Journal of Thermal Engineering, Vol. 3, No. 5, pp. 1505-1514 October, 2017
Yildiz Technical University Press, Istanbul, Turkey

AN EXPERIMENTAL STUDY ON SUBCOOLING PROCESS OF A TRANSCRITICAL CO_2 AIR CONDITIONING CYCLE WORKING WITH MICROCHANNEL EVAPORATOR

T. Dang[1,*], K. Vo[1], C.Le[2], T. Nguyen[1]

ABSTRACT

An experimental study on subcooling process of a transcritical CO_2 air conditioning cycle working with microchannel evaporator was done. In this cycle there are two different subcoolers namely S1 and S2 were installed and tested. The experimental data show that the COP of the cycle working with the subcooler S2 is better which is at 7.2. The evaporator pressure, the subcooler pressure, the subcooling temperature and the compressor current corresponding to the above-mentioned COP are 44 bar, 75 bar, 26 ºC, and 2.4 A, respectively. A total comparison between the present study and other literatures was also indicated which confirms that the results gained by the present study look better.

Keywords: *CO_2 refrigerant, Air conditioning system, Subcooling, Heat transfer, Microchannel*

INTRODUCTION

Energy conservation and environmental protection relating to greenhouse gas emissions for sustainable development are now global issues in general, whereas environmentally friendly refrigerants and high efficiency heat exchangers are specific issues attracting attention of scientists working in the field of air conditioning engineering in particular. Corresponding to this topic, natural refrigerants and microchannel heat exchangers have been considered as one of good solutions in order to solve this problem. In the study published by Lorentzen [1], the energy consumption of CO_2 air conditioning system is equal nearly 80% of that consumed by the air conditioning system working with R12 at the same temperatures. Kuang et al. [2] studied a semi-empirical correlation of gas cooling heat transfer of supercritical carbon dioxide in microchannels. Based on their experimental data, a new semi-empirical correlation was developed to predict the gas cooling heat transfer coefficient of supercritical CO_2 in microchannels within an error of 15%. These experimental data were obtained in an 11-port microchannel tube with an internal diameter of 0.79 mm and with a pressure range from 8 to 10 MPa and mass flux range from 300 to 1200 kg/m^2s. A comparative study of a cascade cycle for simultaneous refrigeration and heating operating with ammonia, R134a, butane, propane, and CO_2 as working fluids was presented by Colorado et al. [3]. In this paper, Ammonia, R134a, butane and propane were evaluated in the low temperature cycle and carbon dioxide (CO_2) is used in the high temperature cycle. The results showed that the cascade system using butane in the low temperature cycle increased the Coefficient of Performance (COP) up to 7.3 % in comparison with those obtained with NH_3–CO_2. On the other hand, the cascade systems operating with the mixtures R134a–CO_2 and propane-CO_2 presented similar results reaching COPs up to 5% higher than those obtained with the NH_3–CO_2 system. However, the investigations in [3] did not experimentally perform.

Regarding to the compact heat exchangers, the cooling concept of microchannel heat sinks was first approached by Tuckerman and Pease [4]. With CO_2 refrigerant and compact heat exchangers, an overview of the boiling heat transfer characteristics and the special thermophysical properties of CO_2 in a horizontal tube was investigated by Zhao and Bansal [5]. Due to the large surface tension, the boiling heat transfer coefficient of CO_2 was found to be much lower at low temperatures but it increased with the vapor quality. However, this study was only reviewed for horizontal tube. Using numerical simulation, Cheng and Thome [6] studied on cooling of microprocessors using CO_2 flow boiling in a micro-evaporator. In this study, CO_2 had high heat transfer coefficients and low pressure drops in the multi-microchannel evaporator. However, the operation pressure of CO_2 was higher than that of R236fa. Based on the analysis and comparison, CO_2 appeared to be a promising

This paper was recommended for publication in revised form by Regional Editor Kwok-wing Chau
[1]Department of Thermal Engineering, HCMC University of Technology and Education, Vietnam
[2]Department of Heat and Refrigeration Engineering, HCMC University of Technology – VietnamNational University – Ho Chi Minh City, Vietnam
**E-mail address: trungdang@hcmute.edu.vn*
Manuscript Received 6 March 2017, Accepted 7 May 2017

coolant for microprocessors at low operating temperatures but also presented a great technological challenge as other new cooling technologies.

Regarding to two-phase flow in microchannels, a comprehensive review of CO_2 flow boiling heat transfer and two-phase flow for both macrochannel and microchannel investigations was presented by Thome and Ribatski [7]. The results showed that CO_2 gives higher heat transfer coefficients than those of conventional refrigerants. At low/moderate vapor qualities, the heat transfer coefficient increases with both saturation temperature and heat flux being almost independent of mass velocity. Ducoulombier et al. [8] studied carbon dioxide two-phase flow pressure drops in a single horizontal stainless steel microtube having the inner diameter of 0.529 mm. Experiments were carried out in adiabatic conditions for four saturation temperatures of – 10; – 5; 0; 5^o C and mass fluxes ranging from 200 to 1400 kg/m^{2}s. The apparent viscosity of the two-phase mixture was larger than the liquid viscosity at low vapor qualities, namely at the lowest temperatures. Cheng et al. [9] updated flow pattern map for CO_2 evaporation inside tubes. The updated map was applicable for a wider range of conditions: tube diameters from 0.6 to 10 mm, mass velocities from 50 to 1500 kg/m^{2}s, heat fluxes from 1.8 to 46 kW/m^2, and saturation temperatures from – 28 to +25 ^{o}C. The new CO_2 two-phase flow pressure drop model predicted that the CO_2 pressure drop was better than the former methods. Boiling heat transfer of carbon dioxide inside a small-sized microfin tube was investigated by Dang et al. [10]. The experimental results indicated that heat flux has a significant effect on the heat transfer coefficient and the coefficient does not always increase with mass flux. In addition, the experimental results also shown that using microfin tubes may considerably increase the overall heat transfer performance.

