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Keywords: Simulation, experimental study, double-pipe evaporator, heat transfer, Carbon Dioxide

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# Article Experimental and Numerical Study of Double-Pipe Evaporators Designed for CO<sub>2</sub> Transcritical Systems

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**Abstract:** The performance of a CO<sub>2</sub> double-pipe evaporator was studied through experiments and a simulation model that was established by the steady-state distribution parameter method and experimentally verified while using a CO<sub>2</sub> transcritical water-water heat pump system. The effects of different operating parameters on heat transfer performance were studied over a range of evaporation temperatures (-5 to 5 °C), mass velocity (100-600 kg/m<sup>2</sup>s), and heat flux (5000-15,000 W/m<sup>2</sup>). It was found that the dryout quality increased at a small evaporation temperature, a large mass velocity, and a small heat flux. The simulation yield means relative error (RE) of heat transfer for the evaporation temperature and that of the CO<sub>2</sub> pressure drop for the chilled water inlet temperature were 5.21% and 3.78%, respectively. The effect of tube diameter on the performance of CO<sub>2</sub> double-pipe evaporator is probed through simulations. At the same time, this paper defines a parameter  $\alpha$ , which is the proportion of the pre-dryout region to the whole heat transfer region. A larger  $\alpha$  value is desirable. A further theoretical basis is provided for designing an efficient and compact CO<sub>2</sub> evaporator.

Keywords: CO<sub>2</sub>; heat transfer; double-pipe evaporator; experimental study; simulation

# 1. Introduction

Significant attention has been directed towards alternative energy conversion systems due to environmental concerns [1]. Important steps are being taken to drastically reduce the production and consumption of powerful greenhouse gasses known as hydrofluorocarbons (HFCs) and limit global warming. Initiatives like the Kigali Amendment to the Montreal Protocol on Substances that Deplete the Ozone Layer entered into force on 1 January 2019. The current Kigali amendment contract has already made practical arrangements for the implementation of the amendment.  $CO_2$  is an alternative refrigerant to replace chlorofluorocarbons (CFCs) and hydro chlorofluorocarbons (HCFCs) due to its zero ODP (ozone depression potential) and low GWP (global warming potential) [2]. CO<sub>2</sub> has low surface tension, a low gas-liquid density ratio, and low viscosity, which leads to more evaporation cores and it improves its heat transfer coefficient [3]. Under the same experimental conditions, the heat transfer coefficient of boiling  $CO_2$  is twice that of other refrigerants [4–6]. In 1992, the experimental results of the first CO<sub>2</sub> automotive air conditioning system using a transcritical cycle were presented [7]. Currently, many researchers are focusing on the heat transfer performance of pure  $CO_2$ . The use of  $CO_2$  as a refrigerant had some disadvantages, such as the degradation of the COP as compared to other refrigerants [2]. The optimization of system components can effectively improve the efficiency of the component [8]. As a key component, evaporators play an important role in the energy efficiency and physical size of systems.

Much effort has been made to improve the performance of the  $CO_2$  heat exchanger. Jin et al. [9] and Rin et al. [10] used the finite volume method to predict the performance of an evaporator for a  $CO_2$ air-conditioning system. Sarkar [11] and Bai et al. [12] demonstrated a dual-evaporator transcritical  $CO_2$  refrigeration cycle. Kravanja et al. [13] developed a double pipe heat exchanger and investigated the performance of the evaporator. Pettersen et al. [14] compared the performance of microchannel evaporators and plate-fin evaporators designed, with the results illustrating that the evaporator with a 2.0-mm inner diameter (ID) tube had the highest capacity per unit volume.

