Performance Investigation of Transcritical

Carbon Dioxide Refrigeration Cycle

by

Allya Radzihan Binti Reduan

Dissertation submitted in partial fulfillment of

the requirements for the

Bachelor Of Engineering (Hons)

(Mechanical Engineering)

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CERTIFICATIONS

CERTIFICATIONS OF APPROVAL

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Approved by,

_______________________,

(Dr. Aklilu Tesfamichael Baheta)

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TRONOH, PERAK

JANUARY 2014

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 of persons.

________________________________,
ALLYA RADZIHAN BINTI REDUAN

ABSTRACT

Carbon dioxide has a high critical pressure and low critical temperature that is 7.36 MPa and 31.1^{o}C respectively causing the cycle to work in a transcritical nature. At this state, heat is rejected at a supercritical pressure and heat absorption occurs at a subcritical level. Above the critical properties, the pressure and temperature of CO_2 becomes independent of one another. Thus, in this region, specifying the operating conditions would be harder. It is important to identify and control the optimum heat rejection pressure so that the cycle will give the highest COP. Thus, the objective of this project was to investigate the performance of a transcritical carbon dioxide compression refrigeration cycle and validate its coefficient of performance. To achieve this objective, CO_2 refrigeration cycle model was modeled and simulated for analysis of the parameters. The analysis was carried out using EXCEL by tabulating the data and through graphs. The outcome of this experiment showed that COP has an almost linear relationship with cycle parameters and optimum pressure does exist and changes according to the cycle parameters. Moreover, parameters such as T_3, P_2 and T_1 were identified to have significant effect in obtaining a better COP value as well as lower optimum pressures. Based on these simulations, it was also identified that lower T_3 gives better COP and lowers the optimum pressure. For evaporator temperature however, higher T_1 gives better COP and lowers the optimum pressure. Apart from that, both P_2 and P_1 cannot be too close to critical point properties at the same time as it destroy the capability of the system to refrigerate.

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ABBREVIATION AND NOMENCLATURE

CO_2 – carbon dioxide

R134a - 1,1,1,2-TETRAFLUOROETHANE

COP – coefficient of performance

GWP – global warming potential

ODP – ozone depletion potential

ppm – parts per million

MPa – Megapascals

h – enthalpy values

$\dot{W}_{2-1}$= Work input to the compressor

$\dot{Q}_{1-4}$= Heat absorbed in the evaporator

s_1, s_2 = entropy values at point 1 and 2

x_4 = quality at point 4 (evaporator inlet)

$\eta_{is,c}$ = compressor isentropic efficiency

$\dot{m}$ = refrigerant mass flow rate

P = Pressure

<u>Subscripts:</u>

1 = evaporator outlet

2 = gas cooler inlet

3 = gas cooler outlet

4 = evaporator inlet

f4 = saturated liquid at point 4

g4 = saturated gas at point 4

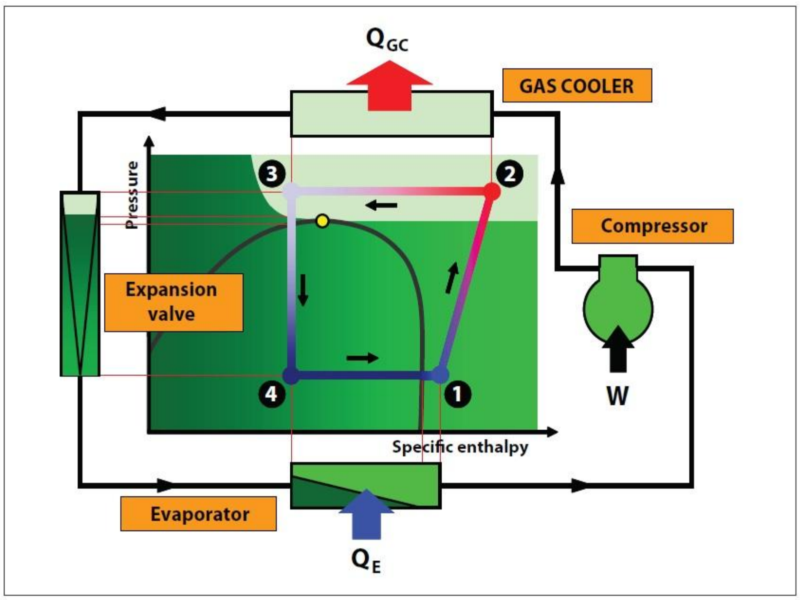
CHAPTER 1
INTRODUCTION

1.1 Background of Study

Refrigerant can be defined as a working fluid of a refrigerator that cools and provide refrigeration [1]. In the world today, there are many types of refrigerants used in application such as storage of food and beverages, cooling of factory equipments as well as to provide air conditioning to maintain an acceptable comfort level. There are many acceptable characteristics of refrigerant that can be used as a cooling medium. For example, the refrigerant must be safe, stable, easily available and can be obtained at an effective cost. Moreover, an effective refrigerant must be efficient, possessed acceptable range of COP and does not create any negative side effects like toxics and health problems when used.

Nowadays, one of the main refrigerants used is R-134a which is a form of hydrocarbons. However, concerns rose regarding the global warming effect from long term use of R-134a due to its high global warming potential (GWP). This issue urged researchers to find other forms of refrigerant be it synthetic or natural [2]. One of the natural refrigerants considered was carbon dioxide. First ever usage of carbon dioxide for cooling purposes was recorded in 1866 [3, 4] when it was first harnessed for ice production by Thaddeus S. C. Lowe [3, 4]. Following a period of further development by scientist, the first carbon dioxide plant was installed in 1890 in the marine industries [5]. It became one the main refrigerant in marine refrigeration [4] unit by 1950-1960 [6]. Carbon dioxide has many excellent qualities that contribute to its popularity in the refrigerating business. Firstly, carbon dioxide is a natural, non-flammable and non-toxic [4, 5, 7] refrigerant with no ozone depletion potential [5]. Moreover, it has negligible global warming potential [5] compare to conventional hydroflurocarbon refrigerants which produce 1300 times more global warming [2, 8]. Apart from that, carbon dioxide exists abundantly in the atmosphere and inert which makes it compatible with normal lubricants and common machine construction materials [7].

Apart from that, carbon dioxide or R-744 has a few thermodynamics properties that make it ideal to be used as a refrigerant for refrigerating cycles. For example, carbon dioxide has high latent heat, high volumetric refrigeration capacity (3-10 times higher than hydrofluorocarbon based refrigerants) [6] and excellent heat transfer properties [7, 8] making it suitable for heat transfer [4]. Another characteristic of carbon dioxide is that its critical pressure and critical temperature of carbon dioxide is 7.37 MPa and 31.1^{O}C respectively [8, 9, 14]. The low critical temperature causes the heat rejection process to occur above the critical point and heat absorption process to happen below the critical point. This condition is called transcritical state [7, 10, 12]. Figure 1 below illustrates in more detail the basic transcritical refrigeration cycle process for carbon dioxide.



Critical point
Supercritical fluid region
* Components are coloured according to the phase transition they undergone

FIGURE 1.1: Transcritical Cycle Process and System Components [11]

At process 1-2, the cycle experienced a one-stage compression where the refrigerant is compressed to supercritical conditions. Then, at point 2-3, the heat rejection process occurs at elevated condition above the critical point while maintaining a constant pressure. The condenser used in conventional refrigeration cycle is replaced by gas cooler in this process. Here, the temperature varies continuously beginning from the inlet (point 2) to the outlet temperature (point 3). Next, expansion occurs from stage 3-4 at constant enthalpy in the expansion valve. At point 3, the condition is supercritical and point 4 is a mixture. Last but not least, heat absorption occurs inside the evaporator at constant pressure (process 4-1).

There are four distinct features [8, 9] of the transcritical carbon dioxide refrigeration cycle.

(i) Heat is rejected at supercritical pressure and the fluid exists in the superheated condition.

This process happens due to the low critical temperature of carbon dioxide. Above critical point, the refrigerant exists in a superheated region where both pressure and temperature are independent of one another [8, 12]. To determine the pressure in this region, it is determined by the refrigerant charge instead of the saturation pressure. In order to achieve the desired coefficient of performance (COP), it is important for the high side pressure to be controlled [8].

(ii) Existence of optimum pressure which gives maximum COP.

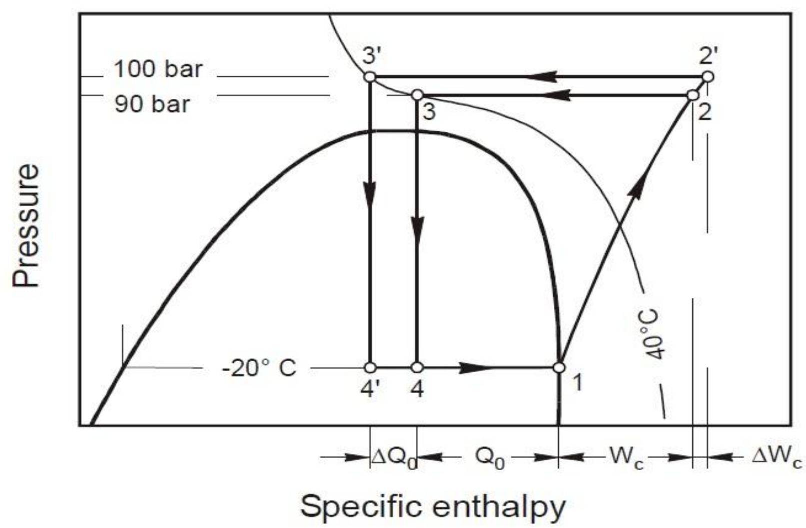


FIGURE 1.2: P-h diagram Of Two Transcritical Refrigerating Cycle [4]

In the transcritical carbon dioxide refrigeration cycle, exists an optimum pressure where the maximum value of coefficient of performance can be achieved [13]. Figure 1.2 explains the existence of optimum pressure in the cycle. In the diagram, all parameters of the system were set constant and gas cooler pressures were varied which affect fluid enthalpy [8]. It can be seen that by moving the cycle points from 1-2-3-4 at 90 bar to 100 bar at 1-2’-3’-4’, both the refrigerating capacity and work input of the compressor increased [4]. Isotherm line also becomes steeper as shown in Figure 1.2 [8]. At this point the COP will increase. However, as the isotherm lines gets steeper, then enthalpy value difference between point 1 and 4 decreases thus causing the refrigerating capacity increment to decrease, thus causing the cycle unable to compensate the increase of compressor work input. Here, the

COP will decrease after reaching the maximum value. This optimum pressure is not constant and changes depending on different parameter settings [13] of the cycle.

(iii) The pressure level of the system is relatively high ranging from 3 MPa to 12 MPa [9].

This characteristic is related to the heat rejection process which occurs above the critical point of carbon dioxide. High pressure for a refrigerating system can be a disadvantage to carbon dioxide as manufacturer may struggle to find the suitable material that can withstand such pressure for everyday use as well as create an imminent hazard risk of explosion and failures. However, this property results in high volumetric capacity [9] which allow carbon dioxide to store internal energy without undergo any phase change. Moreover, compare with hydrofluorocarbon type refrigerant, high operating pressures in either gas cooler or evaporator results in a more efficient heat transfer [9].

(iv) During heat rejection process, the refrigerant experience large temperatures glide [8].

Above the critical point, heat is rejected by cooling dense single phase gas at constant pressure. To accommodate this process, the heat exchanger used is called gas cooler instead of condenser. Previous studies have shown that it is better to cool the refrigerant to a lower temperature at gas cooler exit [9] as the coefficient of performance value is bigger. Besides, the COP value may be higher than a system utilizing HFC refrigerant if this condition is fulfilled.

Based on carbon dioxide transcritical refrigeration cycle features elaborated, some disadvantages relating to the system performance was identified. One of the disadvantages is, due to the high pressure level where heat rejection occurs, there is a need to control the pressure. The pressure can be kept constant however it is not the most energy efficient method [12]. Some of the method suggested is to adopt dynamic pressure control [12]. In order to do so, it is important to understand the effect of other parameters on the pressure. The gas cooler optimum pressure also influenced the highest value for COP the cycle can produce [15]. This pressure however is not constant and dependent on other parameters of the cycle [15].

Apart from that, transcritical carbon dioxide refrigeration cycle also experience performance issues with low coefficient of performance value [4]. One of the reasons is carbon dioxide refrigeration cycle experienced huge expansion loss compare to refrigeration cycle utilizing conventional refrigerants [7]. Gas cooling above the critical pressure also can penalize the COP if a close temperature approach is not reached [4]. The COP value in carbon dioxide refrigeration cycle is also influenced by many variables such as the refrigerating effect and the compression work [4]. These two quantities affect if the COP increases or decreases. Apart from that, parameters such as evaporation temperature, gas cooler outlet temperature, discharge pressure and isentropic efficiency also influence the cycle performance [5].

Based on these disadvantages, the problem statement of the project was identified and elaborated more in the next section.

1.2 Problem Statement

Transcritical carbon dioxide refrigeration cycle performance has lower performance [4] compare to refrigeration cycle utilizing hydrofluorocarbon refrigerants. Carbon dioxide cycle coefficient of performance is influenced mainly by the refrigeration effect and compression work [4] and if the properties involved are identified and controlled the COP can be improved. For example, for refrigeration effect, properties like evaporator temperature and pressure may increase or reduce the COP. Thus, it is important to understand the effect of each parameter on the coefficient of performance.

