



Simulation Optimization on Heat Transfer Characteristics of Carbon Dioxide in Microchannel Evaporator



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Abstract

Simulating model based on a specific CO₂ microchannel evaporator was established through control volume method with MATLAB, in which both wet and dry conditions for air side, and two-phase and overheat zones for CO₂ side have been considered during the evaporative process. Simulation results showed little discrepancy with previous experimental data which validates the model. And then the heat transfer characteristics in microchannel evaporator were simulated under different inlet air parameters. It was shown that air

velocity has the greatest impact on heat transfer effect, followed by air temperature, and air humidity at last. Meanwhile, the dry-out point also has an important impact on heat transfer performance: before the dry out happens, the heat transfer coefficient of the CO₂ increased with higher air temperature, relative humidity and velocity, while after the dry out occurs, there has been a drastic decline of convective heat transfer coefficient. Therefore, the dry-out point should be postponed for better performance. Then, structural optimization has been made by utilizing two-stage series evaporators. Corresponding simulation results showed that 37.5% area of the original experimental device can still achieve 90.5% heat transfer rate of the former one. So, this method can greatly improve the heat transfer effect of the CO₂ microchannel evaporator.

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Introduction

The main devices of heat transfer in the refrigerating cycle of CO₂ have been going through the development from the finned tube style to the microchannel style. Compared to the traditional heat exchanger, the microchannel heat exchanger is usually smaller and higher heat transfer coefficient, but its pressure resistance and drop is higher, which may easily cause congestion and imbalanced distribution of fluid. CO₂ can cover the shortage of microchannel heat exchanger due to its low ratio between liquid and gas density [1]. However, when the hydraulic diameter is smaller than 3mm, the two-phase flow and heat transfer regulation differs from the normal size. More noticeable microscale effect can be observed in narrow passageway [2].

Many research institutions have studied on this issue that mainly focus on the boiling heat transfer coefficient of the two-phase field, critical heat flux, dry-out point, two-phase flow pattern and pressure drop model [3]. Cheng et al. have discovered that the critical dryness of carbon dioxide was generally between 0.5 and 0.7, which was much lower than that of R22 with a critical dryness usually between 0.8 and 0.9 [4]. Then, they have considered the characteristics of intermittent flow, annular flow, dry-out inception and mist flow to modify the boiling heat transfer correlation under the basis research of Wojtan [5]. Zhang has established a two-dimensional distributed parameter model for the CO₂ microchannel evaporator and proposed a modified heat transfer correlation after comparing to the experimental data [6].

Several appropriate heat transfer correlations are selected according to the heat transfer characteristics of CO₂ in microchannel evaporator, and comprehensively considered the different heat transfer characteristics of wet and dry conditions on air side along with two-phase region and overheated region of CO₂. Parameter distribution simulation model of CO₂ microchannel evaporator has been established and verified through the experimental results. Finally, structural optimization has been proposed and verified through further simula-

tion under different channel number, air temperature, humidity and velocity have been analyzed for studying their impact on heat transfer performance.

I. Experiment Research

A. Microchannel evaporator

Experiment research on a parallel flow micro-channel evaporator which is composed of 36 parallel flat tubes, each of which has 18 microchannels with equivalent diameter of 1.096 mm. The two-phase CO₂ coming from the collecting pipes flows into the microchannel and exchanges heat with the air in the louver fin between the microchannels. **Figure 1** is the structure of the microchannel evaporator and its detailed 3D diagram. The calculated main structural parameters are shown in **Table 1**.

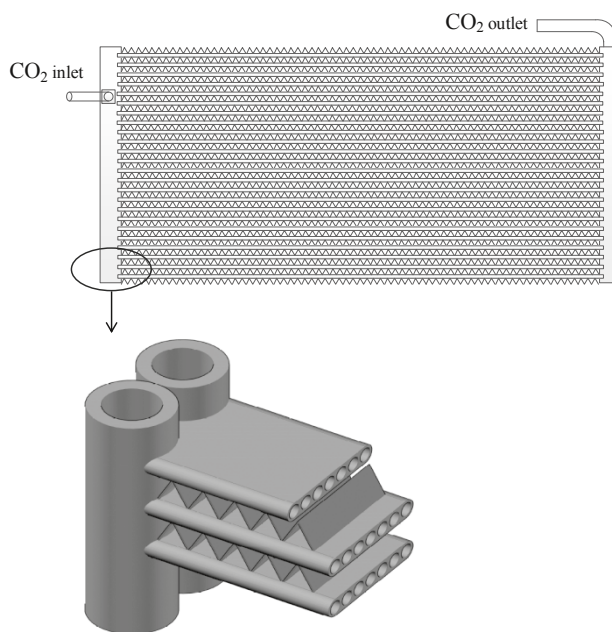


Figure 1. Diagram of the microchannel evaporator.

