Postprint: Study on the Cooling Heat Transfer Mechanism of Supercritical Pressure CO2 in Horizontal Tubes

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Abstract

A numerical investigation of convective heat transfer of supercritical pressure CO2 in horizontal tubes under cooling conditions was conducted using the SST k-ω model, analyzing the effects of fluid properties, heat flux, diameter, and buoyancy on the flow and heat transfer characteristics near the pseudo-critical point, and examining the heat transfer mechanism of supercritical pressure CO2 from the field synergy perspective. The results show that buoyancy effects cause asymmetric temperature fields and secondary flow phenomena in the flow cross-section; the convective heat transfer coefficient at the lower wall reaches its peak earlier than that at the upper wall, but remains smaller than that at the upper wall; increasing heat flux has a minor effect on the heat transfer coefficient but shifts its peak toward the inlet section; increasing both heat flux and diameter enhances the influence of buoyancy effects on fluid heat transfer characteristics; and the field synergy principle can explain the non-uniform heat transfer phenomenon at the same cross-section.

Full Text

Preamble

A Study on the Cooling Heat Transfer Mechanism for Supercritical Pressure CO_2 in Horizontal Tubes

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Abstract

The convective heat transfer of supercritical pressure CO_2 in a horizontal tube under cooling conditions is numerically investigated using the SST k-ω turbulent model. The effects of thermophysical properties, heat flux, tube diameter, and buoyancy on the heat transfer characteristics near the pseudo-critical point are analyzed, and the heat transfer mechanism is examined from the perspective of field synergy principle. The results demonstrate that buoyancy effects induce asymmetric temperature distributions and secondary flow phenomena in the cross-section. The peak heat transfer coefficient on the bottom wall appears earlier than on the top wall, but its magnitude is smaller. Increasing heat flux has minimal impact on the peak value of the heat transfer coefficient but causes the peak to shift toward the inlet. Larger heat flux and larger diameter both intensify the influence of buoyancy on heat transfer characteristics. The field synergy principle can effectively explain the non-uniform heat transfer phenomena at the same cross-section.

Keywords: supercritical pressure CO_2; convective heat transfer; field synergy principle; buoyancy effect; numerical simulation

Introduction

Supercritical CO_2 offers numerous advantages including non-toxicity, non-flammability, chemical stability, and low cost. Due to its dramatic property variations near the critical and pseudo-critical points, supercritical CO_2 exhibits excellent flow and heat transfer characteristics, making it highly promising for applications in nuclear reactors, solar energy systems, refrigeration systems, and other advanced energy technologies [1, 2]. Compared with conventional constant-property fluids, supercritical CO_2 undergoes drastic thermophysical property changes near the critical region, resulting in unique and complex heat transfer behavior that has become a focal point of recent research [3].

Dang and Hihara [4, 5] employed experimental and numerical methods to investigate the effects of diameter, mass flow rate, and heat flux on convective heat transfer characteristics of supercritical pressure CO_2 in horizontal tubes, assuming negligible buoyancy effects. Du et al. [6] discovered that buoyancy effects actually enhance heat transfer near the pseudo-critical point in horizontal tubes. Guo Zengyuan et al. [7-9] proposed the field synergy principle, which posits that heat source intensity depends not only on fluid properties and velocity but also on the synergy between velocity and temperature gradient fields. When the angle between velocity and temperature gradient vectors is less than 90°, smaller angles yield better heat transfer performance.

This study employs numerical methods to investigate convective heat transfer characteristics of supercritical pressure CO_2 near the pseudo-critical point (Tpc = 307.8 K, p = 8 MPa), focusing on analyzing the effects of heat flux, diameter, and buoyancy on local convective heat transfer intensity. The research further examines the local heat transfer mechanism from the field synergy perspective to provide theoretical foundations for the development and design of high-efficiency heat exchangers.

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1.1 Physical Model

Due to the severe thermophysical property variations of supercritical pressure CO_2, a three-dimensional model was established to accurately simulate local convective heat transfer characteristics in horizontal tubes, as shown in [Figure 1: see original paper]. The model consists of horizontal tubes with diameters of 2, 4, and 6 mm, and a total length of 840 mm, comprising a 240 mm adiabatic inlet section and a 600 mm cooling section. This configuration ensures flow approaches fully developed conditions at the cooling section entrance, minimizing inlet effects.

1.2 Governing Equations

The governing equations in Cartesian coordinates are as follows [10, 11]:

Continuity equation:
Momentum equation:
Energy equation:

where Φ represents viscous dissipation and t denotes the turbulent viscosity based on the turbulence model.

