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Optimizing Supercritical Heat Exchangers: A Study on Entrance Effect and Multi-Stage Heating Approach#

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ABSTRACT

Supercritical heat exchangers encounter a variety of non-uniform heat flux during operation, which significantly influence the flow and heat transfer processes. Numerical simulation is employed to investigate the process of supercritical water entering and leaving the heated section, which are divided into three stages: Thermal Establishment Stage, Axially Asymptotic Developed Stage, and Thermal Removal Stage. During Thermal Establishment Stage, influenced by entrance effect, the affected range in this study extends up to 150 z/d. This stage is characterized by a heat transfer coefficient higher than that of the stable state with the same bulk enthalpy. Following sufficient heating, supercritical water reaches the Axially Asymptotic Developed Stage, where the flow and heat transfer processes become independent of inlet parameters. After heating, the top wall temperature drops rapidly, primarily due to the convective heat transfer of the supercritical fluid inside the tube. The Multi-Stage Heating Approach proposed on the basis of this study can effectively relieve the heat transfer deterioration and significantly enhances the overall heat transfer performance of the heat exchanger near the pseudo-critical temperature. Results indicate that, compared to conditions without entrance effect, the overall heat transfer coefficient can increase by up to 28.15%, 58.01%, and 92.24% for one-stage, two-stage, and three-stage heating, respectively.

Keywords: Supercritical water heat exchangers, advanced energy technologies, the entrance effect, horizontal tubes, multi-stage Heating Approach

NONMENCLATURE

Abbreviations	
HTD	Heat transfer deterioration
SST	Shear Stress Transport
SWBT	Supercritical Water Buoyancy-Tuned Turbulent Prandtl
HTC	Heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$)
AADS	Axially Asymptotic Developed Stage
Symbols	
Cp	Specific heat capacity ($J \cdot kg^{-1} \cdot K^{-1}$)
D	Inner diameter (m)
G	Mass flux ($kg \cdot m^{-2} \cdot s^{-1}$)
P	Pressure (MPa)
Pe_t	Turbulent Peclet number
Pr_t	Turbulent Prandtl number
q	Heat flux ($kW \cdot m^{-2}$)
R	Radial length (m)
T	Temperature (K)
V	Velocity ($m \cdot s^{-1}$)
y^+	Non-dimensional distance (m)
k	turbulent kinetic energy ($kg \cdot m \cdot s^{-2}$)
ω	Specific dissipation (s^{-1})
μ	Dynamic viscosity (Pa·s)
μ_t	Turbulent eddy viscosity (Pa·s)
Subscripts	
b	Bulk
cr	Critical
ave	Average
in	Inlet
pc	Pseudo-critical

1. INTRODUCTION

With escalating energy consumption, supercritical fluid heat exchangers, renowned for their exceptional heat transfer efficiency, are extensively utilized in supercritical coal-fired power plants [1], supercritical reactors [2] and solar tower power plants [3]. To deepen the understanding of heat transfer characteristics of supercritical fluids, scholars have engaged in comprehensive experimental and numerical simulation studies. These investigations examine the impacts of various parameters, including mass flux, heat flux, pressure, and tube structure, on the heat transfer mechanisms of supercritical fluids, as reviewed by Huang et al. [4], Mao et al. [5], and Xie et al. [6]. Despite

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significant research efforts, the complex challenges associated with heat transfer in supercritical fluids, particularly heat transfer deterioration (HTD), remain only partially understood.

Research on supercritical fluids has traditionally concentrated on scenarios with uniform heating boundary conditions. However, supercritical fluid heat exchangers often face non-uniform thermal flux distributions along both axial and circumferential directions, which add complexity to the flow and heat transfer processes within the tubes. Recently, some scholars have shifted their focus towards the heat transfer performance of supercritical fluids under variable heat fluxes. This shift is particularly motivated by the uneven solar irradiance in solar power stations and the heterogeneous heat flux boundary conditions in boiler water walls. These studies explore the significance of non-uniform heat flux in improving HTD and enhancing heat transfer coefficient (HTC). However, the interaction between property anomalies and variable heat fluxes accentuates local effects, making the establishment of a relationship between these factors fundamental to improving the performance of supercritical fluid heat exchangers. As Li et al. [7] have pointed out, leveraging constructal law to adjust the axial heat flux distribution could minimize thermal resistance and thereby boost overall heat transfer efficiency or reduce peak temperatures.