Dang et al. [11] investigated the heat transfer and pressure drop phenomena of the microchannel and minichannel heat exchangers, both numerically and experimentally. The results obtained from this study indicated that the heat transfer rate obtained from microchannel heat exchanger was higher than those obtained from the minichannel heat exchangers; however, the pressure drops obtained from the microchannel heat exchanger were also higher than those obtained from the minichannel heat exchangers. In addition, at the same average velocity of water in the channels used in this study, the effectiveness obtained from the microchannel heat exchanger was 1.2 to 1.53 times of that obtained from the minichannel heat exchanger. However, in this study, the pure water was the working fluid; they did not study CO_2 in these studies. A published review by Dario et al. [12] summarized the two-phase flow distribution in parallel channels with macro and micro hydraulic diameters. The investigation allowed identifying the main geometrical and operating conditions which influenced the two-phase flow distribution in parallel channels. Yu et al. [13] studied the two-phase flows in microchannels. The results showed that two phase flows have many advantages in heat and mass transfer, comparing to single-phase flows in microchannels. The heat transfer characteristics of R410A in microchannels were measured by Yun et al. [14]. From the studies in [11-14], the microchannel heat exchangers revealed distinctive fluid flow and heat transfer characteristics to compare with the conventional heat exchangers.

Regarding to CO_2 air-conditioning systems, the reducing size of the components was performed by Kim et al. [15]. With technical advantages, the carbon dioxide could be highlighted its high heat transfer coefficients in the supercritical region and its high pressure levels combined with low specific volumes. A study on minimizing COP loss from optimal high pressure correlation for transcritical CO_2 cycle was performed by Yang el at. [16]. In this study, the optimal high pressure correlations using simple curve-fitting method were widely applied in maximizing COP of transcritical CO_2 cycles or systems. Baheta et al. [17] simulated the performance of transcritical carbon dioxide refrigeration cycle by using EXCEL program. In this study, the highest COP was 3.24 at the cooler pressure of 10 MPa. The results indicated that COP increases as rising the evaporative temperature. However, the investigations in [16, 17] did not experimentally perform. Cabello et al. [18] compared four correlations of optimal high pressure from literature reviews with experimental data. The paper was concluded that a small error in pressure could cause a big reduction in COP. However, the investigations in [18] experimentally performed but the COP is very small. A numerical analysis using the finite volume method on a microchannel evaporator for CO_2 air-conditioning systems was fulfilled by Yun et al. [19]. The performance of the microchannel evaporator for CO_2 systems can be improved by varying the refrigerant flow rate to each slab and changing fin space to increase the two-phase region in the microchannels. A design optimization of CO_2subcooler/condenser in a refrigeration system was done by Ge et al. [20]. In this study, the design optimization of the heat exchanger dealt with different structures, controls and system integration at different operating conditions in order to significantly enhance the performance in a CO_2 refrigeration system. As a result, the effect of heat exchanger sizes on the system performance can be enhanced with fan speed controls. In this study, the boiling heat transfer coefficients of R410A in microchannels were much higher than single tubes at the same conditions. Based on the finite volume method, Jin et al. [21] studied an analysis/computer model to predict the performance of an evaporator for a CO_2 mobile air-conditioning system. Despite the pressure drops due to header and port inlets were considered in this work, they can be neglected due to the small proportion, comparing with the total pressure drop of evaporator. In addition, Kim and Bullard [22] investigated the development and verification of a heat exchanger model for evaluating the thermal performance of an evaporator for a CO_2 mobile air-conditioning system. In the study, the predicted model can be used for the performance analysis and the design of a microchannel evaporator. Carbon Dioxide also studied in [23, 24]; however, the main thermodynamic points of the cycle did not indicate clearly.

From the literature reviews above, the studies did not deal with the subcooling process in more detail and the COP values were not high enough. So, it is important to study on subcooling of a CO_2 air conditioning system to enhance their COP. In the following sections, an experimental study on subcooling process of a CO_2 air conditioning system will be done with transcritical mode. There are two subcoolers have been used in this test loop.