1.3 Numerical Methods and Boundary Conditions

Numerical simulations were performed using ANSYS CFX [12]. Thermophysical properties of supercritical CO_2 were calculated using the NIST Standard Reference Database 23 (REFPROP) Version 7 [13] and incorporated into CFX via RGP property files. Compared with linear interpolation methods, RGP files more accurately capture dramatic property variations. The pressure-velocity coupling algorithm and SST k-ω model were employed, combining the robustness and independence of the k- model with the near-wall accuracy of the k-ω model. Convergence was achieved when residuals for all governing equations fell below 10^{-6}.

Boundary conditions included mass flow inlet, pressure outlet, adiabatic inlet section walls, and constant heat flux cooling section walls. Operating parameters covered mass fluxes of 100-300 kg/m^2·s, pressure of 8 MPa, inlet temperature of 340.15 K, heat fluxes of 35-45 kW/m^2, and inlet Reynolds numbers of approximately 3\$×10ˆ{4}\$.

1.4 Grid Independence Verification and Numerical Method Validation

Grids were generated using ANSYS ICEM with radial refinement to ensure near-wall $y^+ < 1$, satisfying SST turbulence model requirements. Grid quality exceeded 0.6, meeting simulation standards. Grid independence verification results are presented in . The relative error in heat transfer coefficient between grid sets 3 and 5 was only 0.2%. Considering computational time and accuracy, grid No. 3 was selected for subsequent simulations.

To validate numerical accuracy, simulation results were compared with experimental data from Dang and Hihara [4] under identical geometric and boundary conditions. The comparison shown in [Figure 2: see original paper] reveals a maximum relative error of 13.5%, confirming the accuracy and reliability of the numerical method.

Table 1 Mesh Independence Verification

No.	Grid Number	h ($W \cdot m^{-2} \cdot K^{-1}$)	Error (%)

2 Field Synergy Principle

For a general three-dimensional model under steady-state conditions without internal heat sources, the integral form of the convective heat transfer energy equation is [7]:

where , cp, and k represent fluid density, specific heat at constant pressure, and thermal conductivity, respectively; δt denotes thermal boundary layer thickness; and qw is wall heat flux.

Neglecting axial heat conduction, Equation (4) can be written in vector form:

Introducing dimensionless variables:

where ub and Tb are bulk velocity and temperature, respectively, and Tw is wall temperature.

Substituting Equation (6) into Equation (5) yields the dimensionless relationship:

where h is the convective heat transfer coefficient, Reynolds number Re = ubD/ , subscript b denotes bulk fluid, and subscript w denotes wall conditions. The term U · T can be expressed as:

where β is the angle between velocity and temperature gradient vectors, known as the field synergy angle.

Equation (7) indicates that for variable-property fluids, Nub depends on flow characteristics, fluid properties, channel diameter, and the synergy between velocity and temperature gradient fields, making it a complex multifactorial relationship.

3 Results and Discussion

3.1 Effect of Heat Flux on Flow and Heat Transfer

To investigate heat flux effects, simulations were conducted using a 6 mm diameter tube at 8 MPa pressure, 340.15 K inlet temperature, with heat fluxes of 35 and 45 kW/m^2.

Figure 3: see original paper presents the axial distributions of bulk fluid temperature (Tb) and local wall temperature (Tw). Under constant heat flux, both fluid and wall temperatures gradually decrease below the pseudo-critical temperature along the flow direction. The top wall temperature decreases faster than the bottom wall temperature due to buoyancy effects caused by dramatic density differences from supercritical CO_2 property variations, resulting in asymmetric wall temperature distributions. Increased heat flux accelerates fluid cooling and significantly enlarges the temperature difference between top and bottom walls, indicating that higher heat flux enhances buoyancy effects on supercritical CO_2 flow and heat transfer.

Figure 3: see original paper shows the axial distribution of convective heat transfer coefficient (h), which first increases to a peak then decreases, consistent with the temperature-dependent trend of supercritical CO_2 specific heat at constant pressure. The bottom wall heat transfer coefficient reaches its peak earlier than the top wall, but with a significantly lower magnitude. Under cooling conditions, near-wall fluid cools first; buoyancy drives hotter, less dense fluid upward, creating smaller temperature differences between the top wall and bulk fluid, thus yielding higher heat transfer coefficients on the top wall. While increased heat flux barely affects the peak heat transfer coefficient value, it advances the peak location toward the inlet because higher heat flux reduces fluid temperature to the pseudo-critical point more rapidly, where the heat transfer coefficient peaks.

