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# Experiments on Expansion and Superheat Processes of a $Co_2$ Cycle Using Microchannel Evaporator

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**ABSTRACT:** *Experiments on expansion and superheat processes of a transcritical $CO_2$ air conditioning system were done. In this study, the cross-sectional area of the expansion valve reduces from 8.195 to 0.091 $mm^2$, the cooler pressure increases and the evaporator pressure decreases; the pressure difference between cooler and evaporator increases. Besides, the power input of compressor increases as decreasing the cross-sectional area. It is also observed that the pressure difference and power input strongly increase as the cross-sectional area is less than 0.4 $mm^2$. The cooler pressure curve and the power input curve are the same rule as varying the cross-sectional area. Moreover, the evaporating temperature decreases from 18.4 to 7.3 ℃ and the superheat decreases from 3.4 to 1.1 ℃ as the cross-sectional area reduces from 3.825 to 0.091 $mm^2$. In addition, the cooling capacity of this cycle is 168.45 kJ/kg, the compressor power is 23.99 kJ/kg, resulting the COP is 7.01; the COP is higher than those obtained from other literature reviews.*

**Keywords:** *Experiment, $CO_2$ refrigerant, expansion, superheat, evaporator, microchannel.*

## I. INTRODUCTION

The environmentally friendly refrigerants and high effectiveness heat exchangers are interesting scientists. With refrigerant field, Zhao and Bansal [1] experimentally investigated for the flow of Carbon dioxide ($CO_2$) boiling in microchannels, with the vapor quality is from 0.05 to 0.3. They concluded that the heat transfer coefficient of $CO_2$ at around 30 °C is found to vary between 4000 and 7500 W/($m^2$.K) at different vapour qualities. 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. The use of natural refrigerants is a complete solution the CFC/HCFC predicament was made by Lorentzen [2]. In this study, the $CO_2$ air conditioning system saved about 20% energy to compare with the use of R12 refrigerant at the same temperature. With technical advantages [3], 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. Chen et al. [4] analyzed and optimized a hybrid $CO_2$ transcritical mechanical compression – ejector cooling cycle. The hybrid cooling cycle is a combination of a $CO_2$ transcritical mechanical compression refrigeration machine (MCRM) powered by electricity and an ejector cooling machine (ECM) driven by heat rejected from the $CO_2$ cooling cycle. Refrigerants R245ca, R601b (Neopentane) and R717 (Ammonia) were investigated as the working fluids. In this study, using the ejector cooling cycle for subcooling the $CO_2$ gas after gas cooler allows increasing the efficiency of the $CO_2$ transcritical cooling cycle up to 25-30%, depending on the refrigerant type of the ejector cooling cycle. Kuang et al. [5] studied a semi-empirical correlation of gas cooling heat transfer of supercritical carbon dioxide in microchannels. Based on the 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/$m^2$s.

Lee et al. [6] studied on the performance of a CO2 air conditioning system using an ejector as an expansion device. The cooling capacity and Coefficient of Performance (COP) in the air conditioning system using an ejector are higher than those in the conventional system at an entrainment ratio greater than 0.76. Baheta et al. [7] 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. A study on minimizing COP loss from optimal high pressure correlation for transcritical $CO_2$ cycle was performed by Yang el at. [8]. In this study, the optimal high pressure correlations using simple curve-fitting method were widely applied in maximizing COP of transcritical

$CO_2$ cycles. Cabello et al. [9] 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. A comparative study of a cascade cycle for simultaneous refrigeration and heating operating with ammonia, R134a, butane, propane, and $CO_2$ as the working fluids was presented by Colorado et al. [10]. 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 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 [6] experimentally performed but the COP is very small and the investigations in [4,5,7] did not experimentally perform.

Regarding to the microchannel heat exchangers, 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. Barlak et al. [12] also conducted experiments on microtubes. Results showed that at low Reynolds number (Re <2000), the pressure drop is less dependent on the ratio between the length and diameter (L / D) and a linear relationship between the pressure drop coefficient and Reynolds. While, the rising generation of high Reynolds number (Re> 2000), the pressure drop is much dependent on the ratio L / D. Schael and Kind [13] studied the flow patterns and heat transfer characteristics of $CO_2$ in the micro fin tube and compared with smooth pipes. The results showed that the thermal conductivity in microchannel fins tube is significantly higher than the smooth tube.

