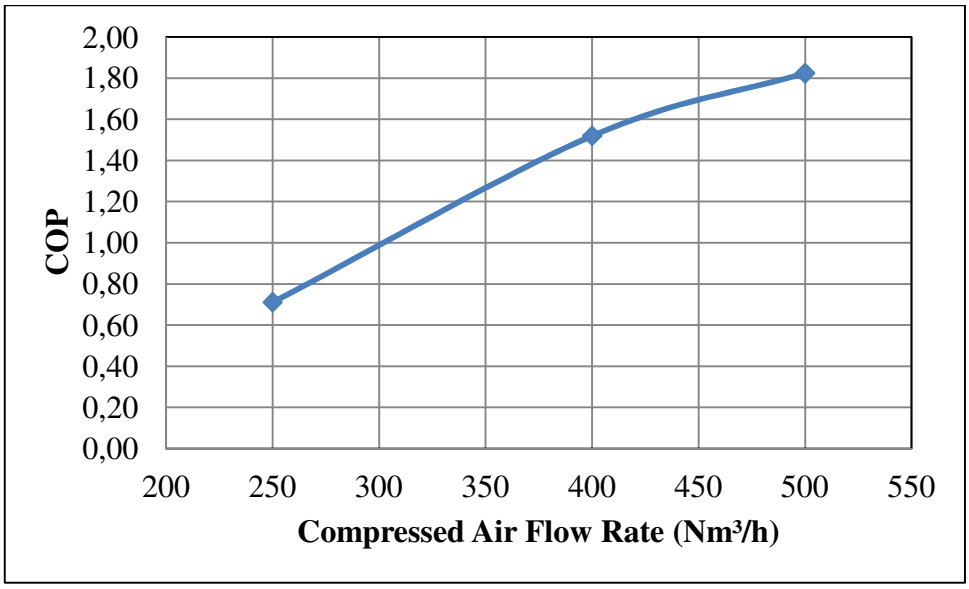

Figure 45. COP of the actual refrigeration cycle with HGBPV

In Figure 45, COP values of the cycle are not very high especially for lower flow rates as expected. For lower compressed air flow rates, the load on the evaporator is decreased while because of the hot gas by pass valve, the compressor power consumption remains same. In the end these facts decreases the COP values to 0.7 values for 250 Nm³/h. It is obtained that increasing compressed air flow rate increases the COP values because of the reduction in the requirement of the hot gas injection.

Similar behavior is observed on 2nd law efficiency of the cycle. The more compressed air flow rate (evaporator load) the more efficient the refrigeration cycle is.

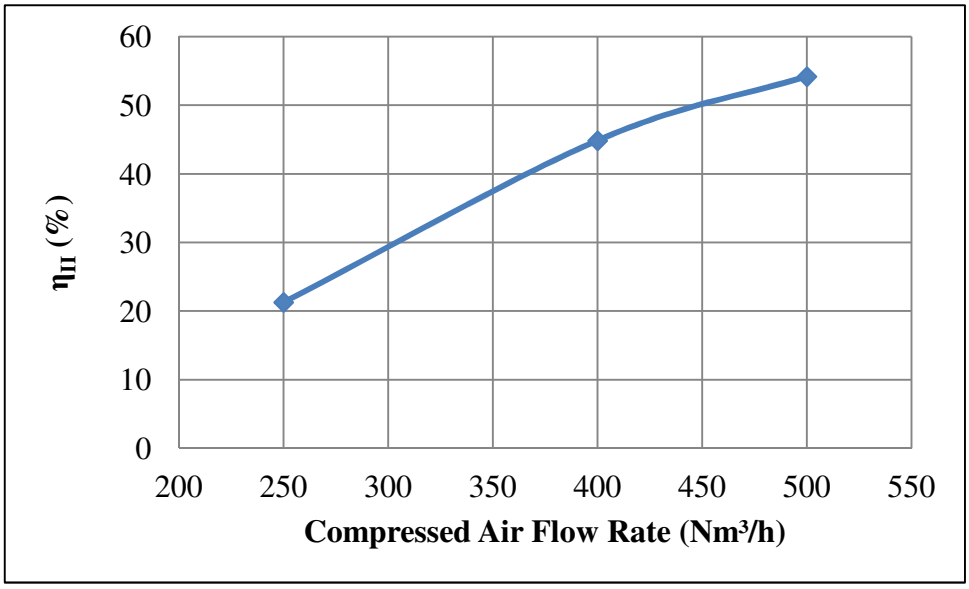

Figure 46. 2nd law efficiency of the actual refrigeration cycle with HGBPV

2nd law efficiency is found by dividing the reversible work to actual compressor work. As told previously, the compressor work does not change for each load conditions because of the artificial load while Reversible work is increasing. This results in increase in efficiency with increasing evaporator load.

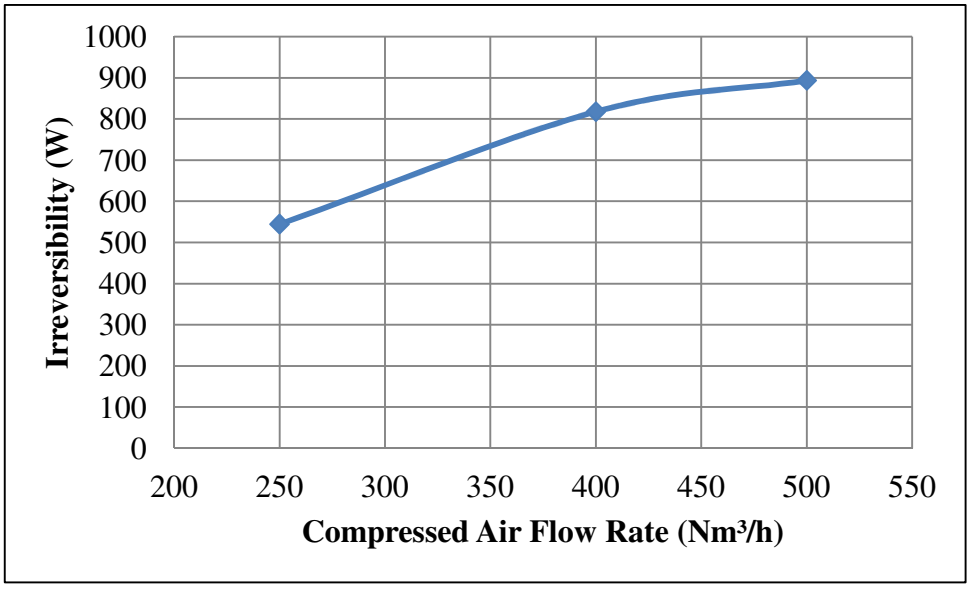

Figure 47. Irreversibility of the actual refrigeration cycle with HGBPV

The sources of irreversibility on the cycle are the friction due to the movement of piston compressor components, friction due to flow in tubes and in heat exchangers, unstrained expansion and compression, heat transfer through finite temperature differences in evaporator, condenser and pipes.

Table 3. Actual, required and reversible compressor works for related compressed air flow rates

Flow	$\dot{W}_{comp-actual}$	$\dot{W}_{comp-exact}$	Overload
Nm³/h	W	W	%
250	-1618.01	-424.15	73
400	-1614.25	-1071.44	33
500	-1602.61	-1403.45	12

By reminding the work values of the compressor by Table 3, the necessity of using hot gas by pass valve should be questioned. Especially for very low load conditions, such as 250 Nm³/h compressed air flow rate, the work used in the

refrigeration cycle is 73% larger than the required (exact) work. The excessive work (so the power) in the daily usage is not acceptable. If this low capacity application period is not taking too much time during the work shift then it may be suspended. However if the low capacity usage is very much, then an alternative solution should be used. The most commonly used solution for variable evaporator load condition is variable speed compressor application. As understood from the name, the compressor speed is controlled depending on the evaporator load. This solution is much more expensive than the hot gas by pass valve application, therefore before using, feasibility of the refrigeration cycle should be analyzed deeply. In following figures, the differences between hot gas by pass and variable speed application are examined.

