STUDY ON HEAT TRANSFER CHARACTERISTICS OF FLUE GAS CONDENSATION IN NARROW GAP HEAT EXCHANGERS

by

Yungang WANG[a*], Jian JIAO[a], Yanjun DAI[a], Zhiwu KE[b], Qinxin ZHAO[a*], and Feixiang LI[c]

[a] Key Laboratory of Thermo-Fluid Science and Engineering, Ministry of Education, Xi'an Jiaotong University, Xi'an, Shaanxi, China
[b] Wuhan Second Ship Design and Research Institute, Wuhan, China
[c] Hubei Special Equipment Inspection and Testing Institute, Wuhan, Hubei, China

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Flue gas after wet desulfurization contains a large quantity of water vapor, as well as a small amount of fine particulate matter, acid and other pollutants. Direct emissions will cause haze and environmental pollution. Flue gas condenser can effectively recover latent heat of vaporization and remove these pollutants. However, the traditional flue gas condenser has disadvantages such as large volume, low heat exchange efficiency and easy blockage. To overcome these problems, a new type narrow gap heat exchanger, which is easy to detach and clean, is proposed. Then, the flue gas condensation characteristic is experimentally investigated. The factors including velocity and temperature of flue gas, flow and temperature of cooling water are studied on heat transfer coefficient and the condensation rate. The results indicate that the overall heat transfer coefficient is seven times of convection heat transfer coefficient, and the condensation rate can reach more than 60%. As the cooling water temperature increases, the condensation rate and condensation heat transfer coefficient gradually decrease. With the increasing of the flue gas velocity, condensation rate gradually decreases and condensation heat transfer coefficient gradually increases. Finally, the dimensionless correlation of Nusselt number is carried out, and the error is within 10%.

Key words: *condensation rate, flue gas, heat transfer, latent heat, narrow gap heat exchanger*

Introduction

At present, most thermal power plants are equipped with wet desulfurization technology [1]. The flue gas after wet desulfurization contains water vapor, SO_x, acidic droplets and extremely fine particles, such as silicate and $CaSO_4$. These pollutants can induce haze and cause serious air pollution. To solve this problem, deep condensation decontamination of flue gas has been applied gradually [2]. The core is to reduce the flue gas temperature to the dew point temperature of water vapor in the flue gas and below, to recover the sensible heat and the latent heat, thereby improving thermal efficiency. The water in flue gas can be recovered by water vapor condensation. At the same time, some pollutants such as

* Corresponding author, e-mail: ygwang1986@xjtu.edu.cn, zhaoqx@xjtu.edu.cn

NO_x and SO_2 in the flue gas will also be dissolved in the condensate, and then it can be effectively removed. This method can not only reduce the emission of pollutants and eliminate the *white smoke plume*, but also recover the water to achieve energy saving and consumption reduction. Therefore, efficient flue gas condensation heat exchanger has become the key to the application of this technology. Nowadays, most of the commonly used flue gas condensers are tubular heat exchangers. This equipment has the characteristics of large volume, large terminal difference, and small flue gas cooling range. It is difficult to realize the deep condensation of flue gas in the limited space of the boiler tail. Compared with tubular heat exchangers, compact heat exchangers have a larger heat transfer area per unit volume. So compact heat exchangers can significantly reduce the heat transfer terminal difference and achieve a larger reduction of flue gas temperature in a limited space. However, the structure of traditional compact heat exchangers is complex and limited by the existing welding technology. It is easy to plug and low temperature corrosion and other problems, can only be used in clean flue conditions. Nevertheless, with the development of ultra-low emission modification of boiler, advanced materials, advanced welding technology and superhydrophobic process, it has created powerful conditions for the application of plate fin heat exchangers in the condensation of relatively clean flue gas. Under the aforementioned background, a new type of narrow gap flue gas condenser is presented and investigated in this paper. The flue gas side structure is formed by the split-fin structure produced by extrusion aluminum process. The production and manufacture of this new compact heat exchanger is relatively simple.