Table 1. Main structural parameters of microchannel evaporator.

Upwind surface Width/Height (mm)	Air direction depth (mm)	Volume (cm ³)	Heat exchange area (m ²)		Equivalent diameter (mm)
			Air side	Refrigerant side	
810/50	25	7087.5	9.46	2.28	1.096

B. Experimental system

The experimental table of the CO₂ microchannel evaporator was set up (see **Figure 2**). The conditions of the evaporator side are provided by the Enthalpy Different Laboratory. The platinum resistances and pressure transmitters were installed in the evaporator inlet and outlet in order to measure the temperature and pressure of CO₂. Thermocouples were fixed on the surface of the evaporator to measure its tube temperature. The temperature, humidity and speed of air side were measured by thermometer, hydrometer and anemometer. Finally via the electronic expansion valve, the dryness and mass flow rate of CO₂ at the inlet of the evaporator was adjusted.

C. Experimental results

The 18th flat tube was analyzed and divided into 9 sections, which is 90 mm with measuring points set in the center of each section. The incipient air temperature is set to 23°C, and relative humidity is 25%, so the dew point temperature is 2.14°C. The experiment measured CO₂ mass flow rate, inlet dryness, pressure, evaporation temperature, wall temperature, air temperature, humidity and speed which are shown in **Table 2** and **Table 3**. The convective heat transfer coefficient and heat transfer amount of each section of this flat tube would be calculated according to the experimental data, which are shown in **Table 2** and **Table 3**. **Table 4** is the convective heat transfer coefficient and heat transfer amount along the length distribution.

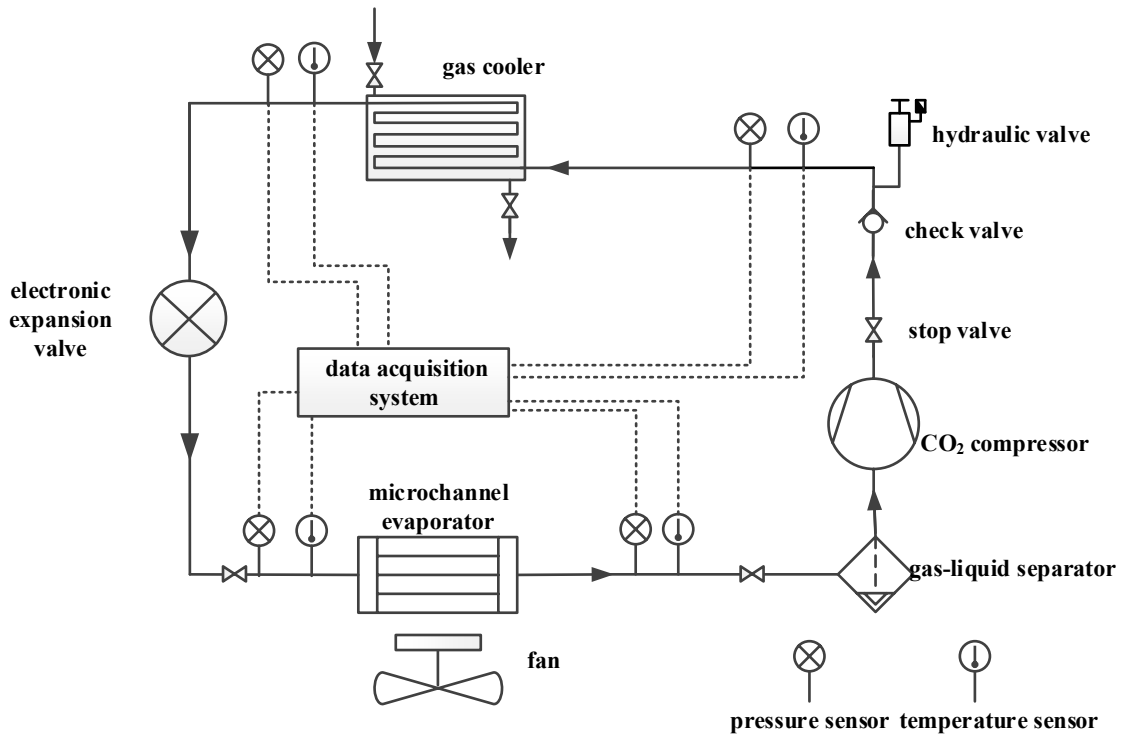


Figure 2. The experiment system diagram.

Table 2. Experimental measurement values of CO₂ side.

Category	Mass flow (g/s)	Inlet dryness	Inlet pressure (MPa)	Outlet pressure (MPa)	Outlet Temperature (°C)
Measured values	15.67	0.28	3.22	3.18	11.58

Table 3. Distribution of Parameters.