The entrance effect, typically caused by the development of hydraulic and thermal boundary layers, is one of the most common and fundamental phenomena of axial heat flux variations. Compared to fluids with constant properties, the entrance effect of supercritical fluids extends over a longer distance and has a more pronounced impact. Tian et al. [8] examined the flow and heat transfer processes of R134a in horizontal tubes and found that the entrance effect could influence the system up to 150 z/d. Numerical analyses indicate that stratification and secondary flow structures, stemming from buoyancy effects, play a pivotal role during this stage. These effects persist over extended lengths when buoyancy and stratification are pronounced. The principle of the entrance effect has been widely adopted to improve the heat transfer performance of heat exchangers, including the installation of vortex generators to disrupt the boundary layer for enhancing the efficiency. In addition, the improved effect of the entrance effect on heat transfer may be capable of being used to relieve HTD, but this requires further research.

This paper effectively simulates the heat exchange process of supercritical water around the heated section using a previously established turbulence model by the author. The process is divided into three stages and each stage is described in detail, followed by a discussion of how Multi-stage Heating Approach can enhance the performance of supercritical heat exchangers.

1.1 Physical model and boundary conditions

Based on the actual dimensions of tubes in boilers, tube with an internal diameter of 26mm and a wall thickness of 3mm are selected for the simulation study. The schematic diagrams of the two physical models are shown in Fig. 1. Model A is 9m long and includes a 1.5m inlet section, a 6m heated section, and a 1.5m outlet section, which is used to establish temperature variation curves under different inlet temperatures. Model B is 15m long, including a 1.5m inlet section, a 12m intermediate section, and a 1.5m outlet section. The intermediate section is divided into eight segments, and the labels indicate the specific locations where heat flux is applied (for example, model B1245 indicates that heat flux is applied in segments 1, 2, 4, and 5). Constant mass flux and pressure outlet boundary conditions are applied at both the inlet and outlet. Considering that the axial and radial pressure drops are negligible compared to the working pressure, it is assumed that the thermal properties are solely functions of temperature under constant pressure conditions. Thermal properties are sourced from the NIST Standard Reference Database (REFPROP) Version 9.1.

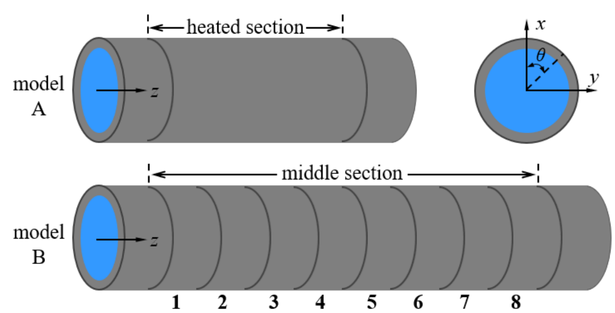


Fig. 1 Schematic of the model

1.2 Governing Equations and turbulence model

The flow and heat transfer characteristics for tubes with continuous and discontinuous heat flux were studied numerically using the commercial software Ansys 2022 R1. The z-axis is designated as the positive direction of flow, with the gravity direction aligned negatively along the y-axis. The governing equations include the continuity equation, momentum equation, energy equation, and a two-equation turbulence model.

The SST k-ω turbulence model is employed due to its demonstrated predictive capability in supercritical fluid flow and heat transfer processes.[9, 10] The SIMPLE algorithm was used for pressure–velocity coupling and the second-order upwind method was adopted for discretization of the transport equations. The above equations can be found in the Ref. [11]. In the previous study, a new Pr_t model is proposed to improve the predictive accuracy of numerical simulation, which is named Supercritical Water Buoyancy-Tuned Turbulent Prandtl Model (SWBT model). It is denoted in Eq. (1).

$$
Pr_t = \begin{cases} 0.3 & \mu_t / \mu < 0.2 \\ 0.3 + 0.01 \cdot \dfrac{P}{P_{cr}} Pe_t \text{-} 0.1 \cdot \dfrac{P_{cr}}{P} \cdot \dfrac{C_p}{C_{p,pc}} & 0.2 \leqslant \mu_t / \mu \leqslant 10 \\ 0.85 \text{-} 0.15 \cdot \dfrac{P_{cr}}{P} \cdot (\dfrac{C_p}{C_{p,pc}})^{0.5} & \mu_t / \mu > 10 \end{cases} \tag{1}
$$

1.3 Grid independence and experiment validation

For model A, the mesh number of 9.38 million is selected as the maximum relative error is less than 1%. In the same way, the mesh number for model B is 16.76 million. The distance between the tube wall and the first node (y+) is always less than 1 in every case to improve the prediction accuracy in the near-wall region.