Heat transfer in condensing process has been studied for a long time. Nusselt [3] analyzed the condensation process of pure vapor laminar flow film on vertical plates as early as 1916, and put forward the analytical solution. It is believed that the major thermal resistance comes from the liquid film formed on the wall surface. Then, Colburn and Hougen [4, 5] put forward the theory of gas-liquid interfacial mass diffusion. It is still widely used up to now. They believed that the gas film was consist of the condensate film and non-condensable gas. Osakabe *et al.* [6-9] carried out mathematical analysis on the energy and mass balance in the process of real flue gas condensation heaters, and studied the influence of flue gas flow and components on condensation heat transfer by experiment. Jeong *et al.* [10-12] modified the Colburn-Hougen model and proposed a new analytical method. Lee *et al.* [13] modified the turbulent diffusion layer model considering non-condensable gas, and obtained the turbulent diffusion coefficient of natural convection. Vyskocil *et al.* [14] established a condensing model based on component transport model, which is appropriate for compressible and incompressible fluid. Li *et al.* [15] compared the boiler efficiency before and after flue gas drying biomass. Wang *et al.* [16] experimentally studied on several factors based on the staggered bundle heater, including steam volume fraction, gas velocity, cooling water flow and temperature. Wang *et al.* [17] experimentally discussed condensation heat-transfer performance of symmetrical inner finned tubes in real flue gas. Shi *et al.* [18] studied a compact heater with finned tubes for recovering exhaust heat. Liu *et al.* [19] studied the influence of wall roughness on sensible heat transfer and flow resistance in wet air condensation. Zhang *et al.* [20] investigated the wettability and condensation heat-transfer performance of flue gas containing steam outside finned tubes with different anticorrosive coatings. Rifert *et al.* [21-22] reviewed the work on condensation in smooth horizontal tubes published between 1955 and 2013. Meanwhile, many studies [23-27] have shown that small amounts of non-condensing gases will seriously reduce heat transfer compared to pure steam condensation. Besides, Wang [28] studied the condensation heat transfer characteristics and corrosion resistance of different coated tube surfaces. Among them, the surface of Ni-P-Cu and PTFE coated tube has the best condensation heat transfer characteristics and corrosion resistance. Therefore, it can be well used in the recovery of waste heat from low-temperature flue gas.

Although there have been many studies on condensation processes containing non--condensable gases, few works have been performed on deep condensation characteristics in unclean flue gas condition. Especially for compact heat exchangers, such as narrow gap heat exchanger (NGHE), few studies on the characteristics of condensation heat transfer in actual flue gas are involved. Therefore, it is necessary to study the performance of the NGHE in this paper for engineering application.

Testing system and operating conditions

Testing system

The testing system is composed of combustion system, condensation heat transfer system and measurement system, as shown in fig. 1. To reduce heat loss, insulation is added around the combustion system, condensation heat transfer system and flue pipes.

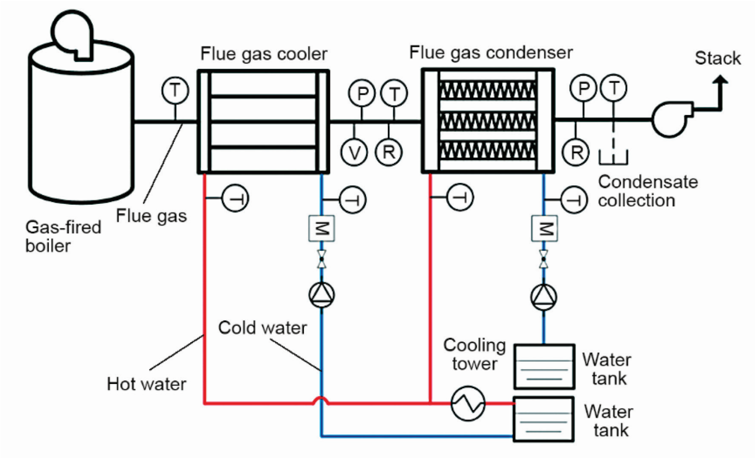

Figure 1. Testing system diagram

The combustion system contains a 350 kW gas-fired boiler and natural gas pipelines. The outlet temperature of flue gas is 150~250 °C, and the load of gas-fired boiler can be adjusted by fan speed of burner, to adjust flue gas velocity in narrow gap. The dominant components of natural gas in the experiment are exhibited in tab. 1. As we can see that the main component of natural gas is methane, and its volume fraction is as high as 93.18%. In order to ensure natural gas burnout, the excess air coefficient is maintained at 1.3 during the test, so the flue gas compositions are stable. Then, it can be calculated according to natural gas contents and excess air coefficient.