Secondly, the optimum discharge pressure in the cycle gives a maximum coefficient of performance. This optimum discharge pressure is the pressure above the critical point. Ability to control the optimum pressure enables the coefficient of performance to be manipulated to the desired value. However, as it exists above the critical point, the value is a function of more than one parameter and changes depending on the parameters. For example, previous research has shown that this pressure is not only affected by the gas cooler pressure but also the evaporator properties. To control this pressure, an understanding of the relationship of each parameter on the optimum pressure must be achieved. Apart from that, parameter combination that gives the most significant effect on the maximum value for COP also must be identified.

Based on the problem statement outlined, the paper served to provide an understanding to the issues and may provide ways on how to improve the problems.

1.3 Objective & Scope of Study

Thus the objective of this project is to investigate the performance of a transcritical carbon dioxide compression refrigeration cycle and validate its coefficient of performance.

One of the scopes of this study will focused on developing the thermodynamic model of the transcritical refrigeration cycle. Here, the basic operation of a transcritical carbon dioxide refrigeration cycle is analyzed by identifying the refrigeration cycle components and processes happening in the components. At this stage thermodynamic governing equation is use to represent the processes occurring inside the components.

Apart from that, this study focuses on simulating the transcritical carbon dioxide refrigeration cycle model. Once the simulation model coefficient of performance is validated, various parameter manipulations are conducted to analyze the performance of the transcritical carbon dioxide refrigeration cycle. Last but not least, through the results also, the influence of different parameters on the performance and optimum pressure is investigated.

CHAPTER 2
LITERATURE REVIEW

Carbon dioxide is the primary greenhouse gases created through various forms of human activities. It is readily available in the Earth atmosphere and used in various processes such as for photosynthesis and greenhouse effect. However, human activities such as fossil fuel combustion have increased the concentration of carbon dioxide in the atmosphere above the safe level. As of 2013, the recorded concentration of carbon dioxide is 399.98 ppm compare to 360 ppm in 2011 [16]. Many solutions were suggested and adapted to reduce the carbon dioxide level in the atmosphere and one of it is reconsidering its application as a refrigerant.

Many researches were conducted to analyze carbon dioxide refrigeration capacity by testing its properties and comparing its refrigeration performance with other known refrigerants. One way to measure carbon dioxide performance as a refrigerant is by quantifying it using coefficient of performance. Among many issues which restrict carbon dioxide usage as a refrigerant in refrigeration cycle is its low coefficient of performance. In Sarkar et al (2008) [17], thermodynamic analysis and optimization studies was conducted between transcritical nitrous oxide (N_2O) and transcritical carbon dioxide through simulation. Comparisons were also done between the two refrigerants due to its similar properties although the outcome shows that the performances between both refrigerants are dissimilar. Sarkar et al (2008) shows that coefficient of performance for carbon dioxide refrigeration cycle is lower compare to the coefficient of performance of nitrous oxide (N_2O). In the model created for the simulation, an internal heat exchanger was added into the refrigeration system model which improved carbon dioxide system performance slightly.

Brown et al (2000) [18] supported this view through the result obtained by simulation to investigate carbon dioxide performance in automotive air conditioning system. In his research, a comparative evaluation between R-134a and carbon dioxide performance were made with an addition of liquid-line heat exchanger to the carbon dioxide air conditioning semi-theoretical cycle model. The outcome shows that R-134a using conventional refrigeration model has better performance compared to CO_2. Here, the author suggested that the low performance of carbon dioxide was due to the large entropy generation in gas cooler. Another author, Brown (2006) [19] also identifies that CO_2 COP is lower through his research. The differences between Brown et al (2000) research and Brown (2006) research was that, Brown (2006) runs a simulation to compare between one-stage and two stage CO_2 refrigeration cycle. Outcome of the research shows that despite having lower COP in one-stage refrigeration cycle, the COP achieved was slightly higher in a two-stage refrigeration cycle. An experiment conducted by Yamasaki et al (2004) [20] also shows that a higher COP of CO_2 can be achieved through equipments modifications which caters to CO_2 transcritical state as well as manipulating the right parameters to obtain the highest COP. Research conducted by these researches shows that despite having lower performance compare to other refrigerants, carbon dioxide system performance can be improved when conventional cycle undergoes modifications to cater to the specific properties of carbon dioxide and its cycle parameters. This shows the importance of understanding each parameters involved in carbon dioxide refrigeration cycle.

Through the research conducted, one of the characteristics of CO_2 refrigeration cycle identified was at certain optimum pressure, the coefficient of performance is at its maximum. This view was supported by an experiment conducted by Cabello et al (2008) [21]. In this experiment, the author focuses on the evaluation of the CO_2 refrigeration cycle energy efficiency by varying evaporator temperature, gas cooler exit temperature as well as the gas cooler pressure. The outcome of the experiment was that an optimal pressure for the maximum COP does exist for transcritical refrigeration cycle. After the optimum pressure, as the pressure increases, the COP value however decreases. Cabello et al (2008) also concludes that the optimum pressure is not constant and varied according to different parameter setting. In his experiment, the finding shows that higher gas cooler exit temperature and evaporator temperature resulted in higher optimum pressure. Through research paper review written by Ma et al (2012) [8], the author supported the outcome by stating that there must be a maximum value of COP along with the changes in system operating pressure, when other parameters are set to constant. Moreover, this view is supported in a research conducted by both Brown (2006) and Xue et al (2010) [22] although the simulation model and the parameters focus by both authors are different. Brown (2006) opted for one stage and two stage refrigeration cycle and varies every parameter in the cycle. Xue et al (2010) however maintained a one stage refrigeration model with the main components (evaporator, gas cooler, compressor and throttling device) and vary only the heat exchanger specification as well as refrigerant superheat conditions.

Thus, from previous researches, it was known that the ability to control high-side pressure will provide the most optimum COP. However, due to the transcritical state of CO_2 refrigeration cycle, the pressure will not be constant as it is influenced by various parameters of the cycle. In McEnaney (1999) [23], through his study of CO_2 application in mobile air conditioning supports this view. The outcome of the research shows that maximum COP was obtained by controlling the conditions at evaporator and gas cooler. Kim et al (2004) [6] also supports this view through his report by reviewing various research papers. Some of the operating conditions that affect the system optimum pressure are for example evaporator temperature, gas cooler exit temperature and pressure as well as components efficiency as mentioned in Sarkar (2010) [7]. Sarkar et al (2004) [15] also supported this view through simulation works on the transcritical heat pumps for simultaneous cooling and heating by identifying that optimum pressure was a function of various parameters such as gas cooler pressure and evaporator temperature.

Some of the effects of these parameters on COP are for example in Cabello et al (2008) [21] it was mentioned that the optimum pressure was also influenced by the refrigerant outlet temperature as well as the evaporating temperature. In a transcritical refrigeration cycle, the higher the temperature of refrigerant at gas cooler outlet, the higher the optimum pressure obtained. However, higher gas cooler outlet temperature, the COP value decreases. Sarkar et al (2004) [15] and Sarkar et al (2008) [17] compliments this view where their research shows that higher evaporator temperature and lower gas cooler exit temperature creates better cycle performance. Moreover, Perez-Garcia et al (2012) and Xue et al (2010) [5, 22], also supports that evaporation temperature influenced the performance of the cycle and added another variable which is the isentropic efficiency. Isentropic efficiency usually involves analyzing the compressor actual performance and the performance under idealized conditions for the same inlet and exit conditions. This parameter was highly affected by the enthalpy conditions at stated process. In Chen (2011) [24], the compressor and expander specification was also considered as the factors that influenced the performance of carbon dioxide transcritical cycles. With higher value of compressor efficiency a higher COP can be obtained. Brown (2006) [19] supports this view by showing that there is a linear relationship between compressor efficiency and the COP.

Here, it can be concluded that in order to obtain the maximum value of COP, the optimum pressure for the system must be achieved and controlled. Since the pressure is not constant and influenced by other working parameters, the relationship between the parameters and to what extent it influence the system COP must be understood. With this understanding only that the parameters that significantly affect the refrigeration cycle of COP could be control and CO_2 refrigeration system COP can be improved.

CHAPTER 3
METHODOLOGY

This section outlines the methodology used in this project. There are three main sections which is thermodynamic model formulation, simulation model and lastly the parameters variation. In the first section, the components selected and equations used to represent the processes inside the components were outlined. Base parameters and assumptions made were also explained in this section. In the simulation model section, few parameters were assigned to this model and the model was simulated using Microsoft Excel. Last but not least, in the parameters variation section, once the COP value obtained was validated, the parameters were varied.

3.1 Thermodynamic Model Formulation

Firstly, the transcritical carbon dioxide refrigeration cycle was modeled. The model consists of four components which was the compressor, gas cooler, throttling device and evaporator. During the modeling, assumptions [17, 25, 26] were made such as the refrigeration cycle was in steady state and no heat was loss to the surroundings. Apart from that, all of the components in the cycle were assumed to experience negligible pressure and temperature loss. The compressor was assumed to be isentropic and the throttling device was isenthalpic.

To represent the processes in the components, thermodynamic governing equations were used. After simplification, these equations relate all the parameters through enthalpies at each component. Inside the evaporator, the refrigerant absorbs heat from the surrounding and the amount of heat can be calculated using equation 1:

$$\dot{Q}_{1-4} = \dot{m} \times (h_1 - h_4) \tag{1}$$

The enthalphy of h_1 and h_2 is in kJ/kg whereas the mass flow rate is in kg/s. Once the refrigerant exits the evaporator, it moves in to the compressor where it is compressed to superheated region. The process is isentropic thus

$$s_1 = s_2 \tag{2}$$

In the compressor, power input was required for the refrigerant to be compressed and the required value depends on the enthalpy value at evaporator exit and gas cooler inlet. The power input is obtained from equation 3 below.

$$\dot{W}_{2-1} = \dot{m} \times (h_2 - h_1) \quad (3)$$

Once the refrigerant enters the gas cooler inlet at superheated condition, heat rejection process will occur. Here, in order to do so, the refrigerant will experience large temperature glide and exits the gas cooler outlet at a slightly higher temperature. In the gas cooler, the heat rejection process occurs at constant pressure. The heat loss in this region can be quantified using:

$$\dot{Q}_{2-3} = \dot{m} \times (h_2 - h_3) \quad (4)$$

The value of h_2 is influenced by the value of s_1. However, this condition was true at isentropic efficiency of compressor at 100%. When $\eta_{is,c}$ used is not at optimum level, the value of h_2 will differ. For COP calculation, the value of h_2 used will be h_{2a} where it represents the actual enthalpy at P_2. To obtain h_{2a}, the following formula is used.

$$\eta_{is,c} = \frac{h_2 - h_1}{h_{2a} - h_1} \quad (5)$$

Then, the refrigerants enters throttling device where it was expanded and experienced adiabatic process. The enthalphy at both gas cooler exit and evaporator inlet is equal as represented below:

$$h_3 = h_4 \quad (6)$$

Enthalpy at point three or four is dependent on the quality input at point 4 or the temperature at point 3. When temperature at point 3 was assigned as input, the value of enthalpy was interpolated from the properties table and when quality at point 4 was used as the input parameter, the value was obtained by using the equation:

$$h_4 = h_{f4} + x_4(h_{g4} - h_{f4}) \quad (7)$$

The mass flow rate for refrigerant is obtained by calculating the heat transfer between evaporator and the surroundings. Here, the medium of heat transfer is air at $35°C$ with mass flow rate of 0.1611 kg/s. To obtain the mass flow rate, equation 6 is used:

$$\dot{m}_{co_2} = \frac{\dot{m}_{air} \times (h_{air,35°C} - h_{air,-13°C})}{(h_1 - h_4)} \quad (8)$$

From these equations, a thermodynamic model of carbon dioxide refrigeration cycle was integrated with a simulation model. At this stage, Microsoft Excel was utilized. Firstly base conditions were set as shown in Table 3.1.

TABLE 3.1: Base Parameters

Heat Transfer Medium	Air	
	Gas Cooler Conditions	Evaporator Conditions
Mass Flow Rate, $\dot{m}_{air}$ (kg/s) [27]	0.8889	0.1611
Inlet Temperature, $T_{air,in}$ (°C) [27]	32	32
Enthalpy,$h_{@32℃}$	305.22	305.22
Outlet Temperature, $T_{air,out}$ (°C)	42	-13
Enthalpy, $h_{@42,-13℃}$	315.27	260.09
Refrigerant	**Carbon Dioxide**	
Pressure Range (MPa)	8-13	2-6
Temperature Range (°C)	-	-15-25

3.2 Simulation Model

In the model few parameters were selected to simulate the model. The parameters were gas cooler pressure, P_2 = 12 MPa, evaporator pressure, P_1 = 4 MPa and gas cooler exit temperature, T_3 at 40°C. The properties like enthalpy value, temperature and entropy value at these parameters were obtained from NIST website at http://webbook.nist.gov/chemistry/fluid/. Figure 3.1 below illustrates the initial thermodynamic model created.

