

PERFORMANCE ANALYSIS OF TRANSCRITICAL CO₂ REFRIGERATION SYSTEM FOR SUPERMARKET APPLICATION

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ABSTRACT: Environmental issues becomes a priority on the refrigeration system development. The main issues are global warming and ozone depletion which are emitted from the conventional refrigeration system. One of the natural and clean refrigerants is Carbon dioxide (CO₂). CO₂ refrigeration system however still has low performance when operating at high ambient temperature. This study aim is to investigate the performance of CO₂ transcritical refrigeration system in high outdoor temperature. The research was carried out by theoretical study and numerical analysis of the refrigeration system using the EES (Engineering Equation Solver) program. Data input and simulation validation were obtained from experimental and secondary data. The result showed that the coefficient of performance (COP) decreased gradually with the outdoor temperature variation increasing. With temperature input increases between 25°C – 45°C, the performance (indicated by overall COP) decreased by 3.0 %. These results will be significantly important for a preliminary reference to improve the CO₂ refrigeration system design in hot climate temperature application.

Keywords: Global warming, Ozone depletion, CO₂ refrigeration system, EES (Engineering Equation Solver), COP

1. INTRODUCTION

Recently, the use of environment-friendly natural refrigerant substituting chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) has been widely investigated by several researchers. Among the natural refrigerants (carbon dioxide, ammonia, hydrocarbon, etc.), carbon dioxide (CO₂) satisfies a lot of thermal characteristics, such as low viscosity, high volumetric capacity, excellent heat transfer coefficients, no toxicity and is inflammable. In addition, CO₂ has zero Ozone Depletion Potential (ODP), negligible Global Warming Potential (GWP) and relatively low cost. So CO₂ is an excellent alternative to the conventional refrigerants [1] - [3].

Carbon dioxide can be used in almost all refrigeration system applications [4] and it is now becoming common in supermarket applications [5]. Other applications, CO₂ has been introduced for automotive air conditioners, vending machines and heat pump water heaters. For example in heat pump water heater, the supercritical operation (i.e., rejection of heat above the critical point) is an additional benefit because the temperature is matching between the water and supercritical CO₂, which improves the coefficient of performance [6],[7]. As a secondary refrigerant, CO₂ can be used as low-temperature stage refrigerant in cascade systems, typically with ammonia or R-507A as the high-temperature refrigerant, in large industrial systems. Medium-sized commercial systems can

also use CO₂ as the low-temperature stage refrigerant in cascade arrangements with HFCs or hydrocarbons as the high-temperature stage refrigerant. In early CO₂ refrigeration systems for supermarket applications, the cascade arrangement was also preferred to avoid high pressures in condensing side and supercritical operation [8].

In warm climate, the supercritical system is an excellent option for CO₂ refrigeration system because the critical point of CO₂ is at a relatively low temperature at 31°C, at relatively high pressure (73.8 bar), with the triple point occurs at -56.6°C at a pressure of 5.2 bar [9]-[11]. Tsamos [12] compared the performance between booster system with gas bypass compressor and other design (i.e. cascade system with gas bypass), it was found that the booster system gets 5% energy saving in a warm climate. So this system will also be investigated in this study.

Heat exchangers improvements also have been done toward CO₂ refrigeration system. Firstly, they concerned about gas cooler/condenser optimization [13-16], which obtained that the gas cooler design can effect significantly in supercritical condition. Secondly, improvement used application of Internal Heat Exchanger (IHX) [3],[6],[17]. Aprea and Maiorino [3], Torrella et al. [6] conducted experiments investigating the influence of the internal heat exchanger (IHX) on carbon dioxide supercritical refrigerating plants and the performance of the system. It was confirmed that the use of the IHX increases the COP of the system

by 10%. In addition, the use of the IHX was associated with an increase in compressor discharge temperature, reaching increments up to 10°C at the evaporating temperature of 15°C. Moreover, Rigola et al. [17] added that there are specific conditions to reach maximum performance of a CO₂ supercritical refrigeration system using an internal heat exchanger. The first condition, when the ambient temperature of 35°C and the optimal discharge pressure is between 95 and 100 bar, the COP increases by 20%. Secondly, when the ambient temperature increases become 43°C, so the optimal gas cooler pressure is between 105 and 110 bar, with IHX the COP can be increased up to 30%. These studies show that IHX can improve the performance of the system if it is applied with well-designed and analysis.

