Numerical study on Flow Characteristics of Microchannel Shell-and-Tube Heat Exchanger with Supercritical CO2

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Keywords: Microchannel tube, supercutical fluid, Microscale fluids

Abstract: The microchannel shell-and-tube heat exchangers with supercritical CO_2 have the characteristics of small volume and high operating temperature. Based on the heat transfer formula and heat balance equation, the practical conditions, such as the variable properties of supercritical fluid, heat dissipation in high temperature conditions and uneven mass flow distribution in microchannels, are considered to establish a thermal calculation model suitable for supercritical fluid and microscale heat transfer, and uses MATLAB programming for numerical calculation. The research found that the heat dissipation cannot be ignored under the high temperature conditions even microchannel heat exchanger volume is small. The heat dissipation rate decreases with the increasing heat transfer power per unit volume. Affected by heat dissipation, the flow distribution between the outermost and middle layer has a large gap. The mass flow rate in columns of tubes decreases along the direction of high-temperature airflow affected by the order of contact with the high temperature flow. The results show that increasing the unit heat power can reduce the heat dissipation, and reducing the diameter of microchannel can reduce the uneven flow distribution.

1. Introduction

The supercritical CO2 Brayton cycle has advantages of small device size and high system efficiency, which is widely used in nuclear power, gas power, waste heat utilization and other fields. To satisfy the requirements on system performance and bulk simultaneously, a compact, efficient and reliable heat exchanger is required for fast heat transfer [1-3]. The micro-scale heat transfer with high heat transfer coefficient has good pressure resistance that can content the high operating pressure of supercritical fluid [4-6].

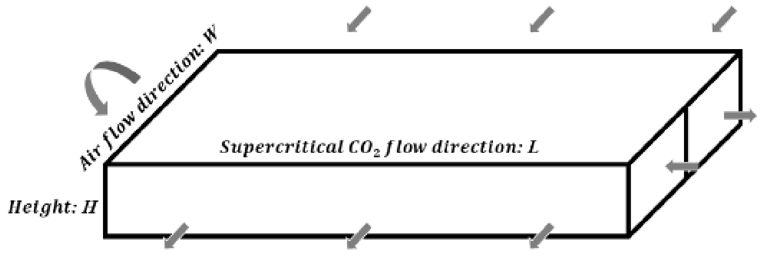
The micro-tube heat exchanger with supercritical CO2 needs to consider many practical conditions. For example, the physical properties of supercritical fluid that vary drastically with temperature and pressure, the uneven distribution of the flow rate in the microchannel, and the heat dissipation to the environment under high temperature working conditions, etc. All these factors have varying degree impact on the heat calculation of the heat exchanger [7,8]. Therefore, it is necessary to improve the design method of the compact heat exchanger with supercritical CO2, which can provide a more accurate design scheme for the practical application and optimization design of the subsequent heat exchanger.

This paper studied the microchannel shell-and-tube exchanger by numerical method. By discretizing the heat transfer into a finite number of heat transfer nodes, the equivalent thermal resistance model is established by considering the variables physical properties, the unevenness of the flow distribution and the heat dissipation to the environment in the thermal calculation to effectively predict the heat exchanger performance. According to the calculation results, the performance of supercritical CO2 microchannel shell-and-tube exchanger was studied, and the influence of pressure and diameter on flow distribution unevenness and heat dissipation to environment was analyzed that can provide better theoretical support for the deterministic design method of compact heat exchangers.

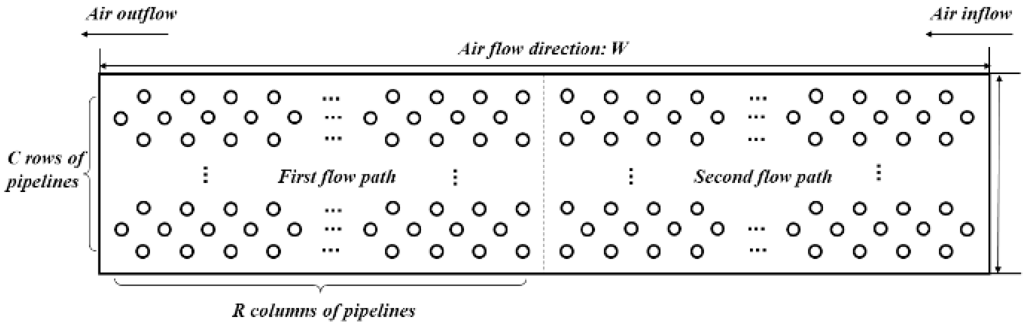
2. Mathematical and numerical method

2.1 Mathematical model

The heat exchanger structure is shown in Fig.1. The shell side is supercritical CO_2 (dual flow), and the shell side is air (single flow). Divided the calculation area of the heat exchanger into the same uniform non-repetitive control volume. The hot and cold fluids in the control volume satisfy the mass and energy conservation. It is assumed that the heat rising the temperature of supercritical CO_2 is all derived from the heat source air. A part of the heat is used for heat exchange, while another part of the heat is dissipated in the height direction causing by the heat dissipation to environment. Temperature distribution network is shown in Fig.2. The flow direction of air and supercritical carbon dioxide is perpendicular to each other. During the heat exchange flow path, only the air in the flow direction, the supercritical CO_2 in the axial direction of microtube, and the temperature change on the inner and outer walls of microtube are concerned [9-13].