Measuring points	1	2	3	4	5	6	7	8
Wind speed (m/s)	1.4	1.9	1.6	1.4	1.8	1.8	1.9	1.8
Outlet temperature (°C)	22.4	22.1	21.7	21.1	19.0	8.2	6.5	5.9
Outlet humidity (%)	26.5	26.8	27.6	28.9	32.7	66.1	60.0	61.1
Wall temperature (°C)	22.8	22.1	21.5	20.5	17.1	6.0	-1.7	-1.9

Table 4. The convective heat transfer coefficient and heat transfer amount along the length distribution.

Length (mm)	45	135	225	315	405	495	585	675	765
Convective heat Transfer coefficient (W/m ² K)	5033.6	4742.3	820.9	156.7	149.2	148.2	147.2	146.3	145.2
Heat exchange amount (W)	30.64	30.90	22.00	5.94	2.14	1.23	1.13	1.02	1.01

II. Simulation Model

A. Heat transfer correlation of CO₂ side

1) Overheated region

According to the different evaporator outlet states of CO₂, the refrigerant flow can be divided into two-phase region and overheated region. As for the overheated region, different heat transfer correlations were selected according to the Reynolds number: When $Re \geq 2300$, the convective heat transfer coefficient was calculated by Gnielinski formula [7]; When $Re < 2300$, the convective heat transfer coefficient was calculated by Sieder-Tate formula [8].

2) Two-phase region

The currently available CO₂ boiling heat transfer correlations are mainly Shah, Gungor and Winterton, Hwang, Yoon, and Cheng correlations. In Cheng correlation, the whole two-phase region is divided into 3 phases as the intermittent flow, annular flow and mist flow according to the boundary point of intermittent flow and annular flow and the dry-out point. Compared with the experimental data in the reference [9], the Cheng correlation was considered to be the most accurate in this experimental condition. So the Cheng correlation was selected in our simulation.

B. Heat transfer correlation of air side

When the wall temperature is below the air dew point temperature, dew will occur on the surface of the flat tube. So the dry and the wet working conditions should be both considered to analyze the heat transfer on the air side. Many research focus on dry condition, while regardless of wet condition. In the wet condition, the surface thermal resistance increases and the heat transfer coefficient will be much smaller. The correlations developed by Kim and Bullard can predict accurately about the heat and mass transfer performance of the shutters both in dry and wet conditions [10]. So their correlation was used in our simulation model.

C. Controlling equations

In order to make the process of calculation easier, the mathematical model of CO₂ microchannel evaporator was assumed as follows:

- 1) CO₂ is equally divided into flow each microchannel;
- 2) No thermal conduction or heat resistance exists between microchannels;

- 3) CO₂ side and air side are both steady flow;
- 4) The air on the condensation water surface is saturated and the thermal resistance of the condensed water is negligible;
- 5) The effect of lubricating oil and noncondensing gas is not considered.

Making the flat tube and the 1/2 louver fin on the upside and underside of it as the research object, the control unit is shown in **Figure 3**.

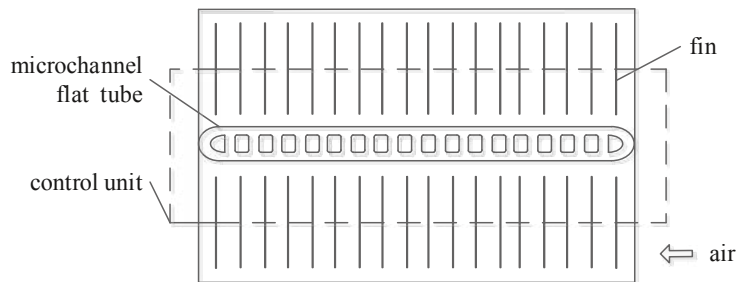


Figure 3. Sectional view of control unit.

As shown in **Figure 4**, every control volume can be regarded as a small cross flow heat exchanger, which was analyzed by energy conservation:

$$\text{Air side heat exchange: } Q_{a,j} = M_{a,d} (h_{ai,j} - h_{ao,j}) \quad (1)$$

$$\text{CO}_2 \text{ side heat exchange (two-phase): } Q_{r,j} = M_r (x_{o,j} - x_{i,j}) i_{tp} \quad (2)$$

$$\text{CO}_2 \text{ side heat exchange (overheated): } Q_{r,j} = M_r (h_{o,j} - h_{i,j}) \quad (3)$$

Where Q represents heat transfer rate (W), M the mass flow rate, (kg/s), h the enthalpy(kJ/kg), x the dryness, and I the latent heat of vaporization(kJ/kg).

Where subscript abbreviation a represent air, r refrigerant, i inlet, o outlet, tp two-phase, j the j control volume.