Using numerical simulation, Cheng and Thome [14] 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 coolant for microprocessors at low operating temperatures but also presented a great technological challenge as other new cooling technologies. Gasche [15] experimentally investigated the evaporation of $CO_2$ inside the tubular microphone channel with a diameter of 0.8mm. The average heat transfer coefficient of 9700 W/($m^2$.°C) was achieved with a standard error 35%. For the vapor quality is low (less 0.25), the slow flow is predominant; whereas, for the vapor quality is high (over 0.50), the annular flow is better. From the studies in [1-15], the microchannel heat exchangers revealed distinctive fluid flow and heat transfer characteristics to compare with the conventional heat exchangers. Dang et al. [16, 17] investigated the $CO_2$ air conditioning systems. The results showed that conventional compressor is not suitable for using high pressure. The vapor quality increases from 0.50 to 0.52 when the $CO_2$ refrigerant enters the evaporator at position of 200mm, with the cooler pressure of 85 bar, evaporator pressure of 37 bar, and the $CO_2$ flow rate of 5.2 g/s. However, the expansion and superheat processes did not mention in [16, 17].

Regarding to two-phase flow in microchannels, Pettersen [18] studied about two-phase flow in microchannel tubes using $CO_2$ refrigerant. The results showed that the heat transfer is much influenced by the vapor quality, especially in volume flow and temperature. 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 [19]. 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. [20] 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.

From the literature reviews above, the studies did not deal with the expansion and superheat processes of a $CO_2$ air conditioning system using microchannel evaporators. In addition, their COP values were not high enough. So, it is important to study on expansion and superheat processes of the $CO_2$ cycle using microchannel evaporator to enhance their COP.

## II. EXPERIMENTAL SETUP

The experimental test loop for $CO_2$ air conditioning system using a microchannel evaporator is indicated in Fig. 1 [16]. The $CO_2$ refrigerant enters the compressor and then it is compressed to a higher pressure. The superheated vapor is routed through a cooler and is cooled by flowing inside tubes.

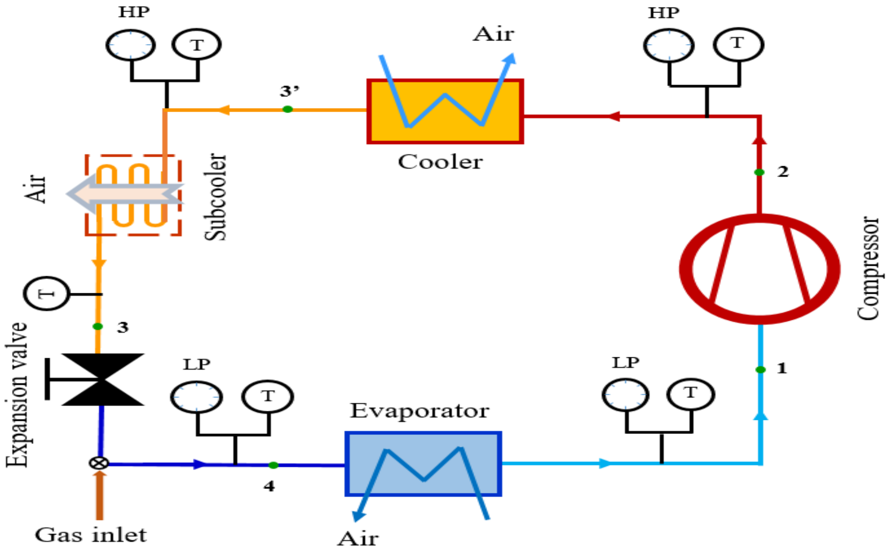


**Fig. 1** The test loop for $CO_2$ air conditioning cycle

The liquid refrigerant continues to move to a subcooler. Next, the liquid subcooled refrigerant continues to move to an expansion valve; the refrigerant pressure is dropped dramatically because of the expansion process. After flowing through the expansion valve, the refrigerant is routed through the channels in the evaporator to cool air. The saturated vapor from the evaporator is routed back into the compressor to complete the cycle. There are five temperature sensors and four pressure gauges were installed to get these parameters. A compressor of Dorin with model CD 180H was used in this system. A copper tube cooler was designed and manufactured with heating capacity of 6 kW. The cooler was tested at the pressure of 150 bar [16]. The dimensions of the microchannel evaporator are shown in Fig. 2. The design cooling capacity for this microchannel evaporator is 2700 W. The microchannel evaporator was tested; it did not tear or deform at the pressure of 90 bar [17].

