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Heat and mass transfer characteristics of simulated high moisture flue gases

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Abstract The heat transfer process occurring in a condensing heat exchanger where noncondensible gases are dominant in volume is different from the condensation heat transfer of the water vapor containing small amount of noncondensible gases. In the process the mass transfer due to the vapor condensation contributes an important part to the total heat transfer. In this paper, the Colburn-Hougen method is introduced to analyze the heat and mass transfer process when the water vapor entrained in a gas stream condenses into water on the tube wall. The major influential factors of the convective-condensation heat transfer coefficient are found as follows: the partial pressure of the vapor p_v, the temperature of the outer tube wall T_w, the mixture temperature T_g, Re and Pr. A new dimensionless number *Ch*, which is defined as condensation factor, has been proposed by dimensional analysis. In order to determine the relevant constants and investigate the convection-condensation heat and mass transfer characteristics of the condensing heat exchanger of a gas fired condensing boiler, a single row plain tube heat exchanger is designed, and experiments have been conducted with vapor-air mixture used to simulate flue gases. The experimental results show that the convection-condensation heat transfer coefficient is 1.5∽2 times higher than that of the forced convection without condensation. Based on the experimental data, the normalized formula for convention-condensation heat transfer coefficient is obtained.

Keywords Condensation · Noncondensible gas · Heat transfer · Mass transfer · Condensing boiler

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A	heat transfer area m^2
Ch	condensation factor
c_p	specific heat at constant pressure, J/(kg·K)
G	mass flux Kg/(m^2·s)
h	heat transfer coefficient $W/(m^2 \cdot K)$
J	J-factor
Nu	Nusselt number
pa	pressure
Pr	Prandtl number
Q	heat transfer rate
q	heat flux W/m^2
r	latent heat, kJ/kg
Re	Reynolds number
Sc	Schmidt number
T	temperature, C or K
λ	heat conductivity m $W/(m \cdot K)$
ρ	density, kg·m^3
g	gas
h	moistened hot air
i	interface
v	vapor
w	water

1 Introduction

The exit flue gas temperature of a conventional boiler is usually higher than 150 C, sometimes as high as 200 C, to avoid low temperature corrosion. 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. Since 1970s, condensing boilers have been developed and have found wide applications in Europe and North America. In such a boiler, the exit flue gas temperature is reduced to 40–50 C so that the water vapor can be condensed and the latent heat released can be recovered. As such, the thermal efficiency of the boiler can be significantly increased. If the low heating value is still taken as the calculation basis, the efficiency can be as high as, or higher than 100%. Previous research has shown that SOx, NOx, dust and soot etc, which are the constituents of the flue gas, can be partially, even totally dissolved in the condensed water, and the pollutants emitted to the environment can be noticeably reduced. Therefore, it is of great significance both to energy conservation and environmental protection to utilize condensing boilers.

Due to the change of the energy policy of China and the emphasis to environmental protection, gas fired boilers are taking a larger market share. When natural gas is used as the fuel of boiler, as high as 20% of volumetric fraction of water vapor in combustion products will be generated, which is much higher than that when anthracite or bituminous coal is used as the fuel. More water vapor in flue gases means that more latent heat can be recovered, the thermal efficiency of the boiler can be more greatly improved by decreasing exit flue gas temperature. On the other hand, natural gas is much more expensive than coal, the operational cost of a gas fired boiler is generally much higher than that of a coal fired boiler. Thus, it is more profitable to recover the latent heat by condensing the water vapor in the flue gases of a gas fired boiler [1–3].

On the condensation heat transfer of the water vapor containing small amount of noncondensible gases, numerous information is available in the literature. In contrast, in the condensing heat exchanger of a condensing boiler, the heat transfer process is much different. The flue gas with water vapor at a superheated state flows across the tube bank, the water vapor entrained in the noncondensible gases that are dominant in volume (the volumetric percentage of noncondensible gases is as high as 80%–85%) condenses into liquid water while the tube wall temperature of the heat exchanger is below the saturation temperature of the water vapor, the tube wall will be partially or totally wetted by water film. The water film is surrounded by the gas boundary layer around the tube, the condensation of water vapor occurs at the interface of the water film and the boundary layer due to the diffusion driven by the vapor concentration difference across the layer. The heat transferred through the film includes two parts, which are the sensible heat due to the forced convection of the gases and the latent heat due to the vapor condensation respectively. Along the flow path of flue gases the water vapor content is gradually decreased because of its being condensed, which gives rise to a continuous reduction of corresponding partial pressure, thus saturation temperature. As the heat is released the flue gas constituents vary, therefore the physical properties of the flue gas continuously change, the heat transfer intensity will vary from point to point. Obviously, the heat transfer process is always accompanied by a mass transfer process, therefore, what happens in a condensing heat exchanger is a combined convection and condensation heat transfer process.