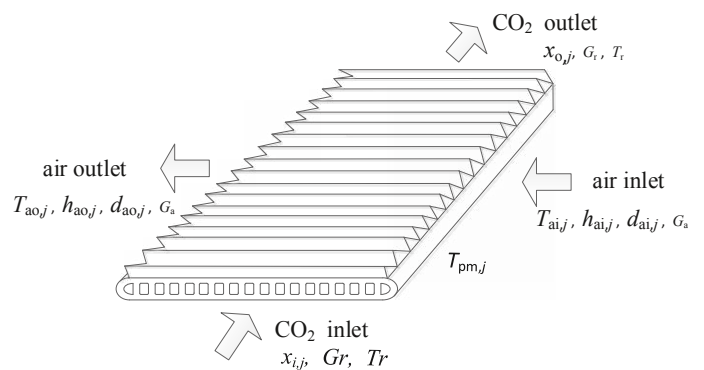


Figure 4. The figure of control volume.

Heat transfer between air and pipe wall under dry and wet conditions:

$$Q_{a,j} = \alpha_{ad} \eta_{ad} A_{a,j} (T_{am,j} - T_{pm,j}) \quad (4)$$

$$Q_{a,j} = \beta_{aw} \eta_{aw} A_{a,j} (h_{am,j} - h_{s,wm,j}) \quad (5)$$

Where α represent sensible heat transfer coefficient ($W/m^2 \cdot K$), β the mass transfer coefficient ($kg/m^2 \cdot s$), η the cooling efficiency, A the heat transfer area (m^2), and T the temperature (K).

Where subscript abbreviation: d represents dry air, w the water film, m average, and s saturated.

D. Simulation process design

The heat transfer existing both in two-phase region and overheated region in normal heat pump system, because a certain overheated degree at evaporator outlet is usually required in order to ensure CO_2 enter into the compressor with gas phase. In this condition, the point with dryness equal 1 of CO_2 was calculated first to divide the heat transfer process into the two-phase region and the overheated region. The specific calculation process is shown in **Figure 5**, in which the state parameters of CO_2 and air were obtained from MATLAB by manipulating the REFPROP.