Moreover, there is a focus on improving the performance of different forms of CO<sub>2</sub> evaporators. Patino et al. [15] developed a finite-volume mathematical model to study a concentric counter-current evaporator for CO<sub>2</sub>. Silvia et al. [16] used a new method for feeding the flooded evaporators in R744 plants. Wiebke et al. [17] developed a one-dimensional discretized steady-state model to study the performance of a CO<sub>2</sub> minichannel evaporator with parallel channels. Ge and Tassou [18] utilized the CFD technique-based evaporator models to optimize the design of  $CO_2$  cabinet evaporators. Ke et al. [19] used the variable property segmented method to investigate the heat transfer and pressure drop characteristics of supercritical CO<sub>2</sub> (S-CO<sub>2</sub>). Son et al. [20] used artificial neural networks to predict the inner pinch for a S-CO<sub>2</sub> heat exchanger. Cui et al. [21] simulated novel airfoil fins for heat exchanger using S-CO<sub>2</sub>. Paradise [22] investigated a hybrid photovoltaic/thermal (PV/T) solar evaporator while using  $CO_2$  through a semi-transient numerical model. The special properties of  $CO_2$ pose serious challenges to the design and optimization of conventional heat exchangers [23]. Therefore, the thermal physical properties in heat transfer and flow of  $CO_2$  have to be taken into account when designing heat exchangers. Particular attention should be paid to the components in transcritical  $CO_2$  refrigeration systems due to the high operating pressure of these systems, as well as the special thermophysical and transport properties of CO<sub>2</sub> in the near-critical region to design efficient and compact heat exchangers.

This paper studies the performance of  $CO_2$  double-pipe evaporators for transcritical  $CO_2$  through experiments and a simulation model that was established by steady-state distribution parameter method. In this paper, Matlab software (MathWorks, Beijing, China) is used to study the performance of the evaporator from structural parameters. The efficacy of the model for the optimization of  $CO_2$  double-pipe evaporator is verified. Moreover, this paper defines one parameter,  $\alpha$ , the proportion of pre-dryout region to the whole heat transfer region.

#### 2. Experimental Installation and Test Procedure

Experiments were conducted based on a water-water heat pump system under different operating conditions to test the performance of the  $CO_2$  double-pipe evaporator. As shown in Figure 1, the test rig consisted of the heat pump and water flow loops. The system comprised of a compressor, an oil separator, a gas cooler, a mass flow meter, a regenerator, a throttling valve, an evaporator, a gas-liquid separator, a water pump, a water flowmeter, an electric heater, hot and cold thermal storage tanks, a platinum resistance temperature sensor, and a pressure transmitter. Table 1 lists detailed specifications of the major components of the heat pump system. The gas cooler and evaporator were double-type heat exchangers with a counter-flow pattern between the refrigerant and water.



1-compressor; 2-oil separator; 3-gas cooler; 4-mass flow meter; 5-regenerator; 6-throttling valve; 7evaporator; 8-gas-liquid separator; 9-water pump; 10-water flow meter; 11-electric heater; 12-water tank; 13-inlet valve; 14-outlet valve; T-thermocouple; P-pressure transmitter

Components	Specifications	
Compressor	Manufacturer	Dorin
-	Model no.	CD380H
	Rated capacity (kW)	3.3
Gas cooler and Evaporator	Manufacturer	Developed in house
_	Туре	Double tube
	Diameter of inner tube (mm)	12
	Length of tube (mm)	16,000
Regenerator	Manufacturer	Developed in house
	Type	Double tube
	Diameter of inner tube (mm)	12
	Diameter of outer tube (mm)	19

Table 1. Specifications of the main components of the system.

The gas-liquid separator between the evaporator and the compressor prevents the liquid shock of the compressor and stores the refrigerant. The oil separator between the gas cooler and the compressor separates the lubricating oil from the refrigerant. Water flow loops for the evaporator and gas cooler include two water pumps and two water tanks. A variable speed pump and a valve control the water flow rate.

Platinum resistance temperature sensors were installed to measure the refrigerant-side and water-side temperatures of the system; among them, four platinum resistance temperature sensors were installed in each water tank, as shown in Figure 1. Seven pressure sensors were installed to measure the pressure in the refrigerant side of the setup. One mass flow meter was installed to measure the refrigerant's mass flow rate. Two turbine flow meters were installed to measure the water flow rates. Table 2 summarizes the specifications and uncertainties of sensors. An electric heating tube is provided in each of the chilled water tanks and the cooling water tank. An air-cooled unit is also

connected outside the cooling water tank. Intelligent control of the temperature of the two tanks is realized. When the water temperature is different from the preset temperature, the unit automatically starts the cooling or heating device.