EXPERIMENTAL SETUP

Figure 1 indicates the experimental test loop for CO_2 air conditioner. This system has five main components: a CO_2 compressor, a gas cooler, a subcooler, a thermal expansion valve, and a microchannel evaporator. The CO_2 refrigerant enters the compressor in superheated vapor state and then it is compressed to a higher pressure corresponding higher temperature state. The superheated vapor is sent to a cooler where it is cooled by flowing inside tubes, the liquid refrigerant is then sent to a subcooler where it is subcooled by the surrounding air and finally it is sent to an expansion valve where it becomes wet saturated vapor and its pressure is decreased dramatically. The mixture has lower temperature than the temperature of the enclosed space and it is routed through the channels in the evaporator to cool air. The saturated vapor from the evaporator is superheated and is routed back into the compressor to complete a cycle. For five main thermodynamic points of this cycle, five temperature sensors and four pressure gauges were installed to get these parameters. In this study, the CO_2 air conditioning cycle was done with transcritical mode.

The dimensions of the microchannel evaporator are shown in Figure 2. The material for this heat exchanger is aluminum, used as a substrate with the thermal conductivity of 237 W/(m°C), density of 2,700 kg/m^3, and specific heat at constant pressure of 904 J/(kg°C). The evaporator has six passes with 29 microchannels. Each microchannel is rectangular in shape, with the width of 1.2 mm and the depth of 600 μm. The total heat transfer area of this microchannel evaporator is 2.5 m^2. The design cooling capacity for this

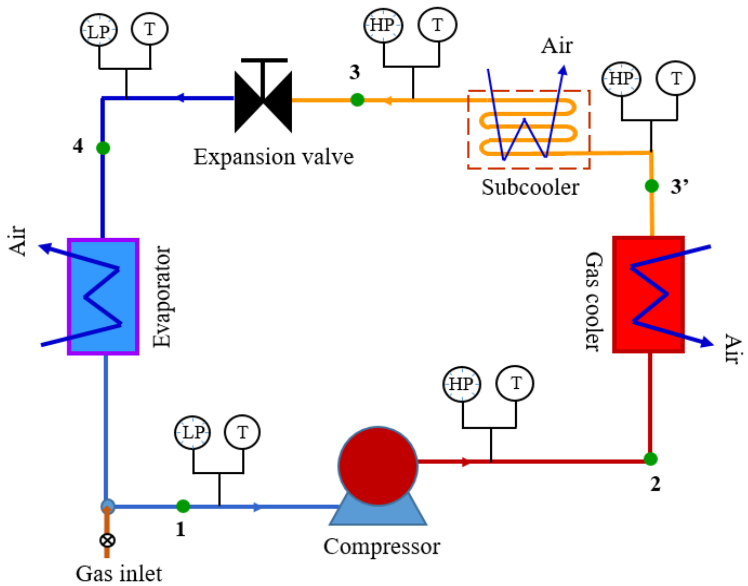


Figure 1. The experimental test loop for CO_2 air conditioning cycle

microchannel evaporator is 2700 W. The cooler and evaporator were tested with the hydraulic testing method. The former and the latter did not tear or deform at the pressure of 150 bar and 90 bar, respectively [25].

There are two subcoolers used in this test loop. Geometric parameters of the devices are listed in Table 1 and Figure 3. Copper tubes with diameter of 6.4 mm were used to manufacture for the subcoolers. The heat transfer area of the subcooler 2 (S2) is two times higher than that of the heat transfer area of the subcooler 1 (S1). In this study, the air in the space to be cooled was used to subcool the refrigerant in the subcoolers.

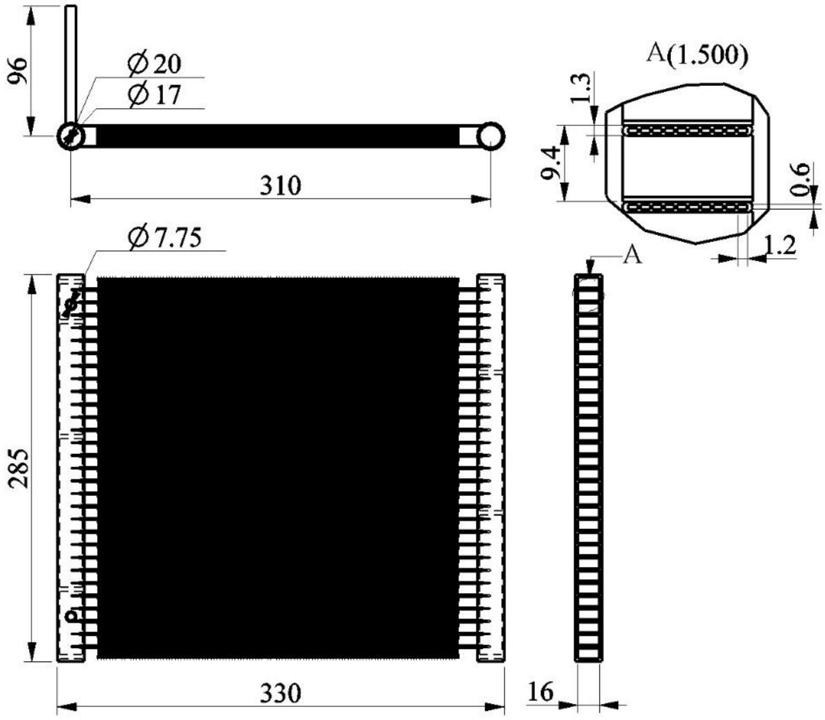
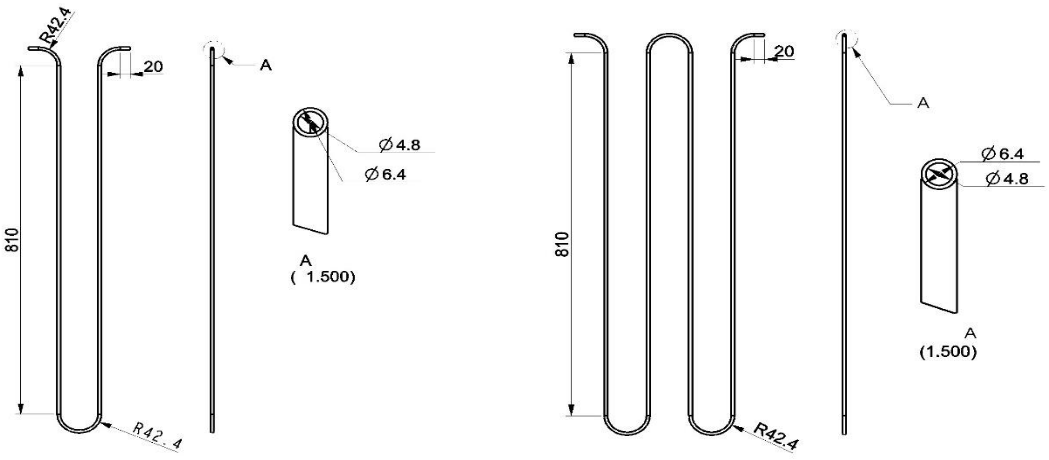