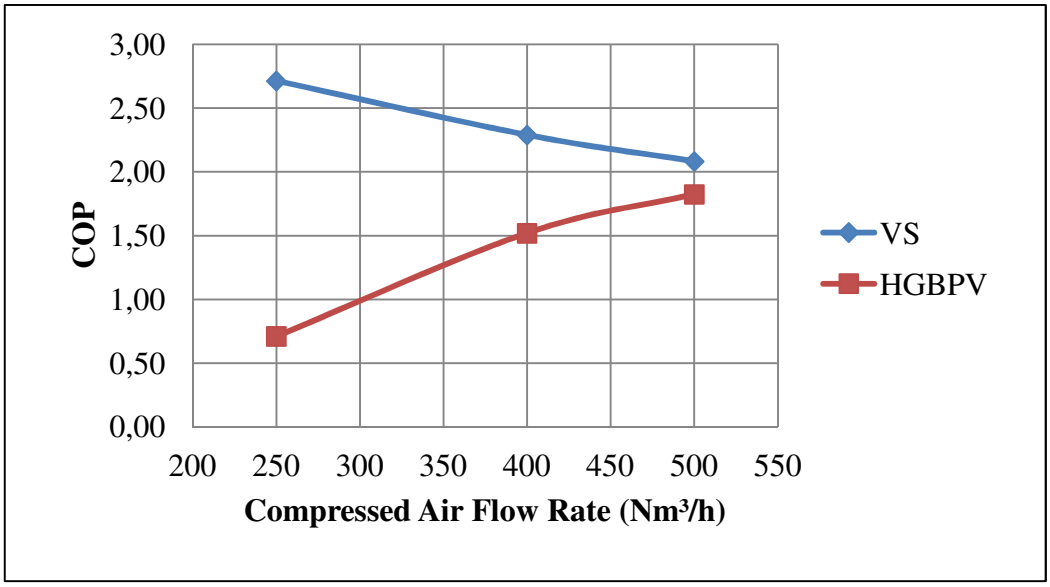

Figure 48. COP values of the cycle for both Variable Speed and Hot Gas By Pass applications

In Figure 48, the advantage of the variable speed compressor is observed. Especially for the 250 Nm³/h compressed air flow rate, the variation is very large. The COP value of the variable speed compressor is by increasing compressed air

flow rate and coming closer to hot gas by pass application. As expected, similar behavior is seen on 2nd law efficiency like seen in Figure 49.

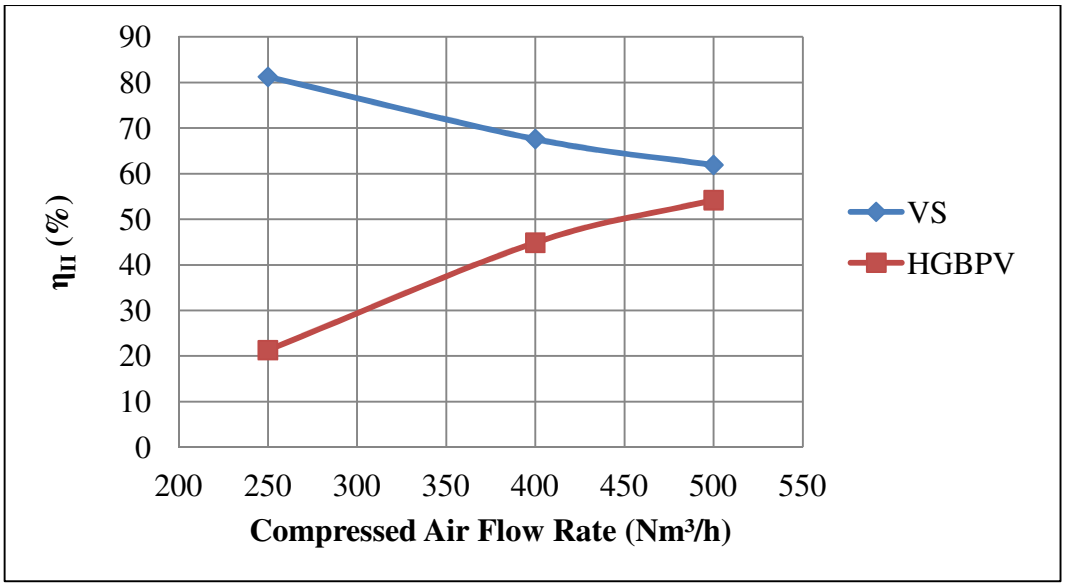

Figure 49. 2nd law efficiency of the cycle for both variable speed and hot gas by pass applications

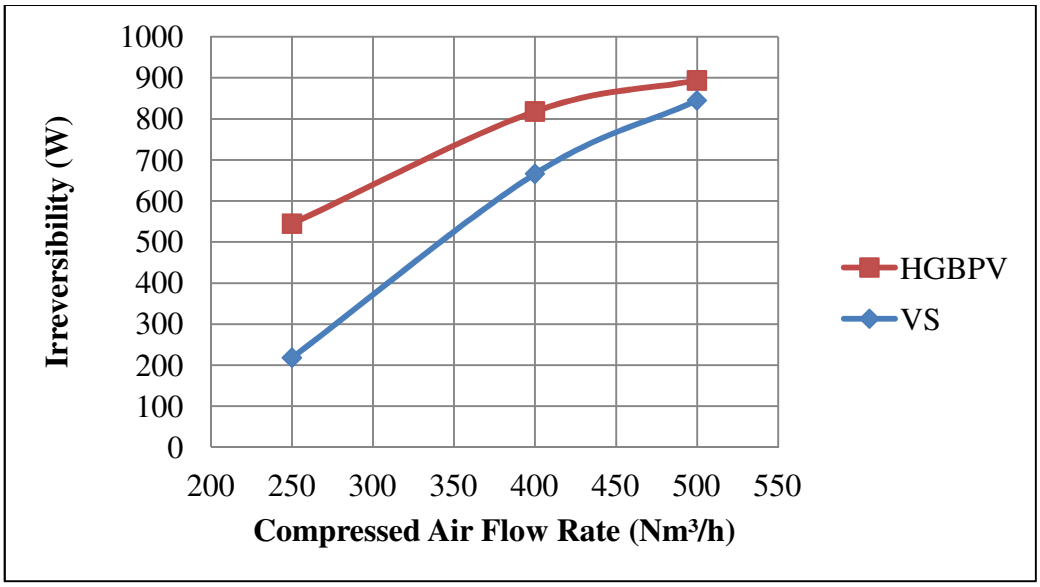

Figure 50. Irreversibility of the cycle for variable speed and hot gas by pass applications

In Figure 50, the irreversibility values of both variable speed compressor and hot gas by pass valve application could be seen. The irreversibility value of hot gas by pass application at 250 Nm^3/h air flow rate condition is much higher compared to variable speed compressor application the major reason behind this could be the excessive compressor work creates more friction which is a main source of irreversibility.

CHAPTER 6

CONCLUSIONS

6.1. Summary and Conclusions

In this thesis, the preparations of the experimental setup to perform real time compressed air flow experiments were presented. The “variable load” concept is investigated. Actual refrigeration cycle with hot gas by pass valve and alternative cycle with variable speed compressor were compared.

The preparation of the experimental setup, the devices that have been used, operation of the devices, experimental conditions, calculations and the results were described and discussed in detail. The measurements were seemed to be meaningful.

During the preparation of the setup, the most difficult part was the leakage on the connections of the tubes. It was very difficult to have a completely sealed refrigeration cycle with such complicated piping and instrumentation. A problem occurred with the small size piston compressor and it was replaced with a scroll type.

Total heat transfer rates and the actual work of the compressor for each experiment have been found. Maximum heat transfer rate is at 500 Nm^3/h compressed air flow rate and is 2923 W while minimum rate is 1150 W at 250 Nm^3/h compressed air flow rate as expected. However, the actual work

($\dot{W}_{comp-actual}$) is 1600 W at all compressed air flow rate experiments. The required/exact work ($\dot{W}_{comp-exact}$) that is required to transfer the heat from relevant compressed air flow rate to the refrigerant is obtained. This fact, of course, reduced the coefficient of performance (COP) in low compressed air flow rates. The reason that compressor works all the time at same level is the application of hot gas by pass valve on the refrigeration cycle. This valve injects the hot gas at the outlet of the compressor to the outlet of the evaporator to avoid any liquid flood to the compressor inlet at lower loads. Similar effect has been observed in 2nd law efficiency of the refrigeration cycle as well. The irreversibility values have been found for each experiment and compared. Since the work of the compressor is similar in each case and irreversibility values were increasing by increasing load on evaporator, the major factor effect could be told as the heat transfer to the finite temperature difference on the cycle.

An alternative solution for variable load condition is presented. The variable speed compressor application is commonly used in industry. It is assumed that the exact amount of work could be required by a variable speed compressor. The obtained results from the cycle with hot gas by pass valve and variable speed compressor applications were compared in detail. It is found that for lower loads, variable speed compressor gives better results in terms of COP (up to 73%), 2nd law efficiency (up to 73%) and irreversibility (up to 60%). However the major disadvantage of using such compressor is the initial cost. It is not recommended to use a variable speed compressor if the compressed air flow variation is not too much (in this thesis 50% has been evaluateed) or the period that the dryer works under no load or very low load condition is very short.

6.2. Future Work

The experimental setup is ready to carry out for other experimental conditions (i.e. different ambient temperature, different air inlet temperatures, smaller-larger compressor-condenser combination, smaller-larger air- air heat exchanger etc.). For future studies, a digital mass flow meter is strongly suggested. The

experiments were carried out by using piston type (primary) compressor. The scroll compressor (secondary) may be used to compare the results with piston type.

The evaporator and economizer can be modeled by considering the thermal and mass transfer. Enough number of RTD probes were located on economizer and evaporator. By using the measurements that have been recorded could be used for modeling. A CFD analysis can be performed on the fins to obtain numerical results to compare with experimental values.