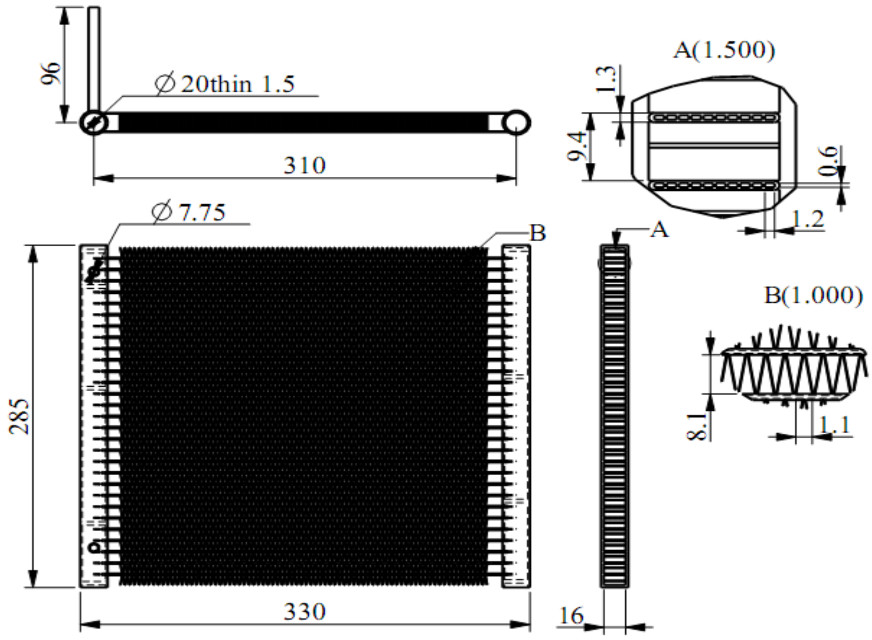


**Fig. 2** Dimensions of the microchannel evaporator [17]

Fig. 3 shows basic dimensions of the expansion device. This valve has the inner diameter of 4 mm, coresponding with the cross-sectional area of 12.57 mm$^2$. This valve has 5.75 rounds: the valve is closing from 5.75 to 0.00 rounds, corresponding with the area is from 12.57 to 0.00 mm$^2$. The apparatuses used for the experiments such as infrared thermometer, thermal camera, thermocouples, thermostat, pressure gauge, anemometer, clamp meter, etc. At thermodynamic points of the cycle, the temperature values were recorded and checked by the four types of the thermometers. Their accuracies and ranges are listed in Table 1. Fig. 4 shows a photo of the experimental system.

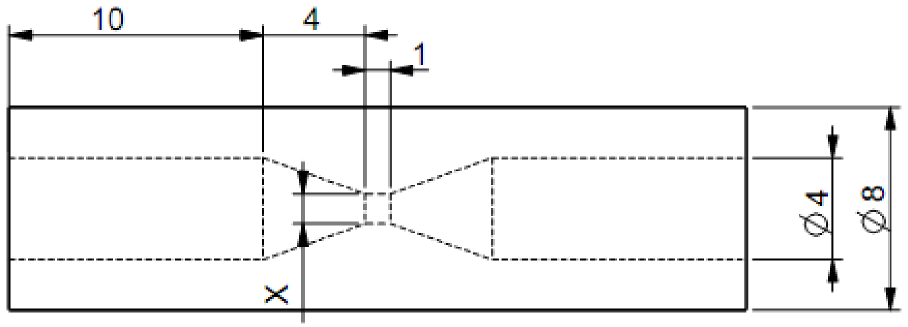


**Fig. 3** Dimensions of the expansion device

**Table 1.** Accuracies and ranges of testing apparatuses [16]

| Testing apparatus | Accuracy | Range |
|---|---|---|
| Thermocouples | ± 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$^2$ |
| Anemometer | ± 3 % | 0 ~ 45 m/s |
| Clamp meter | ± 1.5 % rdg | 0 ~ 200 A |