(a) The structure of microchannel shell-and-tube heat exchanger



(b) The structure of pipelines

Figure 1. Schematic diagram of heat exchanger structure and tube arrangement

For the middle rows of tubes, the heat of convective heat transfer and dissipation are derived from high temperature air:

$$m_a C_{p,a(r,k,c)}\left(T_{a,i(r,k,c)} - T_{a,o(r,k,c)}\right) + \frac{k_a}{\Delta z}\Delta x \Delta y\left(T_{a,m(r,k,c-1)} - T_{a,m(r,k,c)}\right) = m_c C_{p,c(r,k,c)}\left(T_{c,o(r,k,c)} - T_{c,i(r,k,c)}\right) + \frac{k_a}{\Delta z}\Delta x \Delta y\left(T_{a,m(r,k,c)} - T_{a,m(r,k,c+1)}\right) \tag{1}$$

For the upper row of tubes, $\frac{k_a}{\Delta z}\Delta x \Delta y\left(T_{a,m(r,k,c-1)} - T_{a,m(r,k,c)}\right)$ is replace by $h_{a(r,k,c)}\Delta x \Delta y\left(T_{si(r,k)} - T_{a,m(r,k,c)}\right)$.

For the bottom row of tubes, $\frac{k_a}{\Delta z}\Delta x \Delta y\left(T_{a,m(r,k,c)} - T_{a,m(r,k,c+1)}\right)$ is replace by $h_{a(r,k,c)}\Delta x \Delta y\left(T_{a,m(r,k,c)} - T_{so(r,k)}\right)$

The heat of convective heat transfer is all used to transfer to the cold side:

$$m_c C_{p,c(r,k,c)}\left(T_{c,o(r,k,c)} - T_{c,i(r,k,c)}\right) = UA\Delta T_{m(r,k,c)} \tag{2}$$

Meanwhile, the inside of shell is heated by high temperature air, and the external side is radiating by the natural convection heat transfer and heat radiation:

$$q = h_{a(r,k,c)}\left(T_{a,m(r,k,c)} - T_{si(r,k)}\right) \tag{3}$$

$$q = \frac{k_s}{\delta_s}\left(T_{si(r,k)} - T_{so(r,k)}\right) \tag{4}$$

$$q = h_e\left(T_{so(r,k)} - T_e\right) + \varepsilon\sigma {T_{so(r,k)}}^4 \tag{5}$$

Finishing out the quadratic equation of the outside temperature of shell $T_{so(r,k)}$ and solving $T_{so(r,k)}$ by interpolation. Then the inside temperature of shell $T_{si(r,k)}$ is solved.

2.2 Flow distribution model

When the heat exchanger is in multiple flow path, the dynamic viscosity of supercritical CO_2 changes greatly with temperature. If calculated according to the same mass flow rate, the pressure drop along each channel in the same flow path is nonequal. Currently, the maximum pressure difference in the same flow path becomes the driving force of fluid distribution in parallel tubes. When the tube length, diameter, friction and density is constant, mass flux G_c, will increase with pressure, until the pressure drop between the microtubes is equal, that is, the flow rate distribution must content the principle of equal flow resistance [14].