Table 1. Volume fraction of each component

Component	CH_4	C_2H_6	C_3H_8	C_4H_{10}	N_2	CO_2	O_2
Volume fraction	93.18%	1.5%	0.22%	0.06%	2.24%	2.16%	0.64%

The condensing heat transfer system consists of two heat exchangers including flue gas cooler and flue gas condenser. To explore the characteristics of low temperature flue gas condensation heat transfer, flue gas first enters the flue gas cooler to be cooled to the specified temperature, and then enters the flue gas condenser for further cooling. The flue gas condenser is arranged vertically, and the condensate is collected by the device.

The flue gas cooler is a tubular heater. The external diameter of pipe is 20 mm, and pipe thickness is 3 mm. The flue gas condenser is the NGHE as shown in fig. 2. The narrow gap channel structure and the heat exchange unit are exhibited in figs. 2(a) and 2(b), respectively. The flue gas channel is inserted by two plate-fin structures to form a narrow gap channel, while the cooling water channel is arranged in a serpentine shape, entering from the bottom, and flowing out from the top. The material used for the unit body is 6063-T6. Its main alloy elements are magnesium and aluminum, and it has excellent corrosion resistance, excellent processing performance, high toughness and high strength characteristics. The overall structure of NGHE is formed by two flue gas channels, three water side channels, heat exchanger shell and water header, as shown in fig. 2(c). After optimization analysis, and considering the economic factors, a common size of the mold is selected for heat exchanger processing. The main structural dimensions of flue gas condenser are displayed in tab. 2.

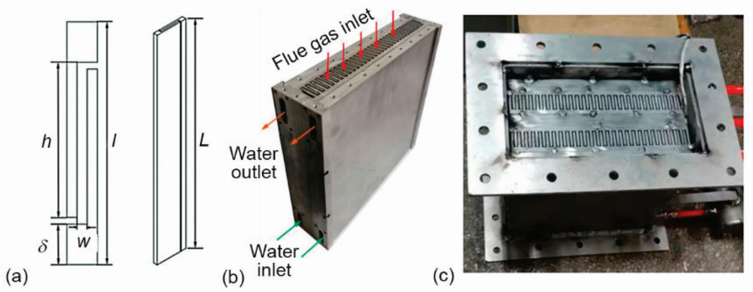

Figure 2. Diagram of NGHE and the narrow gap structure; (a) narrow gap channel, (b) heat exchange unit of NGHE, and (c) flue gas condenser

Table 2. Flue gas condenser parameters

Parameter	Unit	Value
Total height, H	[mm]	140
Total width, W	[mm]	260
Total length, L	[mm]	250
Fin Height, h	[mm]	24
Gap width, w	[mm]	1.5
Spacing of cooling water channel, l	[mm]	36
Wall thickness, δ	[mm]	6

The measurement system includes the measurement of temperature, flow, velocity, pressure and so on, as shown in tab. 3. The temperature of flue gas and cooling water is measured by a digital thermometer. The wall temperature of NGHE was measured by the K-

type thermocouples. The relative humidity is measured by a high temperature hygrometer arranged at flue gas inlet and outlet side. The cooling water flow and natural gas flow are measured using an electromagnetic flowmeter and a Roots flowmeter in the natural gas pipeline, respectively. In addition, the flue gas inlet and outlet pressure are monitored by pressure sensors. The gas velocity in the duct is measured by a wind speed transmitter placed in the steady flow section in front of NGHE. During the experiment process, the condensate is collected by beaker and measured by an electronic scale. To facilitate the collection and measurement, NGHE is vertically arranged in the gas duct.

Table 3. Measuring equipment

Program	Device	Type	Measuring range	Accuracy
Flue gas temperature	Digital thermometer	DTM-180A	(–50) –300 °C	±0.1 °C
Flue gas humidity	Humidity transmitter	FK-TH800	0-100% RH	±1% RH
Cooling water temperature	Digital thermometer	DTM-180A	(–50) –300 °C	±0.1 °C
Cooling water flow	Electromagnetic flowmeter	CKLDG-D15	0.06-6.5 m³ per hours	±0.5%
Wall temperature	*K*-type thermocouple	SMPW-GG-K	0-400 °C	±0.1 °C
Differential pressure	Pressure sensor	PCM400H	0-1 kPa	±0.5 Pa
Flue gas velocity	Anemometer	JY-GD680	0-10 m/s	±0.2 m/s
Nature gas flow	Roots flowmeter	ZKTD-JLQ-60	0.07-60 m³ per hours	1.0%~1.5%
Oxygen volume fraction	Flue gas analyzer	Testo 340	0-25 vol.%	±0.2 vol.%
Condensing water quality	Electronic balance	ICS439-QA6 (Digital)	0-6kg	0.1g

The flue gas cooler and flue gas condenser adopt countercurrent heat transfer. The cooling water flow is regulated by a valve on the cooling water circuits, and its temperature is adjusted through a thermostatic water tank. In addition, the inlet flue gas temperature of condenser is regulated by cooling water temperature and flow of the flue gas cooler.

