Experimental Study on the Hydrodynamics and Condensation Heat Transfer Characteristics of Wet Flue Gas in a Horizontal Smooth Tube

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Keywords: Flue gas, wet air, condensation, horizontal tube.

Abstract: The flow and heat transfer characteristics of wet air inside a horizontal smooth tube was studied experimentally. A horizontal annular heat exchanger was used in the experimental, with the wet air Reynolds number ranging from 5000 to 13,000 and inlet specific humidity ranging from 0.05 to 0.2 kg/kg. The effects of the surface subcooling, the wet air flow rate and the inlet specific humidity on the condensation heat transfer coefficient (HTC) were investigated. The results show that the condensation heat transfer coefficient (HTC) decreases with the increasing of surface subcooling, while increases with the increasing of the wet air flowrate and inlet specific humidity. A correlation of wet air heat transfer coefficient along horizontal smooth tube had been developed. The correlation showed a good agreement with experimental results.

1. Introduction

Since the combustion products do not contain harmful gases, natural gas is gradually replacing coal for the sake of environmental protection. In general gas boiler design, the outlet flue gas temperature of conventional boiler is usually higher than 150°C. At such temperatures, the water vapor entrained in the flue gases does not condense, and the latent heat cannot be reclaimed, which leads to a considerable heat loss. How to reduce the heat loss of flue gas and improve the utilization efficiency of gas-fired boiler is a hot issue. Zhu et al. [1]used direct contact heat exchanger combined with closed absorption heat pump to recover flue gas waste heat. The results show that the thermal efficiency of gas-fired boiler can be increased by 13.6%. Lee et al.[2] proposed a total heat exchange system to recover heat from flue gas by indirect contact heat exchange, and humidify the combustion air by spraying water on the combustion air side. In above studies, for improving the thermal efficiency, the recovery of latent heat can not be ignored. Defu Che [3] and others considered the influence of noncondensable gas on heat and mass transfer process, analyzed the condensation heat transfer process of vapor in flue gas by Colburn Hougen[4] method, and obtained the normalized equation of conventional condensation heat transfer coefficient of flue gas. While Sulin Wang [5] and others analyzed the heat and mass transfer process of flue gas during condensation in finned heat exchanger tubes, and optimized the design of condensation heat exchanger by using the research results, so as to improve its thermal performance.

The research on sensible and laten heat utilization of flue gas mainly adopts condensation method, that is, the heat of flue gas is transferred to low temperature medium through heat exchanger[6,7]. For natural gas, the combustion products do not contain sulfur dioxide, which provides conditions for the application of condensation method. Experiments on the flue gas condensation in multi-row staggered tube bundle heat exchangers were carried out by Husnain[8], the empirical formula of the heat transfer coefficient of multi-row staggered tube bundle heat exchangers was determined. In Mohammadi[9] study, the heat recovery of the exhaust gas of a condensing boiler is analyzed by utilizing a shell and corrugated tube heat exchanger as condensing boiler. Zhang et al[10]designed and optimized a heat exchanger to recover heat from chimney flue gas using hybrid strengthening technique, and verified the results with experimental data.

Essentially, the flue gas condensation is a kind phenomenon of vapor condensation in the presence of noncondensable gas(NCG).In 1929, Othmer[11]first explored the phenomenon of steam condensation with a small amount of air through experiments. The results showed that even with a small amount of air, the heat transfer efficiency would be greatly reduced, and the heat transfer coefficient obtained was significantly less than the value predicted by Nusselt theory. After that, many scholars had conducted in-depth research on this phenomenon. W. C. Lee and J. W. rose[12,13] studied the condensation heat transfer characteristics of pure steam and mixture containing noncondensable gases, and put forward the correlation for condensation with noncondensable gas. Siddique[14]carried out an experimental study on the condensation of noncondensable gas vapor in a vertical tube, and fitted a wide range of experimental correlations according to the experimental data. Colburn and Hougen [4] proposed a heat and mass transfer analogy method to study the film condensation process with noncondensable gas. Gianfranco Caruso[15,16,17]had studied the condensation of noncondensable gases in horizontal and inclined tubes.