The first step in testing is to determine full charge under standard conditions: water temperatures of 25 °C and 12 °C entering the gas cooler and CO<sub>2</sub> evaporator, respectively. The water flow rate in the test setup was measured at the selected locations while using a turbine flowmeter and the evaporation temperature in the test setup was measured using platinum resistance temperature sensors. The pressure and heat flux entering the evaporator were kept at 8 MPa and 10,000 W/m<sup>2</sup>, respectively, and the CO<sub>2</sub> mass velocity entering the evaporator varied from 100 to 600 kg/m<sup>2</sup>s. The CO<sub>2</sub> mass velocity and heat flux entering the evaporator were set at 200 kg/m<sup>2</sup>s and 10,000 W/m<sup>2</sup>, respectively, and the evaporation temperature varied from -5 °C to 5 °C. The CO<sub>2</sub> mass velocity and pressure entering the evaporator. Temperature, pressure, and mass velocity were monitored while using a data collection system. The performance of the double-pipe evaporator was tested by the control variable method and Table 3 lists the test conditions.

The mean heat transfer coefficient of refrigerant is calculated, as follows [24]:

$$h_r = \frac{1}{\frac{1}{\frac{1}{K} - \frac{\delta}{\lambda} \ln \frac{d_{w1}}{d_m} - (\frac{1}{h_w} + \gamma_w) \frac{d_{w1}}{d_{w2}} - \gamma_i}}$$
(1)

$$K = \frac{Q_r}{A\Delta t} \tag{2}$$

$$Q_r = c_p \frac{\nu_w \rho_w}{3600} (T_{wi} - T_{wo})$$
(3)

The heat transfer coefficient of chilled water  $(h_w)$  is from Dittus-Boelter [25].

Parameter	Device	Uncertainty	Full Scale
Temperature	Platinum resistance temperature sensor	±0.01 °C	−50−400 °C
Pressure	Pressure sensor	±0.25%	1-40 MPa
Power	Electric power transmitter	±0.2%	0-866 W
Water flow	Turbine flowmeter	±1.3%	$0-1.6 \text{ m}^3 \cdot \text{h}^{-1}$
CO <sub>2</sub> mass flow	Electromagnetic mass flowmeter	±0.2%	0-250 kg·h <sup>-1</sup>

Table 2. Specifications and uncertainties of sensors.

Table 3.	Test	conditions
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Parameters	Unit	Full Scale	Set Value	
Compressor exhaust pressure	MPa		8	
Evaporation temperature	°C	-5 to 5	0	
$CO_2$ mass velocity	kg/m <sup>2</sup> s	100-600	200	
Heat flux	kW/m <sup>2</sup>	5000-15,000	10,000	

## 3. Experimental Results and Discussion

The effects of evaporation temperature, mass velocity, and heat flux on the heat transfer coefficient and cooling capacity of  $CO_2$  double-pipe evaporator were studied. The dryout, which causes a drop in the heat transfer coefficient, forms two regions: the pre-dryout and post-dryout. The mass quality corresponding to dryout is denoted as  $x_{cr}$ .

#### 3.1. Effect of Evaporation Temperature

Figure 2 shows the impact of evaporation temperature on the  $CO_2$  heat coefficient. In the pre-dryout zone,  $h_r$  varied slightly with *T*, since nuclear boiling is the main heat transfer mechanism.

However,  $h_r$  decreased with the increase in *T* in the post-dryout zone. This is due to the fact that the nuclear boiling heat transfer is converted to convective heat transfer.  $x_{cr}$  was 0.685 for an evaporation temperature of -4 °C. However, the  $x_{cr}$  of -2, 0, 2, and 4 °C was decreased by 13.87%, 25.55%, 37.23%, and 47.45% as compared with 0.685, respectively. This is due to the surface tension of the fluid and the liquid-gas density ratio decreasing with increasing *T*, leading to the rupture of the liquid film on the pipe wall.

The cooling capacity, corresponding to 2 °C, decreased by 1.32% as compared to the -5 °C case, as shown in Figure 3. However, the cooling capacity at 5 °C decreased by 59.7% when compared to the 2 °C case. This was attributed to the decrease in the temperature difference between the refrigerant and the water with increasing *T*.



