Numerical Investigation on Conjugate Heat Transfer in Helical Coiled Tube under Vertical Placement

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Abstract. Conjugate heat transfer of supercritical CO₂ inside helical coiled tube under vertical placement was investigated numerically in the present study. It can be concluded that the flow field of supercritical fluid is affected by both the buoyancy force and centrifugal force in conjugate heat transfer process. The dual effects of buoyancy force and centrifugal force lead to the deflection of second flow direction for the vertical placement, and further results in the heat transfer deterioration region on top generatrix wall for the downward flow larger than that for the upward flow. However, the overall heat transfer performance is slightly better than that in the upward flow in tube side.

1. Introduction

Due to the super thermos-physical properties, thermal stability and eco-friendliness, supercritical CO₂ has good application prospect in fields such as new nuclear power plant, solar power station, enhanced geothermal system and recovery or reuse of low grade thermal energy, etc [1]. Many experiments of heated circular tubes cooled with supercritical CO₂ have been performed. Comprehensive reviews of earlier experimental studies on heat transfer to supercritical fluid in straight pipes were provided [2, 3]. In addition, it has been widely reported in literature that heat transfer rates in helical coils are higher compared to a straight tube. However, the combined effects of the centrifugal force and the variation of the thermal-physical properties make the heat transfer mechanism more complex. A review on heat transfer and friction coefficient correlations in helical or curved tubes were presented by Vashisth et al[4].

So far, there have been few studies about the influences of centrifugal force on the convective heat transfer of supercritical fluid. Ciofalo et al.[5] numerically studied the relationship of the buoyancy and centrifugal force in the curved tube under the laminar condition. The results showed that the centrifugal force aggravates the density difference in the cross section. Zhang et al.[6, 7] studied the heat transfer characteristics of S-CO₂ flowing in a vertical helical coiled tube. The analysis on the average heat transfer coefficient along the flow direction showed that the heat transfer is dominated by forced convection, and the influence of buoyancy force on heat transfer can be ignored. Lazova et al.[8] experimentally studied the heat transfer performance of supercritical R-404a in a helically-coiled-tube heat exchanger. Results showed that the increasing of mass flow rate can obviously improve the efficiency of heat exchanger. The flow and heat transfer characteristics of China No.3 aviation kerosene in a heated curved tube under supercritical pressure are numerically investigated by Li and Huai[9].
To the author's knowledge, there has been no research on turbulent conjugate heat transfer to supercritical fluid in the helical coiled tube heat exchanger. Thus, it is highly needed to understand this phenomenon because of its wide applications in power generation systems and reactor facilities. Moreover, the conjugate heat transfer features of supercritical fluid in helical coiled tube are much different from those in straight pipes. This study was performed to investigate the combined effects of the centrifugal force and buoyancy force on the mixed turbulent flow and conjugate heat transfer of supercritical fluid at 7.58MPa.

2. Geometric and Performance Parameters of the Heat Exchanger

Fig. 1 shows the physical model of the heat exchanger studied here for the numerical simulation. The high temperature CO2 transmits the heat to the low temperature supercritical CO2 by the tube wall. Fig. 1 shows 4 typical generatrixes, where the top is \( \eta=0^\circ \), the outer \( \eta=90^\circ \), the bottom \( \eta=180^\circ \) and the inner \( \eta=270^\circ \).

![Fig. 1 Schematics of the heat exchanger with helical coiled tube](image)

3. Methodology

The governing equations of the problem to be investigated are given as:

\[
\frac{\partial u_i}{\partial x_i} = 0
\]  

\[
\frac{\partial}{\partial x_j}\left[ \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \frac{\partial u_k}{\partial x_k} \delta_{ij} \right) - \rho \mu \mu_j - \delta_{ij} p \right] = 0
\]  

\[
\frac{\partial}{\partial x_j}\left[ \frac{1}{r} \frac{\partial T}{\partial x_j} - \rho u_j C_p T \right] + \mu \Phi_v = 0
\]

where \( \mu \Phi_v \) is the viscous heating term in energy equation.

\[
\Phi_v = \frac{\partial u_i}{\partial x_i} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \frac{\partial u_k}{\partial x_k} \delta_{ij} \right) \text{ and } \Gamma = \mu / Pr
\]

The AKN (Abe-Kondoh-Nagano) \( k-\varepsilon \) model was chosen to consider the turbulence and heat transfer of supercritical CO2 inside the heat exchanger with helical coiled tube [10, 11].