Although there have been many researches on the condensation flow and heat transfer of steam with NCG. These studies are mainly focused on condensation with relative less NCG. The research on steam condensation with high concentration of NCG such as the flue gas is not enough. In this paper, experiments were carried out to study the heat transfer characteristics of the flue gas in the smooth horizontal tubes.

The main composition of wet flue gas is similar to that of wet air, and their condensation heat transfer mechanism are basically the same. Both belong to vapor condensation phenomenon containing a large amount of NCG. Considering the specific humidity of the actual flue gas and the convenience of the experiment, wet air with inlet specific humidity in the range of 0.05-0.2kg/kg is taken as the research object for simulating flue gas. For the smooth tube, the effects of inlet specific humidity, the wall subcooling and the flow rate of wet air on the overall heat transfer performance were investigated.

2. Experimental Details

2.1. Experimental Apparatus

P: Pressure transmitter CWB: Constant temperature water bath
T: Pt100 thermal resistance V: Value
D: Humidity/Temperature probe

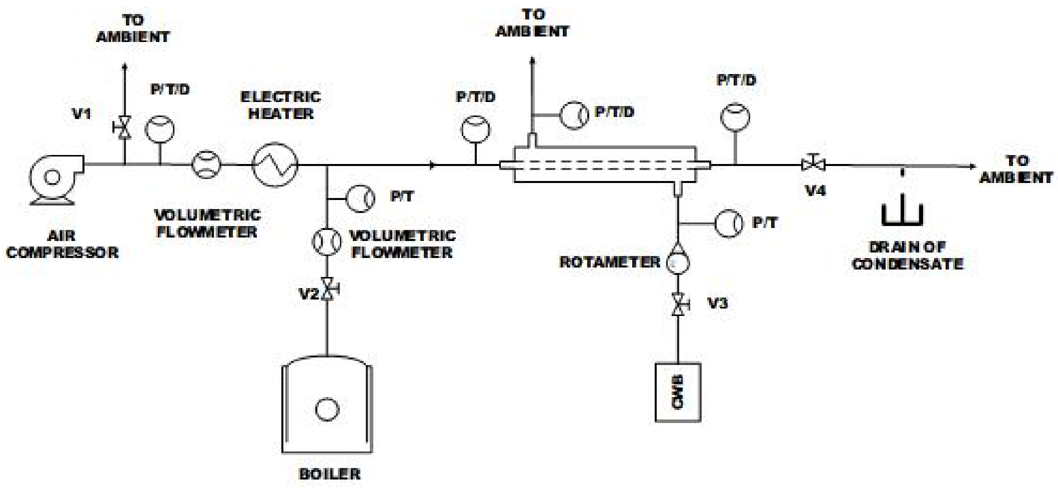

Figure 1: Schematic diagram of the experimental system.

Table 1: Uncertainties in the measured parameters.

Parameter	Uncertainty
Temperature	±0.1℃
Relative humidity	±0.8%
Cooling water mass flow rate	±0.5%(0~80m^3/h)
Wet air volume flow rate	±1%（3~60m^3/h）
Pressure drop	±0.3%(-200Pa~125kPa)

