Thermodynamic Analysis and Comparison of CO_2/Propane and CO2/Isobutane Cascade Refrigeration System

by

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Dissertation submitted in partial fulfilment of
the requirements for the
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CERTIFICATION

CERTIFICATION OF APPROVAL

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A project dissertation submitted to the
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Approved by,

(Prof. Dr. Vijay Raj Raghavan)

UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

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CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

NOR HIDAYATUL SOLEHAH BT SULAIMAN

ABSTRACT

This study thermodynamically analyzes a cascade refrigeration system that uses carbon dioxide and propane (R744-R290) or carbon dioxide and isobutane (R744-R600a) as an alternative to ammonia-carbon dioxide system (R717-R744). Parametric study will be used to determine the optimal condensing temperature of the cascade-condenser to maximize the COP given specific parameters for food processing application under fixed cooling capacity and condensing temperature. The parameters that have been varied are the evaporating temperature for low temperature circuit, the cascade condenser temperature difference, the low temperature circuit condensing temperature and degree of subcooling and superheating. The results show that COP increases with evaporating temperature (Tbe), but decreases as temperature difference in cascade condenser (ΔTcas) increases. Moreover, an increase of superheat reduced the COP of the system while increase of subcooling increased the COP of the system.

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ABBREVIATIONS AND NOMENCLATURES

Nomenclature			*Subscripts*
COP	Coefficient of Performance	c	Condenser
h	specific enthalpy (kJ/kg)	be	Evaporator
$\dot{m}$	mass flow rate (kg/s)	Cas	Cascade condenser
P	Pressure (MPa)	mc	Condensing temperature of cascade condenser
Q	Heat transfer rate (kW)	max	maximum
s	specific entropy (kJ/kg.K)	htc	high temperature circuit
T	Temperature (°C)	*Greek symbol*	
$\dot{W}$	Power (kW)	Δ	change

CHAPTER 1

INTRODUCTION

1.1 Project Background

The industrial refrigeration field, like any other field, is becoming more and more competitive especially in food processing, fisheries, meat packing or any other similar industry, where process temperature requirement is around –40°C to –50°C. The cooling foods history starts with chemicals addition such as sodium nitrate or potassium nitrate to water to cause the water to freeze or cool.

The science of refrigeration continues to develop. Synthetic refrigerants such as Chlorofluorocarbon (CFC) refrigerants have been phased out since 1996 in the developed countries as per Montreal Protocol and its amendments from United Nations Environment Programme (UNEP) [1]. The old refrigerant known to most people as "freon," a trade name, was replaced with HFC 134a, a new refrigerant less injurious to the ozone and still just as effective in keeping food cold. However, later this refrigerant has been identified to cause the depletion of ozone layer and has high global warming potential (GWP). With respect to global environmental protection, the use of natural refrigerant in refrigeration systems has been demonstrated to be a complete solution to permanent alternative fluorocarbon-based refrigerant.

One of the most promising alternatives to replace the synthetic refrigerants is carbon dioxide, CO_2. Carbon dioxide (R744) was a commonly used natural refrigerant in vapor compression refrigeration systems for over 130 years, but it has only been fully exploited during the last decade. Because of that, researchers are now interested in reviewing the state of the art of CO_2 and exploring its likely application. The reason why interest in carbon dioxide is increasing is the result of concern about greenhouse gas emissions, global warming, and the commitment to compliance with the Kyoto Protocol. Some of the characteristics of CO_2 make it a good alternative for use in large-scale refrigeration plants operated at low temperatures. The most obvious advantages of carbon dioxide are that it is nontoxic, incombustible and has no odor. However there is a trade off of using carbon dioxide. Carbon dioxide is known to have low triple-point temperature and high pressure. Its low critical point makes it a poor performer at typical conventional subcritical vapor compression cycle. Besides, due to the higher pressures found with the use of carbon dioxide it becomes necessary to implement certain principles to limit the pressure increases at higher temperatures.

A cascade refrigeration system is one method to provide this capability. With the use of cascade refrigeration system, there will be a significant decrease in the compressor work compared to a single compressor as in single vapor compression system. Besides that, refrigeration capacity also shows an improvement (Figure 1.1).In this process a separate refrigeration system uses a different refrigerant to condense the CO_2. The CO_2 is maintained at relatively low pressures by the low temperatures created by the separate refrigeration system. The second refrigerant will be at the upper side of the cascade cycle. With this type of system configuration, standard refrigeration components are used in the CO_2 refrigeration system. This cascade cycle of CO_2 can produce evaporator temperature close to -50°C. Such temperature is usable, for example, in the long term meat preservation section of supermarkets or hypermarkets. Normally, the recommended storage temperature for meat preservation is at the range of -40°C to -50°C. This range of temperature can slows down bacterial growth efficiently.

It is common to find ammonia (NH_3 or R-717) being used as the higher temperature refrigerant to condense the CO_2 used in the lower temperature refrigeration system. NH_3 has been served the industrial refrigeration industry particularly in the food and beverages industry for years. However, ammonia has a pungent smell; it is toxic and hazardous to human being, moderately flammable and has relatively large swept volume requirements at temperature under -35°C. As safety and large quantities of ammonia are of a concern, the author will discuss on the potential secondary refrigerants available in the market to replace ammonia in the high temperature circuit of CO_2 cascade system. As safety and large quantities of ammonia are of a concern, study of propane (R290) and isobutane (R-600a) will be done to evaluate the thermodynamic performance of the two-stage cascade refrigeration systems using these refrigerants at high temperature circuit and CO_2 at low temperature circuit [2].

In the design phase of a CO_2/secondary conventional refrigerant cascade refrigeration system, an important issue is the means of determining the optimal condensing temperature of a cascade condenser (T_{mc}) and optimal COP under particular design conditions, such as evaporating temperature (T_{be}), effect of sub-cooling (ΔT_{sc}) and superheating (ΔT_{sh}), the temperature difference between the high and low circuits in cascade-condenser (ΔT_{cas}) and its corresponding maximum COP (COP_{max}).

Hence, the main aim of the study is to conduct a parametric analysis to determine the optimum condensing temperature of cascade condenser (T_{mc}) in the low temperature circuit, to give the system maximum COP (COP_{max}) under various design parameter such as degree of subcooling (ΔT_{sc}) and superheating (ΔT_{sh}), evaporating temperature (T_{be}) and temperature difference in the cascade condenser of the system (ΔT_{cas}). Besides that, the optimum COP and ΔT_{cas} will be discovered by conducting parametric analysis on the effect of ΔT_{cas} on the system COP and resulted surface area of cascade condenser.

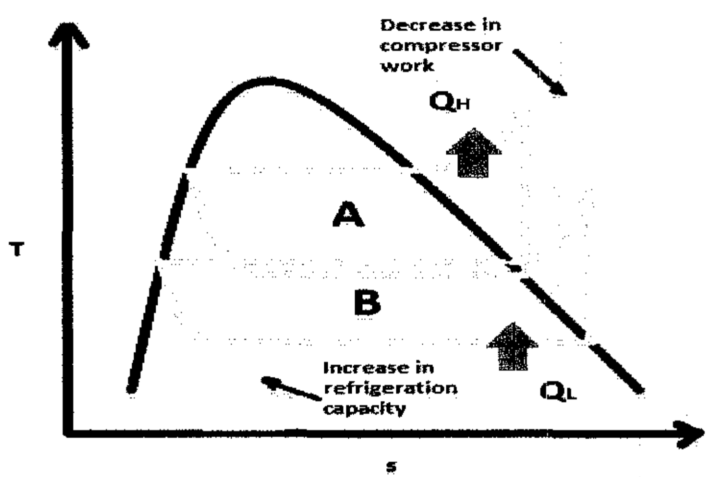

Figure 1.1: The Temperature-entropy diagram of cascade system [Thermodynamic an Engineering Approach, Cengel Boles]

1.2 Problem Statement

1) Ammonia has been identified as a highly toxic substance and can be very poisonous in high concentration. If there is any unexpected leakage from the refrigeration system, it can eventually cause fatal to human being. In August 2009, six men died from inhaling ammonia that leaked from a faulty refrigeration system at a jetty in Kampung Bagan Pasir, Malaysia [3].

2) Ammonia is not compatible with copper. It is very corrosive to copper and can result in a leakage of a system. Therefore, it cannot be used in any system with copper parts. Besides that, ammonia also is considered a strong oxidizer. Thus, safety measures should be taken to separate ammonia and ammonia products from incompatible materials, such as copper, brass, bronze, galvanized steel, tin, or zinc

3) Ammonia should be separated from oxidizers, combustible materials, heat, sparks, and open flame. As a liquefied gas, ammonia is flammable. Sources of ignition usually include smoking or open flames.

4) At room temperature, ammonia is a colorless, flammable gas with a pungent, suffocating odor. It becomes a clear, colorless liquid under increased pressure. Ammonia dissolves in water to form ammonium hydroxide, a corrosive solution.

1.3 Objective

1) To find the best alternative refrigerant to replace ammonia (R-717) as a refrigerant at the upper cycle in the cascade refrigeration system. The alternative natural refrigerants that will be studied are Propane (R290) and Isobutane (R600a). The cycle is designed to carry the purpose of long term meat preservation at a very low temperature.

2) Employs thermodynamic energy or parametric analysis to determine the optimal condensing temperature of the cascade-condenser (T_{mc}) in a low-temperature

CO2/second refrigerant cascade refrigeration system to give the system maximum COP under various values of the design parameters, such as degree of subcooling (ΔT_{sc}) and superheating (ΔT_{sh}), evaporating temperature (T_{be}) and temperature difference in the cascade condenser of the system (ΔT_{cas}).

3) To find the optimal COP and optimal temperature difference of cascade condenser (ΔT_{cas}) of the cascade refrigeration system by considering the tradeoff between COP and surface area of cascade condenser.

1.4 Scope of Study

1.4.1 Natural Refrigerant

After the Montreal Protocol, there was tremendous effort within the refrigeration and air conditioning industry to find proper replacement for CFCs. In this respect, the thermodynamic aspects of replacement refrigerants, in particular, the consequences for system operating efficiencies and the desired operating temperatures and pressures for conventional refrigeration equipment, are being studied.

When concerns about the depletion of the stratospheric ozone layer and global warming impact to the ozone layer emerged, scientists and researchers start to reconsider natural and green refrigerants such as carbon dioxide (CO_2), hydrocarbons, and ammonia as these substances have negligible direct global-warming impact and ozone-depletion potential.

Ammonia, hydrocarbons, and carbon dioxide have a broader range of application, and are used in much more conventional systems. Despite a generally excellent safety record there is a strict limit on the allowable charge of hydrocarbon systems, which makes them unsuitable for use in industry systems unless relevant safety standards can be applied [4]. In many ways ammonia is ideal for large industrial systems where its mild flammability, toxicity, pungent smell and low threshold limit value do not present problems. It is however, unsuited to the domestic, automotive and small commercial refrigeration, food processing and heat pump systems. This leaves carbon dioxide as the only natural refrigerant to find favor across the broad spectrum of industrial refrigeration and air conditioning systems.

1.4.2 Supermarket Refrigeration

Supermarket can be defined as a self grocery store in a larger scale with wider variety of product selections by the customers. Frozen foods like meat, poultry and fish are supposed to be kept in the cabinets in different certain cold temperatures [5]. Considering the basic refrigeration cycle, in supermarket applications, the difference between evaporating and condensing temperature is very large. Systems are usually operating at a constant condensing temperature of about 40°C and the evaporating temperature of about-40°C to -50°C for the frozen products. Therefore, cascade refrigeration system solutions become favorable and they are well adaptable in the food preservation industry.

1.4.3 Cascade Refrigeration System Description

Figure 1.2 schematically depicts a cascade refrigeration system. A two-stage cascade system employs two vapor compression units working separately with different refrigerants – the high temperature circuit and low temperature circuit. They are interconnected in such a way that the evaporator of one system is used to serve as condenser of the lower temperature system [6]. This type of heat exchanger is referred as cascade condenser in the system. CO_2 is the refrigerant of the low temperature circuit while R290 or R600a is the refrigerant of the high temperature circuit. Figure 1 indicates that the condenser in this cascade system rejects heat Q_H at condensing temperature of T_c to the ambient air T_o, approximately 30°C to 35°C. Meanwhile, the evaporator of this cascade system absorbs a refrigerated load QL from the cold refrigerated space at T_{be}. T_{mc} and T_{me} correspond to condensing and evaporating temperature of the cascade condenser. Temperature difference between the cascade condenser and evaporator is denoted by ΔT_{cas}. The refrigeration capacity Q_L of the studied system is 350 kW (for 100 tons of refrigeration) at T_c = 45°C. To evaluate the performance of cascade refrigeration system, T_c and Q_L are kept constant while varying other system parameters such as T_{be}, ΔT_{cas}, ΔT_{sc} and ΔT_{sh}.