Based on the data obtained from the previous experiments, the accuracy of the numerical simulation method used in this paper was validated considering thermal equilibrium, as shown in Fig. 2. The simulation results effectively reproduced the temperature distribution along the flow direction, including results within the range affected by the entrance effect.

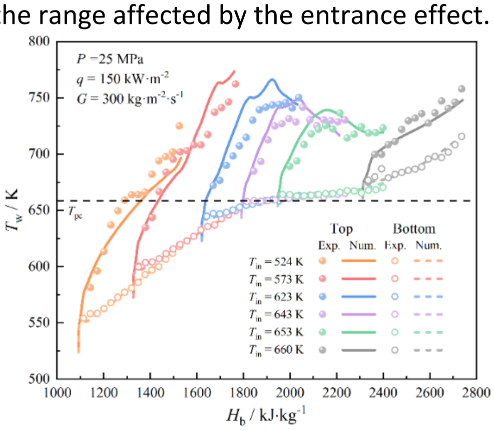
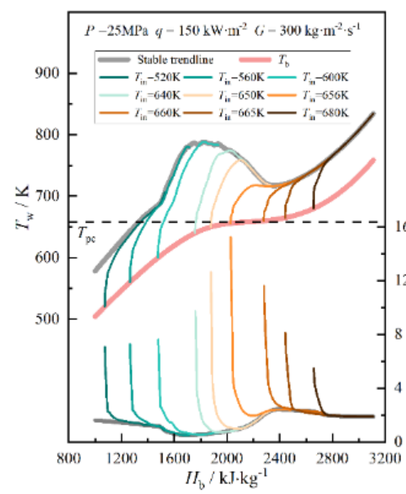


Fig. 2 The results of experiments and SWBT model at heat flux of 150 kW·m^{-2}

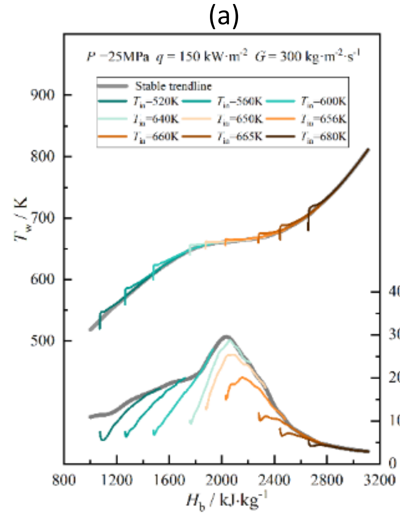
2. RESULTS AND DISCUSSIONS

2.1 Definition of relevant concepts

As mentioned in the introduction, scholars have primarily conducted experiments to investigate the heat transfer characteristics of fully developed supercritical fluids. Nevertheless, the influence of the entrance effect on the flow and heat transfer of supercritical fluids within tubes cannot be overlooked, as it significantly impacts the precise prediction of wall temperatures or heat transfer coefficients. Fig. 3 illustrates the variation in top wall and bottom wall temperatures and HTCs at different inlet temperatures, derived from model A. Similar to the experimental studies by Yu et al. [12] and Lei et al. [13], who analyzed data from the latter parts of the tube to establish a stable temperature trendline, the simulation results also show that after a certain period, the wall temperature and HTDs tend to align with a stable trendline. The state is later referred to as the Axially Asymptotic Developed Stage (AADS). This implies that even as the flow continues to develop axially, the flow and heat transfer are no longer influenced by the history of previous flows. With constant heat input, the fluid at varying mainstream enthalpy levels achieves a nearly distinct state of axial progression. At this juncture, heat transfer can be expressed as a function of bulk parameters, regardless of the axial distance from inlet.



