MARMARA UNIVERSITY

INSTITUTE FOR GRADUATE STUDIES IN PURE AND APPLIED SCIENCES

EXPERIMENTAL ANALYSIS AND PERFORMANCE EVALUATION OF A CASCADE REFRIGERATION SYSTEM USING CO_2 AS REFRIGERANT

ALİ ÖZYURT

MASTER THESIS

Department of Mechanical Engineering

Thesis Supervisor

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Ali ÖZYURT, a Master of Science student of Marmara University Institute for Graduate Studies in Pure and Applied Sciences, defended his thesis entitled "**Experimental Analysis and Performance Evaluation of a Cascade Refrigeration System Using CO2 as Refrigerant**", on July 8th 2015 and has been found to be satisfactory by the jury members.

Jury Members

Asst. Prof. Barış YILMAZ (Advisor)

Marmara University

Assoc. Prof. Ebru MANÇUHAN (Jury Member)

Marmara University

Prof. Dr. Galip TEMİR (Jury Member)

Yıldız Technical University

APPROVAL

Marmara University Institute for Graduate Studies in Pure and Applied Sciences Executive Committee approves that Ali ÖZYURT be granted the degree of Master of Science in Department of Mechanical Engineering/Mechanical Engineering Program on 22.07.2015 (Resolution no: 2015/15-02

Director of the Institute

Prof. Dr. Uğur YAHŞİ

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Ali ÖZYURT

June, 2015

Istanbul, TURKEY

TABLE OF CONTENTS

ÖZET

SOĞUTUCU AKIŞKAN OLARAK CO2'İN KULLANILDIĞI BİR KASKAT SOĞUTMA SİSTEMİNİN DENEYSEL ANALİZİ VE PERFORMANS DEĞERLENDİRMESİ

Günümüzde soğutma sanayisinde kullanılan sentetik soğutkanların ozon tabakasında neden olduğu negatif etkilerden ötürü, bu soğutkanların kullanımının kısıtlanması veya tamamen yasaklanması gündemdedir. Sentetik soğutkanlarla çalışan soğutma sistemlerinden yüksek oranda sızıntıların olması sera gazlarının atmosfere yüksek oranda salınmasına neden olmaktadır. Bu sebeple alternatif soğutkanlar için araştırma yapma ihtiyacı zorunlu hale gelmiştir.

Bu çalışmada, ülkemizin iklimine uygun olarak karbondioksitin 2 basamaklı kaskat sistemde kullanıldığı bir soğutma sisteminin tasarımı, geliştirilmesi ve üretimi yapılarak CO2'in akışkan olarak kullanıldığı bir soğutma sistemi üzerinde deneysel çalışmalar gerçekleştirilmiştir. Üretilen sistemde optimum çalışma şartları oluşturularak yapılan ölçümlerin sonuçları değerlendirilmiş ve Türkiye iklim şartlarına uygun sistemlerin üretiminin gerçekleştirilmesi sağlanmıştır.

Gerçekleştirilen deneyler sonucunda evaporatör ve sistem kapasitelerinin katalog datalarına uygun ve ekipman verimliliklerinin invertörlü kullanımla birlikte standardın üstünde değerlerde olduğu görülmüştür. Sistem henüz test aşamasındadır ve performans ve verimlilik seviyelerinin daha yukarı çekilerek piyasada rekabetçi ve tercih edilebilir olması sağlanacaktır.

ABSTRACT

EXPERIMENTAL ANALYSIS AND PERFORMANCE EVALUATION OF A CASCADE REFRIGERATION SYSTEM USING CO2 AS REFRIGERANT

Synthetic refrigerants are being called off or laid up because of their negative effects on the ozone which are used in refrigeration industry nowadays. Refrigeration systems using synthetic compounds as a refrigerant are caused large amount of greenhouse gases emission to atmosphere due to the leakage in the systems. Therefore, proposing less harmful and environmental friendly alternative refrigerants for such systems is obligatory.

In this study, it is planned to design, develop and produce a two-stage cascade refrigeration system which is suitable for the climate of our country. Thus, such a refrigeration system with carbon dioxide as refrigerant is produced. Testing the developed system at optimum operating conditions made it possible to design and suggest a refrigeration system for climate zones of Turkey.

The performed experiments realized that this system met the capacity requirements and gave parallel results with the catalogue data of the equipment. Efficiency levels of the components are higher than standard due to controlling with invertors. This system is still under development and with minor changes and much more experimentation to get higher coefficient of performance and efficiency values that system could be competitive and preferred in the industry.

SYMBOLS

comp,HTC: High temperature cycle compressor

comp,LTC: Low temperature cycle compressor

fan: energy consumption by fans

h: enthalpy (kJ/kg)

I: Amperes (A)

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

$\dot{Q}$ **:** Heat transfer rate (kW)

P: Power (kW)

Total,HTC: Total energy consumption of high temperature cycle

Total,LTC: Total energy consumption of low temperature cycle

V: Volts (V)

$\dot{W}$: Power consumption (kW)

φ : Phase angle (^{0})

ABBREVIATIONS

AHU – Air Handling Unit

CFC – Chlorofluorocarbons

COP – Coefficient of Performance

DX – Direct Expansion

EEV – Electronic Expansion Valve

EPA – Energy Protection Agency

EVD – Expansion Valve Driver

GWP – Global Warming Potential

HCFC – Hydro Chlorofluorocarbon

HFC – Hydro Fluorocarbon

HFO – Hydro Fluoro Olefin

HTC – High Temperature Cycle

LCCP – Life Cycle Climate Performance

LT – Low Temperature

LTC – Low Temperature Cycle

MTC – Medium Temperature Circuit

ODP – Ozone Depletion Potential

PHE – Plate Heat Exchanger (Cascade Condenser)

PID – Proportional – Integral – Derivative

PLC – Programmable Logic Circuit

TXV – Thermostatic Expansion Valve

LIST OF FIGURES

LIST OF TABLES

1. INTRODUCTION

Pursuit of air-conditioning comfort has been started from the first ages of mankind by setting up fans in front of the ice blocks, wind blowing to the living spaces through rain water holders and come until today with an increasing technological knowledge over the refrigeration systems.

Refrigerants hold the most important place in the refrigeration by being fuel of the refrigeration systems. We can separate refrigerants as natural and synthetic refrigerants with their composition characteristics.

Synthetic refrigerants are started to use from mid-19th century. Because of their refrigeration performance and safety perspectives, synthetic refrigerants are the most common refrigerant used in the refrigeration industry. Refrigeration systems using synthetic compounds as refrigerant are caused large amount of greenhouse gases to the atmosphere due to the leakage in the systems. Therefore, proposing less harmful, environmental friendly and alternative refrigerants for such systems is obligatory.

Carbon dioxide is a natural refrigerant using in the refrigeration systems and known since the beginning of the last century. Because of its low Global Warming Potential, it is called to be environmental friendly refrigerant. Global Warming Potential of Carbon dioxide is equal to the reference value of 1. It has advantageous thermodynamic properties like low viscosity and low critical temperature. Therefore, Carbon dioxide has a high potential refrigerant regarding new energy efficient applications. It has several advantages compared to other refrigerants because it is a non-toxic, non-explosive and environmental friendly gas. In addition, it can be used in wide range of operating temperatures. Carbon dioxide also allows reducing the size of the units in the refrigeration systems due to its low specific volume and high operating pressure values.

There are various possibilities of using Carbon dioxide efficiently as a refrigerant in centralized cooling systems. It might be used as refrigerant in the single stage trans -

critical refrigeration cycle or secondary refrigerant in the low temperature stage of the cascade systems.

In several studies, it is suggested that Carbon dioxide should be used as a refrigerant in the low temperature stage of the cascade systems in warm climates and it should be used as a working fluid in single stage trans - critical refrigeration cycles in cold climates.

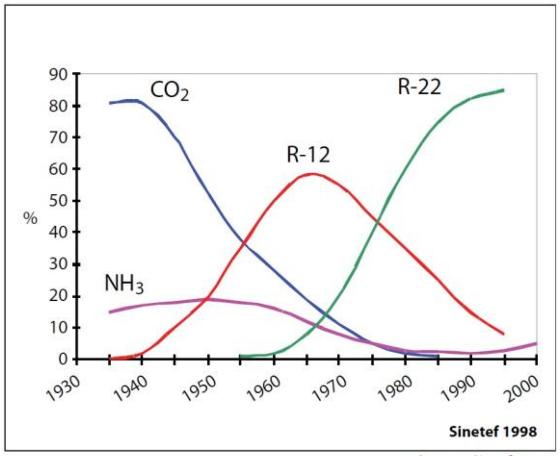
1.1. History of CO_2 as Refrigerant

In late 1800s CO2 started to use and spread in almost all refrigeration areas. Along with ammonia and sulfur dioxide are toxic, flammable and legally restricted to use in refrigeration systems, CO2 refrigeration systems are started to use in wide range of applications like supermarkets, hospitals, air conditioning of trains and passenger ships. This period is continued to mid 1930s and by the production of synthetic refrigerants, application range is started to reduce and finally at the end of 1980s CO2 refrigeration systems are almost removed from refrigeration industry as you can see from the Figure 1.1 below. High working pressure and low cooling capacity at high ambient temperatures are the main causes for phasing out of CO2 from the industry. Because synthetic refrigerants are enabled to use low cost heat exchangers in the system and CO2 systems have no competitive component and system solutions, demand in refrigeration industry transacted to synthetic refrigerants.

In recent years, chlorofluorocarbons (CFCs), hydro chlorofluorocarbons (HCFCs) and their derivatives are used in wide range of applications in commercial refrigeration. By the signing of Montreal Protocol in 1996, CFCs are banned from using and HCFCs are started to phase out from using because of their negative effects on environment and especially on ozone layer. R-12 and R-502 refrigerants are immediately banned from using and R-22 is phased out with a 25% rate from application by the end of 2010.

After CFCs are banned and HCFCs are phased out, commercial refrigeration industry started to use hydro fluorocarbons (HFC) like R-404, R-507, R-410 and R-134. Ozone Depletion Potential (ODP) of HFCs is considerably low but their Global Warming Potential (GWP) is extremely high. Hydro fluoro olefins (HFOs) have considerably low

GWP values which are commercially a new refrigerant in the industry. HFOs are structurally HFCs but to remark their low GWP values they called as HFOs in the industry.



Source: Sintef, 1998

Figure 1.1 Historical Use of Refrigerants

Using wide range of application of low ODP refrigerants in refrigeration industry is caused an increase in researches about reducing GWP of these refrigerants. GWP values of some common refrigerants using in the refrigeration system is shown in Table 1.1. In refrigeration systems global warming potential is effected directly by leakages and indirectly CO2-equivalent emissions corresponding to energy demand of the system. Global warming is mostly effected by indirect emissions not by direct emissions. Researchers have developed a model which is calculating greenhouse gas emission rate of a refrigeration system by adding both direct and indirect effects. Program is called as Life-Cycle Climate Program (LCCP) and it calculates the greenhouse emission of the system over its entire life time.

A standard refrigeration system can hold approximately about 2,500 kg of refrigerant and by the statistical investigation that U.S Environmental Protection Agency (EPA) average 23.5% of the refrigerant is leaked to the atmosphere by the refrigeration industry in 2008. Considering the amount of refrigerant leaked, some legal and financial restrictions have to be applied over the public companies. For that issue the parameter will be the carbon footprint and a standard level will be determined. Every system that has carbon footprint value above the limit the manufacturer will be faced with both legal and financial restrictions. This will be the driving force to phase in to the natural refrigeration system in the refrigeration industry over entire globe.

Table 1.1 ODP and GWP for Various Refrigerants

REFRIGERANT	TYPE	ODP	GWP
R-12	CFC	0.82	10600
R-22	HCFC	0.034	1700
R-404A	HFC	0	3800
R-410A	HFC	0	2000
R-290 (Propane)	Natural	0	20
R-717 (Ammonia)	Natural	0	<1
R-744 (CO2)	Natural	0	1
HFO-1234yf	HFO	0	4

A fluid which is produced by nature's bio-chemical processes or existed freely in the nature is called natural fluid. These fluids have zero or low effect on the environment. That is why these fluids are the alternatives for CFCs, HCFCs and HFCs. Air, water, ammonia, hydrocarbons and carbon dioxide are the main common natural refrigerants.

Air is using in different gas power cycles without any phase change and approach to considerably low temperatures. As far as we know that the power cycle is Brayton, to reach theoretical efficiency is quite difficult so this issue is restricted the application range of air as refrigerant.

Water is used as a vapor in gas power cycles with large capacity machines. The main problem of water as a refrigerant is its thermodynamic properties such as high freezing point temperature which quite difficult to use water at refrigeration systems below 0 ^{0}C. That is the reason water is not an appropriate alternative for the refrigerants with high ODP and GWP.

Ammonia, hydrocarbons and CO2 is the most common natural refrigerants using in the industrial and commercial refrigeration systems. Hydrocarbons have very high flammability levels. This disadvantage makes hydrocarbons hard to use unless necessary safety regulations and standards have been handled across whole facility that the refrigeration system will work. Because of the safety regulations; hydrocarbons are not suitable for water chillers, industrial, commercial and domestic refrigeration systems. Ammonia on the other hand with lower flammability level is the most appropriate natural refrigerant for the industrial refrigeration systems. In industrial systems smell and toxicity is not a disadvantage so these conditions make ammonia ideal for this kind of applications. But toxicity and smell is prohibiting ammonia from use in domestic and commercial refrigeration systems. The only refrigerant left for wide range of applications like automotive industry, commercial and domestic refrigeration applications etc. is CO2 as an ideal selection.

Low ozone depletion and low global warming potential refrigerant need is increasing day by day. So interest on natural refrigerants like ammonia, hydrocarbons and carbon dioxide is also increasing. CO2 is the best alternative for HFC refrigerants in the low temperature applications because of its non-vacuum thermodynamic properties under -40 ^{0}C with high efficiency with respect to the synthetic refrigerants. Because CO2 is a non-toxic and non-flammable gas, it has a high safety degree corresponding to both natural and synthetic refrigerants. CO2 is non ozone depleting refrigerant and its global warming potential is 1 which is the base level. CO2 is a low cost refrigerant comparing to hydrocarbons and HFCs. Beside its physical advantages, CO2 has high volumetric cooling capacity which makes CO2 as the best refrigerant not just only the low temperature applications but for the all kinds of refrigeration applications where cooling demand occurs. All these advantages make CO2 as the ideal alternative refrigerant for the refrigeration industry.

CO2 is very different refrigerant among others with respect to its distinctive properties. The most important one is being high pressure levels at ambient conditions. At sub-critic levels volumetric capacity is similar to other refrigerants. As you can see in Figure 1.2, CO2 has 5.2 bars pressure level at -49.2 ^{0}C which is the solidification starts afterwards and at 31 ^{0}C it has a pressure value of 72 bars.

After 31 ^{0}C, CO2 is reaching its super-critical phase and liquid phase cannot be distinguished from its gas phase. Also at this level liquefaction of gas phase is no longer valid. A Trans-critical CO2 system which is most common solution can operate in this super-critical range. Because of liquefaction problems trans-critical system solutions are more efficient when operate in sub-critical region which requires average ambient temperature level under 31 ^{0}C.

The other important property of CO2 is its high volumetric cooling capacity which means at same volume levels of refrigerants CO2 can absorb more heat than all the other refrigerants in the industry. This property comes with the advantage of less refrigerant amount required for CO2 comparing to other refrigerants for the same cooling effect. The most important problem again is its high pressure levels in standard refrigeration range. This must requires a good consideration of pressure levels of equipment used in the system. **[1]**

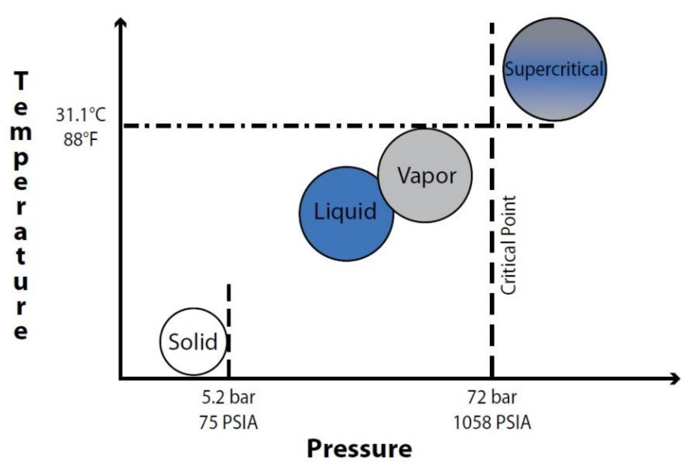


Figure 1.2 CO_2 Temperature – Pressure Characteristics

1.2. CO2 System Solutions

Structure of CO2 refrigeration systems has no difference from structure of conventional refrigeration systems. The only important consideration is the high operating pressure levels of the system and compatibility of the equipment used. There are two main solutions to every refrigeration system which are non – cascade and cascade system solutions. In the Table 1.2 below; the compatibility of different CO2 refrigeration system solutions is shown.

1.2.1. Direct expansion

The most common and simplest refrigeration cycle design is Direct Expansion refrigeration and it consists of compressor, condenser, flow control device and an evaporator which are basic refrigeration components and you can see in Figure 1.3. Because of high operating pressures like 62 bars at room temperature of 25 ^{0}C, direct expansion system is not a suitable solution for CO2 refrigeration.

Table 1.2 Compatibility of Different System Solutions for CO2

System Type		Compatibility
Non-Cascade	Direct Expansion (DX)	NO
	Trans-critical	YES - but average ambient temperature should not be exceed 31 ^{0}C
Cascade	Sub-critical DX	YES
	Secondary Loop	YES

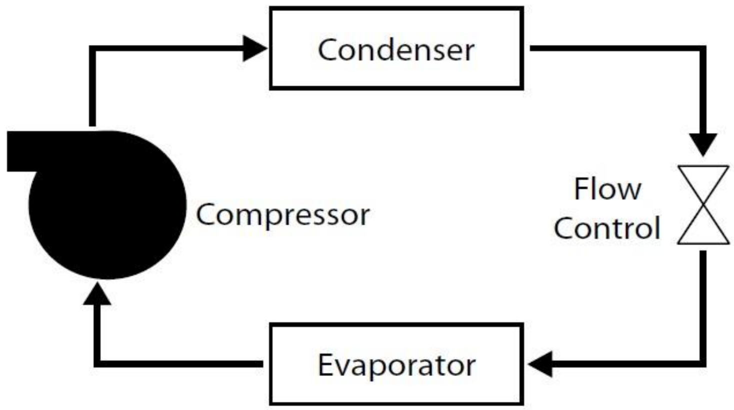


Figure 1.3 Direct Expansion Refrigeration Cycle

Trans – critical refrigeration system for CO2 operates over triple point temperature of 31 ^{0}C. When thermodynamic properties of CO2 are considered operating pressure of the CO2 system is much higher than those HFC refrigeration systems. At this level of operation condenser is no longer works for condensation of a gas but works as a gas cooler to cool of the gas as possible as low because there is no difference between liquid and gas phase of CO2 at super – critical conditions.

The most important advantage of such system solution for CO2 is enabling a totally environmental refrigeration system for various applications. Considering direct effect on global warming trans – critical CO2 refrigeration system solutions is the best solution for total elimination of the problem.

Design cost increases while diluting the high operating pressure effect on the trans – critical systems. Other disadvantage of the trans – critical CO2 systems is low efficiency over at high ambient temperatures. As average temperature is generally much higher over the world explains why trans – critical CO2 systems are not common. Trans – critical system solution is best where the ambient temperatures below 31 ^{0}C and the system operate as direct expansion systems. At sub – critical levels system has more efficiency with respect to the other operating conditions.

1.2.2. Cascade systems

A two – stage cascade refrigeration system uses two pairs of cycles, working individually with different refrigerants connected with a heat exchanger called as ‘’cascade condenser’’. The stages of two – stage cascade refrigeration system are; high temperature cycle and low temperature cycle. Refrigerant used in low temperature cycle has lower boiling point to ensure the corresponding cycle characteristic. Cascade condenser evaporates the high temperature cycle refrigerant to condensate low temperature cycle refrigerant. Cascade condenser enhances thermal coupling between two cycles of the system. The schematic of the two – stage cascade refrigeration system is showed in Figure 1.4a.

The lnp – h diagram of the referred cascade refrigeration system in Figure 1.4a illustrated in Figure 1.4b. As shown, using cascade system reachable evaporation temperature is much lower than a conventional refrigeration system could reach. Operating range of high temperature cycle of the cascade system is similar to conventional DX refrigeration system. As the system getting lower in lnp – h diagram, cooling capacity increases so in necessary conditions cascade refrigeration system is the best solution.