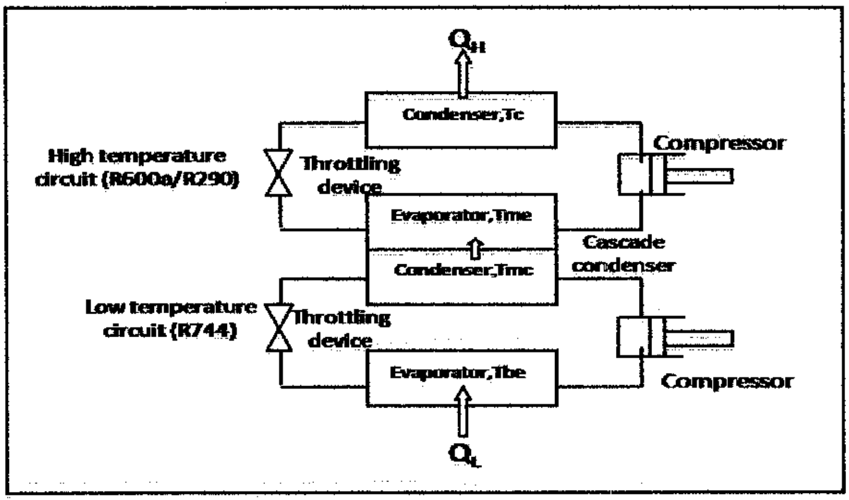

Figure 1.2: Schematic diagram of a R744/R290 or R600a cascade refrigeration system [1]

CHAPTER 2

LITERATURE REVIEW

2.1 Properties of CO_2, Ammonia, Propane and Isobutane

2.1.1 Carbon Dioxide (CO_2 or R744)

As the most typical natural refrigerant, CO_2 has a great safety features since it is non-flammable, non-toxic, and non-explosive. It has an excellent availability thus it is cheap while it does not participate in ozone layer depletion. However, increasing amount of CO_2 as a major product for fossil fuel burning in the globe is supposed to be a main participant to global warming. Meanwhile, its application in refrigeration would not contribute to global warming. It has negligible Global Warming Potential (GWP) and even zero if it is drawn from the waste product of industrial process. In addition, CO_2 has fairly good compatibility with wide range of construction materials and oils being used in refrigeration technology, such as metal, plastic and rubber [7].

Moreover, CO_2 have good thermodynamic properties. Heat transfer to CO_2 is often characterized as being superior to other refrigerants. High vapor density, low surface tension by one order of magnitude, and low vapor viscosity considerably influence the convective and nucleate boiling characteristic of CO_2. This result in heat transfer coefficient of CO_2 that is greater than those of conventional refrigerant by 2-3 times at the same saturation temperature while its two phase pressure drops are significantly smaller. The low pressure drop and the low volume flow rate, due to the high CO_2 vapor density, will also contribute to minimizing the energy consumption of the pump in the indirect circuit which will give CO_2 a major advantage compared to brine based system.

The high operating pressures of CO_2 about five times the typical system operating pressure also provides a potential opportunity for reduction in system size and weight. CO_2 high volumetric capacity due to its high working pressure enabling small equipment components and diameter lines to be used. This has favored CO_2 in automotive air conditioning and cascade refrigeration plants applications where limited space is available.

Its only technical disadvantage is the high critical pressure (73.84 bar) and the low critical temperature (31.06°C). The pressure is much higher than that of the other natural and synthetic refrigerants. This means for CO_2 cycles, newly developed components must be redesigned [8]. Besides that, CO_2 low critical point makes it a poor performer at typical commercial operating conditions. For that purpose, other cycle shall be used instead of conventional vapor compression cycle which can enhance the efficiency of CO_2 refrigeration system. Cascade refrigeration cycle provides a solution where CO_2 can be used at the low temperature stage and another refrigerant can be used at the high temperature stage.

The fact that CO_2 has no smell is a disadvantage while high concentration of the CO_2 in case of leakage may result a serious problem. So, its detection using proper detection devices is very crucial in order to ensure proper handling or measures can be taken in a case of CO_2 leakages. However, the slow leakage through some pore will be slow and proper ventilation can take care of it.

2.1.2 Ammonia (NH_3 or R717)[9]

Among natural refrigerants, R717 holds one of the first places as an alternative to R22 and R502. Production of ammonia all over the world reaches 120 million tons, and only small portion of it (up to 5%) is used in the refrigeration equipment.

Ammonia does not deplete ozone layer (ODP = 0) and does not directly contribute to increase of greenhouse effect (GWP = 0). However, ammonia gas has sharp rank smell which is harmful for the human body. Tolerance concentration in the air is 0.02 mg/dm^3, which corresponds to its inclusion volume fraction 0.0028%. Ammonia is explosive whenever exposed to open fire with ignition temperature with air of 651°C.

Ammonia vapors are lighters than that the air. In fact, it is well soluble in water (one unit of water can resolve 700 units of ammonia which excludes moisture freezing in the system). However, ammonia hardly resolves mineral oils. It does not affect ferrous metals, aluminum and phosphorous bronze, but at the presence of moisture it destroys non-ferrous metals such as zinc, copper and its alloys. To ensure the ammonia not react with non-ferrous metals, moisture mass proportion in ammonia should not exceed 0.2%.

One of the disadvantages of ammonia is higher value of adiabatic line compared to R22 and R12, which causes considerable increase of discharge temperature. In connection with this, they present strict requirements to thermal stability of refrigeration oils used in combination with ammonia during a long period of time at operation of a refrigeration facility. Condenser should have developed heat-transfer area which results in increase of its steel intensity.

In addition, ammonia has caused difficulties with refrigeration equipment production due to its high activity towards copper and copper alloys. Besides that, due to ammonia high toxicity and combustibility, welded connections are thoroughly controlled. High conductivity of R717 also had caused creation of half-hermetic and hermetic compressors are impeded.

2.1.3 Propane (R290)

Propane has ozone depletion potential (ODP) of zero, and global warming potential (GWP) of 3. While using this refrigerant, there is no problem with selection of structural materials for the parts of compressor, condenser and evaporator as it is inert and easily resolved in mineral oils.

Propane is non-toxic, non-caustic and will not create an environmental hazard if released as a liquid or vapor into water or soil. If spilled in large quantity, the only environmental damage that may occur is freezing any organism or plant life in the immediate area. There are no long term effects following a propane spill even if the quantities are excessively large. The only damage and potential danger exists if the vapor is ignited following a spill. And even then, there are no long term effects of ignited propane that can be damaging to the environment. Propane liquid and vapor are environmentally sound and friendly in their unused states (prior to combustion) if released. Among the advantages of propane are [10]:

- Propane is not damaging to freshwater or saltwater ecosystems, underwater plant or marine life.

- Propane is not harmful to soil if spilled on the ground. Propane will not cause harm to drinking water supplies.
- Propane vapor will not cause air pollution. Propane vapor is not considered air pollution.
- Propane vapor is not harmful if accidentally inhaled by birds, animals or people.
- Propane will only cause bodily harm if liquid propane comes in contact with skin (boiling point -44°F).

However, propane main disadvantage as a refrigerant is it is fire risky and can cause explosion when exposed to open fire. In industrial refrigeration facilities, propane has been used for many years already. During recent years, more and more often it is suggested to use propane in refrigeration transport facilities. In Germany, since 1994 there were manufactured more than 1000 domestic refrigerators on propane, isobutane and their blends. Similar refrigerators are being manufactured in China, Brazil, Argentina, India, Turkey and Chile.

Propane also can be immediately charged into the system where there was ozone-depleting refrigerant before. In the USA it is prohibited to use hydrocarbons in domestic refrigerators as American Environment Protection Agency is foreseeing up to 30 000 fires per year in case of their use. However, in New Zealand hydrocarbons are allowed to use in commercial refrigeration equipment. While locating commercial refrigeration equipment operating on propane in public places, it is necessary to follow safety rules. In case of exceeding of assigned charging norms (more than 2.5 kg of R290), refrigeration equipment should be installed in a separate, specially equipped place which increases capital outlays.

Propane is used also in heat pumps. In Lillehammer (Norway) a heat pump is operating on propane with power of 45 kWt with half-hermetic compressor and plate heat-exchangers. In the system of a heat pump propane mass is a little more than 1 kg therefore, the equipment is placed in a separate building.

2.1.4 Isobutane (R600a) [11]

Refrigerant R600a or isobutane, is a possible replacement for other refrigerants, which have high impact on the environment especially in domestic refrigerators. It has zero ozone depletion potential (ODP) and a negligible global warming potential (GWP). Furthermore it is a substance which is a part of petrol gases from natural sources. The refrigerant R600a has been in use in the past, in refrigerators up to the 1940's, and has now again found a wide use in domestic refrigerators and freezers in Europe, especially in Germany, where more than 90% of refrigerators are manufactured using R600a as refrigerant. Because of the availability of isobutane all over the world it has been discussed widely for CFC replacement.

Isobutane (R600a) is a possible refrigerant for this application, with good energy efficiency, but with a very different characteristic in several points, which implies the design to be made or adopted for this refrigerant. Special care has to be taken to the flammability of isobutane. The properties of R 600a differ from other refrigerants commonly used in household applications, as shown in Table 2.1. This leads to a different design of details in many cases.

Table 2.1: Refrigerant data comparison of R600a, R134a and R12

Refrigerant	**R600a**	**R134a**	**R12**
Name	Isobutane	1,1,1,2-Tetra-fluoro-ethane	Dichloro-di-fluoro-methane
Formula	$(CH_3)_3CH$	CF_3-CH_2F	CF_2Cl_2
Critical temperature (°C)	135	101	112
Molecular Weight (kg/mol)	58.1	102	120.9
Normal boiling point (°C)	-11.6	-26.5	-29.8

Pressure at -25°C (bar)	0.58	1.07	1.24
Liquid density at -25°C (kg/l)	0.60	1.37	1.47
Vapor density at -25/+32°C (kg/m^3)	1.3	4.4	6.0
Volumetric capacity at -25/55/32°C (kJ/m^3)	373	658	727
Enthalpy of vaporization at -25°C (kJ/kg)	376	216	163
Pressure at 20°C (bar)	3.0	5.7	5.7

Refrigerant R600a is mostly used with mineral compressor oils, so material compatibility is almost identical to R12 situation from oil side. Use of alkyl benzenes is also possible. R600a is chemically inactive in refrigeration circuits, so no specific problems should occur in the system. Solubility with mineral oil is at least as good as was with R 12. Direct material compatibility is less problematic. On some rubbers, plastics and especially chlorinated plastics however, problems have been observed, but these materials are normally not present in refrigerator systems.

The main disadvantage discussed in connection with R600a use is the risk based in its flammability. This leads to necessity for very careful handling and safety precautions. Because of the flammability of isobutane in a wide concentration range safety precautions are necessary, on the appliance itself and in the manufacturing factory. The risk assessments behind these two situations are quite different. Main common starting point is that accidents need to have two essential preconditions. One is the flammable mixture of gas and air and the other is the ignition source of a certain energy level or temperature. The minimum ignition temperature for isobutene is 460°C. These two have to be present together for combustions, so avoidance of this combination has to be proven.

2.2 Thermodynamic Analysis of CO_2-NH_3 Cascade Refrigeration System

Two-stage cascade refrigeration systems as shown in Figure 2.1 are suitable for industrial applications, especially in the supermarket refrigeration industry, where the evaporating temperature of frozen-food cabinets ranges from -30°C to -50°C [12]. In these systems, two single-stage units are thermally coupled through cascade condensers. The high-temperature circuit of a cascade refrigeration system can normally be charged with ammonia (R717) whereas carbon dioxide (R744) may be used in the low-temperature circuit of the refrigeration system. Ammonia is a naturally available refrigerant with few application constraints such as toxicity and flammability [13]. However, the risk associated with toxic and flammable refrigerants can be highly minimized by confining the high-temperature circuit to the plant room area or the rooftop of a supermarket. Carbon dioxide has a disadvantage of reaching a high pressure (7.4 MPa) at 31°C, leading to redesign of pipes and fittings [14]. A traditional direct expansion low-temperature refrigeration system [15] involves a large pressure lift between evaporating and condensing temperatures resulting in an increase in the compression ratio and reduction of the volumetric efficiency of the compressors.