Previous research shows that the heat transfer coefficient of convection-condensation combination is higher than that of the forced convection without condensation, but on the same order, however, much lower than that of the condensation of pure steam [3–13]. Nevertheless, the mechanistic understanding on this phenomenon is insufficient, empirical or semi-empirical correlations are still in wide use for designing the condensing heat exchangers, further insight is necessary.

Idem et al [6–8] and Jacobi et al [9–11] investigated the heat and mass transfer performance of a finned-tube condensing heat exchanger both experimentally and analytically, using the same test setup. The experimental results were reduced to the form of $J = C\,\mathrm{Re}^{\mathrm{m}}$ for the forced convection heat transfer, where

$$J = \left(\frac{h}{Gc_p}\right)\mathrm{Pr}^{\frac{2}{3}}$$

is defined as sensible heat transfer J-factor. For the latent heat transfer related to the mass transfer, by analogy to convective heat transfer, a similar expression, which is $J = C'Sc^{\mathrm{m}'}$, is given. C, C', m, m' are determined by experiments.

Osakabe et al [12–13] proposed that the total heat flux q_{total} consists of the convection heat flux q_{conv} and the condensation heat flux q_{cond}, i.e.,

$$q_{total} = q_{conv} + q_{cond}$$

q_{total} was obtained from the temperature difference and flow rate of the cooling water. The convection heat flux is expressed as

$$q_{conv} = h_{conv}(T_g - T_w)$$

where T_g and T_w are the temperature of bulk gas and the temperature of tube wall respectively. The condensation heat flux q_{cond} can be expressed as

$$q_{cond} = h_{cond}r(C_g - C_w)$$

where C_g and C_w is the mass concentration of bulk vapor and the mass concentration of saturated steam at the wall temperature T_w respectively, r is latent heat. The Nusselt number Nu for the average convective heat transfer coefficient is

$$Nu = C\mathrm{Re}^{0.6}\mathrm{Pr}^{m}\left(\frac{\mathrm{Pr}}{\mathrm{Pr}_{wall}}\right)^{0.25}$$

Similarly, for the calculation of the mass transfer coefficient h_{cond}, the Nusselt number and the Prandtl number in the above equation are simply replaced by the Sherwood number and the Schmidt number respectively, following equation is obtained:

$$Sh = C\mathrm{Re}^{0.6}Sc^{m}\left(\frac{Sc}{Sc_{wall}}\right)^{0.25}$$

It can be seen that in previous research the processes of convective heat transfer and mass transfer (condensation), that is, sensible heat transfer and latent heat transfer, are separately considered to obtain the total heat transfer capacity. In reality, both the convection heat transfer and the mass transfer are simultaneously performed in the process.

In this paper, the total heat transfer coefficient is represented by one equation, which has been obtained by analyzing the heat transfer and the mass transfer process, a new dimensionless number known as condensation factor is put forward. Experiments have been conducted to determine the relevant constants. In the experiments, the hot moistened air is used to simulate the flue gas of a natural gas fired boiler, the partial pressure of the vapor divided by the total pressure of the fluid stream ranges from 3% to 20%, the temperature is varied between 100C and 200C, one row of plain carbon steel tubes are used as the heat exchanger. Based on the experimental data, a normalized correlation is put forward to predict Nu_h of the flue gases, the correlation is the function of partial vapor pressure p_v, Reynolds number Re, Prandtl number Pr, wall temperature T_w and mixture bulk temperature T_g, and the correlation can be conveniently used to design condensing heat exchangers.