(a)



(b)

Fig. 3 Variation of wall temperature and heat transfer coefficient at different inlet temperatures. (a) Top generatrix. (b)Bottom generatrix

The inlet of the heated section represents a distinct instance where axial heat flux transitions from being absent to present. Similarly, at the outlet of the heating section, the heat flux shifts back from present to absent. To the best of the authors' knowledge, this specific process has not yet been explored. However, as a fundamental type of heat flux variation, studying this process is crucial for enhancing our understanding of how supercritical fluid heat transfer characteristics change under variable axial heat fluxes, ultimately aiding in the improvement of heat exchanger performance. Therefore, based on model B1234, simulations were conducted at an inlet temperature of 620 K, and the heat transfer process was divided into three stages, as depicted in Fig. 4.

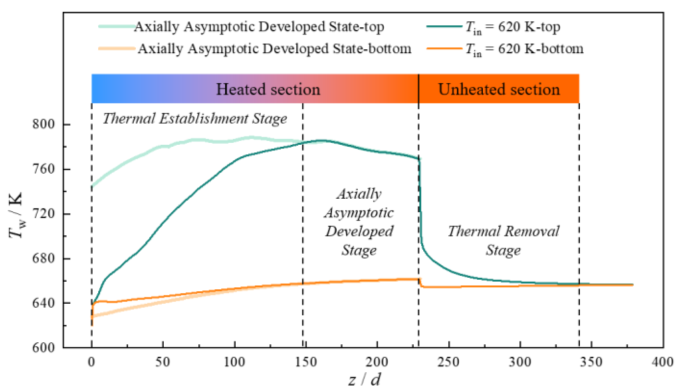


Fig. 4 Different stages of heat transfer in supercritical fluid flow

Drawing on the research findings of Tian et al. [8], the Stage III is identified as the Axially Asymptotic Developed Stage and Stable trendline is called Axially Asymptotic Developed State. Areas that have entered the heated section but have not yet reached this state are referred to as the Thermal Establishment Phase. Once the heat input ceases, the process whereby the wall temperature returns to uniformity is known as the Thermal Removal Phase. The specific definitions of these three stages in Fig. 4 are as follows:

Thermal Establishment Stage: This stage emphasizes the process of introducing heat energy from the environment into the system and gradually reaching a stable thermal state. It describes how heat energy is guided, accumulated, and stabilized within the system, establishing a new thermal equilibrium unaffected by the developmental history.

Thermal Removal Stage: This stage describes the transition of the system from a heated state to one without heat input. This transition may involve the dissipation of thermal energy to the surrounding environment or the deactivation of the heat source.

Thermal Adaptation Stage: This term includes the Thermal Establishment and Removal stages and describes the system's response and adaptation to changes in thermal load. It highlights how the system adjusts itself to reach a new equilibrium state in response to changes in external thermal conditions.

Axially Asymptotic Developed Stage: In this state, the flow and heat transfer processes are almost unaffected by spatial location or developmental history.

2.2 Heat transfer characteristics at different stages

In contrast to fluids with consistent properties, supercritical fluids exhibit an entrance effect that can extend up to 150 times the diameter of the tube. This region is identified as the Thermal Establishment Stage, previously defined. During this stage, there is a tendency for heat transfer deterioration, characterized by lower temperatures at the top wall compared to areas not influenced by entrance effects. To analyze the temperature and velocity distributions in these two states, the cross-section at z/d=50 is examined, as depicted in Fig. 5.

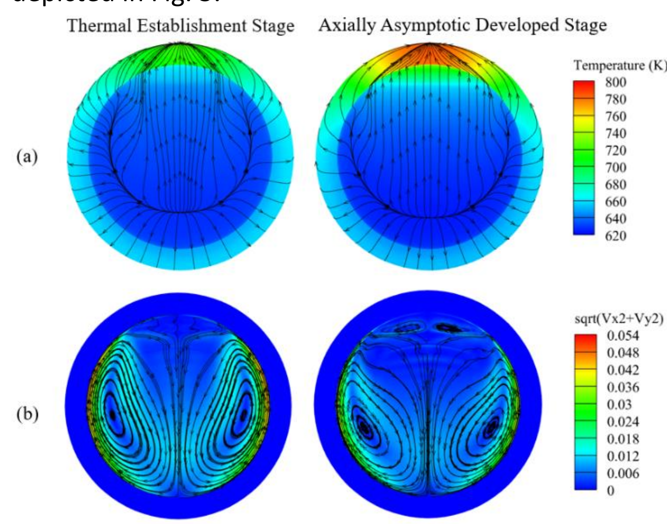
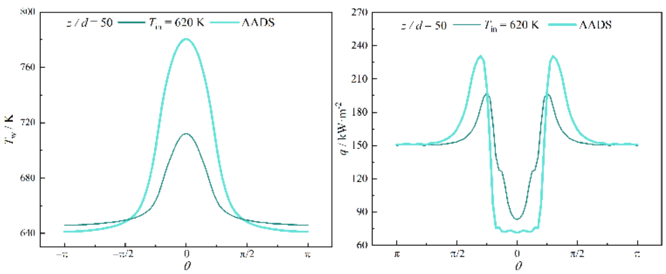