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APPENDIX A

SPECIFICATIONS OF DEVICES IN THE EXPERIMENTAL SETUP

A.1. COMPRESSORS

Compressor Identification	Compressor 1	Compressor 2
Manufacturer	Tecumseh	Copeland
Country of Origin	France	U.S.A.
Type/Model	Piston / FH4518Y	Scroll / ZR22K3E-PFJ
Power Input (kW)	1.42	0.92
Cooling Capacity (kW)	4.5	3.35
COP	3	3.66

A.2. CONDENSERS

Condenser Identification	1	2	3	4
Type/Model	VT50	VT100	VT400	VT500
Maximum Heat Rejection (W)	1250	2100	3117	3867
Manufacturer	Karyer			
Country of Origin	Turkey			

A.3. EVAPORATORS

Evaporator Identification	Evaporator 1	Evaporator 2
Type/Model	VT400	VT400
Cooling Capacity (kW)	3	3
Manufacturer	Mikropor	
Country of Origin	Turkey	

A.4. EXPANSION VALVES

Manufacturer	Danfoss
Model	T2 Orifice 4
Cooling Capacity (@0°C Evaporation)	5.2 kW max
Country of Origin	Denmark

A.5. HOT GAS BY-PASS VALVE

Manufacturer	Danfoss
Model	CPCE 12
By-Pass Capacity	7.6 kW max
Country of Origin	Denmark

A.6. FLOW METERS

Flow Meter	Flow Meter 1	Flow Meter 2
Model	VA40	H250
Type	Glass Cone	Metal Cone
Scale (kg/s)	0.011 – 0.1	0.011 – 0.1
Accuracy (%)	1	1.6
Manufacturer	Krohne	
Country of Origin	Germany	

A.7. PRESSURE TRANSMITTERS

Manufacturer	Enelsan
Model	Etrans-P
Current Range	4-20 mA
Pressure Range	0-20 barg
Accuracy	0.2%
Power Supply	10-36 V DC
Country of Origin	Turkey

A.8. RESISTANCE TEMPERATURE DETECTORS (RTD'S)

Manufacturer	Omega
Model	PR-11-2-100-1/4-12-150-E
Current Range	4-20 mA
Temperature Range	-200 – 260 °C
Accuracy	±0.3-0.8°C from -0 to 100°C
Class	A (alpha = 0.00385 Ω/Ω/ºC))
Country of Origin	United Kingdom

A.9 THERMOCOUPLE WIRE

Manufacturer	Cole Parmer
Model	T30-2-506
Temperature Range	-200 – 204°C
Accuracy	±1°C from -65 to 130°C
Type	T
Country of Origin	U.S.A.

A.10 DEW POINT METER

Manufacturer	CS Instruments
Model	FA410
Range	-20 / 50°C T_{dp}
Current Range	4-20 mA
Accuracy	±1°C
Country of Origin	Germany

A.11 DATA ACCUSATION DEVICE

Manufacturer	Datataker
Model	DT85
Current Range	4-20 mA
Error Limit	±1°C from -65 to 130°C
Reading Accuracy	0.01% of full scale
Country of Origin	Australia

APPENDIX B

MATLAB ® CODE TO IMPLEMENT THE VALUES

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%
% FILENAME                      : RefrigerantAirDryer.m
% WRITTEN BY                    : Mustafa USLU
%                                 Middle East Technical Univeristy
% FIRST WRITTEN DATE            : MAY 2012
% LAST MODIFICATION DATE        : SEPTEMBER 2012
%
% MODIFICATION HISTORY
% SEPTEMBER 2012 - REFRIGERATION THERMODYNAMICS PARTS COMPLETED
%
%
%%%%%%%%%%%%%%%%%%%%%%%%%COMMENTS ON
CODE%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%
% This code is written only for reading and implementing the data
that is
% read by data logger and saved in Excel file (xls) format.
%
% This code works with a code called "refpropm.m" Version 8
written by NIST
% and needs to be placed in the same directory where refpropm.m
is(generally
% in the directory C:\Program Files\REFPROP\MATLAB\).
% Otherwise this code will not work. The version of Matlab is
R2009b
%
% There are two sections in this code. First section deals with
the heat
% exchanger modeling while the second section does thermodynamical
% performance of the dryer.
%
% The Input Parameters are:
%
% Pair - Compressed Air Pressure (barg)
% Icomp - Average Refrigerant compressor current rate (amps)
% V - Voltage (V)
% Wcompe - Electrical power input to Refrigerant Compressor (W)
% nce - Efficiency of the compressor of converting electrical
power to work
% fluid - defined fluid/refrigerant to use in "refpropm" code
```

```
% FlowRate - Flow rate of compressed air under normal conditions
(Nm³/h)
% XLS - The data read from Excel File contains all temperature and
% n - time step size
% RH - Relative Humidity
% Preference - Reference pressure (kPa)
% Treference - Reference temperature (K)
% Read Data from Excel File:
% Tevapin - Evaporator inlet temperature (°C)
% T1e - Evaporator outlet temperature (°C)
% T1 - Compressor inlet temperature (°C)
% T2 - Compressor outlet temperature (°C)
% T2c - Condenser inlet temperature (°C)
% T3 - Condenser outlet temperature (°C)
% Tamb - Ambient temperature (°C)
% T1i - Compressed air inlet temperature (outer tube)(°C)
% T1o - Compressed air outlet temperature (outer tube)(°C)
% T2i - Compressed air inlet temperature (inner tube)(°C)
% T2o - Compressed air outlet temperature (inner tube)(°C)
% Tdp - Dew point temperature (°C)
% P1 - Suction (low) pressure of refrigerant compressor (kPa)
% P2 - Discharge (high) pressure of refrigerant compressor (kPa)
% Tave - Average / mean temperature (K)
% rhoa - Density of air (kg/m³)
% CR - Compression rate
% ha - Enthalpy of air inlet (J/kg)
% sa - Entropy of air inlet (J/kg.K)
% Cp - Specific heat (J/kg)
% k - Conductivity (W/m.K)
% a - Diffusivity (m²/s)
% mu - Dynamic viscosity (Pa-s)
% v - Kinematic viscosity (m²/s)
% Pr - Prandtl Number
% Q - Heat transfer rate (W)
% Pd - Saturation pressure of the water (kPa)
% w - Specific humidity
% mw - water mass flow rate (kg/s)
% Qtotal - Total heat transfer rate (W)
% h - Enthalpy of refrigerant (J/kg)
% S - Entropy of refrigerant (J/kg.K)
% x - Quality of the refrigerant (mv/ml)
% mref - Refrigerant mass flow rate (kg/s)
% Wcomp - Compressor work (W)
% COP - Coefficient of performance
% ns - Isentropic compession efficiency
% I - Irreversibility
% Itotala - Total Irreversibility of actual system
% Itotale - Total Irreversibility of variable speed compressor
system
% nIIa - 2nd law efficiency of actual system
% nIIe - 2nd law efficiency of variable speed compressor system

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%
clear all;
% clc

Pair=7;
```

```
Pair=Pair*100+101.325; % converts the pressure in to absolute kPa
Preference=101.325; %kPa (According to ISO 2533)
Treference=288; %K (According to ISO 2533)
Icomp=6.3; % Amps  Measured Current
V=224;% Volts  Measured voltage
Wcompe=V*Icomp;
nce=0.72; % Electrical convergence efficiency of the compressor

fluid='r134a'; % defined in Refprop Matlab code
FlowRate=400;% Nm³/h  | Flowrate of compressed air read in the lab
XLS=xlsread('400m3h.xlsx');  % reads the Excel document and create
a matrix
n=size(XLS,1); % to determine how many steps are required
RH1=1; %Relative Humidity of Compressed Air at the inlet
RH2=1; %Relative Humidity at the bottom where condensation occurs



%%
Qtotal=zeros(n,1); % These matrices will be used in the following
loop
Qcond=zeros(n,1);
Qcomp_cond=zeros(n,1);
mref=zeros(n,4);
Wcomp=zeros(n,3);
COP=zeros(n,2);
ns=zeros(n,1);
UA=zeros(n,2);


for i=1:1:n;

Tevapin=XLS(i,2)+273.15;
T1e=XLS(i,3)+273.15;
T1=XLS(i,4)+273.15;
T2=XLS(i,5)+273.15;
T2c=XLS(i,6)+273.15;
T3=XLS(i,7)+273.15;
Tamb=XLS(i,8)+273.15;
T1i=XLS(i,11)+273;
T1o=XLS(i,12)+273;
T2i=XLS(i,14)+273;
T2o=XLS(i,15)+273;
Tdp=XLS(i,16)+273;

P1=XLS(i,9)*100+101.325;
P2=XLS(i,10)*100+101.325;
Tave1=(T1i+T1o)/2;
Tave2=(T2i+T2o)/2;

%Density of air and compression rate
rhoa1=refpropm('D','T',T1i,'P',Pair,'nitrogen','oxygen','argon',..
.
    [0.7557, 0.2316, 0.0127]); % kg/m³ Inlet air density
rhoa2=refpropm('D','T',Treference,'P',101.325,'nitrogen','oxygen',
'argon',...
```

```
    [0.7557, 0.2316, 0.0127]); %  kg/m³ air density under normal
conditions
CR=rhoa1/rhoa2; % compression rate