Cascade refrigeration system solution has a wide range of application such as; liquefaction of industrial gases or petroleum products, deep freezing etc. As the most important advantage; since pair of refrigerants used in system at different operating ranges for different cooling capacities could be achieved with several refrigerants. With the thermodynamic properties in the lower stage of the system temperatures below – 80 ^{0}C could be reached.

To enhance proper heat transfer rate inside the cascade condenser, capacity of the heat exchanger should be greater than the compressor cooling capacity of the high temperature cycle as cascade condenser is the evaporator of this cycle. Also an optimum difference should be designated between evaporation temperature of high temperature and condensation temperature of low temperature cycle of the cascade system which is a disadvantage as efficiency of the system decreases. Superheating in the cascade refrigeration system is a condition dependent variable which is changing with respect to the demand and operation location. Generally superheating level of the system should be arranged as liquid return to the compressor is inhibited. [**2]**

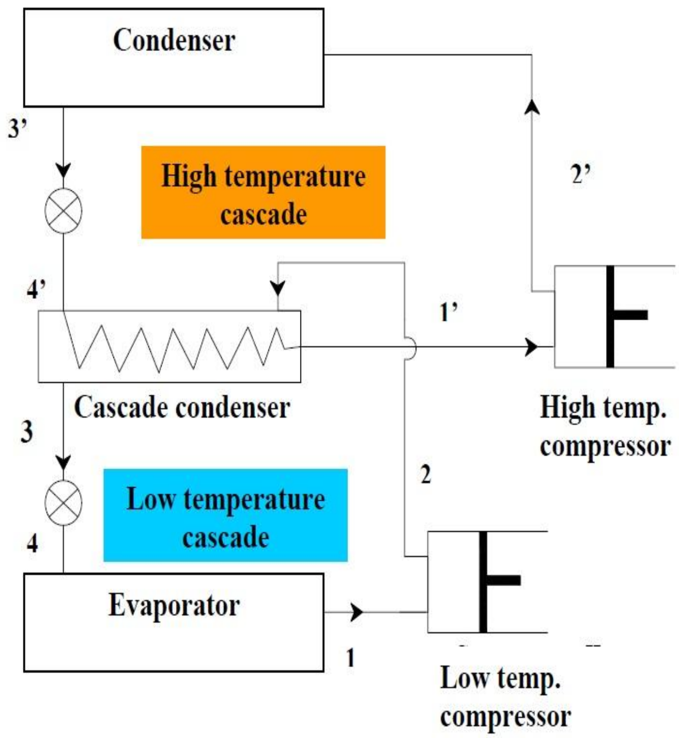


Figure 1.4a A Cascade Refrigeration Cycle

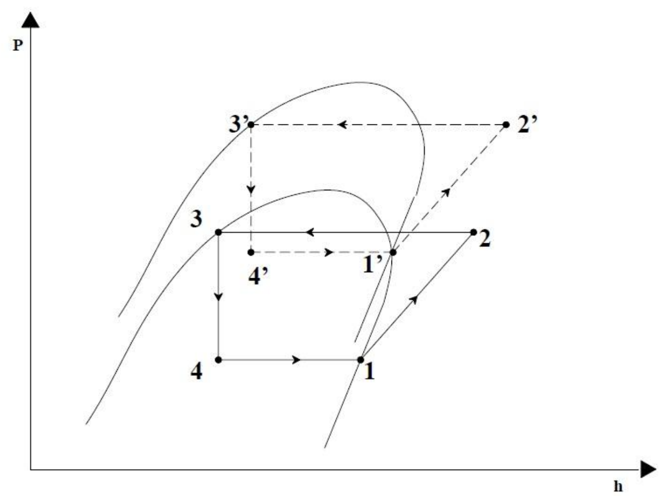


Figure 1.4b LnP-h Diagram of a Cascade Refrigeration Cycle

1.2.2.1. Cascade subcritical direct expansion

A sub – critical direct expansion cascade refrigeration system which used working fluid of R404a in high temperature cycle and CO2 in low temperature cycle is shown in Figure 1.5 for conjunction of medium and low temperature applications.

The most important advantage of this system solution is refrigerant charge reduction to the system. As it is operating in the sub – critical range of CO2 standard refrigeration components should be used. The operating pressure level is much lower than the standard trans – critical refrigeration system. So safety precautions to handle high pressure level are no longer necessary.

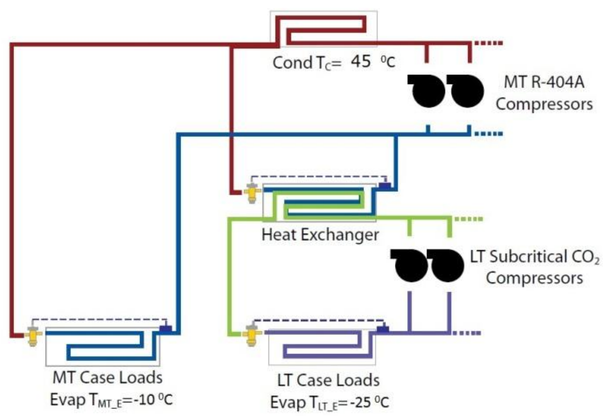


Figure 1.5 Subcritical CO2 Cascade – (DX)

1.2.2.2. Secondary loop refrigeration (CO2 Brine Refrigeration)

Schematic of refrigeration system solution that CO2 is circulated as in liquid phase over the entire low temperature cycle is shown in Figure 1.6. This is also called as secondary loop or brine refrigeration system. The distinctive property of such system is using pump to circulate CO2 in the system instead of compressor. This system solution is generally used in applications which have evaporation temperatures of -10 ^{0}C or above. In the evaporator, refrigerant could not fully evaporate and mixture of liquid and gas refrigerant returns to flooded cascade condenser to supply saturated or sub – cooled liquid refrigerant to the pump for avoiding gas return.

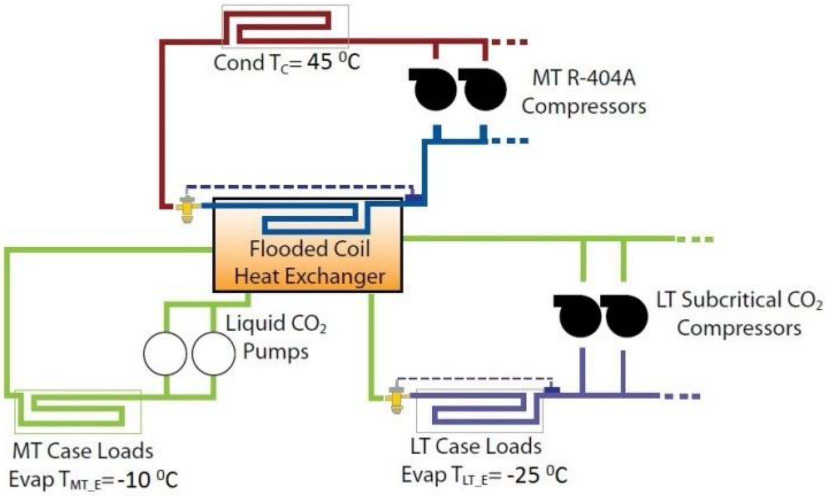


Figure 1.6 Secondary Loop CO_2 System

1.3. Benefits

Using CO2 as a refrigerant comes with several advantages – most of them are mentioned earlier – which are;

- CO2 has a high volumetric capacity which concluded with the advantage of pipe and compressor size reduction. Compressor and pipe size reduction enables the shorter pipelines, reduced installation costs and high energy saving potentials with a compact design.
- CO2 allows inhibiting or reducing the charging levels of high GWP refrigerants to systems. So direct effect of global warming is reduced or inhibited by reducing the leak rates of refrigerants to the atmosphere.
- With a minimum GWP value of 1; CO2 is so called environmental friendly refrigerant.
- Since high GWP refrigerants are synthetic which means that they produced by controlled chemical reactions, purchasing costs are higher than CO2 which is easily available in the atmosphere.

1.4. Literature Review

Synthetic refrigerants are being called off or laid up because of their negative effects on the ozone which are used in refrigeration industry nowadays.

Annual refrigerant leakage rate approaches to 14% according to a detailed research committed over 220 supermarkets in Norway. This information shows that, industry affects the refrigerant emissions in the atmosphere at high rates **[3]**. At these circumstances, releasing of green house gasses to the atmosphere is at improving rates and searching over the alternative refrigerants and reducing annual leakage rate is being obligatory. For example; Sweden government had banned the refrigeration systems using CFCs as refrigerant and had stopped production of these refrigerants between 1995 and 2000. Also, production of refrigeration systems using HCFCs refrigerants was stopped after 1998 and refilling of these refrigerants to the existing HCFC refrigeration systems.

Carbon dioxide is a well-known and natural refrigerant using since the beginnings of 20^{th} century. CO_2 is known as environmental friendly refrigerant because of its low GWP. GWP of CO_2 is equal to the reference value of 1, because used CO_2 is already produce by other processes or sources so no need to produce again as refrigerant. CO_2 stands out as a high potential refrigerant considering new efficient energy products. Also CO_2 has distinctive thermodynamic properties like; low viscosity, high operating pressure and low critical temperature.

Considering the centralized systems; there are three possibilities of using CO_2 as refrigerant:

- As primary refrigerant in single stage trans-critical refrigeration systems
- As secondary refrigerant or primary refrigerant in Low Temperature Circuit (LTC) of a cascade refrigeration system
- As primary refrigerant in LTC and Medium Temperature Circuit (MTC) of a cascade refrigeration system.

Lorentzen [**4]** and Pearson **[5]** showed that CO_2 has high potential in Low Temperature (LT) application in their studies. CO_2 has distinctive application advantages with respect to other refrigerants such as; environmental friendly, non-toxic, non-explosive, available and applicable in wide range of temperature. Also CO_2 has such an advantage of being compatible with the lubricators used in refrigeration systems **[6]**. CO_2 application enhances to reduce size of the refrigeration system equipment combining of low specific volume and high pressure values **[7]**. Although high operating pressure value is known as a disadvantage for a refrigerant, EURAMMON **[8]** clears that; with the rising operating pressure of CO_2 system gains some advantages. The compressors are relatively small and so is the pipe work. The gas is very dense and a result gives good transport and heat transfer performance. The system efficiency is relatively unaffected by pressure drop because the saturation temperature profile is so flat. It is fair to say that high operating pressure of CO_2 offers opportunities to create significantly more efficient systems than the current best practice provided the whole system is designed to capitalize on these advantages.

Lorentzen **[4]** clears that using CO_2 in trans-critical and cascade refrigeration systems is fairly acceptable and CO_2 has relatively good Coefficient Of Performance (COP) values comparing to commonly used synthetic refrigeration systems.

CO_2 cascade refrigeration systems are recommended as best option for supermarket refrigeration industry which uses food cabinets in the -30 ^{0}C to -50 ^{0}C variable evaporation temperature range **[9]**.

Cason **[10]** analyzed COP values of CO_2 at the range of -10 ^{0}C and -30 ^{0}C evaporation temperatures in MTC and LTC of refrigeration system respectively. Using experimental and numerical analyses of the study shows that COP values of CO_2 reduces with increasing ambient temperatures.

Girotto **[11]** investigates the efficiency of CO_2 based refrigeration systems and compare the results with R404A refrigeration system. In this study, the efficiency of the MTC refrigeration system is relatively low with respect to R404A refrigeration system efficiency.

Eggen and Aflekt **[12]**, Pearson and Cable **[13]** and Van Riessen **[14]** are presented applicable examples of cascade refrigeration systems on refrigeration supermarkets. Eggen and Aflekt **[12]** developed a prototype of a cascade refrigeration system for air-conditioning of a supermarket in Norway. Van Riessen [**14**] studied on the economic and technical feasibility of a refrigeration system based on Netherlands. Sawalha **[15]** and Likithammanit **[16]** designed and produced a CO_2/NH_3 cascade refrigeration systems for mid-range supermarkets in Sweden.

Belozerov **[17]** investigated the effect of CO_2 evaporation temperature on COP of the cascade system. Getu and Bansal **[9]** published thermodynamic analyses of cascade system. Dopazo **[18]** analyzed the effect of design and application conditions on the COP of the cascade system theoretically including compressor efficiency. Bingming **[19]** got experimental results on CO_2/NH_3 refrigeration systems using screw compressor.

Sawalha **[20]** established a system to calculate the system performance of two major CO_2 trans-critical refrigeration systems using numerical method. The system includes two separate, centralized parallel and at medium temperature nutrition tanks accompany with direct expansion cycle. One of these nutrition tanks feeding medium temperature level cabinets and the other is feeding LTC freezers. Two stage cascade refrigeration system reaches its maximum COP value at 10 ^{0}C to 40 ^{0}C variable climate conditions. In this study, practitioner concluded that NH_3/CO_2 cascade refrigeration system is applicable on warm climate zones (except parallel ones) and refrigeration system using CO_2 as primary refrigerant is applicable on cold climate zones.

Cavallini **[21]** analyzed the potential development, theoretical and experimental studies of a two-stage cascade refrigeration system which contains two compression stages including sub-cooling between these compressions and working at the standard refrigeration conditions. This study is unique and lead with respect to its topic and publishing time. Gathered experimental results show that it is possible to obtain maximum energy efficiency with optimum pressure value at the upper cycle of the tras-critical application. Also the effects of the refrigerant temperature at lower cycle on the system performance were investigated. The system performance is increased 25% by optimization of the experimental results using a numerical analysis.

S.G. Kim **[22]** analyzed experimental study, performance evaluation and simulation of an auto-cascade refrigeration system. A zeo-tropical compound – combination of CO_2, NH_3, R134a and R290 – is used in experimental study. Study aims to investigate the effects of different zeo-tropical compounds (combination of different ratios of different substances) on the system performance. On the study, this system has some several advantages as low compound volume and low operating pressure with respect to the simulation and experimental studies. But there is a major disadvantage which is lower COP of the auto-cascade refrigeration system compared to the cascade refrigeration systems. This disadvantage makes the auto-cascade refrigeration systems unlikely and need improved overall performance.

Acül **[23]** stated that because of CO_2 has low critical point temperature, compression line of the system has some difficulties working over the critical point temperature. These difficulties could be solved by using cascade system applications or doing optimization studies on gas cooler applications at a temperature over the critical point. That's why appropriate selection of the system with respect to design considerations is important. Trans-critical and sub-critical systems are working at high efficient levels on the relatively warm and cold climate zones. It is emphasized that using cascade cycles is the most efficient application on the hot climate zones. This study involves software and development studies which is necessary to design the most important equipment of a refrigeration system; gas coolers and evaporators. Software studies are supported by the prototype tests in the laboratory. As a conclusion; a software program developed which provides designing of finned-tube CO_2 heat exchanger.

Bansal **[24]** declared that CO_2 is more appropriate refrigerant for LTC applications compared to other systems and more advantageous compared to other refrigerants. Heat transfer coefficient values of CO_2 is calculated in the experiments made for the flow boiling heat transfer values, are reducing with the reducing temperature and changing between 3500 and 7500 W/m^2K^{-1} values with respect to different vapor qualities. It is showed that cascade system has 60% more efficient system performance compared to existing systems. At cold climate zones; trans-critical applications and at warm or hot climate zones cascade cycle applications of CO_2 is suggested.

A. da Silva **[25]**, showed that, CO_2 cascade refrigeration system has greater system performance compared to known R404A and R22 single-stage refrigeration systems. Also concluded that, cascade system has several advantages due to experimental and theoretical studies;

- Low energy consumption
- Longer compressor life due to low compression ratio
- Higher pressure and density values at low compression stage
- Requires smaller pipe diameters
- Requires low refrigeration nutrition
- Low refrigerant cost (CO_2)
- Low GWP and tax values
- High enthalpy value, high sub-cooling temperature at liquid phase and high cooling capacity

Also; production of cascade system costs less and needs less equipment compared to other refrigeration systems.

Bingmin and Huagen **[26]**, studied on feasibility of NH_3/CO_2 cascade refrigeration system in large-scale industrial applications. Main objective of the study is examining the effect of operation conditions on the system performance. After experimental and numerical analyses, concluded that;

- NH_3/CO_2 cascade refrigeration system has higher COP values compared to single-stage NH_3 refrigeration system at under evaporation temperature of -40 ^{0}C.
- Results from experimental studies compared with previous numerical results of some practitioners like Lee **[27]**.
- Increasing condensing temperature of CO_2 come up with sharp decrease in cooling capacity of the system. By fixation of condensing temperature of CO_2 the problem is

handled and also maximized the COP values of the system. (During the experiments; NH_3 condensing temperature and CO_2 evaporation temperature are fixed.)

- Increasing temperature difference at heat exchanger of cascade system caused slight decrease at cooling capacity but sharp decrease at COP values. On the other hand, decreasing temperature difference caused no effect on cooling capacity but slight decrease at COP values.
- Increasing superheating temperature of CO_2 caused decrease on both cooling capacity and COP but these decreases are negligible deviations.

Sawalha **[28]** used a multi-stage cascade refrigeration system consisting of an NH_3 HTC and a CO_2 MTC and LTC for a supermarket refrigeration process in thesis study. In this study; power consumption and COP of the cascade refrigeration system for required evaporator temperature at constant environment conditions. Also a comparison analysis is made for cascade refrigeration system with single-stage refrigeration systems. Theoretical and numerical analysis of single stage and cascade refrigeration systems is made and compared with the experimental results. Thus an optimization process is occurred.

Dopazo **[29]** tested a cascade refrigeration system which numerically modeled and analyzed at previous study **[18]**. The results obtained from numerical and theoretical analysis are used as operating conditions on the performed experiments. System performance and power consumption is calculated for a specified ambient temperature and a specified CO_2 evaporator capacity. Experimental results of the theoretical and numerical study and studies of Bansal **[9]** and Lee **[27]**, thus feasibility of the corresponding design is indicated.

2. MATERIALS AND METHODS

2.1. Experimental Facilities

Experimental facilities on this study consist of; R404A/CO_2 Cascade Refrigeration System, Adiabatic Testing Room, Refrigeration Unit and Cooling Tower. Brief explanation about corresponding systems can be found in the following sections.

2.1.1. Adiabatic testing room

Adiabatic Testing Room in Figure 2.1 is constituted as a shell room and a testing room one within the other. It is used to prevent the system conditions from the variant ambient conditions and collect proper results from the tests. Dimensions of the Adiabatic Testing Room are designed with respect to the TS EN 14511 Standard.

Evaporators are tested inside this room. The room made of polyurethane filled with insulation material of 150 mm thick and 40 kg/m^3 density. Testing Room has 5.3x3.3x2.8 m dimensions. There is a door to get in and out of the room with dimensions of 1x2 m. Also 6 pieces of 165x400 mm deadwoods put under the Testing Room enabling homogeneous distribution of air flow and temperature around the room.