Depend on previous studies reported, it can be argued that CO₂ refrigeration system will become excellent alternative natural refrigeration system, however, the study about the transcritical system in tropical countries especially Indonesia is null, so this study will be very important for the environment-friendly refrigeration system development.

2. SIMULATION DESIGN

Carbon dioxide (CO₂) systems, supercritical or sub-critical booster refrigeration systems are widely used in supermarkets [10],[18]. Figure 1 shows the

design of CO₂ refrigeration booster system (without Internal Heat Exchanger) which has been investigated in this study. The system has four pressure regions: high, intermediate, medium and low, with two-stage compressors (low stage and high stage compressor). Evaporating systems are covering Medium Temperature (MT) and Low Temperature (LT) evaporators. The system also comprises two expansion valves. The first valve is bypass valve. This valve has a duty to mix the expanded vapor from the receiver with refrigerant from low stage compressor -LP and MT evaporator. The mixture then flows to the high stage compressor-HP. The second valve is expansion valve - ICMT type which operates to decrease pressure from high pressure to medium pressure region. Two electronic expansion valves, AKV-MT and AKV-LT adjust the MT and LT evaporator pressure, respectively.

Figure 2 illustrates the p-h diagram which automatically depends on the design shown in Figure 1. The refrigeration cycle in Figure 1 and 2 can be traced to thermodynamic processes characterized by numbers from 1 to 14 with the following processes: The 1-2 process is a vapor compression in a high-pressure compressor (HP) with high pressure superheat refrigerant conditions until transcritical conditions (above the critical point). The compressor performance is determined by the isentropic efficiency. Process 2-3 is heat rejection processed by a gas cooler (cooling in one