Figure 3. Effect of *T* on cooling capacity.

#### 3.2. Effect of Mass Velocity

Figure 4 shows the impact of mass velocity (*G*) on the heat transfer coefficient ( $h_r$ ), where an increase of *G* resulted in an increase in  $h_r$ . This is due to a high *G* value increasing the concentration of droplets in the vapor flow, which increases the collisions between the droplets and the wall. The  $x_{cr}$  was 0.36 when *G* was 100 kg/m<sup>2</sup>s; the  $x_{cr}$  increased by 40.5%, 70.3%, 95.8%, and 122% for 200, 300, 400, and 500 kg/m<sup>2</sup>s, respectively, which thereby confirms that  $x_{cr}$  is sensitive to *G*. This is explained by the fact that the concentration of liquid droplets in the vapor flow increased under the effect of large *G*. The collision between the droplets and the wall increased, which causes a slow reduction in the thickness of the liquid film until the appearance of dry zones on the wall. In the post-dryout zone,  $h_r$  decreased with the increase in *x*. This can be explained by the fact that convective heat transfer is the dominant mechanism when there is high vapor quality, CO<sub>2</sub> primarily exists as a gas, and the nuclear boiling is suppressed. As can be seen from Figure 5a, the maximum cooling capacity was obtained when *G* was 500 kg/m<sup>2</sup>s. This was due to the mass velocity being too high to allow the fluid to adequately perform the heat transfer. As can be seen from Figure 5b, the cooling capacity increases with increasing tube length at the same mass velocity. This is due to the increased fluid flow time over a larger tube length under the same operating conditions.



Figure 5. Effect of G on cooling capacity.

#### 3.3. Effect of Heat Flux

Figure 6 shows the effect of heat flux (*q*) on heat transfer. In the pre-dryout zone,  $h_r$  remained unchanged by the heat flux due to the heat transfer mechanism of nuclear boiling. However,  $h_r$  and  $x_{cr}$  decreased with increasing heat flux.  $x_{cr}$  was 0.7 when *q* was 5000 W/m<sup>2</sup>, and then the  $x_{cr}$  decreased by 14.3%, 28.6%, 35.7%, and 42.9% for 7500, 10,000, 12,500, and 15,000 W/m<sup>2</sup>, respectively. This was due to the heat transfer mechanism changing from nuclear boiling to convective heat transfer, resulting in a rapid decrease in the heat transfer coefficient.  $x_{cr}$  decreased with increasing heat flux due to the evaporation of liquid film intensifying with increasing heat flux, and the liquid film on the wall of the pipe eventually evaporates and dryout occurs. Nuclear boiling is inhibited and the heat transfer coefficient gradually decreases in the high vapor quality region. At this point, heat flux has a weaker influence on the heat transfer coefficient, which led to the change in fluid from annular flow to mist flow.



**Figure 6.** Effect of heat flux on  $h_r$ .

As a result, the heat transfer deteriorated. As can be seen from Figure 7, the maximum cooling capacity of 9 kW was obtained when q was 5 kW/m<sup>2</sup>. This is due to the heat exchange area before drying being larger when q was smaller.



Figure 7. Effect of heat flux on cooling capacity.

It is necessary to study the influence of tube diameter on the pre-dryout and dryout regions by establishing a model in order to delay dryout and determine the range of the pipe diameter.

#### 4.1. Establishment of a CO<sub>2</sub> Double-Pipe Evaporator Model

The steady-state distributed-parameter method was used to establish the CO<sub>2</sub> double-pipe evaporator. As shown in Figure 8, CO<sub>2</sub> and chilled water flowed outside and inside the tube, respectively. The evaporator was divided into *n* controlled volume elements along the pipe length, and each part included CO<sub>2</sub> fluid, chilled water, and the pipe wall. For the governing equations of the various parts in the micro-element, the flow of fluid in the inner and outer tubes is regarded as one-dimensional flow [26]. The changes in fluid conductivity, fluid density, and specific heat as compared to those of CO<sub>2</sub> are negligible when the oil concentration is small [27]. In addition, as long as a separate liquid CO<sub>2</sub> phase exists, the saturated temperature of the CO<sub>2</sub>-oil mixture is assumed to be equivalent to that of pure CO<sub>2</sub> and the properties of the CO<sub>2</sub>-oil mixture are the same as those of pure CO<sub>2</sub> [27]. The aim of this simulation is to meet the requirement that the length to diameter ratio of the pipeline be greater than 70, that is *L/d* > 70, and thus the inlet effect is not required in determining the heat transfer characteristics of the fully developed fluid [28,29]. The steady-state distributed-parameter model was based on the following assumptions:

- (1) The flow of fluid in the inner and outer tubes was regarded as one-dimensional flow [26].
- (2) The outer wall was considered to be adiabatic without considering leakage heat loss.
- (3) The loss of water-side pressure drop in the evaporator and the momentum equation were not considered.
- (4) When the refrigerants underwent phase transition, the two phases of the fluid were in a state of thermal equilibrium.
- (5) The effect of the lubricating oil and other substances on all heat transfer processes was not considered [27].



Figure 8. CO<sub>2</sub> double-pipe evaporator model.

For each microelement, the heat absorption on the refrigerant side, the heat release on the chilled water side, and the total heat transfer of the evaporator were the same. The heat transfer temperature difference in the microelement was calculated by the logarithmic mean temperature difference method. The heat transfer characteristics are calculated with Equations (4)–(11) according to thermodynamics and Li's research [24].

The refrigerant-side energy is calculated by

$$Q_{r,n} = m_r (h_{ro,n} - h_{ri,n}) \tag{4}$$

The energy equation of the chilled water-side is calculated by

$$Q_{w,n} = c_{pw} \times m_w \times (t_{wi,n} - t_{wo,n})$$
<sup>(5)</sup>

Total heat transfer in evaporator is calculated by

$$Q_n = K_n \times A_n \times \Delta t_n \tag{6}$$

The energy conservation equation of each part is calculated by

$$Q_{r,n} = Q_{w,n} = Q_n \tag{7}$$

The total heat transfer coefficient equation is calculated by

$$K_{n} = \frac{1}{\left(\frac{1}{h_{r,n}} + \gamma_{w}\right) + \frac{\delta}{\lambda} \ln \frac{d_{w1}}{d_{m}} + \left(\frac{1}{h_{w}} + \gamma_{r}\right) \frac{d_{w1}}{d_{w2}}}$$
(8)

Logarithmic mean temperature difference is expressed, as follows:

$$\Delta t_n = \frac{\Delta t_{1,n} - \Delta t_{2,n}}{\ln(\Delta t_{1,n} - \Delta t_{2,n})} \tag{9}$$

$$\Delta t_{1,n} = t_{wi,n} - t_{wo,n} \tag{10}$$

$$\Delta t_{2,n} = t_{wo,n} - t_{ri,n} \tag{11}$$

The water-side heat transfer coefficient is calculated as follows [25]:

$$h_w = \frac{N u_w \lambda_w}{d_{wi}} \tag{12}$$

$$Nu = 0.023 Re^{0.8} Pr^{0.3} \tag{13}$$

The correlation form established by Yoon [30] is used for the calculation of the  $CO_2$ -side heat transfer coefficient since the equivalent diameter of the model established in this paper belongs to the range of conventional pipe diameters. The heat transfer coefficient of pre-dryout is expressed, as follows:

$$h_{r,n} = \sqrt{(Eh_l)^2 + (Sh_{pool})^2}$$
 (14)

$$E = \left[1 + 9.36 \times 10^3 x \Pr\left(\frac{\rho_l}{\rho_g} - 1\right)\right]^{0.11}$$
(15)

$$S = \frac{1}{1 + 1.62 \times 10^{-6} E^{0.69} \operatorname{Re}_{l}^{1.11}}$$
(16)

The heat transfer coefficient of post-dryout is expressed, as follows:

$$h_{r,n} = \frac{\theta_{dry}h_g + (2\pi - \theta_{dry})h_{wet}}{2\pi}$$
(17)

$$h_{wet} = Eh_l \tag{18}$$

$$E = 1 + 3000Bo^{0.86} + 1.12 \left(\frac{x}{1-x}\right)^{0.75} \left(\frac{\rho_l}{\rho_g}\right)^{0.41}$$
(19)