Fig. 5 Temperature and radial velocity distribution of Thermal Establishment Stage and Axially Asymptotic Developed Stage. (a) Temperature contour and temperature gradient. (b) Radial velocity contour and streamlines

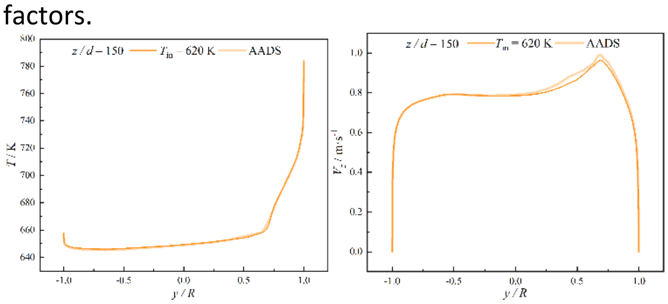
The diagram illustrates that a temperature gradient develops across the entire section from the wall to the fluid core, and vertically from the top wall to the bottom wall, with the lowest temperature near the bottom center of the TUBE. Although the temperature difference between the top and bottom is greater in the AADS, due to its well-developed nature and clear temperature stratification, the secondary flow at this stage is actually weaker than in the corresponding Thermal Establishment Stage. The Thermal Establishment Stage continuously approaches the AADS with the drive of the heat flux. Fig. 6 shows the differences in inner wall temperature and heat flux distribution along the circumferential direction between these two states. Entrance effect enhances heat transfer in the upper half of the tube, but do not benefit the lower half, leading to a more uniform wall temperature distribution. The presence of wall temperature difference induces a redistribution of wall heat flux, with nearly half of the heat near the top wall transferring to the side walls. Compared to the Thermal Establishment Stage, the AADS exhibits a broader range of higher temperatures and lower heat flux at the top wall, indicating less effective heat transfer at this location.



(a) (b)

Fig. 6 Comparison of wall temperature and heat flux between two states of z/d=50 cross section

The characteristics of the AADS have been comprehensively analyzed in the research conducted by Tian et al. [8], as shown in Fig. 7. Despite variations in inlet temperatures, the temperature and velocity distributions along the y-direction at this stage are nearly identical. This consistency demonstrates that after extending over a certain distance, the flow and heat transfer of supercritical fluids can converge to a nearly unique stable state. This finding provides a solid foundation for developing deterministic heat transfer correlations. Additionally, when accounting for the effects of the entrance region or changes in heat flux, existing dimensionless analysis methods for supercritical fluids can be enhanced by incorporating correction factors.



(a) (b)

Fig. 7 Comparison of temperature and velocity in y-direction between the two states

Research on the Thermal Removal Stage primarily investigates the distance required for wall temperature recovery and the dynamics of flow and heat transfer during this recovery process. As depicted in Fig. 4, following the heated section, the top wall temperature rapidly declines and stabilizes after approximately 100 times the z/d distance. Despite the temperature difference between the wall and the core fluid driving significant secondary flow within the tube, as convective heat transfer progresses, the temperature disparity between the top and bottom walls lessens. Eventually, the temperature difference between the solid and the fluid phases fades, establishing a temperature gradient from the bottom wall to the top wall, as illustrated in Fig. 8. Within a 0.5m range post-heating, the top wall temperature drops to a considerably lower level. Previous studies have indicated that solid conduction can notably relieve heat transfer stress at the top. Fig. 9 displays the longitudinal trends of top and bottom wall temperatures during the Thermal Removal Stage, both with and without the presence of solids. At this stage, the impact of solid conduction is minimal, largely due to the rapid decrease in wall temperature difference, underscoring that achieving uniform wall temperatures is predominantly dependent on the convective heat transfer within the tube.