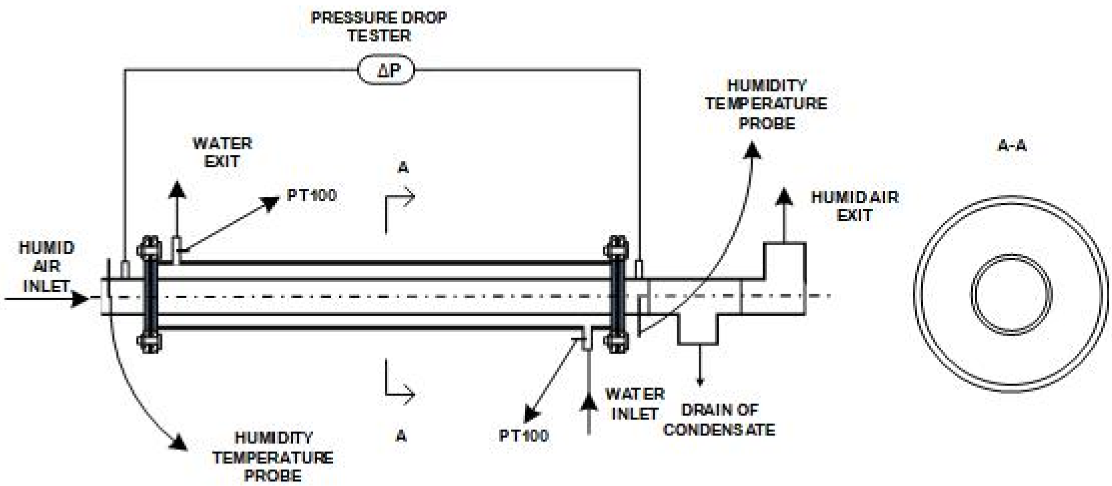

Figure 2: Details of the experimental testing section.

Figure. 1 shows the schematic diagram of the experimental system, which is composed of gas supply system, water supply system, experimental testing section, data-acquisition system and related instruments. The experiment was carried out in a horizontal annular tube, with wet air flowing in the inner tube and cold water flowing in the annular space. The wet air generated by the gas supply system and the cooling water generated by the water supply system exchanged heat in the experimental testing section. The heat released by the condensation of wet air was absorbed by cooling water. After heat exchange, the cooling water and wet air were discharged into the environment.

The details of the experimental testing section is shown in Figure 2. The mass flowrate of the cooling water was measured by electromagnetic flowmeter. The volume flowrate of the wet air was measured by vortex flowmeter. The temperature of the wet air in the test section was measured by Pt100 thermal resistance, and the relative humidity of the wet air were measured by the humidity probe. The uncertainties of the measurement devices are shown in Table 1.

For wet air experiments, the volume flux was varied in the range of 12.5m3/h-30m3/h. And to explore the influence of wall undercooling, the temperature of cooling water was controlled at 19.5 ℃ and 9.5 ℃. The structural parameters of the experimental testing section are shown in Table 2.

Table 2: The structural parameters of the experimental testing section.

Parameter	Value
The outer diameter of the inner tube(do)	38mm
The wall thickness of the inner tube(e)	2.5mm
The outer diameter of the outer tube(Do)	51mm
The wall thickness of the outer tube(e)	3mm
The heat conductivity coefficient of tube (kt)	16.3W/(m2·K)

2.2. Data Reduction

To study the condensation heat transfer characteristics of the wet air, it is necessary to determine the internal heat transfer coefficient of wet air. The heat coefficient was calculated as follows:

$$\frac{1}{K}=\frac{d_o}{h_m d_i}+\frac{d_o \ln\left(d_o / d_i\right)}{2k_t}+\frac{1}{h_c} \tag{1}$$

Where hm is the wet air condensation heat transfer coefficient (HTC), K is the overall coefficient of heat transfer, hc is the cooling water side surface heat transfer coefficient which was determined by the Gnielinski [18]equation:

$$h_c=\frac{(f_c/8)(Re_c-1000)Pr_c}{1+12.7\sqrt{f_c/8}(Pr_c^{2/3}-1)}\left[1+(\frac{d}{l})^{2/3}\right]\frac{k_c}{d} \tag{2}$$

$$f_c=(1.82\lg Re_c-1.64)^{-2} \tag{3}$$

The logarithm mean temperature difference (LMTD) method was used to calculate the total heat transfer coefficient K:

$$K=\frac{Q}{S_o\cdot\Delta T_{ln}} \tag{4}$$

$$Q=W_c Cp_c(T_{c,in}-T_{c,ex}) \tag{5}$$

$$\Delta T_{In}=\frac{\left(T_{m,ex}-T_{c,in}\right)-\left(T_{m,in}-T_{c,ex}\right)}{\ln\left(\frac{T_{m,ex}-T_{c,in}}{T_{m,in}-T_{c,ex}}\right)} \tag{6}$$

