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Research Article

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Experimental Investigation on a Microchannel Evaporator of CO_2 Air Conditioning System with an Internal Heat Exchanger

Tronghieu Nguyen
Department of Thermal Engineering, HCMC University of Technology and Education, Vietnam

Thanhtrung Dang
Department of Thermal Engineering, HCMC University of Technology and Education, Vietnam

Kimhang Vo
Department of Thermal Engineering, HCMC University of Technology and Education, Vietnam

Abstract—

This paper presented experimental investigation on a microchannel evaporator of CO_2 air conditioning system with an internal heat exchanger. The cycle was experimented under the ambient temperature from 30 to 31ºC. The pressure would be adjusted by changing the cross-sectional area of the expansion valve. The pressure of cooler increases from 75 to 78 bar; the pressure of evaporator decreases from 42 to 40bar; temperature evaporator decreases from 7.4 to 5.5°C. The refrigerant temperature drops approximately 2 °C after passing through the internal heat exchanger. At the cooler pressure of 75bar, the cooling capacity is 168.8kJ/kg, the compressor power is 24.04kJ/kg, resulting the COP is 7.02.

Keywords— CO_2 refrigerant, microchannel, evaporator, internal heat exchanger

I. INTRODUCTION

In the recent decades, microchannel heat exchangers have been applied in refrigeration and air conditioning, because they provide larger heat transfer area per unit volume and they are smaller and lighter than those obtained from conventional heat exchangers. Moreover, CO_2 is used as a green refrigerant. Its Global Warming Potential (GWP) is negligible compared to synthetic refrigerants. Compared with the conventional evaporator, microchannel evaporator has remarkable effects on heat transfer and fluid flow behaviours because of dimension, shape, surface roughness, etc. The Poiseuille and Nusselt numbers are higher when the relative surface roughness is larger [1]. Reduced scale leads to enhance fluid compressibility effects [2, 3] and increased roughness leads to increase the drag coefficient [4,5].

In theory analysis, Srinivasan et al. [6] performed energetic analysis of CO_2 vapour compression refrigeration cycle using a new fundamental equation of state. Fartaj et al. [7] studied the main factors affecting the performance of CO_2 automobile air-conditioning system based on the second law of thermodynamic analysis. Kim et al. [8] developed a microchannel evaporator model for a CO_2 mobile air-conditioning system by finite volume method. This model in the study predicted the experimental data with reasonable accuracy, and could be used for the performance analysis and designing of a microchannel evaporator. Sarkar et al. [9] presented the exergetic analysis and optimization of a transcritical carbon dioxide based heat pump cycle for simultaneous heating and cooling applications. Chen and Gu [10] carried out the theoretical analysis and simulation work for the optimum high pressure of CO_2 transcritical refrigeration systems with internal heat exchangers. Chau et al. [11] studied heat transfer phenomena in microchannel evaporator of a CO_2 air conditioning system by numerical simulation. The numerical results are in good agreement with those obtained from experimental data at the same condition.

In component designs, Huai et al. [12] studied flow boiling characteristics of carbon dioxide in multiport minichannels, which had 10 circular channels. The results indicate that pressure drop along the test section is very small; two-phase CO_2 flow exhibits a higher heat transfer coefficient than that of the single-phase liquid or vapor flow. Pettersen et al. [13] developed some compact heat exchangers for CO_2 air conditioning systems. Microchannel heat exchangers give the best overall efficiency. Ma and Wang [14] estimated the design load of microchannel heat exchanger used in CO_2 vehicle air conditioner. They proposed design philosophy about heat exchanger size, material selection, dehumidifier [15]. Jin el al. [16] predicted the performance of an evaporator for a CO_2 mobile air-conditioning system by the finite volume method and then they compared with the experimental data with errors ±12.3%.

In numerical simulation and experimental data, CO_2 refrigerant and compact heat exchanger investigated and experimented. 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 by Dang et al. [17]. The conventional compressor is not suitable for CO_2 refrigerant due to high pressure, so the COP (Coefficient of Performance) of cycle is very low. With CO_2 compressor, the cycle can be achieved COP of 3.07 at the evaporative temperature of 10°C by Nguyen et al. [18].

From literature reviews above, there are no studies on a microchannel evaporator of CO_2 air conditioning system with internal exchanger and subcooling. So, this study will experimentally discuss on a microchannel evaporator of CO_2 air conditioning system with an internal exchanger to finding out the effect of the heat exchanger on the evaporator temperature.