Furthermore, environmental issues of global warming are forcing the supermarket owners to adopt alternative technologies offering lower refrigerant charge and reduced environmental impact. Therefore, natural refrigerants are increasingly receiving attention for their use in supermarket refrigeration systems. Especially, the use of R744 in the low-temperature circuit of cascade systems is recently becoming commercially more attractive alternative. A small concern with cascade refrigeration systems is the initial installation cost being 10% higher than the traditional direct expansion systems [16]. But this cost can be negated with less refrigerant charge requirements and the environmental advantage of the cascade system due to less direct emissions.

An improvement, which can also be observed in cascade systems (with the same fluid in both circuits) is the reduced amount of superheat in the discharge temperature of the high-temperature circuit that results in a reduced capacity of the high-temperature condenser and an increased refrigeration effect [17]. High-temperature circuit condenser, cascade condenser and evaporator losses can also be reduced if the sizes of the heat exchangers are properly optimized. Several types of cascade condensers such as plate, shell-and-plate or shell-and-tube heat exchangers can be employed for cascade systems to couple the low- and high-temperature circuits [18].

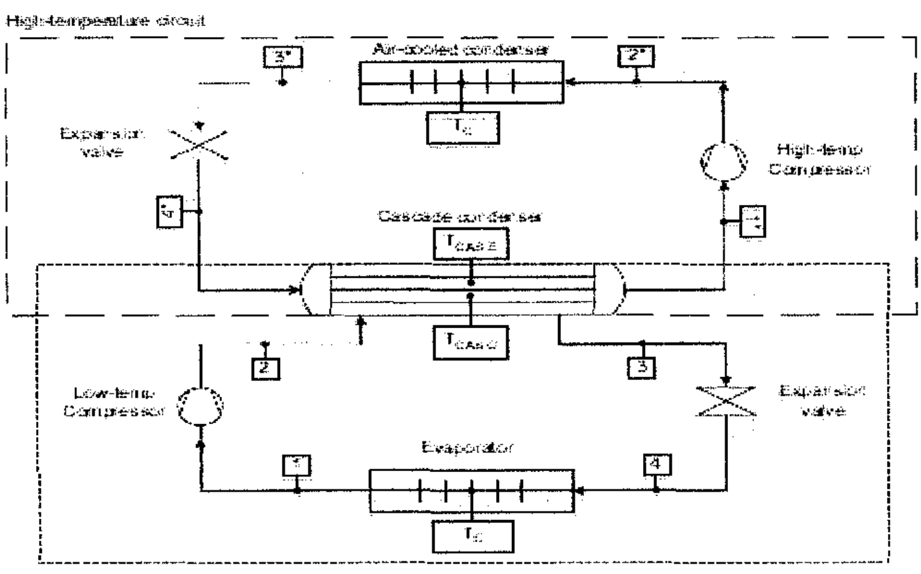

Figure 2.1: Schematic of a two stage cascade refrigeration system [12]

2.3 Conclusion on the Literature Review

Several researchers have evaluated the thermodynamic performance of the two-stage cascade refrigeration systems. Lee et al. (2006)[19] analyzed a carbon dioxide–ammonia (R744– R717) cascade system thermodynamically to determine the optimum condensing temperature of R744 in the low-temperature circuit. Bansal and Jain (2007)[20] evaluated the optimum cascade condensing temperatures of R744 for different refrigerants such as R717, R290, R1270 and R404A, which are in the high temperature circuits of a cascade system. However, the aforementioned studies lack the main design parameters such as degree of subcooling and superheat in order to optimize the condensing temperature of R744 for maximum COP of a two-stage carbon dioxide–second refrigerant cascade refrigeration system. Ammonia (R717) has higher heat transfer than R600a and R290, but its vapor pressure, corrosion and toxicity are higher. The toxicity is particularly a disadvantage in domestic application especially in food preserving industry. Hydrocarbon refrigerants such as R600a and R290 have environmental advantages and are safe in small quantities. These advantages make R600a and R290 desirable as the secondary refrigerants in the applications where toxicity is important. Table 2 shows the comparison on the environmental impacts between R744, R717, R600a and R290. Therefore, the main aim of the current research is to conduct a thermodynamic analysis on a carbon dioxide–isobutane (R744–R600a) and carbon dioxide–propane (R744–R290) cascade refrigeration system to optimize the condensing temperature of R744, which can give the maximum COP of the system and the optimum COP the system can have with consideration to the cost.

Table 2.2: Comparison on the environmental impacts between R744, R717, R600a and R290

Refrigerant	R744	R717	R600a	R290
Class	Natural	Natural	Hydrocarbon	Hydrocarbon
Atmospheric lifetime (years)	~100	Few hours	<1	<1
Ozone Depletion Potential (ODP)	0	0	0	0
Global Warming Potential (GWP)	1	0	8	8

CHAPTER 3

METHODOLOGY

3.1 Project Flow

This project presents a parametric analysis to evaluate the thermodynamic performance of the two-stage cascade refrigeration systems using propane (R290) or isobutane (R600a) at the upper stage and carbon dioxide (R744) at lower stage. In completing this project, some tasks and methodology have been allocated. The project flow is shown as below:

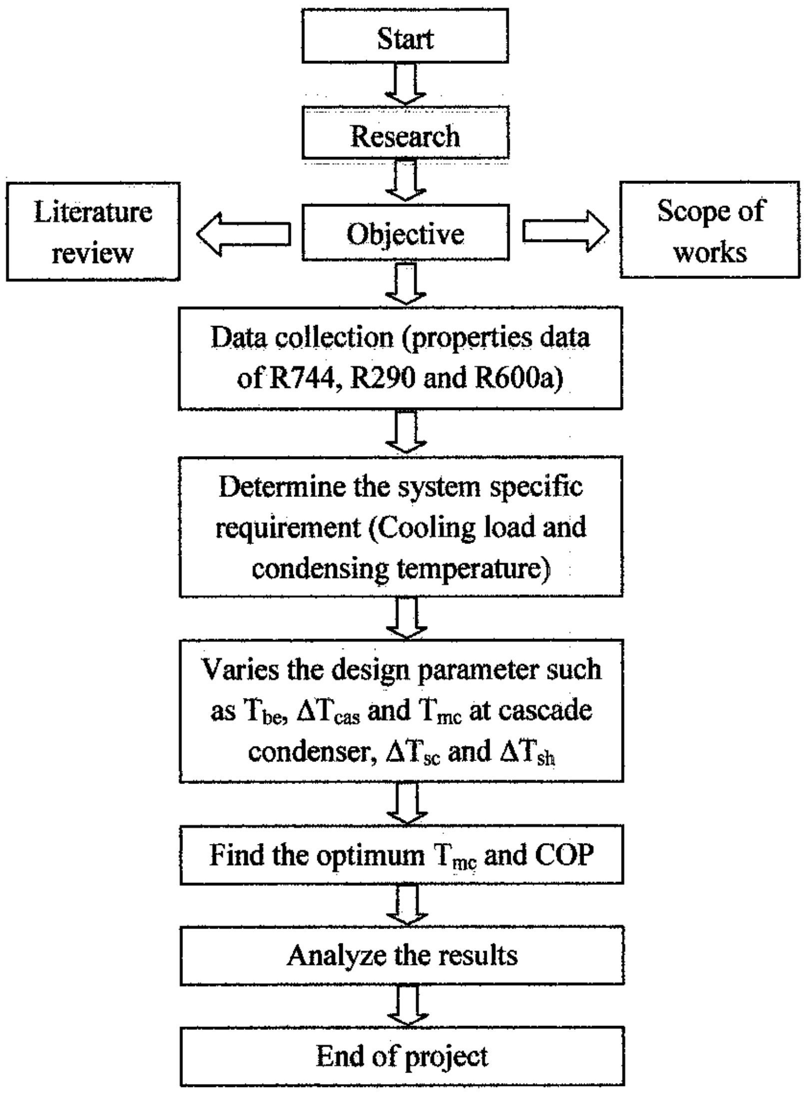

Figure 3.1: Process flow chart

3.2 Project Activities

3.2.1 Data Collection

The Reference Fluid Thermodynamic and Transport Properties Database (REFPROP): Version 9.0 (REFPROP) "database" is actually a program and does not contain any experimental information, aside from the critical and triple points of the pure fluids. The program uses equations for the thermodynamic and transport properties to calculate the state points of the fluid or mixture. These equations are the most accurate equations available worldwide. The REFPROP database is published by the NIST. With the help of REFPROP database, the properties of R744, R600a and R290 are collected to enable the calculation of Coefficient of Performance (COP) for each refrigerant.

3.2.2 System specific requirement

A parametric study for a typical refrigeration system for food processing has been done for which the basic requirements are:

- Process temperature: – 40°C
- Evaporating temperature: – 45 °C to – 50°C (for low-stage or low temperature cycle)
- Condensing temperature: 45°C
- Capacity of plant required: 100 TR (350 kW)

To simplify the thermodynamic analysis the following assumptions are made:

1) All components are assumed to be in steady-state and steady flow. The changes in potential and kinetic energy are negligible.
2) The high and low temperature circuit compressors are isentropic.
3) The heat loss and pressure drops in the piping connecting the components are negligible.
4) All throttling devices are iso-enthalpic
5) Heat transfer in cascade heat exchanger with the ambient is negligible
6) Heat transfer process in heat exchangers is isobaric.

3.2.3 Variation in design parameters

Figure 3.2 shows the P-h diagram of the cascade system with the subcooling and superheating. The parameters that have been varied are furnished below [21]:

i. The evaporating temperature for low temperature circuit, T_{be} is varied from -40°C to -50°C
ii. The cascade condenser temperature difference, ΔT_{cas} is varied from 2°C to 5°C.
iii. Low temperature circuit condensing temperature, T_{mc} is varied from -20°C to 5°C.
iv. Degree of subcooling, ΔT_{sc} and superheating, ΔT_{sh} is varied from 0°C to 8°C.

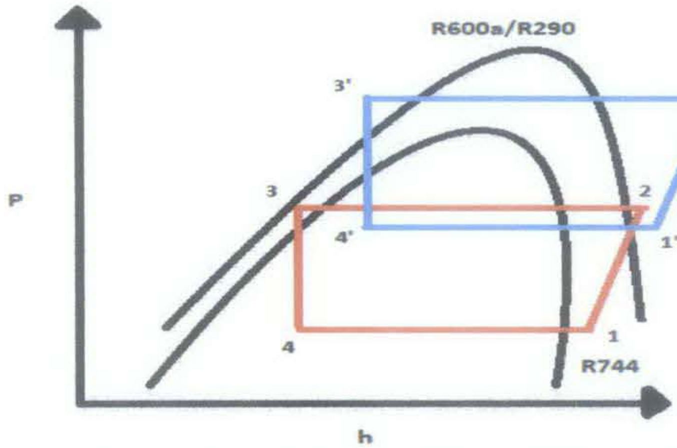

Figure 3.2: P-h diagram for cascade system with subcooling and superheating [2]

3.2.4 Find the optimum T_{mc} and COP

All refrigerant thermophysical properties were obtained from the REFPROP 9 for several state points as shown in Figure 5, and are directly calculated for the system analysis with Microsoft Excel. Based on the above assumptions, balance equations are applied to find the mass flow rate of each cycle, the work input to the compressor, the heat transfer rates of the condenser and the cascade-condenser and the COP as follows:

If the flow rate of CO_2 in low stage is $\dot{m}_{CO2}$ and the Q_L in low temperature circuit is 350 kW, then

$$\dot{m}_{co2} = \frac{350}{h_1 - h_3} \text{ (kg/s)}$$

The capacity of the cascade condenser, Q_{cas}

$$Q_{cas} = 350 \frac{(h_2 - h_3)}{(h_1 - h_4)} \text{ (kW)}$$

The flow rate of R290 or R600a in high temperature circuit is

$$\dot{m}_{htc} = \frac{Q_{cas}}{h_1' - h_3'} \text{ (kg/s)}$$

The total power of the compressor, $\dot{W}$ is

$$\dot{W} = \dot{m}_{co2}(h_2 - h_1) + \dot{m}_{htc}(h_2' - h_1')$$

Coefficient of Performance, COP of the cycle is

$$COP = \frac{350}{\dot{W}}$$

The surface area of cascade condenser is obtained by this equation;

$$Q_{cas} = U_o A \Delta T_{LMTD}$$

Where U_o = Overall heat transfer coefficient (Assume 1200 $W/m^2.K$)

A = Surface area of cascade condenser

ΔT_{LMTD} = Log mean temperature difference

$$= \frac{T4' - T3}{T1' - T2}$$

CHAPTER 4
RESULT AND DISCUSSION

4.1 Selection of high temperature circuit refrigerant

Comparison of system performance between R290 and R600a was investigated to select an appropriate refrigerant to be used at high temperature circuit of the cascade cycle with R744 at low temperature circuit. Figure 6 shows the variation of COP at different cascade condenser temperatures T_{mc}, for R290 and R600a. Other operating parameters such as condensing temperature (T_c), evaporating temperature (T_{be}), temperature difference in cascade condenser (ΔT_{cas}), degree of superheat (ΔT_{sh}) and subcooling (ΔT_{sc}) were kept constant. From Figure 4.1, the COP of the cycle changes with the condensing temperature of cascade condenser and there is a maximum value. This is because, when T_{mc} falls, the COP of low temperature circuit rises, but the COP of high temperature circuit decreases, and vice versa. The maximum COP (COP_{max}) of the system for the given condition is highest at T_{mc} = -5°C for both refrigerants. In general, the COP_{max} of R600a is higher than R290 which makes R600a more appropriate for use with R744 compared to R290. However, this is only valid when effects of superheating and subcooling are not considered.