Figure 2. Dimensions of the microchannel evaporator

Table 1. Geometric parameters of the subcoolers

No.	Pass number	Heat transfer area (m^2)
S1	2	0.034
S2	4	0.068



a) Subcooler1 (S1) b) Subcooler2 (S2)

Figure 3. Dimensions of the subcoolers

The equipments used for the experiments as follows:

- Thermocouples, T-type (with the sensor diameter of 300 μm)
- Thermostat, EW – 181 H, made by Ewelly
- Infrared thermometer, Raynger@ST, made by Raytek
- Thermal camera, Fluke Ti9, made by Fluke, USA
- Pressure gauge, made by Pro – Instrument
- Clamp meter, Kyoritsu 2017, made by Kyoritsu.

Thermocouples were pasted at the five main points of the cycle. The data acquisition system for recording the electronic signals was implemented to obtain data from thermocouples (T100); the system was integrated through instant monitoring software to record and analyze the data received, as shown in Figure 4. To increase the exactness of the temperature data, the infrared thermometer and thermal camera were used to duple check in this study. The thermostat EW-181H was used to record the room temperature. The five pressure gauges were installed in this test loop to obtain the pressure data.Accuracies and ranges of testing apparatus are listed in Table 2.

Table 2. Accuracies and ranges of testing apparatuses [25]

Testing apparatus	Accuracy	Range
Thermocouples (T-type)	± 0.1 °C	0 ~100 °C
Thermal camera	2 %	-20~250 °C
Infrared thermometer	± 1 °C of reading	- 32 ~ 400 °C
Pressure gauge	± 1 FS	0~100 kgf/cm^2
Clamp meter	± 1.5 % rdg	0 ~ 200 A

Figure 4. A photo of the experimental system
(1-Expansion valve; 2-Microchannel evaporator; 3-Subcooler; 4-Cold room; 5-Data acquisition system; 6-Gas cooler; 7- Compressor)

Governing equations

To analyze the thermodynamic parameters of the CO_2 air conditioning system and, the governing equations were given below:

The heat transfer rate for subcooler was calculated as

$$q_{3'-3} = c_p\left(T_{3'} - T_3\right) \tag{1}$$

The power consumption was determined using

$$w_{1-2} = \left(h_2 - h_1\right) \tag{2}$$

The isenthalpic process was presented by

$$h_3 = h_4 \tag{3}$$

The heat transfer rate for evaporator was calculated as

$$q_{4-1} = (h_1 - h_4) \tag{4}$$

The COP of the cycle (ignoring the heat transfer of subcooler) was quantified by

$$COP = \frac{q_{4-1}}{w_{1-2}} \tag{5}$$

RESULTS AND DISCUSSION

The experiments of the CO_2 air conditioner were done with transcritical mode. The two subcoolers were used in turn to get the main thermodynamic parameters. The Dorin compressor with Model CD 180H was used in this system. The compressor was connected with the Voltage of 380V, phases of 3, and frequency of 50Hz.

The experimental results of the subcooler S1

The CO_2 air conditioning system with S1 was experimented more times to record data. The results obtained from this experiment are very stable. Experimental data for the air conditioning system were obtained under the ambient temperature of 29.5 ºC. Table 3 shows the experimental points of the cycle for S1, at 44 bar and 75 bar for evaporator and subcooler pressures, respectively. The points were plotted on the p-h diagram, using EES software as shown in Figure 5 (EES is an acronym for Engineering Equation Solver). With the tube inner diameter of 4.8 mm and the length of 1.7 m, the pressure drop of the subcooler S1 obtained in these experiments is negligible small.