%Mass flow rate of air
mair=FlowRate/3600*rhoa2; % kg/s

% Enthalpy and entropy of air
ha1i=refpropm('H','T',T1i,'P',Pair,'nitrogen','oxygen','argon',...
    [0.7557, 0.2316, 0.0127]);%J/kg Enthalpy of Air Inlet
ha2o=refpropm('H','T',T2o,'P',Pair,'nitrogen','oxygen','argon',...
    [0.7557, 0.2316, 0.0127]);%J/kg Enthalpy of Air outlet

sa1i=refpropm('S','T',T1i,'P',Pair,'nitrogen','oxygen','argon',...
    [0.7557, 0.2316, 0.0127]);%J/kg-K Entropy of Air Inlet
sa2o=refpropm('S','T',T2o,'P',Pair,'nitrogen','oxygen','argon',...
    [0.7557, 0.2316, 0.0127]);%J/kg-K Entropy of Air Outlet


%%
Cpa=refpropm('C','T',T1i,'P',Pair,'nitrogen','oxygen','argon',...
    [0.7557, 0.2316, 0.0127]);    % J/kg-K specific heat of the
air (assuming constant

Pd1=refpropm('P','T',T1i,'Q',1,'water');  % kPa Pressure of the
water inside the mixture
hw1=refpropm('H','T',T1i,'Q',1,'water');  % J/kg Enthalpy of the
water
w1=0.622/(Pair/(RH1*Pd1)-1);              % kgw/kga specific
humidity
mw1=mair*w1;                              % kg/s mass rate of
water inside compressed air

Pd2=refpropm('P','T',Tdp,'Q',0,'water'); % kPa pressure of the
water inside the mixture
hw2=refpropm('H','T',Tdp,'Q',0,'water'); % J/kg Enthalpy of the
water in
w2=0.622/(Pair/(RH2*Pd2)-1);             % kgw/kga specific
humidity
mw2=mair*w2;                             % kg/s mass of water
inside compressed air in

mw=mw1-mw2;                  % kg/s mass of water that is condensed
Qw=mw*(hw1-hw2);             % heat that is required to condense
that amount of water in
Qa=mair*Cpa*(T1i-T2o);   % heat that is required to have that cold
air out in W
Qtotal(i,1)=Qa+Qw;           % total heat transfer in W
%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%
%%%%%%%%%%%%%%%%%Refrigeration part Analysis%%%%%%%%%%%%

%Compressor
h1=refpropm('h','T',T1,'P',P1,fluid); % J/kg Enthalpy at the inlet
of the
                                      % compressor
```

```
S1=refpropm('S','T',T1,'P',P1,fluid); % J/kg.K Entropy at the
inlet of the
                                      % compressor

h2=refpropm('h','T',T2,'P',P2,fluid); % J/kg Enthalpy at the
outlet of the compressor
S2=refpropm('S','T',T2,'P',P2,fluid); % J/kg.K Entropy at the
outlet of the compressor

% For isentropic compression
S2s=S1;  % For isentropic case, inlet and outlet entropies of the
compressor are equal
T2s=refpropm('T','P',P2,'S',S2s,fluid); % K Isentropic Outlet
temperature
h2s=refpropm('h','T',T2s,'P',P2,fluid); % J/kg Enthalpy at the
outlet of the compressor

%Condenser
h2c=refpropm('h','T',T2c,'P',P2,fluid); %J/kg Enthalpy at the
inlet of the condenser
S2c=refpropm('S','T',T2c,'P',P2,fluid); %J/kg.K Entropy at the
inlet of the condenser


P3=P2;
h3=refpropm('h','T',T3,'P',P3,fluid); % J/kg Enthalpy at the
outlet of the condenser
S3=refpropm('S','T',T3,'P',P3,fluid); % J/kg.K Entropy at the
outlet of the condenser

%Expansion valve
h4=h3; %Assuming isenthalpic expansion
P4=P1; % Outlet pressure of the expansion valve is equal to the
inlet of the compressor
Q4=refpropm('Q','h',h4,'P',P4,fluid); % Quality of the refrigerant
at the inlet of the evaporator
T4=refpropm('T','P',P4,'Q',Q4,fluid); % K Temprature at related
state
S4=refpropm('S','P',P4,'Q',Q4,fluid); % J/kg.K Entropy at the
inlet of the evaporator

%Evaporator
h1e=refpropm('h','T',T1e,'P',P1,fluid); % J/kg Enthalpy at the
outlet of the evaporator
S1e=refpropm('S','T',T1e,'P',P1,fluid); % J/kg.K Entropy at the
outlet of the evaporator

mref(i,1)=Qtotal(i,1)/(h1e-h4);   % kg/s Required Refrigerant flow
through evaporator
mref(i,2)=XLS(i,17);              % kg/s Refrigerant  flow through
compressor (read from flow meter)
mref(i,3)=mref(i,2)-mref(i,1);    % kg/s Refrigerant flow through
hot gas by pass valve

mref(i,4)=mref(i,3)/mref(i,2)*100;% percentage of by pass
Wcomp(i,2)=mref(i,2)*(h2-h1)/nce;     % W Actual Compressor work
```

```
Wcomp(i,1)=mref(i,1)*(h2-h1)/nce;     % W Compressor work for the
flow in evaporator
Wcomp(i,3)=Qtotal(i,1)*(Tamb-(((T4+T1e)/2))/(((T4+T1e)/2)))/1000;
% Reversible work
Qcond(i,1)=mref(i,1)*(h3-h2c);    % W Condenser heat transfer rate
Qcomp_cond(i,1)=mref(i,1)*(h2c-h2); % W Heat loss from the pipe
that connects compressor to condenser
COP(i,1)=Qtotal(i,1)/Wcomp(i,1); % Coefficient of performance if
only the right compressor was used
COP(i,2)=Qtotal(i,1)/Wcomp(i,2);    % Coeffficient of Performance
for Actual Case
ns(i,1)=(h2s-h1)/(h2-h1);       % isentropic efficiency of the
compressor

T0=Tamb; % Amb?ent temperature is assumed to be the reference
temperature

IComp=(mref(i,2)*(S2-S1))*T0; % W Irreversibility of the variable
speed compressor work
ICond=(mref(i,1)*(S3-S2c)-Qcond(i,1)/Tamb)*T0; % W Irreversibility
of the condenser
IExp=(mref(i,1)*(S4-S3))*Tamb; % W Irreversibility of the
expansion valve
IEvap=(mref(i,1)*(S1e-S4)-Qtotal(i,1)/((T4+T1e)/2))*T0; % W
Irreversibility of the evaporator
IComp_Cond=(mref(i,1)*(S2c-S2)-Qcomp_cond(i,1)/Tamb)*T0; % W
Irreversibility of the heat loss between the line compressor and
condenser
ICompe=(mref(i,1)*(S2-S1))*T0; % W Irreversibility of actual
compressor work

Itotala=IComp+ICond+IExp+IEvap+IComp_Cond; % W Total
Irreversibility of actual cycle
Itotale=ICompe+ICond+IExp+IEvap+IComp_Cond; % W Total
Irreversibility of the cycle with variable speed compressor


% nIIa=abs(Qtotal(i,1)*(1-Tamb/((T4+T1e)/2)))/Wcomp(i,2)*100; %
Actual 2nd law efficiency
nIIa=Wcomp(i,3)/Wcomp(i,2)*100; % 2nd Law efficiency with variable
speed compressor
nIIe=Wcomp(i,3)/Wcomp(i,1)*100; % 2nd Law efficiency with variable
speed compressor

XLS(i,16)=Qtotal(i,1);
XLS(i,17)=Qcond(i,1);
XLS(i,18)=Qcomp_cond(i,1);
XLS(i,19)=0;
XLS(i,20)=COP(i,1);
XLS(i,21)=COP(i,2);
XLS(i,22)=mref(i,1);
XLS(i,23)=mref(i,2);
XLS(i,24)=mref(i,3);
XLS(i,25)=ns(i,1);
XLS(i,26)=Itotale;
XLS(i,27)=Itotala;
XLS(i,28)=abs(Itotala-Itotale)/Itotala*100;
XLS(i,29)=nIIe;
```

```
XLS(i,30)=nIIa;
XLS(i,31)=Wcomp(i,1);
XLS(i,32)=Wcomp(i,2);
XLS(i,33)=Wcomp(i,3);

end


xlswrite('500.xls',XLS) % to write all the data and results into a
new Excel File

ThermodynamicResults=mean(XLS); % For steady state operation gives
the idea


disp('The Calculation Completed Succesfully!'); % Shows the end of
the calculation


%%% Compressor Efficiency Testing%%%


clear all;
clc


fluid='r134a'; % defined in Refprop Matlab code
FlowRate=400;% Nm³/h  | Flowrate of compressed air read in the lab
XLS=xlsread('Comp_Test.xlsx');   % reads the Excel document and
create a matrix
n=size(XLS,1); % to determine how many steps are required



%%
Qtotal=zeros(n,1); % These matrices will be used in the following
loop
Qcond=zeros(n,1);
Qcomp_cond=zeros(n,1);
mref=zeros(n,4);
Wcomp=zeros(n,2);
COP=zeros(n,2);
ns=zeros(n,1);
UA=zeros(n,2);