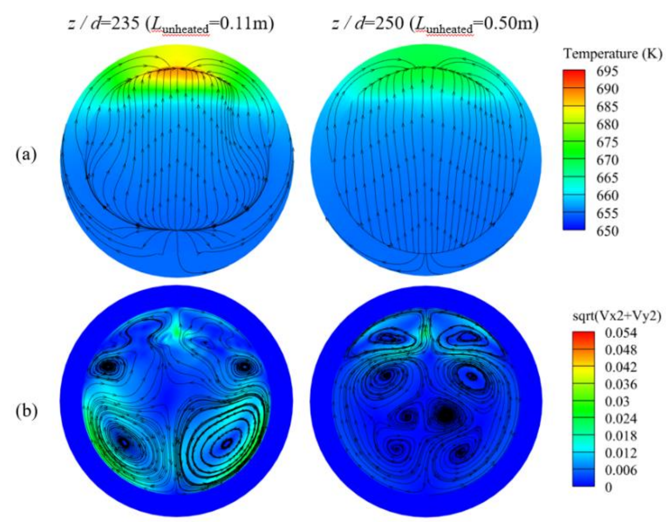


Fig. 8 Temperature and radial velocity distribution at different locations. (a) Temperature contour and temperature gradient. (b) Radial velocity contour and streamlines

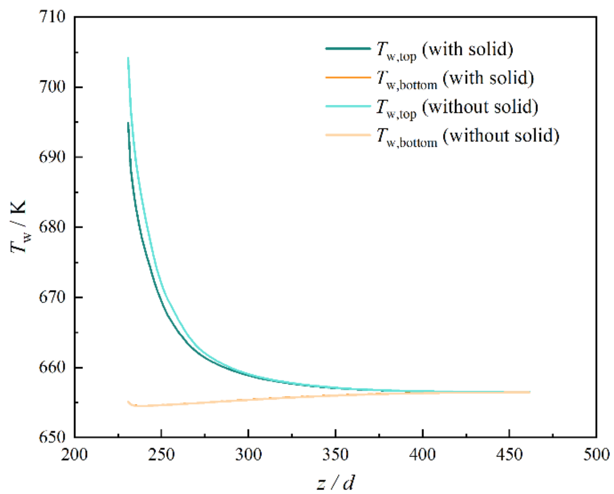


Fig. 9 Comparison of wall temperature with and without solid along the axial direction

2.3 Discussion on Multi-stage Heating Approach for supercritical fluids

Clearly, the entrance effect plays a positive role in improving the performance of supercritical fluid heat exchangers, as evidenced by lower wall temperatures and temperature differences. With the overall heat transfer and heat flux remaining constant, the length or number of heat exchangers can be adjusted to relieve HTD and enhance the overall heat transfer coefficient. Using heat transfer parameters of exchangers unaffected by entrance effects as a reference, numerical simulations were conducted for different heating approaches within corresponding enthalpy ranges, including single-stage heating (model B1234), two-stage heating (model B1245), and four-stage heating (model B1345). The overall heat transfer coefficient of the heat exchanger is calculated based on the average wall temperature, average fluid temperature, and average heat flux of the heated section. δ represents the degree of improvement in the overall heat transfer coefficient under different heating approaches compared to that without considering the entrance effect. The formulas for calculating the average heat transfer coefficient and the relative error are as follows:

$$\mathrm{HTC} = \frac{q}{T_{\mathrm{w,ave}} - T_{\mathrm{b,ave}}} \tag{2}$$

$$\delta = \frac{(\mathrm{HTC}_{\mathrm{model\,B}} - \mathrm{HTC}_{\mathrm{AADS}})}{\mathrm{HTC}_{\mathrm{AADS}}} \times 100\% \tag{3}$$

Fig. 10 displays a comparison of wall temperatures along the length of the heat exchanger for various heating approaches. Utilizing the entrance effect significantly reduces the top wall temperature and the maximum circumferential temperature difference. However, it is crucial to recognize that a multi-stage heating approach can lead to pronounced variations in axial temperature due to shifts in heat flux. Fig. 11 illustrates the comparative effects of the entrance effect on enhancing the overall heat transfer coefficient at different inlet temperatures. The enhancement is more marked when the inlet temperature approaches the pseudo-critical temperature. For instance, with an inlet temperature of 656 K, the improvement effect in model B1234 peaks at 28.15%. At an inlet temperature of 640 K, models B1245 and B1357 show maximum improvement effects of 58.01% and 92.24%, respectively. Considering a heat exchanger with a heating length of 6m, shorter heating lengths tend to show more pronounced improvements. Therefore, the Multi-stage Heating Approach can effectively mitigate heat transfer deterioration and significantly boost the overall heat transfer performance near the pseudo-critical temperature.