Table 3. The experimental points of the cycle for S1

I (A)	p1 (bar)	t1 (ºC)	p2 (bar)	t2 (ºC)	p3' (bar)	t3' (ºC)	p3 (bar)	t3 (ºC)	p4 (bar)	t4 (ºC)
2.4	41.5	9.7	75	56	75	31.7	75	28.6	44	9

(where I is current, p is pressure and t is temperature)

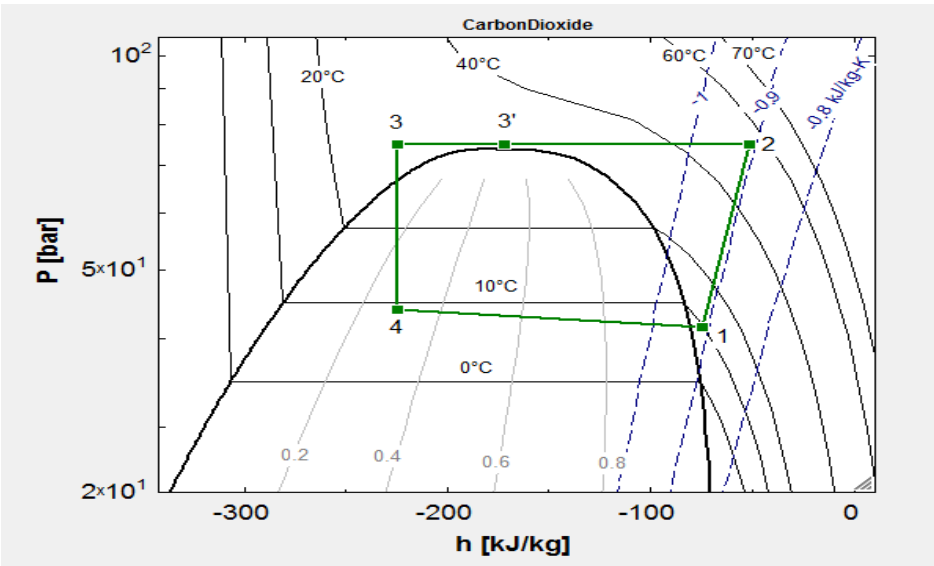


Figure 5. The thermodynamic points of the cycle on p-h diagram for S1

Table 4. Enthalpies of the main points in Table 3 and Figure 5

h_1 (kJ/kg)	h_2 (kJ/kg)	h_3 (kJ/kg)	h_4 (kJ/kg)	q_{4-1} (kJ/kg)	w_{1-2} (kJ/kg)	COP
-74.47	-51.86	-224.8	-224.8	150.3	22.6	6.6

The results in Table 3 show that the subcooling temperature reduces 3.1 ºC. Besides, the pressure drops of the microchannel evaporator are so high. The pressure difference between the inlet of evaporator and the compressor suction port is also due to the suction force of the compressor. This pressure difference causes an increase both the evaporator cooling capacity and the compressor power, as shown in Figure 5; however, the COP of this cycle is no more changes. From Figure 5 and Table 4, the cooling capacity is 150.3 kJ/kg, the compressor power is 22.6 kJ/kg, resulting the COP is 6.6. Compared with Daikin residential air conditioners (ACs) [26], the FTKV25NVMV (using R32 refrigerant) and FTKV71NVMV (using R410 refrigerant) serials provide a high COP of 4.63 and 3.55, respectively. All models have received Singapore's 5 Tick Energy Label. With Trane residential air ACs, the COP is only around 2.7 – 3.0 for present products [27]. Compared with other commercial ACs, it is indicated that the COP in this study is higher than the COPs from the conventional air conditioning system.

The experimental results of the subcooler S2

Experimental data for the air conditioning system with S2 were obtained under the ambient temperature of 30 ºC. It is observed that the subcooling temperature of S2 is double the subcooling temperature of S1, it agrees with the heat transfer area, as shown in Table 5 and Figure 6. In addition,the pressure drop of the subcooler S2 obtained in the experiments is negligible small by the tube length of 3.4 m only. From Figure 6 and Table 6, the cooling capacity is 160.6 kJ/kg, the compressor power is 22.3 kJ/kg, resulting the COP is 7.2. The results indicated that the subcooling temperature reduces from 28.6 to 26 ºC; the cooling capacity increases 10.3 kJ/kg. From Tables 3 and 5, it is shown that the compressor power using two subcoolers is the same; the currents of system are 2.4 A for two cases. They indicated the agreement between the results on p-h diagram and the absorbed current.

