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Effect of Combustion in Furnace on Heat Transfer Process of Supercritical Fluid

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The conjugate heat transfer from combustion in furnace to supercritical fluid in tube has been numerically investigated in this study. The mathematic model of the conjugate process has been constructed. The effects of the combustion status in furnace on the conjugate heat transfer process were analysed. The results demonstrate that the heat transfer deterioration of supercritical fluid in the tube will only occur in high heat flux regions such as the high temperature recirculation zone. The swirling air can suppress the high temperature recirculation zone near the furnace wall, which is beneficial to inhibit the heat transfer deterioration of supercritical fluid in the tube.

1. Introduction

Supercritical units have the advantages of high efficiency and good variable-load performance, which have become important development directions for the power industry (Kurganov et al., 1993). The literature survey was conducted to heat transfer of supercritical fluid flowing in the channels (Pioro et al., 2004). Experimental and numerical studies on CO_2 convective heat transfer at supercritical pressure in vertical microtubes (Jiang et al., 2008). The experimental results show that under the conditions studied, the flow direction, buoyancy and flow acceleration have little effect on the local wall temperature (Pioro et al., 2005). Wang studied the abnormal heat transfer mechanism of supercritical pressure water in the so-called large specific heat zone. It is believed that the heat transfer enhancement of supercritical pressure water in the large specific heat region is the result of a comprehensive effect caused by the rapid change of the thermophysical properties of the supercritical pressure water in the large specific heat region (Wang et al., 2010).

The current research on the heat transfer characteristics of supercritical fluid is mainly simplified by the assumption of constant wall heat flux. Over the past few years, there is a good deal of experimental evidences for both local enhancement and local deterioration in turbulent mixed convective heat transfer of supercritical fluids. The majority of these experiments on supercritical water or CO_2 flowing in uniformly heated vertical tubes can provide the data for wall temperature and heat transfer coefficient. Results from these early experiments showed that turbulent heat transfer of supercritical fluid is very complicated, especially for upward flow.

However, the heat flux on the water-cooled wall in actual boiler varies significantly along the furnace height. This is due to the coupling effect of combustion process in the furnace and the heat transfer process in the water-cooled wall (Yang et al., 2019). Therefore, the uniformly heat flow assumption to study the heat transfer process of supercritical fluid is far from the actual situation. In order to better study the effect of the combustion in the furnace on the coupled process. The mathematical model of this coupled process in the furnace and the tube has been constructed. The influence of the combustion process in the furnace on the heat transfer process of supercritical fluid in the tube is studied. The effects of air supply methods and fuel capacity on the heat transfer characteristics of supercritical fluid are analysed.

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2. Physical model

The general style guidelines are first given, followed by specific cases. First of all, avoid lower-level heading immediately following the higher-level one. It is recommended to have at least one sentence in-between. This research takes a small vertical fuel-oil boiler as the object. The fuel combustion of heat release is in furnace, and heat absorption of supercritical fluid is in the tube. The inside diameter of the furnace is 300 mm, and the inside diameter of the water-cooled tube is 6 mm. The length of the furnace and water-cooled tube is 2.5 m, and the water-cooled wall material is Super304H steel. The operating parameters are shown in Table 1.

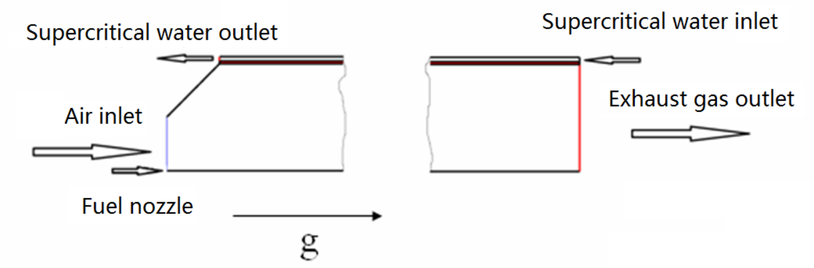

Figure 1: Schematic diagram of vertical fuel-oil boiler

Table 1: Operating parameters of vertical fuel-oil boiler

Air inlet temperature /K	Air axial velocity /m·s^{-1}	Air tangential velocity /rad·s^{-1}	Fuel axial velocity /m·s^{-1}	Fuel radial velocity /m·s^{-1}	Fuel tangential velocity /m·s^{-1}	Fuel temperature /K	Fuel mass flow /g·s^{-1}	Water mass flowrate /kg·m^{-2}·s^{-1}
303	6	0, 6, 120, 240	40	25	25	303	5-10	83-500