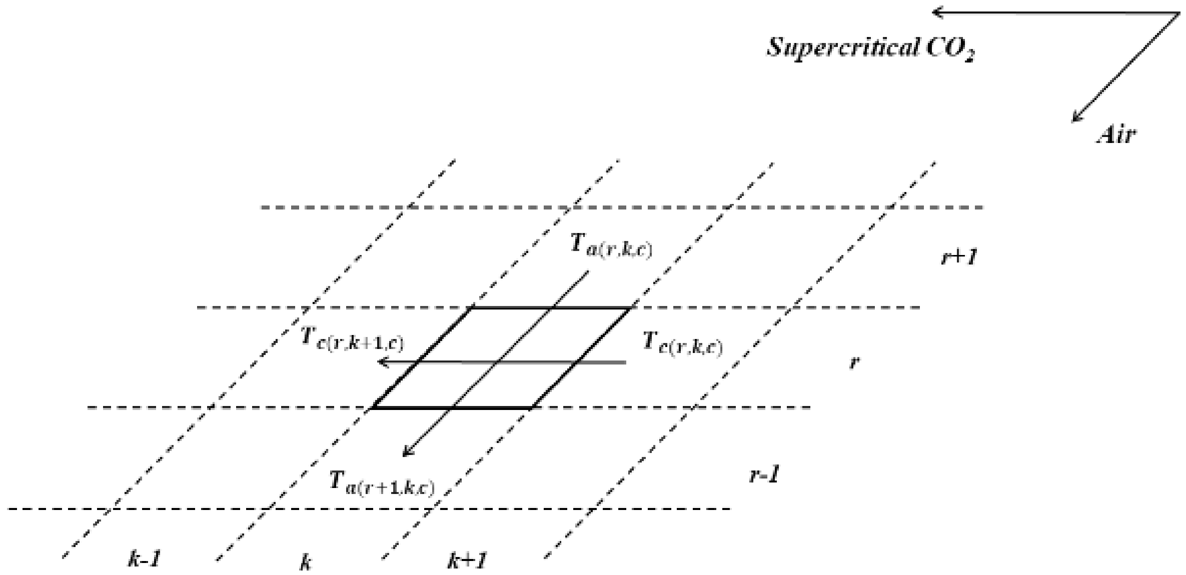


Figure 2. Temperature distribution network

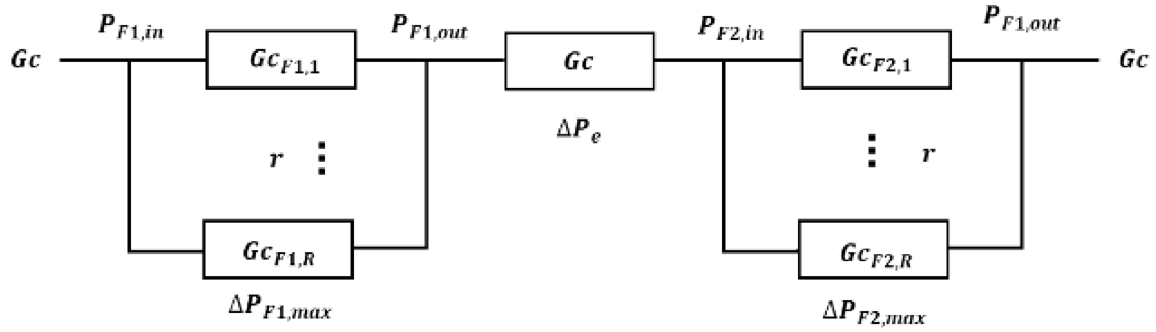


Figure 3. Flow resistance diagram

Simplify the pressure drop into a flow resistance diagram (Fig. 3). $P_{F1,in}$ is the pressure at the entrance of first flow path, the total mass flux of the fluid is G_c. Under the effect of the maximum pressure drop $\Delta P_{F1,max}$, the fluid flows into the microtubes of the first flow path. The friction and the average density are different due to the variable temperature distribution. Therefore, under the same pressure drop, the mass flow rate in each flow channel is expressed as $Gc_{F1,1}$、$Gc_{F1,2}$、··· 、$Gc_{F1,R}$. After that, the fluid flowing out of the first flow path tube turns in the head, and the pressure drop of ΔP_e is generated, and then flows into the second flow path, the flow distribution is completed under the effect of the maximum pressure drop $\Delta P_{F2,max}$. In the process of flow distribution, the mass flux needs to satisfy the following equal relationship:

$$Gc_{F1,1} + Gc_{F1,2} + \cdots + Gc_{F1,R} = Gc \tag{6}$$

Therefore, the flow rate of branches and the total flow rate must be distributed according to the following ratios:

$$\beta_r = Gc_{F1,\mathrm{r}} / Gc \tag{7}$$

The unevenness of flow distribution in each tube:

$$\sigma_r = \frac{qmc_r - qmc_{avr}}{qmc_{avr}} \tag{8}$$

The total flow distribution unevenness of microchannel heat exchanger:

$$\mathrm{S} = \sqrt{\frac{1}{R-1}\sum_{r=1}^{R}(qmc_r - qmc_{avr})^2} \tag{9}$$

3. Calculation method

Thermal calculations require iterative calculations using MATLAB software programming. The definite process is shown in the Fig.4.

The diameter of microtube is 1.2 mm. The mass flow rate of air is 0.15 kg/s and the mass flow rate of supercritical CO_2 is 0.22 kg/s. The working conditions are shown in the Table 1.