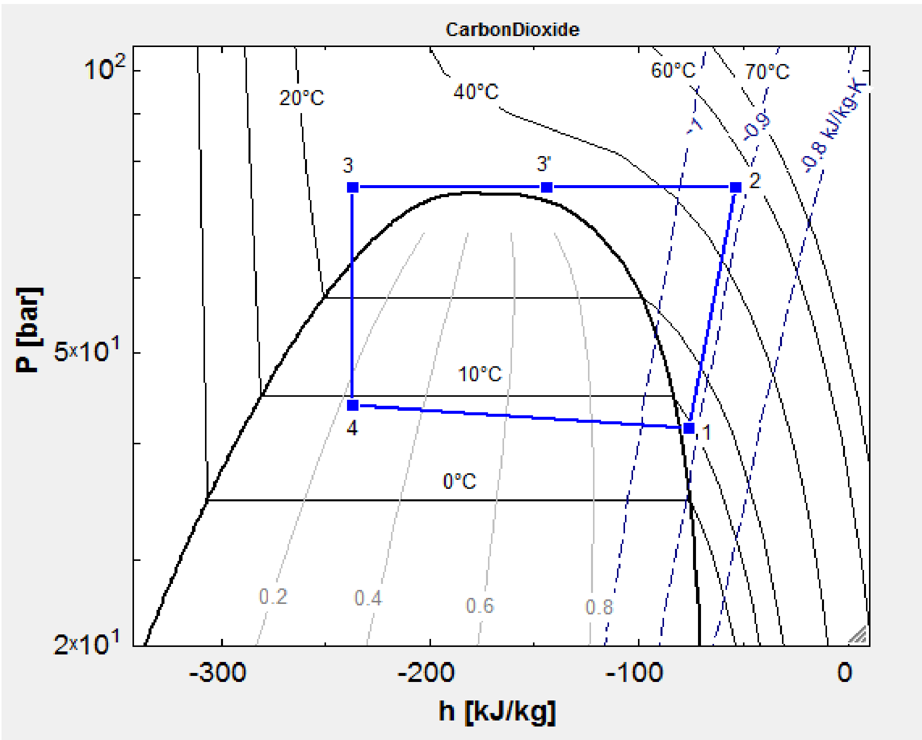


Figure 6. The thermodynamic points of the cycle on p-h diagram for S2

Table 5. The experimental points of the cycle for S2

I (A)	p1 (bar)	t1 (ºC)	p2 (bar)	t2 (ºC)	p3' (bar)	t3' (ºC)	p3 (bar)	t3 (ºC)	p4 (bar)	t4 (ºC)
2.4	41.5	8.7	75	55.5	75	32	75	26	44	9

Table 6. Enthalpies of the main points in Table 5 and Figure 6

h_1 (kJ/kg)	h_2 (kJ/kg)	h_3 (kJ/kg)	h_4 (kJ/kg)	$q_{4\text{-}1}$ (kJ/kg)	$w_{1\text{-}2}$ (kJ/kg)	COP
-76.51	-54.22	-237.2	-237.2	160.6	22.3	7.2

A comparison between the present study and other literatures is indicated in Figure 7. In [17] the COP of 3.9 was achieved at 40 bar and 100 bar evaporator and subcooler pressures, respectively. From Table 1 in [28], the COP of 3.24 was achieved at 34.9 bar and 90 bar evaporator and subcooler pressures, respectively. From Figure 7, it is observed that the compressor power in [17] and [28] is higher than that obtained from the present study; while the cooling capacity in [17] and [28] is lower than that obtained from the present study. In addition, the main thermodynamic points of the cycle in this study have not published by other literature reviews; almost the literature reviews studied with the gas cooler pressure around 90 – 100 bar. Finding new approaches to enhance COP, the results [18] were only published with COP < 2; the results in [29] were achieved with COP = 2.08. From above comparisons, the results in present study are higher than other reviews. In this study, a new approach carried out; authors did not use an internal heat exchanger to transfer heat from the hot refrigerant in the gas cooler discharge line to the cold refrigerant in the compressor suction line. The subcoolers were installed in the cold room (the expansion device was also installed in the cold room, as shown in Figure 4). The hot refrigerant in the gas cooler discharge line is cooled by the air in the cold room (the superheating of the cold refrigerant in the compressor suction line is not increased). The experimental results indicated that the cooling reduction of the cold room is negligible small as transferring heat for the subcooler (because the refrigerant state in the subcooler is superheat). Figure 8 shows the temperature profile of expansion valveobtained from the thermal camera. The temperature data are the same with those obtained from the thermocouples.

