Simulation of Transcritical CO₂ Refrigeration System with Booster Hot Gas Bypass in Tropical Climate

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Abstract. A Simulation computer becomes significant important for performance analysis since there is high cost and time allocation to build an experimental rig, especially for CO₂ refrigeration system. Besides, to modify the rig also need additional cos and time. One of computer program simulation that is very eligible to refrigeration system is Engineering Equation System (EES). In term of CO₂ refrigeration system, environmental issues becomes priority on the refrigeration system development since the Carbon dioxide (CO₂) is natural and clean refrigerant. This study aims is to analysis the EES simulation effectiveness to perform CO₂ transcritical refrigeration system with booster hot gas bypass in high outdoor temperature. The research was carried out by theoretical study and numerical analysis of the refrigeration system using the EES 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. The results show the program can calculate the performance of the refrigeration system with quick running time and accurate. So, it will be significant important for the preliminary reference to improve the CO₂ refrigeration system design for the hot climate temperature.

1. Introduction
Waltrich et al.[1] developed a model to investigate the thermo-hydraulic behaviour of compact fan-cooled tube-fin heat exchangers used as gas coolers in CO₂ refrigeration systems. The model was based on the mass, momentum and energy conservation equations applied to both the refrigerant and air streams. The model predictions for the temperature profiles, heat transfer rate and the air-side pressure drops were compared with experimental data obtained using a purpose-built test facility. It was observed that the model is able to predict the overall performance of heat exchangers in terms of heat transfer rate and pressure drop with errors within 10% and 15% bands, respectively. Ge and Cropper[2] presented a distributed model for the steady state simulation of finned-tube air-cooled gas coolers. The model was validated against data published in the literature and was used to investigate the effect of different pipe circuit arrangements on the performance of the gas coolers.

CO₂ can be used in almost all refrigeration system applications and is now becoming common in supermarket applications [3]. As primary refrigerant, CO₂ has been proposed for use in mobile air conditioners, supermarket display cases, and vending machines. In the application in heat pump water heaters for example, the supercritical operation (i.e., rejection of heat above the critical point) is beneficial because it allows good temperature matching between the water and supercritical CO₂, which improves the coefficient of performance [5],[6]. As a secondary refrigerant, CO₂ can be used as the 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 and supercritical operation [7].

2. Simulation CO₂ Refrigeration System Design

Being environmentally friendly systems, supercritical or sub-critical booster refrigeration systems are widely used in supermarkets [8],[9]. Figure 1 shows a typical booster cycle for supermarket application. The system has four pressure regions: high, intermediate, medium and low, with two stage compressors (low stage and high stage compressor) and two evaporating systems which are Medium Temperature (MT) and Low Temperature (LT) evaporators. The system also comprises two bypass valves (BPV). The first valve mixes the expanded vapour from the receiver with refrigerant from the low stage compressor (Comp LP) and MT evaporator. The mixture then flows through an internal heat exchanger (IHX) before entering the high stage compressor (Comp HP). In this system, a second bypass valve (BPV-2) is included to bypass. This situation may occur at the system operates in the sub-critical condition.

Figure 1. Schematic diagram CO₂ refrigeration design

Figure 1 shows the designed CO₂ cooling system and Figure 2 illustrates the p-h diagram of the designed system. This cooling system is a bypass booster system that has four pressure areas, high (HP), medium (IP), medium pressure (MP) and low (LP). The main components of high pressure areas include two high temperature compressors (HT) and gas cooler / condenser (in transcritical conditions called gas coolers and subcritical called condensers). Expansion valve (ICMT) decrease pressure from high pressure to medium pressure areas [10],[11]. A mechanical component for the medium pressure area of the ICM valve that controls the pressure on the receiver and lowers the vapour pressure of the refrigerant from the receiver to the medium pressure level of the system. The medium pressure area also includes an MT evaporator (temperature medium), and an electronic expansion valve (AKV-MT) as well as components for additional load on the system when required. Low Temperature Area consists of five main components that include: LT compressor, expansion valve (AKV-LT), LT evaporator.
The refrigeration cycle in Figure 1 and 2 can be traced to thermodynamic processes characterized by numbering from 1 to 14 with the following processes: The 1-2 process is a vapour compression in a high pressure compressor (HP) with high pressure superheat refrigerant conditions until transcritical conditions (above the critical point) where the compressor performance is determined from the isentropic efficiency. Process 2-3 is heat rejection processed by gas cooler (cooling in one phase) due to transcritical condition [12] then the parameter used here is approach temperature (T_{airin}-T_{ref out}), 3-4 is adiabatic expansion process (isenthalpic) by an expansion valve. Because of relatively high pressure, so the expansion valve type is ICMT [13], the refrigerant state in point 4 is mixed and experiencing further cooling on the receiver so that the refrigerant on liquid state in 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 booster.

3. Thermodynamic Analysis
In developing EES-based simulations some thermodynamic equations were developed to simulate the CO₂ refrigeration system with a system booster, as follows:

a. COP calculation:
The system consists of two levels of evaporator namely evaporator LT and evaporator MT then COP calculation there are three namely, COPMT, COPLT and COP{	ext{tot}} with the following formula:

\[
COP_{MT} = \frac{h_{11} - h_{6}}{h_{2} - h_{1}} \tag{1}
\]

\[
COP_{LT} = \frac{h_{10} - h_{9}}{h_{11} - h_{10}} \tag{2}
\]

\[
COP_{tot} = \frac{(h_{11} - h_{6}) + (h_{10} - h_{9})}{(h_{2} - h_{1}) + (h_{11} - h_{10})} \tag{3}
\]
b. Mass flow rate distribution