Experimental conditions

The flue gas after natural gas combustion is basically composed of CO_2, N_2, O_2, and H_2O. According to the combustion formulas, eqs. (1)-(3), the contents of natural gas and excess air coefficient, the flue gas compositions can be calculated.

$$\mathrm{C_mH_n} + \left(m + \frac{n}{4}\right)\mathrm{O_2} = m\mathrm{CO_2} + \frac{n}{2}\mathrm{H_2O} \tag{1}$$

$$\mathrm{H_2S} + 1.5\mathrm{O_2} = \mathrm{SO_2} + \mathrm{H_2O} \tag{2}$$

$$\mathrm{H_2} + 0.5\mathrm{O_2} = \mathrm{H_2O} \tag{3}$$

Generally speaking, many factors can affect the heat transfer characteristic of NGHE, including the flue gas velocity, v_g, flue gas temperature, T_g, cooling water temperature, $T_{c,in}$, cooling water flow, m_c, *etc.* Specifically, v_g has a significant effect on the residence time of the flue gas in heat exchangers, T_g determines the temperature difference with water, and m_c and $T_{c,in}$ will affect the surface temperature of the plate fin. Thus, these factors are emphatically studied in this paper, and the variable ranges of every factor are shown in tab. 4.

Table 4. Variable range of each parameter

Parameter	Unit	Value
Flue gas velocity, v_g	[ms^{-1}]	10~15.5
Flue gas temperature, T_g	[°C]	60~90
Flow of cooling water, m_c	[m^{3}h^{-1}]	0.2~0.8
Temperature of cooling water, T_c	[°C]	20~35

Analysis method

For NGHE, the condensation rate, ω, is the ratio of condensate flow, m_{cond}, to the amount of steam in the flue gas, m_v, the calculation formula is:

$$\omega = \frac{m_{cond}}{m_v} \cdot 100\% \tag{4}$$

The total heat transfer, Q_{tot}, is composed of two parts, namely, convection heat transfer, Q_{conv}, and the condensation heat transfer, Q_{cond}, the calculation formula is:

$$Q_{tot} = Q_{conv} + Q_{cond} \tag{5}$$

All the equipment is fully insulated, so heat dissipation can be neglected. The heat discharge of flue gas and the heat absorption of cooling water are basically equal. Therefore, eq. (5) can be converted to:

$$Q_{tot} = m_c c_{p,w}(t_{c,out} - t_{c,in}) \tag{6}$$

Condensation heat transfer, Q_{cond}, can be computed by the condensate flow, m_{cond}, and latent heat coefficient, γ, as shown in eq. (7). And the latent heat changes with dewpoint under the vapor partial pressure set in the experiment:

$$Q_{cond} = m_{cond}\gamma \tag{7}$$

Depending on the [29], total heat transfer coefficient, h_{tot}, is calculated by:

$$h_{tot} = \frac{Q_{tot}}{AT_m} \tag{8}$$

where A is the total heat exchange area and T_m – the logarithmic mean temperature difference. Meanwhile, h_{conv} is calculated by:

$$h_{conv} = \frac{Q_{conv}}{AT_m} \tag{9}$$

In addition, Nusselt number on flue gas side is calculated by:

$$\mathrm{Nu} = \frac{hd}{\lambda} \tag{10}$$

where λ is heat conductivity of condensing wall.

Results and discussion

Effects of cooling water flow rate and temperature

When flue gas velocity v_g = 12 m/s and inlet flue gas temperature T_g = 90 °C, the variation of condensation rate, ω, with cooling water flow, m_c, is shown in fig. 3. It can be seen that the condensation rate increases with cooling water flow, and the growth trend is gradually decreasing under different inlet cooling water temperature. When $T_{c,in}$ = 20 °C, ω changes from 40% to 56%. As $T_{c,in}$ increases, the condensation rate decreases. Even so, most condensation rates are above 30%. Because the increase of cooling water flow can greatly reduce the wall temperature, heat transfer temperature difference becomes larger. Then, condensation rate increases, and condensation heat transfer coefficient increases sharply.