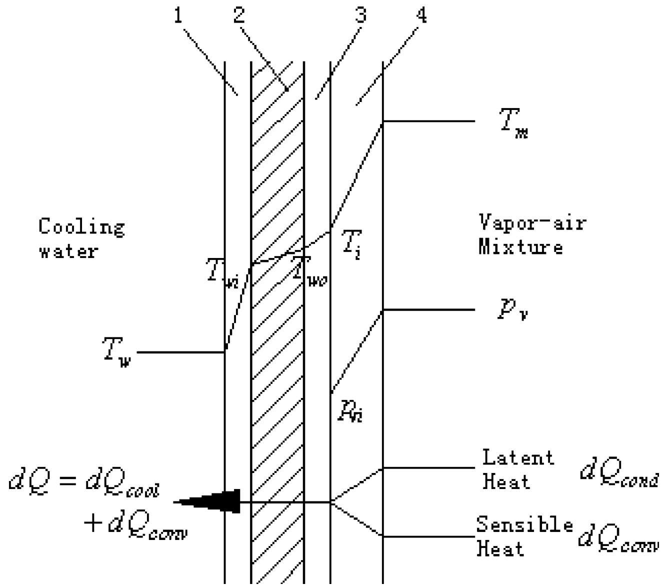

Fig. 1 Simplified Colburn-Hougen Model 1- Cooling Water Boundary Layer; 2- Tube Wall; 3- Condensate Film; 4- Noncondensible Gas Boundary Layer

2 Analyses

The condensation process in the presence of noncondensible gases is different from that of pure vapor. The temperature the condensation occurs at is variant along the heat exchanger, in addition to heat transfer, mass transfer plays an important role. The process can be analyzed by assuming that a boundary layer of noncondensible gases exists in the vicinity of condensate film, and the total resistance to heat transfer and mass transfer in the gas phase is lumped in this layer.

According to Colburn-Hougen model [14], a condensation process in which noncondensible gases exist can be analyzed. The driving forces of heat and mass transfer, or the temperature and pressure distribution are illustrated in Fig. 1

While the tube temperature is below the dew point of the water vapor in the moistened hot air stream, the condensation of the vapor begins, the tube surface is gradually wetted by the condensate, and a liquid film 3 is formed, which is referred to as condensate film. A gas boundary layer 4 in which the convective heat transfer and the mass transfer take place, is assumed to exist outside the condensate film, and the temperature is reduced from bulk gas temperature T_g to interfacial temperature T_i, the partial pressure of the water vapor is decreased from the partial pressure p_v of bulk vapor to the partial pressure p_{vi} at the interface. T_g–T_i is the driving force of heat transfer, and p_v–p_{vi} is the driving force of mass transfer.

The vapor reaches the outer surface of the condensate film, i.e., the interface, where its condensation occurs, through the noncondensible gas layer. If the area dA of an infinitesimal element at the outer side of the boundary layer is considered, the heat dQ_{cond} transferred at the surface of the condensate film due to vapor condensation is

$$dQ_{cond} = h_m r(\rho_v - \rho_{vi})dA \tag{1}$$

where h_m is mass transfer coefficient, r is latent heat of vaporization and A is area, ρ_v, ρ_{vi} are the concentrations of the vapor in bulk stream and at interface respectively.

From the ideal gas law

$$c_i = \frac{p_i}{RT} \tag{2}$$

where c_i is the molar concentration and p_i is the partial pressure, R is the universal gas constant.

The density of gas is

$$\rho_i = c_i M_i \tag{3}$$

where ρ_i is density and M_i is molecular weight.

From Eq. (2) and Eq. (3), Eq. (1) can be rewritten as

$$dQ_{conv} = h_m r M_v \left(\frac{p_v}{RT_g} - \frac{p_{vi}}{RT_i}\right)dA \tag{4}$$

The sensible heat transferred due to forced convection is

$$dQ_{conv} = h_g(T_g - T_i)dA \tag{5}$$

where h_g is the convective heat transfer coefficient of the mixture.

Thus, the total heat transferred from the mixture bulk to the condensate film is

$$\begin{aligned} dQ &= dQ_{conv} + dQ_{cond} \\ &= h_m r M_v \left(\frac{p_v}{RT_g} - \frac{p_{vi}}{RT_i}\right)dA + h_g(T_g - T_i)dA \end{aligned} \tag{6}$$

The heat transfer coefficient correspondingly is

$$h'_h = \frac{dQ}{(T_s - T_i)dA} = h_m r M_v \left(\frac{p_v}{RT_g} - \frac{p_{vi}}{RT_i}\right) / (T_g - T_i) h_g \quad (7)$$

According to the analogy of heat and mass transfer, the relationship between heat transfer coefficient and mass transfer coefficient is

$$h_m = \xi h_g \frac{1}{\rho c_{pm}} \quad (8)$$

where c_{pm} is the specific heat of vapor-air mixture, ξ is a constant close to unity.