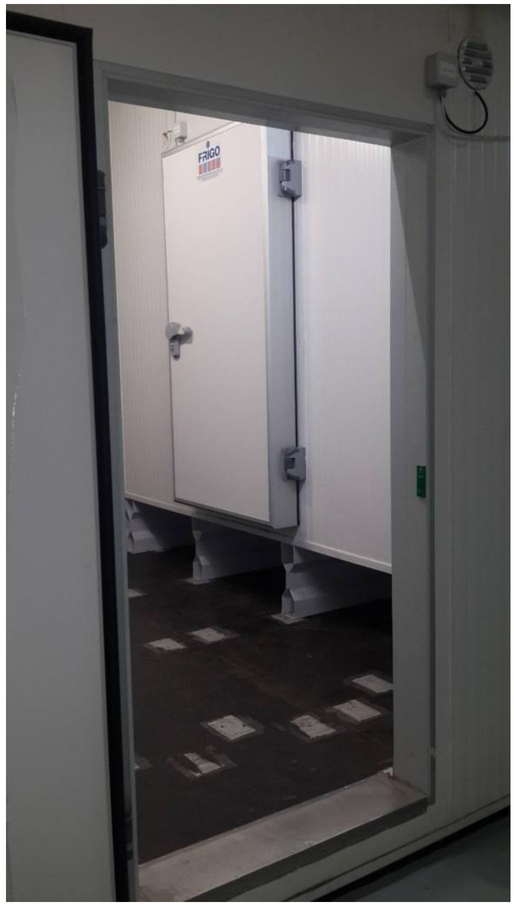


Figure 2.1 Adiabatic Testing Room

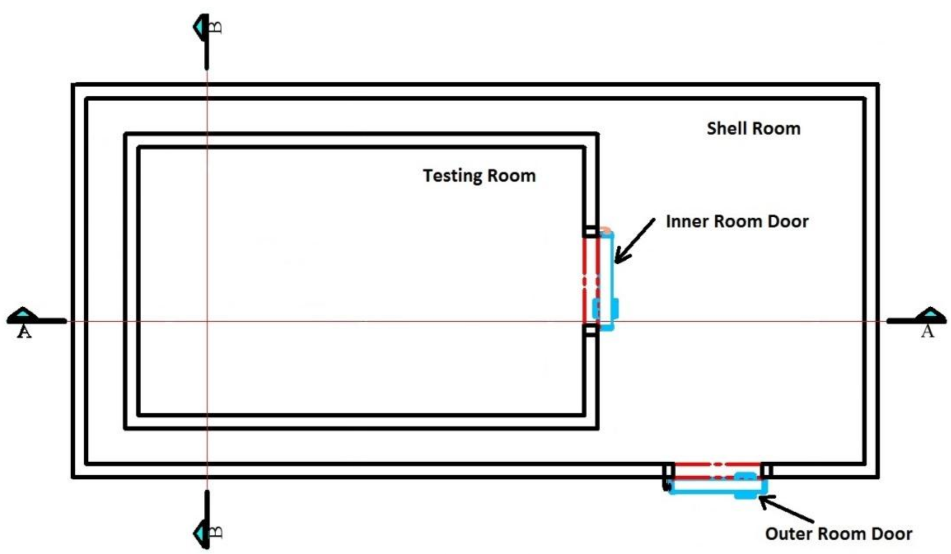


Figure 2.2 Adiabatic Testing Room Schematic View

Shell room is constituted for insulation of Testing Room from the outer ambient conditions. The room made of polyurethane filled with insulation material of 150 mm thick and 40 kg/m^3 density. Shell Room has 9.04x4.4x3.9 m dimensions. There is a door to get in and out of the room with dimensions of 1x2 m. An Air Handling Unit system with R404A refrigerant is used to reach desired temperature values. Testing Room and the Shell Room temperatures are adjusted as the same to minimize the heat loss from the testing facility to the surroundings.

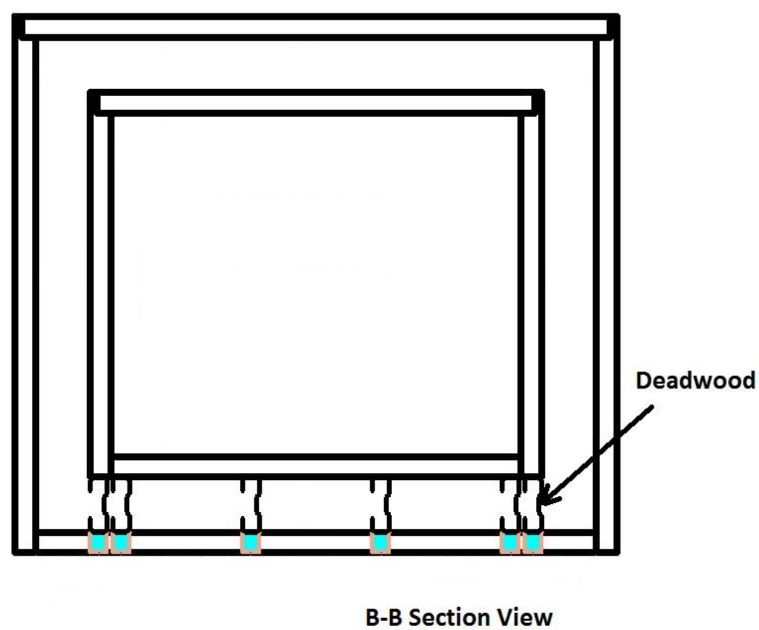


Figure 2.3 Adiabatic Testing Room B-B Section View

2.1.2. R404A/CO_2 cascade refrigeration system

Cascade refrigeration system consists of HTC and LTC basically. Main purpose of these systems is facilitating the refrigerants which are more efficient at low ambient (condensation) temperatures like CO2. R404A (Freon) is used as HTC refrigerant in the cascade refrigeration system. R404A enhanced using CO2 at efficient temperature levels with reducing the condensation temperature of LTC at internal heat exchanger. R404A is selected because it is commonly used refrigerant in the industry. R404A/CO2 cascade refrigeration system has the following components (Figure 2.4).

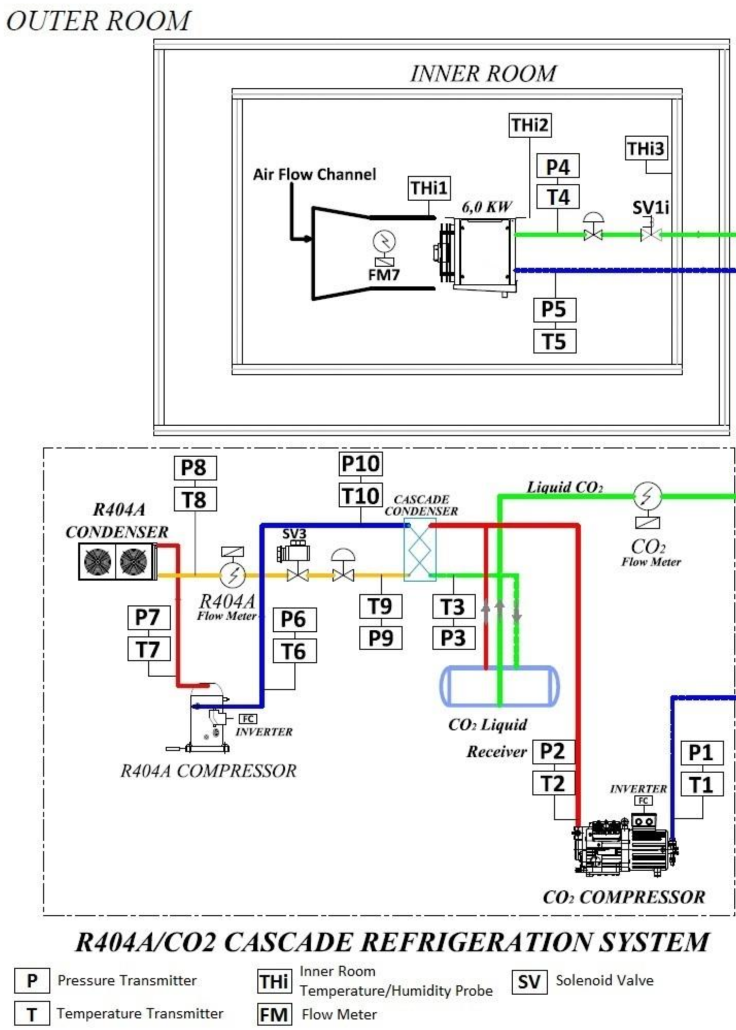


Figure 2.4 R404A/CO2 Cascade Refrigeration System

2.1.2.1. High temperature cycle (HTC)

High Temperature Cycle (HTC) consists of R404A refrigeration unit. The refrigeration unit has an air-cooled condenser with fan speed control with respect to condensation pressure, liquid receiver, thermostatic expansion valve (TXV) and cascade condenser as evaporator.

Figure 2.5 HTC Scroll Compressor and sensor connections

A scroll compressor (Figure 2.5) is used in HTC. The compressor has an operating temperature range between -10 / +40 ^{0}C. Compressor has also a cooling capacity of 8.3 kW. Operating pressure range of the compressor is between 3.7 and 16 bars for suction and compression pressures respectively. The compressor has a frequency converter to handle the capacity changes in the system according to the experiment that is practiced. Energy consumption of R404A compressor is read by the inverter that is controlling which can be seen in Figure 2.6

A standard thermostatic expansion valve with an appropriate orifice is used to feed the cascade condenser properly. A vertical liquid receiver is used to provide saturated liquid R404A to the tested thermostatic expansion valve for decent heat transfer between the refrigerants.

Figure 2.6 R404A Inverter placed on the Electrical Control Unit

Temperature, pressure and flow meter measuring points on the high temperature cycle are illustrated schematically in Figure 2.7.

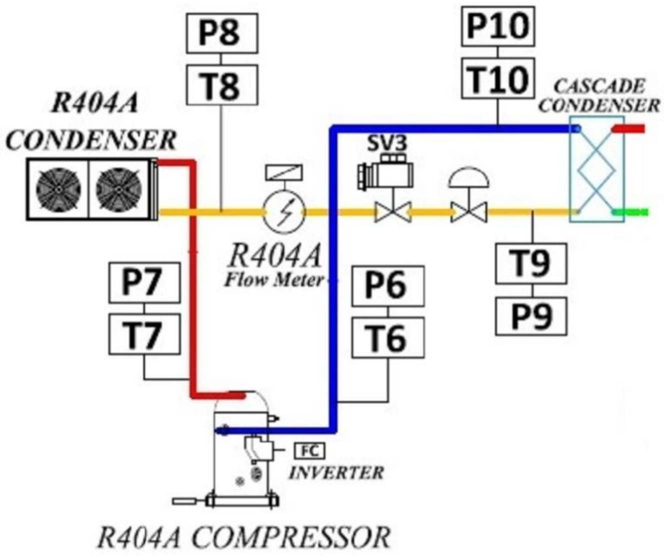


Figure 2.7 High Temperature Cycle (All components included)

2.1.2.2. Low temperature cycle (LTC)

Low Temperature Cycle (LTC) consists of CO_2 refrigeration unit. The refrigeration unit has cascade condenser, liquid receiver, Electronic Expansion Valve (EEV) and an evaporator.

Beside standard refrigeration components a set of auxiliary equipments are mounted on the system. Oil tank, oil pump and oil separator is used to control compressor lubrication. Also a suction line accumulator is used to avoid liquid CO2 return to the compressor. Mechanical pressure indicators are also located onto several high pressure outlet regions of the system. The objective here is complete observation of the system and robustness against electrical failure over the system.

A single stage reciprocating semi-hermetic compressor is used in LTC. Operating temperature of the compressor is between -40 / -10 ^{0}C. Compressor has also a cooling capacity of 6,44 kW and power consumption of 1.88 kW. Operating pressure range of the compressor is between 10 and 25 bars for suction and compression pressures respectively. The compressor has a frequency converter to handle the capacity changes in the system according to suction pressure value.

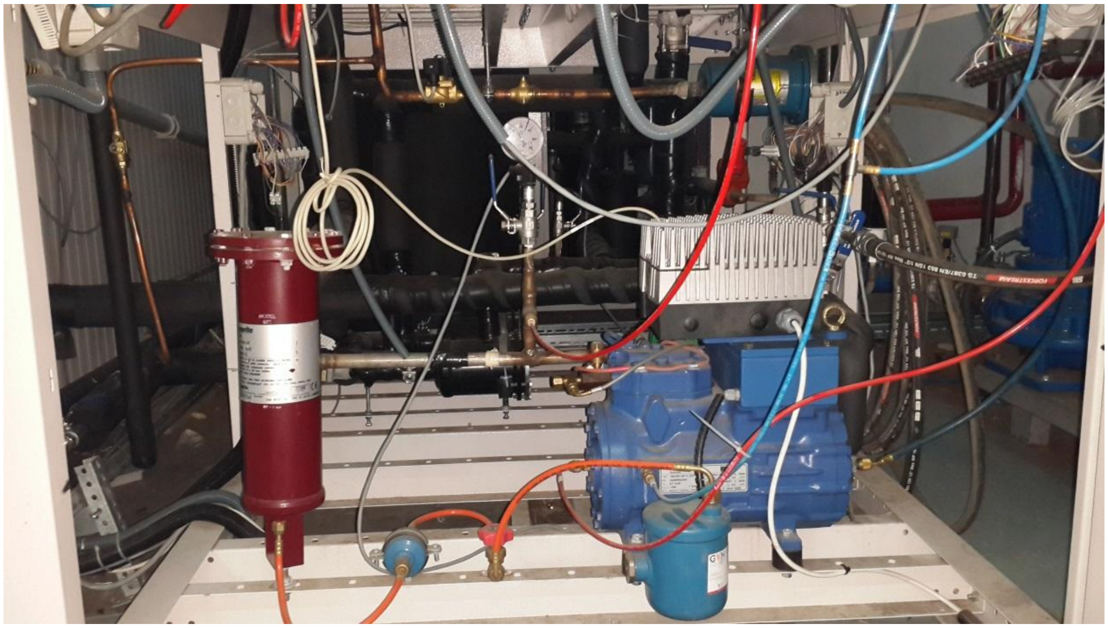


Figure 2.8 CO2 Compressor and Lubrication Components

Figure 2.9 LTC PHE and Liquid Receiver

An electronic expansion valve is used to regulate the pressure of the refrigerant which is entering to the evaporator. This valve is a slide type solenoid valve with an orifice for expansion. Valve body can be combined with 6 interchangeable orifices to cover 7 different capacity ranges. Electronic expansion valve (EEV) operates between -40 ^{0}C and -10 ^{0}C for gas and liquid CO2 temperatures respectively within the range of 26 and 10 bars pressure values and it has a capacity value of 6 kW in this operating range.

Carel Expansion Valve Driver (EVD) is used to control steps of the electronic expansion valve. Driver controls refrigerant superheat and optimizes the efficiency of the refrigerant circuit. The driver can handle low superheat, high evaporation pressure and low evaporation pressure problems. Driver controls the expansion valve via the pressure and temperature sensors located at the exit of the evaporator that is tested. The possibility of control by a PLC unit with 4-20 mA or 0-10 Vdc analog input signals is a feature that could be considered with this product. EEV and electronic Proportional – Integral – Derivative (PID) driver can be seen in Figure 2.10 and Figure 2.11, respectively.

Figure 2.10 CAREL E2V Electronic Expansion Valve

A horizontal liquid receiver (Figure 2.9) is used to provide vapor CO_2 refrigerant to the internal heat exchanger and saturated liquid CO_2 to the tested evaporator for decent heat transfer between the refrigerants. The horizontal liquid receiver has a volumetric capacity of 60 L, working under pressure of 25 bars, approximately at a temperature of -10 ^{0}C.

Figure 2.11 Electronic Expansion Valve Control Unit and Driver

2.1.2.3. Cascade condenser

A single circuit brazed plate heat exchanger is used as cascade condenser of the system. Gaseous CO_2 coming from compressor circulates in primary side of the cascade condenser to be fully condensed before entering the liquid receiver. In the secondary side of the cascade condenser; liquid R404A circulates for evaporation process. Evaporation temperature of the secondary side is -15 ^{0}C and with 5 ^{0}C difference on the exchanger CO_2 condensation temperature is -10 ^{0}C. 5 ^{0}C differences is found to be the optimum value of the cascade condenser in this operating range from the thermodynamic calculations made on EES software.

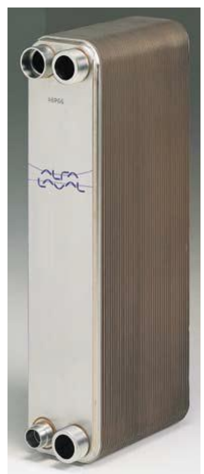


Figure 2.12 Brazed Plate Heat Exchanger

2.1.2.4. Refrigeration cycle unit

Refrigeration Cycle consists of two compressors with same operating conditions, shell and tube condenser, liquid receiver, expansion valves and R404A evaporators based in the Air Handling Units (AHUs).

There are two semi-hermetic single-stage reciprocating R404A compressors. Total cooling capacity of the compressors will handle the capacities of two different evaporators in two different air handling units. Compressors are working simultaneously between +45/-40 ^{0}C and +45/-10 ^{0}C temperatures with respect to the temperature of the Shell and Testing

Rooms corresponding to the Standard Conditions. At operating conditions of +45 / -40 ^{0}C each compressor has 5 kW cooling capacity and 4.7 kW power consumption and at operating conditions of +45 / -10 ^{0}C each compressor has 24 kW cooling capacity and 11.4 kW power consumption. Operating suction pressure values change between 0.7 and 19.6 bars respectively.

Since CO2 evaporator is placed inside the Testing Room; Air Handling Unit inside this room is closed to test evaporator in the ambient conditions to get proper results. So solenoid valve of this line is closed to block refrigerant pass.

A horizontal liquid receiver is used to provide saturated liquid R404A refrigerant to the evaporators of AHUs.

Figure 2.13 Refrigeration Unit Cycle

2.1.2.5. Cooling tower

Niba cooling tower is operated with some subsidiary components such as shell and tube condenser and pumps. Heat transfer from the water supplied by cooling tower to the refrigerant is performed by shell and tube condenser. Circulation of cooling water through condenser and cooling tower is performed by pumps. Cooling capacity of the tower is 39.7 kW. Cooling water circulates at 12.3 m^3/h volume flow rate. Heat rejection from cooling water to the outside air is provided by fans with 2.2 kW power. Water is entered to cooling tower at the temperature of +35 ^0C and exited at temperature of +30 ^0C for shell and tube condenser of the system.

To provide water need for shell and tube condenser, two circulation pumps are used. Pump has 6 m^3/h volume flow rate and pumps water at +30 ^0C coming from the cooling tower. At the exits of the pumps there are two throttling valves to compensate the cooling water need of condenser. The condenser of the Refrigeration Unit Cycle needs 6 m^3/h of volume rate. As you can see in the Figure 2.14 there are 2 pumps mounted on the system. One of them used and is controlled with inverter the other one is a backup in case of breakdown and to prevent high pressure fault of the refrigeration system at the discharge of the compressors.

Figure 2.14 Circulation Pumps

2.2. Instrumentation And Data Logging

Instrumentation devices are mounted for controlling and the monitoring the instantaneous conditions of the refrigerants and the equipments. Pressure transducers, thermocouples, manometers, differential pressure switches and humidity/temperature sensors monitor the conditions of the CO_2, Freon and the room air both mechanically and electronically. Energy analyzer, thermocouple signal transmitter and inverters control and monitor the electrical parameters of the controlled equipments. A data logging system combined with PLC programming restores the received data from instrumentation devices.

2.2.1. Instrumentation devices

2.2.1.1. Temperature measurements

Temperature measurements are realized with K-type thermocouples for temperature measuring over the refrigeration lines.

K-type thermocouples have measuring range of -200 ^{0}C to +1350 ^{0}C and they are inexpensive, highly available type of thermocouples with good temperature sensitivity. Since thermocouples produce 0-5 V output signal for measured temperature and selected PLCs are requiring 4-20 mA input signal a thermocouple transmitter from 0-5 V to 4-20. A temperature/humidity sensors are used to measure temperature and relative humidity of air in the Figure 2.15. They calculate psychometric parameters such as the dew or frost point. Depending on the probe model, sensor can measure conditions within the range of 0 to 100 %RH and -100 to 200°C (-148 to 392°F). The electronics operating range is limited to -40 and +60 °C.

Figure 2.15 Rotronic Temperature and Humidity Sensor

2.2.1.2. Pressure measurements

Piezo-resistive type pressure transmitters are used for monitoring pressure values of the refrigerants. Because of different operating temperatures of R404A and CO_2, transmitters selected at different operating ranges. Due to low temperature values while refrigeration system operating, temperature effect on pressure measurements has added to calculations. Keller enhances the best temperature tolerances with its temperature tolerant circuit in the transmitters. Circuit has a temperature sensor to measure and calibrate the value of the measured pressure data. Thus, controlling and monitoring system gets the most accurate pressure data. Since CO_2 works at higher pressure levels this comes an advantage of using Keller transmitters.

2.2.1.3. Flow measurements

There are 4 different flow meters in the experimental system which are; CO2 liquid flow, R404A liquid flow, water mass flow and air mass flow. Appropriate flow meter devices are selected for each requirement.

R404A liquid flow meter is a turbine flow meter (Figure 2.16). Liquid R404A flows through the turbine housing causing an internal rotor to spin. As the rotor spins, an electrical signal is generated in the pickup coil. This pulse data is translated from the turbine into calibrated flow units on the readout of computer.

Figure 2.16 R404A Liquid Turbine Flow Meter

CO2 liquid flow meter (Figure 2.17) has same reading characteristics with the R404A liquid flow meter but there is a mechanical difference in the measuring device which is not a turbine but 2 pieces of oval gears. They are connected in elliptical shape with the threads and liquid passing through these gears force them to rotate and a pulse is generated to be read from the computer as same logic with R404A flow meter. Safety precautions might be considered because any solid sedimentation or flash gas formation of CO2 inside or around the flow meter results with wrong measurement of fluid flow.

Water flow meter is a pulse out standard water meter with a mechanical indicator of measuring cumulative fluid flow rate. Using pulse module, fluid flow rate can be read from the electronic pulse send from the flow meter.