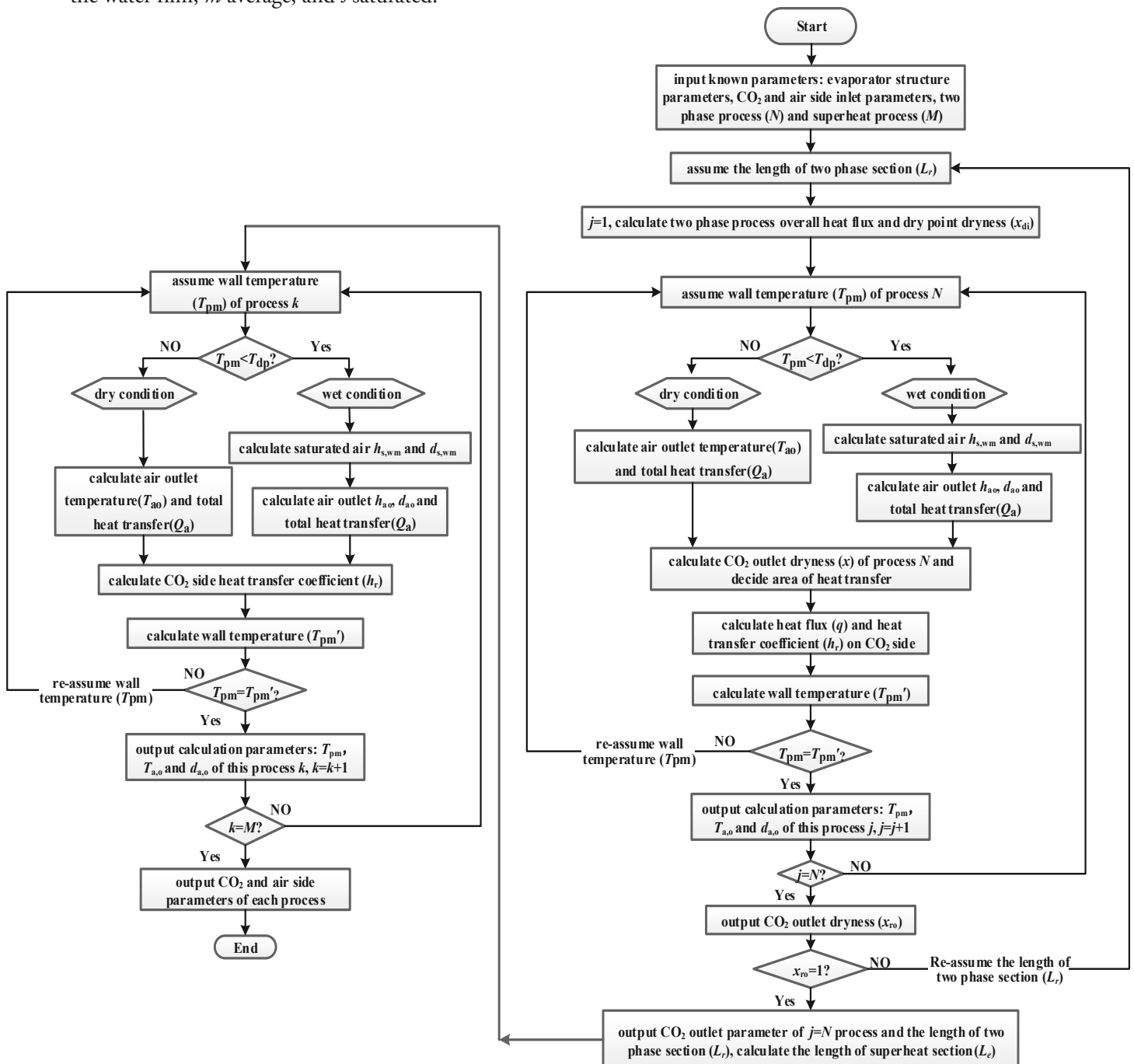


Figure 5. Simulation process of heat transfer.

II. Results and Discussion

A. Comparison of experimental and simulation results

Comparison between simulating and experimental values of CO₂ temperature, tube wall temperature, air

inlet and outlet temperature and the convective heat transfer coefficient are respectively shown in **Figure 6** and **Figure 7**.

The relative errors between experimental and simulated values were calculated. The relative error of CO₂

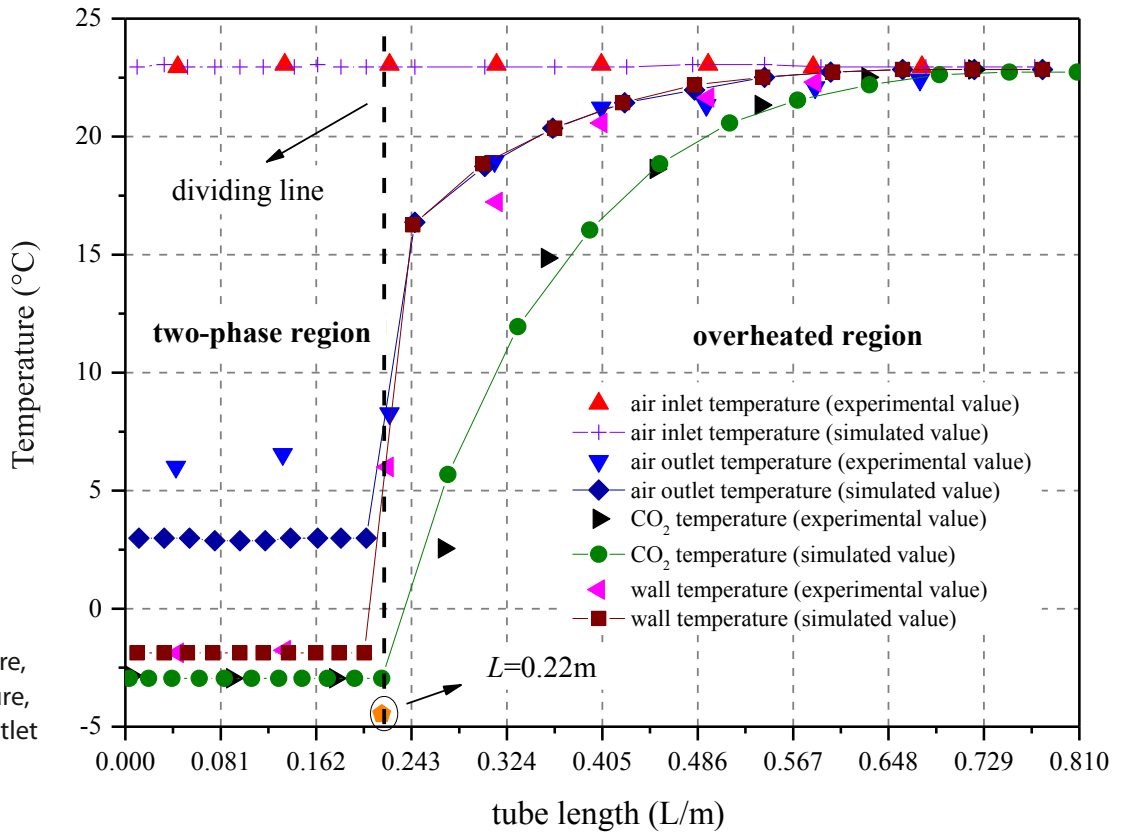


Figure 6. Comparison of CO₂ temperature, wall temperature, air inlet and outlet temperatures.