3. Methodology

There are processes of evaporation, homogeneous reaction, radiation and convective heat transfer in the furnace, and forced convective heat transfer processes in the tube with supercritical fluid (Sang, 2015). In this study, the coupled process is simplified to a two-dimensional steady-state process (Fan et al., 2001).
Continuity equation:

$$\frac{\partial}{\partial x_i}(\rho u_i) = S_{\mathrm{m}} \tag{1}$$

Momentum equation:

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

Energy equation:

$$\frac{\partial}{\partial x_i}\left(u_i\left(\rho E + p\right)\right) = \frac{\partial}{\partial x_i}\left(k_{\mathrm{eff}}\frac{\partial T}{\partial x_i} - \sum_{j'} h_{j'} J_{j'} + u_j\left(\tau_{ij}\right)_{\mathrm{eff}}\right) + \sum_{j'}\left[\frac{h_{j'}^0}{M_{j'}} + \int_{T_{ref},j'}^{T_{ref}} c_{p,j'} dT\right] R_{j'} + S_{\mathrm{h}} \tag{3}$$

Material transport equation:

$$\frac{\partial}{\partial t}(\rho Y_i) + \nabla\cdot(\rho\vec{v}Y_i) = -\nabla\vec{J}_i + R_i + S_i \tag{4}$$

RNG k−ε models have been recommended by many scholars to obtain good prediction results. The turbulence model will be applied to this study. In an actual furnace, liquid fuel is rapidly evaporated and burned. The combustion reaction rate is controlled by turbulent mixing, so a vortex dissipative combustion model is used to simulate the homogeneous combustion process in the furnace. In addition, the DO model

and the Grey Gas Weighted Average Model (WSGGM) were used to calculate the radiative heat transfer process.
The NIST Standard Reference Database was used for calculating the temperature and pressure dependent properties of CO_2 (Sharabi et al., 2008). The distance between the first calculating node and the wall must be very small ($y^+<0.5$) (Du et al., 2010). The sensitivity of calculation results to mesh was carefully checked and the grid independent solutions are obtained. The QUICK scheme was used for discretization of momentum and energy equations. The PRESTO！ scheme was used for coupling the pressure and the velocity fields (Sharabi et al., 2008). Note that the inside and outside surface temperature of tube was calculated by the conjugate process. The convergence criterion for normalized residual of individual equation was set to be less than 10^{-6}.

4. Results and discussion

In the boiler, combustion conditions in the furnace will affect the heat transfer of supercritical fluid in the tube. Figure 2 shows the effect of different air swirl strength on temperature distribution in the furnace when G=83 kg·m^{-2}·s^{-1} in the tube. It can be seen from Figure 2 that the temperature distribution in the furnace will change significantly with the increase of the intensity of air swirl. As the increase of the air swirl intensity, the high temperature recirculation zone near the furnace entrance gradually disappeared, and combustion occurs mainly in the centre of the furnace. In addition, as the intensity of the air swirl increases, the temperature of the exhaust fume also increases. Because the recirculation zone near the wall disappears, the local heat exchange capacity will decrease accordingly. Disappearance of the recirculation zone and burning in the furnace centre can reduce the risk of heat transfer deterioration in the tube. But they will cause the exhaust fume temperature at the furnace outlet to rise.