Table 1. The working conditions of microtube shell-and-tube exchanger

Symbol	Unit	Value
$T_{a,i}$	K	923
P_a	MPa	0.6
$T_{c,i}$	K	325
P_a	MPa	8.8-10.8

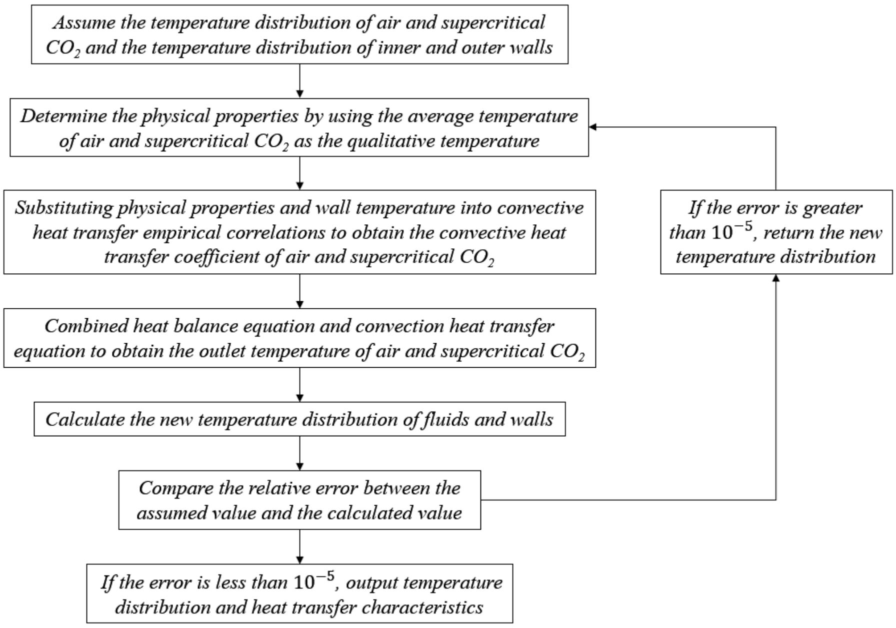


Figure 4. Thermal calculation schematic diagram

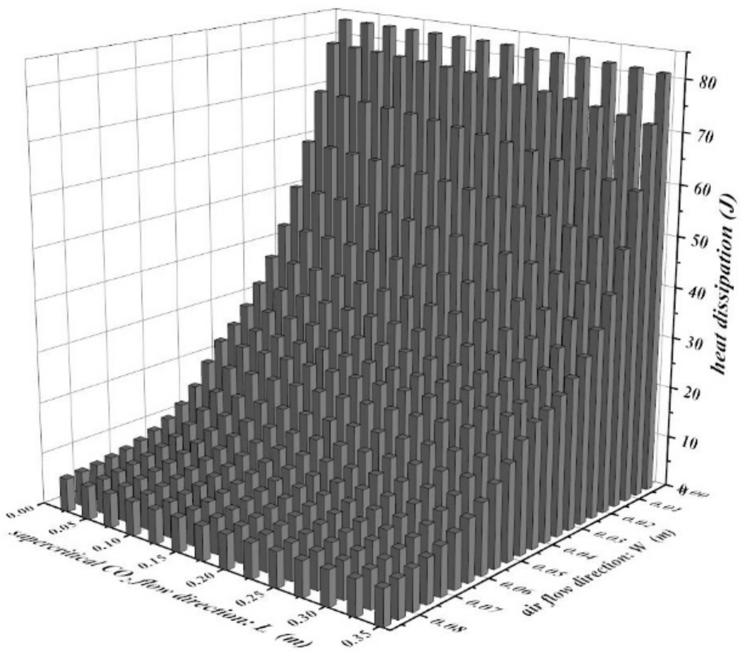


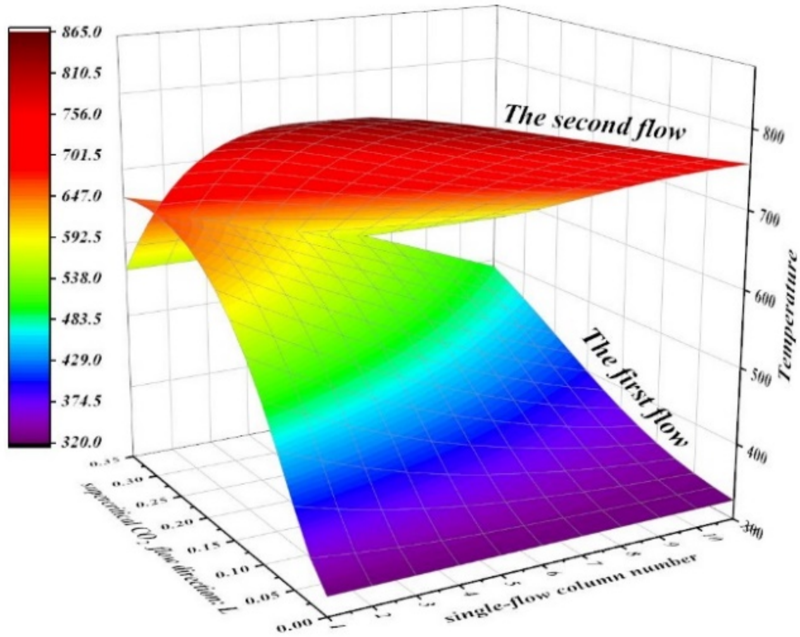
Figure 5. The heat dissipation to environmentResult and Analysis Conclusion

Taking the microchannel shell-and-tube heat exchanger with $P_c = 10.8MPa$and $d_o = 1.2mm$ ($d_i = 0.9mm$) as an example, the heat transfer characteristics were calculated. The analysis focused on the differences in mass flow distribution and heat dissipation through shell after considering actual operating conditions.