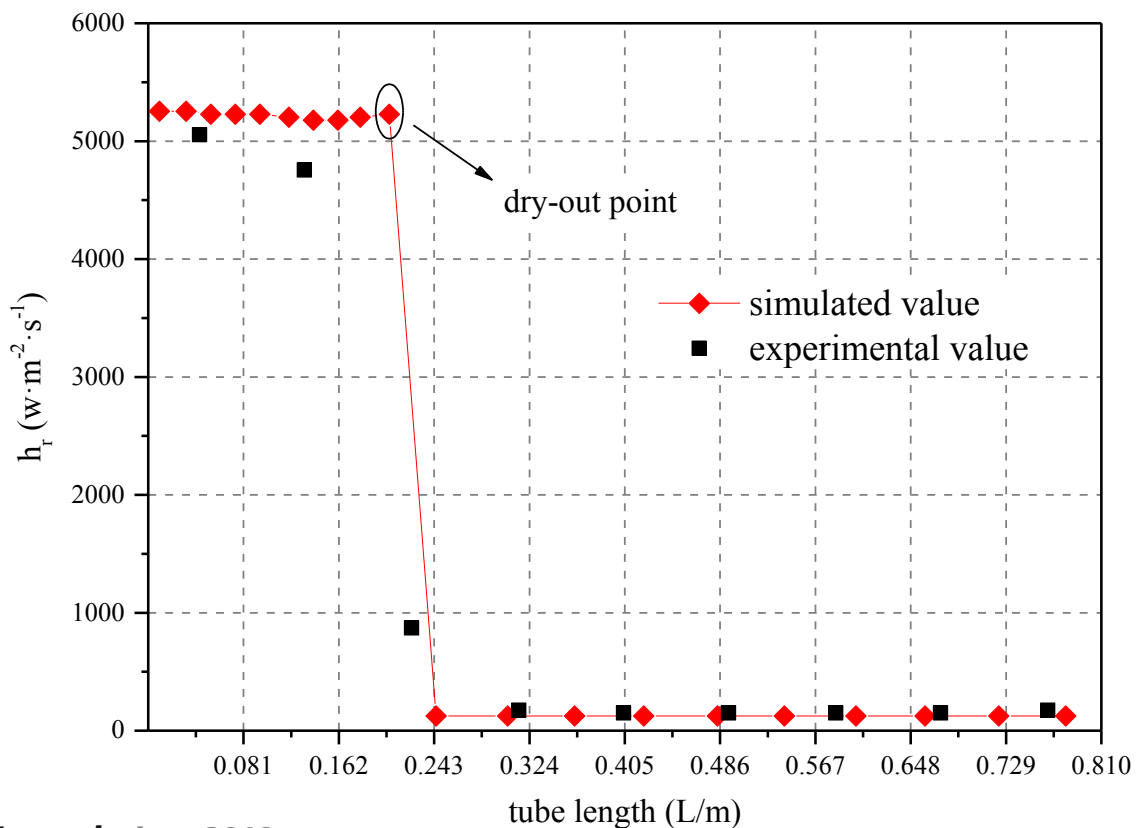


Figure 7. Comparison of the convection heat transfer coefficient.

temperature and wall temperature is under 10%. But the relative error of convective heat transfer coefficient is about 18%. This is because that microchannel evaporator tended to have an uneven flow distribution problem during actual operation which was omitted in the model. Which may cause too much or less CO₂ mass flow rate in some microchannels. Meanwhile, similar changing trend of simulated and experimental values can be seen from **Figure 7** with acceptable errors range, which is usually 20% for a simulation of engineering application. So the simulation model is highly reliable for further analysis and optimization.

B. Sturctural optimization

Considering that the location of the drying point is at the first 1/4 length in the channel as is shown in **Figure 7**, heat transfer efficiency decreases sharply at the latter part of the channel. Therefore, we divide the former one evaporator into two ones with a gas-liquid separator installed between them. The length of the first evaporator is halved to 0.405 m, together with halved flat tubes number of 18. Simulation results show a dryness of 0.68 at the outlet, as is shown in **Figure 8**. Then the two-phase flow of the first evaporator flows into the gas-liquid separator before it enters the compressor. And the remaining 32% liquid refrigerant enters the next evaporator which also has the length of 0.405m. Different number of flat tubes of the second evaporator have been simulated which are 18, 13, 9 and 5 respectively in order to study their impact on heat transfer efficiency under the same inlet air parameters of former experiments, while the CO₂ inlet dryness is assumed to be 0 in the second-stage evaporator.

Figure 9 shows that when flat tubes number is decreased to 9, CO₂ at the outlet of the second stage evaporator is in the overheat zone with overheat temperature of 6°C as is shown in **Figure 10**. When the number decreased to 5, it turned into two-phase zone. Considering only the postpone of dry-out point, in order to ensure the over-heated state at the outlet, the number of flat tubes should be 9 for optimization of the secondary evaporator, which can reduce the heat transfer area to 37.5% of the original one. Former heat transfer capacity of the evaporator was 3.46 kW, and after the optimization, it has become 2.48 kW for the first stage evaporator, and 0.65 kW for the second stage evaporator, which adds to 90.5% of the former one.