II. METHODOLOGY

A. Experimental Setup

The experimental test loop of the CO_2 cycle is shown in Fig. 1. A microchannel evaporator was tested on this system for finding its heat transfer phenomena. After compression 1-2 to a transcritical pressure, the gas is cooled 2-3'' in a gas cooler. There is no condensation in transcritical area. The gas is cooled further 3''-3' in an internal heat exchanger. 3''-3' segment includes the suction line and the line after cooler and they are insulated with a length of 400mm. The CO_2 gas is cooled further 3'-3 in the subcooler. The expansion 3-4 in the expansion valve drops dramatically pressure and the refrigerant is a mixture of liquid and vapor. Next, the mixture is endothermically evaporated 4-1' in the evaporator, making the cooling capacity. After, the saturated refrigerant has flown through 1'-1 point, it absorbs heat in the internal heat exchanger and becomes the superheat state. At main points of this cycle, four pressure gauges were installed to get the pressure data. The data acquisition device for recording the electronic signals was implemented to obtain data from the thermocouples; the system was integrated through instant monitoring software to record and analyse the data received.

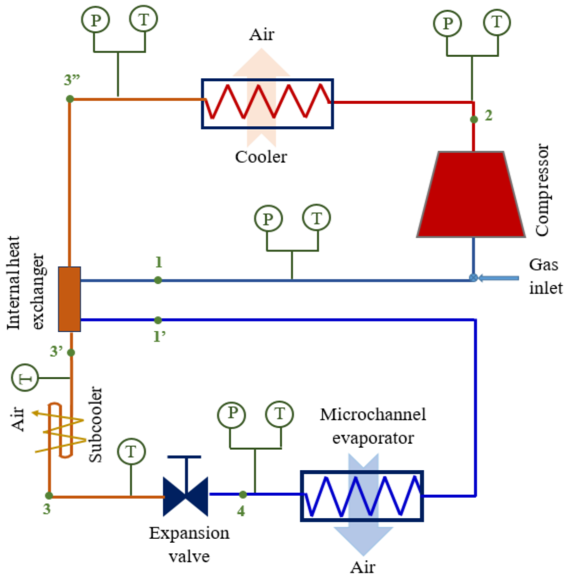

Fig. 1 The experimental test loop for CO_2 air conditioning cycle

In this study, the experiments of the CO_2 air conditioner were done using the Dorin compressor with Model CD 180H. The compressor was connected with the Voltage of 380V, three-phase, and frequency of 50Hz. Copper tubes were used in the cooler; these tubes have outer diameter of 9.4 mm. This cooler was designed and manufactured with heating capacity of 6 kW [18]. The cooler was tested with the hydraulic testing method; it did not tear or deform at the pressure of 150 bar. A microchannel evaporator which is aluminium was used in this test loop. The microchannel evaporator has four passes with 29 microchannels, as shown in Fig. 2. Each microchannel is rectangular in shape, with the width of 1.2 mm and the depth of 0.6 mm. The total heat transfer area of microchannel evaporator is 2.53 m^2. The design cooling capacity for this microchannel evaporator is 2.7 kW [19]. The evaporator was also tested with the hydraulic testing method and this evaporator did not tear or deform at the pressure of 90 bar.

The suction diameter has larger than the discharge diameter. The refrigerant in discharge tube is upstream flow but the refrigerant in suction tube is downstream flow. The discharge and suction tubes are placed in contact with the length of 400mm. The refrigerant from the cooler flows through the IHX in upward and cool refrigerant vapour flows from the evaporator through the IHX in downward, as show in Fig.3.