%Reference State
T0=15; %°C
P0=101.325; %kPa

% T0=T0+273.15;
%
% U0r=refpropm('U','T',T0,'P',P0,fluid); % Internal energy of the
refrigerant at dead state
%  h0r=refpropm('h','T',T0,'P',P0,fluid);  %  Enthalpy  of  the
refrigerant at dead state
```

```
%   s0r=refpropm('S','T',T0,'P',P0,fluid);   %   Entropy   of   the
refrigerant at dead state
% V0r=1/refpropm('D','T',T0,'P',P0,fluid); % Specific Volume of
the refrigerant at dead state
Results=zeros(n,3);

for i=1:1:n;
Tcompin=XLS(i,2)+273.15; % Compressor inlet temperature converted
to K
Tcompout=XLS(i,3)+273.15;   %   Compressor   outlet   temperature
converted to K
Pcompin=XLS(i,4)*100+101.325;   %   Compressor   inlet   Pressure
converted to kPa
Pcompout=XLS(i,5)*100+101.325;   %   Compressor   outlet   Pressure
converted to kPa
mref_read=XLS(i,6); % Read mass flowrate
I_read=XLS(i,7); % Read Amp Value
V_read=XLS(i,8); % Read Voltage value

W_electrical=I_read*V_read;


%Compressor
h1=refpropm('h','T',Tcompin,'P',Pcompin,fluid); % enthalpy at the
inlet of the compressor in j/kg
h2=refpropm('h','T',Tcompout,'P',Pcompout,fluid);  %  enthalpy  at
the outlet of the compressor in j/kg


W_comp=mref_read*(h2-h1);

ne=W_comp/W_electrical*100;

Results(i,1)=W_electrical;
Results(i,2)=W_comp;
Results(i,3)=ne;

XLS(i,9)=Results(i,1);
XLS(i,10)=Results(i,2);
XLS(i,11)=Results(i,3);
end


xlswrite('comp_Test_Results.xls',XLS) % to write all the data and
results into a new Excel File