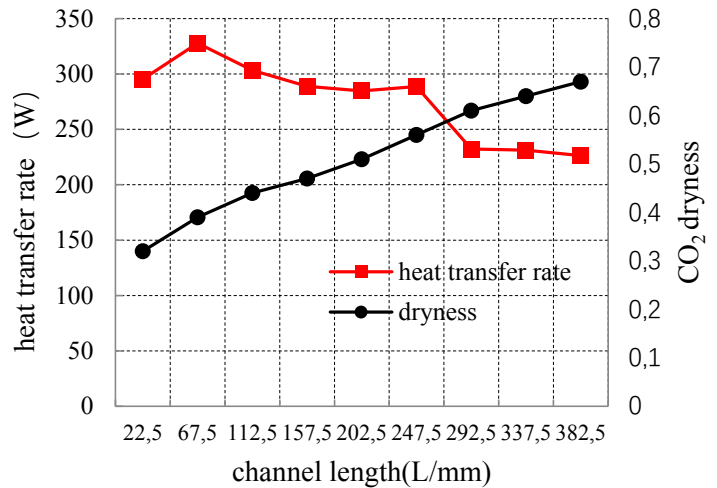


Figure 8. Heat transfer along channel of first-stage evaporator.

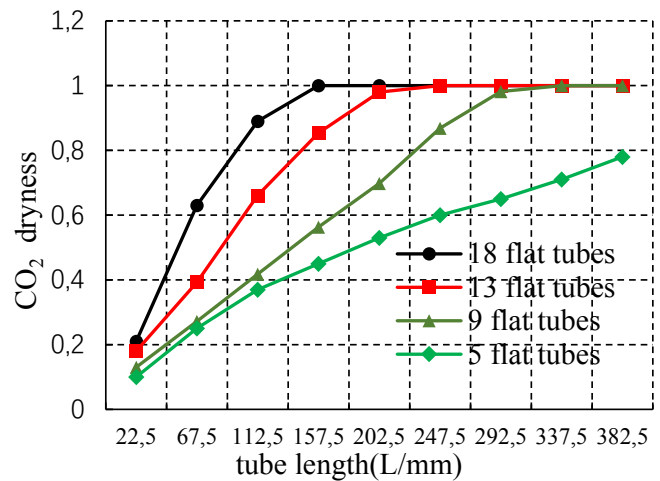


Figure 9. CO₂ dryness of second-stage evaporator under different flat tubes number.

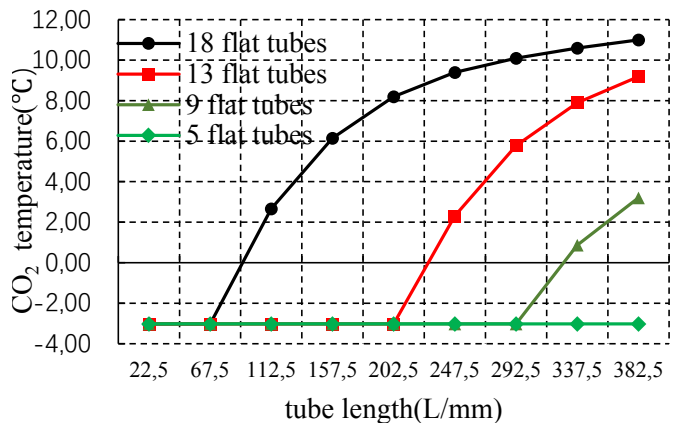


Figure 10. CO₂ temperature of second-stage evaporator under different flat tubes number.

C. Simulation analysis

In order to further reduce the size of evaporator, we plan to figure out the proper operation conditions in which the outlet CO₂ will be in the overheated zone. for evaporator with less flat tubes. Different inlet air temperature, humidity and velocity have been analyzed through simulation for studying their impact on such smaller evaporators as is shown in **Table 4**. Heat transfer rate in each interval was calculated for securing the location of dry-out point. **Figure 11 to 13** showed the impact of inlet air parameters on heat transfer performance:

Table 4. Research conditions.

Category	Air inlet temp (°C)	Relative humidity (%)	Air face velocity (m/s)
Figure 12	23~38	25	2
Figure 13	23	25~70	2
Figure 14	23	25	2~5

We can see the same pattern in heat transfer rate which is similar to former device that the dry-out point marks a threshold of a drastic decline of heat transfer rate after it. While under all inlet conditions, total heat transfer rate remains to be about 600 W. **Figure 12 and 13** show that when air temperature and humidity increase, dry out happens earlier. But the increase of velocity may also improve total heat transfer rate Moreover, air temperature has more significant effect than humidity on the location of dry-out point. Therefore, in order to create a proper condition for smaller evaporator like this in order to have better heat transfer performance, we found that under air temperature of 28°C, humidity of 40% and velocity of 5 m/s, CO₂ can reach the dry-out point in the evaporator with 5 flat tubes with overheated CO₂ at the outlet.