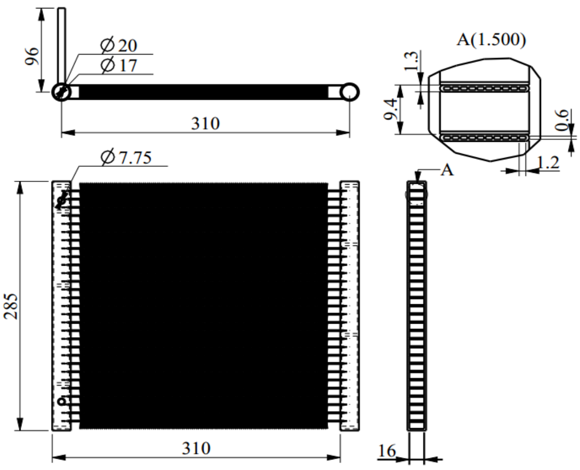

Fig. 2 Dimensions of the microchannel evaporator

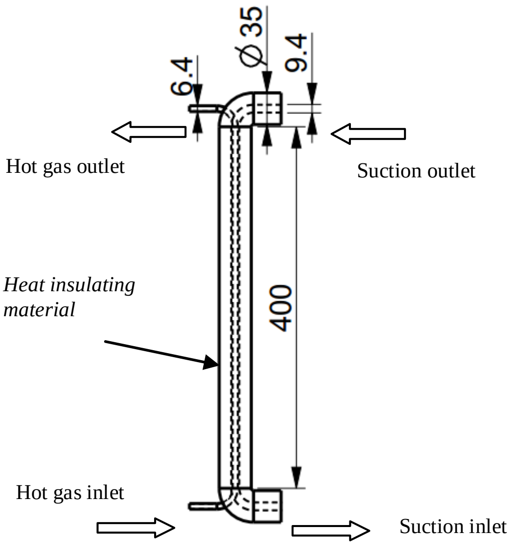

Fig. 3 Dimensions of the internal heat exchanger

B. Governing equations

To analyse the thermodynamic parameters of the CO_2 air conditioning system and the performance index of the microchannel evaporator, the governing equations were given below [19]:

The heat transfer rate for gas cooler was calculated as

$$\dot{Q}_{2-3''} = \dot{m}_c\, c_p \left(T_2 - T_{3''}\right) \tag{1}$$

The heat transfer rate for internal heat exchanger was calculated as:

$$\dot{Q}_{3''-3'} = \dot{m}_c\, c_p \left(T_{3''} - T_{3'}\right) \tag{2}$$

The heat transfer rate for subcooler was calculated as:

$$\dot{Q}_{3'-3} = \dot{m}_c\, c_p \left(T_{3'} - T_3\right) \tag{3}$$

The power input was determined using:

$$\dot{W}_{1-2} = \dot{m}_c \left(h_2 - h_1\right) \tag{4}$$

The isenthalpic process was presented by:

$$h_3 = h_4 \tag{5}$$

The heat transfer rate for evaporation was calculated as:

$$\dot{Q}_{4-1'} = \dot{m}_c \left(h_{1'} - h_4\right) \tag{6}$$

Finally, the COP of the cycle was quantified by:

$$COP = \frac{\dot{Q}_{4-1'}}{\dot{W}_{1-2}} \tag{7}$$

(negligible heat transfer rate of subcooler)

where $\dot{m}_c$ is the mass flow rate of carbon dioxide
c_p is specific heat at constant pressure.

III. RESULT AND DISCUSSION

Table 1 shows experimental parameters of the CO_2 air conditioning cycle with an internal heat exchanger at the cooler pressure of 75 bar. The parameters were plotted on the p-h diagram using EES software, as shown in Fig. 4.

Table 1. Thermodynamic parameters of the CO2 cycle

p1 (bar)	t1 (°C)	t1' (°C)	p2 (bar)	t2 (°C)	p3 (bar)	t3'' (°C)	t3' (°C)	t3 (°C)	p4 (bar)	t4 (°C)	COP	$q_{4-1'}$ (kJ/kg)	w_{1-2} (kJ/kg)
40.5	9.7	8.3	75	57	75	33	31.7	24.4	42	7.4	7.022	168.8	24.04

(where p is pressure and t is temperature)