In the above three equations, Q is the heat flux and ΔTln is the logarithmic mean temperature difference. $c_{p,c}$ is the water specific heat, W_c is the water mass flow rate, $T_{c,in}/T_{c,ex}$ are the inlet and outlet temperatures of the cooling water, respectively. So is the heat transfer area based on the external surface of the inner tube. $T_{m,in}/T_{m,ex}$ are the inlet and outlet temperatures of the wet air. According to Eq(1)(2)and(4) , hm can be calculated.

The fan friction coefficient obtained from the measured pressure drop was used to evaluate the flow resistance characteristics.

$$f_m = \frac{\Delta p d_c \rho_m}{2G_m^2 l} \tag{7}$$

3. Result and Discussions

3.1. Character of Flow Resistance

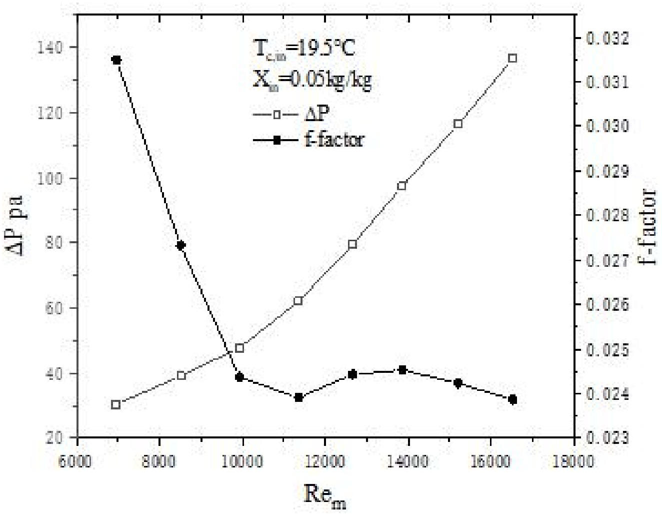

Figure 3: Variation of pressure drop and f-factor with reynolds number.

Figure 3. shows the variation of pressure drop and flow resistance coefficient with Reynolds number when the inlet moisture content is 0.05kg/kg and the cooling water temperature is 19.5 °C. The pressure drop gradually increases with the increase of Reynolds number. Different from the pressure drop, when the Reynolds number is in the range of 8000 to 10000, the decrease is faster, and with the flow rate increasing again, the decrease of F-factor was not obvious.

3.2. Character of Heat Transfer

In order to explore the condensation heat transfer characteristics, the effects of the surface subcooling, the wet air flow rate and the inlet specific humidity on the condensation heat transfer coefficient (HTC) were investigated. And a new correlation is proposed based on the experimental results.

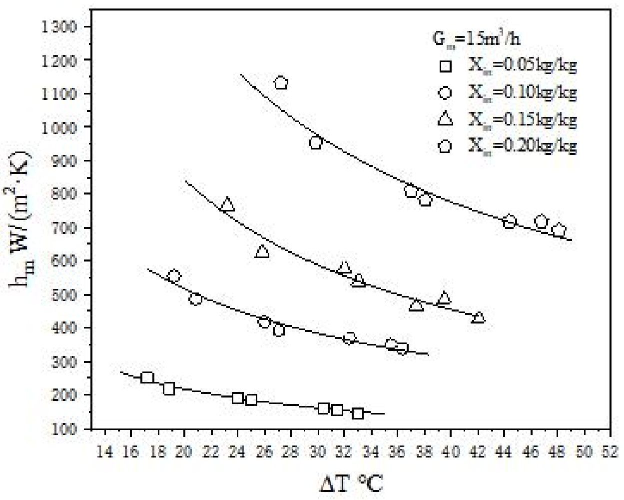

Figure 4: Variation of condensation heat transfer coefficient with wall subcooling.