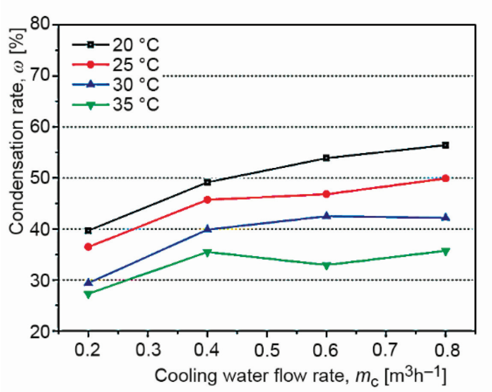

Figure 3. Chart of condensation rate variation with cooling water flow

Figure 4 indicates the changes of each heat transfer coefficient and cooling water flow, m_c, at different inlet cooling water temperature, $T_{c,in}$. When $T_{c,in}$ remains constant, total heat transfer coefficient, h_{tot}, and condensation heat transfer coefficient, h_{cond}, increase with cooling water flow, but convection heat transfer coefficient, h_{conv}, has little change. In addition, when $T_{c,in}$ gradually increases, the change of total heat transfer coefficient with cooling water flow becomes smaller and smaller. Especially, when $T_{c,in}$ = 35 °C, the condensation heat transfer coefficient changes slightly. This is because when $T_{c,in}$ is high, the increase of cooling water flow can not make the heat exchanger wall temperature change greatly, so the heat transfer temperature difference decreases, the heat transfer driving force decreases. Accordingly, the heat transfer driving force decreases, then the condensation rate reduces, the condensation heat transfer coefficient increases slightly.

When cooling water flow rate, m_c, retains 0.8 m^3 per hours, the changes of heat transfer coefficient under different inlet cooling water temperature, $T_{c,in}$, are displayed in fig. 5. It can be observed that h_{tot} is 320 W/m^{2}K as $T_{c,in}$ = 20 °C. It is about five times that of h_{conv}. When $T_{c,in}$ increases to 35 °C, h_{tot} decreases to 167 W/m^{2}K. It is just about 2.8 times that of h_{conv}. The variation trend of h_{tot} and h_{cond} is consistent, which means the condensation heat transfer is dominant. Besides, with $T_{c,in}$ increasing and m_c decreasing, the heat transfer driving force between flue gas and cooling water reduces, and then the condensation rate and heat transfer coefficient are getting smaller and smaller.

Effect of flue gas velocity

Flue gas velocity, v_g, directly affects condensation heat transfer effect, as shown in fig. 6. It can be seen that as cooling water flow rate, m_c, is 0.8 m^3 per hours, inlet cooling water temperature $T_{c,in}$ = 20 °C and flue gas temperature T_g = 90 °C, the condensation rate decreases with flue gas velocity significantly. The condensation rate reduces from 68% to 34%, when v_g changes from 11 to 15.5 m/s. The condensation rate reduces nearly half. To analyze the reason, it is just because when the flue gas flow increases, the residence time of flue gas in NGHE is shortened, resulting in the condensing process is not sufficient. Thus, the condensate increasing per unit time is less than the vapor content increasing in the flue gas, and condensation rate decreases.