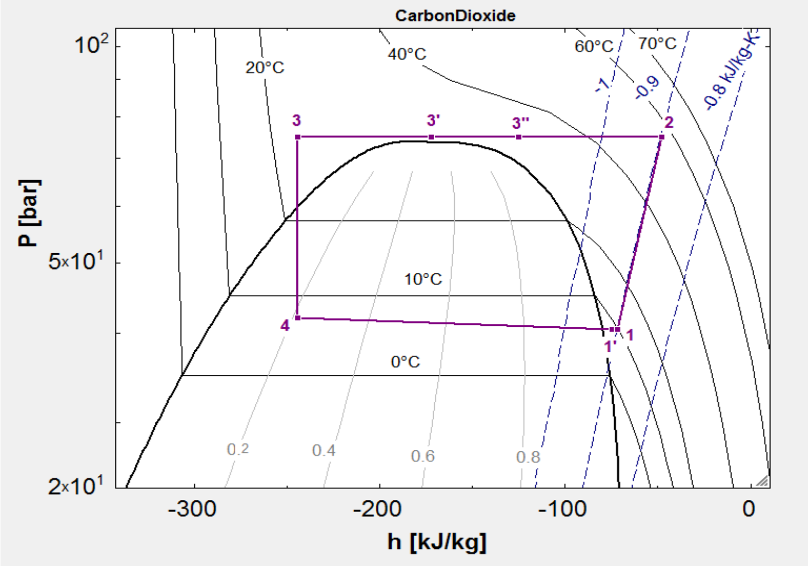

Fig. 4 Experimental points of the cycle on p-h diagram

The experimental relationships between the cooler pressure, the discharge temperature t2 and the suction temperatures t1 are shown in Fig. 5. The results indicated that the discharge temperature decreases as reducing the cooler pressure but the suction temperature increases as reducing the cooler pressure. At the cooler pressure of 75 bar, the suction temperature t1 is 9.7ºC and the discharge temperature t2 is 57 ºC. The Figure 6 shows relationships of the cooler temperature t3'', the internal heat exchanger temperature t3', and the subcooler temperature t3 with the cooler pressure. The experimental results indicated that the cooler temperature is quasi uniform (around 33ºC); the internal heat exchanger temperature and the subcooler temperature slightly increase as raising the cooler pressure. The differential temperature between the inlet and outlet temperatures of the internal heat exchanger temperature is about 2ºC. It is indicated that the efficiency of the internal heat exchanger is not so good because the contact between the two copper tubes in the internal heat exchanger is only two straight lines and the length of internal heat exchanger only has 400mm.