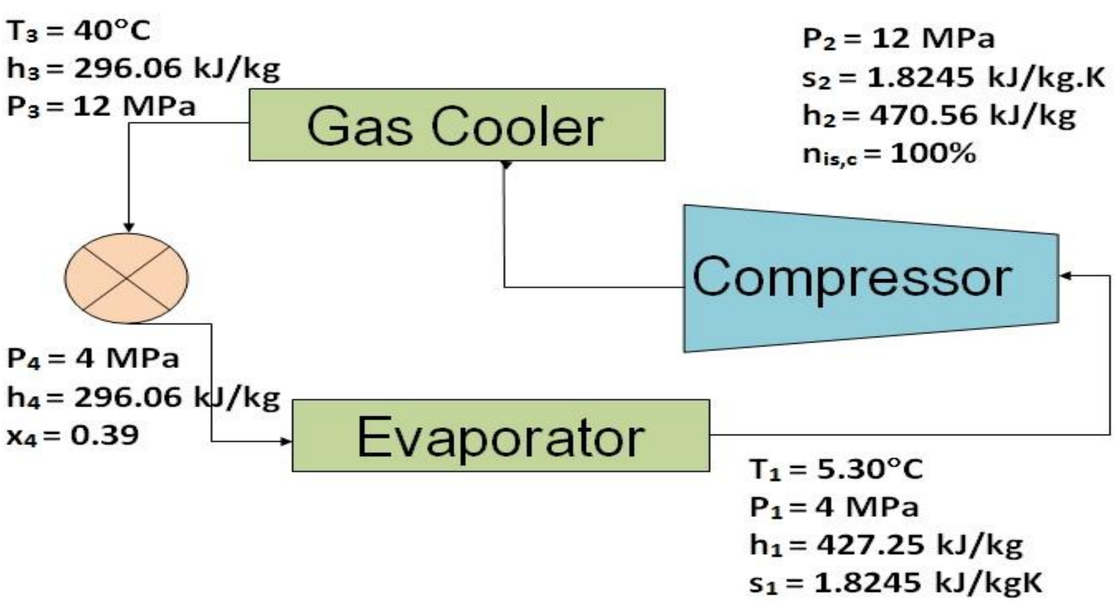


FIGURE 3.1: Schematics of The Thermodynamic Model

Based on this model, the COP was calculated using equation 9.

$$\mathrm{COP} = \frac{\dot{Q}_{1-4}}{\dot{W}_{2-1}} \qquad (9)$$

The value calculated is shown in Table 3.2.

TABLE 3.2: Heat absorbed, Work Input and COP Value

Q_L	7.27
$\dot{W}$	2.40
COP	3.03

The COP value obtained at gas cooler pressure of 12 MPa and evaporator pressure of 4 MPa was 3.03. Next, the COP value was validated using equation 10 to ensure that the cycle design was valid. The validation method was done by ensuring firstly the COP value was more than 1 and secondly by calculating the percentage error. Benchmark value was obtained from Brown (2006) which conducts similar form of analysis. The percentage error between the values must be less than 10% for the cycle design to be valid.

3.3 Parameter Variations

When the COP value was validated, the analysis proceeds by using the refrigeration cycle design to analyze the system performance and the effect of transcritical carbon dioxide refrigeration cycle parameters on the cycle performance. Cycle parameters were varied in two parts. Firstly, only one parameter was varied while others were kept constant. Parameters varied in this section were gas cooler pressure, gas cooler outlet temperature, evaporator pressure, evaporator temperature, quality and compressor efficiency. In this section, the main objective was to investigate the effect of each parameter on the system performance. Here also, the influence of each parameter on the optimum pressure was also analyzed. At this stage, once the optimum pressure for the cycle was identified, the pressure was utilized throughout the variation.

In the second part, two parameters were varied and others were kept constant to understand if the parameters combinations have significant effect on the cycle performance. The parameter combination analyzed was as follows:

1. Gas cooler pressure and gas cooler exit temperature
2. Gas cooler pressure and evaporator temperature
3. Gas cooler pressure and evaporator pressure
4. Gas cooler pressure and compressor efficiency
5. Gas cooler pressure and quality

At this stage also, the best parameters combination which gives better cycle performance was also identified. Simulation tool utilized in this project was Microsoft Excel, presented in the form of graphs and table. From the graphs, the patterns and significance of the results were discussed and analyzed in chapter 5. Figure 3.2 summarizes the methodology of this project through a flow chart.

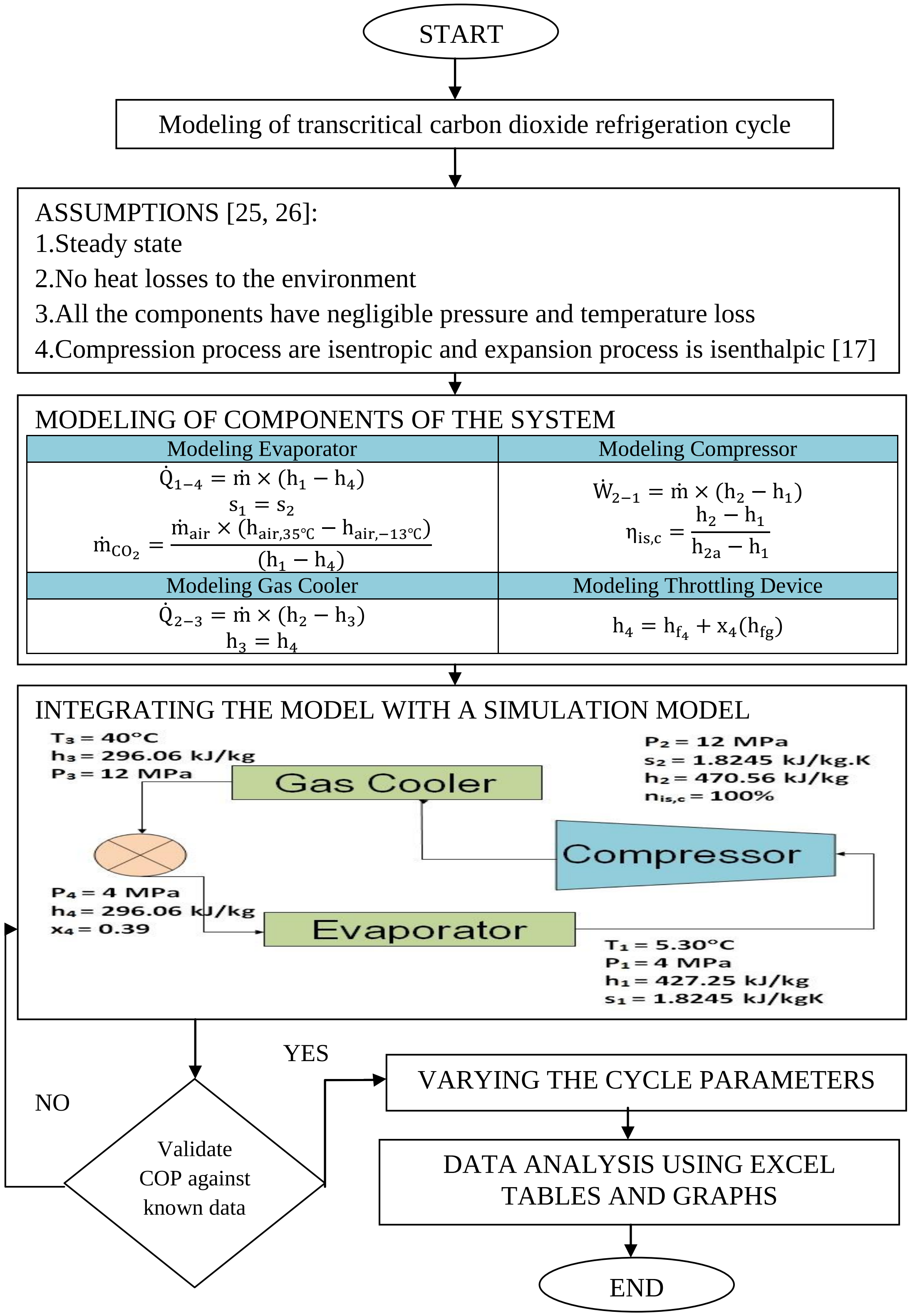


FIGURE 3.2: Methodology Flow Chart

CHAPTER 4

GANTT CHART & KEY MILESTONES

In this section, the planning for this project and the milestones achieved were outlined in Table 4.1 and Table 4.2. First half of the project focuses more on gaining information and deciding the direction of the project. The second half focuses more on running the simulation and analysis of the data.

TABLE 4.1: Gantt Chart For FYP 1 And Key Milestone

No	Detail/Week	1	2	3	4	5	6	7		8	9	10	11	12	13	14
1	**Selection of Project Topic**								MID-SEMESTER BREAK							
	a) First meeting with supervisor			√												
2	**Preliminary Research Work**															
	a) Identification of project scope of study and direction					√										
	b) Extended proposal preparation															
3	**Submission of Extended Proposal** - Submission of Form 04						√									
4	**Proposal Defense** - Submission of Form 05										√					
5	**Project Work Continues**															
	- Understanding and studying MATLAB software (heat transfer system modeling)						√									
	- Understanding the process that happens inside each refrigeration cycle components									√						
	- Modeling of CO_2 refrigeration cycle (developing equations for compressor, gas cooler)										√					
7	**Submission of Interim Report** - Submission of Form 06															√

Legend

√ - Key milestone   - Duration

TABLE 4.2: Timeline For FYP 2

No	Detail/Week	1	2	3	4	5	6	7		8	9	10	11	12	13	14	15
1	**Project Work Continues**																
	- Continuation of FYP 1 incomplete tasks																
	- Modeling of CO_2 refrigeration cycle (developing equations for expansion valve, evaporator)																
	- Integration of the model and COP validation with known CO_2 refrigeration system from journal papers																
	- Varying operation parameters																
	- Tabulating data & creating graph of data																
2	**Submission of Progress Report** - Submission of Form 07								**MID-SEMESTER BREAK**	√							
3	**Project Work Continues**																
	- Analysis of data - Identifying which parameter have the most significant impact on COP - Additional modification to data collection																
4	**Pre-SEDEX** - Preparation of poster & presentation to examiner - Submission of Form 08												√				
5	**Submission of Draft Report** - Creating a complete report and adding points suggested during pre-SEDEX - Report reviewed by supervisor													√			
6	**Submission of Dissertation (soft bound)** - Editing the report - Submission of Form 10														√		
7	**Submission of Technical Paper** - Submission of Form 09														√		
8	**Oral Presentation** - Submission of Form 11															√	
9	**Submission of Project Dissertation (Hard Bound)**																√

CHAPTER 5
RESULTS

5.1 Cycle Design Validation

Once the design was modeled, the COP value was calculated using equation 9. From research conducted by Brown (2006) [19], the COP value calculated when P_2 = 12 MPa, P_1 = 4 MPa, T_3 = 40℃ and x_4=0.4 was 2.90. The percentage error obtained was 4.5% thus the COP was validated.

$$\text{Percentage Error} = \frac{3.03\text{-}2.90}{2.90}\times 100 = 4.5\% \tag{10}$$

5.2 Parameters Variation

At this stage, various parameters input were manipulated and analyzed to understand their influence on the COP. In part 1, only one parameter was varied to see its effect on COP and then two parameters were manipulated to see the influence of its relationship on the cycle COP in part 2. Complete set of data calculated from parameters variation using Excel is attached in Appendix 1. This section illustrates the outcome and relationship of these parameters in the form of graphs.

Part 1

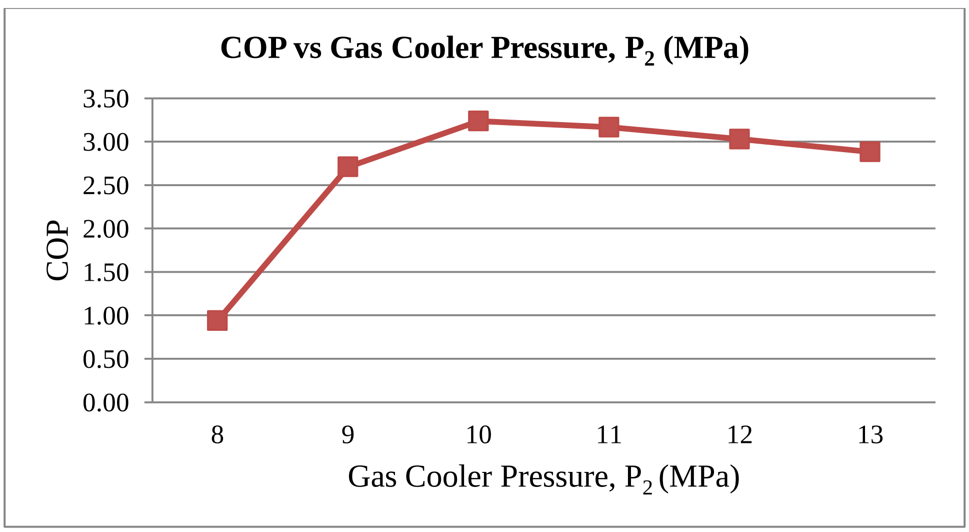


FIGURE 5.1: Variation of COP vs Gas Cooler Pressure

Figure 5.1 shows the COP vs gas cooler pressure graph. In this simulation, the input parameters were P_1 = 4 MPa, T_3 = 40℃, $\eta_{is,c}$ = 100%. The graph shows that there was an optimum pressure trend emerging whereby the COP increases as the gas cooler pressure increase, up to a certain pressure where the COP value decreased. For this cycle design, the optimum pressure was at 10 MPa where the COP was the highest at 3.24. Another pattern that can be identified from Figure 5.1 was the overall value of COP was relatively low, ranging from 0.9-3.3. Apart from that, at 8 MPa, the COP value obtained was below than 1.

The existence of optimum gas cooler pressure can be explained by the shape of the constant-temperature line. As the gas cooler exit temperature was kept constant, and the gas cooler pressure increased, the isotherm line becomes steeper thus affecting the value of enthalpy. As shown in Figure 5.1, after optimum pressure was achieved, the increase in refrigerating capacity was low compared to compression work. Based on equation 9, this reduces the COP value.

Average COP value for the design cycle was relatively low compare to previous research done by Brown (2006) [19] as in this simulation, the input parameter of T_3 was used instead of the x_4. In Brown (2006) [19], the quality was set at 0.2 whereas by selecting gas cooler exit temperature at 40^oC, the quality was higher averaging at 0.5 thus resulting in the lower value of COP. At 8 MPa, the COP value was below than 1 because when calculated, the quality at point 4 was 0.89 resulting in smaller refrigeration capacity at 8 MPa compare to the compressor work. This shows that, in order to improve COP value at the selected input parameters, gas cooler pressure should not be too close to the critical pressure.