Turbulent kinetic energy:

\[
\frac{\partial}{\partial x} (\rho k) + \frac{1}{r} \frac{\partial}{\partial r} (r \rho v k) = \frac{\partial}{\partial x} \left[ \left( \mu + \frac{\mu_k}{\sigma_k} \right) \frac{\partial k}{\partial x} \right] + \frac{1}{r} \frac{\partial}{\partial r} \left[ \left( \mu + \frac{\mu_k}{\sigma_k} \right) \frac{\partial k}{\partial r} \right] + P_k + G_k - \rho \varepsilon
\]

where the production term due to the mean velocity gradient is:

\[
P_k = \mu \left[ 2 \left( \frac{\partial u}{\partial x} \right)^2 + \left( \frac{\partial v}{\partial r} \right)^2 + \left( \frac{\partial v}{\partial x} \right)^2 + \frac{\partial u}{\partial r} \frac{\partial v}{\partial x} \right] \]

2
Turbulence dissipation rate:

\[
\frac{\partial}{\partial x} \left( \rho \mu \varepsilon \right) + \frac{1}{r} \frac{\partial}{\partial r} \left( r \rho \nu \varepsilon \right) = \frac{\partial}{\partial x} \left[ \left( \mu + \frac{\mu_t}{\sigma_v} \right) \frac{\partial \varepsilon}{\partial x} \right] + \frac{1}{r} \frac{\partial}{\partial r} \left[ \left( \mu + \frac{\mu_t}{\sigma_v} \right) \frac{\partial \varepsilon}{\partial r} \right] + C_{\varepsilon_1}f_1 \frac{1}{T_i} (P'_k + G'_k) - C_{\varepsilon_2}f_2 \frac{\rho \varepsilon}{T_i} \tag{7}
\]

The turbulent viscosity is defined as:

\[\mu_t = \rho C_{\mu} \nu^2 T_i = \rho C_{\mu} f_{\mu} \frac{k^2}{\varepsilon} \tag{8}\]

More details about the model can be found in Ref.[12].

The National Institute of Standards and Technology (NIST) database was used to calculate the thermophysical properties of CO\textsubscript{2}. Fig. 2 provides the grid used for simulation, the grid number reaches one million. The calculation sensitivity on mesh was checked and the grid independent solutions were obtained. SIMPLEC algorithm as adopted for the Pressure-velocity coupling. The verification of numerical simulation has been completed in the previous article [11].

![Fig. 2 Local structured grid on helical coil](image)

4. Results and discussions

Fig. 3 reveals the wall temperature profile along the coiled tube for different flow directions when \(T_{\text{shell,0}}\) is 600 K, \(G_{\text{shell}}\) is 200 kg m\textsuperscript{-2} s\textsuperscript{-1}, \(G_{\text{Tube}}\) is 500 kg m\textsuperscript{-2} s\textsuperscript{-1} and \(T_{\text{Tube,0}}\) is 250 K. The top-generatrix wall temperature and inner-generatrix temperature were higher than the temperature at the other two generatrixes when \(\theta\) is lower than 360°. The present computation shows that the buoyancy causes a strong secondary flow along the gravity direction, which leads to the accumulation of low-density fluid on the top. The deterioration of heat transfer is magnified on the top resulting from the effects of the buoyancy, so the wall temperature on the bottom wall is lower than those on the top wall. In addition, the local heat transfer deterioration region on top-generatrix for the vertical downward flow is larger than that for the vertical upward flow.

When 1080°<\(\theta\)<2160°, the local wall temperature on all four generatrixes have significantly decreased. The wall temperature decrease of the top-generatrix is the most obvious, and the decrease in temperature reached 7%. The wall temperature of the top-generatrix is very close to the bottom-generatrix. This shows that continuously stratified fluid caused by buoyancy has been significantly improved. The effect of buoyancy becomes negligible compared to the centrifugal force, so the effects of the centrifugal force are predominant.
Fig. 3 The wall temperature along the coiled tube

(a) Helical upward flow             (b) Helical downward flow

Fig. 4 shows the variation of the average heat transfer coefficient under when $T_{\text{shell,0}}$ is 600 K, $G_{\text{shell}}$ is 200 kg/m$^2$/s$^{-1}$, $G_{\text{tube}}$ is 500 kg/m$^2$/s$^{-1}$ and $T_{\text{Tube,0}}$ is 250 K. The maximum the heat transfer coefficient in the helical upward flow is significantly lower than that in the helical downward flow. The angle between the gravity direction and the flow the is less than 90°, therefore the high enosity fluid directly collides toward the bottom wall which can worsen the cooling condition on the top. In addition, after $\theta>2160°$, the heat transfer coefficients for the different flow directions are almost the same, indicating that the buoyancy effect has completely disappeared.