Eq. (7) becomes

$$h'_h = h_g \left[1 + \xi \frac{1}{\rho c_{pm}} r M_v \left(\frac{p_v}{RT_g} - \frac{p_{vi}}{RT_i}\right) / (T_g - T_i)\right] \quad (9)$$

According to previous studies, the heat resistance exists mainly in the noncondensible gas boundary layer, although the thickness of condensate boundary layer is very thin. Under present experimental conditions, the volumetric fraction of noncondensible gases is above 80% so that the heat resistance of condensate is negligible, as such

$$h'_h = h_h \quad (10)$$

where h_h is the convection-condensation heat transfer coefficient on the outer tube wall. If the heat resistance of the condensate film is ignored

$$T_i = T_w \quad (11)$$

Finally, the convection-condensation heat transfer coefficient is obtained as follows:

$$h_h = h_g \left[1 + \xi \frac{1}{\rho c_{pm}} r M_v \left(\frac{p_v}{RT_g} - \frac{p_{vi}}{RT_w}\right) / (T_g - T_w)\right] \quad (12)$$

As we know, the influential factors of h_g are fluid velocity u, characteristic length l, density ρ, viscosity η, heat conductivity λ and specific heat c_p. From above equation, the influential factors of convection-condensation heat transfer, in addition, are the partial pressure of vapor p_v, bulk gas temperature T_g and tube wall temperature T_w.

In above equation, T_g–T_w represents the driving force of sensible heat transfer, $\frac{p_v}{RT_g} - \frac{p_{vi}}{RT_w}$ represents the driving force of latent heat transfer. Since saturation state is attained at the interface, p_{vi} is the partial pressure of saturated vapor, and it is dependent only on the interfacial temperature $T_i(T_w)$. Essentially, $\frac{p_v}{RT_g} - \frac{p_{vi}}{RT_w}$ represents the degree of departure from saturation, i.e., the wall subcooling, thus it can be represented also by a temperature difference. The temperature difference is $T_{sat(pv)}$–T_w, where $T_{sat(pv)}$ is the saturation temperature corresponding to the partial pressure of the bulk vapor. As such, following functional relationship yields

$$h_h = f(w, l, \rho, \mu, \lambda, c_p, T_g - T_w, T_{sat(pv)} - T_w) \quad (13)$$

If dimensional analysis of Eq. (13) is made, a new dimensionless number Ch can be obtained:

$$Ch = \frac{T_{sat(pv)} - T_w}{T_g - T_w} \quad (14)$$

$T_{sat(pv)}$ is the saturation temperature corresponding to the partial pressure of vapor p_v. Ch is known as condensation heat transfer factor, or simply condensation factor.

The normalized correlation of convention-condensation heat transfer can be expressed as

$$Nu_h = a\mathrm{Re}^b \mathrm{Pr}^c Ch^d \quad (15)$$

The Prandtl number Pr in the investigated range can be considered to be a constant, and like the forced convection heat transfer, c can be taken as 1/3.

3 Experimental

The experimental setup is illustrated in Fig. 3.

Air is forced by draft fan into primary electric heater, where it is heated to about 300 C, then goes into primary water sprayed section, in which water is pumped into the nozzle and atomized, and transformed into steam due to heat absorption from the hot air. The air vapor mixture flows over the secondary electric heater and enters the secondary water sprayed section. The moisture-laden gas stream flows downwards in the heat exchanger, where condensation takes place. After the vapor entrained in the gas stream is condensed, latent heat is released. The condensed water is collected by a tray and is weighed. The cooling water flows into the heat exchanger from bottom, in a counter-flow arrangement with the gas stream. The flow rates of air, sprayed water and cooling water are changed by regulating valves and electrical power is adjusted by voltage regulator in the experiments. The flow rates of sprayed water and cooling water are measured by rotameters.