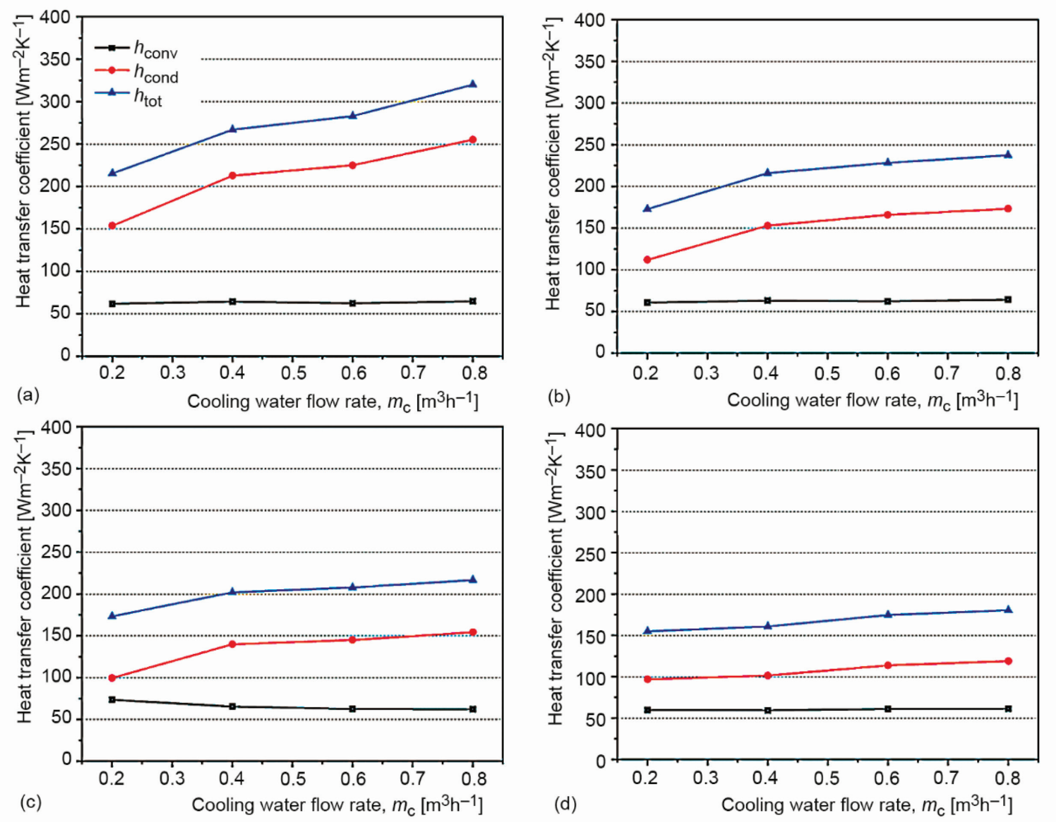

Figure 4. The changes of each heat transfer coefficient and cooling water flow; (a) $T_{c,in}$ = 20 °C, (b) $T_{c,in}$ = 25 °C, (c) $T_{c,in}$ = 30 °C, and (d) $T_{c,in}$ = 35 °C

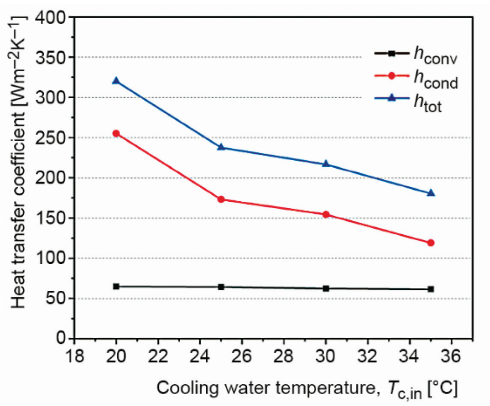

Figure 5. The changes of heat transfer coefficient and temperature of cooling water

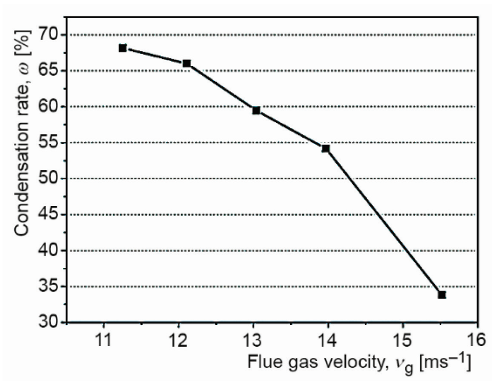

Figure 6. The changes of condensation rate and flue gas velocity

Figure 7 shows the variation of each heat transfer coefficient with different flue gas velocity, v_g. The results indicate that total heat transfer coefficient, h_{tot}, and condensation heat transfer coefficient, h_{cond}, increase obviously, while convection heat transfer coefficient, h_{conv}, just increases slightly. Especially when v_g is 15.5 m/s, h_{tot} reaches 365.7 W/m^{2}K, which is about seven times that of h_{conv}. Meanwhile, v_g is an important factor affecting the resistance of NGHE. As shown in fig. 8, the resistance increases from 360 to 720 Pa when v_g varies from 11 to 15.5 m/s. The resistance is nearly doubled.