MeanNe=mean(Results);%  Mean  values  of  overall  heat  transfer
coefficients

disp(MeanNe(1,3));
```

APPENDIX C

PRESSURE TRANSMITTER CALIBRATION

Table C. 1. The calibration results for the pressure transmitter

Pressure	Current
barg	mA
1	4.79
2	5.61
3	6.40
4	7.19
5	8.01
6	8.79
7	9.62
8	10.40

Pressure	Current
barg	mA
9	11.20
10	12.21
11	12.79
12	13.60
13	14.50
14	15.21
15	16.00

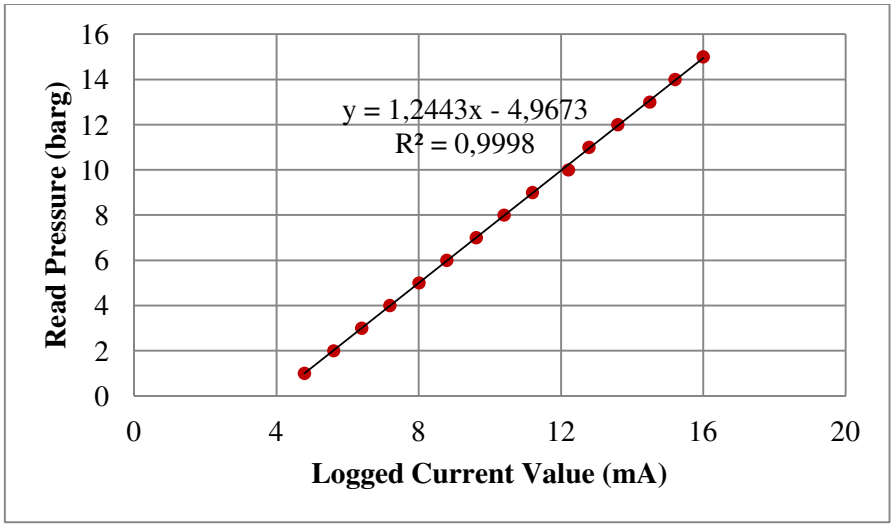

Figure C. 1. Calibration curve for the pressure transmitter

APPENDIX D

MEASURED DATA DURING EXPERIMENTS

D.1. MEASURED DATA OF 250 Nm³/h EXPERIMENTS

Table D. 1. Measured temperature and pressure values of 250 Nm³/h experiments

Time Step	Evap Inlet	Evap Outlet	Comp Inlet	Comp Outlet	Cond Inlet	Cond Outlet
seconds	°C	°C	°C	°C	°C	°C
0	4.24	9.19	10.36	72.81	53.03	26.04
5	4.32	9.13	11.81	72.47	53.64	25.92
10	4.50	9.06	11.08	71.84	53.58	26.37
15	4.56	8.99	11.06	72.26	53.94	25.95
20	4.55	8.88	11.30	72.05	53.65	26.55
25	4.48	8.81	11.98	72.02	53.49	26.73
30	4.50	8.83	11.96	72.54	53.25	25.89
35	4.56	8.99	13.04	72.67	53.43	26.39
40	4.42	8.95	11.71	72.69	53.30	26.57
45	4.06	8.82	13.30	72.94	52.48	26.96
50	4.15	8.90	12.44	72.51	53.39	26.31
55	4.26	8.85	12.97	72.91	53.84	26.24
60	4.51	8.70	12.16	73.56	54.19	26.61
65	4.59	8.58	11.68	72.84	54.09	26.26
70	4.49	8.60	12.13	72.50	54.01	26.49
75	4.47	8.61	11.63	71.55	53.94	25.86
80	4.24	8.44	12.48	71.87	53.86	26.53
85	3.90	8.34	12.20	72.33	53.57	26.59
90	4.15	8.40	11.83	71.98	53.35	26.12
95	4.27	8.46	12.18	72.45	53.43	26.00
100	4.44	8.38	11.98	72.86	53.23	25.96
105	4.34	8.09	12.87	72.98	52.85	26.13
110	4.27	8.08	11.53	72.78	52.51	25.79

Table D.1. Continued

Time Step	Exp Inlet	Ambient	Air Inlet	AirOutlet	Comp Inlet	Comp Outlet
seconds	**°C**	**°C**	**°C**	**°C**	**barg**	**barg**
0	26.58	24.43	31.18	25.03	2.38	10.06
5	27.80	24.54	31.26	25.62	2.36	10.12
10	26.14	24.39	31.20	25.56	2.36	10.11
15	27.12	24.47	31.17	25.39	2.35	10.11
20	27.12	24.75	31.29	25.31	2.35	10.04
25	26.53	24.78	31.26	25.02	2.35	10.03
30	26.92	24.39	31.22	25.14	2.35	9.96
35	26.14	24.51	31.31	25.31	2.37	9.96
40	27.35	24.69	31.33	25.28	2.36	10.07
45	26.98	24.57	31.26	25.59	2.37	10.03
50	26.22	24.46	31.35	25.82	2.38	10.05
55	27.38	24.81	31.36	25.37	2.38	10.09
60	27.60	24.46	31.30	25.08	2.35	10.08
65	27.50	24.56	31.37	25.39	2.34	10.07
70	27.44	24.60	31.39	25.79	2.36	9.97
75	27.35	24.59	31.35	25.28	2.38	10.04
80	27.68	24.53	31.40	25.47	2.36	9.99
85	27.34	24.45	31.43	25.47	2.36	9.95
90	27.54	24.50	31.39	25.16	2.37	9.98
95	27.55	24.50	31.42	25.37	2.38	9.94
100	27.78	24.60	31.49	25.26	2.36	10.01
105	27.85	24.47	31.43	24.96	2.35	10.01
110	27.79	24.47	31.44	25.28	2.35	10.01

D.2. MEASURED DATA OF 400 Nm³/h EXPERIMENTS

Table D. 2. The measured temperature and pressure values of 400 Nm³/h experiments

Time Step	Evap Inlet	Evap Outlet	Comp Inlet	Comp Outlet	Cond Inlet	Cond Outlet
seconds	°C	°C	°C	°C	°C	°C
0	3.58	16.70	6.26	75.18	58.01	36.22
5	3.66	16.65	7.91	74.70	59.23	38.18
10	3.64	16.58	7.09	75.63	56.59	38.23
15	3.61	16.56	7.84	75.86	57.38	38.31
20	3.57	16.62	7.89	76.13	57.32	38.63
25	3.54	16.70	7.50	76.10	58.68	38.82
30	3.53	16.83	6.10	74.73	58.51	38.27
35	3.43	16.94	5.06	75.31	59.04	37.96
40	3.41	17.04	4.51	74.89	59.22	37.71
45	3.39	17.17	5.22	75.08	59.13	37.67
50	3.38	17.29	4.60	75.07	59.05	37.50
55	3.37	17.31	4.63	74.88	58.92	37.37
60	3.39	17.30	4.27	74.58	58.74	37.23
65	3.43	17.23	4.50	73.99	57.42	36.95
70	3.44	17.10	4.40	74.30	58.53	36.93
75	3.42	17.04	4.00	73.86	56.98	36.63
80	3.42	17.18	5.01	74.53	59.30	36.68
85	3.39	17.25	5.20	74.50	59.09	37.35
90	3.38	17.34	5.15	75.20	58.79	37.11
95	3.38	17.43	5.91	74.69	58.53	36.22
100	3.36	17.52	5.97	74.85	58.26	35.99
105	3.32	17.59	5.55	74.98	58.24	35.95
110	3.35	17.60	6.57	74.56	56.29	35.42

Table D.2. Continued

Time Step	Exp Inlet	Ambient	Air Inlet	AirOutlet	Comp Inlet	Comp Outlet
seconds	°C	°C	°C	°C	barg	barg
0	24.11	24.41	35.53	27.61	2.27	11.43
5	22.97	25.00	35.64	27.53	2.25	11.50
10	22.61	24.67	35.60	27.85	2.23	11.54
15	23.42	25.12	35.56	27.92	2.23	11.55
20	22.26	24.72	35.67	27.86	2.26	11.55
25	23.47	24.87	35.55	27.62	2.29	11.54
30	23.20	24.82	35.74	27.33	2.30	11.54
35	23.87	24.57	35.60	27.91	2.34	11.54
40	23.52	24.52	35.74	27.71	2.34	11.55
45	21.55	24.36	35.72	27.91	2.32	11.55
50	23.75	24.34	35.69	27.70	2.28	11.53
55	23.55	24.30	35.80	27.58	2.25	11.52
60	23.07	24.42	35.66	27.38	2.25	11.53
65	22.08	24.36	35.83	28.69	2.22	11.54
70	20.83	24.42	35.70	28.12	2.24	11.53
75	24.12	24.22	35.74	27.99	2.27	11.54
80	22.16	24.21	35.70	27.72	2.27	11.55
85	22.99	24.44	35.60	27.83	2.29	11.55
90	23.11	24.21	35.72	28.64	2.33	11.54
95	23.22	24.16	35.53	28.05	2.34	11.53
100	22.65	24.29	35.66	27.85	2.31	11.49
105	23.14	24.37	35.50	27.56	2.28	11.46
110	24.65	24.40	35.53	27.39	2.25	11.49

D.3. MEASURED DATA OF 500 Nm³/h EXPERIMENTS

Table D. 3. The measured temperature and pressure values of 500 Nm³/h experiments

Time Step	Evap Inlet	Evap Outlet	Comp Inlet	Comp Outlet	Cond Inlet	Cond Outlet
seconds	**°C**	**°C**	**°C**	**°C**	**°C**	**°C**
0	2.38	16.32	11.67	82.97	68.34	43.40
5	2.27	16.31	11.61	82.48	67.87	43.82
10	2.19	16.41	11.66	81.96	68.69	44.03
15	2.07	16.63	11.46	81.85	66.99	44.08
20	2.03	16.93	11.13	82.30	67.31	43.80
25	2.06	17.14	11.23	82.40	67.14	43.29
30	2.16	17.29	11.27	82.68	65.04	43.02
35	2.29	17.23	11.58	82.24	66.92	42.69
40	2.43	16.97	11.42	82.21	67.24	42.70
45	2.53	16.57	11.22	82.23	67.68	42.89
50	2.47	16.29	11.74	82.60	67.61	43.29
55	2.36	16.13	11.35	82.91	67.81	43.75
60	2.26	16.19	11.18	82.60	67.79	44.07
65	2.11	16.37	11.18	82.02	67.06	44.14
70	2.02	16.56	11.04	81.65	66.24	43.99
75	1.96	16.78	11.01	81.74	65.98	43.58
80	1.95	16.94	11.12	82.02	66.24	43.17
85	2.03	17.05	11.26	82.27	66.09	42.85
90	2.23	16.97	11.10	82.54	66.15	42.52
95	2.40	16.53	11.08	82.52	65.82	42.49
100	2.48	16.11	11.48	82.27	66.03	42.73
105	2.43	15.85	11.33	82.20	66.40	43.08
110	2.35	15.72	11.22	82.34	65.51	43.62

Table D.3. Continued

Time Step	Exp Inlet	Ambient	Air Inlet	AirOutlet	Comp Inlet	Comp Outlet
seconds	**°C**	**°C**	**°C**	**°C**	**barg**	**barg**
0	24.11	24.41	35.11	27.91	2.27	11.43
5	22.97	25.00	35.09	28.77	2.25	11.50
10	22.61	24.67	35.17	28.47	2.23	11.54
15	23.42	25.12	35.06	27.44	2.23	11.55
20	22.26	24.72	35.22	27.71	2.26	11.55
25	23.47	24.87	35.11	27.57	2.29	11.54
30	23.20	24.82	35.22	27.87	2.30	11.54
35	23.87	24.57	35.15	27.43	2.34	11.54