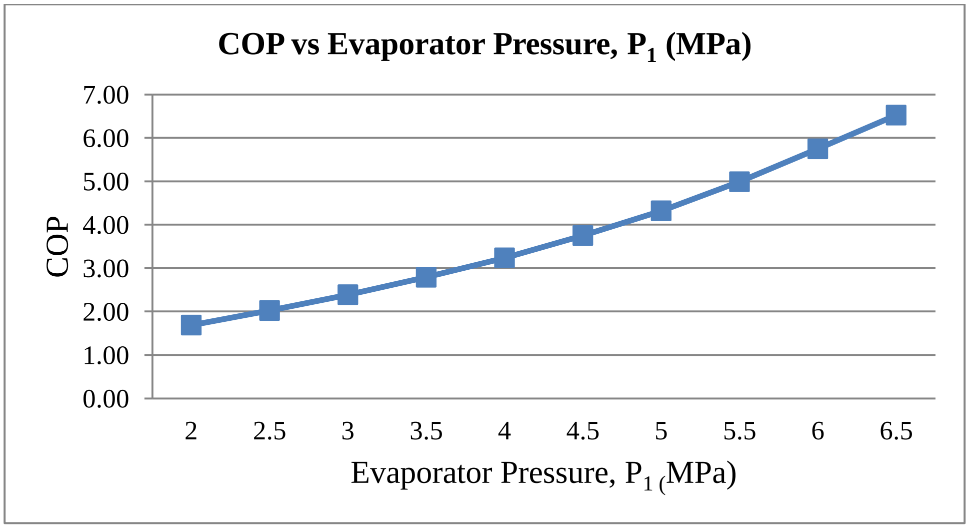


FIGURE 5.2: Variation of COP vs Evaporator Pressure

The COP vs evaporator pressure graph was shown in Figure 5.2 where the input parameters were selected from P_2 = 10 MPa, T_3 = 40^{o}C, $\eta_{is,c}$ = 100%. It was observed that the COP value increased almost linearly as the evaporator pressure approaches the critical pressure of carbon dioxide. As the evaporator pressure increases, the difference between enthalpy values at point 1 and 2 decreases. This was due to the fact on the p-h diagram, the enthalpies became closer and the entropy line becoming more vertical. According to equation 3, the work input also decreases as the pressure increased. Less work input to the system, the COP value increases according to equation 9.

Apart from that, no optimum pressure trend was observed when evaporator pressure was varied. Here, the pressure 10 MPa was selected as in the previous simulation the pressure was the optimum pressure for the cycle. The simulation also only varied the evaporator pressure up to 6.5 MPa as the quality value was zero at this pressure thus making the cycle invalid. Thus, this analysis shows that if the performance of carbon dioxide refrigeration cycle were to be improved, a higher value of evaporator pressure and closer to the critical pressure should be used.

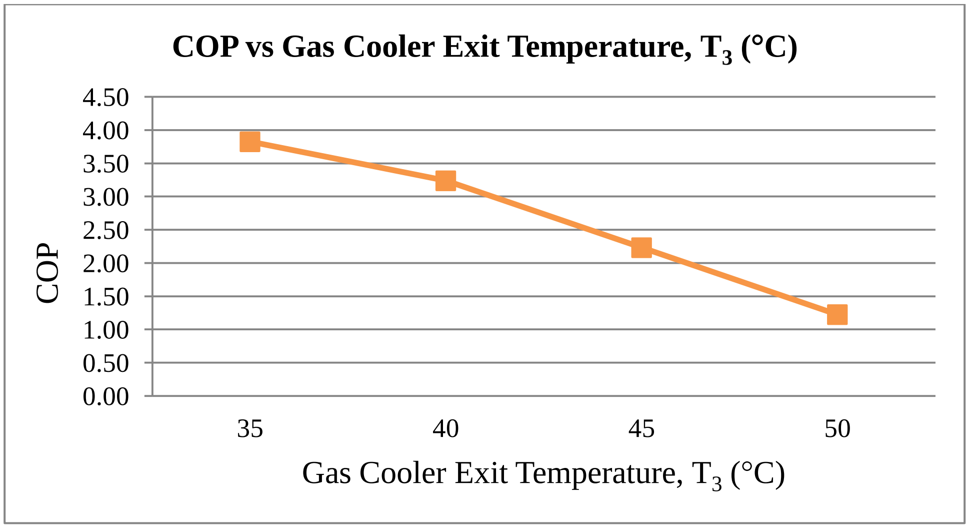


FIGURE 5.3: Variation of COP vs Gas Cooler Exit Temperature

In Figure 5.3, the COP vs gas cooler exit temperature graph shows that at input of P_1 = 4 MPa and P_2 = 10 MPa, as the gas cooler exit temperature increased, the COP decreased. A linear relationship can be observed between the COP and the gas cooler exit temperature. The highest value of COP = 3.82 was at 35 ^{o}C. At temperature more than 50^{o}C, the COP value decrease to a value less than zero thus omitted from the analysis as it indicates the cycle have failed to provide refrigeration. From Figure 5.3 also, it was observed that at lower gas cooler exit temperature the COP value was higher.

When the temperature elevates, the enthalpy value at point 3 increases thus reducing the differences between h_1 and h_4. For example at 35^{o}C, the enthalpy difference was 134.92 kJ/kg as compared to 114.21 kJ/kg at 40 ^{o}C. The refrigerant mass flow rate also increases thus increasing the compressor work input. From equation 9, it shows that when compressor work increases, the COP value will decrease. Moreover as T_3 increases, the temperature glide in the gas cooler becomes smaller thus increasing the quality value at point 4. Thus, from the graph, it can be deduced that a lower exit temperature was desired for COP value improvement.

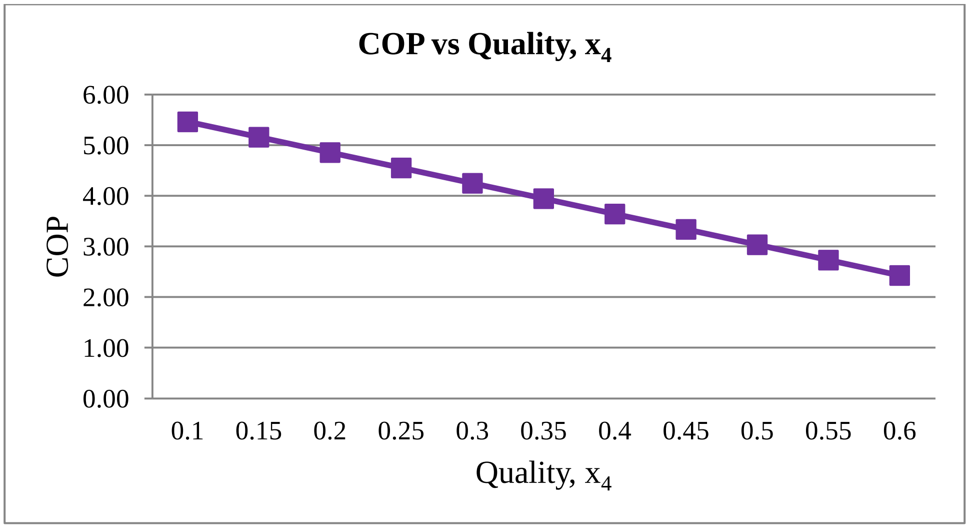


FIGURE 5.4: Variation of COP vs Quality

The relationship between COP and quality at point 4 in the cycle was illustrated in Figure 5.4. The input parameters were set at P_1 = 4 MPa, P_2 = 10 MPa and compressor isentropic efficiency at 100%. This shows that as the quality value increased with an increment of 0.05, the COP value decreases at an increment of 0.3. No optimum pressure was observed in the graph shown in Figure 5.4. Another noteworthy observation was, at lower value of quality, the temperature falls below the critical temperature of carbon dioxide as shown in Table 5.1. At quality above 0.3, however the temperature exceeds the critical temperature of carbon dioxide.

TABLE 5.1: Temperature at Gas Cooler Exit

Quality, x_4	Gas Cooler Evaporator Temperature, T_3 (^oC)
0.1	17
0.15	21
0.2	25
0.25	28
0.3	31

Firstly, the COP decreases as the quality increases in the cycle because, the enthalpy difference between h_1 and h_4 decreases thus reducing the refrigerating capacity of the cycle. As the refrigerating capacity decreases, the mass flow rate increases, thus increasing the value of work input into the compressor. Thus from equation 9, it was shown that the COP value decrease. Increasing quality at point 4 also increases the amount of moisture content in the system which requires larger work input to the compressor.

Apart from that, at 10 MPa, specifically for this cycle design, it was advisable that, the quality value exceeds 0.3 as below this value, the exit temperature falls below the critical temperature of carbon dioxide. The focus of this project was to analyze transcritical refrigeration cycle thus only quality (0.35-0.6) with exit temperature above critical point was selected.

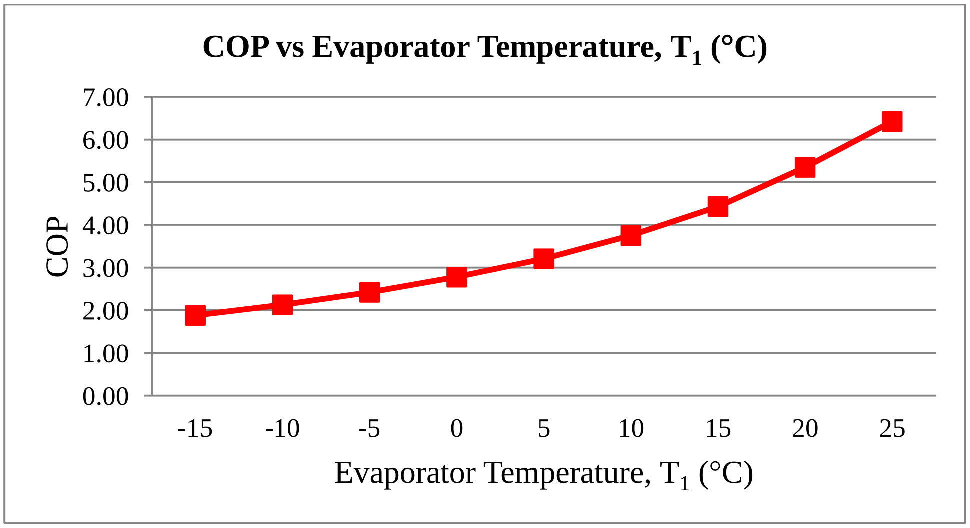


FIGURE 5.5: Variation of COP vs Evaporator Temperature

As Figure 5.5 shows, when the evaporator temperature increases, the COP value of the cycle increased. This corresponds to the relationship between COP and the evaporator pressure presented in Figure 5.2. As the evaporator temperature approaches the critical point, the cooling performance for carbon dioxide refrigeration cycle increases. Here, the input parameters were at P_2 = 10 MPa, T_3 = 40^{0}C and at isentropic efficiency of 100%. The outcome of this simulation also supports the findings in Figure 5.4 where the quality ranges from 0.3 to 0.6 for gas cooler exit temperature of 40^{o}C. From this graph also, the application for carbon dioxide can be deduced. That is heat pump (10^{o}C-25^{o}C) will mostly benefit from the high COP value rather than for freezing purposes (0^{o}C-(-10^{o}C)) where the value of COP was relatively lower.

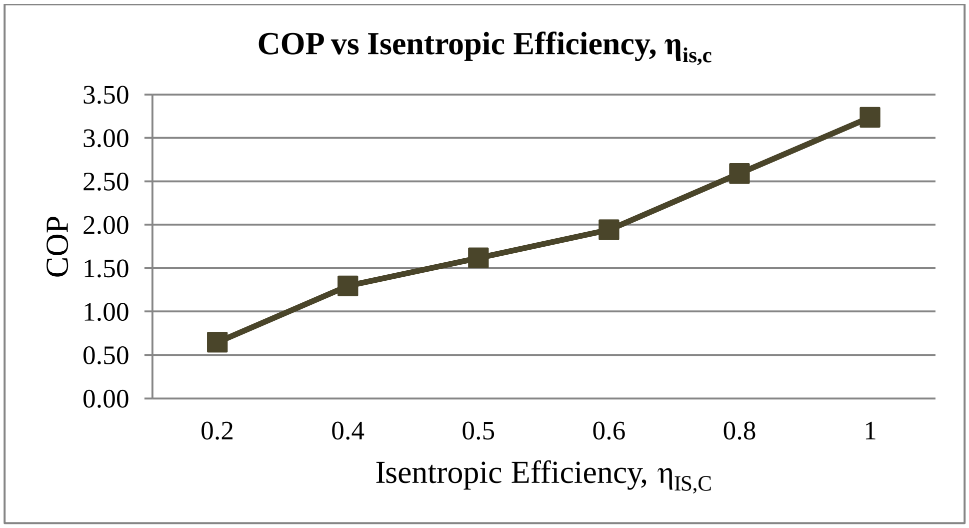


FIGURE 5.6: Variation of COP vs Isentropic Efficiency

In Figure 5.6, the graph shows the relationship between COP and compressor isentropic efficiency. As the compressor efficiency improves, the COP value also increases. With better compressor performance, the work input required by the system decreases thus according to equation 9 increases the value of COP. This simulation uses P_1 = 4 MPa, P_2 = 10 MPa with T_3 = 40℃ as the input parameters. From this calculation also, when the compressor efficiency was the lowest, $\eta_{is,c}$ = 0.2, the COP value falls below 1. At this compressor efficiency also, the actual enthalpy and temperature was very high at h_{2a} = 603.63 kJ/kg and T_{2a} = 130^{o}C as compared to the isentropic properties value. This shows that, high temperature at gas cooler inlet is not a favourable property to the COP.