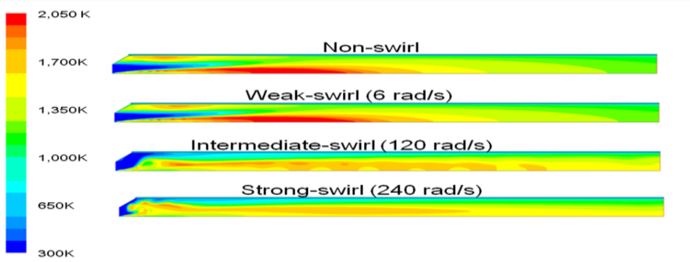

Figure 2: Effect of air swirl strength

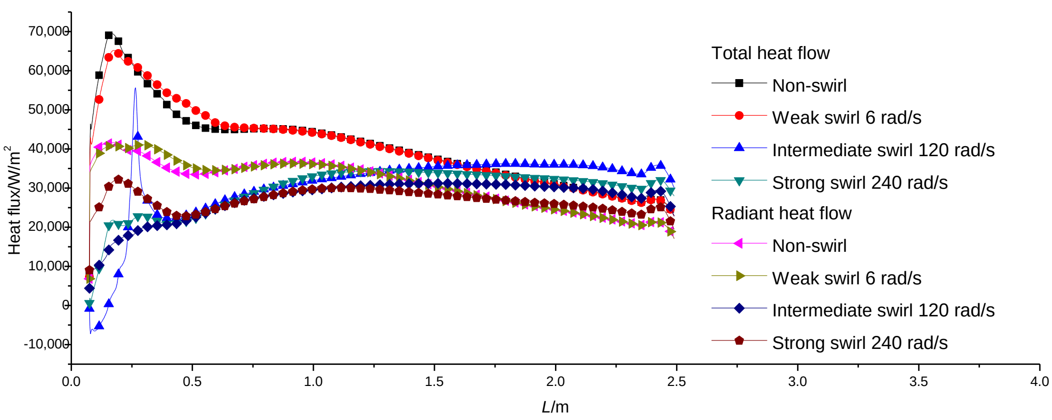

Figure 3: Heat flux distribution under different air supply modes

Figure 3 shows the heat flux distribution at different air swirling strength when G = 83 kg·m^{-2}·s^{-1} in the tube. Due to the swirling effect, the local high heat flux density region at the entrance of the furnace disappears. The increase of the exhaust fume temperature leads to an increase in the heat flux at L > 1.5 m in the second half of the furnace. The local negative heat flux at the furnace inlet is due to the absence of recirculation zone, which causes heat to be transferred from the tube wall to the gas in the furnace. The swirling air supply method reduces the average heat flux on the wall. This is because the high temperature area of the combustion is in the centre of the furnace to reduce the average heat flux. In addition, the swirling air supply reduces the overall temperature in the furnace, resulting in a decrease in the radiant heat flux. The swirling air supply causes the exhaust fume velocity near the wall to be greatly reduced, resulting in a decrease in the convective heat transfer. The disappearance of the local high temperature region leads to the disappearance of the high temperature difference convection heat transfer region.

Figure 4 shows the wall temperature distribution of the tube under different air supply modes when G = 83 $kg \cdot m^{-2} \cdot s^{-1}$ in the tube. In the non-swirl and weak swirl (6 rad/s), a local high wall temperature region appeared near the furnace inlet. This high temperature region coincides with the high heat flux region in Figure 3, and the maximum temperature is close to 700 K. With the increase of the swirl intensity, the high temperature zone of the wall temperature will decrease significantly. When the supply air swirl speed reaches 240 rad/s, the supply air has reached a strong swirl state. The wall temperature distribution is relatively uniform, there is no obvious local rise, and the maximum temperature drops by 25 K. In addition, it can be seen that the wall temperature is slightly higher under the condition of strong swirling flow in the second half of furnace, but the temperature difference does not exceed 5 K.

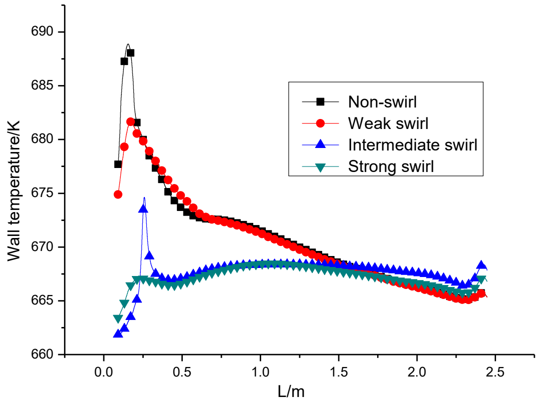

Figure 4: Wall temperature distribution on the tube

Figure 5 shows the temperature distribution of the fuel capacity 10 g/s when G = 500 $kg \cdot m^{-2} \cdot s^{-1}$ in the tube. When there is non-swirl or intermediate-swirl, the combustion in the furnace will appear unstable, and the main combustion zone appears near the wall and in the second half of the furnace. Because a large range of recirculation combustion zone appear near the wall surface, the exhaust fume temperature is too high. Due to limited furnace space, fuel combustion is insufficient. With the increase of the swirl intensity, the temperature field in the furnace will change significantly. The local recirculation high temperature zone at the entrance disappeared, so that the combustion occurred stably in the central area. The range of the main combustion region is significantly increased, and the combustion efficiency of the fuel is significantly improved.

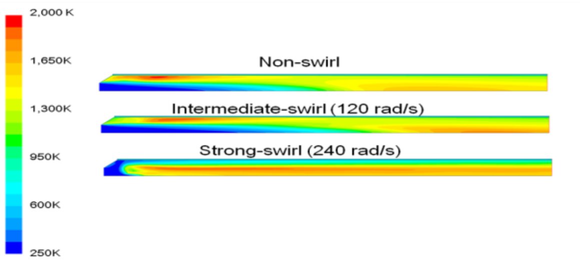

Figure 5: Temperature field in the furnace under the fuel capacity 10 g/s

Figure 6 shows the distribution of heat flux in the furnace of the fuel capacity 10 g/s when G = 500 $kg \cdot m^{-2} \cdot s^{-1}$ in the tube. When there is non-swirl, the maximum heat flux density increased from 70 kW/m^2 to 90 kW/m^2 relative to the 5 g/s fuel. With the increase of the swirl, the heat load distribution in the furnace tends to be uniform, the maximum heat flux decreased by 50 %. This phenomenon is similar to the conclusion obtained in the previous section. Relative to the working condition of 5 g fuel, its average heat flux has increased, but the maximum increase has only increased from 41 kW/m^2 to 55 kW/m^2. This shows that when the furnace is unchanged, increasing in fuel capacity does not significantly increase the heat transfer of the furnace.