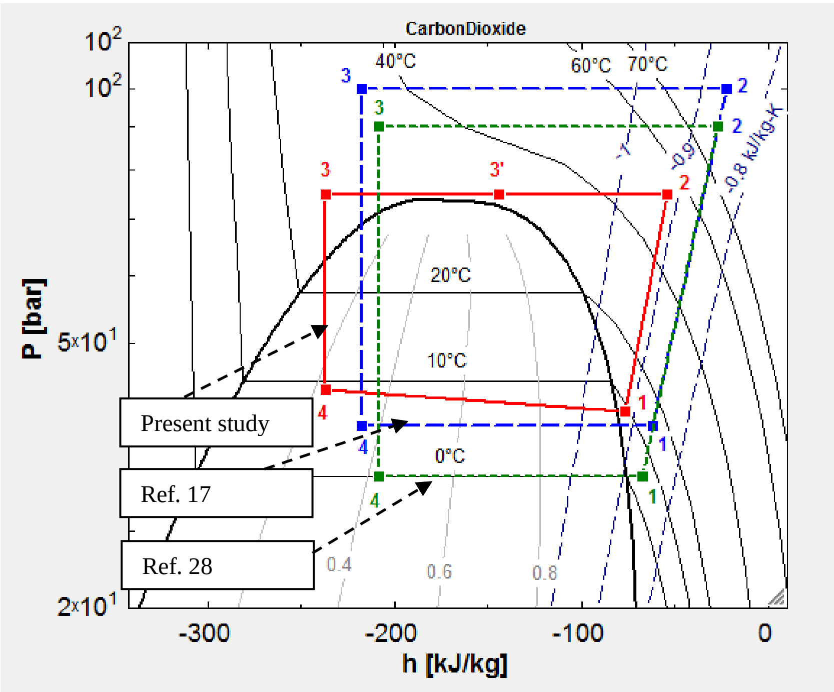


Figure 7. A comparison between the present study and the other literatures

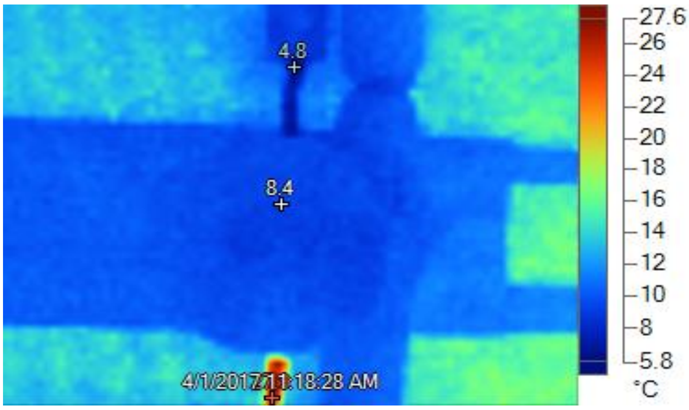


Figure 8. Temperature profile of expansion valve

CONCLUSIONS

An experimental investigation on subcooling process of a CO_2 air conditioning system was done with transcritical mode. In this test loop, there are two subcoolers used in turn, these two heat exchangers were installed in the cold room, the hot refrigerant in the gas cooler discharge line is cooled by the air in the cold room.

The experimental points were plotted on the p-h diagram, using EES software. Based on the experimental results, the conclusions can be summarized as follows:

1. The main thermodynamic points of the cycle in this study are new to achieve high COP with the gas cooler pressure of 75 bar; they have not published by other literature reviews.

2. With the subcooler S1, the COP of 6.6 achieved at the evaporator pressure of 44 bar, the subcooler pressure of 75 bar, the subcooling temperature of 28.6 ºC, and the compressor current of 2.4 A. With the subcooler S2, the COP of 7.2 achieved at the evaporator pressure of 44 bar, the subcooler pressure of 75 bar, the subcooling temperature of 26 ºC, and the compressor current of 2.4 A.

3. A COP comparison between the present study and the other literatures was indicated; the results in present study are higher than other reviews (almost literature reviews achieved COP < 5 and the gas cooler pressure around 80 – 100 bar).

ACKNOWLEDGEMENTS

The supports of this work by the project No. 35/2015/HĐ-SKHCN (sponsored by HCMC Department of Science and Technology, Vietnam) are deeply appreciated.

NOMENCLATURE:

I	compressor current (A)
p	Pressure (bar)
t	Temperature (*ºC)*
h	Specific enthalpy (kJ/kg)
c_p	Specific heat at constant pressure (kJ/kg.K)
q_{4-1}	Cooling capacity (kJ/kg)
$q_{3'-3}$	Heat transfer rate for subcooler (kJ/kg)
w_{1-2}	Power consumption (kJ/kg)
COP	Coefficient of Performance

REFERENCES

[1] G. Lorentzen, The use of natural refrigerants: a complete solution to the CFC/HCFC predicament, Int. J. Refrig, 18, 1995,190 – 197.
[2] GouhuaKuang, Michael Ohadi, SerguieDessiatoun, Semi-Empirical Correlation of Gas Cooling Heat Transfer of Supercritical Carbon Dioxide in Microchannels, HVAC&R Research, 14, 2008, 861 – 870.
[3] D. Colorado a , J. A. Hernández, W. Rivera, "Comparative study of a cascade cycle for simultaneous refrigeration and heating ammonia, R134a, butane, propane, and CO_2 as working fluids, International Journal of Sustainable Energy, 31, 2012, 365 – 381.
[4] D. B. Tuckerman, R.F.W. Pease, High-performance heat sinking for VLSI, IEEE Electron device letters, 2, 1981, 126 – 129.