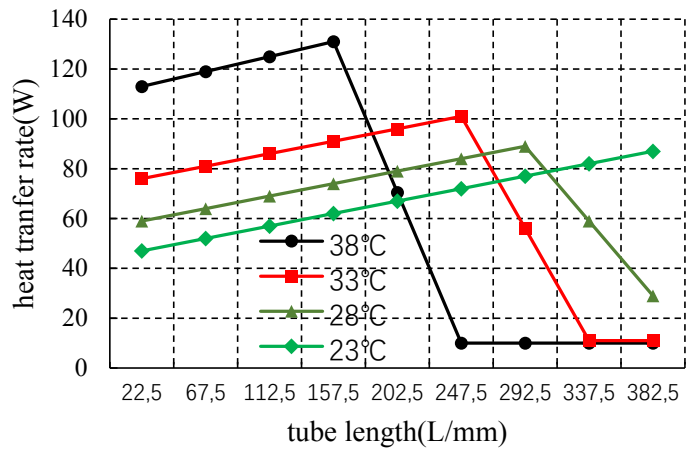


Figure 11. Influence of air temperature on the heat transfer rate.

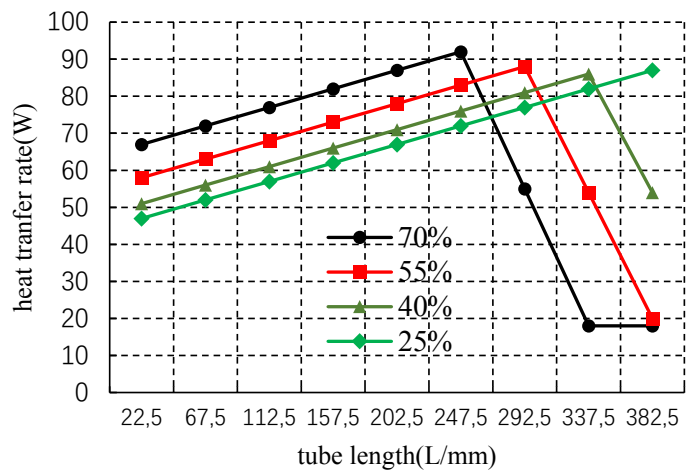


Figure 12. Influence of air relative humidity on the heat transfer rate.

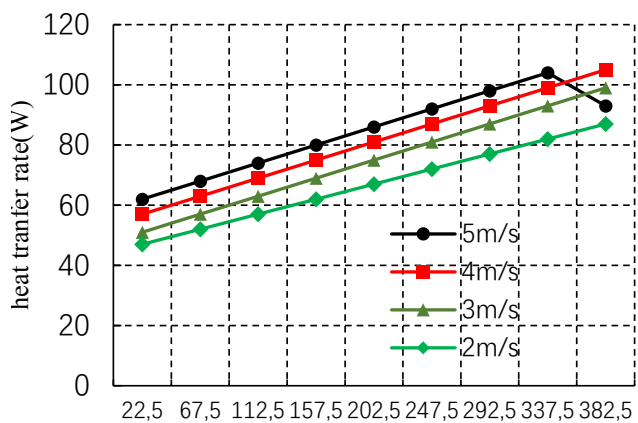


Figure 13. Influence of air speed on the heat transfer rate.

Conclusions

Considering dry and wet conditions on air side and different heat transfer characteristics of CO₂ in two-phase and overheated region, a distributed parameter simulation model of the CO₂ microchannel evaporator was established. The heat transfer in the two-phase region was calculated by Cheng correlation, while in overheated region, it was selected according to the Reynolds number. Comparison between experimental and simulated values in terms of CO₂ temperature, wall temperature, inlet and outlet air temperature and convective heat transfer coefficient showed little discrepancy which verifies the simulation. Both inlet parameters impact on heat transfer performance and structural optimization have been realized through simulation process according to which we can draw the following conclusions:

- 1) The comparison between experimental and simulation results shows little discrepancy within 18% which verifies the simulation method.
- 2) Convective heat transfer coefficient reached the maximum at the dry-out point and then decline drastically which causes heat transfer deterioration. In the overheated region, the heat transfer coefficient is way smaller compared to that of the two-phase

region. Therefore, the later the dry-out happens, the better the cooling efficiency of the device.

- 3) Structural improvement of the evaporator has been made by separating one evaporator into two with gas-liquid separator between them. Results show that 37.5% area of the original experimental device can still achieve 90.5% heat transfer rate of the former one.
- 4) Dry out occurs in a 5-tube evaporator when the air temperature is 28°C, with humidity of 40% or air velocity of 5 m/s. Higher air temperature or relative humidity makes the dry out happen earlier, while none of which has apparent impact on the total heat transfer. Higher air velocity not only makes the dry-out occur earlier, but also improves total heat transfer rate. ■

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