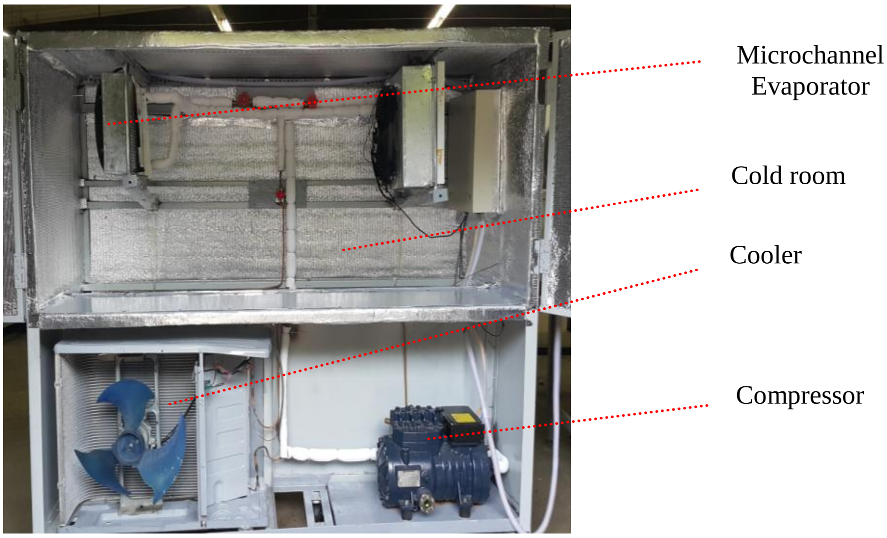


**Fig. 4** A photo of the experimental system

## III. RESULTS AND DISCUSSION

The experiments of the transcritical $CO_2$ air conditioning cycle working with a microchannel evaporator were done under the ambient temperature around 31 °C. Before operating, the pressures of cooler and evaporator were balance at 50 bar. From Fig. 5, it is observed that when the cross-sectional area of the expansion valve reduces from 8.195 to 0.091 mm$^2$, the cooler pressure increases and the evaporator pressure decreases; the pressure difference between cooler and evaporator increases. Besides, the pressure difference clearly indicates as the cross-sectional area is less than 0.4 mm$^2$. The results also indicated that the power input of compressor increases as decreasing the cross-sectional area. It is also observed that the power input strongly increases as the cross-sectional area is less than 0.4 mm$^2$. The Fig. 5 also indicates that the cooler pressure curve and the power input curve are the same rule as varying the cross-sectional area.