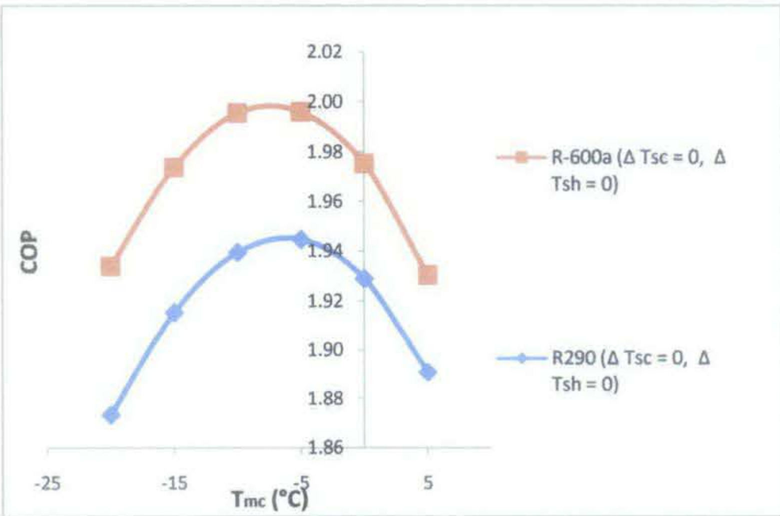

Figure 4.1: Variation of system performance at different cascade condensing temperatures (Tc = 45°C, Tbe = -40°C, ΔTcas = 2°C, ΔTsh= 0°C, ΔTsc = 0°C)

4.2 Effect of degree of superheating and subcooling

To evaluate the effect of superheating and subcooling on the system performance, the effect of having different and the same degree of subcooling and superheating in both high and low temperature circuit of cascade system was individually and jointly analyzed while keeping other parameters constant. These parameters are T_c = 45°C, T_{be} = -40°C, ΔT_{cas} = 2°C, T_{mc} = -5°C.

4.2.1 Effect of degree of subcooling

The parametric study for subcooling effect is divided into three main parts;

a) *Subcooling in R744 cycle only*

   Degree of subcooling in low temperature circuit (R744) was varied from 0°C to 8°C. At the same time, degree of subcooling and superheating in high temperature circuit was kept constant at 0°C.

b) *Subcooling in R290/R600a cycle only*

   Degree of subcooling in high temperature circuit (R290/R600a) was varied from 0°C to 8°C. At the same time, the degree of subcooling and superheating in low temperature circuit was kept constant at 0°C.

c) *Subcooling in both R744 and R290/R600a cycles with the same degree of subcooling.*

   Degree of subcooling in both cycles was varied simultaneously from 0°C to 8°C while the superheat is kept at 0°C.

From Figures 4.2 and 4.3, conclusions can be drawn on the effect of having different and same degree of subcooling in both high and low temperature circuits.

a) *Subcooling in R744 cycle only*

   It was observed that COP of the system is increased but to a lower extent than recorded in the case of subcooling in both cycles and in R600a/R290 cycle only.

b) *Subcooling in R290/R600a cycle only*

   The COP is slightly higher than subcooling in R744 cycle only. However it is at much smaller amount than recorded for subcooling in both cycles.

c) Subcooling in both R744 and R290/R600a cycles with same degree of subcooling.

Sub cooling in both cycles has resulted in an increase in COP of the system. In fact, it yields the highest COP of all. It increases the COP by up to 9% higher than the reference COP (2.00 for R600a-R744 system and 1.95 for R290-R744 system). Figures 4.4 and 4.5 illustrate the maximum COP of R600a-R744 and R290-R744 systems with variation in cascade condenser temperature (T_{mc}) and degree of subcooling in both cycles. It is observed that in general, the COP_{max} increases when the degree of subcooling is increased. In this case, COP is highest at ΔT_{sc} = 8°C and lowest at ΔT_{sc} = 0°C. The COP_{max} occurs at T_{mc} = -5°C for both systems.

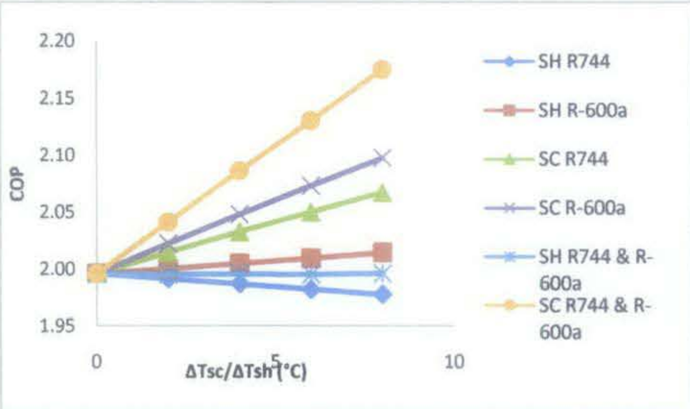

Figure 4.2: Effect of superheating and subcooling on R600a-R744 cascade system

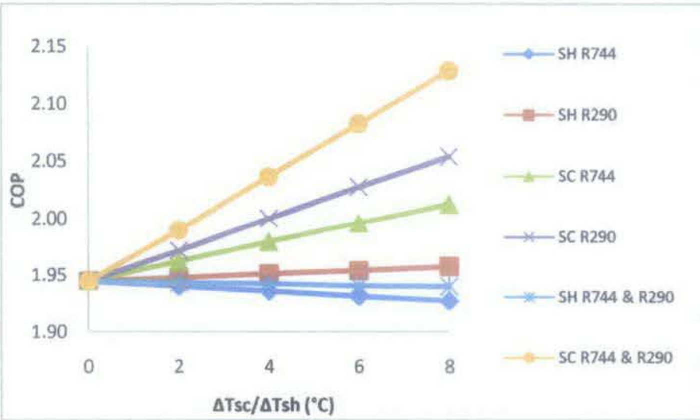

Figure 4.3: Effect of superheating and subcooling on R290-R744 cascade system

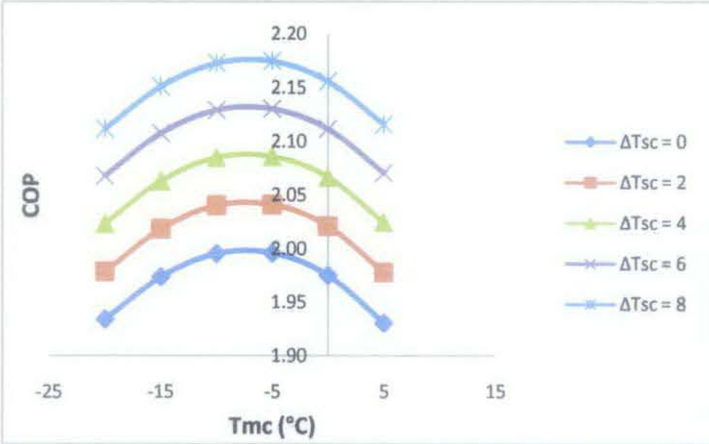

Figure 4.4: Effect of varying degree of subcooling on R600a-R744 system performance

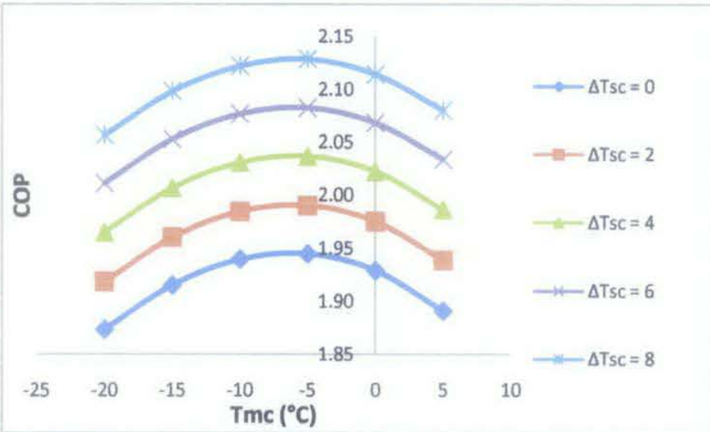

Figure 4.5: Effect of varying degree of subcooling on R290-R744 system performance

4.2.2 Effect of degree of superheating

The parametric study for superheating effect is divided into three main parts;

a) Superheating in R744 cycle only

Degree of superheating in low temperature circuit (R744) was varied from 0°C to 8°C. At the same time, degree of subcooling and superheating in high temperature circuit was kept constant at 0°C.

b) Superheating in R290/R600a cycle only

Degree of superheating in high temperature circuit (R290/R600a) was varied from 0°C to 8°C. At the same time, the degree of subcooling and superheating in low temperature circuit was kept constant at 0°C.

c) Superheating in both R744 and R290/R600a cycles with the same degree of subcooling.

Degree of superheating in both cycles was varied simultaneously from 0°C to 8°C while keeping the subcooling at 0°C.

The effect of superheating in both high and low temperature circuit can be made by referring to Figures 4.2 and 4.3.

a) Superheating in R744 cycle only

It yields the lowest COP among all by reducing the reference COP by about 0.9%. The decrease in COP is more dominant compared with superheat in both cycles. Moreover, COP is further decreased with increase in degree of superheat.

b) Superheating in R290/R600a cycle only

This resulted in a very small increase in reference COP as degree of superheat increased.

c) Superheating in both R744 and R290/R600a cycles with same degree of subcooling.

Superheat in both cycles decreased the COP of the system but showed a negligible difference with the reference COP (without superheating and subcooling). However the COP is slightly higher than in the case of superheat in R744 cycle only. Figures 4.6 and 4.7 depict the change of maximum COP of the system at difference values of superheat as the condensing temperature at cascade condenser is varied between -20°C to 5°C. For each value of superheat, the change in COP is very small with a nearly negligible effect. However, superheat has the opposite effect of subcooling on the COP_{max}. As degree of superheat increase, the COP is found to be decreased.

As conclusion, for specific operating conditions and assumption of 100% isentropic efficiency of compressors at both cycles, Figures 4.2 and 4.3 show that the COP_{max} of cascade system lies in the region confined by the graphs of subcooling in both cycles in upper limit and superheat in both cycles in lower limits for any such systems. Superheat at both cycles is chosen to be the lower limit as it has negligible deviation from the reference COP ($\Delta T_{sc} = \Delta T_{sh} = 0°C$).

Therefore, the same degree of subcooling and superheating in both cycles gives the average COP of the cascade refrigeration systems. Figure 4.8 shows the variation of system performance for the same degree of subcooling and superheat at different secondary refrigerants and values of superheat and subcooling.