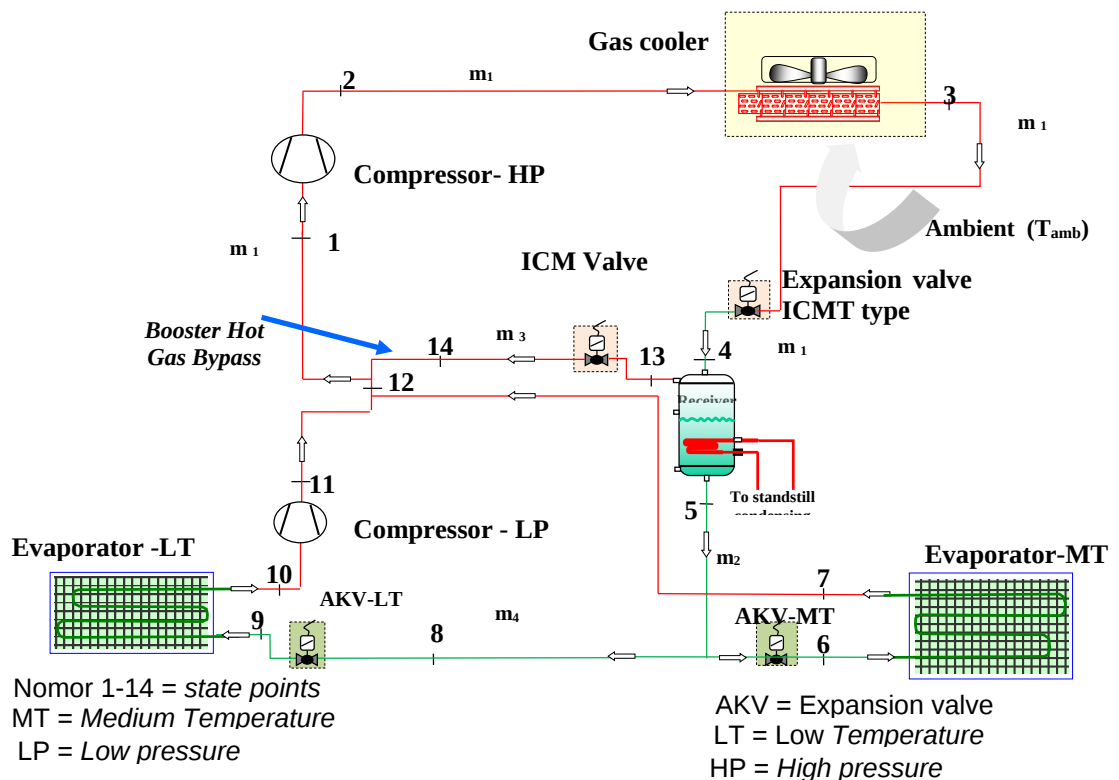


Fig.1 Schematic diagram of CO₂ refrigeration booster system design

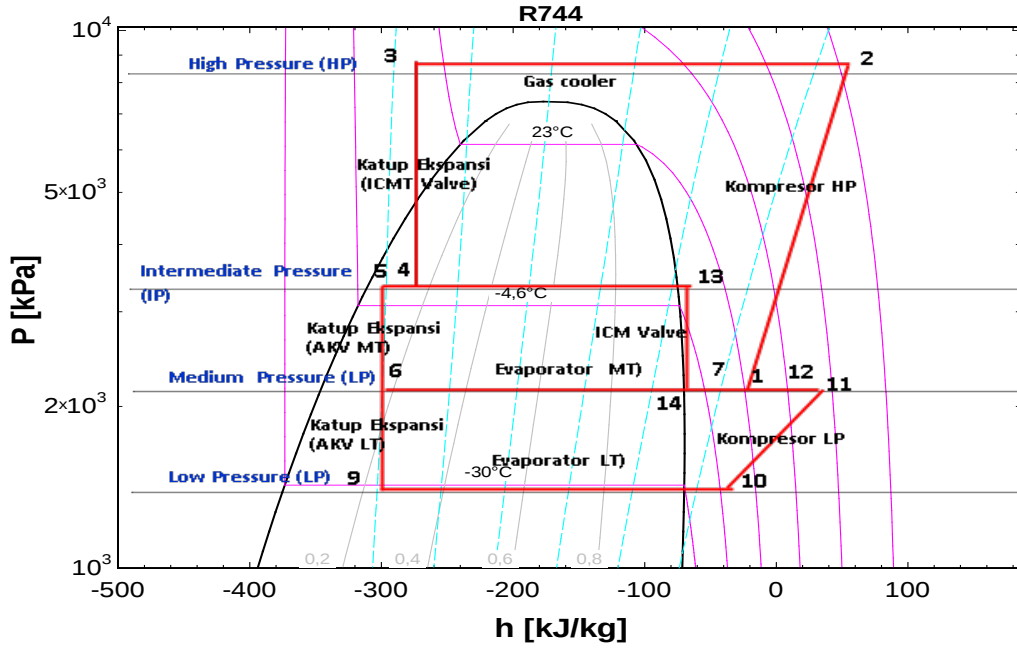


Fig.2 P-h diagram of the system design

phase) due to transcritical condition [14] then the performance parameter used here is approach temperature refer to Eq. (10). The State of 3-4 is adiabatic expansion process (isenthalpic) which is done by an expansion valve. Because of relatively high pressure, the expansion valve type is ICMT [19], the refrigerant state in point 4 is mixed vapor and liquid and further cooling occurs on the receiver to get the refrigerant in a liquid state (the 4-5 process). The 5-6 and 6-9 processes are an adiabatic (isenthalpic) expansion process with two levels of pressure including, medium temperature and low-temperature systems. The 10-11 process is a compression that corresponds to the isentropic efficiency. The 13-14 process is an expansion made by the ICM valve for hot gas bypass.

3. THERMODYNAMIC ANALYSIS

In order to build EES (Engineering Equation Solver) simulations, thermodynamic equations and analytical assumptions were developed to simulate the CO₂ refrigeration system with booster system, the thermodynamic analysis is as follow:

3.1 Performance Equations

Equations have been developed according to the system design and the p-h diagram as shown in Fig.1 and Fig.2, respectively.