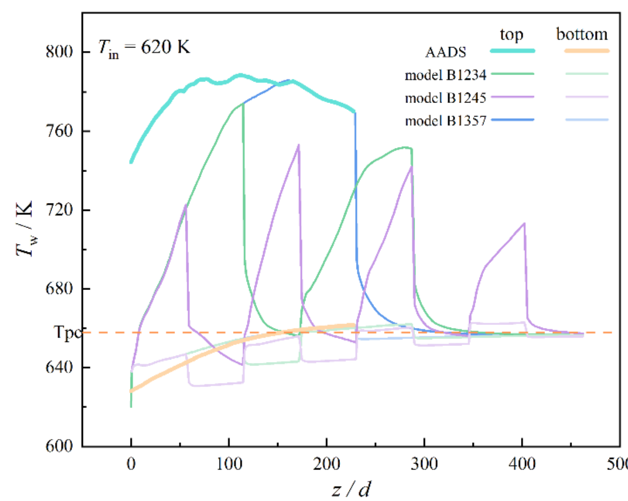


Fig. 10 Comparison of wall temperature with different heated methods at inlet temperature 620 K

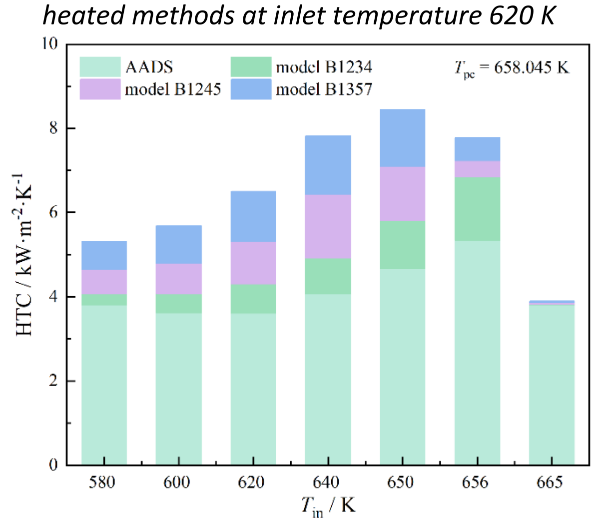


Fig. 11 Comparison of the global heat transfer coefficient at different inlet temperatures

3. CONCLUSIONS

This paper examines the changes in heat transfer characteristics of supercritical water in horizontal tubes both before and after the heated section through numerical simulation. The flow and heat transfer process are categorized into three stages: the Thermal Establishment Stage, the Axially Asymptotic Developed Stage (AADS), and the Thermal Removal Stage. Detailed analyses of each stage are provided, and the effectiveness of the Multi-stage Heating Approach is evaluated. The key findings include:

(1) The area affected by changes in heat flux, termed the Thermal Adaptation Stage, includes the Thermal Establishment Stage influenced by the entrance effect and the Thermal Removal Stage after heating ceases. For supercritical water, the impact of the Thermal Establishment Stage can extend up to 150 z/d, characterized by lower wall temperatures and better heat transfer coefficients compared to the stable state at the same bulk enthalpy. After the heated section, the top wall temperature rapidly decreases, becoming almost identical to the bottom wall after a distance of 100 z/d, mainly due to the convective heat transfer of the supercritical fluid within the tube.

(2) After traversing a certain distance, supercritical fluids can achieve a nearly unique state under a given bulk enthalpy value, known as the AADS. At this stage, the temperature and velocity fields are nearly identical regardless of different inlet temperatures.

(3) The Multi-stage Heating Approach can effectively relieve the HTD and significantly improve the overall heat transfer coefficient of the heat exchanger near the pseudo-critical temperature, while keeping the heating length and heat flux constant. Simulation results show that the overall heat transfer coefficients of one-stage heating, two-stage heating and three-stage heating can be increased by up to 28.15%, 58.01% and 92.24% compared to those without entrance effect.

ACKNOWLEDGEMENT

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