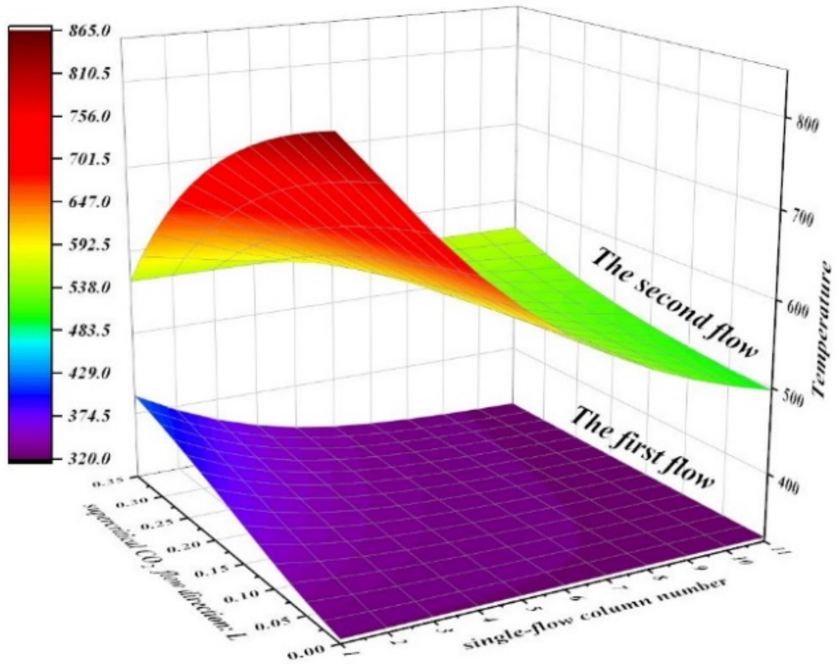
3.1 Temperature distribution and heat dissipation

Fig.5. shows the heat dissipation through the shell to the environment. It can be seen from the figure that the heat dissipation decreases with the direction of air flow, and the drop rate reaches 92.2%. It can clearly be seen that the higher the temperature, the more heat loss, which is 8.876 kJ under this working condition, accounting for 8.63% of the total power. Under high temperature working conditions, even the heat dissipation of microchannel heat exchanger cannot be ignored.

Comparing the temperature distribution of supercritical CO_2 and air at different locations shown in Fig. 6-7. Whether it is supercritical CO_2 or air side, the temperature of the outermost rows, which is heat dissipation to environment through shell, is lower than the temperature distribution at the intermediate rows closest to the ideal state in the same flow path. Supercritical CO_2 flowing out of the first flow path is sufficiently mixed in the head and then flows into the second process, so the inlet temperature of each process is the same. in the same process, the heat transfer is enough when the heat transfer temperature difference is relatively large. The heat transfer capacity of the outmost fluid decreases with heat exchange and heat dissipation.