Figure 2.17 CO2 Liquid Oval Gear Flowmeter

4 pieces of pitot tubes has been used as air flow meter located inside a cylindrical channel in front of the evaporator. Pitot tubes measure the static and total pressure difference of air and calculates the dynamic pressure of the air which equals to the velocity of the air. Since there are 4 pitot tubes, average velocity has to be transmitted to the PLC units to store the data. So a differential pressure transmitter has been used.

2.2.2. Data logging

Measurement from all sensors gathered in a Programmable Logic Circuit (PLC) unit and controlled with automation system with respect to the read data. Program logics are constructed for each device that running the system with respect to the control scenario designated before. To control entire system remotely; a web based interface has been established.

2.2.2.1. PLC units

Direct digital controller PLC unit has 16 universal inputs for different measuring device reading and total of 12 universal outputs for controlling devices in the system. PLC unit is also allowed to write 10 programs for controlling and 40 analog and 40 binary value objects. Each PLC unit can store 8 different readings from measuring devices with trend object feature. Totally 8 units of PLC is used to control entire experimental facility.

Figure 2.18 PLC Controlling Units

2.2.2.2. Control programs

Control programs wrote for each cycle in the experimental facility via using Total Control program. Since refrigeration cycles are connected and each cycle is controlled by the measured data; an automation scenario has been established by considering data priorities and safety precautions.

2.2.2.3. Web based user interface

Web based user interface allows the user to control whole system remotely and monitoring system behaviors. Parameter settings of each component are determined using this interface. Power consumption of each controlled component could be monitored from the interface. In Figure.14 a part of the interface could be seen.

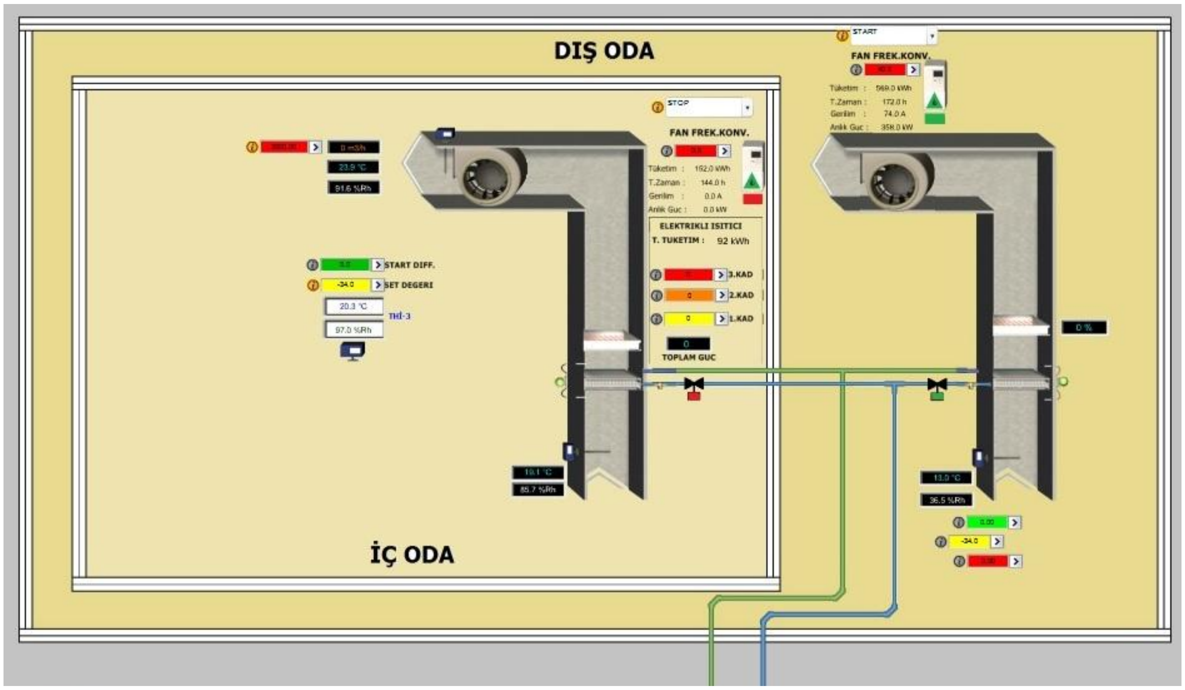


Figure 2.19 Web Based User Interface

3. RESULTS AND DISCUSSION

The aim of this study is to determine the system performance and efficiency of the evaporator of a R404A/CO_2 cascade refrigeration system by measuring humidity, temperature, pressure, flow rate, mass flow rate, power consumption and cooling capacity. Additionally the potential of industrial applications for our system is investigated.

The experiments are handled at SC5 Standard Conditions of temperature for evaporator at capacity of 7.75 kW. Evaporator design is made by using Coils program. Standard conditions are specified on the ''TS EN 328 – Heat Exchanger-Test Procedures for Establishing the Performance of Forced Convection Unit Air Coolers for Refrigeration'' standard. The standard conditions are given in Table 3.1. In this study, the experiments are realized for SC5 standard condition.

Table 3.1 TS EN 328 - Standard Conditions

Standard Condition	Evaporation Temperature (^oC)	Room Temperature (^0C)
SC1	0	10
SC2	-8	0
SC3	-25	-18
SC4	-31	-25
SC5	-40	-34

3.1.1. Experiments for R404A/CO_2 cascade refrigeration system

The evaporator is placed into the Adiabatic Testing Room which can be conditioned for different Standard Conditions. Evaporator which has 7.75 kW cooling capacity is tested for SC5 standard condition.

3.1.1.1. High temperature cycle (HTC)

As mentioned before HTC is used for regulating the pressure of liquid receiver feeding the CO2 expansion valve. In temperature graphics you can see that average evaporation temperature of R404A at cascade condenser is -11.47 ^{0}C and at the outlet of the cascade condenser temperature is -3.63 ^{0}C with the same pressure level leading to superheat level of 2 K. Automation system controls the speed of the compressor with respect to the CO2 liquid receiver pressure. As the pressure goes up R404a compressor speeds up when the pressure is in the range we set, R404a compressor slows down. The temperature and pressure peaks and falls is a conclusion of this speed regulation. Also condensation pressure and temperature levels are important for the appropriate evaporation temperature and pressure at the thermostatic expansion valve inlet.

Figure 3.1 HTC Split Unit

Pressure and temperature variations of the system in the performed experiment are shown in the figures below.

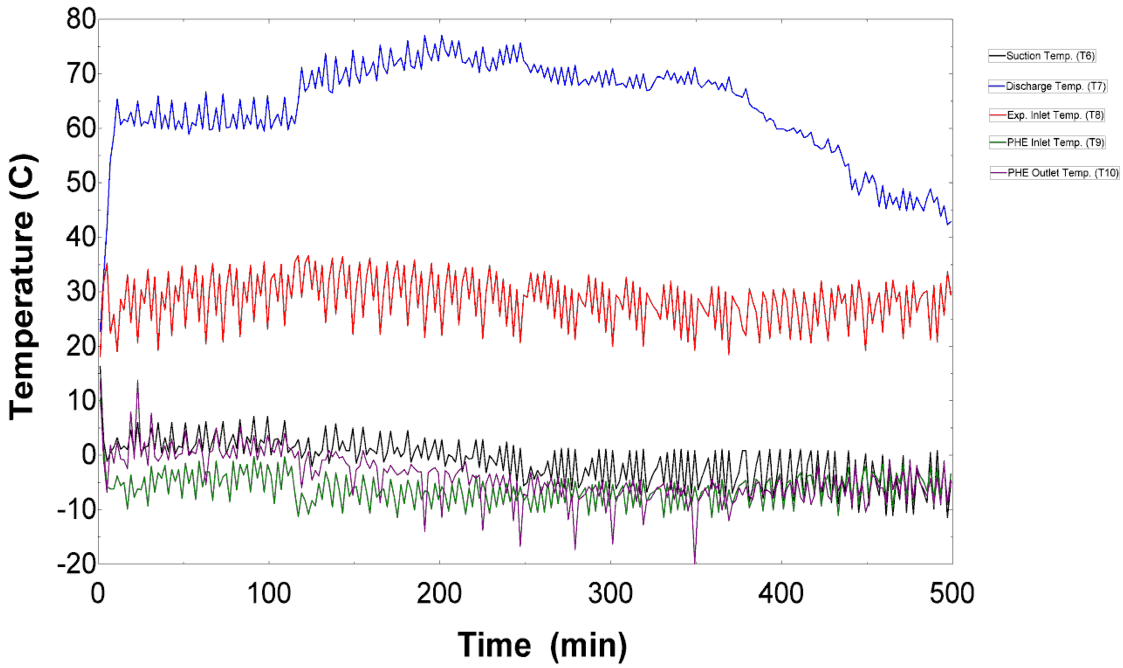


Figure 3.2 Temperature Variations in the High Temperature Cycle

As you can see in the Figures 3.2 & 3.3; outlet of the air cooled condenser is almost same value for every pressure level entering the condenser. The pressure and temperature values at the outlet of the condenser is adjusted using fan speed control devices located onto the fans of the condenser which has a pressure setting of 15 bars and if the pressure value of the refrigerant entering to the condenser is greater than the set point; first fan starts to cool down if it cannot provide sufficient pressure drop at the outlet, the second fan starts to rotate and cools down the refrigerant. The fan selection is done with respect to the operating range and location of the HTC considering thermodynamic properties of the refrigerant R404A.

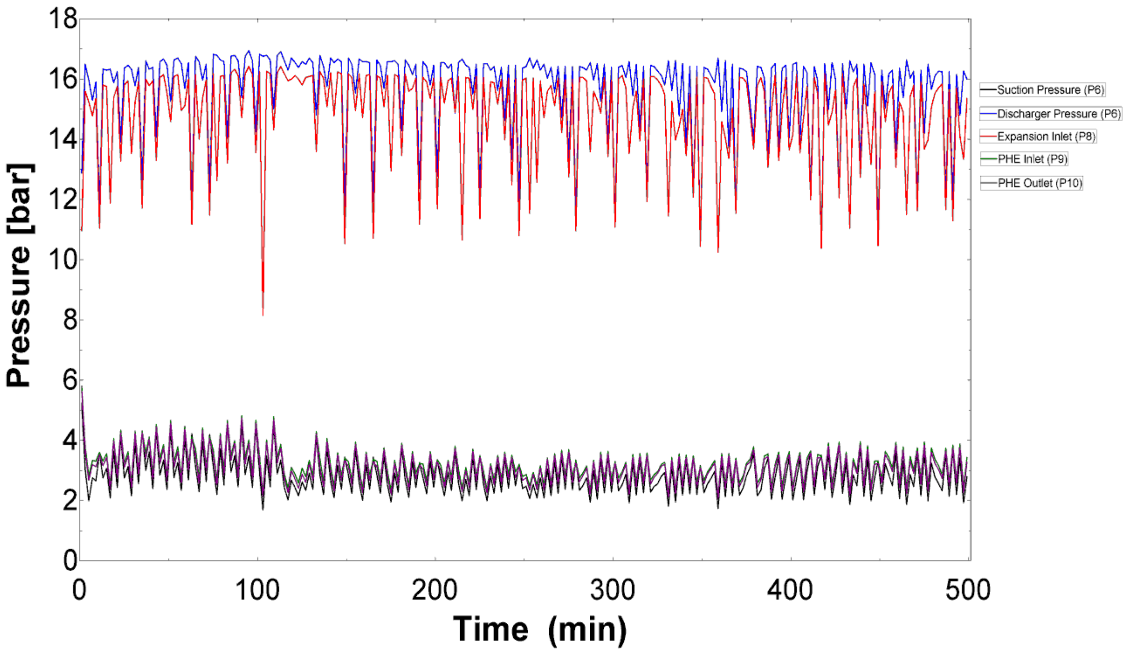


Figure 3.3 Pressure Variations in the High Temperature Cycle

As the temperature and pressure values are recorded via PLC units, the average of the collected and restored data has been calculated. Enthalpy values are specified using superheat and saturated properties of the refrigerant with respect to the calculated temperature and pressure values and tabulated in Table 3.2.

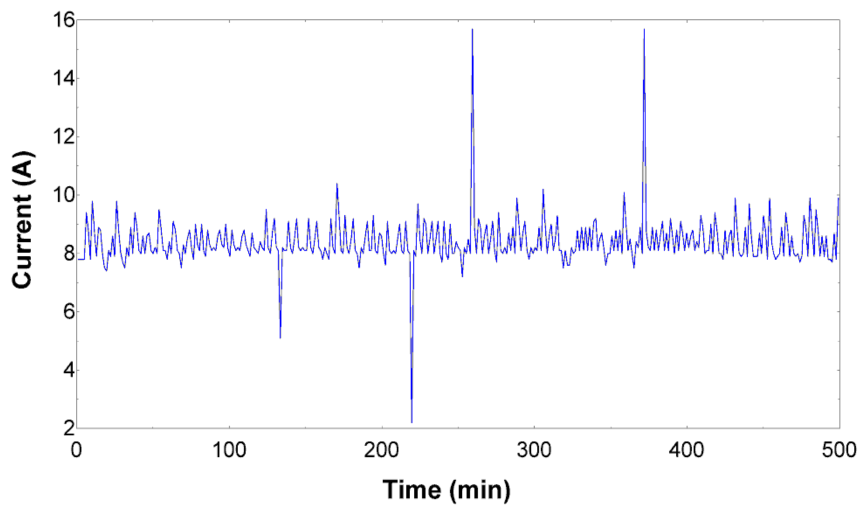


Figure 3.4 Current Drawn by HTC Scroll Compressor

As can be seen in the Figure 3.4 the current drawn by the scroll compressor in the entire experiment has an average of 8.4 Amperes. Current peaks and lows are caused by the slowdowns and startups of compressor with respect to the CO2 receiver pressure. Power consumption calculations can be made by using equation of;

$$P = V \cdot I \cdot \sqrt{3} \cdot \cos\varphi \tag{4.1}$$

V= voltage supply from the main electric line

I= Current drawn by the compressor

$\cos\varphi$= is a constant with value of 0.89

when the calculations are made by the given data and equations above, the power consumption of the HTC compressor is;

$$P = (380) \cdot (8.40) \cdot \sqrt{3} \cdot (0.89)$$

$$P = 4920W$$

Fan power consumption values must also be added for calculating total power consumption value. There are 2 identical fans on the HTC condenser and consumption values are equals to 245 W.

$$P_{fan} = 2 \cdot 245 = 490 \text{ W}$$

$$P_{total,HTC} = 490 + 4920 = 5410 \text{ W}$$

In the experimentation period total of 1920 L liquid refrigerant pass through liquid turbine flow meter. Flow meter is located just before the thermostatic expansion valve to ensure saturated liquid refrigerant to measure properly. The thermal condition around the flow meter is +28.45 ^{0}C and since refrigerant is saturated liquid specific density of the refrigerant is 1027.5 m^3/kg.

Total experiment duration is 453 minutes;

$$\dot{m}_{R404A} = \frac{1920}{453 \cdot 60} = 0.070 \text{ L/s}$$

$$\dot{m}_{R404A} = \frac{(0.070)}{1000} = 7 \cdot 10^{-5}\,\mathrm{m^3/s}$$

$$\dot{m}_{R404A} = (7 \cdot 10^{-5}) \cdot (1027.4) = 0.074\,\mathrm{kg/s}$$

Average temperature and pressure values and corresponding enthalpy values are tabulated below.

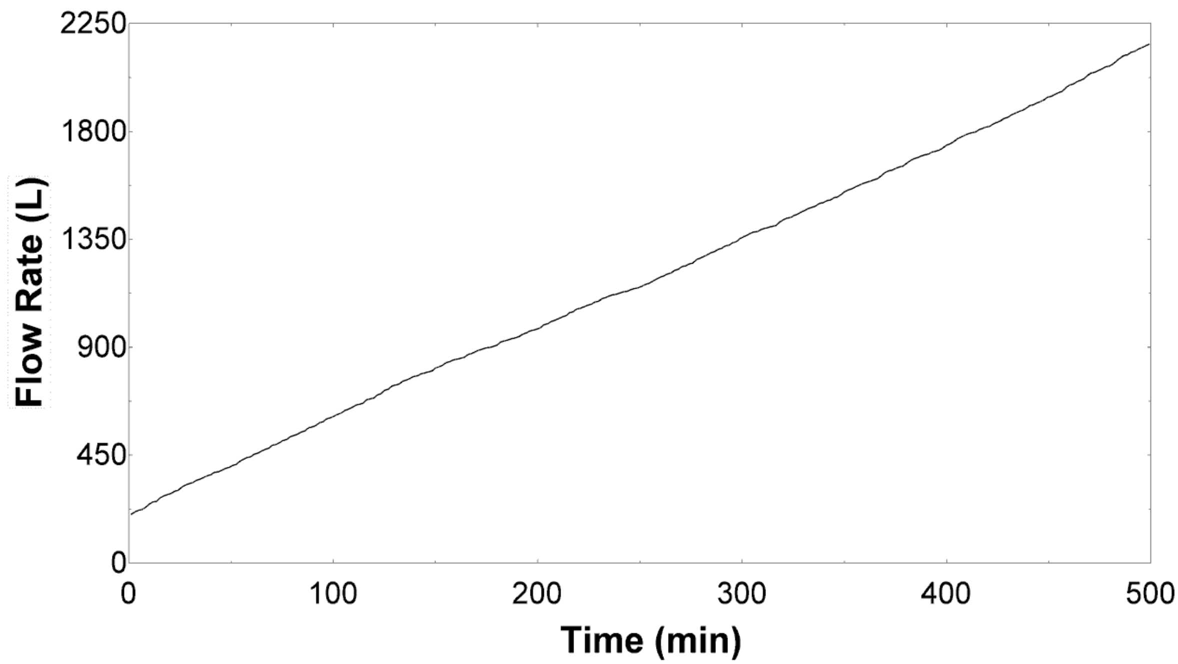


Figure 3.5 Cumulative Flow Rate of R404A Refrigerant in Liters

Capacity calculations of the High Temperature Cycle made by using following equations with respect to the values in Table 3.2;

$$\dot{Q} = \dot{m} \cdot \Delta h = \dot{m} \cdot (h_{outlet} - h_{inlet})$$

(4.2)

$$\dot{Q}_{compressor} = \dot{m}_{R404A} \cdot (\mathrm{h}_{outlet} - \mathrm{h}_{inlet}) = (0.074) \cdot (232.9 - 197.5) = 2.62\ \mathrm{kW}$$

$$\dot{Q}_{condenser} = \dot{m}_{R404A} \cdot (\mathrm{h}_{outlet} - \mathrm{h}_{inlet}) = (0.074) \cdot (103.6 - 232.9) = \text{-}9.57\ \mathrm{kW}$$

$$\dot{Q}_{PHE} = \dot{m}_{R404A} \cdot (\mathrm{h}_{outlet} - \mathrm{h}_{inlet}) = (0.074) \cdot (200.5 - 50.86) = 11.07\ \mathrm{kW}$$

Table 3.2 Average Pressure, Temperature and Enthalpy Values of HTC

Measurement Points	High Temperature Cycle		
	Pressure (bar)	**Temperature (^{0}C)**	**Enthalpy (kJ/kg)**
Compressor Inlet	1.52	-2.87	197.5
Compressor Outlet	16.53	68.30	232.9
Expansion Inlet	15.93	31.65	103.6
PHE Inlet	1.98	-11.47	50.86
PHE Outlet	1.89	-3.63	200.5

3.1.1.2. Low temperature cycle (LTC)

Saturated vapor CO_2 is entered to compressor at -40 ^{0}C temperature and 10 bars pressure values and exited from the compressor as superheated vapor at 25 bars of pressure value. Superheated CO_2 vapor is transferred to the internal heat exchanger which is used as the condenser of LTC. CO_2 is rejected heat to R404A coming from the HTC and condensing to -10 ^{0}C temperature and 25 bars pressure values. CO_2 is transferred to the liquid tank to be ensured that refrigerant passing through EEV and evaporator is saturated liquid. CO_2 is exited from EEV at -40 ^{0}C temperature and 10 bars pressure values and transferred to the tested evaporator. CO_2 is extracted heat from the air inside the Testing Room which is conditioned with respect to Standard Conditions. After evaporation, CO2 is transferred to the compressor as superheated vapor at 5K superheat and 10 bars of pressure values. At the compressor inlet additional 13K superheat value is occurred due to heating effect on the suction line. Thus LTC is completed.

Pressure and temperature variations of the system in the performed experiment are shown in the Figure 3.6 & 3.7 below.