The corresponding pipe lengths in the two regions were recorded as  $L_1$  and  $L_2$ . The mass quality corresponding to dryout ( $x_{cr}$ ) is expressed, as follows [29]:

$$x_{cr} = 38.27 \operatorname{Re}_{l}^{2.12} (1000 Bo)^{1.64} B d^{-4.7}$$
<sup>(20)</sup>

Bo is the boiling number. Bd is the Bond number, which is calculated, as follows [29]:

$$Bd = \frac{g(\rho_l - \rho_g)De}{\sigma}$$
(21)

*De* is the hydraulic diameter.  $L_1$  and  $L_2$  are calculated, as follows:

$$L_1 = \frac{A_{cr}}{\pi De} \tag{22}$$

$$L = \frac{A}{\pi De}$$
(23)

$$L_2 = L - L_1 \tag{24}$$

 $A_{cr}$  is the heat exchange area when the mass quality is  $x_{cr}$ . L is the total tube length, m. The parameter  $\alpha$  is defined, as follows:

$$\alpha = L_1 / L \tag{25}$$

This parameter indicates the proportion of the dryout stage to the entire heat transfer stage, where a larger value of  $\alpha$  is desirable.

This model utilized the Matlab package for programming and calculation. The input was divided into three categories: the structural parameters of the evaporator, such as pipe length and diameter; the refrigerant import and export parameters, such as mass flux, evaporation temperature, and pressure; and, the import and export parameters of chilled water, such as flow rate, pressure, and temperature. First, the outlet temperature of chilled water was assumed to calculate the inlet temperature of chilled water, until the difference between the inlet temperature and the given temperature was between -5% and 5%.

Figure 9 shows a flowchart of the simulation.



Figure 9. Flowchart of simulation.

#### 4.2. Validation of the Model

The simulation values of  $h_r$  and pressure were compared with the experimental results in order to verify the accuracy of the model, as shown in Figure 10. The RE is calculated, as follows:

$$RE = \frac{|simulation value - experimental value|}{experimental value} \times 100\%$$
(26)

Figure 10a shows the effects of the chilled water flow rate on the heat transfer. The heat transfer coefficient was found to proportionally increase with the increase in the chilled water flow rate. As can be seen from Figure 10a, the simulation value is slightly higher than the experimental value. This was attributed to variations in the experimental conditions. The temperature continued to decrease although there was an electric heating wire in the chilled water tank to maintain the constant temperature of the chilled water, which resulted in a decrease in the heat transfer temperature difference, resulting in decreased heat transfer. With the increase in the evaporation temperature, both the simulation value and the experimental value of the average heat transfer coefficient exhibited lower fluctuations,

as shown in Figure 10b. When the evaporation temperature is -1 °C, RE showed a minimum value of 4.41%. When the evaporation temperature was 1 °C, RE showed a maximum value of 5.94% with a 5.21% mean value. Figure 10c shows the effects of the chilled water inlet temperature on the CO<sub>2</sub> pressure drop. The pressure drop was found to increase proportionally with the increase in the chilled water inlet temperature. When the chilled water inlet temperature was 12 °C, RE has a maximum value of 4.57% and a mean of 3.78%.



**Figure 10.** Comparison of experimental and calculated results. (a) Effect of chilled water flow on  $h_r$ ; (b) Effect of evaporation temperature on  $h_r$ ; and, (c) Effect of chilled water inlet temperature on CO<sub>2</sub> pressure drop.

This model is used to simulate and analyze the performance of a double-pipe evaporator designed for CO<sub>2</sub> transcritical systems since the overall trend of experimental and simulated values is consistent.