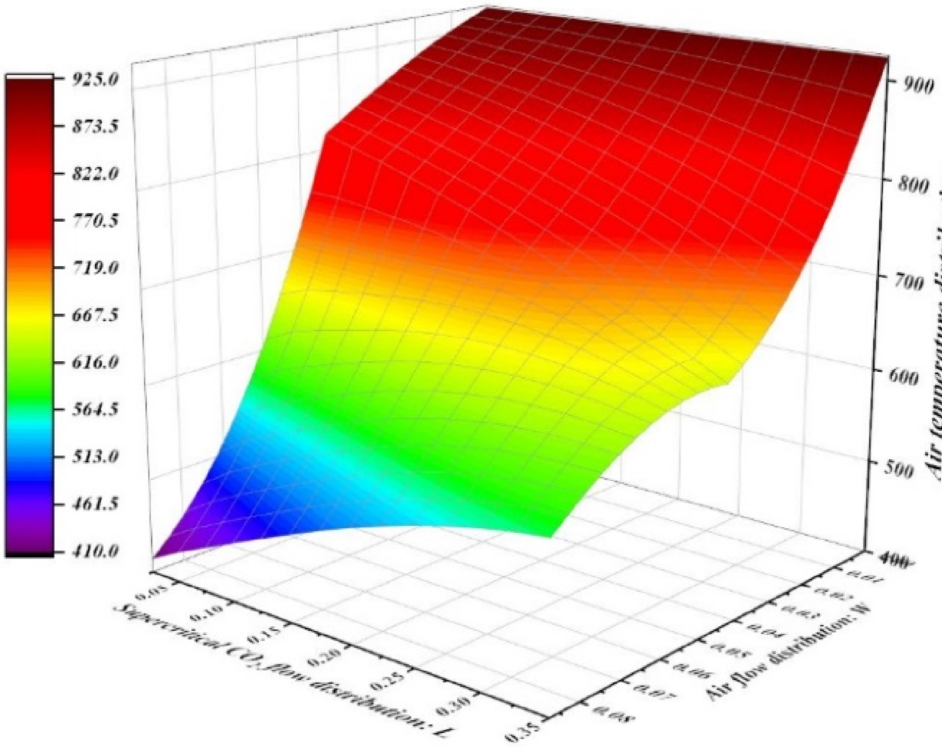


(a) The intermediate layer temperature distribution of supercritical CO2

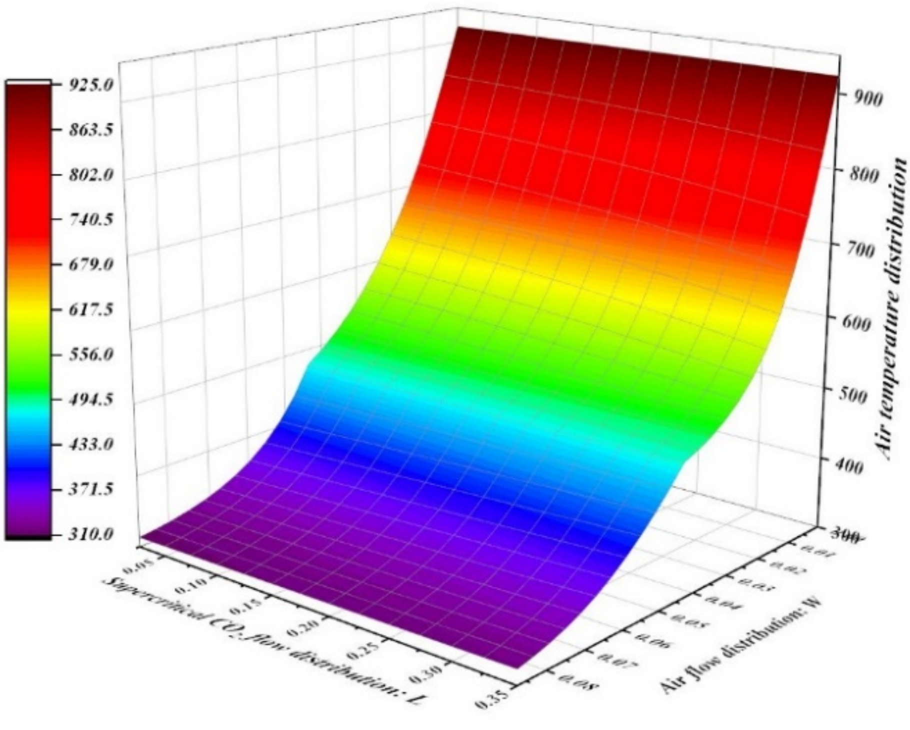


(b) The outermost layer temperature distribution of supercritical CO2

Figure 6. Comparing the temperature distribution of supercritical CO2 at different locations



(a) The intermediate layer temperature distribution of air



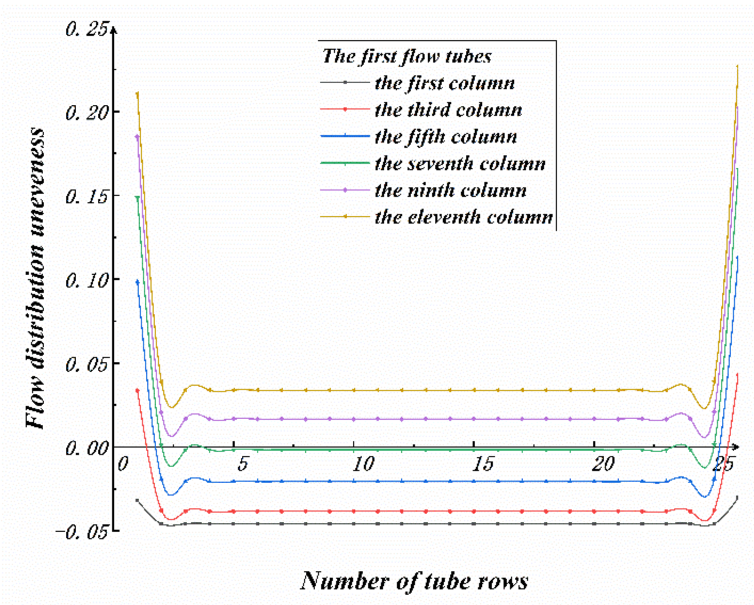
(b) The outermost layer temperature distribution of air