The room temperature, the moistened gas stream temperature, the condensing heat exchanger tube wall

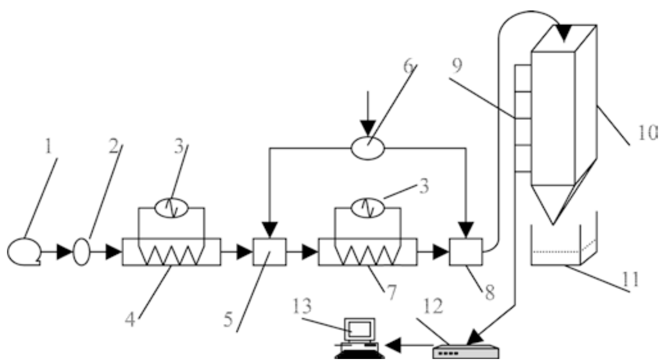

Fig. 2 Experimental Setup 1 – Forced Draft Fan; 2 – Orifice Flowmeter; 3 –Transformer; 4 –Electrical Heater I; 5 –Water Spraying Section I; 6- Water Pump; 7 –Electrical Heater II; 8 – Water Spraying Section II; 9 – Thermocouples;10 – Condensing Heat Exchanger; 11 – Condensed Water Collecting Pan;12 – Data Acquisition System; 13 – Computer

temperature and the cooling water temperature are measured by nickel chromium-nickel silicon thermocouples with high sensitivity and good thermoelectrical performance.

The heat exchanger is composed of horizontal carbon steel tubes, its characteristic geometry is described in Table 1.

Wilke method [15] and Wassiljewa method [16] are used to calculate the viscosity and heat conductivity of the mixture respectively. Other physical properties such as specific volume, density, kinematic viscosity can be obtained in terms of the law of mixed gases.

After the test loop is in stable operation, the heat Q transferred through tube wall from the vapor-air mixture is equal to the heat absorbed by the cooling water, i.e.,

$$Q = G_w c_w (T'_w - T''_w) \tag{16}$$

where G_w is the mass flow rate of cooling water, c_w is the specific heat of cooling water, T'_w and T''_w is the inlet and outlet temperature of cooling water respectively.

The convection-condensation heat transfer coefficient is calculated by

$$h_h = \frac{Q}{A(\bar{T}_m - \bar{T}_{wo})} \tag{17}$$

where $\bar{T}_w$is the average temperature of vapor-air mixture and $\bar{T}_w$is the average temperature of outer tube wall.

The uncertainties in the measured properties were estimated to be as in Table 2.

The method of Kline and McClintock [17] was employed to evaluate the uncertainties of the experimental results. The maximum uncertainty in the Reynolds number was estimated to be 8%. The estimation of the uncertainty in the Nusselt number can be completed by combining Eq. (16) and Eq. (17), and the maximum uncertainty was 5.1%.

Table 1 Characteristic Geometry of Heat Exchanger

Outside diameter	20 mm
Wall thickness	2 mm
Cross-section	500 mm × 40 mm
Longitudinal pitch	40 mm
Number of tubes	70

Table 2 Experimental uncertainties

Property	Uncertainty	Range
Cooling water flow rate	0.02 m^3/hr	0.4–1.6 m^3/hr
Moistened air flow rate	0.004 m^3/s	0.05–0.08 m^3/s
Inlet cooling water temperature	0.1 C	10–13.5 C
Outlet cooling water temperature	0.1 C	24.6–36 C
Inlet moistened air stream temperature	0.1 C	150–250 C
Tube wall temperature	0.1 C	16.3–30 C

4 Results and Dicussions

Experiments have been performed to examine the effects of various factors, including cooling water flow rate, gas flow rate (Reynolds number), volumetric fraction of water vapor etc, on the convention-condensation heat transfer coefficient.

The relationship of Nusselt number Nu_h versus Reynolds number Re is shown in Fig. 3. Fig. 4 presents the relationship of Nusselt number Nu_h versus condensing factor Ch. The solid line in the figures represents the best fitted line of the experimental data.

Re is raised, the forced convection takes greater part, i.e., the sensible heat transfer is boosted, which leads to the increase of h_g in Eq. (12).