[5] Xiumin Zhao, P.K. Bansal, Flow boiling heat transfer characteristics of CO_2 at low temperatures, International Journal of Refrigeration, 30, 2007, 937 – 945.
[6] Lixin Cheng, John R. Thome, Cooling of microprocessors using flow boiling of CO_2 in a micro-evaporator: Preliminary analysis and performance comparison, Applied Thermal Engineering, 29, 2009, 2426 – 2432
[7] John R. Thome and GherhardtRibatski, State-of-the-art of two-phase flow and flow boiling heat transfer and pressure drop of CO_2 in macro- and micro-channels, International Journal of Refrigeration, 28, 2005, 1149 – 1168
[8] MaximeDucoulombier, StéphaneColasson, Jocelyn Bonjour, Philippe Haberschill, Carbon dioxide flow boiling in a single microchannel – Part I: Pressure drops, Experimental Thermal and Fluid Science, 35, 2011, 581 – 596
[9] Lixin Cheng, GherhardtRibatski, Jesús Moreno Quibén, John R. Thome, New prediction methods for CO_2 evaporation inside tubes: Part I – A two-phase flow pattern map and a flow pattern based phenomenological model for two-phase flow frictional pressure drops, International Journal of Heat Mass Transfer, 51, 2008, 111-124.
[10] Chaobin Dang, NoboriHaraguchi, EijiHihara, Flow boiling heat transfer of carbon dioxide insidea small-sized microfin tube, International Journal of Refrigeration, 33, 2010, 655 – 663.
[11] Thanhtrung Dang and Jyh-tong Teng, Comparisons of the heat transfer and pressure drop of the microchannel and minichannel heat exchangers, Heat Mass Transfer, 47, 2011, 1311 – 1322.
[12] E.R. Dario, L. Tadrist, J.C. Passos, Review on two-phase flow distribution in parallel channels with macro and micro hydraulic diameters: Main results, analyses, trends, Applied Thermal Engineering, 59, 2013, 316 – 335.
[13] Zhao Yu, Orin Hemminger, Liang-Shih Fan, Experiment and lattice Boltzmann simulation of two-phase gas–liquid flows in microchannels, Chemical Engineering Science, 62, 2007, 7172 – 7183
[14] Rin Yun, Jae Hyeok. Heo, Yongchan Kim, Evaporative heat transfer and pressure drop of R410A in microchannels, International Journal of Refrigeration, 29, 2006, 92 – 100
[15] Manhoe Kim, JosteinPettersen, Clark W. Bullard, Fundamental process and system design issues in CO_2 vapor compression systems. Progress in Energy and Combustion Science, 30, 2004, 119 – 174.
[16] Liang Yang, Hui Li, Si-weiCai, Liang-liang Shao, Chun-luZang, Minimizing COP loss from optimal high pressure correlation for transcritical CO_2 cycle" Applied Thermal Engineering, 89, 2015, 656 – 662.
[17] AkliluTesfamichaelBaheta, Suhaimi Hassan, AllyaRadzihan B. Reduan, Abraham D. Woldeyohannes, Performance investigation of transcritical carbon dioxide refrigeration cycle, Procedia CIRP, 26, 2015, 482 – 485.
[18] R. Cabello, D. Sa´nchez, R. Llopis, E. Torrella, Experimental evaluation of the energy efficiency of a CO_2 refrigerating plant working in transcritical conditions, Applied Thermal Engineering, 28, 2008, 1596 – 1604.
[19] Rin Yun, Yongchan Kim, Chasik Park, Numerical analysis on a microchannel evaporator designed for CO_2 air-conditioning systems, Applied Thermal Engineering, 27, 2007, 1320 – 1326.
[20] Y.T. Ge , S.A. Tassou, I. DewaSantosa, K. Tsamos, Design Optimisation of CO_2Subcooler/Condenser in a Refrigeration System, Energy Procedia,61, 2014, 2311 – 2314.
[21] Jifeng Jin, Jiangping Chen, Zhijiu Chen, Development and validation of a microchannel evaporator model for a CO_2 air-conditioning system, Applied Thermal Engineering, 31, 2011, 137 – 146.
[22] Man-Hoe Kim, Clark W. Bullard, Development of a microchannel evaporator model for a CO_2 air-conditioning system, Energy, 26, 2001, 931 – 948.
[23] Paul Maina and Zongjie Huan, Effects of Various Parameters on the Efficiency of a CO_2 Heat Pump: A Statistical Approach, Journal of Thermal Engineering, Vol. 1, Issue 4, 2015, pp. 236-278
[24] Chi-Chuan Wang, Chen-Xi Zhu, and Yi-Chun Tang, Performance and Flow Distribution of the Plate Heat Exchanger with Supercritical Fluid of Carbon Dioxide, Journal of Thermal Engineering, Vol. 1, Issue 3, 2015, pp. 143-151.
[25] Tankhuong Nguyen, Tronghieu Nguyen, Thanhtrung Dang, and Minhhung Doan, An experiment on a CO_2 air conditioning system with Copper heat exchangers, International Journal of Advanced Engineering, Management and Science, Vol. 2, 2016, 2058-2063.
[26] http://www.daikin.com.vn/en/products/residential/split/ftkv/index.html
[27]http://www.tranevietnam.com/en/Products/Dx-Unitary-System/Residental-Split-Air-Conditioners/Premio-New-High-Wall-Type-Air-Conditioner/
[28] Young Chul Kwon, DaeHoon Kim, Jae Heon Lee, Jun Young Choi and Sang Jae Lee, Experimental study on heat transfer characteristics of internal heat exchangers for CO_2 system under cooling condition, Journal of Mechanical Science and Technology Vol. 23, 2009, 698 – 706.
[29] Jae Seung Lee, Mo Se Kim, and Min Soo Kim, Studies on the performance of a CO2 air conditioning system using an ejector as an expansion device, International Journal of Refrigeration, Vol. 38, 2014, pp. 140-152.