3.1.1 COP calculation

The system consists of two levels/stages of evaporator namely evaporator LT and evaporator

MT then COP also are calculated, COP_{MT}, COPLT and COP_{tot} with the following formula:

$$COP_{MT} = \frac{h_{11}-h_6}{h_2-h_1} \quad (1)$$

$$COP_{LT} = \frac{h_{10}-h_9}{h_{11}-h_{10}} \quad (2)$$

$$COP_{tot} = \frac{(h_{11}-h_6)+(h_{10}-h_9)}{(h_2-h_1)+(h_{11}-h_{10})} \quad (3)$$

3.1.2 Mass flow rate distribution

Equations of distribution of m (mass flow rate) from the simulation diagram obtained:

$$m_1 = m_2 + m_3 \quad (4)$$

$$m_2 = m_4 + m_5 \quad (5)$$

3.1.3 Compressor work

Compressor works are calculated from the refrigerant side. High and low-pressure compressors equation are obtained as follows:

$$W_{HP} = \dot{m}_1.(h_2 - h_1) \quad (6)$$

$$W_{LP} = \dot{m}_1.(h_{11} - h_{10}) \quad (7)$$

3.1.4 Heat rejection

The heat rejection in gas cooler is calculated from the refrigerant side as follows:

$$Q = \dot{m}_1.(h_2 - h_3) \quad (8)$$

3.1.5 Energy balance

The flow rate of the refrigerant cycle is calculated according to the energy balance of the compressor. The energy balance has shown that the

refrigerant flow can be calculated indirectly from the energy balance between the refrigerant and the electrical power required by the compressor with the assumption of adiabatic energy transfer. The energy of the compressor is calculated by the equation:

$$\dot{m}_1 \cdot (h_2 - h_1) = V A \cos \phi \quad (9)$$

where $\cos \phi$ is power factor = 0,85 ,
 V = (volt), A = current (A)

3.1.6 Approach temperature (AT)

Approach temperature for the heat exchanger is defined as the minimum temperature difference between the two fluids, for air-cooled gas cooler, the approach temperature is assumed as the temperature difference between the refrigerant outlet ($T_{ref out}$) and the air inlet (T_{air-on}) as described by Ge and Tassou [20].

$$AT = T_{ref out} - T_{air-on} \quad (10)$$

The switch point between sub-critical and supercritical is determined by the critical point of R744 , $P_{crit-a} = 73.77 \text{ bar-absolute}$ or $P_{crit-gauge} \approx 72.77 \text{ bar-gauge}$

3.1.7 Superheated and Sub-cooling

Conditions of super-heat gas and sub-cooling levels under liquid conditions are determined by the equation:

$$\Delta T = T_{sat} \pm T_{ref out} \quad (11)$$

3.1.8 Optimum pressure of gas cooler

The optimum pressure is taken from Ge and Tassou [8] as described at following equations:

$$Y = 2.3426 x + 11.541, \quad \text{with } R^2 = 0.9991 \quad (12)$$

Where, Y = optimum pressure of gas cooler (bar) and x = ambient temperature ($^{\circ}\text{C}$)

3.2 Assumptions Development

Whereas in simulating of EES Program [21] there are some control variables that are assumed based on empirical condition as follows:

- The efficiency of the heat exchanger gas cooler type of finned tube: 0.85
- Evaporator efficiency of finned tube type: 0.8
- Isentropic efficiency of LP and HP compressors: 80%
- The pressure drop in the expansion valve is adiabatically / isenthalpic
- Checking of the mass flow rate is using the energy balance equation in compressors
- Approach temperature is optimal for 3 K
- The saturated liquid side (no degree of sub-cooling) or saturated state
- Superheated on LT evaporator = 8K

- Superheated on HT evaporator = 10K

All the properties of carbon dioxide and air are based on conditions in Indonesian climate.

The EES specification has been used as follow: Program Engineering Equation Solver (EES), Academic Commercial V10.289-3D, Mechanical Engineering Department, Bali State Polytechnic (Politeknik Negeri Bali).

4. PRELIMINARY RESULTS AND DISCUSSIONS

Preliminary results and analysis of COP based on ambient temperature were found that the gradual increasing ambient temperature from temperature 26°C to 48°C was obtained a gradual decrease of COP. Medium Temperature (MT) COP is lower than COP of Low Temperature (LT), this is due to transcritical conditions in temperature medium system that heat rejection conditions at a high temperature and pressure so that thermal efficiency and the isentropic compressor is getting lower.