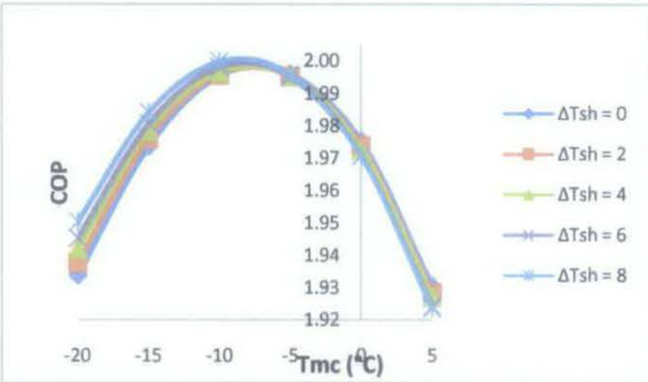

Figure 4.6: Effect of varying degree of superheating on R600a-R744 system performance

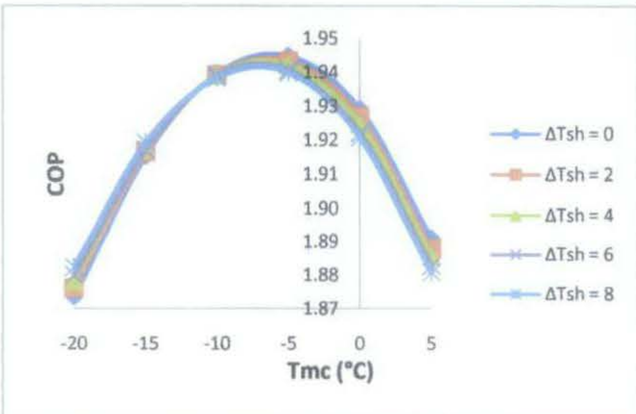

Figure 4.7: Effect of varying degree of superheating on R290-R744 system performance

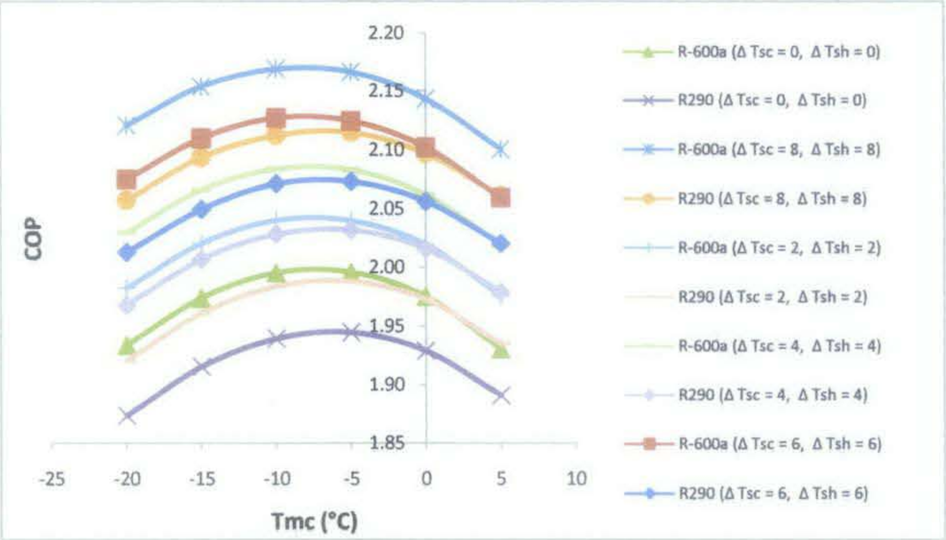

Figure 4.8: Variation in system performance for same degree of subcooling and superheating

4.3 Effect of T_{be}, ΔT_{cas} and T_{mc}

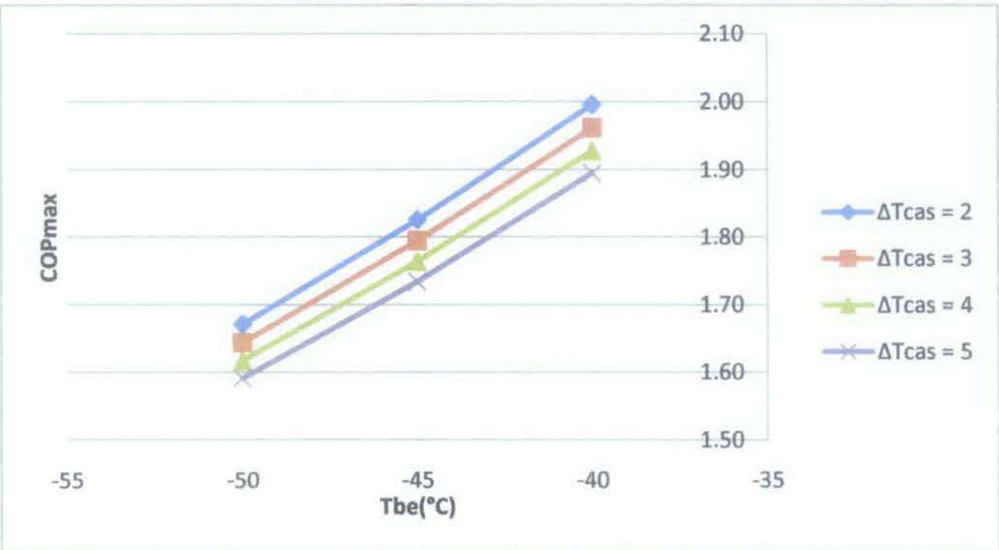

Figure 4.9: Effect of evaporating temperature (T_{be}) on R600a-R744 cascade system performance

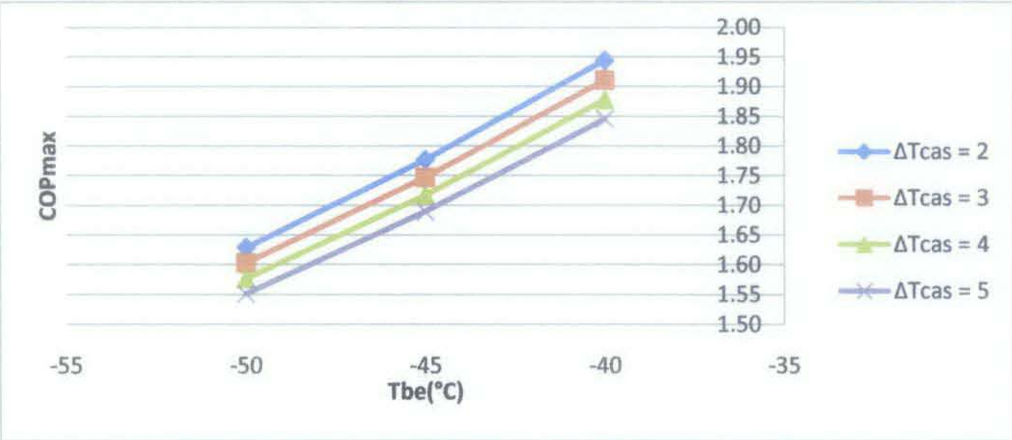

Figure 4.10: Effect of evaporating temperature (T_{be}) on R290-R744 cascade system performance

Figures 4.9 and 4.10 portray the variation of COP with the change in evaporating temperature (T_{be}) at different ΔT_{cas} for R600a-R744 and R290-R744 cascade cycles. The evaporating temperature (T_{be}) was varied from -40°C to -50°C while keeping the condensing temperature (T_c) at 45°C. The two figures show that increase T_{be} increases the COP_{max}. Figure 4.9 and 4.10 reveal a linear relationship. COP was found maximum at a system with $\Delta T_{cas} = 2$°C.

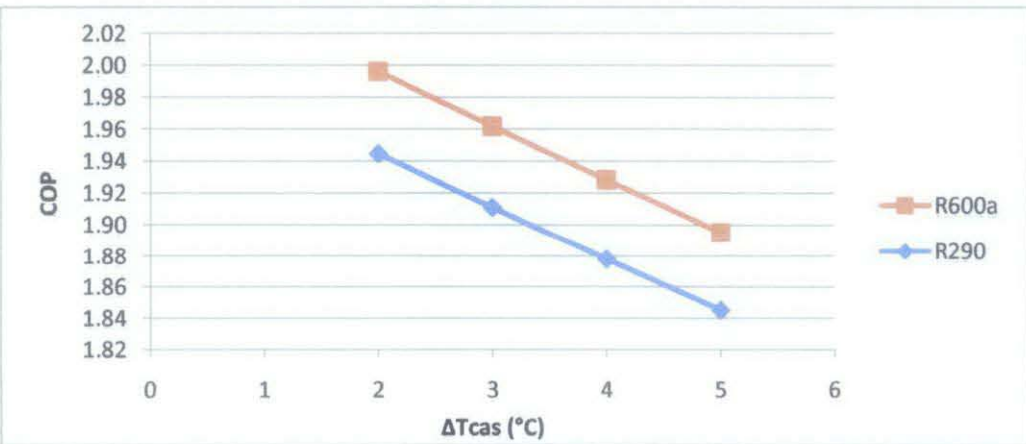

Figure 4.11: Effect of temperature difference of cascade condenser (ΔT_{cas}) on R600a-R744 and R290-R744 cascade system performance at T_{be} = -40°C

From Figure 4.11, the temperature difference in cascade condenser (ΔT_{cas}) was varied from 2°C to 5°C by holding T_c at 45°C and T_{be} at -40°C. This time, COP was found linearly decreasing with increase in ΔT_{cas}.COP is maximum at ΔT_{cas} = 2°C for both R600a-R744 and R290-R744 system.

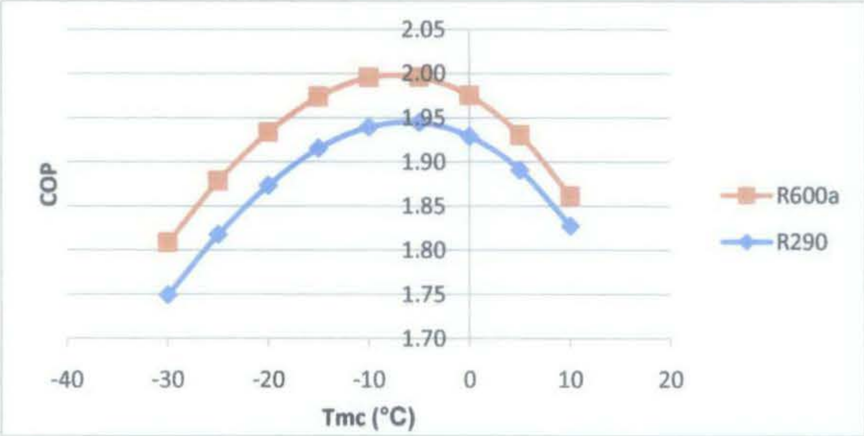

Figure 4.12: Effect of condensing temperature of cascade condenser (T_{mc}) on R600a-R744 and R290-R744 cascade system performance at T_{be} = -40°C

The effect of varying condensing temperature at cascade condenser (T_{mc}) can be seen by varying T_{mc} from -30°C to 10°C and fixing T_c at 45°C, T_{be} at -40°C and ΔT_{cas} at 2°C. From Figure 4.12, it shows that there was an optimum temperature at which COP of the system was the highest.