From the simulation outlined in part 1, a few points can be summarized. Firstly, optimum pressure trend exists when gas cooler pressure is varied. Secondly, for evaporator pressure, evaporator temperature and gas cooler exit temperature as these properties approaches the critical point, the COP value becomes larger although this relation does not apply to gas cooler pressure. The COP value is lower when gas cooler pressure is closer to critical pressure. For isentropic efficiency of the compressor, as the efficiency improves, the COP value also increased. The quality value however, as it increased, reduced the value of COP significantly. For this specific cycle design also, only certain value of quality can be used in order to maintain a transcritical refrigeration cycle.

Part 2

In this section, two parameters were varied to understand the effect on carbon dioxide transcritical refrigeration cycle. Some of the observations expected to be seen in this section was the effect of two parameters on the COP as well as on the optimum pressure value. From this section also, parameters that have the most significant on the COP value was analyzed and identified.

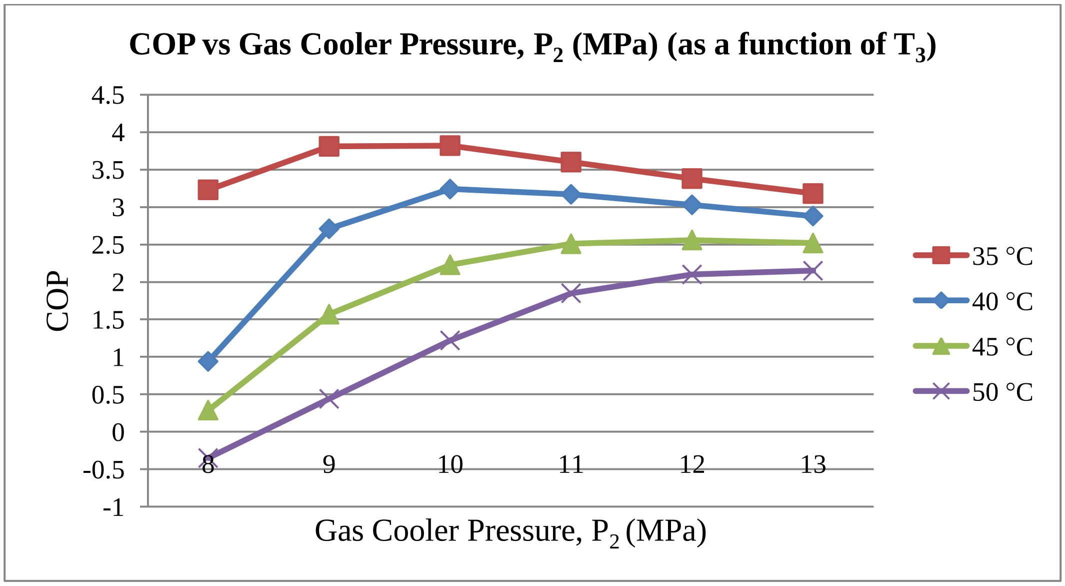


FIGURE 5.7: Variation of COP vs Gas Cooler Exit Temperature (as a function of P_2)

From Figure 5.7, The COP value was plotted against the gas cooler pressure as a function of gas cooler exit temperature. The input parameter used was the evaporator pressure, P_1 = 4 MPa. In this simulation, two parameters, gas cooler exit temperature and gas cooler exit pressure were varied to see the effect of these parameters combination on the COP. Firstly, the results obtained in this simulation supported the results from Figure 5.3, whereby lower exit temperature has better COP than higher temperature.

Apart from that, from this graph a distinct optimum pressure trend was observed at temperature of 35^{o}C and 40^{o}C. As the gas cooler exit temperature increases, the optimum pressure also increases. For example at 35^{o}C and 40^{o}C, the highest COP value was recorded at optimum pressure of 10 MPa. However, at temperature of 45^{o}C, a less distinct optimum pressure trend was observed. At this temperature, the COP value rises until a maximum value was achieved and then decrease in small increments. For example, at 45^{o}C, the COP value increased up to 2.56 at 12 MPa and decrease to 2.52 at

13 MPa. As for $50^{o}C$, no optimum pressure trend was identified within the pressure range analyzed however the highest COP recorded for the pressure were 2.15 at 13 MPa.

Here it can be deduced that gas cooler exit temperature has a significant effect on the gas cooler optimum pressure. This supports the various outcome of researches conducted previously by other researchers. Outcome of this simulation can be explained by the isotherm line existing above the critical point. At higher evaporator temperature, the enthalpy difference was bigger. Thus it would take higher pressure for the work input to overcome the refrigerating capacity of the system to achieve optimum pressure at higher temperatures.

Apart from that, it was also observed that at 8 MPa, the COP value was the lowest and in agreement with the results illustrated in Figure 5.1. At pressure 8 MPa also, at higher temperature, the COP value was very small (below 1). However, at $35^{o}C$, the value suddenly increased to 3.23. This was due to the effect of enthalpy value at h_4. At higher temperature, the value was bigger compared to enthalpy at $35^{o}C$, thus enthalpy difference at refrigerating capacity was smaller as the temperature increases. This reduces the refrigerating capacity significantly at higher temperature compare to at $35^{o}C$.

Gas cooler exit temperature at higher temperature also shows that at pressures higher than 8 MPa, the COP value increased sharply. For example, at $40^{o}C$ and $45^{o}C$, the value hiked from 0.94 to 2.71 and 0.29 to 1.57 respectively at 9 MPa. The value of COP at $35^{o}C$ however, increased gradually with smaller increments over the pressures as compared to other temperatures. Negative value of COP was also observed at $55^{o}C$ and 8 MPa which shows that the cycle has failed to provide refrigeration.

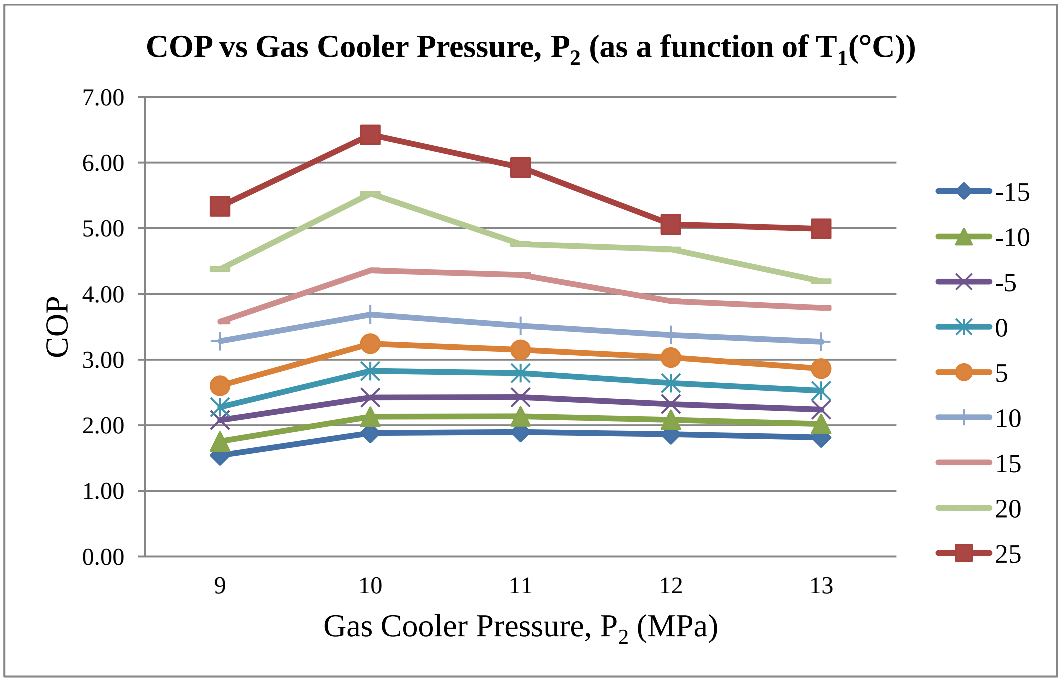


FIGURE 5.8: Variation of COP vs Gas Cooler Pressure (as a function of T_1)

In Figure 5.8, the COP value is plotted against the gas cooler pressure as a function of evaporator temperature. The input value used in this simulation cycle was the isentropic efficiency was at 100% and the gas cooler exit temperature set at 40°C. At temperature higher and closer to the critical point of carbon dioxide, the COP value was bigger. This supports the finding discovered in Figure 5.5. Apart from that, by varying the gas cooler pressure, optimum pressure for the cycle was observed although the pattern was more distinct at higher evaporator temperature especially at 5^{o}C and above. Starting at 0^{o}C and below, the optimum pressure trend was less distinct as the COP increments as the pressure increased becomes smaller.

Figure 5.8 also illustrates that at a lower evaporator temperature (-15^{o}C-(-5^{o}C)), the maximum COP value happens at higher pressure that was 11 MPa, although at higher evaporator temperature (0^{o}C-25^{0}C), the maximum COP value happens at 10 MPa. Previous research discussed that at lower evaporator temperature, the optimum pressure at gas cooler should be higher however this characteristic was influenced by a constant gas cooler exit temperature. This simulation proofs the findings as Figure 5.8 simulation was conducted with the gas cooler exit temperature set to a constant value. Findings in this simulation also prove that gas cooler exit temperature has a more significant effect on the COP compare to the evaporator temperature.

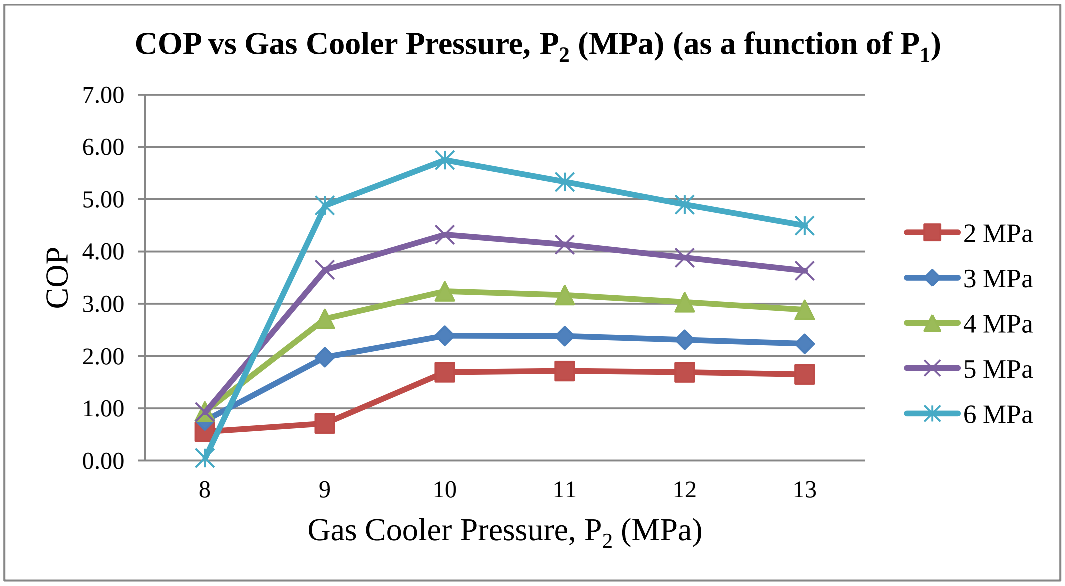


FIGURE 5.9: Variation of COP vs Gas Cooler Pressure (as a function of P_1)

The graph shown in Figure 5.9 illustrates the relationship between the COP and gas cooler pressure as a function of evaporator pressure. In this simulation, the input parameters were the gas cooler exit temperature which was set at 40^0C and $\eta_{is,c}$=100%. From the graph, it shows that at higher P_1, the COP of the cycle reached up to 5.75 and an optimum pressure value was identified at 10 MPa. The outcome of these parameter simulations also was in agreement with Figure 5.2 whereby higher evaporator pressure gives better COP.

At 8 MPa, it was observed that the COP value calculated was the lowest among the other pressures which agrees with previous simulations results. One anomaly identified also was at gas cooler pressure of 8 MPa and evaporator pressure of 6 MPa, the COP value falls to 0.05. The reason for this outcome was the refrigerating capacity was too small which resulted from the small enthalpy value difference (0.4kJ/kg). This produce large mass flow rate of carbon dioxide in the evaporator thus creating larger work input requirement to the compressor. Thus according to equation 9, the COP value becomes smaller.

Here, it shows that for this design cycle, when both gas cooler and evaporator pressure become too close to the critical point, the refrigeration system fails. In order for the system to increase its COP at 6 MPa and 8 MPa, the gas cooler exit temperature must at least be higher than 40°C. This will give higher enthalpy value at h_4, improving the refrigeration capacity and thus improving the COP.