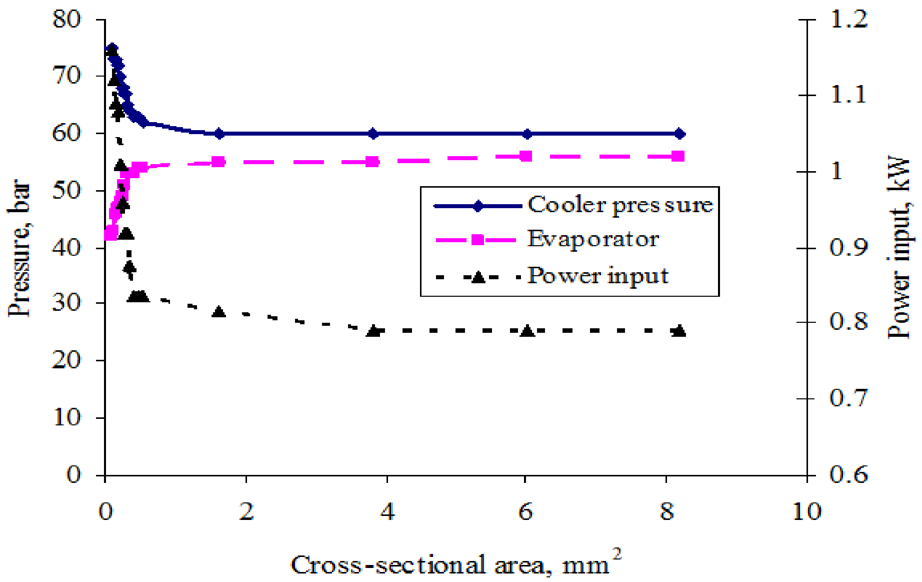


**Fig. 5** Cross-sectional area versus pressure and power input

A relationship between the cross-sectional area and superheat and evaporating temperature is shown in Fig. 6. The results indicated that the evaporating temperature decreases from 18.4 to 7.3 °C and the superheat decreases from 3.4 to 1.1 °C as the cross-sectional area reduces from 3.825 to 0.091 mm$^2$.

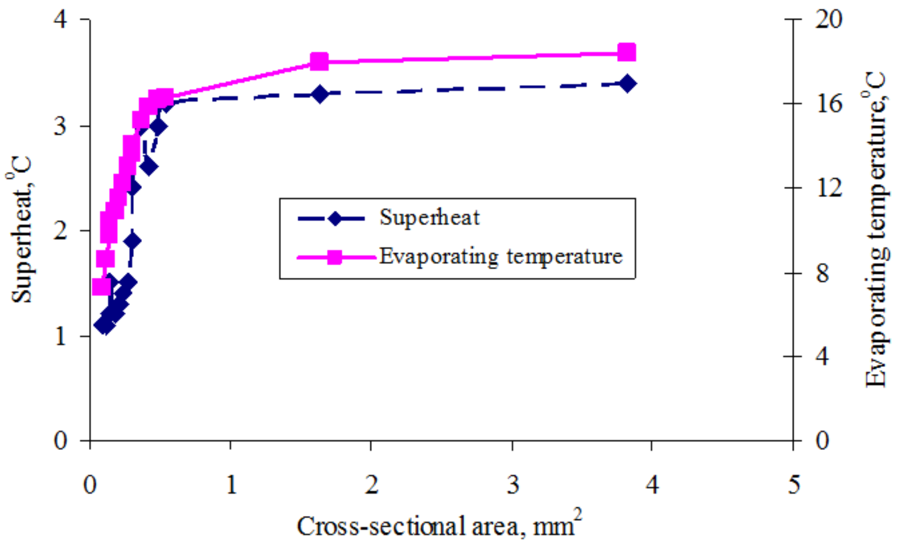


**Fig. 6** Cross-sectional area versus superheat and evaporating temperature

**Table 2.** Thermodynamic parameters of the $CO_2$ cycle

| p1 (bar) | t1 (°C) | p2 (bar) | t2 (°C) | p3 (bar) | t3' (°C) | t3 (°C) | p4 (bar) | t4 (°C) |
|---|---|---|---|---|---|---|---|---|
| 40.6 | 8.4 | 75 | 62.0 | 75 | 32.8 | 25.3 | 42 | 7.3 |

(where p is pressure and t is temperature)

The experimental parameters of the $CO_2$ air conditioning cycle at the cooler pressure of 75 bar are listed in Table 2. From Fig. 7 and Table 2, the cooling capacity is 168.45 kJ/kg, the compressor power is 23.99 kJ/kg, resulting the COP is 7.01. A comparison between the present study and another literature is also shown in Fig. 7. In [7], the COP of 3.24 was achieved at 40 bar and 100 bar evaporator and cooler pressures, respectively. From the figure, it is observed that the compressor power in [7] is higher than that obtained from the present study; while the cooling capacity in [7] is lower than that obtained from the present study. The experimental points were plotted on the p-h diagram, using EES software.

A photo about temperature profile of the microchannel evaporator was taken by a thermal camera, as shown in Fig. 8. The temperature profile of this evaporator is uniform and in good agreement with the evaporating temperature (corresponding with the evaporator pressure).