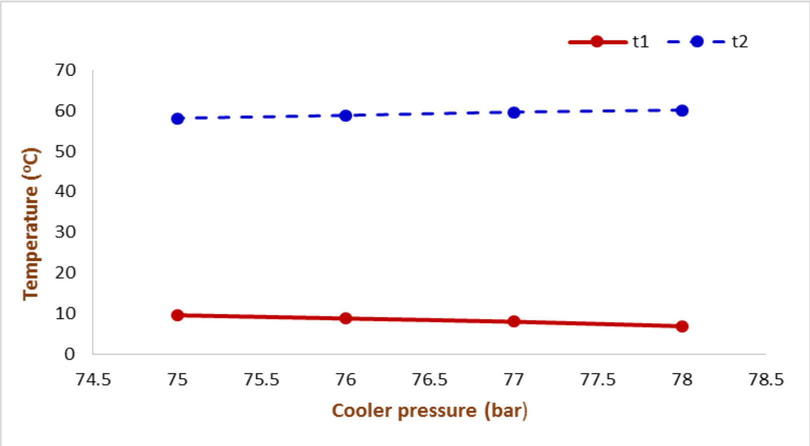

Fig. 5 Cooler pressure versus suction and discharge temperatures

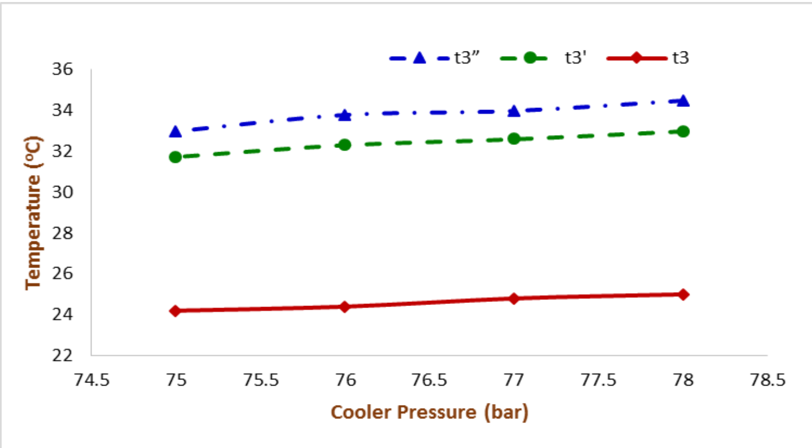

Fig. 6 Cooler pressure versus cooler temperature

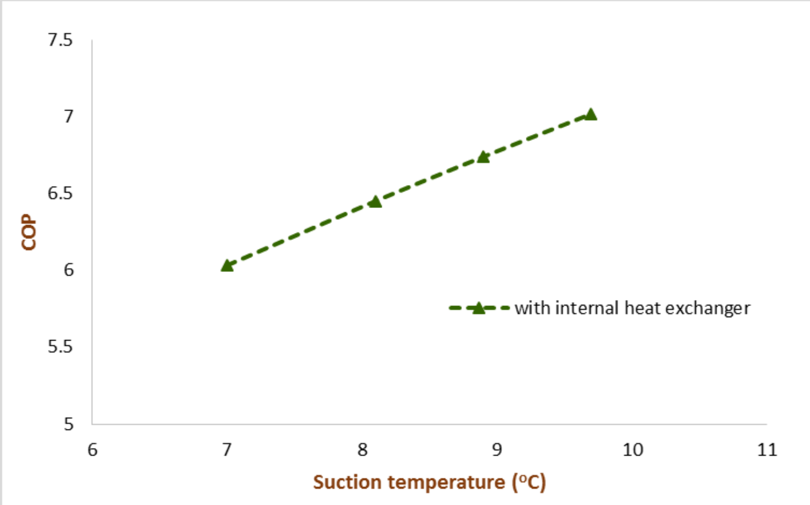

Fig. 7 COP of CO_2 air condition system with the internal heat exchanger

The Figure 7 shows COP of CO_2 air conditioning system with the internal heat exchanger. The experimental results indicated that COP increases from 6.03 to 7.02 as evaporating temperature increases from 5.5 to 7.4 ^{0}C.

A comparison between the present study and other literatures is indicated in Fig. 8. In [20] the COP of 3.2 was achieved at the evaporator pressure of 39.7 bar and the subcooler pressures of 97.5 bar and. The results in present study are higher than another review.

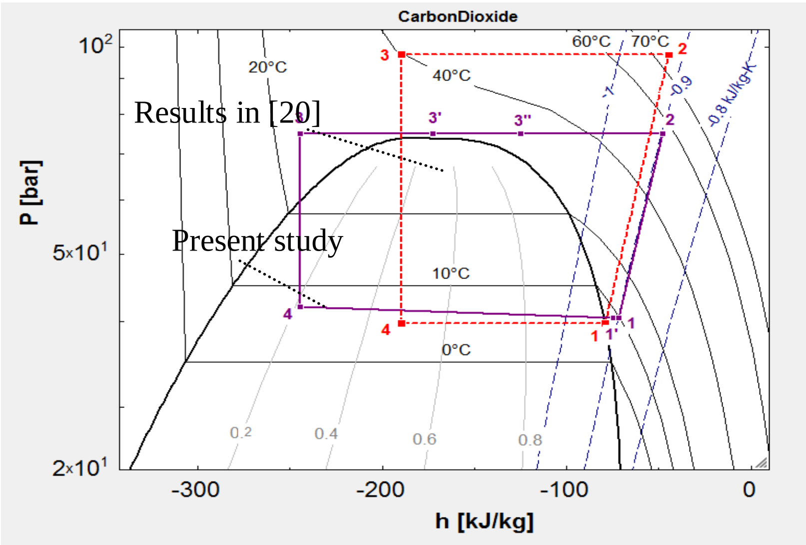

Fig. 8 Comparison between the present study and [20] on the p-h diagram

IV. CONLUSIONS

An experiment on CO_2 air conditioning system with an internal exchanger using microchannel evaporator was done. The cycle with internal heat exchanger was experimented under the ambient temperature from 30 to 31ºC.

In this study, the discharge temperature decreases as reducing the cooler pressure but the suction temperature increases as reducing the cooler pressure. At the evaporator pressure of 42 bar, the suction temperature t1 is 9.7 ºC and the discharge temperature t2 is 57 ºC.

The differential temperature between the inlet and outlet temperatures of the internal heat exchanger is approximately 2ºC. It is indicated that the efficiency of the internal heat exchanger is not so good because the contact between the two copper tubes in the internal heat exchanger is only two straight lines and the length of internal heat exchanger is short.

At the cooler pressure of 75 bar, the cooling capacity is 168.8 kJ/kg, the compressor power is 24.04 kJ/kg, resulting the COP is 7.02. The COP of present study is better than those obtained from other literature reviews.

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