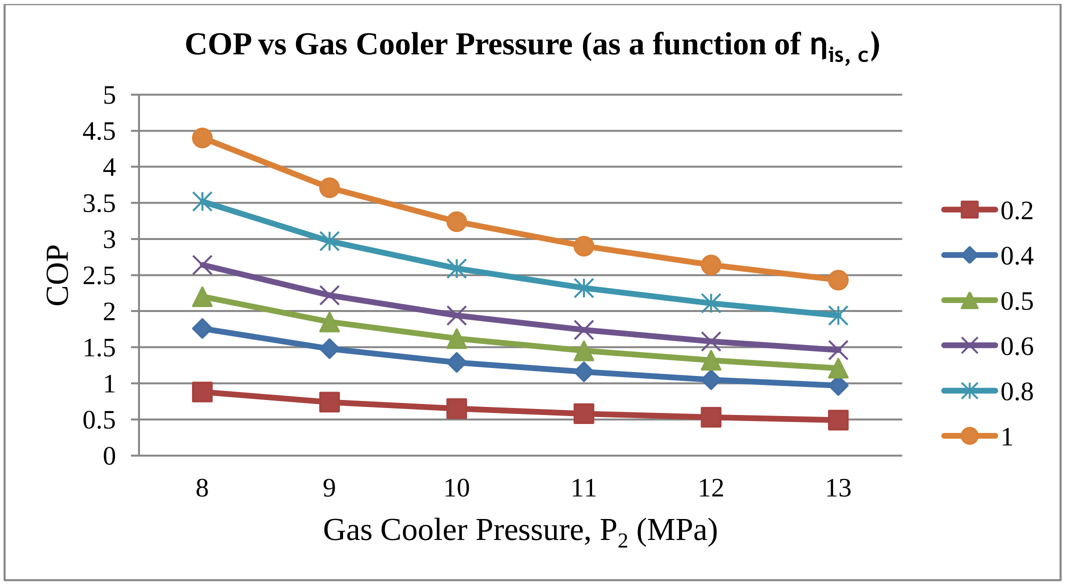


FIGURE 5.10: Variation of COP vs Gas Cooler Pressure (as a function $\eta_{is,c}$)

In Figure 5.10, COP was plotted against the gas cooler pressure as a function of isentropic efficiency of the compressor. The simulation was conducted by using input parameters of gas cooler temperature at 40°C and evaporator pressure, P_1 set at 4 MPa. At the highest compressor efficiency, the COP value was the highest and lowest at compressor efficiency of 0.2. This result corresponds to the findings discovered in Figure 5.6. Apart from that, no optimum pressure trend was observed when these two parameters were varied. For every gas cooler pressure also, the COP value doubled at each $\eta_{is,c.}$ This indicates that isentropic efficiency of compressor has no significant impact on the gas cooler optimum discharge pressure.

Contrary to previous results, the highest value of COP recorded was at gas cooler pressure of 8 MPa and the lowest was at 13 MPa for all efficiency value. This can be explained by the enthalpy value at the gas cooler pressures. As the pressure increases, the actual enthalpy value, h_{2a} also increases. This increases the work input value to the compressor as the gas cooler pressure increases. Thus according to equation 9, as the evaporator heat absorption was kept constant and the work input to the compressor increased, the COP value of the system will decrease. This also proves that the COP value for the cycle was dependent on the compressor efficiency although the significance of its effect depends on the gas cooler pressure requirements.

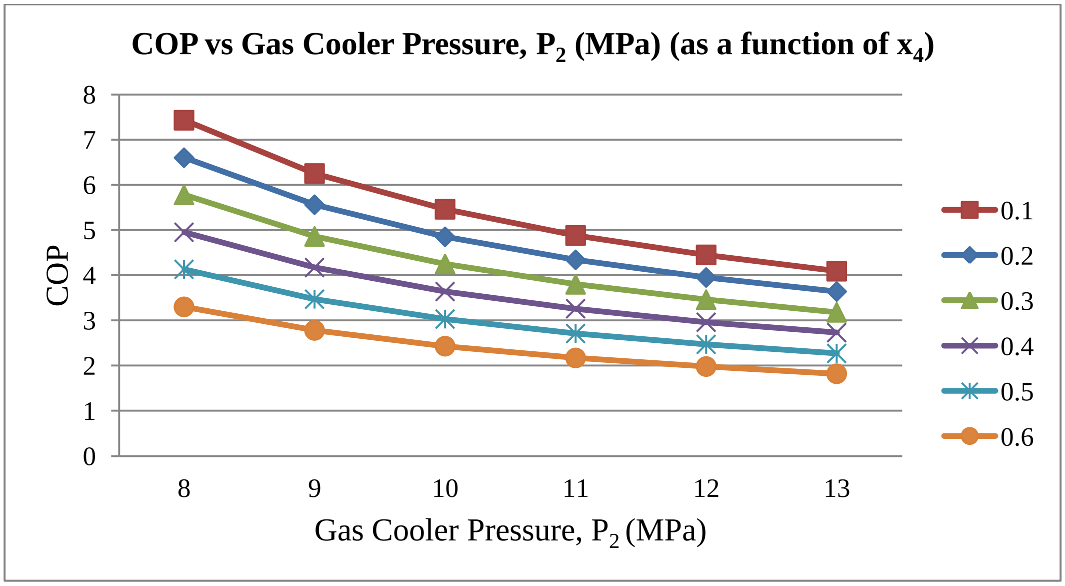


FIGURE 5.11: Variation of COP vs Gas Cooler Pressure (as a function of x_4)

Last but not least, the graph in Figure 5.11 illustrates the COP vs gas cooler pressure graph as a function of quality. In this simulation, the evaporator pressure was set at P_1 = 4 MPa, Quality in this simulation was varied at an increment of 0.1 and pressure was varied at an increment of 1 MPa. The outcome of this result was that at lower quality value, the COP was higher. This was because, at lower quality value, the refrigerating capacity or $\dot{Q}_{1\text{-}4}$ was larger compare to the work input in the compressor. Thus, according to equation 9, the COP was at a higher value.

In this simulation, no optimum pressure trend was identified although quality value determines the gas cooler exit temperature according to the model. Previous simulations have shown that gas cooler exit temperature influenced the optimum pressure of the system. This shows that quality value at evaporator inlet have no significant effect on the gas cooler optimum discharge pressure. Moreover, one pattern identified from the graph was the increment value of COP was similar at every quality value. For example, when quality was increased from 0.2 to 0.4, the COP value double from the value obtained when quality was increased by 0.1.

Apart from that, highest COP value obtained among the pressures was identified at 8 MPa with values ranging from 3.30-7.43. This outcome differs from the findings in previous simulations. Despite outstanding values of COP, further calculations shows that the temperature at these quality points fall below the critical temperature of the cycle as shown in Table 5.2.

TABLE 5.2: Temperature Value at Gas Cooler Exit

	8	9	10	11	12	13
0.1	15	16	17	17	18	18
0.2	22	24	25	26	27	27
0.3	28	30	31	33	35	35
0.4	31	33	37	39	42	42
0.5	33	37	41	43	49	49
0.6	35	40	44	48	51	53

Results obtained also shows that, for this cycle design, the suitable quality value must be 0.3 and above to ensure the cycle remains a transcritical cycle. Table 5.2 also shows that as the gas cooler pressure increases, the quality value at evaporator inlet decreases in order to achieve above critical gas cooler exit temperature. This outcome was undesirable as the main focus of this paper was to analyze transcritical carbon dioxide refrigeration cycle not subcritical cycles.

In a summary, from these simulations, a few characteristics of the cycle were identified. Firstly, the gas cooler pressure, gas cooler exit temperature and evaporator temperature parameter was found to have a more significant effect on carbon dioxide transcritical refrigeration cycle optimum pressure. Secondly, at higher gas cooler exit temperature, the optimum pressure obtained was higher. Evaporator temperature also influenced the existence of optimum pressure at higher temperature ranges compare to below freezing temperature range. COP values were affected by most parameters especially the gas cooler pressure, gas cooler exit temperature and evaporator temperature and pressure. Higher evaporator pressures and evaporator temperatures gives better value of COP. To obtain better COP, having gas cooler pressure and evaporator pressure was not advised. However, the outcome for gas cooler exit temperature was the opposite as lower temperature gives better COP value. Moreover, parameters combination of gas cooler with compressor isentropic efficiency and quality gives better COP at lower pressure value. However, these two parameters also has less significant effect on the cycle COP.

CHAPTER 6

CONCLUSION

As research uncovers most conventional refrigerant used today contributes to earth global warming potential, researchers shift their focus on finding substitutes for these refrigerants. One refrigerant in particular was carbon dioxide where it has negligible GWP, non-toxic, non-flammable and highly abundant in the environment. Carbon dioxide also was discovered to have excellent heat transfer properties and high volumetric capacity which enables it to transfer heat effectively. However, it was discovered that, despite these excellent properties, the refrigeration cycle faced performance issues.

Thus, throughout this project, carbon dioxide transcritical refrigeration cycle performance was successfully understood, investigated and validated through literature review and Microsoft Excel simulation. The approach used was to create a model of transcritical refrigeration cycle and simulating it according to the set parameters. Refrigeration cycle modeling was conducted using a mathematical model with set base parameters as the guideline. The cycle consists of four processes that were quantified in the form of equations as explained in Chapter 3. Based on the designed carbon dioxide refrigeration cycle, the value of COP obtained was more than 1, at 3.03 to be exact. This value was validated using a research paper published by Brown (2006) which utilized similar analysis method to ensure that the cycle was feasible. The percentage error obtained was 4.5%, thus the cycle was validated.

Once the model was validated with other known model, selected parameters were then analyzed and compared to see which parameter have the most significant impact on the system COP. In this section, the simulation was divided into two parts where the first part analyzes the parameters by varying only one variable with respect to the COP value. The purpose was to understand the relationship between individual parameters to the cycle COP. Through the second part, two variables were manipulated to understand the relationship between each parameters and its influence on the COP. At this stage, the parameters combinations which affect the refrigeration cycle cooling performance significantly were identified.

Based on simulation results, few characteristics of the cycle was identified. Firstly, COP for carbon dioxide transcritical refrigeration cycle has an almost linear relationship with most of its cycle parameters. Compare to conventional refrigeration cycle also, transcritical CO_2 refrigeration cycle also has an optimum pressure where the maximum COP value was obtained. This pressure was not constant and varies according to the cycle parameters input. From the simulations also, gas cooler exit temperature, gas cooler pressure and evaporator temperature was identified as the parameters that has a significant effect on the cycle optimum pressure and COP value. Parameters such as the compressor isentropic efficiency, quality and evaporator pressure has less significant effect compare to the others.

Moreover, the best combinations of these parameters were identified based on the cycle results. Firstly, for the cycle design, lower gas cooler exit temperature, gives better COP and lower optimum pressure value. For evaporator temperature however, higher evaporator temperature gives higher COP and lower pressure value. With lower pressure value, problems such as containment issues can be eliminated. Apart from that, it was also observed that, both gas cooler pressure and evaporator pressure cannot be too close to the critical properties at the same time as it lowers the COP value of the refrigeration cycle. Thus, the ability to control the gas cooler pressure, gas cooler exit temperature, evaporator temperature as well as the ability to finding a balance between each parameter combinations will give better COP as well as lower optimum pressures.

Here, it is recommended that in the future more research can be conducted to provide better understanding about the cycle. One analysis that can be conducted was more experimental based research to validate the data and information obtained from the simulations. Moreover, it is hoped more research that can derive accurate equations to quantify and explain the effect of parameters on COP and optimum pressure can be conducted. These research data also should be validated by experiments to ensure its accuracy. Based on these outcomes, it is hoped that a better understanding of controlling carbon dioxide transcritical refrigeration cycle COP can be achieved. Apart from that, with the identification of the parameters that affect the COP significantly, it is hoped that future design of CO_2 refrigeration cycle can be improved.

REFERENCES

[1] New Oxford American Dictionary, 2010, "refrigerant," In Stevenson, A., and Lindberg, C.(Eds.) : Oxford University Press. Retrieved from http://www.oxfordreference.com/view/10.1093/acref/9780195392883.001.0001/m_en_us1283874 on 22 Jun 2013.

[2] Brown, J.S. and Domanski, P.A., 2000, "Semi-theoretical Simulation Model for a Transcritical Carbon Dioxide Mobile A/C System," at *SAE Technical Paper Series 2000-01-0989*. Detroit; Society of Automotive Engineers, Inc. (ASE).

[3] Robinson, D. M. and Groll, E. A., 1996, "Using Carbon Dioxide in a Transcritical Vapor Compression Refrigeration Cycle," at *International Refrigeration and Air Conditioning Conference 345*.

[4] Cavallini, A. 2004, "Properties of CO2 as a refrigerant," In *European Seminar-CO2 as a Refrigerant, Mediolan*.

[5] Perez-Garcia, V., Belman-Flores, J-M., Navarro-Esbri, J. and Rubio-Maya, C., 2012, "Comparative study of transcritical vapor compression configurations using CO_2 as refrigeration mode base on simulation," *Applied Thermal Engineering.* **51**. 1038-1046.

[6] Kim, M-H., Pettersen, J. and Bullard, C-W., 2004, "Fundamental process and system design issues in CO_2 vapor compression systems," *Progress in Energy and Combusion Sciences* **30**: 119-174.

[7] Sarkar, J., 2010, "Review On Cycle Modifications of Transcritical CO_2 Refrigeration and Heat Pump Systems," *Journal of Advanced Research in Mechanical Engineering.* **1**. 22-29.

[8] Ma, Y., Liu, Z., and Hua, T., 2012, "A review of transcritical carbon dioxide heat pump and refrigeration cycles," *Energy.* **55**. 156-172.

[9] Neksa, P., Walnum, H. T., and Hafner, A., 2010, "CO_2 – A Refrigerant From The Past With Prospects Of Being One Of The Main Refrigerants In The Future," Retrieved from http://www.sintef.no/project/CREATIV/Publications/Journal%20and%20conference/D4.1.6%20%20Keynote_GL2010_NeksaWalnumHafner-CO2-Arefrigerantfromthepastwithprospectsofbeing%5B1%5D.pdf on 26 June 2013.

[10] Riffat, S. B., Afonso, C. F., Oliveira, A.C., and Reay, D. A., 1996, "Natural Refrigerants For Refrigeration And Air-Conditioning Systems," *Applied Thermal Energy*. **17.** 33-42.