Figure 3 also shows the advantage of LT system that is working in constant COP. This is because the compressor LP which drive the LT system much lower work than HP compressor, and constant work occurs because of a multi-stages system that LT system only effected by refrigeration load. In another hand, for the MT system, the higher temperature will effect to increase the discharge pressure that becomes the HP compressor duty.

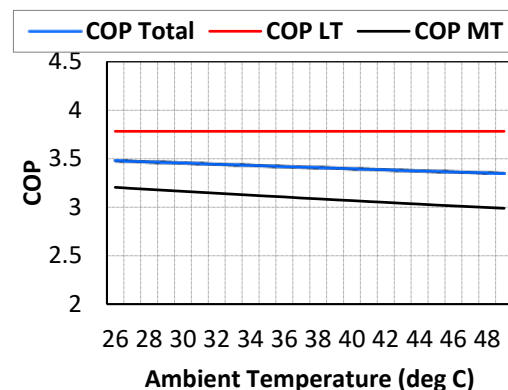


Fig.3 Variation of COP with ambient temperature

Figure 4 shows the pressure also has significant effect on the COP. this is related to the compressor work. Depend on the thermodynamics analysis the higher pressure can effect the compressor work heavier. To solve this problem, the previous researcher found some ways. First, installing the eligible gas cooler that can minimize the discharge pressure. Because the transcritical condition of the gas cooler is significantly important to control the discharge pressure, and usually using higher capacity than that the condenser [14]-[16]. Second,

other solution, adding the internal heat exchanger that will be conducted in the next study in this research project. Previous researcher using an internal heat exchanger (IHX) to reduce the exergy losses [2] but in really for the hot climate, the IHX need to be investigated more detail in operational optimization. In the next, this research project will be analysis the combination IHX in order to improve the performance of the system.

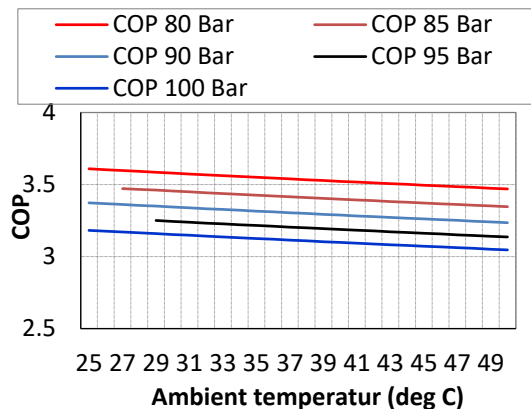


Fig.4 COP characteristic in different discharge pressure

The performance of CO₂ refrigeration transcritical system in hot climate show slightly lower than other conventional refrigeration system (such as, R22, R134a) which have COP around 4,5. However, depending on the refrigerant price, safety, environmentally friendly and non-toxic incentives, the CO₂ system still more eligible to be implemented in big refrigeration capacity, such as supermarket [20].

In summary, it can be argued that the heat exchanger in discharge pressure especially the condenser/gas cooler is significantly important to improve the performance of the system [14]. Because it works above critical point then the previous researcher indicated the performance of the gas cooler with approach temperature (AT) [20]. Supporting by combination IHX ineligible state points (positions) will contribute increase performance the CO₂ refrigeration transcritical system. In future, this system will can compete with other conventional system in term of performance system (COP).

5. CONCLUSIONS

CO₂ Refrigeration for supermarket application, when operated in the tropical country, has lower performance comparing with conventional refrigeration systems such as R22 and R134a. Moreover, in a tropical climate, ambient temperature show plays an important rule for the

performance (COP) of the CO₂ refrigeration system. The COP decreases gradually when the ambient temperature is increasing. As much as 3.0% of the COP decreased when ambient temperature increase from 26°C until 45°C. Discharge pressure also effects to the COP system that the COP higher on lower pressure. Increasing pressure from 80 bar until 100 bar was recorded that COP decrease of 3.6 to 3.2. However, the CO₂ transcritical system has some disadvantages in term of high pressure and temperature operation in the tropical country, but other factors such as the refrigerant price, safety and nontoxicity for food will make this system still eligible to apply for supermarket application. Finally, to improve CO₂ refrigeration performance in a tropical country can be recommended that using a combination of IHX system will become the best practice and be conducted in the next study.

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