Figure 4. shows that the variations of the condensation heat transfer coefficient (HTC) plotted as a function of surface subcooling(ΔT) under different inlet specific humidity. As can be seen that, for the same inlet specific humidity, the HTC decreased as ΔT increases. In the case of the inlet specific humidity is 0.05kg/kg, HTC decreased from 249 W/m2·K to 144 W/m2·K with ΔT from 17℃ to 33℃. And with the increase of inlet specific humidity, the change of surface heat transfer coefficient with wall undercooling is more obvious. The results show that there is a negative correlation between the wall subcooling and the heat transfer coefficient.

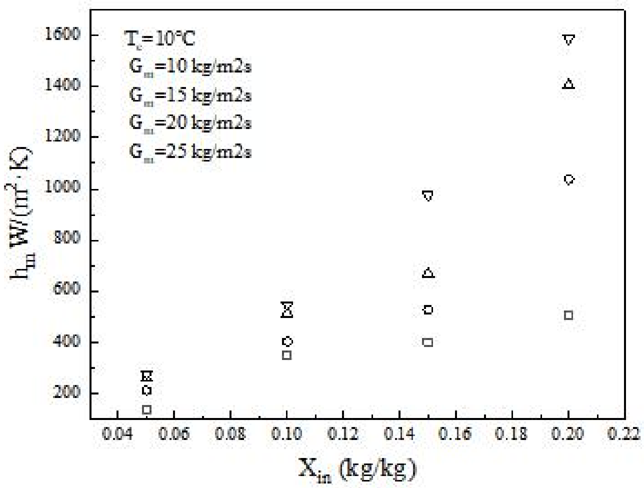

Figure 5: Variation of condensation heat transfer coefficient with the inlet specific humidity.

Figure 5. shows the effect of inlet specific humidity on heat transfer coefficient at different flow rates. The changing trend is different from that of wall undercooling. It can be observed that the HTC enhances with the rise of inlet specific humidity under the same conditions. This effect is gradually intense with the increase of flow velocity inside the tube. The main reason is that the higher the inlet specific humidity, there will be more vapor condensation, HTC will also increase. And for high inlet specific humidity, the effect of flow rate on the condensation heat transfer coefficient is greater.

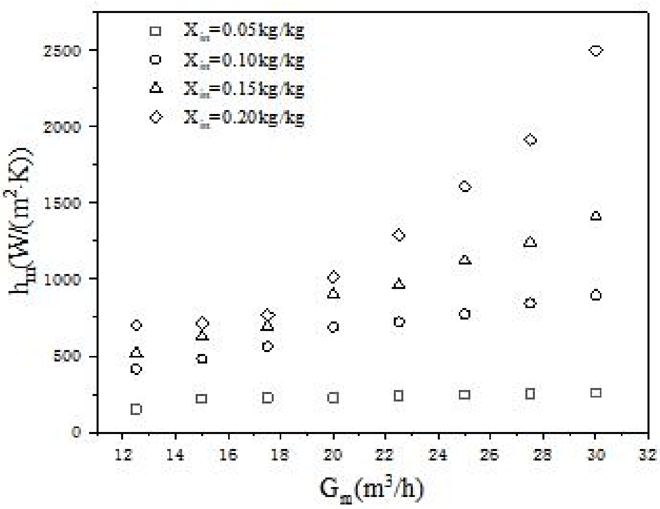

Figure 6: Variation of condensation heat transfer coefficient with the volume flowrate of wet air.