3.2 Effect of Buoyancy on Flow and Heat Transfer

[Figure 4: see original paper] illustrates the local Nub distribution at different heat fluxes. The Nub peak increases slightly with heat flux, suggesting that higher heat flux enhances convective heat transfer intensity. [Figure 5: see original paper] shows the corresponding local field synergy angle distribution. The field synergy angle varies non-uniformly along the flow direction, with smaller angles near the top wall than the bottom wall, indicating better synergy between velocity and temperature gradient fields near the top wall and consequently superior heat transfer characteristics. Under identical conditions, the difference in field synergy angles between top and bottom walls increases with heat flux, demonstrating that higher heat flux amplifies the non-uniformity of convective heat transfer fields induced by buoyancy effects.

To more intuitively understand buoyancy effects, simulations were performed without gravity (g = 0) and compared with cases including gravity (gy = -9.81 m/s^2). [Figure 6: see original paper] presents the effect of gravity on Nub distribution. Local convective heat transfer intensity is slightly higher with gravity than without, indicating that gravity-induced buoyancy enhances heat transfer, primarily in the latter half of the cooling section where bulk fluid temperature falls below the pseudo-critical temperature. Buoyancy effects become more pronounced with increasing heat flux.

[Figure 7: see original paper] shows the corresponding local field synergy angles. Interestingly, field synergy angles without gravity are smaller than those with gravity. For supercritical fluids with dramatic property variations, field synergy principle application cannot be simply attributed to decreasing synergy angles; multiple influencing factors must be considered comprehensively.

[Figure 8: see original paper] displays the axial distribution of the equivalent heat source term (- cU · T) with and without gravity. The equivalent heat source is significantly larger with gravity and shows non-uniform axial distribution, increasing with heat flux. According to Equations (5) and (7), larger equivalent heat sources enhance heat transfer intensity. The equivalent heat source depends not only on velocity, temperature, and their included angle but also on fluid properties. Since supercritical CO_2 properties vary dramatically with temperature, these property variations become the dominant factor governing flow and heat transfer characteristics.

3.3 Effect of Diameter on Flow and Heat Transfer

To examine diameter effects, simulations were conducted using tubes of 2, 4, and 6 mm diameter at 45 kW/m^2 heat flux, 8 MPa pressure, 340.15 K inlet temperature, with constant inlet Reynolds number of 3$×10ˆ{4}$.

[Figure 9: see original paper] shows Nub distributions for different diameters. Before reaching the peak, Nub increases with diameter; after the peak, Nub decreases gradually and becomes less sensitive to diameter. The corresponding field synergy angle distribution is presented in [Figure 10: see original paper]. Before the fluid temperature reaches the pseudo-critical point, the field synergy angle decreases significantly with increasing diameter, indicating that larger diameters improve synergy between velocity and temperature gradient fields, thereby enhancing heat transfer intensity. When fluid temperature falls below the pseudo-critical point, diameter has minimal influence on field synergy angle and consequently limited impact on heat transfer coefficient.

[Figure 11: see original paper] illustrates temperature, radial velocity, and field synergy angle distributions at the z = 390 mm cross-section for 2 mm and 6 mm diameter tubes. The cross-section exhibits asymmetric temperature distribution, with hotter, less dense fluid accumulating in the upper region and cooler, denser fluid concentrating at the bottom. Gravity-induced buoyancy creates secondary flow that enhances fluid mixing. Field synergy angles also show non-uniform distribution, with smaller angles in the upper region than at the bottom. As diameter decreases, temperature and field synergy angle distributions become more uniform, and secondary flow intensity diminishes.

Conclusions

This study numerically investigated convective heat transfer characteristics of supercritical pressure CO_2 in horizontal tubes under cooling conditions. The primary conclusions are as follows:

Buoyancy effects cause asymmetric temperature field distributions in the cross-section, with top wall temperatures exceeding bottom wall temperatures at the same location. Near-wall fluid cools first, creating significant density differences that drive secondary flow formation. The bottom wall heat transfer coefficient reaches its peak earlier than the top wall, but with a lower magnitude, while field synergy angles at the top are smaller than at the bottom. Increasing heat flux shifts the heat transfer coefficient peak toward the inlet, and both increased heat flux and larger tube diameter amplify buoyancy effects on heat transfer characteristics. Compared with the g = 0 case, the gravity case (gy = -9.81 m/s^2) exhibits larger field synergy angles but also larger equivalent heat sources, resulting in overall enhanced heat transfer intensity.

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