Fig. 4 The average heat transfer coefficient along the coiled tube for different flow directions

Fig. 5 shows the contours of velocity at different axial positions. It can be seen from Fig. 5 that the higher flow velocity occurs between the inner and top generatrixes during the helical upward flow at $\theta<1080°$. The angle between the gravity direction and the flow is less than 90°, the fluid with low density in the high temperature region is affected by significant buoyancy. The buoyant force produces fluid acceleration phenomenon, leading to that the high temperature region is accompanied by high velocity region. However, the flow velocity in the higher temperature region is relatively lower in the helical downward flow. This is due to that the angle between the buoyance force and the flow direction is greater than 90°. Therefore, the buoyant force produces the deceleration effect, resulting in the slower flow velocity in the region with high temperature. As the temperature of the main fluid continues to increase, the maximum velocity of the supercritical fluid appears near the outer generatrix, showing a typical fluid velocity distribution in the coiled tube.
Fig. 5 Distribution of axial velocity along coiled tube (The unit is m/s) 
(upper row: Helical flow upward; lower row: Helical flow downward)

Fig. 6 shows the wall temperature profile along the coiled tube for different flow directions when $T_{shell,0}$ is 600 K, $G_{shell}$ is 200 kg·m$^{-2}$·s$^{-1}$, $G_{tube}$ is 4000 kg·m$^{-2}$·s$^{-1}$ and $T_{tube,0}$ is 250 K. When the mass flow rate of the supercritical fluid in tube side is large enough, the flow directions will not affect the coupled heat transfer process of the supercritical fluid. The buoyancy effect caused by the fluid density difference will not affect the flow field in the tube, and the heat transfer of the supercritical fluid in the tube side is mainly affected by the centrifugal force caused by the helical structure.

Fig. 6 The wall temperature for different flow directions at $G_{tube}$=4000 kg·m$^{-2}$·s$^{-1}$

Fig. 7 shows the temperature and velocity distribution in the tube side at $G_{tube}$=4000 kg·m$^{-2}$·s$^{-1}$. It can be seen that the temperature near the inner-generatrix is the highest, while the temperature near the outer-generatrix is the lowest. This is caused by the secondary flow effect in the helical coiled tube. The maximum velocity always appears near the outer-generatrix. Therefore, the buoyancy does not have a significant effect on heat transfer, which is consistent with the wall temperature results. When the mass flow rate of the supercritical fluid is 4000 kg·m$^{-2}$·s$^{-1}$, the buoyancy effect due to the density difference is negligible. The heat transfer and flow of the fluid are only affected by the centrifugal force influences.
Temperature Distribution (The unit is K)

velocity distribution (The unit is m/s)

Fig. 7 Fluid temperature and axial velocity distributions at $G_{\text{Tube}}=4000$ kg·m$^{-2}$·s$^{-1}$

Fig. 8 shows the wall temperature under the constant density assumption. It can be seen that when the supercritical fluid density is assumed to be a constant value, the flow direction has no effect on heat transfer. The difference in heat transfer under different flow directions caused by the buoyancy force no longer exists. However, the temperature difference between the inner and outer generatrices caused by centrifugal force still exists. This shows that even if the density of the fluid is constant, the secondary flow in the helical tube will still cause the inner generatrix wall temperature to be too high.

5. Conclusions

The numerical calculation results on the heat transfer of supercritical CO$_2$ in the vertical helical coiled tube heat exchanger have been provided. Conclusions can be obtained as follows:

1. For the vertical configuration, the dual effects of centrifugal force and buoyancy force cause the deflection of the direction of the 2nd flow.
2. The deterioration of local heat transfer on the upper generatrix wall in the flow downward is higher than local heat transfer deterioration area in the upward flow in tube side, the decrease of temperature in top-generatrix reached 7%, although the performance of heat transfer is higher than that in the flow upward in the tube side.

3. As the mass flux of fluid $G_{\text{Tube}}$ increased to 4000 kg·m$^{-2}$·s$^{-1}$, the effect of buoyancy becomes negligible compared to the effect of centrifugal force. The coupled heat transfer performance of the supercritical fluids have nothing to do with the flow direction.

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