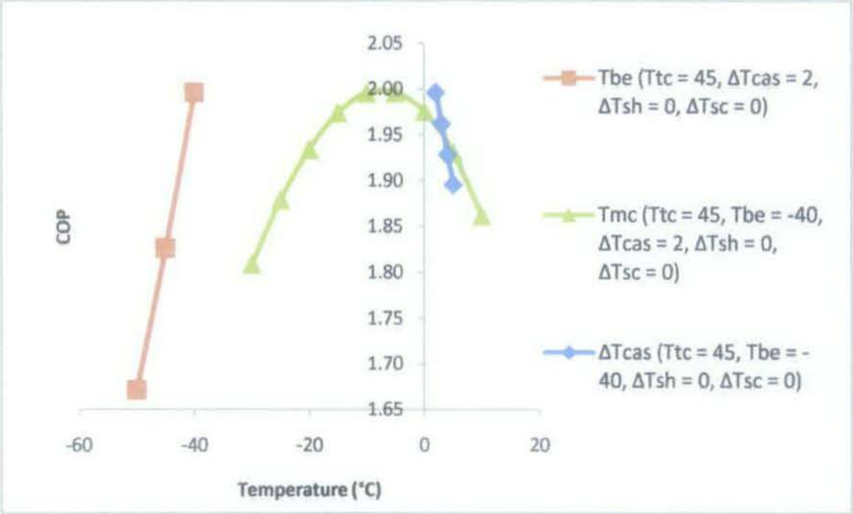

Figure 4.13: Effect of T_{be}, T_{mc} and ΔT_{cas} on R600a-R744 cascade system performance

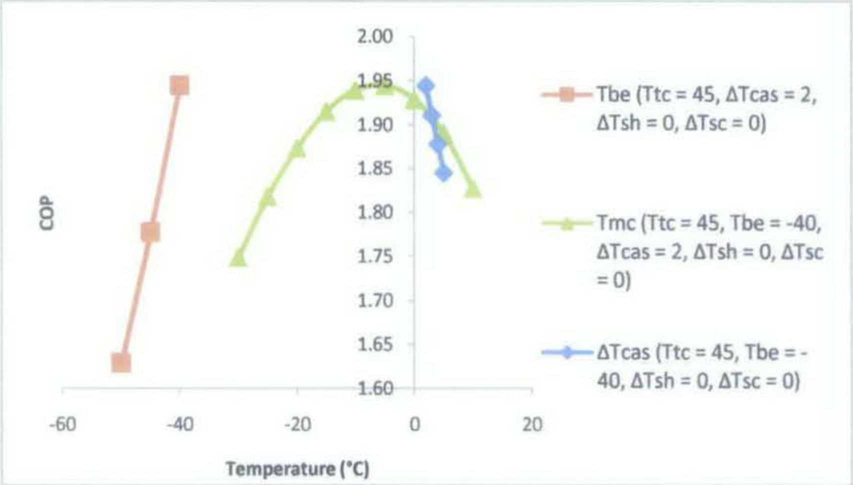

Figure 4.14: Effect of T_{be}, T_{mc} and ΔT_{cas} on R290-R744 cascade system performance

Figure 4.13 and 4.14 present the overall effect of the evaporating temperature (T_{be}), condensing temperature of cascade condenser (T_{mc}), and temperature difference in cascade condenser (ΔT_{cas}) with 0°C in superheating and subcooling for R600a-R744 and R290-R744 cycle. Table 4.1 and 4.2 present the calculated optimum condensing temperatures of the cascade condenser and the corresponding maximum COPs.

Table 4.1: The optimum condensing temperature of cascade condenser and its corresponding maximum COP under various design parameters for R600a-R744 cascade system

T_c (°C)	T_{be} (°C)	ΔT_{cas} (°C)	$T_{mc,opt}$ (°C)	COP_{max}	COP_{carnot}
45	-40	2	-5	1.996	2.742
		3	-5	1.962	2.742
		4	-5	1.928	2.742
		5	-5	1.895	2.742
	-45	2	-10	1.826	2.534
		3	-10	1.795	2.534
		4	-10	1.764	2.534
		5	-10	1.734	2.534
	-50	2	-10	1.671	2.348
		3	-10	1.644	2.348
		4	-10	1.617	2.348
		5	-10	1.591	2.348

Table 4.2: The optimum condensing temperature of cascade condenser and its corresponding maximum COP under various design parameters for R290-R744 cascade system

T_c (°C)	T_{be} (°C)	ΔT_{cas} (°C)	$T_{mc,opt}$ (°C)	COP_{max}	COP_{carnot}
45	-40	2	-5	1.945	2.742
		3	-5	1.911	2.742
		4	-5	1.878	2.742
		5	-5	1.845	2.742
	-45	2	-10	1.777	2.534
		3	-10	1.747	2.534
		4	-5	1.718	2.534
		5	-5	1.689	2.534
	-50	2	-10	1.629	2.348
		3	-10	1.603	2.348
		4	-10	1.577	2.348
		5	-10	1.551	2.348

4.4 Effect of ΔT_{cas} on optimum COP and surface area, A

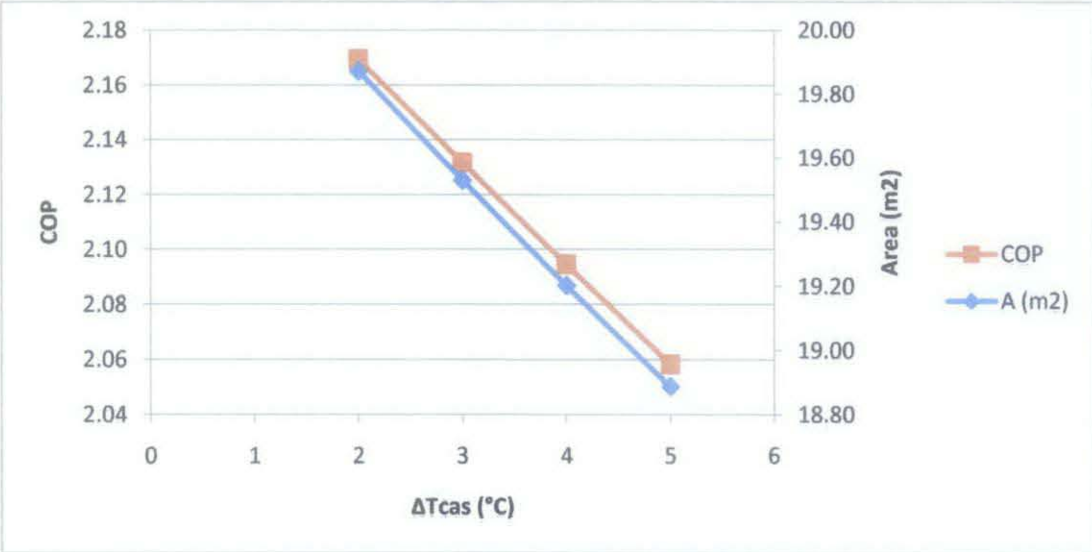

Figure 4.15: Effect of varying ΔT_{cas} on the COP and surface area of cascade condenser for R600a-R744 cascade system

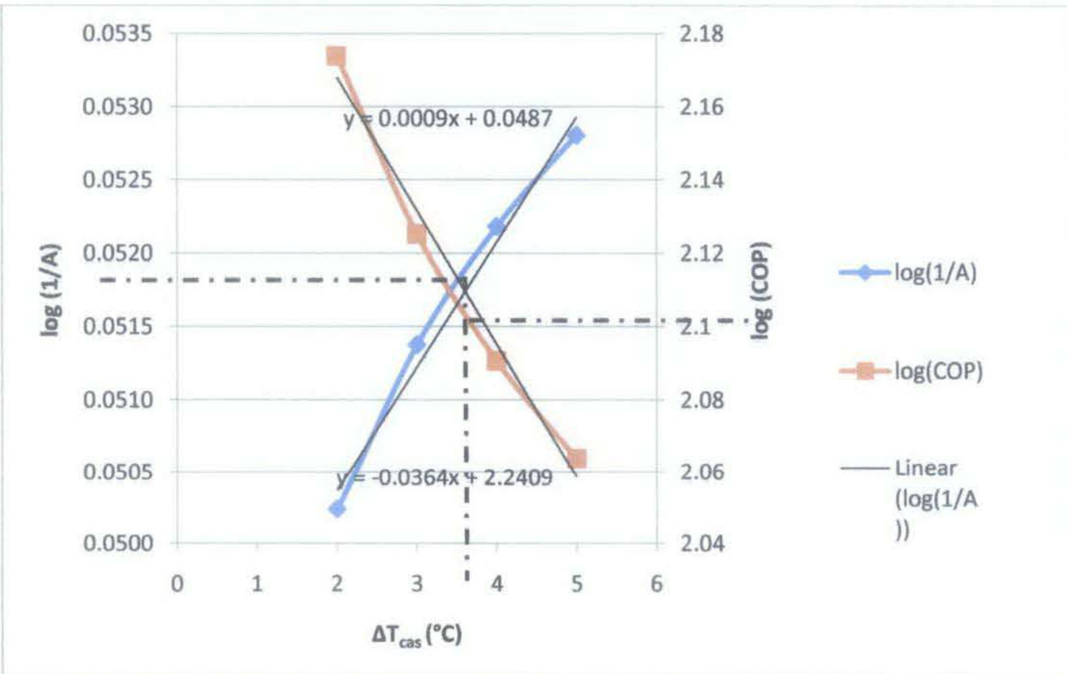

Figure 4.16: The optimum COP and surface area of cascade condenser for R600a-R744 cascade cycle

From Figure 4.15, it can be seen that when ΔT_{cas} is increased, the system performance (COP) is decreasing linearly. Same goes to the surface area of cascade condenser. This means that, it will have a tradeoff between having a high COP and high surface area of cascade condenser. High surface area of heat exchanger is not favorable to the fabricator as it will generate high cost. Therefore, there will be an optimum COP at ΔT_{cas} between 2°C to 5°C.

Figure 4.16 shows the relationship between ΔT_{cas}, logarithm of COP (log COP), and logarithm of reciprocal area (log 1/A). Log (COP) represent the system performance while log (1/A) characterize the reciprocal of cost. As ΔT_{cas} increase, COP of the system increase, however, at the same time, the cost of fabricating increase. Thus, the optimum COP was found at $\Delta T_{cas} = 3.7°C$.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

1. The replacement of refrigerants in the refrigeration industry is a problem to be solved. Natural refrigerant is the promising alternative refrigerant. CO2 has the advantage in low temperature circuit in cascade system due to its state of the art. Refrigerant to be used in high temperature circuit should exhibit a good COP and have good characteristics such as low toxicity and non-corrosiveness.
2. The thermodynamic analysis result of R290-R744 and R600a-R744 cascade systems demonstrates that the cycle has an optimum condensing temperature of the cascade condenser at a certain evaporating temperature (T_{be}) and certain temperature difference of a cascade condenser (ΔT_{cas}).
3. By the comparison with R290-R744 and R600a-R744 cascade refrigeration systems, the COP of R600a-R744 is larger than that of R290-R744 cycle for the same conditions.
4. A parametric study leads to the following conclusions
   a) An increase of superheat reduced the COP of the system
   b) An increase of subcooling increased the COP of the system
   c) An increase in evaporating temperature increased the COP of the system
   d) An increased in temperature difference in cascade condenser reduced the COP of the system
5. The maximum COP is obtain when ΔT_{cas} is at the lowest ($\Delta T_{cas} = 2°C$). However, in heat transfer, as ΔT_{cas} decreased, the area of heat transfer will increased as well. This will raise the cost of fabricating the heat exchanger. Therefore, there will be trade off in between ΔT_{cas} and area of heat transfer. This proofs that optimum COP exists in the cascade system and in R600a-R744 cascade system $COP_{optimum}$ is at ΔT_{cas} = 3.7°C

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APPENDICES

Appendix A: Gantt chart for FYP 2

No.	Detailed Work	1	2	3	4	5	6	7		8	9	10	11	12	13	14	15
1	**Literature review**																
2	**Project Work** • Collect the properties of R600a, R290 and R744 • Familiarize with REFPROP 9 database																
3	**Project Work** • Define the specific requirement of the system • Varies the design parameter • Find the optimum COP and Tmc of R600a/R290 – R744 cascade system • Extract any related figures or graphs • Result revision																
4	**Submission of Progress Report**								Mid - Sem Break								
5	**Project Work Compilation** • Revision and compilation of project work • Preparation of Project Poster • Preparation of Dissertation Preparation of Technical Report																
7	**Submission of Draft Report**																
9	**Submission of Technical Report**																
10	**Oral Presentation**																
11	**Submission of Project Dissertation (Hard Bound)**																

R744 - Carbon Dioxide Saturation Properties - Temperature Table (-20°C - 5°C)