Figure 7. Comparing the temperature distribution of air at different locations

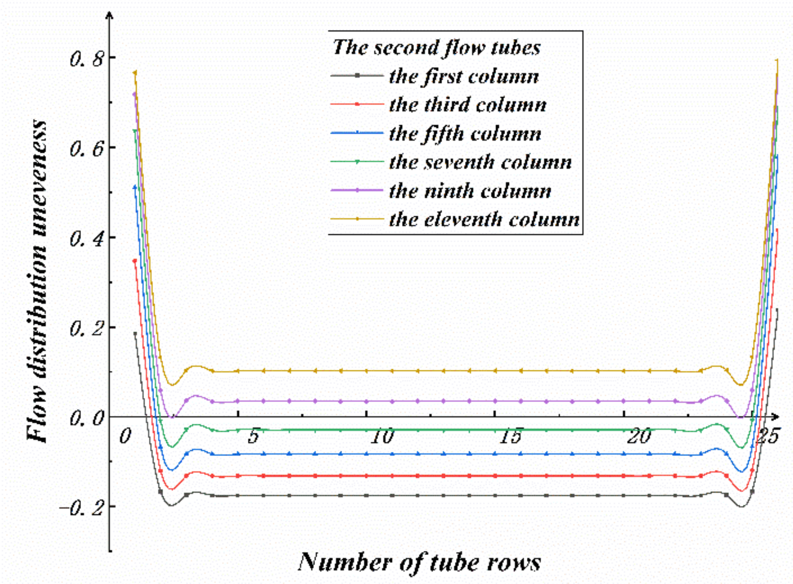
3.2 Flow Distribution Unevenness

Since the supercritical CO_2 side is double-flow, the variation of the flow distribution unevenness in each flow path with the number of tube rows is shown in the Fig.8. The flow distribution unevenness of the first flow path is significantly higher than that of the second flow path. The mass flow distribution unevenness in the middle position of each column is basically identical while the difference is distinct in various columns. In the same flow path, the flow distribution unevenness of tubes that are contact with high-temperature air earliest is negative and the flow distribution unevenness of tubes that the last contact is conversely positive.

The analysis shows that the temperature difference between the hot and cold fluids at the location near the entrance of high-temperature air is notable that could cause enough heat transfer to make obviously temperature rising of supercritical CO_2. The dynamic viscosity at the above location is higher and the friction coefficient increases accordingly. Then, the mass flow rate in the tube is lower than the average value according to the flow distribution calculation formula. Meanwhile, the temperature distribution near the shell is lower than the center position, the dynamic viscosity is less affected by the temperature than the center position, and the friction coefficient is smaller, so the mass flow rate is larger.



(a)The flow distribution unevenness of various tube rows in first flow



(b) The flow distribution unevenness of various tube rows in second flow

Figure 8. The variation of the flow distribution unevenness in each flow with the number of tube rows

3.3 Influence of pressure and diameter on total unevenness and heat dissipation

Fig 9-10 respectively show the trend of total flow distribution unevenness (S) and heat dissipation rate (Q_{loss}/Q) with P_c for different diameters. The black dotted-lines indicate the calculation result of do=1.0mm, and the red color indicates do = 1.6 mm. In Fig. 9 P_c almost has no effect on the total unevenness while diameter has great influence that the larger the diameter is, the higher the total unevenness S.

The temperature distribution range and trend of the heat exchangers with the same power are similar, so the dynamic viscosity is basically the same. The analysis shows that larger diameter makes G_c increase, which increase Re. According to the Darcy-Weisbach Formula, the friction coefficient decreases. In the flow distribution formula, ΔP_{max}, ρ_m, L and d_i in the same flow path are consistent. The reduction of the friction coefficient increases mass flux, resulting in more mass flow. However, the mass flow will continue to affect the Reynolds number, resulting in uneven distribution of mass flow, which also makes the total flow unevenness of the large diameter heat exchanger higher.

The heat dissipation rate shown in Figure 10 decreases as P_c increases. As P_c rises, the microchannel number required for the same power heat exchanger is less, the heat exchanger becomes more compact, the heat exchange power per unit volume increases, and the heat dissipation rate decreases. The heat transfer power per unit volume of the small-diameter heat exchanger increases with the increasing P_c, which leads to the more severe the heat dissipation rate of such heat exchanger.