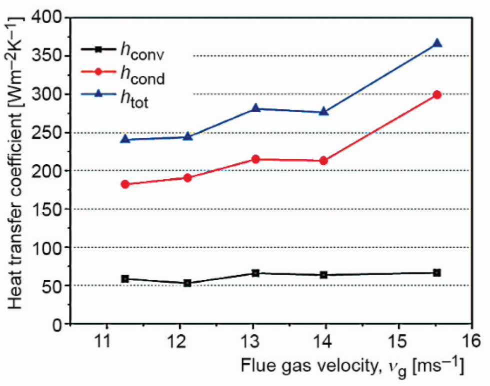

Figure 7. Relationship between flue gas velocity and heat transfer coefficient

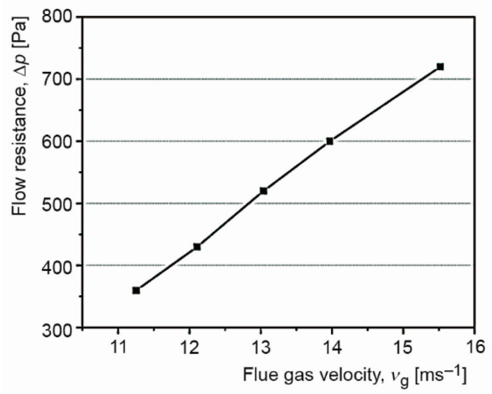

Figure 8. Relationship between flue gas velocity and flow resistance

Effect of flue gas temperature

Flue gas temperature, T_g, also has effect on condensation rate and heat exchange coefficient. When flue gas velocity v_g = 10 m/s, inlet cooling water temperature, $T_{c,in}$ = 20 °C, the effects of cooling water flow rate, m_c, on condensation rate is displayed in fig. 9. Obviously, condensation rate increases with cooling water flow rate, but has little change with the increase of flue gas temperature. With the increase of cooling water rate, the wall temperature of NGHE decreases. Accordingly, the heat transfer temperature difference between flue gas and NGHE wall increases, the driving force of heat and mass transfer increases. Then the water partial pressure in flue gas decreases, leading to an increase in condensation rate. In addition, when cooling water flow rate reaches 0.4 m^3 per hours, the condensation rate increases slightly, and condensation rate is more than 60%. Compared with an advanced condensing boiler system [30], the condensing rate of both is at the same level. To investigate the influence of gas temperature, a condition with the flow rate of 0.4 m^3 per hours is selected for further study, and the results are shown in fig. 10. It is observed that when flue gas temperature increases from 60 °C to 90 °C, convective heat transfer coefficient, h_{conv}, has no obvious change, and always stays at about 50 W/m^{2}K, while h_{cond} increases slowly. Namely, when the inlet water temperature and flow rate are appropriate, the change of flue gas has little effect on heat transfer coefficient.

The dimensionless correlation of Nusslet number

The presented results show that cooling water temperature, $T_{c,in}$, cooling water flow, m_c, flue gas temperature, T_g, and flue gas velocity, v_g, have different effects on the condensation heat transfer in NGHE. Specifically, m_c, $T_{c,in}$, and T_g mainly affect the condensing wall temperature. Flue gas dewpoint also has an effect on condensation heat transfer, the effect of temperature is expressed by dimensionless coefficient, Ln. In addition, Re reflects the effect of flue gas velocity. The condensation rate reflects degree of heat transfer from condensation. Ln and Re are defined:

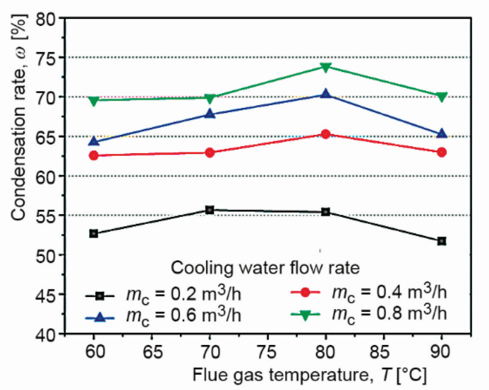

Figure 9. The changes of condensation rate and flue gas temperature

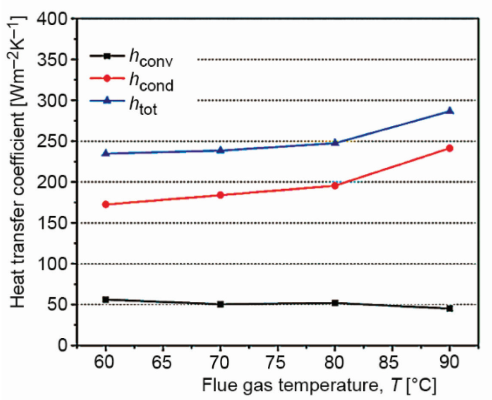

Figure 10. Relationship between flue gas temperature and heat transfer coefficient

$$\mathrm{Re} = \frac{\rho v d}{\mu} \tag{11}$$

$$\mathrm{Ln} = \frac{t_\mathrm{s} - t_\mathrm{w}}{t_\mathrm{g} - t_\mathrm{w}} \tag{12}$$