40	23.52	24.52	35.18	27.63	2.34	11.55
45	21.55	24.36	35.18	27.87	2.32	11.55
50	23.75	24.34	35.09	27.87	2.28	11.53
55	23.55	24.30	35.18	28.00	2.25	11.52
60	23.07	24.42	35.03	28.02	2.25	11.53
65	22.08	24.36	35.12	27.66	2.22	11.54
70	20.83	24.42	34.97	27.28	2.24	11.53
75	24.12	24.22	34.98	27.15	2.27	11.54
80	22.16	24.21	34.94	27.34	2.27	11.55
85	22.99	24.44	34.86	27.53	2.29	11.55
90	23.11	24.21	34.99	28.12	2.33	11.54
95	23.22	24.16	34.86	27.54	2.34	11.53
100	22.65	24.29	34.98	27.56	2.31	11.49
105	23.14	24.37	34.85	27.89	2.28	11.46
110	24.65	24.40	34.94	28.22	2.25	11.49

APPENDIX E

RESULTS

E.1. RESULTS OF 250 Nm³/h EXPERIMENTS

Table E. 1. The results of 250 Nm³/h experiments

Time Step	$\dot{Q}_{Total}$	$\dot{Q}_{cond}$	$\dot{Q}_{comp-cond}$	COP (VS)	COP (hgbpv)
s	W	W	W	-	-
0	1156.59	-1336.19	-145.59	2.62	0.69
5	1117.73	-1294.81	-133.92	2.73	0.69
10	1114.62	-1292.12	-130.03	2.72	0.69
15	1127.40	-1309.05	-131.40	2.70	0.69
20	1149.06	-1335.03	-135.11	2.70	0.71
25	1171.54	-1360.67	-138.99	2.74	0.73
30	1155.51	-1339.92	-141.41	2.71	0.71
35	1149.98	-1334.71	-140.93	2.75	0.72
40	1156.70	-1340.34	-143.48	2.69	0.71
45	1120.21	-1294.24	-147.20	2.74	0.70
50	1110.04	-1287.58	-135.44	2.74	0.69
55	1150.14	-1337.26	-139.90	2.75	0.72
60	1170.17	-1364.31	-144.99	2.66	0.71
65	1154.25	-1345.35	-138.12	2.68	0.71
70	1120.62	-1307.45	-132.24	2.71	0.69
75	1156.63	-1347.30	-129.69	2.76	0.73
80	1149.17	-1340.63	-132.46	2.77	0.73
85	1153.20	-1344.67	-138.42	2.71	0.72
90	1173.98	-1365.53	-139.54	2.73	0.73
95	1158.54	-1348.52	-140.20	2.72	0.72
100	1179.07	-1369.92	-147.49	2.69	0.72
105	1198.50	-1391.69	-154.21	2.73	0.75
110	1171.36	-1357.16	-151.45	2.67	0.71

Table E.1. continued

Time Step	$\dot{m}_{ref-evap}$	$\dot{m}_{ref-comp}$	$\dot{m}_{ref-HGBPV}$	Irreversibility (VS)	Irreversibility (hgbpv)
s	kg/s	kg/s	kg/s	W	W
0	0.0068	0.0260	0.0192	235.12	597.11
5	0.0066	0.0260	0.0194	209.21	537.49
10	0.0066	0.0260	0.0194	216.50	545.14
15	0.0066	0.0260	0.0194	216.46	550.61
20	0.0068	0.0260	0.0192	220.06	549.21
25	0.0070	0.0260	0.0190	222.42	535.07
30	0.0068	0.0260	0.0192	220.38	550.37
35	0.0068	0.0260	0.0192	218.65	533.33
40	0.0069	0.0260	0.0191	222.55	553.83
45	0.0067	0.0260	0.0193	210.33	523.82
50	0.0066	0.0260	0.0194	212.56	535.69
55	0.0068	0.0260	0.0192	214.27	530.06
60	0.0069	0.0260	0.0191	226.80	561.60
65	0.0068	0.0260	0.0192	220.88	553.06
70	0.0067	0.0260	0.0193	211.67	541.66
75	0.0068	0.0260	0.0192	213.66	532.01
80	0.0068	0.0260	0.0192	209.14	518.22
85	0.0069	0.0260	0.0191	217.06	541.62
90	0.0070	0.0260	0.0190	218.59	540.76
95	0.0069	0.0260	0.0191	217.57	548.50
100	0.0070	0.0260	0.0190	223.07	555.22
105	0.0071	0.0260	0.0189	220.93	534.72
110	0.0069	0.0260	0.0191	222.75	560.72

Table E.1. continued

Time Step	nII (VS)	nII (hgbpv)	$\dot{W}_{comp-actual}$	$\dot{W}_{comp-eaxact}$	$\dot{W}_{Rev}$
s	%	%	W	W	W
0	78.25	20.53	-1683.01	-441.53	-345.51
5	81.82	20.72	-1617.81	-409.76	-335.26
10	81.03	20.55	-1617.67	-410.29	-332.48
15	80.80	20.66	-1633.22	-417.55	-337.39
20	80.88	21.20	-1622.10	-425.16	-343.88
25	81.75	21.89	-1598.81	-428.01	-349.91
30	81.18	21.28	-1623.66	-425.67	-345.58
35	82.14	21.51	-1594.73	-417.60	-343.03
40	80.46	21.23	-1632.09	-430.55	-346.43
45	82.02	21.04	-1592.51	-408.54	-335.09
50	81.69	20.65	-1604.00	-405.42	-331.20
55	82.24	21.53	-1600.06	-418.89	-344.50
60	79.59	21.27	-1648.87	-440.73	-350.76
65	80.33	21.13	-1637.01	-430.57	-345.87
70	81.17	20.77	-1616.38	-413.61	-335.73
75	82.69	21.73	-1594.17	-418.92	-346.41
80	82.99	21.80	-1580.20	-415.17	-344.55
85	81.29	21.46	-1609.71	-424.85	-345.37
90	81.85	21.90	-1606.56	-429.83	-351.83
95	81.43	21.48	-1616.66	-426.41	-347.22
100	80.68	21.65	-1633.20	-438.30	-353.64
105	81.76	22.36	-1607.70	-439.78	-359.55
110	80.15	21.37	-1644.18	-438.34	-351.34

E.2. RESULTS OF 400 Nm³/h EXPERIMENTS

Table E. 2. The results of 400 Nm³/h experiments

Time Step	$\dot{Q}_{Total}$	$\dot{Q}_{cond}$	$\dot{Q}_{comp-cond}$	COP (VS)	COP (hgbpv)
s	W	W	W	-	-
0	2455.18	-2788.61	-288.11	2.31	1.53
5	2493.70	-2858.20	-268.49	2.37	1.62
10	2441.56	-2751.39	-325.35	2.29	1.53
15	2424.07	-2745.70	-313.40	2.31	1.53
20	2452.00	-2777.47	-323.69	2.29	1.54
25	2454.73	-2805.31	-299.95	2.27	1.53
30	2536.23	-2891.99	-287.40	2.29	1.59
35	2417.50	-2764.26	-273.48	2.22	1.47
40	2476.39	-2832.50	-269.20	2.23	1.50
45	2450.09	-2797.98	-270.52	2.25	1.50
50	2476.95	-2823.59	-273.83	2.24	1.50
55	2522.62	-2871.74	-277.61	2.25	1.54
60	2519.84	-2864.62	-275.01	2.26	1.54
65	2379.46	-2681.83	-271.87	2.31	1.48
70	2426.14	-2756.40	-263.36	2.28	1.49
75	2449.03	-2756.43	-284.79	2.29	1.51
80	2478.30	-2827.70	-258.95	2.30	1.53
85	2437.45	-2779.23	-259.22	2.29	1.51
90	2345.03	-2668.63	-264.95	2.25	1.43
95	2382.71	-2703.94	-263.04	2.33	1.48
100	2445.73	-2769.06	-276.14	2.33	1.52
105	2454.53	-2776.84	-279.15	2.31	1.51
110	2487.03	-2776.23	-308.42	2.39	1.58

Table E.2. continued

Time Step	$\dot{m}_{ref-evap}$	$\dot{m}_{ref-comp}$	$\dot{m}_{ref-HGBPV}$	Irreversibility (VS)	Irreversibility (hgbpv)
s	kg/s	kg/s	kg/s	W	W
0	0.0152	0.0230	0.0078	647.59	800.33
5	0.0157	0.0230	0.0073	645.01	769.54
10	0.0154	0.0230	0.0076	659.87	800.01
15	0.0153	0.0230	0.0077	642.17	780.61
20	0.0155	0.0230	0.0075	668.15	805.84
25	0.0155	0.0230	0.0075	668.15	810.05
30	0.0160	0.0230	0.0070	687.42	822.62
35	0.0152	0.0230	0.0078	675.46	840.71
40	0.0155	0.0230	0.0075	694.77	854.50
45	0.0153	0.0230	0.0077	693.58	850.71
50	0.0154	0.0230	0.0076	687.46	844.88
55	0.0157	0.0230	0.0073	695.14	844.15
60	0.0157	0.0230	0.0073	696.64	845.91
65	0.0147	0.0230	0.0083	648.61	807.48
70	0.0150	0.0230	0.0080	679.22	836.03
75	0.0152	0.0230	0.0078	658.06	816.22
80	0.0153	0.0230	0.0077	678.75	829.77
85	0.0152	0.0230	0.0078	663.92	818.12
90	0.0146	0.0230	0.0084	652.08	827.91
95	0.0147	0.0230	0.0083	637.75	802.42
100	0.0150	0.0230	0.0080	657.75	814.72
105	0.0151	0.0230	0.0079	661.81	820.71
110	0.0152	0.0230	0.0078	629.28	772.99

Table E.2. continued

Time Step	nII (VS)	nII (hgbpv)	$\dot{W}_{comp-actual}$	$\dot{W}_{comp-exact}$	$\dot{W}_{Rev}$
seconds	%	%	W	W	W
0	68.53	45.19	-1609.74	-1061.35	-727.36
5	70.07	47.78	-1540.38	-1050.35	-735.95
10	67.62	45.18	-1593.04	-1064.23	-719.68
15	68.34	45.37	-1579.21	-1048.35	-716.48
20	67.41	45.40	-1589.95	-1070.90	-721.89
25	67.03	45.28	-1602.66	-1082.59	-725.66
30	67.60	46.92	-1596.61	-1108.05	-749.08
35	65.78	43.38	-1649.66	-1087.90	-715.62
40	65.81	44.33	-1651.70	-1112.65	-732.19