Temp	Pressure	volume (m^3/kg)		enthalpy (kJ/kg)			entropy (kJ/kg.K)		
°C	MPa	vf	vg	hf	hfg	hg	sf	sfg	sg
-20	1.9696	0.000969	0.019343	154.45	282.44	436.89	0.8328	1.1157	1.9485
-19	2.0310	0.000974	0.018726	156.61	280.20	436.81	0.8411	1.1025	1.9436
-18	2.0938	0.000978	0.018131	158.77	277.93	436.70	0.8494	1.0892	1.9386
-17	2.1581	0.000983	0.017557	160.95	275.63	436.58	0.8577	1.0761	1.9337
-16	2.2237	0.000987	0.017002	163.14	273.30	436.44	0.8659	1.0628	1.9287
-15	2.2908	0.000992	0.016467	165.34	270.93	436.27	0.8742	1.0495	1.9237
-14	2.3593	0.000997	0.015950	167.55	268.54	436.09	0.8825	1.0362	1.9187
-13	2.4294	0.001002	0.015450	169.78	266.11	435.89	0.8908	1.0229	1.9137
-12	2.5010	0.001007	0.014967	172.01	263.65	435.66	0.8991	1.0095	1.9086
-11	2.5740	0.001012	0.014500	174.26	261.15	435.41	0.9074	0.9962	1.9036
-10	2.6487	0.001017	0.014048	176.52	258.62	435.14	0.9157	0.9828	1.8985
-9	2.7249	0.001023	0.013611	178.80	256.04	434.84	0.9241	0.9694	1.8934
-8	2.8027	0.001028	0.013188	181.09	253.42	434.51	0.9324	0.9558	1.8882
-7	2.8821	0.001034	0.012778	183.39	250.78	434.17	0.9408	0.9422	1.8830
-6	2.9632	0.001040	0.012381	185.71	248.08	433.79	0.9492	0.9287	1.8778
-5	3.0459	0.001046	0.011996	188.05	245.33	433.38	0.9576	0.9149	1.8725
-4	3.1303	0.001052	0.011624	190.40	242.55	432.95	0.9660	0.9012	1.8672
-3	3.2164	0.001058	0.011262	192.77	239.71	432.48	0.9744	0.8874	1.8618
-2	3.3042	0.001065	0.010911	195.16	236.83	431.99	0.9829	0.8734	1.8563
-1	3.3938	0.001071	0.010571	197.57	233.89	431.46	0.9915	0.8595	1.8509
0	3.4851	0.001078	0.010241	200.00	230.89	430.89	1.0000	0.8453	1.8453
1	3.5783	0.001085	0.009920	202.45	227.84	430.29	1.0086	0.8311	1.8397
2	3.6733	0.001093	0.009609	204.93	224.72	429.65	1.0172	0.8168	1.8340
3	3.7701	0.001100	0.009306	207.43	221.54	428.97	1.0259	0.8023	1.8282
4	3.8688	0.001108	0.009011	209.95	218.3	428.25	1.0346	0.7877	1.8223
5	3.9695	0.001116	0.008724	212.50	214.98	427.48	1.0434	0.7729	1.8163

Appendix C: Thermodynamic Properties Table for Propane

Temperature (^oC)	Pressure (MPa)	h_f (kJ/kg)	h_g (kJ/kg.K)	s_f (kJ/kg.K)	s_g (kJ/kg.K)
-50	0.07056	82.75	516.48	0.52975	2.4734
-45	0.08905	93.88	522.49	0.57893	2.4575
-40	0.11112	105.12	528.48	0.62751	2.4433
-35	0.13723	116.49	534.45	0.67554	2.4306
-32	0.15502	123.36	538.01	0.70411	2.4236
-30	0.16783	127.97	540.38	0.72306	2.4192
-27	0.18856	134.93	543.93	0.75135	2.4129
-25	0.20343	139.60	546.28	0.77012	2.4090
-22	0.22739	146.64	549.80	0.79815	2.4034
-20	0.24452	151.36	552.13	0.81676	2.3999
-17	0.27203	158.49	555.62	0.84456	2.3949
-15	0.29162	163.28	557.93	0.86303	2.3918
-12	0.32300	170.50	561.37	0.89063	2.3874
-10	0.34528	175.35	563.65	0.90897	2.3846
-8	0.36870	180.22	565.92	0.92726	2.3819
-7	0.38085	182.67	567.05	0.93638	2.3806
-6	0.39329	185.12	568.18	0.94550	2.3794
-5	0.40604	187.59	569.30	0.95461	2.3781
-2	0.44613	195.01	572.65	0.98187	2.3746
0	0.47446	200.00	574.87	1.00000	2.3724
3	0.51943	207.54	578.16	1.02710	2.3692
5	0.55112	212.60	580.33	1.04520	2.3672
8	0.60131	220.25	583.55	1.07220	2.3644
10	0.63660	225.40	585.67	1.09020	2.3626
15	0.73151	238.40	590.89	1.13510	2.3583
20	0.83646	251.64	595.95	1.17990	2.3544
25	0.95207	265.11	600.84	1.22470	2.3507
30	1.07900	278.83	605.54	1.26950	2.3471
35	1.21790	292.84	610.01	1.31430	2.3436
40	1.36940	307.15	614.21	1.35940	2.3399
45	1.53430	321.79	618.12	1.40460	2.3360
50	1.71330	336.80	621.66	1.45020	2.3317

Appendix D: Thermodynamic Properties Table for Isobutane

Temperature (^oC)	Pressure (MPa)	h_f (kJ/kg)	h_g (kJ/kg.K)	s_f (kJ/kg.K)	s_g (kJ/kg.K)
-40	0.028702	112.51	501.35	0.65491	2.3227
-35	0.036797	123.04	507.85	0.69955	2.3154
-30	0.046622	133.68	514.40	0.74369	2.3095
-25	0.058427	144.43	520.99	0.78738	2.3048
-20	0.072477	155.30	527.61	0.83064	2.3013
-15	0.089053	166.29	534.26	0.87351	2.2989
-10	0.108450	177.40	540.93	0.91601	2.2975
-5	0.130980	188.63	547.63	0.95816	2.2969
0	0.156960	200.00	554.34	1.00000	2.2972
5	0.186720	211.50	561.06	1.04150	2.2983
10	0.220610	223.15	567.78	1.08280	2.3000
15	0.258990	234.94	574.50	1.12390	2.3023
20	0.302220	246.88	581.21	1.16470	2.3051
25	0.350670	258.98	587.90	1.20530	2.3085
30	0.404720	271.24	594.57	1.24580	2.3123
35	0.464770	283.67	601.21	1.28610	2.3165
40	0.531210	296.28	607.80	1.32630	2.3211
45	0.604450	309.07	614.34	1.36640	2.3259

Appendix E: COP of R600a-R744 Cascade System

Tmc	ΔT sub/sup	SUPERHEAT CO2 ONLY				SUPERHEAT ISOBUTANE ONLY				SUBCOOLING CO2 ONLY			
		ΔT cas = 2	ΔT cas = 3	ΔT cas = 4	ΔT cas = 5	ΔT cas = 2	ΔT cas = 3	ΔT cas = 4	ΔT cas = 5	ΔT cas = 2	ΔT cas = 3	ΔT cas = 4	ΔT cas = 5
		COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal
5	0	1.930	1.902	1.873	1.845	1.930	1.902	1.873	1.845	1.930	1.902	1.873	1.845
	2	1.926	1.897	1.869	1.841	1.933	1.904	1.876	1.847	1.959	1.930	1.901	1.872
	4	1.921	1.893	1.864	1.837	1.936	1.907	1.879	1.851	1.987	1.957	1.926	1.897
	6	1.916	1.888	1.859	1.832	1.939	1.910	1.882	1.854	2.013	1.983	1.952	1.922
	8	1.912	1.884	1.855	1.828	1.943	1.914	1.885	1.858	2.039	2.008	1.976	1.946
0	0	1.975	1.943	1.912	1.882	1.975	1.943	1.912	1.882	1.975	1.943	1.912	1.882
	2	1.970	1.938	1.908	1.877	1.979	1.947	1.915	1.885	1.999	1.966	1.935	1.903
	4	1.966	1.934	1.903	1.873	1.982	1.951	1.920	1.888	2.021	1.988	1.956	1.924
	6	1.960	1.929	1.898	1.868	1.986	1.954	1.923	1.892	2.043	2.009	1.976	1.944
	8	1.956	1.925	1.894	1.864	1.990	1.958	1.927	1.897	2.064	2.029	1.996	1.963
-5	0	1.996	1.962	1.928	1.895	1.996	1.962	1.928	1.895	1.996	1.962	1.928	1.895
	2	1.991	1.957	1.924	1.891	2.000	1.966	1.933	1.899	2.015	1.980	1.946	1.912
	4	1.987	1.953	1.920	1.887	2.005	1.970	1.937	1.904	2.032	1.997	1.963	1.929
	6	1.982	1.948	1.915	1.882	2.009	1.975	1.941	1.909	2.050	2.014	1.979	1.944
	8	1.978	1.944	1.911	1.878	2.014	1.980	1.946	1.914	2.067	2.031	1.995	1.960
-10	0	1.995	1.960	1.924	1.889	1.995	1.960	1.924	1.889	1.995	1.960	1.924	1.889
	2	1.991	1.955	1.919	1.885	2.001	1.964	1.929	1.894	2.010	1.974	1.937	1.903
	4	1.987	1.951	1.915	1.881	2.006	1.970	1.934	1.900	2.024	1.987	1.950	1.915
	6	1.982	1.947	1.911	1.877	2.011	1.975	1.940	1.905	2.037	2.000	1.963	1.928
	8	1.978	1.943	1.908	1.874	2.017	1.981	1.946	1.911	2.050	2.013	1.975	1.939
-15	0	1.974	1.936	1.900	1.864	1.974	1.936	1.900	1.864	1.974	1.936	1.900	1.864
	2	1.970	1.933	1.896	1.860	1.980	1.942	1.905	1.869	1.985	1.947	1.910	1.874
	4	1.967	1.929	1.893	1.857	1.986	1.948	1.912	1.875	1.995	1.957	1.920	1.883
	6	1.962	1.925	1.889	1.853	1.993	1.955	1.918	1.882	2.005	1.967	1.929	1.893
	8	1.959	1.922	1.886	1.850	1.999	1.962	1.924	1.889	2.015	1.976	1.938	1.901
-20	0	1.934	1.895	1.858	1.821	1.934	1.895	1.858	1.821	1.934	1.895	1.858	1.821
	2	1.931	1.892	1.855	1.818	1.941	1.902	1.864	1.827	1.942	1.903	1.865	1.828
	4	1.928	1.890	1.853	1.816	1.948	1.909	1.871	1.834	1.949	1.910	1.872	1.835
	6	1.925	1.887	1.850	1.813	1.955	1.916	1.879	1.842	1.957	1.917	1.879	1.842
	8	1.922	1.884	1.847	1.811	1.962	1.924	1.886	1.849	1.964	1.924	1.886	1.848

Tmc	ΔT sub/sup	SUBCOOLING ISOBUTANE ONLY				SUPERHEAT CO2 & ISOBUTANE ONLY				SUBCOOLING CO2 & ISOBUTANE ONLY			
		ΔT cas = 2	ΔT cas = 3	ΔT cas = 4	ΔT cas = 5	ΔT cas = 2	ΔT cas = 3	ΔT cas = 4	ΔT cas = 5	ΔT cas = 2	ΔT cas = 3	ΔT cas = 4	ΔT cas = 5
		COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal
5	0	1.930	1.902	1.873	1.845	1.930	1.902	1.873	1.845	1.930	1.902	1.873	1.845
	2	1.949	1.920	1.892	1.864	1.929	1.900	1.871	1.843	1.978	1.949	1.920	1.891
	4	1.967	1.938	1.910	1.882	1.927	1.898	1.870	1.842	2.024	1.995	1.965	1.936
	6	1.984	1.956	1.927	1.900	1.924	1.896	1.867	1.840	2.071	2.040	2.010	1.980
	8	2.001	1.973	1.944	1.917	1.924	1.895	1.867	1.840	2.116	2.085	2.054	2.024
0	0	1.975	1.943	1.912	1.882	1.975	1.943	1.912	1.882	1.975	1.943	1.912	1.882
	2	1.998	1.966	1.935	1.904	1.974	1.942	1.911	1.880	2.021	1.989	1.957	1.926
	4	2.019	1.987	1.957	1.926	1.973	1.941	1.910	1.879	2.067	2.034	2.002	1.970
	6	2.040	2.009	1.978	1.947	1.971	1.940	1.909	1.878	2.111	2.078	2.045	2.013
	8	2.061	2.029	1.999	1.968	1.971	1.939	1.909	1.879	2.156	2.122	2.089	2.056
-5	0	1.996	1.962	1.928	1.895	1.996	1.962	1.928	1.895	1.996	1.962	1.928	1.895
	2	2.022	1.988	1.955	1.921	1.995	1.961	1.928	1.895	2.041	2.006	1.973	1.939
	4	2.048	2.014	1.980	1.947	1.995	1.961	1.928	1.895	2.086	2.051	2.016	1.982
	6	2.073	2.039	2.005	1.972	1.995	1.961	1.928	1.896	2.130	2.094	2.059	2.024
	8	2.097	2.063	2.030	1.996	1.996	1.962	1.929	1.897	2.175	2.138	2.103	2.067
-10	0	1.995	1.960	1.924	1.889	1.995	1.960	1.924	1.889	1.995	1.960	1.924	1.889
	2	2.026	1.990	1.954	1.919	1.996	1.960	1.924	1.890	2.040	2.004	1.968	1.933
	4	2.055	2.019	1.983	1.948	1.997	1.961	1.926	1.892	2.085	2.048	2.011	1.975
	6	2.084	2.048	2.012	1.977	1.998	1.962	1.927	1.893	2.129	2.092	2.054	2.018
	8	2.112	2.076	2.040	2.005	2.000	1.964	1.929	1.895	2.173	2.135	2.097	2.060
-15	0	1.974	1.936	1.900	1.864	1.974	1.936	1.900	1.864	1.974	1.936	1.900	1.864
	2	2.008	1.970	1.933	1.897	1.976	1.939	1.902	1.866	2.019	1.980	1.943	1.907
	4	2.041	2.003	1.966	1.929	1.979	1.941	1.905	1.869	2.063	2.024	1.987	1.950
	6	2.073	2.035	1.998	1.961	1.981	1.943	1.907	1.871	2.107	2.068	2.030	1.992
	8	2.105	2.067	2.029	1.993	1.984	1.947	1.910	1.875	2.151	2.111	2.073	2.035
-20	0	1.934	1.895	1.858	1.821	1.934	1.895	1.858	1.821	1.934	1.895	1.858	1.821
	2	1.971	1.932	1.894	1.857	1.938	1.899	1.861	1.825	1.979	1.940	1.902	1.864
	4	2.007	1.968	1.930	1.892	1.942	1.904	1.866	1.829	2.023	1.984	1.945	1.907
	6	2.043	2.003	1.965	1.927	1.945	1.907	1.870	1.833	2.068	2.027	1.988	1.950
	8	2.078	2.038	2.000	1.962	1.950	1.912	1.875	1.838	2.112	2.071	2.031	1.992