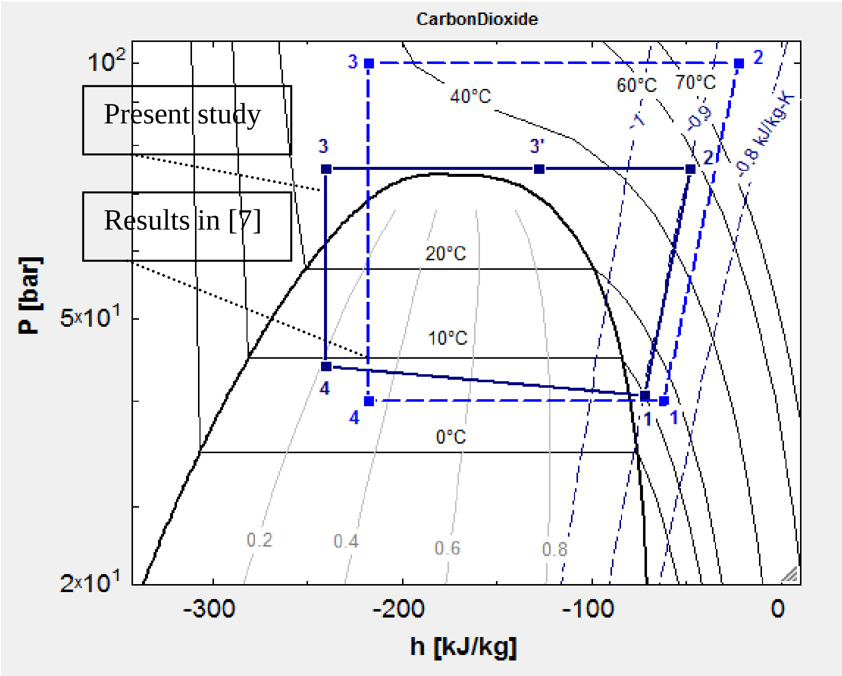


**Fig. 7** A comparison on the p-h diagram

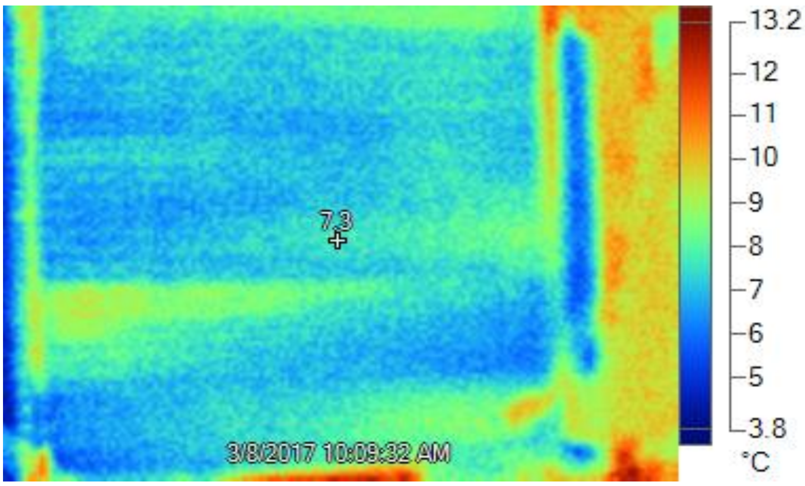


**Fig. 8** A photo of the microchannel evaporator

## IV. CONCLUSION

An experimental investigation on expansion and superheat processes of a transcritical $CO_2$ air conditioning system was done. The experimental points were plotted on the p-h diagram, using EES software.

In this study, the cross-sectional area of the expansion valve reduces from 8.195 to 0.091 $mm^2$, the cooler pressure increases and the evaporator pressure decreases; the pressure difference between cooler and evaporator increases. Besides, the pressure difference clearly indicates as the cross-sectional area is less than 0.4 $mm^2$. The results also indicated that the power input of compressor increases as decreasing the cross-sectional area. It is also observed that the power input strongly increases as the cross-sectional area is less than 0.4 $mm^2$. The cooler pressure curve and the power input curve are the same rule as varying the cross-sectional area. The evaporating temperature decreases from 18.4 to 7.3 °C and the superheat decreases from 3.4 to 1.1 °C as the cross-sectional area reduces from 3.825 to 0.091 $mm^2$.

In addition, the cooling capacity of the cycle is 168.45 kJ/kg, the compressor power is 23.99 kJ/kg, resulting the COP is 7.01. The COP in this study is higher than those obtained from other literature reviews.

## ACKNOWLEDGEMENTS

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