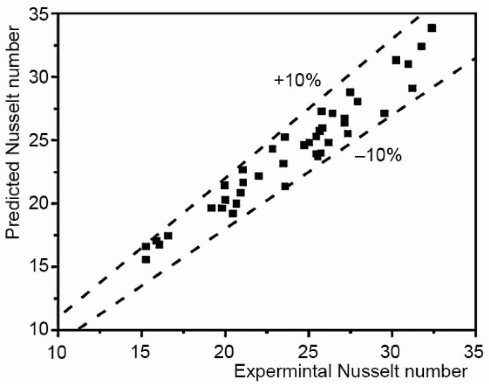

Figure 11. The fitting formula error of Nusselt number

In general, the Nusselt number can be expressed as a custom function of Re, Ln and ω:

$$\mathrm{Nu} = (a\mathrm{Re}^b \mathrm{Pr}^c \omega^d + e)\mathrm{Ln}^f \tag{13}$$

where a, b, c, d, e, f are the influence coefficient and need to be determined. By substituting the experimental results into eq. (13), the dimensionless criterion of flue gas condensation rate, eq. (14) can be obtained after multiple linear regression fitting:

$$\mathrm{Nu} = (0.04\mathrm{Re}^{0.789}\mathrm{Pr}^{0.333}\omega^{\mathrm{d}} + 23.6)\mathrm{Ln}^{0.419} \tag{14}$$

Based on the fitting formula, eq. (14), the predicted and experimental value of Nusselt number is shown in fig. 11. The error of all data points is within ±10%. The fitting formula is applicable to $1400 < \mathrm{Re} < 2500$, $0.2 < \mathrm{Ln} < 1.2$, $0.2 < \omega < 0.9$.

Conclusions

In this study, a new type NGHE is proposed and its heat transfer characteristics are experimentally investigated. Several factors affect the condensing heat transfer performance are mainly discussed, including the cooling water flow and temperature, flue gas flow and flue gas temperature. Following are the conclusions.

- For NGHE, cooling water flow rate and temperature have a significant effect on its heat transfer characteristics. The condensation rate, ω, increases with cooling water flow, m_c, and the growth trend is gradually decreasing under different inlet cooling water temperature, $T_{c,in}$. When cooling water flow rate reaches 0.4 m^3 per hours, the total heat transfer coefficient increases slightly. As $T_{c,in}$ increases, the condensation rate decreases. Even so, most condensation rates are above 30%.
- Flue gas velocity, v_g, directly affects condensation heat transfer effect. With the increase of flue gas velocity, condensation rate, ω, decreases rapidly while the heat transfer coefficient increases correspondingly. The condensation rate reduces from 68% to 34%, when v_g changes from 11 m/s to 15.5 m/s. Also, the total heat transfer coefficient can reach seven times of convection heat transfer coefficient. While the change of flue gas temperature has little effect on heat transfer coefficient. During the growth of flue gas temperature, condensation rate did not change significantly, but the heat transfer coefficients increased slightly.
- According to experimental results, the dimensionless correlation of the Nusselt number in NGHE is fitted as follows: $\mathrm{Nu} = (0.04\mathrm{Re}^{0.789}\mathrm{Pr}^{0.333}\omega^{0.546}+23.6)\mathrm{Ln}^{0.419}$. Compared to the experimental values of the Nusselt number, the error is within ±10%, and the validation range is $1400 < \mathrm{Re} < 2500$, $0.2 < \mathrm{Ln} < 1.2$, $0.2 < \omega < 0.9$.

Acknowledgment

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Nomenclature

A – area, [m^2]
a, b, c, d, e, f – experimental constants, [–]
c_p – specific heat, [$Jkg^{-1}K^{-1}$]
d – hydraulic diameter, [m]
h – heat transfer coefficient, [$Wm^{-2}K^{-1}$]
m – mass flow, [kgs^{-1}]
Q – heat transfer, [W]
Re, Ln, Nu – dimensionless parameter, [–]
T – temperature, [°C]
v – velocity, [ms^{-1}]

Greek symbols

ω – condensation rate, [%]
ρ – density, [kgm^{-3}]
μ – viscosity [Pa·s]
γ – latent heat, [$kJkg^{-1}$]

Subscripts

c – cooling water
g – flue gas
s – saturation
w – wall
in – inlet
tot – total
conv – convection
cond – condensation

Acronym

NGHE – narrow gap heat exchanger

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