Appendix F: COP of R290-R744 Cascade System

		SUPERHEAT CO2 ONLY				SUPERHEAT PROPANE ONLY				SUBCOOLING CO2 ONLY				SUBCOOLING PROPANE ONLY				SUPERHEAT CO2 & PROPANE ONLY				SUBCOOLING CO2 & PROPANE ONLY			
		ΔT cas = 2	ΔT cas = 3	ΔT cas = 4	ΔT cas = 5	ΔT cas = 2	ΔT cas = 3	ΔT cas = 4	ΔT cas = 5	ΔT cas = 2	ΔT cas = 3	ΔT cas = 4	ΔT cas = 5	ΔT cas = 2	ΔT cas = 3	ΔT cas = 4	ΔT cas = 5	ΔT cas = 2	ΔT cas = 3	ΔT cas = 4	ΔT cas = 5	ΔT cas = 2	ΔT cas = 3	ΔT cas = 4	ΔT cas = 5
Tmc	ΔT sub/sup	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal	COP ideal
5	0	1.891	1.862	1.833	1.805	1.891	1.862	1.833	1.805	1.891	1.862	1.833	1.805	1.891	1.862	1.833	1.805	1.891	1.862	1.833	1.805	1.891	1.862	1.833	1.805
	2	1.886	1.857	1.829	1.801	1.892	1.864	1.836	1.807	1.919	1.889	1.860	1.831	1.910	1.881	1.853	1.825	1.888	1.860	1.831	1.803	1.939	1.909	1.880	1.851
	4	1.882	1.853	1.825	1.797	1.895	1.866	1.837	1.810	1.945	1.914	1.885	1.855	1.930	1.901	1.873	1.845	1.886	1.857	1.829	1.801	1.986	1.956	1.927	1.897
	6	1.877	1.848	1.820	1.792	1.897	1.868	1.839	1.811	1.971	1.939	1.909	1.879	1.949	1.921	1.893	1.865	1.883	1.854	1.826	1.798	2.034	2.003	1.973	1.943
	8	1.873	1.844	1.816	1.788	1.899	1.870	1.842	1.814	1.995	1.963	1.933	1.902	1.968	1.939	1.912	1.884	1.881	1.852	1.824	1.797	2.080	2.048	2.018	1.988
0	0	1.929	1.897	1.867	1.836	1.929	1.897	1.867	1.836	1.929	1.897	1.867	1.836	1.929	1.897	1.867	1.836	1.929	1.897	1.867	1.836	1.929	1.897	1.867	1.836
	2	1.924	1.893	1.862	1.832	1.932	1.900	1.869	1.839	1.951	1.919	1.888	1.857	1.952	1.921	1.890	1.860	1.927	1.895	1.865	1.834	1.975	1.943	1.912	1.881
	4	1.920	1.888	1.858	1.828	1.934	1.903	1.872	1.842	1.973	1.940	1.908	1.877	1.976	1.945	1.914	1.884	1.925	1.894	1.863	1.833	2.022	1.989	1.958	1.926
	6	1.915	1.883	1.853	1.823	1.937	1.905	1.875	1.844	1.994	1.960	1.928	1.896	2.000	1.968	1.938	1.907	1.922	1.891	1.861	1.830	2.068	2.035	2.003	1.971
	8	1.911	1.879	1.849	1.819	1.939	1.908	1.877	1.847	2.014	1.980	1.947	1.914	2.022	1.991	1.960	1.930	1.921	1.890	1.859	1.829	2.114	2.080	2.047	2.015
-5	0	1.945	1.911	1.878	1.845	1.945	1.911	1.878	1.845	1.945	1.911	1.878	1.845	1.945	1.911	1.878	1.845	1.945	1.911	1.878	1.845	1.945	1.911	1.878	1.845
	2	1.940	1.906	1.874	1.841	1.948	1.914	1.881	1.849	1.962	1.928	1.895	1.861	1.972	1.938	1.905	1.872	1.943	1.910	1.877	1.845	1.990	1.956	1.922	1.889
	4	1.936	1.902	1.870	1.837	1.951	1.918	1.885	1.852	1.980	1.945	1.911	1.877	2.000	1.966	1.933	1.900	1.942	1.909	1.876	1.844	2.037	2.002	1.968	1.934
	6	1.931	1.898	1.865	1.833	1.954	1.921	1.888	1.856	1.996	1.961	1.926	1.892	2.027	1.993	1.960	1.927	1.941	1.907	1.875	1.843	2.083	2.047	2.012	1.978
	8	1.927	1.894	1.861	1.829	1.958	1.923	1.891	1.858	2.013	1.977	1.942	1.907	2.054	2.020	1.987	1.954	1.940	1.906	1.874	1.842	2.128	2.092	2.057	2.022

-10	0	1.939	1.905	1.870	1.836	1.939	1.905	1.870	1.836	1.939	1.905	1.870	1.836	1.939	1.905	1.870	1.836	1.939	1.905	1.870	1.836	1.939	1.905	1.870	1.836
	2	1.935	1.900	1.866	1.832	1.944	1.908	1.874	1.840	1.953	1.918	1.883	1.849	1.970	1.935	1.900	1.866	1.939	1.904	1.870	1.836	1.984	1.949	1.914	1.879
	4	1.931	1.896	1.862	1.828	1.948	1.912	1.877	1.843	1.966	1.931	1.895	1.861	2.002	1.967	1.932	1.898	1.939	1.904	1.869	1.835	2.031	1.995	1.959	1.924
	6	1.927	1.893	1.858	1.825	1.952	1.916	1.881	1.847	1.979	1.943	1.907	1.872	2.034	1.998	1.963	1.929	1.939	1.904	1.869	1.836	2.077	2.040	2.004	1.968
	8	1.923	1.889	1.854	1.821	1.955	1.920	1.885	1.851	1.992	1.955	1.919	1.884	2.064	2.029	1.993	1.959	1.938	1.904	1.869	1.836	2.122	2.085	2.048	2.012
-15	0	1.915	1.879	1.843	1.808	1.915	1.879	1.843	1.808	1.915	1.879	1.843	1.808	1.915	1.879	1.843	1.808	1.915	1.879	1.843	1.808	1.915	1.879	1.843	1.808
	2	1.912	1.876	1.840	1.805	1.920	1.883	1.848	1.812	1.926	1.889	1.853	1.818	1.950	1.913	1.877	1.842	1.916	1.880	1.844	1.809	1.960	1.924	1.887	1.851
	4	1.909	1.873	1.837	1.802	1.925	1.888	1.852	1.817	1.936	1.899	1.862	1.827	1.985	1.949	1.912	1.877	1.918	1.881	1.846	1.810	2.007	1.970	1.932	1.896
	6	1.904	1.869	1.833	1.798	1.928	1.892	1.856	1.821	1.945	1.908	1.872	1.835	2.020	1.983	1.947	1.911	1.917	1.881	1.845	1.811	2.053	2.015	1.977	1.940
	8	1.901	1.866	1.830	1.795	1.933	1.897	1.861	1.826	1.955	1.917	1.880	1.844	2.054	2.017	1.980	1.944	1.919	1.883	1.848	1.813	2.098	2.060	2.022	1.984
-20	0	1.874	1.837	1.800	1.765	1.874	1.837	1.800	1.765	1.874	1.837	1.800	1.765	1.874	1.837	1.800	1.765	1.874	1.837	1.800	1.765	1.874	1.837	1.800	1.765
	2	1.871	1.834	1.798	1.762	1.879	1.842	1.805	1.770	1.881	1.844	1.807	1.771	1.911	1.874	1.837	1.801	1.876	1.839	1.802	1.767	1.918	1.881	1.844	1.808
	4	1.868	1.832	1.795	1.760	1.884	1.846	1.810	1.774	1.888	1.851	1.814	1.778	1.949	1.912	1.875	1.838	1.878	1.841	1.805	1.769	1.965	1.927	1.889	1.852
	6	1.865	1.829	1.792	1.757	1.889	1.851	1.815	1.779	1.895	1.858	1.821	1.784	1.987	1.950	1.912	1.875	1.880	1.843	1.807	1.771	2.011	1.973	1.934	1.897
	8	1.863	1.826	1.790	1.755	1.894	1.856	1.820	1.784	1.902	1.864	1.827	1.790	2.025	1.987	1.949	1.911	1.882	1.846	1.810	1.774	2.057	2.018	1.979	1.940

Typical Overall Heat Transfer Coefficients in Heat Exchangers

Type	Application and Conditions	*U* W/(m² K)[1]	*U* Btu/(ft² °F h)[1]
Tubular, heating or cooling	Gases at atmospheric pressure inside and outside tubes	5 - 35	1 - 6
	Gases at high pressure inside and outside tubes	150 - 500	25 - 90
	Liquid outside (inside) and gas at atmospheric pressure inside (outside) tubes	15 - 70	3 - 15
	Gas at high pressure inside and liquid outside tubes	200 - 400	35 - 70
	Liquids inside and outside tubes	150 - 1200	25 - 200
	Steam outside and liquid inside tubes	300 - 1200	50 - 200
Tubular, condensation	Steam outside and cooling water inside tubes	1500 - 4000	250 - 700
	Organic vapors or ammonia outside and cooling water inside tubes	300 - 1200	50 - 200
Tubular, evaporation	steam outside and high-viscous liquid inside tubes, natural circulation	300 - 900	50 - 150
	steam outside and low-viscous liquid inside tubes, natural circulation	600 - 1700	100 - 300
	steam outside and liquid inside tubes, forced circulation	900 - 3000	150 - 500
Air-cooled heat exchangers[2]	Cooling of water	600 - 750	100 - 130
	Cooling of liquid light hydrocarbons	400 - 550	70 - 95
	Cooling of tar	30 - 60	5 - 10
	Cooling of air or flue gas	60 - 180	10 - 30
	Cooling of hydrocarbon gas	200 - 450	35 - 80
	Condensation of low pressure steam	700 - 850	125 - 150
	Condensation of organic vapors	350 - 500	65 - 90
Plate heat exchanger	liquid to liquid	1000 - 4000	150 - 700
Spiral heat exchanger	liquid to liquid	700 - 2500	125 - 500
	condensing vapor to liquid	900 - 3500	150 - 700

Notes:

1) 1 Btu/(ft² °F h) = 5.6785 W/(m² K)

2) Coefficients are based on outside bare tube surface

Source: http://www.deltathx.com/uploadsDocs/U.pdf

Appendix H: COP and Surface Area of cascade condenser (A) with variation in ΔTcas for R600a-R744 cascade system

ΔTcas	A (m^2)	COP	log(1/A)	log(COP)	y = 0.0009x + 0.0561	y = -0.0364x + 2.2409
2	19.87	2.17	0.050241	2.173822	0.0579	2.1681
3	19.53	2.13	0.051376	2.125167	0.0588	2.1317
4	19.20	2.09	0.052182	2.090645	0.0597	2.0953
5	18.89	2.06	0.052806	2.063867	0.0606	2.0589