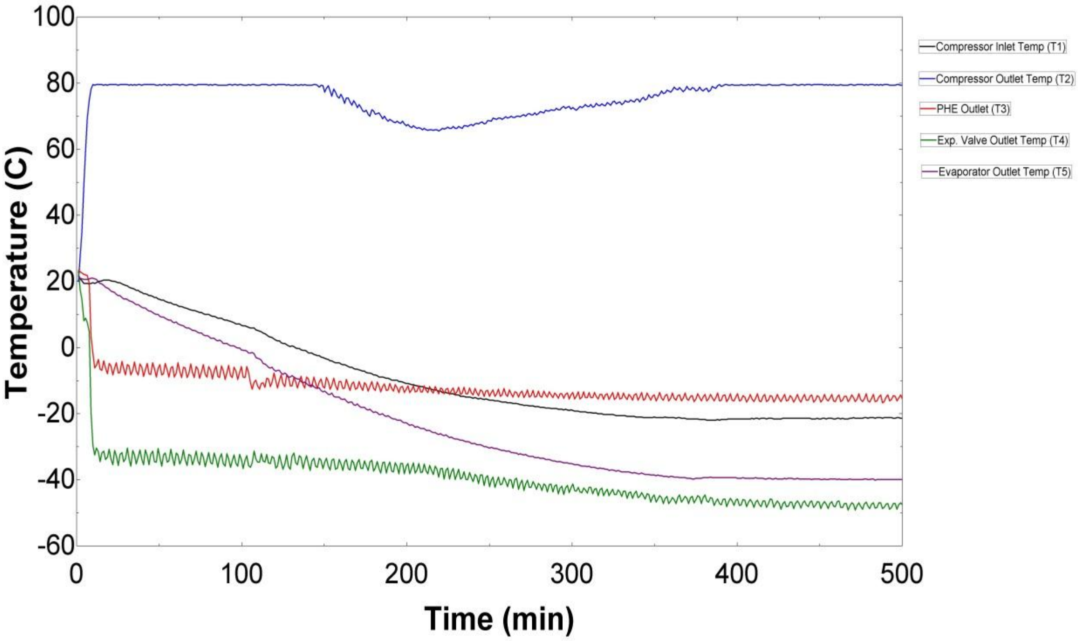


Figure 3.6 Temperature Variations in the Low Temperature Cycle

As you can see from the figures desired pressure and temperature levels are provided for SC5 conditions. Regarding safety problem HTC provided the desired 25 bars and -10 ^{0}C liquid CO2. The electronic expansion valve is set to arrange the 5 K superheat level at the outlet of tested evaporator. Suction pressure of the compressor is arranged by the Electronic Frequency Controller (EFC) of the compressor. This controller enables steady suction pressure of CO2 by controlling the speed of the compressor. EFC is working independent from the automation system we designed.

Compressor power consumption is recorded with an energy analyzer connected to the circuit breaker that controls the energy supply to the compressor. Average hourly value of compressor consumption is equal to 2 kW.

Like HTC condenser, tested evaporator has 2 identical fans with 165W.

$$P_{total,\mathrm{LTC}} = 2 + 2 \cdot 165 = 2330 \ \mathrm{W}$$

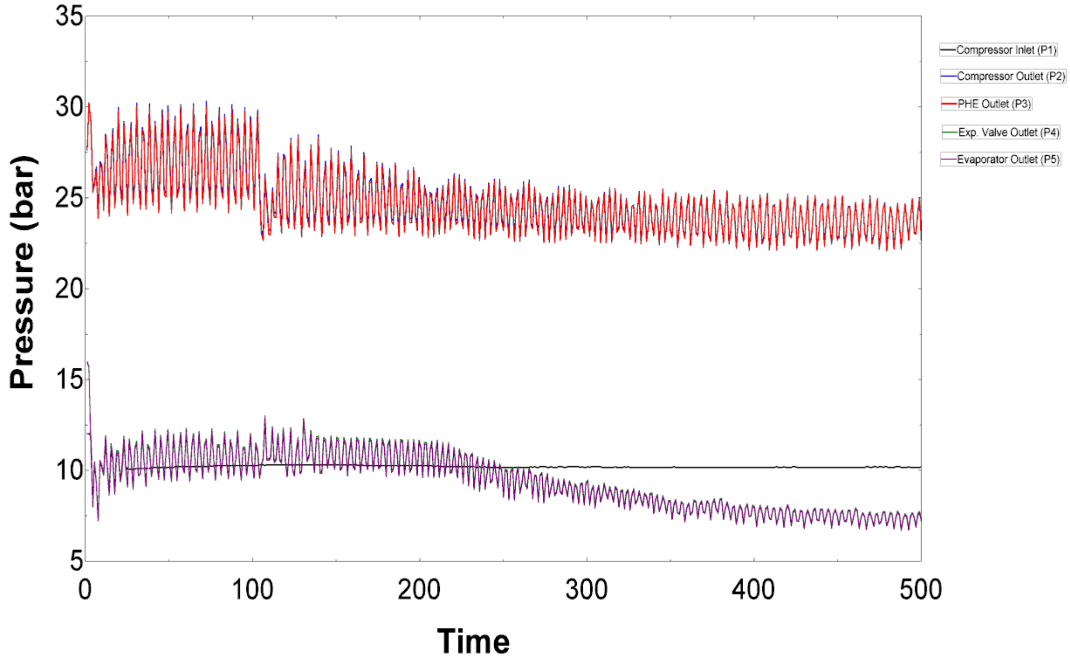


Figure 3.7 Pressure Variations in the Low Temperature Cycle

Figure 3.8 CO2 Compressor Energy Analyzer

Total of 810 liters CO2 is passed through the oval gear flow meter located at the outlet of the liquid receiver. Average temperature and pressure values corresponding to the position of the flow meter are -11.8 ^{0}C and 24.69 bars respectively. At these conditions using thermodynamic properties of CO2 specific liquid density is 983.96 m^3/kg.

$$\dot{m}_{CO2} = \frac{810}{453 \cdot 60} = 0.029 \text{ L/s}$$

$$\dot{m}_{CO2} = \frac{(0.029)}{1000} = 2.9 \cdot 10^{-5}\,\mathrm{m^3/s}$$

$$\dot{m}_{CO2} = (2.9 \cdot 10^{-5}) \cdot (983.96) = 0.0285\,\mathrm{kg/s}$$

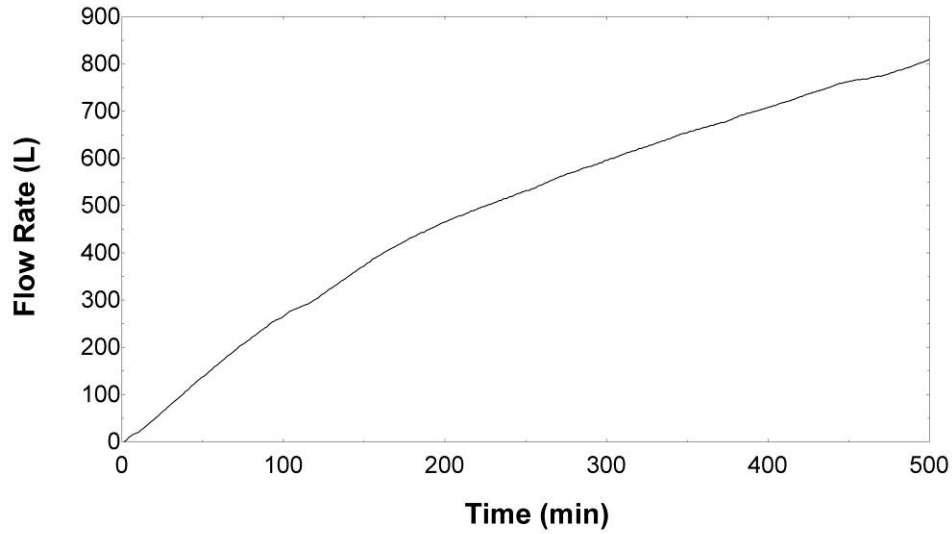


Figure 3.9 Cumulative Flow Rate of CO2 Refrigerant in Liters

Table 3.3 Average Pressure, Temperature and Enthalpy Values of HTC

Measurement Points	**Low Temperature Cycle**		
	Pressure (bar)	**Temperature (^{0}C)**	**Enthalpy (kJ/kg)**
Compressor Inlet	10.21	-21.19	452.11
Compressor Outlet	24.82	79.92	533.27
Expansion Inlet	24.69	-11.87	164.59
Evaporator Inlet	9.42	-39.41	170.29
Evaporator Outlet	9.36	-34.25	434.7

$$\dot{Q} = \dot{m} \cdot \Delta h = \dot{m} \cdot (h_{outlet} - h_{inlet})$$

(4.2)

$$\dot{Q}_{compressor} = \dot{m}_{CO2} \cdot (\mathrm{h}_{outlet} - \mathrm{h}_{inlet}) = (0.0285) \cdot (533.27 - 452.11) = 2.313\ \mathrm{kW}$$

$$\dot{Q}_{PHE} = \dot{m}_{CO2} \cdot (\mathrm{h}_{outlet} - \mathrm{h}_{inlet}) = (0.0285) \cdot (164.59 - 533.27) = \text{-}10.50\ \mathrm{kW}$$

$$\dot{Q}_{Evaporator} = \dot{m}_{CO2} \cdot (\mathrm{h}_{outlet} - \mathrm{h}_{inlet}) = (0.0285) \cdot (434.7 - 170.29) = 7.536 \text{ kW}$$

3.1.1.3. COP calculations

COP is a ratio of refrigeration effect that is provided to the electrical energy consumed by the pump or compressor. But COP of a cascade system is differing from standard COP calculations. Since there are 2 cycles connected with a cascade condenser HTC must be accounted in the COP calculation even if the refrigeration effect is gathered from the LTC evaporator. The COP equation of Cascade Refrigeration System is as follows;

$$COP = \frac{\dot{Q}_{evaporator}}{P_{total,HTC} + P_{total,LTC}} \tag{4.3}$$

$\dot{Q}_{evaporator}$ = Cooling capacity of CO2 evaporator

$P_{total,HTC}$ = Total electrical consumption of HTC

$P_{total,LTC}$ = Total electrical consumption of LTC

$$COP = \frac{7536}{5410 + 2330} = 0.974$$

The result shows us even though we had high COP at the LTC side using a HTC to provide low condensation temperature of CO2 ends up with too much energy consumption. But since cascade systems are adequate for the warm climate applications; this excess power consumption should be afforded in expense of lowering environmental effects of synthetic refrigerants.

3.1.1.4. Air side temperature and humidity variation

Inside the testing room, air temperature, humidity and flow rate values from different locations are measured. The testing room temperature which we try to provide the SC5 condition is measured by a temperature/humidity sensor. Air distribution over the tested evaporator is measured from inlet and outlet conditions. Inlet air temperature and humidity values are measure using the same sensor.

An air channel is manufactured to measure air mass flow rate which is blown by the evaporator fans. 4 pieces of pitot tubes are placed inside the air channel that forming an octagon. The pitot tubes are mounted to the differential pressure transmitter (Figure 3.12) which sends the dynamic pressure value using the total and static pressure difference measured by the pitot tubes. Outlet temperature and humidity values are measured using temperature/humidity sensor which is placed onto the air channel. The air channel connection to the blowing side of the evaporator can be seen on Figure 3.11 below.

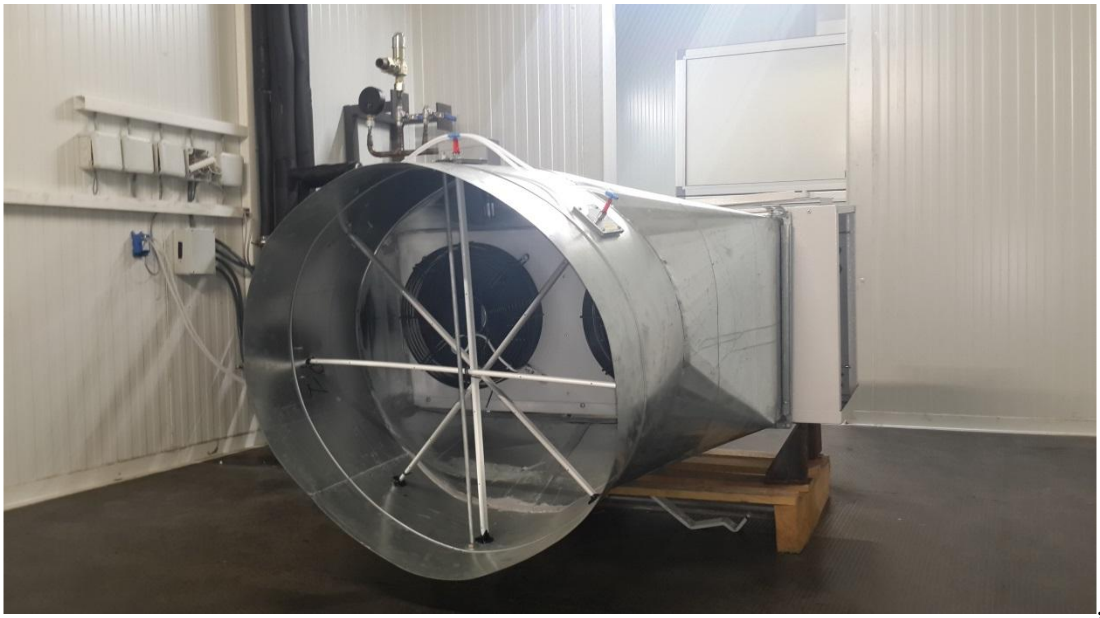
,

Figure 3.10 Air Channel Connection

Figure 3.11 Differential Pressure Transmitter

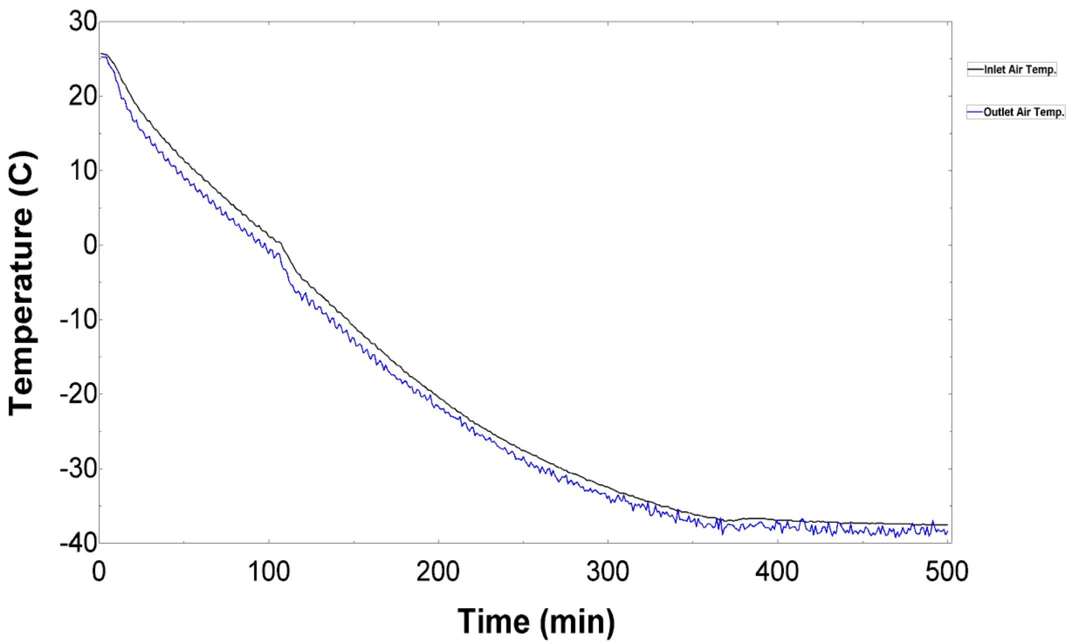


Figure 3.12 Air Temperature Variations at the Inlet and Outlet of the Tested Evaporator

For the entire period of experiment; approximately 3 ^{0}C difference is occurred for the inlet and outlet air temperature of the evaporator as you can see in Figure 3.13

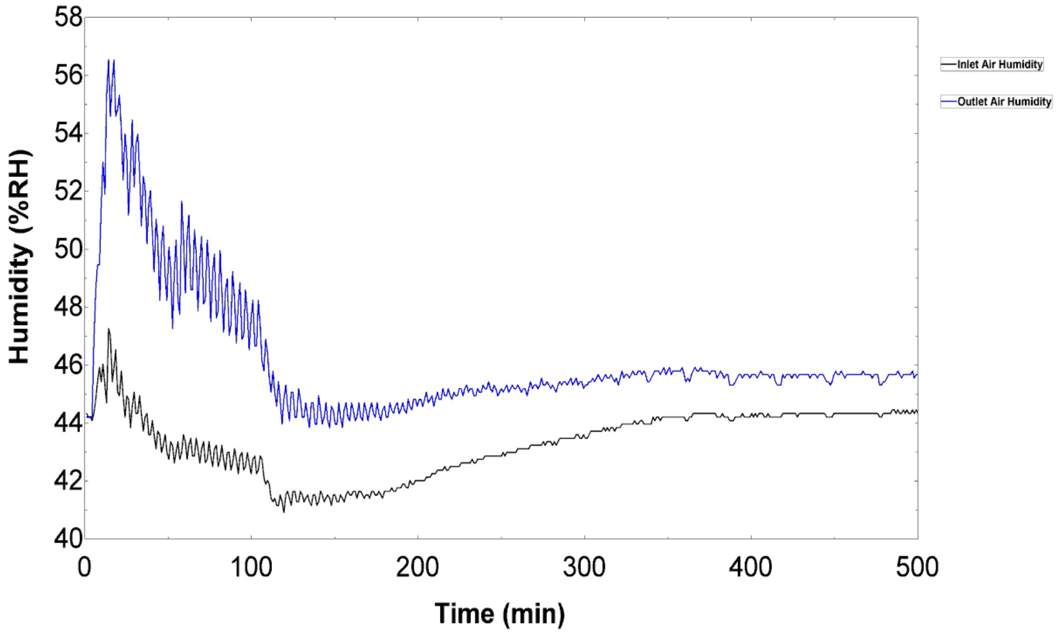


Figure 3.13 Air Humidity Variations at the Inlet and Outlet of the Tested Evaporator

Humidity variations of the air are decreased from 56% for outlet and 47% for inlet air. After testing room is reached the set value of -34 ^{0}C; humidity values are stabilized at 46% for outlet and 44% for the inlet air.