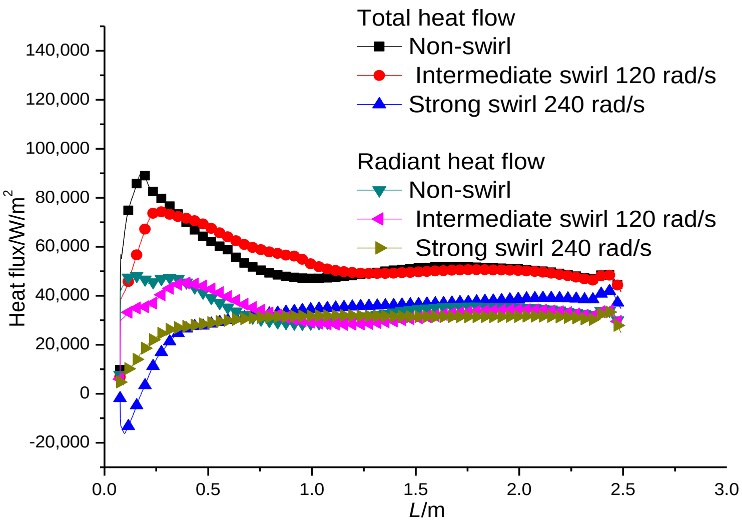

Figure 6: Distribution of heat flux in the furnace under 10 g/s fuel

Figure 7 shows the wall temperature and heat transfer coefficient distribution of the inner wall surface of the tube under different air supply modes. The wall temperature under the strong swirling conditions is much lower than the other two conditions. This temperature difference is caused by the different local heat flux distributions under different air supply modes. But the difference in maximum temperature does not exceed 10 K. Because a large flow of G = 500 kg·m^{-2}·s^{-1} is used, the phenomenon of enhanced heat transfer will occur under larger heat flux. There is no local high heat flux region under strong swirl, which leads to heat transfer enhancement to disappear under strong swirl conditions.

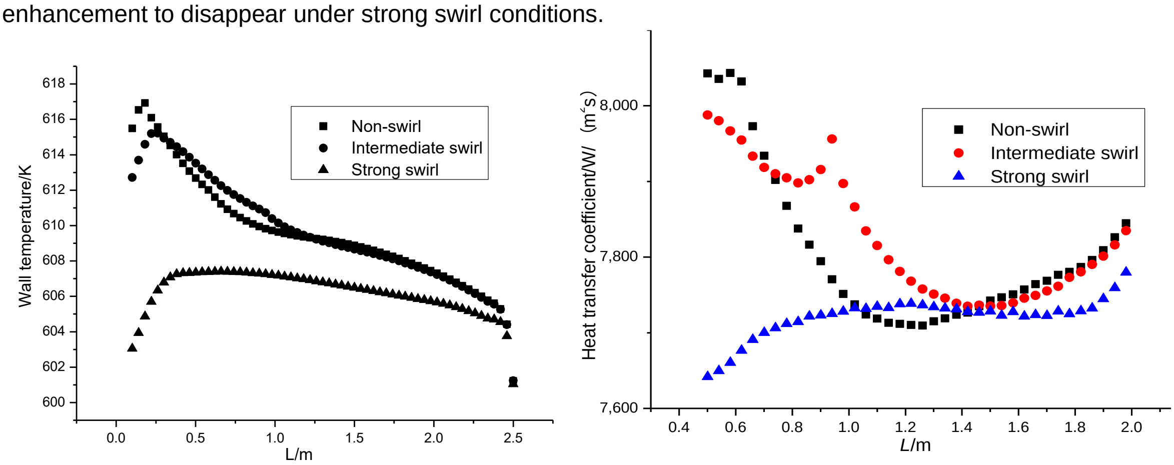

Figure 7: Temperature and heat transfer coefficient distribution

5. Conclusions

Numerical simulations have been performed to investigate on coupling process of supercritical heat transfer and boiler combustion. A coupled calculation model for combustion in the furnace and heat transfer in the tube is established. The deterioration of supercritical fluid heat transfer in the tube will only occur in local high heat flux region such as high temperature recirculation zone. The swirling air method can suppress the high temperature recirculation zone near the wall surface, which is beneficial to prevent local heat transfer deterioration on the wall surface. When the fuel quantity is increased to increase the boiler load, it will cause the flame to directly impact the water-cooled wall if the air swirl strength is not increased synchronously.

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