Numerical simulation of supercritical CO2 characteristics in a double-pipe microchannel gas cooler

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Abstract—In this paper, numerical simulation is considered as a method to explore the heat transfer characteristics of sCO2 under different pressures, CO2 mass flow and CO2 inlet temperature in the double-pipe microchannel heat exchanger. 0.89 mm was adopted as the microchannel diameter. No matter it is under different pressures, different CO2 mass fluxes or different inlet temperatures. The heat transfer coefficient of supercritical CO2 rises sharply to the highest point and then falls slowly. The maximum value of the heat transfer coefficient of CO2 always appears at the position where $T_b < T_{pc}$ with all working conditions. Lower pressure, CO2 mass flux and CO2 inlet temperature lead to higher heat transfer coefficient. No matter what kind of working conditions, CO2 has not been cooled to water temperature.

1. INTRODUCTION
The physical properties near the CO2 critical point are complex. When supercritical CO2 as the refrigerant, microchannel heat exchangers are often used because of their compact structure[1] in a transcritical CO2 refrigeration system[2]–[5].

Cai et al.[6] reported a numerical investigation of the heat exchange process between supercritical CO2 and cooling H2O in a microtube gas cooler at low Reynolds number. He noted that the $h_{CO2}$ is the largest when CO2 temperature appears at $T_b = T_{pc}$. The increase in $h_{CO2}$ is accompanied by an increase in the mass flux of CO2. Zhang et al.[7] numerically studied the local heat transfer characteristics of sCO2 heated in a horizontal semicircular microtube. Pressure, CO2 mass flux, heat flux and tube geometry are included, and the results show that the dramatic changes in the specific heat of the critical zone have a great impact on the heat exchange mechanism. Lei et al.[8] explore the $h_{CO2}$ and $\Delta P$ of supercritical CO2 in horizontal wave microchannels through numerical method. Obviously wave-shaped microchannels show significant advantages in heat exchange performance.
In this paper, numerical simulation methods are used to explore the relationship between $h_{CO_2}$ and $T_{b,CO_2}$ of supercritical CO2 under different pressures, CO2 mass flow and CO2 inlet temperature in the double-pipe microchannel gas cooler aims to provide available basis for the design of gas coolers theoretically. The diameter of the microchannel is 0.89 mm.

2. NUMERICAL SIMULATION

2.1. Numerical model

The geometry and dimensions of the double-pipe micro-channel gas cooler was shown in Fig.1. CO2 flows in the outer 12 microchannels with a diameter of 0.89 mm, water flows in an inner tube with a diameter of 3mm, and the flow heat exchange is counter-current. The heat exchanger length is 30 mm and the shell diameter is 5 mm. Following assumptions were required in the simulation:

- uniform flow on the tube side.
- uniform flow at the shell side inlet.
- heat conduction exists only in the vertical direction of the tube wall.
- ideal counter-flow between CO2 and water.
- the heat exchangers are placed horizontally.

![Fig.1. Numerical model](image)

The global heat transfer processes were simulated using ANSYS fluent 19.0. SST k-ω turbulent model[9]–[12] was selected in this paper. NIST refprop9.11 was used to reference the properties. Mass-flow inlet and pressure-outlet were chosen as the boundary conditions. Table 1 shows the details of the simulated working conditions.

| $G_{CO_2}$ (kg/m²s) | $T_{b, in}$ (K) | P (MPa) |
|---------------------|----------------|---------|
| 100                 | 343.15         | 8       |
| 159                 | 353.15         | 9       |
| 210                 | 363.15         | 10      |

2.2. Governing equations

A steady-state method was used in the simulation. The continuity equation is shown in Eq. 1.

$$\frac{\partial (\rho u_j)}{\partial x_j} = 0$$

Momentum equation can be described by Eq. 2.

$$\frac{\partial}{\partial x_j} (\rho u_i u_j) = \frac{\partial}{\partial x_j} \left[ \mu_{eff} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \mu_{eff} \frac{\partial u_k}{\partial x_k} \right] - \frac{\partial p}{\partial x_j} + \rho g_i$$

where $\mu_{eff}$ is the effective viscosity which is calculated as Eq. 3.

$$\mu_{eff} = \mu + \mu_t$$
where $\mu_t$ is the turbulence viscosity under the turbulence model.

Conservation of momentum is described by Eq. 4.

$$\frac{\partial \left( \rho u_j c_p T \right)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \mu_{\text{eff}} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \mu_{\text{eff}} \frac{\partial u_k}{\partial x_k} \right] + \frac{\partial}{\partial x_i} \left( k_{\text{eff}} \frac{\partial T}{\partial x_i} \right)$$

(4)

where $k_{\text{eff}}$ is the effective conductivity which is calculated as Eq. 5.

$$k_{\text{eff}} = k + k_t$$

(5)

where $k_t$ is the turbulent thermal conductivity under the turbulence model.