### 5. Simulation Results and Discussion

#### 5.1. Effect of Outer Tube Diameter

Figure 11 shows the influence of  $CO_2$  side diameter on heat transfer, where  $h_r$  and  $x_{cr}$  were observed to increase with decreasing diameter, although the drop rate of the  $h_r$  gradually decreased.  $h_r$  increased by 38.9% and  $x_{cr}$  increased by 50%. This shows that the smaller the outer tube diameter, the higher the heat transfer coefficient and the greater the  $x_{cr}$ . This is due to the cross-sectional area of the outer tube decreasing with the decrease in diameter, which resulted in more contact between the  $CO_2$  fluid and the tube wall; smaller diameters promote nuclear boiling heat transfer. The heat transfer coefficient of the smaller pipe diameter slowly decreases as *x* increases in the post-dryout zone. This can be explained by the fact that the convective heat transfer is the main mechanism in the post-dryout zone, and the flow velocity of the working fluid increases with the decrease in the pipe diameter, which enhances the convective heat transfer in the small pipe diameter.



**Figure 11.** Effect of  $d_r$  on  $h_r$ .

Table 4 shows a comparison of the  $\alpha$  of each heat transfer process under different  $d_r$ .  $\alpha$  decreased with increasing  $d_r$ , as is evident from Table 4.  $L_1$  accounted for 67% of the L (3 m) when  $d_r$  was 17 mm; the cooling capacity was 9846.2 W, as seen in Figure 12. The cooling capacity decreased with an increase in  $d_r$ , which is also evident from Figure 12. This was due to  $d_r$  being too large to allow the fluid to make full contact with the wall, causing a reduction in the effective heat transfer area. In addition, under the same operating conditions, the flow velocity of the fluid decreases as the diameter of the pipe increases, so the amount of heat exchange decreases.

**Table 4.** Comparison of  $\alpha$  under different  $d_r$ .

$d_{ m r}$ mm	16	17	18	19	20	21	22	23	24
α	0.64	0.67	0.60	0.57	0.52	0.49	0.45	0.30	0.23



**Figure 12.** Effect of  $d_r$  on cooling capacity.

#### 5.2. Effect of Inner Tube Diameter

Figure 13 shows the influence of the inner tube diameter on  $h_r$ , where  $h_r$  and  $x_{cr}$  were observed to increase with increasing diameter, but the drop rate of  $h_r$  gradually decreased. Overall,  $h_r$  increased by 38% and  $x_{cr}$  increased by 50%. This shows that the larger the inner tube diameter, the higher the heat transfer coefficient and the greater the  $x_{cr}$ . This was due to the decrease in the cross-sectional area of the outer tube with an increase in the inner tube diameter, which resulted in more than sufficient contact between the CO<sub>2</sub> fluid and the tube wall. The heat transfer coefficient of the larger inner tube diameter slowly decreases as *x* increases in the post-dryout zone. This can be attributed to the convective heat transfer being the main mechanism in the post-dryout zone, and the equivalent diameter of the outer tube diameter decreasing with a larger inner tube diameter, which enhances the convective heat transfer in the outer tube diameter.



**Figure 13.** Effect of  $d_{w2}$  on  $h_r$ .

Table 5 shows the comparison of  $\alpha$  of each heat transfer process under different  $d_r$ .  $\alpha$  increased with increasing  $d_{w2}$  for the case of  $d_{w2} < 14$  mm, as is evident from Table 5. For the case of  $d_{w2} > 14$  mm,  $\alpha$  was found to be 0.62.  $L_1$  accounted for 63% of the L (7.6 m) when  $d_{w2}$  was 14 mm. From Figure 14, at a certain fluid velocity, the cooling capacity increases as the pipe diameter increases. The maximum

cooling capacity was 9010 W, when  $d_{w2}$  was 17 mm. This can be explained by the fact that the increase in the diameter of the water side causes the distance between the inner and outer tubes to decrease, which results in an increase in the heat exchange area of the CO<sub>2</sub> fluid and the water, in cases where the outer diameter is constant. In addition, as the water-side pipe diameter increases, the heat transfer coefficient gradually increases. Therefore, the cooling capacity increases with the increase in the water-side pipe diameter.