Figure 6. shows that the effect of the volume flowrate of wet air on the condensation heat transfer coefficient at different inlet specific humidity. When the inlet specific humidity was 0.20kg/kg, HTC increased from 700 W/(m2·K) to 2494W/(m2·K) with Gm from 12.5 m3/h to 30 m3/h. And when the inlet specific humidity was 0.05kg/kg, HTC slightly increased from 151 W/(m2·K) to 254 W/(m2·K) with Gm from 12.5 m3/h to 30 m3/h. That means the increase of flow velocity can enhance the heat transfer coefficient.

Through the above analysis, it is proved that the surface subcooling, mixture velocity, the inlet specific humidity are the important factors that influence the condensing and heat transfer capacity of the wet air inside horizontal tube. All above parameters should be considered in the mixture condensation heat transfer coefficient. The form of correlation proposed by Hasanein[19]and Siddique[20] is adopted. Based on the above analysis and experimental data, a multiple regression approach was performed to correlate the mixture condensation heat transfer coefficient in the following equation:

$$Nu_m = 0.2325\, Re_m^{1.102} X_{in}^{1.187} Ja^{-0.0617} \tag{8}$$

$$Re_m = \frac{\rho_m v_m d_i}{\mu_m} \tag{9}$$

$$Ja = \frac{Cp_m (T_m - T_w)}{r_m} \tag{10}$$

Where Rem, Xin and Ja represent the influence of the wet air velocity, the inlet specific humidity of wet air and wall undercooling respectively, rm is the latent heat of condensation. The application scope of the above correlation is:

$$4763 < Re_m < 16524; \qquad 0.05kg/kg < X_{in} < 0.20kg/kg; \qquad 0.0009 < Ja < 0.045$$

In Figure 7, a comparison between the wet air condensation heat transfer Nusselt number obtained by the above correlation and the data obtained from experiments is shown in Figure. 4. For all the experimental points, 84% of the calculation results were plotted within the error range of 25%, indicating that the correlation in this study can well describe the heat transfer process.

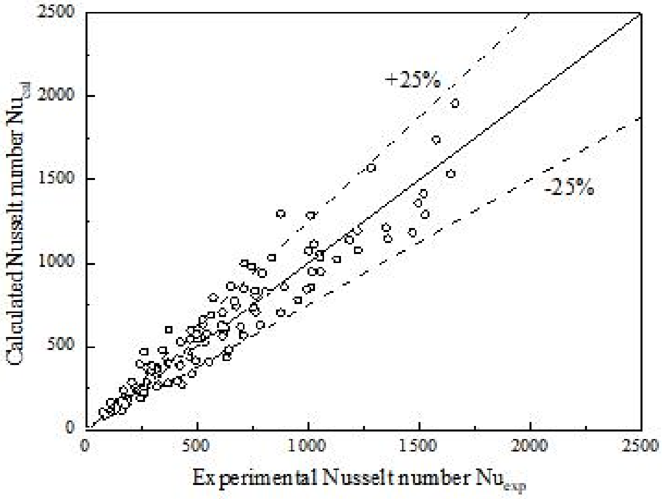

Figure 7: Comparison of experimental and predicted Nusselt numbers.

4. Conclusions

In the present study, an experimental investigation on the heat transfer characteristics of the wet air flow condensation inside the horizontal smooth tube. The major conclusions can be summarized as:

In the process of wet air flow condensation, flow loss is inevitable due to the existence of friction. The drop of pressure gradually increased with the increase of Reynolds number. The f-factor decreased rapidly at a lower flow rate and decreased a little at stable turbulent stage.

The noncondensable gas has a significant effect on the condensation heat transfer coefficient. The experiment results show that the heat transfer coefficient increases with the increase of the inlet specific humidity and flow rate of wet air. And HTC decrease with the increase of the subcooling of the wall.

Considering the influence of the subcooling of the wall, the inlet specific humidity and the flow rate of the wet air on HTC, a correlation of wet air heat transfer coefficient along the horizontal smooth tube has been developed, in good agreement with experimental results.

Acknowledgments

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