45	66.10	43.99	-1635.72	-1088.65	-719.59
50	66.20	44.41	-1650.24	-1107.20	-732.93
55	66.63	45.45	-1641.25	-1119.59	-745.94
60	66.65	45.36	-1640.04	-1116.11	-743.91
65	67.82	43.47	-1610.52	-1032.30	-700.11
70	66.83	43.72	-1626.01	-1063.56	-710.82
75	67.79	44.67	-1624.28	-1070.25	-725.57
80	67.61	45.07	-1618.24	-1078.75	-729.38
85	67.62	44.60	-1612.99	-1063.92	-719.38
90	66.54	42.13	-1643.31	-1040.60	-692.39
95	68.74	43.85	-1604.85	-1023.87	-703.77
100	68.66	44.83	-1608.39	-1050.13	-720.99
105	68.21	44.64	-1623.61	-1062.57	-724.80
110	71.02	46.85	-1575.49	-1039.34	-738.15

E.3. RESULTS OF 500 Nm³/h EXPERIMENTS

Table E. 3. The results of 500 Nm³/h experiments

Time Step	$\dot{Q}_{Total}$	$\dot{Q}_{cond}$	$\dot{Q}_{comp-cond}$	COP (VS)	COP (hgbpv)
s	W	W	W	-	-
0	2916.22	-3540.41	-308.86	2.07	1.81
5	2761.99	-3347.15	-293.70	2.08	1.73
10	2835.55	-3454.65	-273.70	2.10	1.80
15	2983.89	-3599.67	-321.66	2.09	1.88
20	2980.10	-3597.61	-321.83	2.06	1.84
25	2973.69	-3580.20	-324.49	2.07	1.84
30	2953.92	-3503.65	-372.71	2.07	1.82
35	3010.14	-3606.15	-329.37	2.12	1.89
40	2982.97	-3583.27	-319.65	2.11	1.87
45	2941.56	-3550.79	-307.91	2.10	1.84
50	2916.96	-3526.88	-316.35	2.09	1.83
55	2918.46	-3537.96	-320.75	2.06	1.81
60	2874.61	-3486.77	-310.62	2.05	1.78
65	2961.27	-3576.98	-322.67	2.07	1.85
70	2985.09	-3586.90	-333.49	2.08	1.87
75	3011.52	-3608.55	-342.00	2.08	1.88
80	2966.66	-3555.48	-335.40	2.08	1.84
85	2910.89	-3481.16	-335.91	2.09	1.81
90	2847.30	-3402.54	-332.65	2.08	1.76
95	2909.71	-3473.71	-348.30	2.09	1.80
100	2942.74	-3525.31	-344.59	2.10	1.85
105	2846.88	-3424.71	-326.07	2.09	1.79
110	2813.28	-3370.88	-345.89	2.07	1.76

Table E.3. continued

Time Step	$\dot{m}_{ref-evap}$	$\dot{m}_{ref-comp}$	$\dot{m}_{ref-HGBPV}$	Irreversibility (VS)	Irreversibility (hgbpv)
s	kg/s	kg/s	kg/s	W	W
0	0.0193	0.0220	0.0027	845.85	894.21
5	0.0183	0.0220	0.0037	797.40	861.26
10	0.0188	0.0220	0.0032	810.71	864.45
15	0.0198	0.0220	0.0022	859.75	898.32
20	0.0197	0.0220	0.0023	875.20	918.13
25	0.0195	0.0220	0.0025	868.23	914.55
30	0.0193	0.0220	0.0027	862.71	912.60
35	0.0196	0.0220	0.0024	857.59	899.31
40	0.0195	0.0220	0.0025	849.68	893.63
45	0.0193	0.0220	0.0027	842.63	890.16
50	0.0192	0.0220	0.0028	837.14	884.96
55	0.0194	0.0220	0.0026	854.64	901.82
60	0.0191	0.0220	0.0029	843.57	895.08
65	0.0197	0.0220	0.0023	861.18	902.12
70	0.0198	0.0220	0.0022	864.05	903.35
75	0.0199	0.0220	0.0021	872.45	911.22
80	0.0195	0.0220	0.0025	859.93	906.38
85	0.0190	0.0220	0.0030	842.54	897.25
90	0.0186	0.0220	0.0034	826.62	890.33
95	0.0190	0.0220	0.0030	839.69	893.78
100	0.0193	0.0220	0.0027	837.11	883.49
105	0.0188	0.0220	0.0032	814.30	870.04
110	0.0187	0.0220	0.0033	814.77	873.08

Table E.3. continued

Time Step	nII (VS)	nII (hgbpv)	$\dot{W}_{comp-actual}$	$\dot{W}_{comp-exact}$	$\dot{W}_{Rev}$
s	%	%	W	W	W
0	61.60	53.92	-1607.04	-1406.58	-866.52
5	61.85	51.49	-1593.92	-1327.00	-820.72
10	62.38	53.40	-1577.77	-1350.60	-842.56
15	62.07	55.87	-1587.13	-1428.59	-886.73
20	61.23	54.80	-1615.82	-1446.00	-885.43
25	61.56	54.61	-1618.26	-1435.79	-883.81
30	61.60	54.10	-1622.50	-1424.88	-877.77
35	62.88	56.11	-1594.25	-1422.58	-894.47
40	62.80	55.62	-1593.52	-1411.46	-886.39
45	62.36	54.70	-1598.05	-1401.67	-874.10
50	62.14	54.35	-1594.67	-1394.73	-866.72
55	61.06	53.73	-1613.82	-1420.16	-867.17
60	61.00	53.03	-1610.66	-1400.20	-854.13
65	61.54	55.09	-1597.12	-1429.79	-879.85
70	61.84	55.65	-1593.79	-1434.33	-886.95
75	61.88	55.88	-1601.05	-1445.86	-894.71
80	61.90	54.80	-1608.41	-1423.70	-881.34
85	62.00	53.64	-1612.21	-1394.82	-864.82
90	61.84	52.18	-1620.97	-1367.95	-845.88
95	61.97	53.56	-1614.16	-1395.13	-864.52
100	62.53	54.92	-1592.06	-1398.29	-874.33
105	62.17	53.10	-1592.97	-1360.46	-845.82
110	61.51	52.24	-1599.95	-1358.80	-835.83

APPENDIX F

RESULTS OF ELECTRICAL CONVERGENCE EFFICIENCY OF THE COMPRESSOR

Table F. 1. Electical convergence efficiency of the compressor

Time Step	**T_1**	**T_2**	**P_1**	**P_2**	$\dot{m}_{ref-comp}$
seconds	**°C**	**°C**	**barg**	**barg**	**kg/s**
0	6.33	75.93	2.29	11.55	0.02
5	7.99	75.44	2.27	11.61	0.02
10	7.17	76.39	2.25	11.65	0.02
15	7.92	76.62	2.26	11.67	0.02
20	7.97	76.90	2.29	11.67	0.02
25	7.57	76.87	2.31	11.66	0.02
30	6.16	75.48	2.33	11.66	0.02
35	5.11	76.06	2.36	11.66	0.02
40	4.55	75.64	2.37	11.67	0.02
45	5.27	75.83	2.34	11.67	0.02
50	4.65	75.82	2.30	11.65	0.02
55	4.68	75.63	2.28	11.64	0.02
60	4.31	75.33	2.27	11.65	0.02
65	4.55	74.73	2.25	11.66	0.02
70	4.44	75.05	2.26	11.65	0.02
75	4.04	74.59	2.29	11.65	0.02
80	5.06	75.28	2.30	11.67	0.02
85	5.25	75.24	2.31	11.67	0.02
90	5.20	75.95	2.36	11.66	0.02
95	5.97	75.43	2.37	11.64	0.02
100	6.03	75.60	2.33	11.60	0.02
105	5.60	75.73	2.30	11.58	0.02
110	6.64	75.31	2.27	11.60	0.02

Table F.1. continued

Time Step	I	V	$\dot{W}_{comp-electrical}$	$\dot{W}_{comp-actual}$	η_{ee}
seconds	**Amps**	**Volts**	**W**	**W**	**%**
0	6.355	220.0	1398.1	1020.65	73.0
5	6.082	224.7	1366.7	976.75	71.5
10	6.290	226.1	1422.0	1010.10	71.0
15	6.235	226.4	1411.6	1001.35	70.9
20	6.277	225.6	1416.0	1008.14	71.2
25	6.327	224.8	1422.4	1016.19	71.4
30	6.304	224.4	1414.8	1012.36	71.6
35	6.513	223.7	1456.7	1045.95	71.8
40	6.521	223.7	1458.6	1047.24	71.8
45	6.458	224.1	1447.5	1037.12	71.6
50	6.515	224.8	1464.5	1046.31	71.4
55	6.480	225.1	1458.8	1040.62	71.3
60	6.475	225.5	1460.3	1039.86	71.2
65	6.358	226.3	1438.8	1021.16	71.0
70	6.419	225.7	1449.0	1030.97	71.1
75	6.413	225.1	1443.7	1029.88	71.3
80	6.389	225.3	1439.7	1026.05	71.3
85	6.368	224.9	1432.5	1022.73	71.4
90	6.488	223.6	1451.0	1041.93	71.8
95	6.336	223.1	1413.6	1017.57	72.0
100	6.350	222.8	1414.8	1019.80	72.1
105	6.410	223.0	1429.5	1029.44	72.0
110	6.220	224.4	1395.5	998.97	71.6

APPENDIX G

SCHEMATIC VIEW OF TESTING SECTION

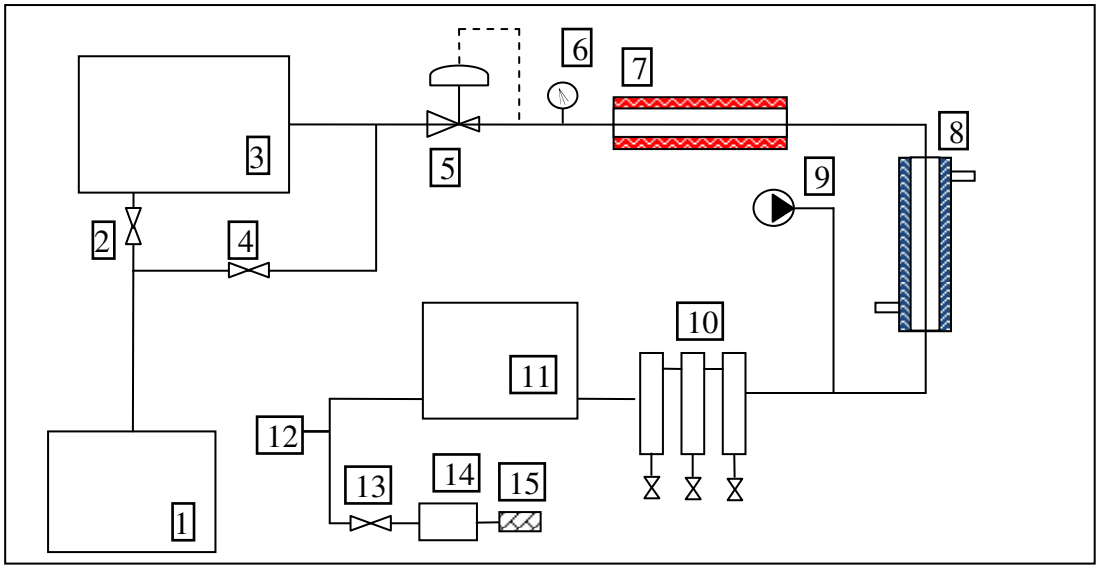

Figure G. 1. Testing section

Table G. 1. Testing section components

Item	Description	Capacity
1	Air Compressor	2100 Nm³/h – 250 kW
2	Manual Valve	3”
3	Storage Tank	3 m³
4	By-Pass Valve	3”
5	Pressure Regulator	8 – 1 barg

(Table G.1. continued)

6	Manometer	10 barg
7	Electrical Heater	40 kW max
8	Air Cooling Heat Exchanger	40 kW max
9	Water Spray Pump	100 gr/s max
10	Liquid Water Separator	
11	Refrigerant Air Dryer	See Figure 21 for details
12	Dew Point Sensor	-20 to 40 °C T_{dp}
13	Flow Rate Control Valve	3”
14	Flow Meter with Monitor	3000 Nm^3/h
15	Silencer	