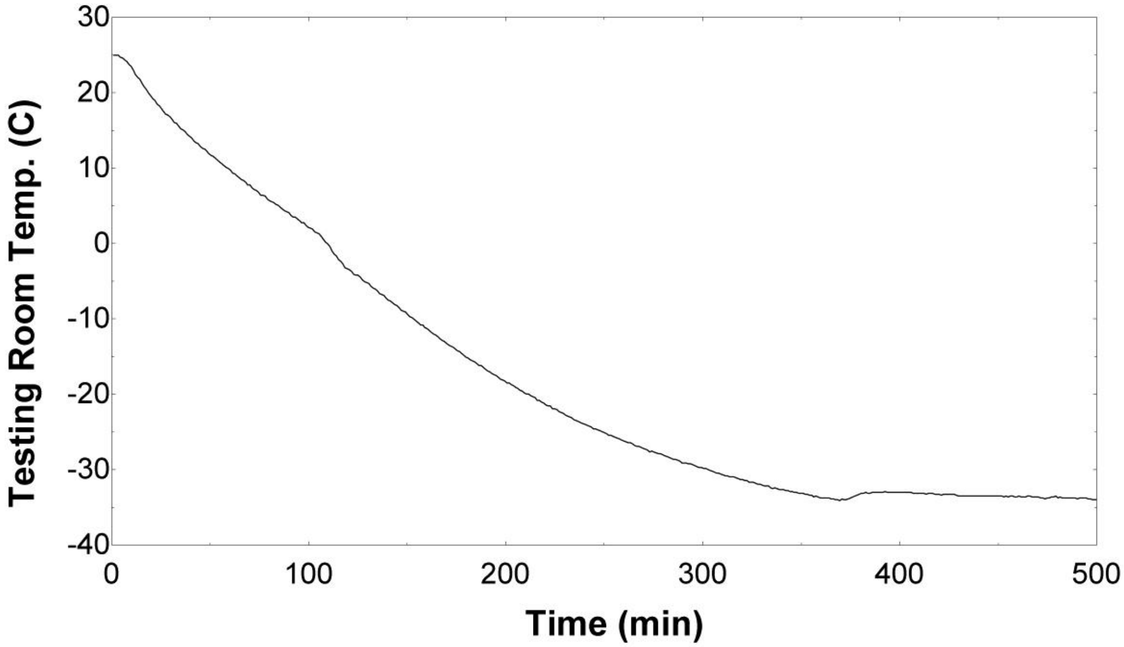


Figure 3.14 Testing Room Temperature Variations during Experiment

One of the main purposes of the experiment is providing testing room temperature of SC5 condition. As you can see from the Figure 3.15 room temperature of -34 ^{0}C is provided after approximately 360 minutes from starting at room temperature of +25 ^{0}C. The room temperature value is stabilized for the next 120 minutes of operation. In Figure 3.16 air flow rate of the evaporator fans can be seen changing between 5120 and 5826 m^3/h and has an average value of 5564 m^3/h has been calculated from the data collected.

$$\dot{m} = \frac{5564}{3600} = 1.545 \text{ m}^3/\text{s}$$

Air density at these conditions is equal to 1.45 kg/m^3

$$\dot{m} = (1.545) \cdot (1.45) = 2.24 \,\text{kg/s}$$

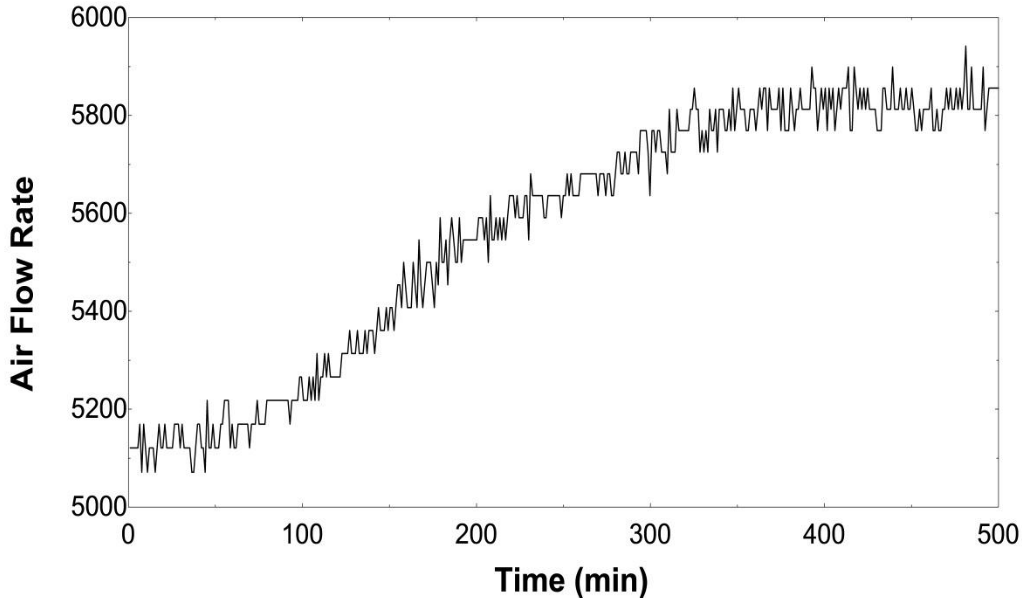


Figure 3.15 Air Flow Rate (m^3/h)

3.1.2. Cooling tower cycle

Water is used as refrigerant in the Cooling Tower Cycle. Two identical pumps are pumping water to Refrigeration Unit Cycle of Air Handling Units. Flow rate need for the cycle is equal to 6 m^3/h. Excess flow rate is passed through water-defrost cycle. Thus, Cooling Tower Cycle is completed.

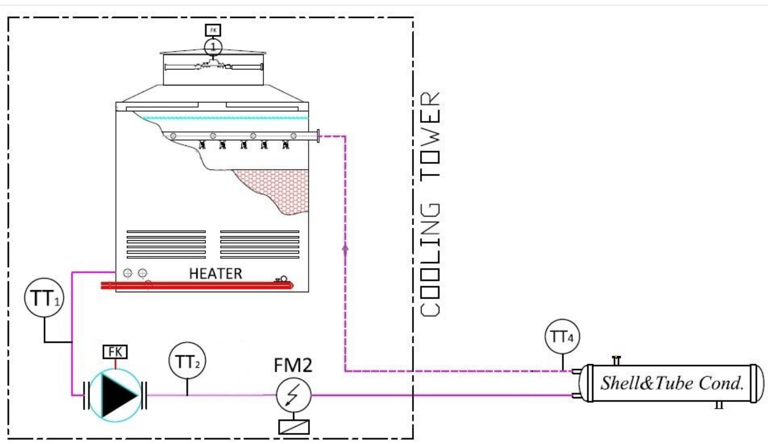


Figure 3.16 Cooling Tower Cycle

Temperatures are measured at the exit and inlet of the pump as tower outlet water temperature and water temperature pumped to the system. The return water temperature is measured at the exit of the shell and tube condenser as tower inlet water temperature.

3.1.2.1. Condenser cycle

Water is transferred from the pump at +30 ^{0}C temperature and 6 m^3/h flow rate to the shell and tube condenser of refrigeration unit cycle. Water is extracted heat from the refrigerant R404A and exited at +35 ^{0}C temperature. Water is rejected heat by cooling tower. Thus, cycle is completed.

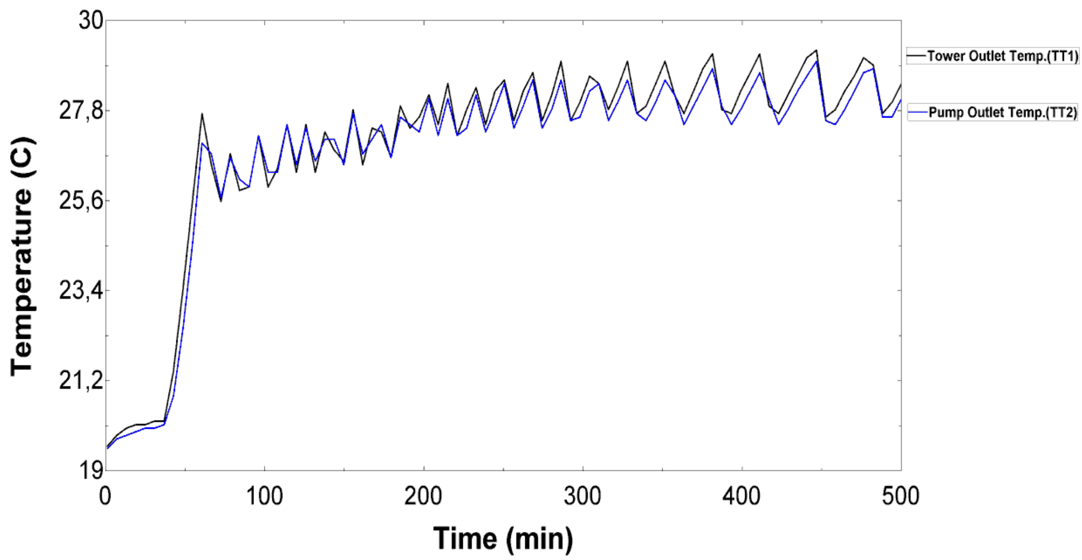


Figure 3.17 Temperature variations of water at cooling tower and circulation pump outlets

Pump outlet temperature has a 26.9 ^{0}C as average which is also the same temperature sent to the shell and tube condenser which condensing the R404A refrigerant. There is a slight difference between tower and pump outlet conditions in temperature values which is caused by the pump work in the water.

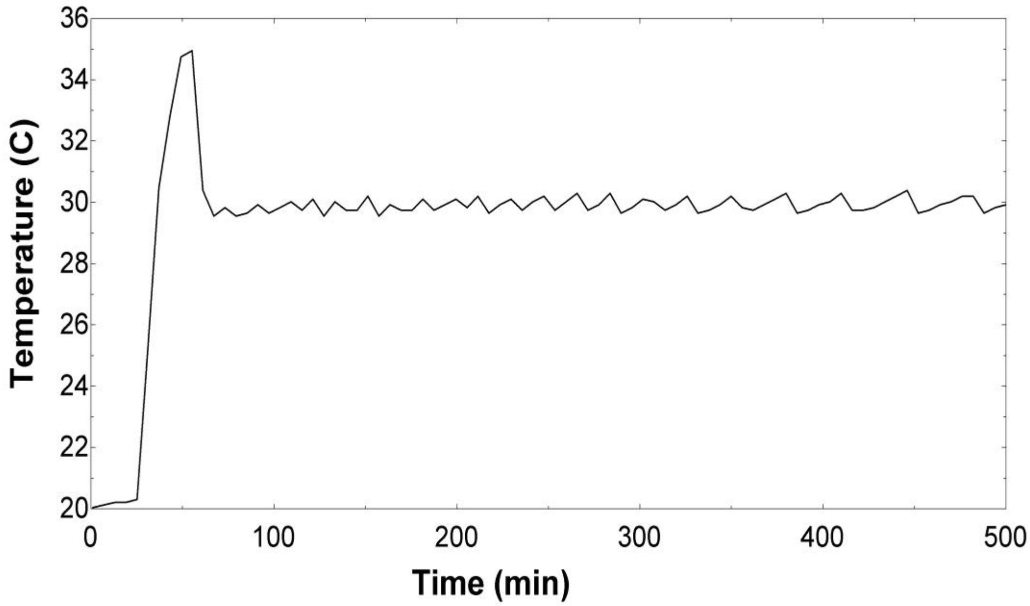


Figure 3.18 Shell and Tube Condenser Outlet Temperature

As you can see in the Figure outlet temperature of the condenser is about 30 ^{0}C. This means that adequate heat transfer is realized inside the condenser with these temperature values.

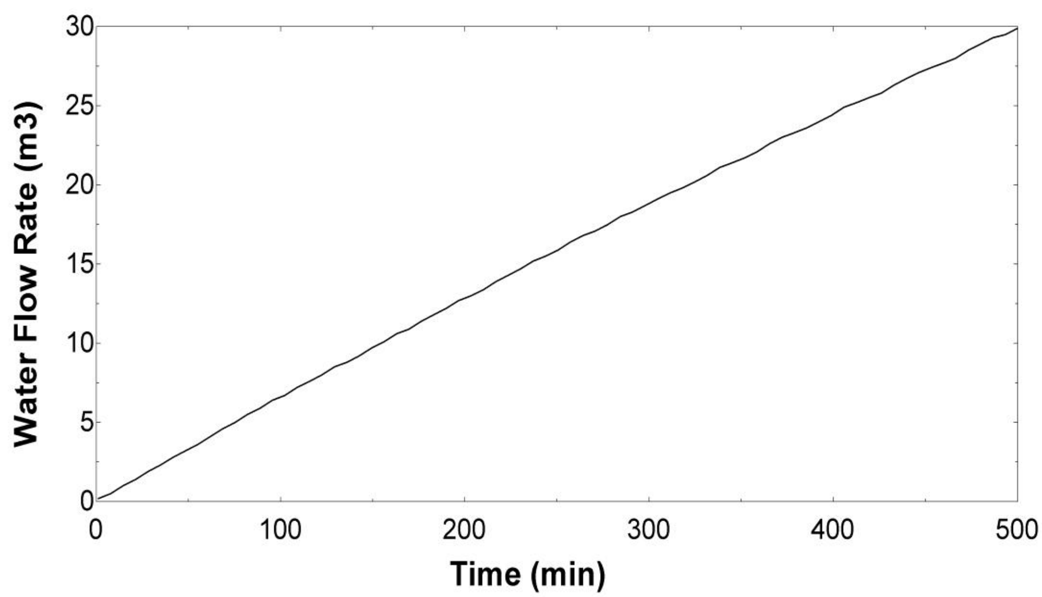


Figure 3.19 Water Flow Rate in the Cooling Tower Cycle

Refrigeration Unit Cycle operates with respect to the measured flow rate of the water. To avoid high temperature fault on the condenser side of the system we have to control that adequate amount of the water is passed through the water lines and reach to the condenser, so 3 m^3 minimum level of water flow is designated. If this amount of water does not pass through system the liquid solenoid of the refrigeration will not be opened and system gives an error to the user. As seen in the Figure 3.20 total of 30 m^3 of water circulates during the experimentation period.

3.1.3. Refrigeration unit cycle

Refrigerant R404A is entered to the compressors at the temperature of Adiabatic Testing Room corresponding to the Standard Conditions. Refrigerant is exited from compressor at +45 ^{0}C and 19.6 bars. R404A is passed through Condenser-2 and become liquid as it is rejected heat to the water coming from the cooling tower. A 30-L liquid receiver is used to provide saturated liquid to the evaporators. Two evaporators and expansion valve fed by the same line and Testing Room evaporator and Shell Room Air Handling Unit evaporators are separated with by-pass lines. Refrigerant is exited from expansion valves with pressure value with respect to the regulated temperature inside the Adiabatic Testing Room. Refrigerant is evaporated in the AHU evaporators then sent to the compressor as saturated vapor.

Compressor speed control is done using data gathered from the compressor suction pressure transmitter. Automation controls the speed considering equal-aging and using invertors connected to each compressor.

Suction pressure value is set to 0.7 bars to reach proper evaporation temperature at evaporators inside the AHUs. Since inner room is set to the temperature value of -34 ^{0}C, outer room temperature has to reach the same value.

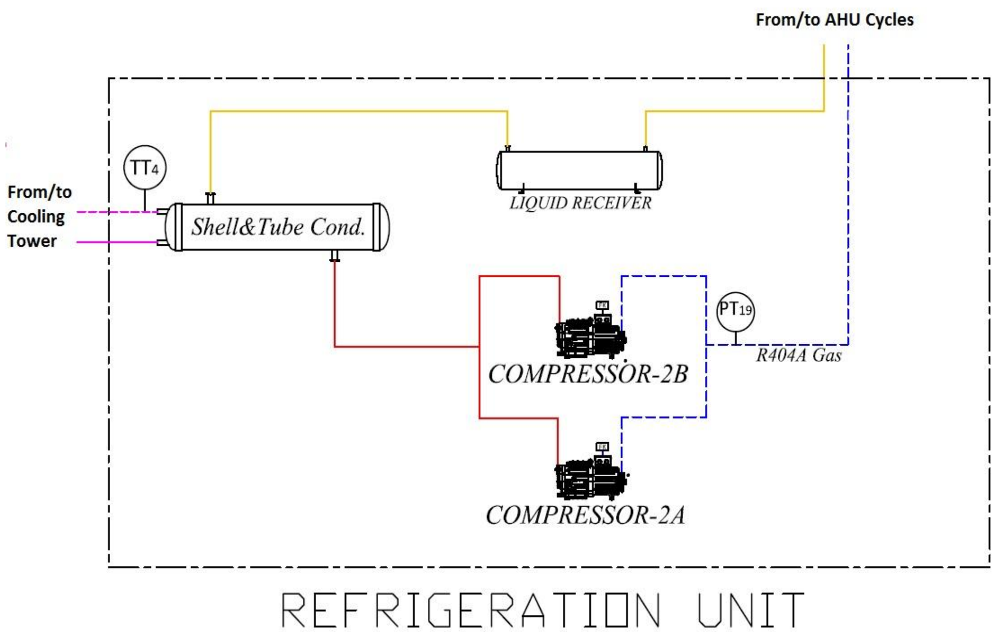


Figure 3.20 Refrigeration Unit Cycle

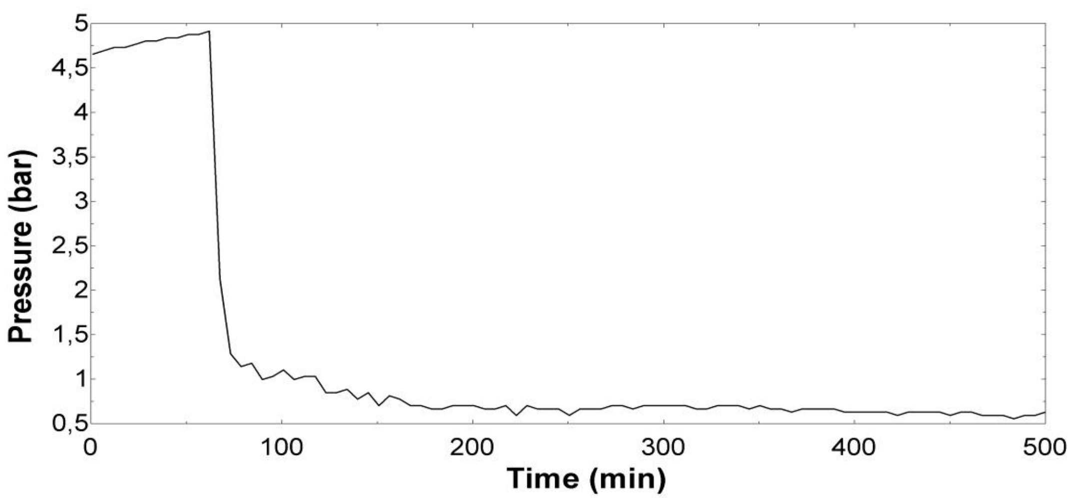


Figure 3.21 Suction Pressure (PT19) Values of Refrigeration Unit Cycle Compressors

4. CONCLUSION

Over the entire period of this thesis study all the works that is planned is accomplished with high accuracy.

A commercial CO_2 Cascade Refrigeration System has been built. To prove that it is applicable in the refrigeration industry several experiments must be done. So a calorimetric testing room regarding to the ASHRAE standards has been built. To control temperature level inside the rooms that experiments took place an auxiliary refrigeration system has been built accompanying with a cooling tower to control condensing temperature of this high capacity refrigeration system. Cooling tower gave an opportunity to control the temperature accurately since which is very important to gather proper data isolated from the ambient effects.

The most important part of the experiments was testing CO_2 evaporator that was manufactured whether it had tabulated capacity and efficiency in the designing period. Whole cascade system based on the capacity value of the evaporator. First of all a CO_2 compressor and its equipments for lubrication and control have been selected. With the designated low temperature cycle a R404A refrigeration unit was selected based on the condensation capacity of the LTC. As HTC control condensation pressure of the CO_2 it was important that doing it so. HTC refrigeration unit is based on the capacity that is needed in cascade condenser. And a cascade condenser is selected with the same logic.

Experiments showed that without any major problem the testing room temperature of -34 ^{0}C is achievable with this set of components. And with the thermodynamic calculations catalogue capacity value of 7.75 kW of tested evaporator is achieved with the accuracy of %97.5. Experiment was last 7.5 hours long and desired temperature is reached at the end of 6th hour. The room temperature was +25 ^{0}C at the beginning.

Using gathered data from the all sensors located on the cascade refrigeration cycle both HTC and LTC; a COP calculation was made. Overall COP of the system calculated using evaporator actual capacity and the electrical consumptions of CO_2 and R404A compressors. Overall COP of the system is found to be as 0.974. That value obliges system development and done more experiments with different condensation and evaporation temperatures to ensure more efficient system.

At the end; this thesis study allowed us to develop, manufacture and test a Cascade Refrigeration System using CO2 as a refrigerant. As the global warming and ozone depletion is increasing day by day, refrigeration systems using synthetic refrigerants have to be replaced with the systems using natural refrigerants. Using natural refrigerants can cause changes in the refrigeration system design and because of its unfamiliarity; people and companies working on this topic have to be encouraged and supported financially.