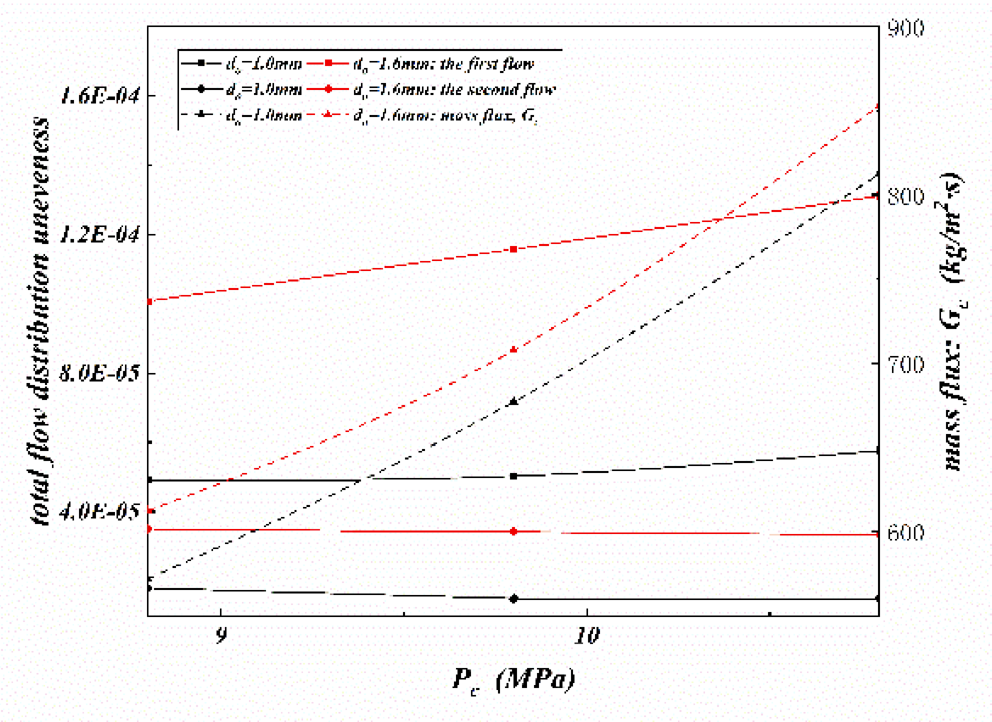


Figure 9. the trend of total flow distribution unevenness with P_c for different diameters

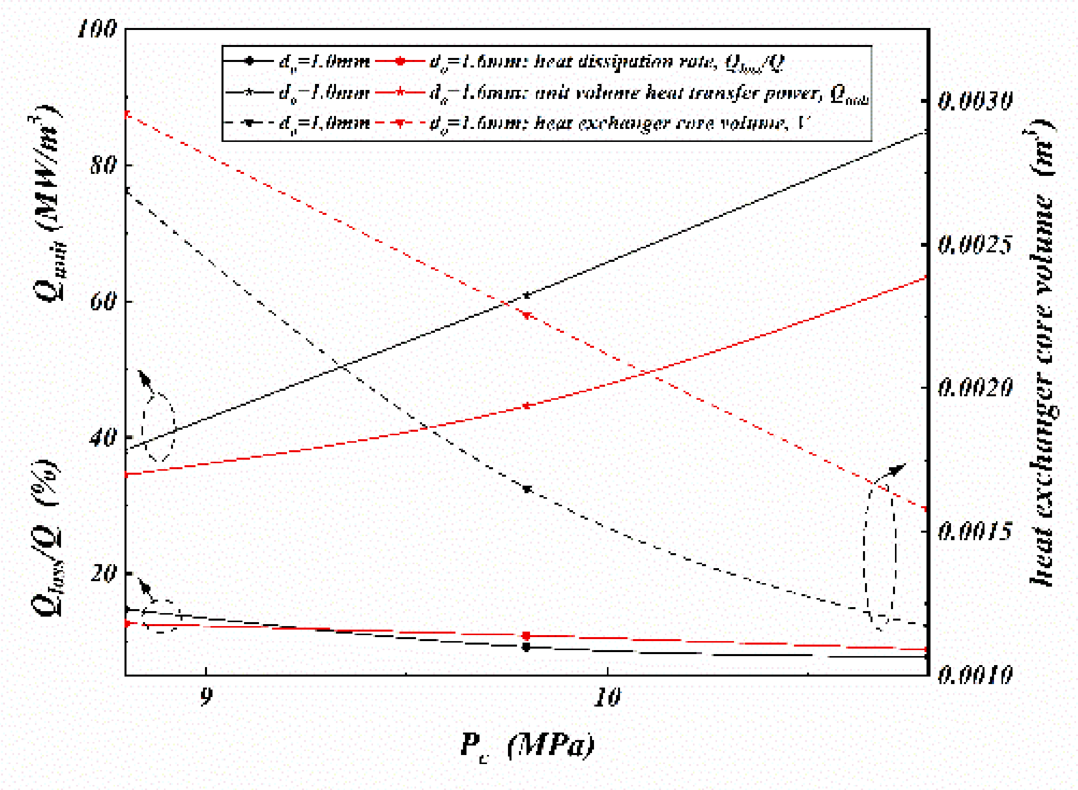


Figure 10. the trend of heat dissipation rate (Q_{loss}/Q) with P_c for different diameters

4. Conclusion

Microchannel shell-and-tube heat exchangers with supercritical CO_2 generally have higher working temperature. Most thermal calculations do not consider the heat dissipation of such compact heat exchangers to the environment and the uneven distribution of mass flow between processes. In this paper, the research on variable physical properties of supercritical fluid, heat dissipation of high temperature working conditions and uneven distribution of mass flow between microchannels are carried out, and summarized as follows:

1) When considering heat dissipation, the temperature distribution of the outermost rows is significantly different from that of the intermediate rows closer to the relative ideal heat transfer.

2) Under high temperature working conditions, the heat dissipation cannot be ignored even the microchannel heat exchanger.

3) When the same diameter is used, the number of microchannel reduces with the increasing of the supercritical CO_2 pressure, which increases the heat exchange power per unit volume, thereby reducing the heat dissipation rate. At the same pressure, the heat transfer power per unit volume with small diameter increases more intensely causing heat dissipation rate decreases more severely.

4) When the same diameter is affected by the temperature distribution, the dynamic viscosity is different, which affects the friction coefficient, resulting in uneven flow distribution between flow paths. While mass flux varying with diameters would affect the Reynolds number, which could cause the friction coefficient to affect the flow distribution.

Acknowledgments

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