10 11 12 13 14 15 16 17 18  $d_{w2} \text{ mm}$ α 0.23 0.28 0.45 0.56 0.63 0.62 0.62 0.62 0.62 10,000 T = 0 °C 9000 = 2.25 m/s8000 Cooling capacity/W 7000 6000 5000 4000

**Table 5.** Comparison of  $\alpha$  under different  $d_{w2}$ .

**Figure 14.** Effect of  $d_{w2}$  on cooling capacity.

13

 $d_{w2}$  / mm

14

15

16

17

18

12

11

10

#### 6. Conclusions

3000

2000

9

This work established a  $CO_2$  double-pipe evaporator model by applying a steady-state distributed parameter method, and then built a  $CO_2$  transcritical water-water heat pump experimental system. The performance of the double-pipe evaporator for  $CO_2$  transcritical systems was studied through simulations and experiments. The test rig consisted of a heat pump and water flow loops. In the double-pipe evaporator,  $CO_2$  and chilled water flowed outside and inside the tube, respectively. The effects of evaporation temperature, mass velocity, and heat flux on the heat transfer coefficient and cooling capacity of the  $CO_2$  double-pipe evaporator were experimentally studied. The results revealed that large mass velocity, small evaporation temperature, and heat flux could delay the dryout. The maximum cooling capacity was obtained when *G* was 500 kg/m<sup>2</sup>s and the maximum cooling capacity of 9 kW was obtained when *q* was 5 kW/m<sup>2</sup>. In the pre-dryout zone, *hr* varied slightly with *T*, since nuclear boiling is the main heat transfer mechanism. Convective heat transfer is the dominant mechanism at high vapor quality.

The simulation value of *hr* and pressure were compared with the experimental results in order to verify the accuracy of the model. The results revealed that the mean relative error (RE) of heat transfer for the evaporation temperature and that of the CO<sub>2</sub> pressure drop for chilled water inlet temperature were 5.21% and 3.78%, respectively. The simulation results showed that, under given conditions, the small outer tube diameter and large inner tube diameter could cause the heat transfer coefficient and  $x_{cr}$  increase. This indicates that the equivalent diameter of the refrigerant side is as small as possible.  $\alpha$  was 67% and 63% for a 17 mm outer tube diameter and 14 mm inner tube diameter,

respectively. A larger  $\alpha$  is desirable; this indicates that the heat exchange zone mainly occurs in the pre-dryout and the heat transfer mechanism is mainly nuclear boiling. A further theoretical basis is provided for designing an efficient and compact CO<sub>2</sub> double-pipe evaporator.

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#### Nomenclature

Α	area (m <sup>2</sup> )
$C_p$	specific heat (J/kg/K)
d	diameter (mm)
De	equivalent diameter (mm)
Ε	enhancement factor
G	mass velocity (kg/m <sup>2</sup> s)
8	gravity acceleration (m/s <sup>2</sup> )
$h_w$	water heat transfer coefficient (W/m <sup>2</sup> /K)
h	enthalpy (kJ/kg)
h <sub>r</sub>	heat transfer coefficient (W/m <sup>2</sup> /K)
$h_l$	heat transfer coefficient of liquid (W/m <sup>2</sup> /K)
h <sub>wet</sub>	heat transfer coefficient of wet pipe wall (W/m <sup>2</sup> /K)
$h_g$	gas phase heat transfer coefficient (W/m <sup>2</sup> /K)
h <sub>pool</sub>	boiling heat transfer coefficient (W/m <sup>2</sup> /K)
K	total heat transfer coefficient (W/m <sup>2</sup> /K)
L	total tube length (m)
$L_1$	tube length in pre-dryout zone (m)
$L_2$	tube length in post-dryout zone (m)
т	mass flux (kg/s)
q	heat flux (W/m <sup>2</sup> )
Q	heat quantity (W)
RE	relative errors
S	suppression factor
t	temperature (°C)
Т	evaporation temperature (°C)
υ	volume flow (m <sup>3</sup> /h)
Greek symbols	
λ	thermal conductivity (W/m <sup>2</sup> /K)
ρ	density (kg/m <sup>3</sup> )
σ	surface tension (N/m)
Pr	Prandtl number
Re	Reynolds number
Nu	Nusselt number
δ	wall thickness (mm)
γ	dirty factor
x	mass quality
Во	boiling number
Bd	Bond number
α	L <sub>1</sub> /L
$\theta_{dry}$	dry angle
Subscripts	
w	water
$w_1$	refrigerant side wall

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