[11] Danfoss A/S., 2008, "Transcritical Refrigeration Systems with Carbon Dioxide (CO_2) How to design and operate a small-capacity (<10 kW) transcritical CO_2 system," Retrieved from http://www.danfoss.com/North_America/NewsAndEvents/Archive/Company+News/2008/How-to-Design-and-Operate-a-Small-Capacity-Transcritical-CO2-System/621BC452-473B-4998-990B-837AA92168B4.html on 26 June 2013.

[12] Nordic Chemical Group (NKG)., 2009, "The transcritical CO_2 Cycle Fact sheet 2.2.4," *Natural Refrigerants for new Application.* (Publication No. AIP 2009:426). Copenhagen K: Denmark

[13] Wang, S., Tuo, H., Cao, F., and Xing, Z., 2013, "Experimental investigation on air-source transcritical CO_2 heat pump water heater system at a fixed water inlet temperature," *International Journal of Refrigeration.* **36.** 701-716.

[14] Pietro, A., 2005, "Multi-Scale Analysis of Heat and Mass Transfer in Mini/Micro Structures," Ph. D. Thesis, Politecnico di Torino, Italy.

[15] Sarkar, J., Bhattacharyya, S., and Ram Gopal, M., 2004, "Optimization of a transcritical CO_2 heat pump cycle for simulateneous cooling and heating applications", *International Journal of Refrigeration.* **27**. 830-838.

[16] McGee, M., 2013, "Earth's CO_2 Home Page," Retrieved from http://co2now.org/ on 20 June 2013.

[17] Sarkar, J. and Bhattacharyya, S., 2008, "Optimization of a transcritical N_2O refrigeration/heat pump cycle," at *8th IIR Gustav Lorentzen Conference on Natural Working Fluids,* Copenhagen.

[18] Brown, J. S., Yana-Motta, S. F., and Domanski, P. A., 2000, "Comparitive analysis of an automotive air conditioning systems operating with CO2 and R134a," International Journal of Refrigeration. 25. 19-32.

[19] Brown, M. 2006,"Simulations for thermodynamic analyses of transcritical carbon dioxide refrigeration cycle and reheat dehumidification air conditioning cycle," *Graduate School Theses and Dissertations.* Retrieved from http://scholarcommons.usf.edu/etd/3876 on 11 June 2013.

[20] Yamasaki, H., Yamanaka, M., Matsumoto, K., and Shimada, G., 2004, Introduction of transcritical refrigeration cycle utilizing CO_2 as working fluid," *International Compressor Engineering Conference.* Purdue.

[21] Cabello, R., Sanchez, D., Llopis, R., and Torella, E., 2008, "Experimental evaluation of the energy efficiency of a CO_2 refrigerating plant working in transcritical conditions," *Applied Thermal Engineering.* **28**. 1596-1604.

[22] Xue, J., Koyama, S., and Kuwahara, K. (Eds.), 2010, Proceedings from 2010 International Symposium on Next-generation Air Conditioning and Refrigeration Technology: *Performance Prediction of A R744 Transcritical Cycle for Air Conditioning*. Tokyo: Japan.

[23] McEnaney, R.P., Yin, J. M., Bullard, C.W., and Hrnjak, P.S. 1999, "An Investigation of Control-Related Issues In Transcritical R744 and Subcritical 134a Mobila Air Conditioning Systems," Retrieved from https://www.ideals.illinois.edu/bitstream/handle/2142/13326/CR019.pdf?sequence=2 on 27 June 2013.

[24] Chen, Y., 2011, "Thermodynamic Cycles using Carbon Dioxide as Working Flid," Retrieved from http://www.diva-portal.org/smash/get/diva2:461426/FULLTEXT01 on 25 June 2013.

[25] Fourie, M. and Dobson, RT., 2011, "Theoretical Simulation and Experimental Validation Of A Critical And Transcritical CO_2 Refrigeration System Using A Dual Capillary Tube Expansion Device," Retrieved from http://www.crses.sun.ac.za/files/services/conferences/annual-student-symposium-2011/9-17nov_fourieM.pdf on 27 June 2013.

[26] Baek, J. S., Groll, E. A., and Lawless, P.B., 2002, "Effect of Pressure Ratios Across Compressor On The Performance Of The Transcritical Carbon Dioxide Cycle With Two-Stage Compression and Intercooling," *International Refrigeration and Air Conditioning Conference.* Paper 583. Retrieved from http://docs.lib.purdue.edu/iracc/583 on 27 June 2013.

APPENDICES

Calculation Tables

1) Thermodyamic Modeling Calculations

TABLE 3: Input Conditions

Gas Cooler	T_3	40	°C
Evaporator	P_1	4	Mpa
	h_1	427.25	kJ/kg
	$s_1 = s_2$	1.8145	kJ/kgK
	$\dot{m}_{air}$	0.1611	kg/s
Compressor	η	100	%

TABLE 4: COP Calculation For Thermodynamic Modeling

Gas Cooler Pressure, P_2 (Mpa)	Gas Cooler Temperature, T_2 (° C)	Gas Cooler Enthalpy, h_2 (kJ/kg)	Enthalpy P_3, h_3 $h_4 = h_3$ (kJ/kg)	Mass Flow Rate CO_2, $\dot{m}$ (kg/s)	Heat rejected, $Q_{1\text{-}4}$ (kJ/s)	Work input to compressor, $\dot{W}_{2\text{-}1}$ (kJ/s)	COP	Quality, x_4
11	83.73	466.69	302.38	0.0582	7.27	2.30	3.17	0.42
11.5	87.41	468.66	298.9	0.0566	7.27	2.35	3.10	0.40
12	90.86	470.56	296.06	0.0554	7.27	2.40	3.03	0.39
12.5	94.41	472.48	293.67	0.0544	7.27	2.46	2.95	0.38
13	97.68	474.30	291.61	0.0536	7.27	2.52	2.88	0.37

2) COP vs Gas cooler Pressure Calculations

TABLE 5: COP vs Gas Cooler Pressure Calculations

Gas Cooler Pressure, P_2 (Mpa)	**Gas Cooler Temperature, T_2 (° C)**	**Gas Cooler Enthalpy, h_2 (kJ/kg)**	**Enthalpy P_3, h_3 $h_4 = h_3$ (kJ/kg)**	**Mass Flow Rate CO_2, $\dot{m}$ (kg/s)**	**Heat absorbed, Q_{1-4} (kJ/s)**	**Work input to compressor, $\dot{W}_{2-1}$ (kJ/s)**	**COP**	**Quality, x_4**
8	57.73	453.18	402.9	0.2986	7.27	7.74	0.94	0.89
9	67.24	458.05	343.78	0.0871	7.27	2.68	2.71	0.61
10	75.88	462.53	313.04	0.0637	7.27	2.25	3.24	0.47
11	83.73	466.69	302.38	0.0582	7.27	2.30	3.17	0.42
12	90.86	470.56	296.06	0.0554	7.27	2.40	3.03	0.39
13	97.68	474.30	291.61	0.0536	7.27	2.52	2.88	0.37

3) COP vs Evaporator Pressure Calculations

TABLE 6: COP vs Evaporator Pressure Calculations Table

Evaporator Pressure, P_1 (Mpa)	**Evaporator Enthalpy, h_1 (kJ/kg)**	**Entropy at P_1, s_1 (kJ/kgK)**	**Gas Cooler Temperature, T_2 (° C)**	**Gas Cooler Enthalpy, h_2 (kJ/kg)**	**Mass Flow Rate CO_2, $\dot{m}$ (kg/s)**	**Heat absorbed, Q_{1-4} (kJ/s)**	**Work input to compressor, $\dot{W}_{2-1}$ (kJ/s)**	**COP**	**Quality, x_4**
2	436.85	1.9461	104.08	510.15	0.05872	7.27	4.30	1.69	0.56
2.5	435.66	1.9087	95	496.23	0.05929	7.27	3.59	2.02	0.53
3	433.61	1.8754	87.58	484.09	0.06030	7.27	3.04	2.39	0.51
3.5	430.80	1.8444	81.23	473.00	0.06174	7.27	2.61	2.79	0.49
4	427.25	1.8145	75.88	462.53	0.06366	7.27	2.25	3.24	0.47
4.5	422.90	1.7848	70.76	452.18	0.06618	7.27	1.94	3.75	0.44
5	417.66	1.7544	66.57	441.87	0.06949	7.27	1.68	4.32	0.42
5.5	411.28	1.7221	62.32	430.96	0.07401	7.27	1.46	4.99	0.39
6	403.32	1.6862	58.54	419.02	0.08053	7.27	1.26	5.75	0.33
6.5	392.84	1.6436	55.13	405.07	0.09111	7.27	1.11	6.53	0.03

4) COP vs Gas Cooler Exit Temperature Calculations

TABLE 7: COP vs Gas Cooler Exit Temperature Calculations

Temperature at Gas Cooler Exit, T_3 (°C)	Enthalpy at Gas Cooler Exit, h_3 (°C)	Quality, x_4	Mass Flow Rate CO_2, $\dot{m}$ (kg/s)	Heat absorbed, Q_{1-4} (kJ/s)	Work input to compressor, $\dot{W}_{2-1}$ (kJ/s)	COP
35	292.33	0.37	0.0539	7.27	1.901	3.82
40	313.04	0.47	0.0637	7.27	2.246	3.24
45	348.56	0.63	0.0924	7.27	3.259	2.23
50	384.07	0.80	0.1684	7.27	5.940	1.22
55	404.55	0.89	0.3202	7.27	11.296	0.64
60	425.02	0.99	3.2603	7.27	115.011	0.06
65	437.84	1.05	-0.6869	7.27	-24.230	-0.30
70	450.65	1.11	-0.3107	7.27	-10.960	-0.66

*rows highlighted represents data which were omitted from analysis

5) COP vs Quality Calculations

TABLE 8: COP vs Quality Calculations

Quality, x_4	Enthalpy at Evaporator, h_4 (°C)	Mass Flow Rate CO_2, $\dot{m}$ (kg/s)	Heat absorbed, Q_{1-4} (kJ/s)	Work input to compressor, $\dot{W}_{2-1}$ (kJ/s)	COP
0.1	234.67	0.03775	7.27	1.33	5.46
0.15	245.37	0.03997	7.27	1.41	5.16
0.2	256.07	0.04247	7.27	1.50	4.85
0.25	266.77	0.04530	7.27	1.60	4.55
0.3	277.46	0.04854	7.27	1.71	4.25
0.35	288.16	0.05227	7.27	1.84	3.94
0.4	298.86	0.05663	7.27	2.00	3.64
0.45	309.56	0.06178	7.27	2.18	3.34
0.5	320.26	0.06795	7.27	2.40	3.03
0.55	330.96	0.07550	7.27	2.66	2.73
0.6	341.66	0.08494	7.27	3.00	2.43

6) COP vs Evaporator Temperature Calculations

TABLE 9: COP vs Evaporator Temperature Calculations

Evaporator Temperature, T_1 (°C)	Entropy at 1, s_1 (kJ/kgK)	Evaporator Enthalpy, h_1 (kJ/kg)	Gas Cooler Temperature, T_2 (° C)	Gas Cooler Enthalpy, h_2 (kJ/kg)	Mass Flow Rate CO_2, $\dot{m}$ (kg/s)	Heat absorbed, Q_{1-4} (kJ/s)	Work input to compressor, $\dot{W}_{2-1}$ (kJ/s)	COP
-15	1.9237	436.27	98.49	501.75	0.058999	7.27	3.86	1.88
-10	1.8985	435.14	92.63	492.47	0.059545	7.27	3.41	2.13
-5	1.8725	433.38	86.99	483.05	0.060416	7.27	3.00	2.42
0	1.8453	430.89	81.41	473.32	0.061692	7.27	2.62	2.78
5	1.8163	427.48	76.19	463.15	0.063531	7.27	2.27	3.21
10	1.7847	422.88	70.74	452.15	0.066191	7.27	1.94	3.75
15	1.7489	416.64	65.85	440.01	0.070178	7.27	1.64	4.43
20	1.7062	407.87	60.22	425.59	0.076668	7.27	1.36	5.35
25	1.6498	394.43	55.62	407.10	0.089328	7.27	1.13	6.42

7) COP vs Isentropic Efficiency

TABLE 10: COP vs Isentropic Efficiency Calculations

Isentropic Efficiency, $\eta_{is, c}$	Gas Cooler Actual Enthalpy, h_{2s} (kJ/kgK)	Gas Cooler Actual Temperature, T_{2s} (° C)	Work input to compressor, $\dot{W}_{2-1}$ (kJ/s)	COP
0.2	603.63	129.69	11.23	0.65
0.4	515.44	95.28	5.61	1.30
0.5	497.80	88.40	4.49	1.62
0.6	486.04	83.81	3.74	1.94
0.8	471.35	80.25	2.81	2.59
1	462.53	75.88	2.25	3.24

8) COP vs Gas Cooler Exit Temperature (as a function of P_2)