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APPENDIX

APPENDIX – A REFRIGERANT PROPERTIES

A-1 R404A P-T CHART

Temp. °C	Pressure kPa		spec. Density kg/m³		spec. Volume m³/kg		spec. Enthalpy kJ/kg			spec. Entropy kJ/kg K	
	liquid	gas	liquid	gas	liquid	gas	liquid	latent	gas	liquid	gas
-105	1,7976	1,5654	1481,1	0,10961	0,00067516	9,1233	67,24	236,98	304,21	0,39147	1,8093
-104	1,9810	1,7316	1478,1	0,12056	0,00067656	8,2948	68,48	236,32	304,80	0,39885	1,8042
-103	2,1801	1,9126	1475,0	0,13241	0,00067796	7,5525	69,72	235,66	305,38	0,40617	1,7992
-102	2,3960	2,1094	1472,0	0,14521	0,00067936	6,8865	70,96	235,01	305,97	0,41343	1,7943
-101	2,6298	2,3231	1469,0	0,15903	0,00068076	6,288	72,20	234,36	306,56	0,42063	1,7895
-100	2,8827	2,5549	1465,9	0,17393	0,00068216	5,7493	73,43	233,72	307,15	0,42777	1,7849
-99	3,1560	2,8060	1462,9	0,18998	0,00068356	5,2638	74,66	233,08	307,74	0,43486	1,7803
-98	3,4508	3,0775	1459,9	0,20723	0,00068496	4,8255	75,89	232,44	308,33	0,44189	1,7759
-97	3,7686	3,3709	1456,9	0,22576	0,00068637	4,4294	77,12	231,80	308,92	0,44887	1,7716
-96	4,1108	3,6874	1454,0	0,24565	0,00068778	4,0709	78,34	231,18	309,52	0,4558	1,7673
-95	4,4787	4,0286	1451,0	0,26696	0,00068919	3,7459	79,56	230,55	310,11	0,46269	1,7632
-94	4,8741	4,3960	1448,0	0,28977	0,0006906	3,451	80,79	229,92	310,71	0,46953	1,7591
-93	5,2983	4,7910	1445,0	0,31416	0,00069202	3,1831	82,01	229,30	311,31	0,47632	1,7552
-92	5,7531	5,2153	1442,1	0,34022	0,00069344	2,9392	83,23	228,68	311,91	0,48307	1,7513
-91	6,2402	5,6706	1439,1	0,36803	0,00069486	2,7171	84,44	228,07	312,51	0,48977	1,7475
-90	6,7614	6,1586	1436,2	0,39768	0,00069629	2,5146	85,66	227,45	313,11	0,49643	1,7438
-89	7,3184	6,6812	1433,2	0,42926	0,00069772	2,3296	86,88	226,83	313,71	0,50305	1,7402
-88	7,9132	7,2401	1430,3	0,46286	0,00069916	2,1605	88,09	226,22	314,31	0,50964	1,7367
-87	8,5478	7,8375	1427,4	0,49858	0,00070059	2,0057	89,31	225,61	314,92	0,51618	1,7333
-86	9,2242	8,4751	1424,4	0,53651	0,00070204	1,8639	90,52	225,00	315,52	0,52268	1,7299
-85	9,9444	9,1552	1421,5	0,57676	0,00070349	1,7338	91,74	224,39	316,13	0,52915	1,7266
-84	10,7110	9,8799	1418,6	0,61944	0,00070494	1,6144	92,95	223,78	316,73	0,53559	1,7234
-83	11,525	10,651	1415,6	0,66464	0,0007064	1,5046	94,17	223,18	317,34	0,54198	1,7203
-82	12,391	11,472	1412,7	0,71248	0,00070786	1,4035	95,38	222,57	317,95	0,54835	1,7172
-81	13,309	12,344	1409,8	0,76308	0,00070933	1,3105	96,59	221,96	318,55	0,55468	1,7142
-80	14,282	13,270	1406,9	0,81654	0,00071081	1,2247	97,81	221,35	319,16	0,56097	1,7113
-79	15,314	14,252	1403,9	0,87298	0,00071229	1,1455	99,02	220,75	319,77	0,56724	1,7084
-78	16,406	15,293	1401,0	0,93253	0,00071378	1,0723	100,24	220,14	320,38	0,57348	1,7056
-77	17,561	16,396	1398,1	0,99531	0,00071527	1,0047	101,45	219,54	320,99	0,57968	1,7029
-76	18,782	17,563	1395,1	1,0615	0,00071677	0,94211	102,67	218,93	321,60	0,58585	1,7002
-75	20,072	18,797	1392,2	1,1311	0,00071828	0,88411	103,88	218,33	322,21	0,592	1,6976
-74	21,433	20,100	1389,3	1,2043	0,0007198	0,83035	105,10	217,73	322,83	0,59812	1,6951
-73	22,869	21,477	1386,4	1,2813	0,00072132	0,78045	106,31	217,13	323,44	0,6042	1,6926
-72	24,382	22,929	1383,4	1,3622	0,00072285	0,73411	107,53	216,52	324,05	0,61027	1,6902
-71	25,976	24,461	1380,5	1,4471	0,00072439	0,69103	108,75	215,91	324,66	0,6163	1,6878
-70	27,654	26,074	1377,5	1,5362	0,00072593	0,65095	109,97	215,31	325,28	0,62231	1,6855
-69	29,418	27,773	1374,6	1,6296	0,00072748	0,61364	111,19	214,70	325,89	0,62829	1,6832
-68	31,273	29,560	1371,7	1,7275	0,00072905	0,57887	112,41	214,09	326,50	0,63425	1,681
-67	33,221	31,438	1368,7	1,83	0,00073062	0,54645	113,63	213,48	327,11	0,64018	1,6788
-66	35,266	33,412	1365,8	1,9373	0,0007322	0,51618	114,85	212,88	327,73	0,64609	1,6767
-65	37,412	35,485	1362,8	2,0495	0,00073378	0,48792	116,08	212,26	328,34	0,65198	1,6747
-64	39,662	37,660	1359,8	2,1668	0,00073538	0,4615	117,30	211,65	328,95	0,65784	1,6727
-63	42,019	39,941	1356,9	2,2894	0,00073699	0,43679	118,53	211,04	329,57	0,66368	1,6707
-62	44,488	42,332	1353,9	2,4174	0,0007386	0,41366	119,75	210,43	330,18	0,6695	1,6688
-61	47,072	44,836	1350,9	2,551	0,00074023	0,392	120,98	209,81	330,79	0,67529	1,6669
-60	49,774	47,457	1348,0	2,6904	0,00074187	0,37169	122,21	209,19	331,40	0,68107	1,6651
-59	52,600	50,199	1345,0	2,8357	0,00074351	0,35264	123,44	208,58	332,02	0,68682	1,6633
-58	55,553	53,066	1342,0	2,9872	0,00074517	0,33476	124,68	207,95	332,63	0,69255	1,6616
-57	58,636	56,063	1339,0	3,1449	0,00074684	0,31797	125,91	207,33	333,24	0,69826	1,6599
-56	61,854	59,192	1336,0	3,3092	0,00074851	0,30219	127,15	206,70	333,85	0,70396	1,6582

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Temp. °C	Pressure kPa		spec. Density kg/m³		spec. Volume m³/kg		spec. Enthalpy kJ/kg			spec. Entropy kJ/kg K	
	liquid	gas	liquid	gas	liquid	gas	liquid	latent	gas	liquid	gas
-55	65,212	62,459	1333,0	3,4801	0,0007502	0,28735	128,38	206,08	334,46	0,70963	1,6566
-54	68,713	65,868	1330,0	3,6579	0,0007519	0,27338	129,62	205,45	335,07	0,71528	1,655
-53	72,361	69,423	1326,9	3,8428	0,00075362	0,26023	130,86	204,82	335,68	0,72091	1,6535
-52	76,161	73,127	1323,9	4,035	0,00075534	0,24783	132,10	204,19	336,29	0,72653	1,652
-51	80,118	76,987	1320,9	4,2346	0,00075708	0,23615	133,35	203,55	336,90	0,73213	1,6505
-50	84,236	81,005	1317,8	4,4419	0,00075882	0,22513	134,59	202,92	337,51	0,73771	1,6491
-49	88,519	85,188	1314,8	4,657	0,00076058	0,21473	135,84	202,27	338,11	0,74327	1,6477
-48	92,972	89,538	1311,7	4,8803	0,00076236	0,20491	137,09	201,63	338,72	0,74881	1,6463
-47	97,600	94,062	1308,7	5,1119	0,00076414	0,19562	138,34	200,99	339,33	0,75434	1,645
-46	102,410	98,763	1305,6	5,352	0,00076594	0,18685	139,59	200,34	339,93	0,75985	1,6437
-45	107,40	103,65	1302,5	5,6008	0,00076776	0,17855	140,85	199,68	340,53	0,76535	1,6424
-44	112,58	108,72	1299,4	5,8586	0,00076959	0,17069	142,10	199,04	341,14	0,77083	1,6412
-43	117,95	113,98	1296,3	6,1257	0,00077143	0,16325	143,36	198,38	341,74	0,77629	1,64
-42	123,53	119,44	1293,2	6,4022	0,00077328	0,1562	144,62	197,72	342,34	0,78174	1,6388
-41	129,30	125,11	1290,1	6,6883	0,00077515	0,14951	145,89	197,05	342,94	0,78717	1,6377
-40	135,29	130,98	1286,9	6,9844	0,00077704	0,14318	147,15	196,38	343,53	0,79259	1,6365
-39	141,49	137,06	1283,8	7,2907	0,00077894	0,13716	148,42	195,71	344,13	0,79799	1,6355
-38	147,91	143,36	1280,6	7,6074	0,00078086	0,13145	149,69	195,04	344,73	0,80338	1,6344
-37	154,55	149,88	1277,5	7,9348	0,00078279	0,12603	150,96	194,36	345,32	0,80876	1,6333
-36	161,42	156,63	1274,3	8,2731	0,00078474	0,12087	152,23	193,68	345,91	0,81412	1,6323
-35	168,53	163,62	1271,1	8,6227	0,00078671	0,11597	153,51	193,00	346,51	0,81947	1,6313
-34	175,88	170,84	1267,9	8,9837	0,00078869	0,11131	154,79	192,31	347,10	0,8248	1,6304
-33	183,47	178,31	1264,7	9,3565	0,00079069	0,10688	156,07	191,61	347,68	0,83012	1,6294
-32	191,31	186,03	1261,5	9,7413	0,00079271	0,10266	157,35	190,92	348,27	0,83543	1,6285
-31	199,42	194,00	1258,3	10,138	0,00079475	0,098634	158,64	190,22	348,86	0,84073	1,6276
-30	207,78	202,23	1255,0	10,548	0,0007968	0,094802	159,93	189,51	349,44	0,84601	1,6267
-29	216,41	210,73	1251,8	10,971	0,00079887	0,09115	161,22	188,80	350,02	0,85128	1,6259
-28	225,32	219,51	1248,5	11,407	0,00080097	0,087667	162,52	188,08	350,60	0,85654	1,625
-27	234,50	228,56	1245,2	11,856	0,00080308	0,084343	163,81	187,37	351,18	0,86179	1,6242
-26	243,97	237,89	1241,9	12,32	0,00080521	0,081171	165,11	186,64	351,75	0,86703	1,6234
-25	253,73	247,52	1238,6	12,797	0,00080737	0,078143	166,41	185,92	352,33	0,87225	1,6226
-24	263,79	257,44	1235,3	13,289	0,00080954	0,07525	167,72	185,18	352,90	0,87747	1,6219
-23	274,15	267,66	1231,9	13,796	0,00081174	0,072485	169,03	184,44	353,47	0,88267	1,6211
-22	284,81	278,19	1228,6	14,318	0,00081396	0,069843	170,34	183,70	354,04	0,88787	1,6204
-21	295,80	289,03	1225,2	14,855	0,0008162	0,067315	171,65	182,95	354,60	0,89305	1,6197
-20	307,10	300,19	1221,8	15,409	0,00081846	0,064897	172,97	182,19	355,16	0,89822	1,619
-19	318,73	311,68	1218,4	15,979	0,00082075	0,062583	174,29	181,43	355,72	0,90339	1,6183
-18	330,69	323,50	1215,0	16,565	0,00082306	0,060368	175,61	180,67	356,28	0,90854	1,6177
-17	343,00	335,66	1211,5	17,169	0,0008254	0,058246	176,93	179,91	356,84	0,91369	1,617
-16	355,64	348,16	1208,1	17,789	0,00082776	0,056213	178,26	179,13	357,39	0,91883	1,6164
-15	368,64	361,01	1204,6	18,428	0,00083015	0,054265	179,60	178,34	357,94	0,92395	1,62
-14	382,00	374,22	1201,1	19,085	0,00083256	0,052397	180,93	177,56	358,49	0,92907	1,6152
-13	395,72	387,80	1197,6	19,761	0,000835	0,050605	182,27	176,76	359,03	0,93419	1,6146
-12	409,81	401,74	1194,1	20,456	0,00083747	0,048886	183,61	175,97	359,58	0,93929	1,614
-11	424,28	416,06	1190,5	21,17	0,00083997	0,047236	184,96	175,15	360,11	0,94439	1,6134
-10	439,13	430,76	1186,9	21,905	0,0008425	0,045652	186,31	174,34	360,65	0,94947	1,6129
-9	454,37	445,86	1183,4	22,66	0,00084505	0,044131	187,66	173,52	361,18	0,95456	1,6123
-8	470,01	461,35	1179,7	23,436	0,00084764	0,04267	189,01	172,70	361,71	0,95963	1,6118
-7	486,05	477,24	1176,1	24,233	0,00085026	0,041265	190,37	171,87	362,24	0,9647	1,6113
-6	502,50	493,54	1172,5	25,053	0,00085291	0,039915	191,74	171,02	362,76	0,96976	1,6107

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Temp. °C	Pressure kPa		spec. Density kg/m^3		spec. Volume m^3/kg		spec. Enthalpy kJ/kg			spec. Entropy kJ/kg K	
	liquid	gas	liquid	gas	liquid	gas	liquid	latent	gas	liquid	gas
-5	519,37	510,26	1168,8	25,895	0,0008556	0,038617	193,10	170,18	363,28	0,97481	1,6102
-4	536,66	527,40	1165,1	26,761	0,00085832	0,037368	194,48	169,32	363,80	0,97986	1,6097
-3	554,38	544,97	1161,3	27,65	0,00086107	0,036167	195,85	168,46	364,31	0,9849	1,6092
-2	572,54	562,98	1157,6	28,563	0,00086386	0,03501	197,23	167,59	364,82	0,98994	1,6087
-1	591,14	581,43	1153,8	29,502	0,00086669	0,033896	198,61	166,71	365,32	0,99497	1,6082
0	610,19	600,33	1150,0	30,465	0,00086956	0,032824	200,00	165,82	365,82	1,00	1,6078
1	629,70	619,69	1146,2	31,456	0,00087246	0,031791	201,39	164,93	366,32	1,005	1,6073
2	649,68	639,52	1142,3	32,473	0,00087541	0,030795	202,79	164,02	366,81	1,01	1,6068
3	670,13	659,82	1138,4	33,517	0,00087839	0,029836	204,19	163,10	367,29	1,0151	1,6064
4	691,06	680,59	1134,5	34,59	0,00088142	0,02891	205,59	162,19	367,78	1,0201	1,6059
5	712,47	701,86	1130,6	35,692	0,00088449	0,028018	207,00	161,25	368,25	1,0251	1,6054
6	734,38	723,62	1126,6	36,823	0,00088761	0,027157	208,41	160,32	368,73	1,0301	1,605
7	756,79	745,88	1122,6	37,985	0,00089078	0,026326	209,83	159,37	369,20	1,0351	1,6045
8	779,71	768,65	1118,6	39,179	0,00089399	0,025524	211,26	158,40	369,66	1,0401	1,6041
9	803,14	791,94	1114,5	40,405	0,00089725	0,024749	212,68	157,44	370,12	1,0451	1,6036
10	827,10	815,75	1110,4	41,664	0,00090057	0,024002	214,12	156,45	370,57	1,0501	1,6032
11	851,59	840,09	1106,3	42,957	0,00090394	0,023279	215,56	155,45	371,01	1,0551	1,6027
12	876,61	864,97	1102,1	44,286	0,00090736	0,022581	217,00	154,45	371,45	1,06	1,6023
13	902,19	890,40	1097,9	45,65	0,00091084	0,021906	218,45	153,44	371,89	1,065	1,6018
14	928,32	916,39	1093,6	47,052	0,00091437	0,021253	219,90	152,42	372,32	1,07	1,6013
15	955,01	942,94	1089,4	48,491	0,00091797	0,020622	221,36	151,38	372,74	1,075	1,6009
16	982,27	970,06	1085,0	49,971	0,00092163	0,020012	222,83	150,32	373,15	1,08	1,6004
17	1010,10	997,76	1080,7	51,49	0,00092536	0,019421	224,30	149,26	373,56	1,085	1,5999
18	1038,50	1026,00	1076,3	53,052	0,00092915	0,018849	225,78	148,18	373,96	1,09	1,5994
19	1067,50	1054,90	1071,8	54,657	0,00093301	0,018296	227,26	147,10	374,36	1,095	1,5989
20	1097,20	1084,40	1067,3	56,306	0,00093695	0,01776	228,75	145,99	374,74	1,1	1,5984
21	1127,40	1114,50	1062,7	58,002	0,00094096	0,017241	230,25	144,87	375,12	1,105	1,5979
22	1158,20	1145,20	1058,2	59,744	0,00094504	0,016738	231,75	143,74	375,49	1,11	1,5974
23	1189,70	1176,60	1053,5	61,536	0,00094921	0,016251	233,26	142,59	375,85	1,115	1,5969
24	1221,80	1208,60	1048,8	63,379	0,00095346	0,015778	234,77	141,44	376,21	1,12	1,5964
25	1254,60	1241,20	1044,1	65,274	0,00095781	0,01532	236,30	140,25	376,55	1,125	1,5958
26	1288,00	1274,50	1039,2	67,223	0,00096224	0,014876	237,83	139,06	376,89	1,13	1,5953
27	1322,10	1308,40	1034,4	69,229	0,00096677	0,014445	239,37	137,85	377,22	1,1351	1,5947
28	1356,80	1343,10	1029,5	71,293	0,00097139	0,014027	240,91	136,62	377,53	1,1401	1,5941
29	1392,20	1378,40	1024,5	73,418	0,00097612	0,013621	242,47	135,37	377,84	1,1451	1,5935
30	1428,30	1414,40	1019,4	75,605	0,00098096	0,013227	244,03	134,11	378,14	1,1502	1,5929
31	1465,10	1451,10	1014,3	77,858	0,00098592	0,012844	245,60	132,82	378,42	1,1552	1,5923
32	1502,70	1488,50	1009,1	80,179	0,00099099	0,012472	247,18	131,52	378,70	1,1603	1,5916
33	1540,90	1526,60	1003,8	82,57	0,00099619	0,012111	248,77	130,19	378,96	1,1654	1,5909
34	1579,80	1565,50	998,5	85,035	0,0010015	0,01176	250,37	128,84	379,21	1,1704	1,5903
35	1619,50	1605,10	993,1	87,577	0,001007	0,011418	251,97	127,48	379,45	1,1755	1,5895
36	1660,00	1645,50	987,6	90,2	0,0010126	0,011087	253,59	126,09	379,68	1,1806	1,5888
37	1701,20	1686,60	982,0	92,906	0,0010184	0,010764	255,22	124,67	379,89	1,1858	1,588
38	1743,10	1728,50	976,3	95,7	0,0010243	0,010449	256,86	123,22	380,08	1,1909	1,5872
39	1785,90	1771,20	970,5	98,586	0,0010304	0,010143	258,51	121,75	380,26	1,1961	1,5864
40	1829,40	1814,60	964,7	101,57	0,0010366	0,0098456	260,17	120,26	380,43	1,2012	1,5855
41	1873,70	1858,90	958,7	104,65	0,0010431	0,0095555	261,85	118,73	380,58	1,2064	1,5846
42	1918,90	1904,00	952,6	107,84	0,0010498	0,0092728	263,53	117,18	380,71	1,2116	1,5837
43	1964,80	1949,90	946,4	111,15	0,0010567	0,0089972	265,23	115,59	380,82	1,2169	1,5827
44	2011,60	1996,70	940,1	114,57	0,0010638	0,0087284	266,95	113,97	380,92	1,2221	1,5817

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Temp. °C	Pressure kPa		spec. Density kg/m³		spec. Volume m³/kg		spec. Enthalpy kJ/kg			spec. Entropy kJ/kg K	
	liquid	gas	liquid	gas	liquid	gas	liquid	latent	gas	liquid	gas
45	2059,30	2044,30	933,6	118,12	0,0010711	0,0084663	268,68	112,31	380,99	1,2274	1,5807
46	2107,80	2092,80	927,0	121,8	0,0010787	0,0082104	270,42	110,63	381,05	1,2327	1,5796
47	2157,20	2142,20	920,3	125,62	0,0010866	0,0079605	272,18	108,90	381,08	1,2381	1,5784
48	2207,40	2192,40	913,4	129,59	0,0010949	0,0077164	273,96	107,12	381,08	1,2434	1,5772
49	2258,60	2243,60	906,3	133,73	0,0011034	0,0074778	275,76	105,30	381,06	1,2488	1,5759
50	2310,60	2295,70	899,1	138,04	0,0011123	0,0072444	277,57	103,44	381,01	1,2543	1,5746
51	2363,60	2348,70	891,7	142,53	0,0011215	0,007016	279,41	101,52	380,93	1,2598	1,5732
52	2417,50	2402,70	884,0	147,22	0,0011312	0,0067924	281,27	99,55	380,82	1,2653	1,5717
53	2472,40	2457,70	876,2	152,13	0,0011413	0,0065732	283,15	97,53	380,68	1,2709	1,5701
54	2528,30	2513,60	868,1	157,28	0,001152	0,0063582	285,06	95,44	380,50	1,2766	1,5685
55	2585,10	2570,50	859,7	162,68	0,0011632	0,0061471	286,99	93,29	380,28	1,2823	1,5667
56	2643,00	2628,50	851,1	168,36	0,001175	0,0059396	288,96	91,05	380,01	1,288	1,5648
57	2701,90	2687,50	842,1	174,35	0,0011875	0,0057355	290,95	88,74	379,69	1,2939	1,5628
58	2761,80	2747,60	832,8	180,69	0,0012007	0,0055343	292,99	86,33	379,32	1,2998	1,5607
59	2822,80	2808,80	823,1	187,41	0,0012149	0,0053359	295,06	83,83	378,89	1,3059	1,5584
60	2884,90	2871,10	813,0	194,57	0,00123	0,0051396	297,18	81,22	378,40	1,312	1,5559
61	2948,10	2934,50	802,4	202,22	0,0012463	0,0049452	299,35	78,47	377,82	1,3183	1,5532
62	3012,40	2999,10	791,2	210,43	0,0012639	0,0047521	301,58	75,58	377,16	1,3247	1,5503
63	3077,90	3064,90	779,3	219,32	0,0012832	0,0045596	303,87	72,53	376,40	1,3313	1,5471
64	3144,60	3132,00	766,6	228,99	0,0013044	0,004367	306,24	69,27	375,51	1,338	1,5436
65	3212,60	3200,40	753,0	239,62	0,001328	0,0041733	308,71	65,77	374,48	1,3451	1,5397
66	3281,80	3270,10	738,2	251,43	0,0013547	0,0039772	311,30	61,97	373,27	1,3525	1,5353
67	3352,30	3341,20	721,8	264,78	0,0013854	0,0037767	314,04	57,80	371,84	1,3602	1,5302
68	3424,20	3413,80	703,3	280,19	0,0014218	0,003569	316,98	53,11	370,09	1,3686	1,5243
69	3497,60	3488,00	681,8	298,59	0,0014667	0,003349	320,22	47,69	367,91	1,3778	1,5172
70	3572,50	3564,00	655,3	321,83	0,0015259	0,0031072	323,96	41,08	365,04	1,3883	1,5081

These data were collected by application REFPRO7.