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

\[ m_1 = m_3 + m_4 + m_5 \]  
(4)

\[ m_2 = m_4 + m_5 \]  
(5)

c. Compressor work

Compressor work are calculated from refrigerant side obtained equation as follows:

\[ W_{HP} = \dot{m}_1 (h_2 - h_1) \]  
(6)

\[ W_{LP} = \dot{m}_1 (h_{11} - h_{10}) \]  
(7)

d. Heat rejection

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

\[ Q = \dot{m}_1 (h_2 - h_3) \]  
(8)

e. 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:

\[ \dot{m}_1 (h_2 - h_1) = V A \cos \phi \]  
(9)

where \( \cos \phi \) is power factor = 0.85, \( V \) = (volt), \( A \) = current (A)

f. Approach temperature (AT)

Approach temperature for 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 and the air inlet as described by Ge and Tassou [14].

\[ AT = T_{ref\ out} - T_{air-on} \]  
(10)

The switch point between sub-criticism and supercritical is determined by the critical point of R744, \( P_{crit-a} = 73.77 \text{ bara} \) or \( P_{crit-g} \approx 72.77 \text{ barg} \)

g. Superheated and Sub-cooling

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

\[ \Delta T = T_{sat} - T_{ref\ out} \]  
(11)

h. Optimum pressure of gas cooler

The optimum pressure is taken from Ge and Tassou [4] with the following equation:

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

Where, \( Y \) = optimum pressure of gas cooler (bar) and \( x \) = ambient temperature

i. Assumptions

Whereas in simulating in EES Program [13] there are control variables that are assumed based on condition and data of survey data as follows:

- Efficiency of heat exchanger gas cooler type finned tube: 0.85
- Evaporator efficiency of fined tube type: 0.8
- Isentropic efficiency of LP and HP compressors: 80%
- The pressure drop in the expansion valve proceeds adiabatically / isenthalpic
- Check the mass flow rate with the balance energy equation in the compressor
- Approach temperature 3 K
- The saturated liquid side (no degree of sub-cooling)
- 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.

4. Simulation Results and Discussion
EES Simulation results show that variation of the COP with increase ambient temperature was found that temperature 26°C to 48°C was obtained gradual decrease of the COP. Medium Temperature COP is lower than COP Low Temperature, this is due to transcritical conditions in temperature medium that heat rejection conditions at high temperature and pressure so that thermal efficiency and isentropic compressor is lower. The result show good agreement with previous result [3],[4],[5],[15]. As shown in the following figure.

![Figure 3](image)

**Figure 3.** EES results, (a) numeric simulation and (b) Variation of COP toward ambient temperature and gas cooler pressure

5. Conclusions
According to previous studies comparison show good agreement in term of ambient temperature variation with the performance (COP). The EES simulation shows best practice program for CO₂ Refrigeration system according to temperature validation on some important point. Where, The COP decrease gradually when the ambient temperature is increasing.

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References
[1] Waltrich M, Hermes C J L, Goncalves J M and Melo C 2010 A first-principles simulation model for the thermo-hydraulic performance of fan supplied tube-fin heat exchangers, *App. Ther. Eng.* 30 2011-2018.
[2] Ge Y T and Cropper R T 2009 Simulation and performance evaluation of finned-tube CO₂ gas coolers for refrigeration systems, *App. Thermal Eng.* 29 957-965.
[3] Pearson A B 2014 *IIR Guide CO2 as a refrigerant.* (International Institute of Refrigeration ISBN: 978-2-36215-006-7)

[4] Sánchez D, Patiño J, Llopis R, Cabello R and Torrella E 2014 New positions for an internal heat exchanger in a CO2 supercritical refrigeration plant. Experimental analysis and energetic evaluation. *App. Therm. Eng.* **63** 129-139.

[5] Torrella E, Sánchez D, Llopis R, Cabello R 2011 Energetic evaluation of an internal heat exchanger in a CO2 transcritical refrigeration plant using experimental data. *Int. J. of Ref.* **34** 40-49.

[6] Ge Y T, and Tassou S A 2011 Thermodynamic analysis of transcritical CO2 booster refrigeration systems in supermarket. *E. Conver. and Management* **52** 1868-1875.

[7] Ommen T and Elmegaard B 2012 Numerical model for thermoeconomic diagnosis in commercial transcritical/subcritical booster refrigeration system. *E. Conver. and Management* **60** 161-169.

[8] Lucas C and Koehler J 2012 Experimental investigation of the COP improvement of a refrigeration cycle by use of an ejector. *Int. J. of Ref.* **35** 1595-1603.

[9] Nakagawa M, Marasigan A R and Matsukawa T 2011 Experimental analysis on the effect of internal heat exchanger in transcritical CO2 refrigeration cycle with two-phase ejector. *Int. J. of Ref.* **34** 1577-1586.

[10] Gupta K, Singh D K and Dasgupta S 2010 Environmental effect on gas cooler design for transcritical carbon dioxide refrigeration system in India context. *J. of Adv. Res.in Mech. Eng.* pp.147-152.

[11] Ta Y B, He Y L and Tao W Q 2010 Exergetic analysis of transcritical CO2 residential air-conditioning system based on experimental data. *Appl. Energy* **87** 3065-3072.

[12] Santosa I D M C, Gowreesunker BL, Tassou S A and Tsamos KM 2017 Investigations into air and refrigerant side heat transfer coefficients of finned-tube CO2 gas coolers. *Int. J. of Heat and Mass Trans.* **107** 168–180.

[13] Danfoss-Optyma OP-MCHC034GSA01G.Product catalogue, available from: www.danfoss.co.uk

[14] F-Chart Software. EES (engineering equation solver); 2017. <http://www.fchart.com>.

[15] Ge Y T and Tassou S A 2009. Control optimisation of CO2 cycles for medium temperature retail food refrigeration systems. *Int. J. of Ref.* **32** 1376-1388.