TABLE 11: COP vs Gas Cooler Pressure (as a function of T_3) Calculations

Gas Cooler Pressure, P_2 (Mpa)	Temperature at Gas Cooler Exit, T_3 (°C)	Gas Cooler Temperature, T_2 (° C)	Gas Cooler Enthalpy, h_2 (kJ/kg)	Enthalpy at Gas Cooler Exit, h_3 (°C)	Mass Flow Rate CO_2, $\dot{m}$ (kg/s)	Heat absorbed, Q_L (kJ/s)	Work input to compressor, $\dot{W}$ (kJ/s)	COP
8	35	57.73	453.18	343.47	0.0868	7.27	2.25	3.23
	40			402.9	0.2986	7.27	7.74	0.94
	45			419.64	0.9548	7.27	24.76	0.29
	50			436.37	-0.7972	7.27	-20.67	-0.35
9	35	67.24	458.05	310.05	0.0620	7.27	1.91	3.81
	40			343.78	0.0871	7.27	2.68	2.71
	45			378.80	0.1500	7.27	4.62	1.57
	50			413.81	0.5410	7.27	16.66	0.44
10	35	75.88	462.53	292.33	0.0539	7.27	1.90	3.82
	40			313.04	0.0637	7.27	2.25	3.24
	45			348.56	0.0924	7.27	3.26	2.23
	50			384.07	0.1684	7.27	5.94	1.22
11	35	83.73	466.69	285.30	0.0512	7.27	2.02	3.60
	40			302.38	0.0582	7.27	2.30	3.17
	45			328.38	0.0735	7.27	2.90	2.51
	50			354.37	0.0998	7.27	3.93	1.85
12	35	90.86	470.56	280.82	0.0496	7.27	2.15	3.38
	40			296.06	0.0554	7.27	2.40	3.03
	45			316.24	0.0655	7.27	2.84	2.56
	50			336.41	0.0800	7.27	3.47	2.10
13	35	97.68	474.30	277.51	0.0486	7.27	2.28	3.18
	40			291.61	0.0536	7.27	2.52	2.88
	45			308.81	0.0614	7.27	2.89	2.52
	50			326.00	0.0718	7.27	3.38	2.15

9) COP vs Gas Cooler Pressure (as a function of evaporator temperature) Calculations

TABLE 12: COP vs Gas Cooler Pressure (as a function of T_1) Calculations

Evaporator Temperature, T_1 (°C)	**Gas Cooler Pressure, P_2 (Mpa)**	**Entropy at 1, s_1 (kJ/kgK)**	**Evaporator Enthalpy, h_1 (kJ/kg)**	**Gas Cooler Temperature, T_2 (° C)**	**Gas Cooler Enthalpy, h_2 (kJ/kg)**	**Gas Cooler Enthalpy, h_3 (kJ/kg)**	**Mass Flow Rate CO_2, $\dot{m}$ (kg/s)**	**Heat absorbed, Q_{1-4} (kJ/s)**	**Work input to compressor, $\dot{W}_{2-1}$ (kJ/s)**	**COP**
-15	8	1.9237	436.27	79	490.37	402.9	0.2179	7.27	11.79	0.62
	9			89	496.28	343.78	0.0786	7.27	4.72	1.54
	10			98	501.75	313.04	0.0590	7.27	3.86	1.88
	11			107	506.83	302.38	0.0543	7.27	3.83	1.90
	12			115	511.58	296.06	0.0519	7.27	3.91	1.86
	13			122	516.04	291.61	0.0503	7.27	4.01	1.81
-10	8	1.8985	435.14	74	481.60	402.9	0.2255	7.27	10.48	0.69
	9			84	487.25	343.78	0.0796	7.27	4.15	1.75
	10			93	492.47	313.04	0.0595	7.27	3.41	2.13
	11			101	497.30	302.38	0.0548	7.27	3.40	2.14
	12			109	501.87	296.06	0.0523	7.27	3.49	2.08
	13			116	506.18	291.61	0.0507	7.27	3.60	2.02
-5	8	1.8725	433.38	68	472.65	402.9	0.2385	7.27	9.37	0.78
	9			78	476.47	343.78	0.0811	7.27	3.50	2.08
	10			87	483.05	313.04	0.0604	7.27	3.00	2.42
	11			95	487.35	302.38	0.0555	7.27	3.00	2.43
	12			103	492.48	296.06	0.0529	7.27	3.13	2.32
	13			110	496.70	291.61	0.0513	7.27	3.25	2.24
0	8	1.8453	430.89	63	463.47	402.9	0.2598	7.27	8.46	0.86
	9			73	469.18	343.78	0.0835	7.27	3.20	2.27
	10			81	472.60	313.04	0.0617	7.27	2.57	2.83
	11			89	476.87	302.38	0.0566	7.27	2.60	2.79
	12			97	481.92	296.06	0.0539	7.27	2.75	2.64
	13			104	486.04	291.61	0.0522	7.27	2.88	2.53

5	8	1.8163	427.48	58	453.78	402.9	0.2958	7.27	7.78	0.93
	9			68	459.64	343.78	0.0869	7.27	2.79	2.60
	10			76	462.77	313.04	0.0635	7.27	2.24	3.24
	11			84	467.21	302.38	0.0581	7.27	2.31	3.15
	12			91	470.83	296.06	0.0553	7.27	2.40	3.03
	13			98	474.94	291.61	0.0535	7.27	2.54	2.86
10	8	1.7847	422.88	53	442.90	402.9	0.3639	7.27	7.28	1.00
	9			62	447.00	343.78	0.0919	7.27	2.22	3.28
	10			71	452.67	313.04	0.0662	7.27	1.97	3.69
	11			79	457.17	302.38	0.0603	7.27	2.07	3.51
	12			86	460.44	296.06	0.0573	7.27	2.15	3.38
	13			92	462.99	291.61	0.0554	7.27	2.22	3.27
15	8	1.7489	416.64	49	433.02	402.9	0.5291	7.27	8.67	0.84
	9			58	436.99	343.78	0.0998	7.27	2.03	3.58
	10			66	440.40	313.04	0.0702	7.27	1.67	4.36
	11			73	443.26	302.38	0.0636	7.27	1.69	4.29
	12			80	447.64	296.06	0.0603	7.27	1.87	3.89
	13			86	449.64	291.61	0.0581	7.27	1.92	3.79
20	8	1.7062	407.87	45	419.64	402.9	1.4629	7.27	17.21	0.42
	9			53	422.50	343.78	0.1134	7.27	1.66	4.38
	10			60	425.02	313.04	0.0767	7.27	1.31	5.53
	11			68	430.03	302.38	0.0689	7.27	1.53	4.76
	12			74	431.77	296.06	0.0650	7.27	1.55	4.68
	13			80	435.59	291.61	0.0625	7.27	1.73	4.19
25	8	1.6498	394.43	40	400.44	402.9	-0.858	7.27	-5.16	-1.41
	9			49	403.92	343.78	0.1435	7.27	1.36	5.34
	10			56	407.10	313.04	0.0893	7.27	1.13	6.42
	11			62	409.96	302.38	0.0790	7.27	1.23	5.93
	12			68	413.88	296.06	0.0739	7.27	1.44	5.06
	13			73	415.02	291.61	0.0707	7.27	1.46	4.99

10) COP vs Gas Cooler Pressure (as a function of evaporator pressure) Calculations

TABLE 13: COP vs Gas Cooler Pressure (as a function of P_1) Calculations

Evaporator Pressure, P_1(Mpa)	**Gas Cooler Pressure, P_2 (Mpa)**	**Entropy at 1, s_1 (kJ/kgK)**	**Evaporator Enthalpy, h_1 (kJ/kg)**	**Gas Cooler Temperature, T_2 (° C)**	**Gas Cooler Enthalpy, h_2 (kJ/kg)**	**Mass Flow Rate CO_2, $\dot{m}$ (kg/s)**	**Work input to compressor, $\dot{W}$ (kJ/s)**	**COP**
2	8	1.9461	436.85	84.58	498.34	0.2142	13.17	0.55
	9			94.79	567.78	0.0781	10.23	0.71
	10			104.08	510.15	0.0587	4.30	1.69
	11			112.59	515.40	0.0541	4.25	1.71
	12			120.41	520.28	0.0516	4.31	1.69
	13			127.81	524.96	0.0501	4.41	1.65
3	8	1.8754	433.61	68.71	473.63	0.2367	9.47	0.77
	9			78.57	479.05	0.0809	3.68	1.98
	10			87.58	484.09	0.0603	3.04	2.39
	11			95.83	488.77	0.0554	3.06	2.38
	12			103.39	493.14	0.0529	3.15	2.31
	13			110.31	497.20	0.0512	3.26	2.23
4	8	1.8145	427.25	57.73	453.18	0.2986	7.74	0.94
	9			67.24	458.05	0.0871	2.68	2.71
	10			75.88	462.53	0.0637	2.25	3.24
	11			83.73	466.69	0.0582	2.30	3.17
	12			90.86	470.56	0.0554	2.40	3.03
	13			97.68	474.30	0.0536	2.52	2.88
5	8	1.7544	417.66	49.17	433.58	0.4926	7.84	0.93
	9			58.31	437.89	0.0984	1.99	3.65
	10			66.57	441.87	0.0695	1.68	4.32
	11			73.99	445.57	0.0631	1.76	4.13
	12			80.64	449.00	0.0598	1.87	3.88
	13			87.17	452.38	0.0577	2.00	3.63

6	8	1.6862	403.32	42.69	411.92	17.3106	148.84	0.05
	9			50.59	415.52	0.1221	1.49	4.88
	10			58.54	419.02	0.0805	1.26	5.75
	11			65.52	422.26	0.0720	1.36	5.33
	12			71.53	425.23	0.0678	1.48	4.90
	13			77.48	428.18	0.0651	1.62	4.49

11) COP vs Gas Cooler Pressure (as a function of $\eta_{is,c}$) Calculations

TABLE 14: COP vs Gas Cooler Pressure (as a function of $\eta_{is,c}$) calculations

Gas Cooler Pressure, P_2 (Mpa)	**Isentropic Efficiency, $\eta_{is,c}$**	**Gas Cooler Temperature, T_2 (° C)**	**Gas Cooler Enthalpy, h_2 (kJ/kg)**	**Gas Cooler Actual Enthalpy, h_{2s} (kJ/kg)**	**Mass Flow Rate CO_2, $\dot{m}$ (kg/s)**	**Work input to compressor, $\dot{W}$ (kJ/s)**	**COP**
8	0.2	58	453.18	556.90	0.06366	8.25	0.88
	0.4			492.08		4.13	1.76
	0.5			479.11		3.30	2.20
	0.6			470.47		2.75	2.64
	0.8			459.66		2.06	3.52
	1			453.18		1.65	4.40
9	0.2	67	458.05	581.25	0.06366	9.80	0.74
	0.4			504.25		4.90	1.48
	0.5			488.85		3.92	1.85
	0.6			478.58		3.27	2.22
	0.8			465.75		2.45	2.97
	1			458.05		1.96	3.71
10	0.2	76	462.53	603.65	0.06366	11.23	0.65
	0.4			515.45		5.61	1.29
	0.5			497.81		4.49	1.62
	0.6			486.05		3.74	1.94
	0.8			471.35		2.81	2.59

	1			462.53		2.25	3.24
11	0.2	84	466.69	624.45	0.06366	12.55	0.58
	0.4			525.85		6.28	1.16
	0.5			506.13		5.02	1.45
	0.6			492.98		4.18	1.74
	0.8			476.55		3.14	2.32
	1			466.69		2.51	2.90
12	0.2	91	470.56	643.80	0.06366	13.79	0.53
	0.4			535.53		6.89	1.05
	0.5			513.87		5.51	1.32
	0.6			499.43		4.60	1.58
	0.8			481.39		3.45	2.11
	1			470.56		2.76	2.64
13	0.2	98	474.3	662.50	0.06366	14.98	0.49
	0.4			544.88		7.49	0.97
	0.5			521.35		5.99	1.21
	0.6			505.67		4.99	1.46
	0.8			486.06		3.74	1.94
	1			474.30		3.00	2.43

12) COP vs Gas Cooler Pressure (as a function of x_4) Calculations

TABLE 15: COP vs Gas Cooler Pressure (as a function of x4) Calculations

Quality, x_4	Gas Cooler Pressure, P_2 (Mpa)	Gas Cooler Temperature, T_2 (° C)	Gas Cooler Enthalpy, h_2 (kJ/kg)	Enthalpy at Evaporator, h_4 (^{O}C)	Mass Flow Rate CO_2, $\dot{m}_{CO2}$ (kg/s)	Work input to compressor, $\dot{W}$ (kJ/s)	COP
0.1	8	58	453.18	234.67	0.03775	0.98	7.43
0.2				256.07	0.04247	1.10	6.60
0.3				277.46	0.04854	1.26	5.78
0.4				298.86	0.05663	1.47	4.95
0.5				320.26	0.06795	1.76	4.13
0.6				341.66	0.08494	2.20	3.30
0.1	9	67	458.05	234.67	0.03775	1.16	6.25
0.2				256.07	0.04247	1.31	5.56
0.3				277.46	0.04854	1.49	4.86
0.4				298.86	0.05663	1.74	4.17
0.5				320.26	0.06795	2.09	3.47
0.6				341.66	0.08494	2.62	2.78
0.1	10	76	462.53	234.67	0.03775	1.33	5.46
0.2				256.07	0.04247	1.50	4.85
0.3				277.46	0.04854	1.71	4.25
0.4				298.86	0.05663	2.00	3.64
0.5				320.26	0.06795	2.40	3.03
0.6				341.66	0.08494	3.00	2.43
0.1	11	84	466.69	234.67	0.03775	1.49	4.88
0.2				256.07	0.04247	1.67	4.34
0.3				277.46	0.04854	1.91	3.80
0.4				298.86	0.05663	2.23	3.26
0.5				320.26	0.06795	2.68	2.71
0.6				341.66	0.08494	3.35	2.17

0.1	12	91	470.56	234.67	0.03775	1.63	4.45
0.2				256.07	0.04247	1.84	3.95
0.3				277.46	0.04854	2.10	3.46
0.4				298.86	0.05663	2.45	2.96
0.5				320.26	0.06795	2.94	2.47
0.6				341.66	0.08494	3.68	1.98
0.1	13	98	474.30	234.67	0.03775	1.78	4.09
0.2				256.07	0.04247	2.00	3.64
0.3				277.46	0.04854	2.28	3.18
0.4				298.86	0.05663	2.66	2.73
0.5				320.26	0.06795	3.20	2.27
0.6				341.66	0.08494	4.00	1.82