$k$ - equation and $\omega$ - equation are described by Eq. 6 and Eq. 7.

$$\frac{\partial \left( \rho u_j k \right)}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \mu + \frac{\mu_t}{\sigma_k} \frac{\partial k}{\partial x_j} \right) + P_k - D_k + S_k$$

(6)

$$\frac{\partial \left( \rho u_j \omega \right)}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \mu + \frac{\mu_t}{\sigma_\omega} \frac{\partial \omega}{\partial x_j} \right) + C_\omega + P_\omega - D_\omega + S_\omega$$

(7)

2.3. Data reduction

After the simulation is completed, the results of the inner and outer fluid temperature and heat flux in the axial line can be output directly. The $T_{b,CO_2}$, the water temperature, and the $h_{CO_2}$ need to be calculated. The CO2 heat transfer coefficient inside was calculated by

$$h_{CO_2} = \frac{q_{w,\text{in}}}{T_{b,CO_2} - T_{w,\text{in}}}$$

(8)

where $q_{w,\text{in}}$ and $T_{w,\text{in}}$ were the average wall heat flux and the internal wall temperature along the axial line, respectively, which were calculated by

$$q_{w,\text{in}} = \frac{\int q dL}{L}$$

(9)

$$T_{w,\text{in}} = \frac{\int T dL}{L}$$

(10)

$L$ was the heat exchange pipe’s length, the axial position in STHE is equal to $z$ and $y$ in CPHE. $L$ in DPHE represents the direction along the helix. The CO2 bulk temperature was calculated by

$$T_{b,CO_2} = \frac{\int \rho u c_p T dA}{\int \rho u c_p dA}$$

(11)

The water heat transfer coefficient was calculated by

$$h_{H_2O} = \frac{q_{w,\text{out}}}{T_{w,\text{out}} - T_{b,H_2O}}$$

(12)

2.4. Reliability verification

2.4.1. Model validation

Since experiments on supercritical CO2 double-pipe heat exchangers have been done previously, this paper directly used the previously obtained experimental data to verify the validity of the simulation. The specific equipment of the experiments and the working conditions can be found in Ref.[13]. Fig.2 showed a comparison of the numerical and experimental results for different pressures. It can be seen that the SST $k - \omega$ turbulence model is very reliable.
Fig. 2. Comparison of numerical and experimental results.

2.4.2. Grid independence
Table 1 shows the ΔP for different grid numbers. It is clearly that the pressure drop in the double-pipe microchannel gas cooler is no longer significant for a grid partition of 713585. Therefore, grids with the numbers 713585 was selected to save computational time.

| Number      | Pressure drop | Error (%) |
|-------------|---------------|-----------|
| 331900      | 973.4         | 1.54      |
| 497850      | 628.8         | 0.39      |
| 663800      | 511.9         | 0.34      |
| 713585      | 399.4         | 0.04      |
| 829750      | 382.5         | 0         |

3. RESULTS AND DISCUSSIONS

3.1. Effects of P.
The operating pressure range of 8-10 MPa in heat exchanger was investigated in this paper. The trend of $h_{CO_2}$ at different P can be seen in Fig. 2. The supercritical CO2 inlet temperature of the double-pipe microchannel heat exchanger is 343.15 K, and the CO2 mass flux is 100 kg/m$^2$s. The cooling water inlet temperature is 299.15 K, and the cooling water mass flux is 250 kg/m2s. Obviously, the $h_{CO_2}$ first rises with rising temperature, and then falls with increasing temperature. This change trend is consistent with the change in the specific heat of supercritical CO2. Lower pressure causes higher heat transfer coefficient peak. The position of this $h_{CO_2}$ peak appears at the point where the CO2 temperature in the tube is slightly higher than the $T_{pc}$. Before the $h_{CO_2}$ reaches its peak, lower pressure has a higher heat transfer coefficient. After the $h_{CO_2}$ reaches its peak value, higher pressure causes a higher $h_{CO_2}$. 
3.2. Effects of CO2 mass flux.
The influence of different $G_r$ on the $h_{CO2}$ cannot be underestimated. Fig. 4 shows the changing trend of CO2 heat transfer coefficient under different CO2 mass fluxes. The CO2 inlet temperature of the gas cooler is 343.15 K, and the pressure is 8 MPa. The cooling water inlet temperature is 299.15 K, and the cooling water mass flux is 250 kg/m$^2$s. It was found that low refrigerant mass flux always causes a greater $h_{CO2}$. Like the trend under pressure, the peak of $h_{CO2}$ also appears at a temperature slightly higher than $T_{pc}$. The difference with pressure is that the change before and after the peak value of the CO2, $h_{CO2}$ is consistent.

3.3. Effects of supercritical CO2 inlet temperature.
The CO2 inlet temperature range of 343.15-363.15 K was investigated in this study. Fig. 5 shows the changing trend of CO2 heat transfer coefficient under different CO2 inlet temperature. The CO2 mass flux is 100 kg/m$^2$s, and the pressure is 8 MPa. The cooling water inlet temperature is 299.15 K, and the cooling water mass flux is 250 kg/m$^2$s. Before the $h_{CO2}$ reaches its peak, the $h_{CO2}$ and temperature are positively correlated, after the CO2 peak, the $h_{CO2}$ and the temperature are negatively correlated. It is clearly hat the low CO2 inlet temperature causes a greater heat transfer coefficient.
4. CONCLUSION

In this paper, numerically simulates were adopted to explore the change process of CO2 heat transfer coefficient in a double-pipe microchannel gas cooler under different pressures, CO2 mass flux and CO2 inlet temperature. This study draws the following conclusions.

- No matter it is under different pressures, different CO2 mass fluxes or different inlet temperatures, the $h_{\text{CO}_2}$ shows a trend of first increasing and then decreasing.
- The peak value of the $h_{\text{CO}_2}$ always appears at the position where the CO2 temperature is slightly greater than the critical point with all working conditions.
- Lower pressure, CO2 mass flux and CO2 inlet temperature lead to higher heat transfer coefficient.
- No matter what kind of working conditions, CO2 has not been cooled to water temperature.

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