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A-2 CO2 P-T CHART

Temp. °C	Pressure kPa	spec. Density kg/m³		spec. Volume m³/kg		spec. Enthalpy kJ/kg			spec. Entropy kJ/kg K	
		liquid	gas	liquid	gas	liquid	latent	gas	liquid	gas
-50	682,34	1154,60	17,925	0,00086613	0,055789	92,943	339,737	432,68	0,57939	2,1018
-49	710,49	1150,80	18,638	0,00086893	0,053654	94,922	338,068	432,99	0,58813	2,0963
-48	739,49	1147,10	19,373	0,00087176	0,051618	96,905	336,385	433,29	0,59684	2,0909
-47	769,37	1143,40	20,131	0,00087462	0,049674	98,891	334,689	433,58	0,60553	2,0855
-46	800,15	1139,60	20,912	0,00087752	0,047819	100,88	332,980	433,86	0,61418	2,0801
-45	831,84	1135,80	21,717	0,00088046	0,046046	102,87	331,260	434,13	0,62282	2,0747
-44	864,45	1132,00	22,547	0,00088343	0,044352	104,87	329,520	434,39	0,63143	2,0694
-43	898,00	1128,10	23,401	0,00088644	0,042733	106,87	327,770	434,64	0,64001	2,0642
-42	932,52	1124,20	24,281	0,00088949	0,041184	108,88	326,000	434,88	0,64858	2,0589
-41	968,01	1120,30	25,187	0,00089258	0,039702	110,89	324,220	435,11	0,65712	2,0537
-40	1004,50	1116,40	26,121	0,00089572	0,038284	112,9	322,420	435,32	0,66564	2,0485
-39	1042,00	1112,50	27,082	0,00089889	0,036925	114,92	320,610	435,53	0,67413	2,0434
-38	1080,50	1108,50	28,071	0,00090211	0,035624	116,95	318,770	435,72	0,68261	2,0382
-37	1120,10	1104,50	29,089	0,00090537	0,034377	118,98	316,920	435,90	0,69107	2,0331
-36	1160,70	1100,50	30,137	0,00090868	0,033181	121,01	315,060	436,07	0,69951	2,0281
-35	1202,40	1096,40	31,216	0,00091204	0,032035	123,05	313,180	436,23	0,70794	2,023
-34	1245,20	1092,40	32,326	0,00091545	0,030935	125,1	311,270	436,37	0,71634	2,018
-33	1289,10	1088,30	33,469	0,00091891	0,029879	127,15	309,360	436,51	0,72474	2,0129
-32	1334,20	1084,10	34,644	0,00092242	0,028865	129,2	307,420	436,62	0,73311	2,0079
-31	1380,40	1079,90	35,854	0,00092598	0,027891	131,27	305,460	436,73	0,74148	2,0029
-30	1427,80	1075,70	37,098	0,0009296	0,026956	133,34	303,480	436,82	0,74982	1,998
-29	1476,30	1071,50	38,378	0,00093328	0,026056	135,41	301,490	436,90	0,75816	1,993
-28	1526,10	1067,20	39,696	0,00093701	0,025192	137,5	299,460	436,96	0,76649	1,988
-27	1577,00	1062,90	41,051	0,00094081	0,02436	139,59	297,420	437,01	0,77481	1,9831
-26	1629,30	1058,60	42,445	0,00094467	0,02356	141,69	295,350	437,04	0,78311	1,9781
-25	1682,70	1054,20	43,88	0,0009486	0,022789	143,79	293,270	437,06	0,79141	1,9732
-24	1737,50	1049,80	45,356	0,0009526	0,022048	145,91	291,150	437,06	0,79971	1,9683
-23	1793,50	1045,30	46,875	0,00095666	0,021334	148,03	289,010	437,04	0,80799	1,9633
-22	1850,90	1040,80	48,437	0,0009608	0,020645	150,16	286,850	437,01	0,81627	1,9584
-21	1909,60	1036,30	50,045	0,00096502	0,019982	152,3	284,660	436,96	0,82455	1,9535
-20	1969,60	1031,70	51,7	0,00096931	0,019343	154,45	282,440	436,89	0,83283	1,9485
-19	2031,00	1027,00	53,402	0,00097369	0,018726	156,61	280,200	436,81	0,8411	1,9436
-18	2093,80	1022,30	55,155	0,00097815	0,018131	158,77	277,930	436,70	0,84937	1,9386
-17	2158,10	1017,60	56,959	0,0009827	0,017557	160,95	275,630	436,58	0,85765	1,9337
-16	2223,70	1012,80	58,816	0,00098734	0,017002	163,14	273,300	436,44	0,86593	1,9287
-15	2290,80	1008,00	60,728	0,00099208	0,016467	165,34	270,930	436,27	0,87421	1,9237
-14	2359,30	1003,10	62,697	0,00099692	0,01595	167,55	268,540	436,09	0,88249	1,9187
-13	2429,40	998,14	64,725	0,0010019	0,01545	169,78	266,110	435,89	0,89078	1,9137
-12	2501,00	993,13	66,814	0,0010069	0,014967	172,01	263,650	435,66	0,89908	1,9086
-11	2574,00	988,06	68,967	0,0010121	0,0145	174,26	261,150	435,41	0,90739	1,9036
-10	2648,70	982,93	71,185	0,0010174	0,014048	176,52	258,620	435,14	0,91571	1,8985
-9	2724,90	977,73	73,471	0,0010228	0,013611	178,8	256,040	434,84	0,92405	1,8934
-8	2802,70	972,46	75,829	0,0010283	0,013188	181,09	253,420	434,51	0,9324	1,8882
-7	2882,10	967,12	78,261	0,001034	0,012778	183,39	250,780	434,17	0,94076	1,883
-6	2963,20	961,70	80,77	0,0010398	0,012381	185,71	248,080	433,79	0,94915	1,8778
-5	3045,90	956,21	83,359	0,0010458	0,011996	188,05	245,330	433,38	0,95756	1,8725
-4	3130,30	950,63	86,032	0,0010519	0,011624	190,4	242,550	432,95	0,96599	1,8672
-3	3216,40	944,97	88,794	0,0010582	0,011262	192,77	239,710	432,48	0,97444	1,8618
-2	3304,20	939,22	91,647	0,0010647	0,010911	195,16	236,830	431,99	0,98293	1,8563
-1	3393,80	933,38	94,596	0,0010714	0,010571	197,57	233,890	431,46	0,99145	1,8509

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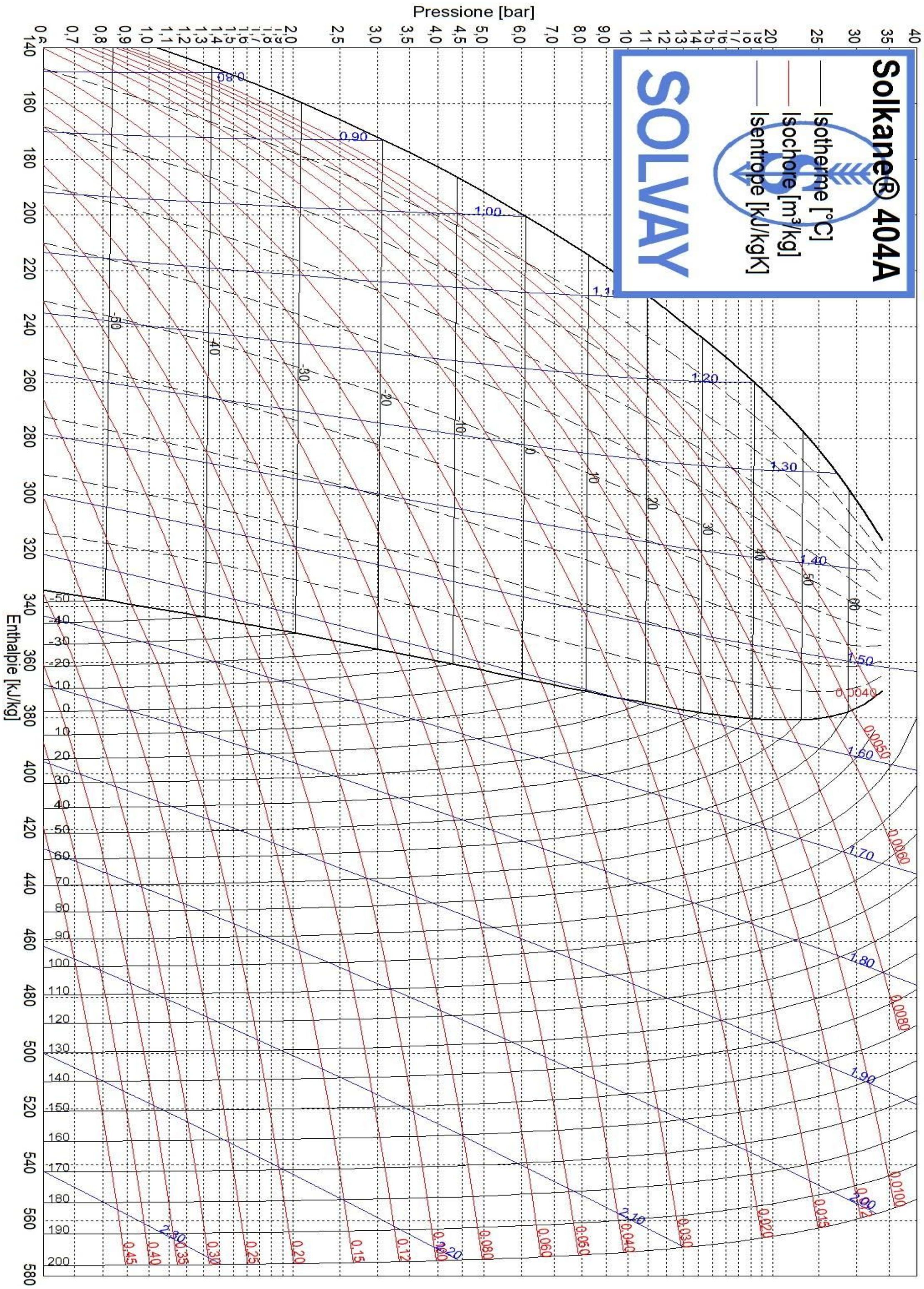
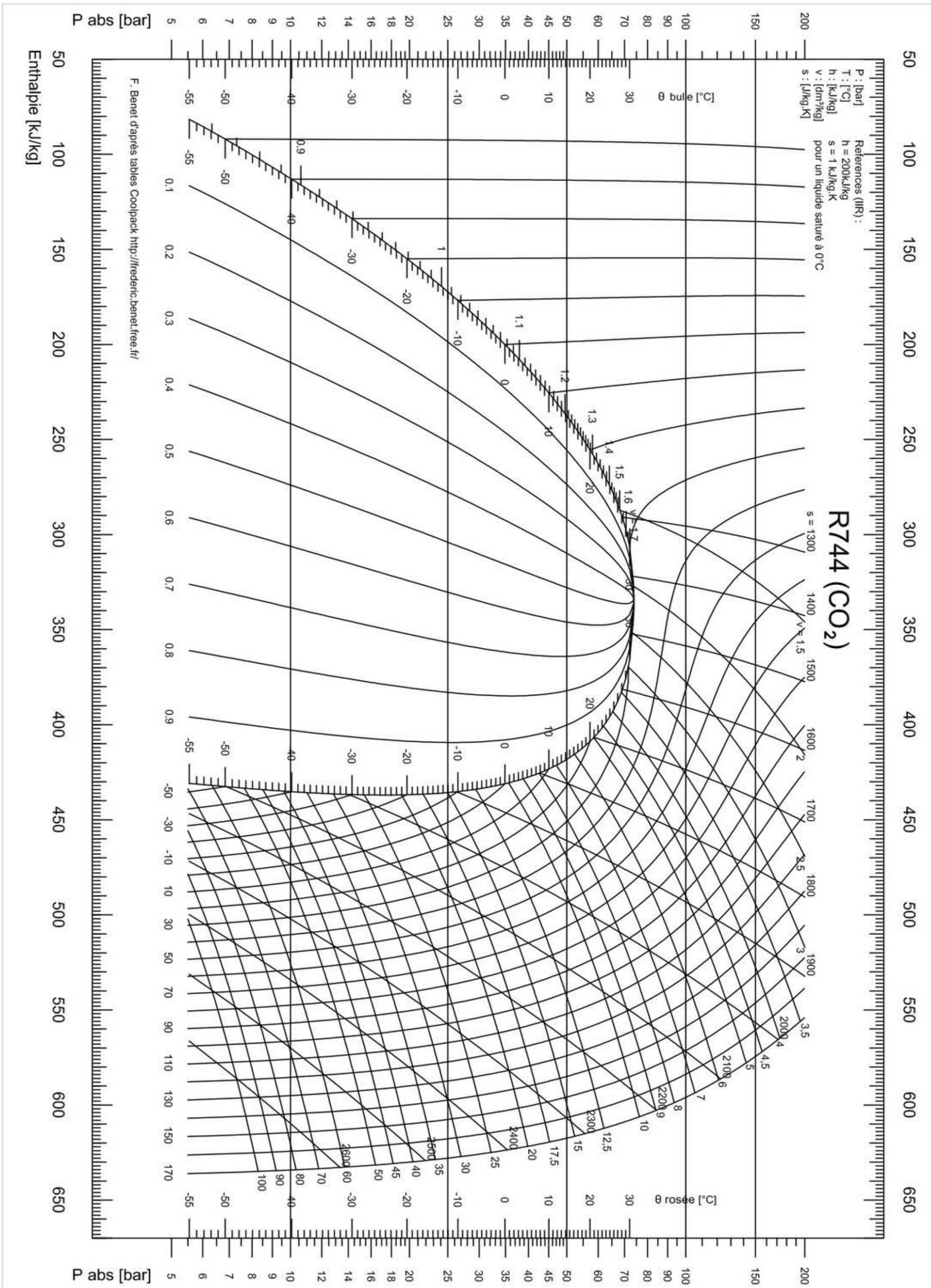
Temp. °C	Pressure kPa	spec. Density kg/m³		spec. Volume m³/kg		spec. Enthalpy kJ/kg			spec. Entropy kJ/kg K	
		liquid	gas	liquid	gas	liquid	latent	gas	liquid	gas
0	3485,10	927,43	97,647	0,0010782	0,010241	200	230,890	430,89	1,00	1,8453
1	3578,30	921,38	100,8	0,0010853	0,0099202	202,45	227,840	430,29	1,0086	1,8397
2	3673,30	915,23	104,07	0,0010926	0,0096085	204,93	224,720	429,65	1,0172	1,834
3	3770,10	908,95	107,46	0,0011002	0,0093056	207,43	221,540	428,97	1,0259	1,8282
4	3868,80	902,56	110,98	0,001108	0,009011	209,95	218,300	428,25	1,0346	1,8223
5	3969,50	896,03	114,62	0,001116	0,0087244	212,5	214,980	427,48	1,0434	1,8163
6	4072,00	889,36	118,41	0,0011244	0,0084454	215,08	211,590	426,67	1,0523	1,8102
7	4176,50	882,55	122,34	0,0011331	0,0081737	217,69	208,120	425,81	1,0612	1,8041
8	4283,10	875,58	126,44	0,0011421	0,0079089	220,34	204,550	424,89	1,0702	1,7977
9	4391,60	868,44	130,71	0,0011515	0,0076508	223,01	200,910	423,92	1,0792	1,7913
10	4502,20	861,12	135,16	0,0011613	0,0073988	225,73	197,150	422,88	1,0884	1,7847
11	4614,90	853,60	139,8	0,0011715	0,0071528	228,49	193,300	421,79	1,0976	1,7779
12	4729,70	845,87	144,67	0,0011822	0,0069125	231,29	189,330	420,62	1,107	1,771
13	4846,60	837,91	149,76	0,0011934	0,0066774	234,13	185,240	419,37	1,1165	1,7638
14	4965,80	829,70	155,11	0,0012053	0,0064472	237,03	181,020	418,05	1,1261	1,7565
15	5087,10	821,21	160,73	0,0012177	0,0062216	239,99	176,650	416,64	1,1359	1,7489
16	5210,80	812,41	166,66	0,0012309	0,0060003	243,01	172,110	415,12	1,1458	1,7411
17	5336,80	803,27	172,93	0,0012449	0,0057828	246,1	167,400	413,50	1,1559	1,7329
18	5465,10	793,76	179,57	0,0012598	0,0055688	249,26	162,500	411,76	1,1663	1,7244
19	5595,80	783,81	186,64	0,0012758	0,0053578	252,52	157,370	409,89	1,1769	1,7155
20	5729,10	773,39	194,2	0,001293	0,0051493	255,87	152,000	407,87	1,1877	1,7062
21	5864,80	762,40	202,32	0,0013116	0,0049427	259,33	146,340	405,67	1,1989	1,6964
22	6003,10	750,77	211,08	0,001332	0,0047375	262,93	140,330	403,26	1,2105	1,686
23	6144,00	738,36	220,62	0,0013543	0,0045326	266,68	133,950	400,63	1,2225	1,6749
24	6287,70	725,02	231,1	0,0013793	0,0043272	270,61	127,090	397,70	1,2352	1,6629
25	6434,20	710,50	242,73	0,0014075	0,0041198	274,78	119,650	394,43	1,2485	1,6498
26	6583,70	694,46	255,86	0,00144	0,0039083	279,26	111,450	390,71	1,2627	1,6353
27	6736,10	676,36	271,01	0,0014785	0,0036898	284,14	102,250	386,39	1,2783	1,6189
28	6891,80	655,28	289,11	0,0015261	0,0034589	289,62	91,580	381,20	1,2958	1,5999
29	7050,90	629,36	312,03	0,0015889	0,0032048	296,07	78,540	374,61	1,3163	1,5763
30	7213,70	593,31	345,1	0,0016855	0,0028977	304,55	60,580	365,13	1,3435	1,5433

These data were collected by application REFPRO7.

A-3 R404A LNP-H DIAGRAM



A-4 CO2 LNP-H DIAGRAM



CURRICULUM VITAE

Ali ÖZYURT was born in Bursa on July 20, 1988. He graduated from the Marmara University, Faculty of Engineering, Mechanical Engineering Department as mechanical engineer on June, 2011. After his bachelor degree, he started Master of Science in Mechanical Engineering at Marmara University, Institue for Graduate Studies in Pure and Applied Sciences. By 2014 he started to work at the Research and Development Department of PANEL SİSTEM Soğutma A.Ş as an engineer.