In the expression of condensation factor, $Ch = \frac{T_{sat(pv)} - T_w}{T_g - T_w}$, the numerator $T_{sat(pv)}$–T_w is the deference between the saturation temperature corresponding to the partial pressure of water vapor and the tube wall

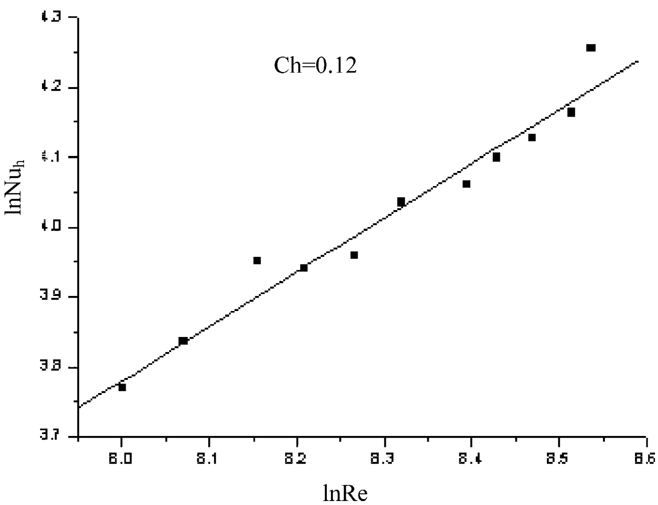

Fig. 3. Relationship of Nu_h and Re

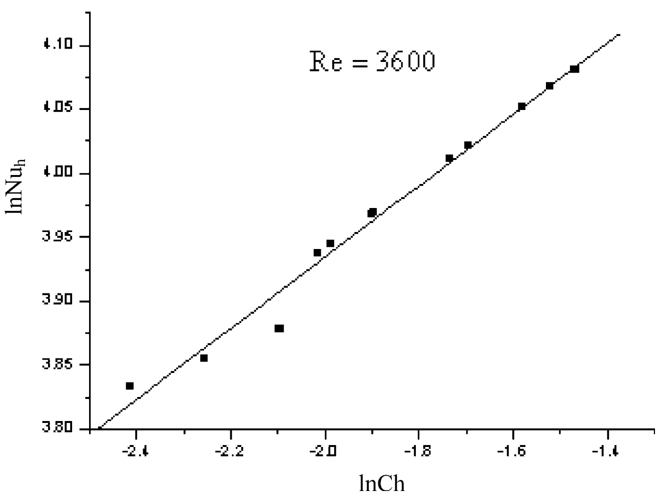

Fig. 4 Relationship of Nu_h and Ch

temperature, i.e., the tube wall subcooling. Higher subcooling means greater driving force of mass transfer, thus, more water vapor can be condensed, and more latent heat can be released, higher Nu_h is obtained. The denominator is the difference between the bulk gas stream temperature and the tube wall temperature. From Newton's cooling law, for a given amount of heat exchanged, larger temperature difference means lower heat transfer coefficient. Generally, the physical properties of the gas in the experimental range are very weakly affected by the temperature, thus, the convective sensible heat transfer coefficient is invariant, according to Eq. (12) the convective-condensation heat transfer coefficient h_h decreases with the temperature difference T_g–T_w.

Multivariate regression method has been employed in order to determine the unknown constants in Eq. (15) in terms of the experimental data, following normalized equation for convention-condensation heat transfer is obtained:

$$Nu_h = 0.1823\mathrm{Re}^{0.7707}\mathrm{Pr}^{1/3}Ch^{0.2615} \tag{18}$$

In above equation, the characteristic temperature is the mean temperature of inlet vapor-air mixture, and the characteristic dimension is the outside diameter of tube, the partial pressure of the vapor at inlet is taken as p_v, T_g is the mean temperature of bulk mixture, T_w is the average temperature along the tube length.

Fig. 5 shows the comparisons between experimental data and calculated results using the obtained nondimensionalized equation for Nu_h throughout the heat exchanger. Good agreement can be found, and the maximum deviation is within ± 5%

The experiments on the heat transfer characteristics of forced convection (without condensation) of air have been conducted to compare the heat transfer characteristic of air and vapor-air mixture, as shown in Fig. 6. The experimental conditions are $Ch = 0.15$, $\mathrm{Pr} = 0.656$. It can be found that the convection-condensation heat transfer coefficient is 1.5∽2 times higher than that of the forced convection without condensation.

The heat transfer process occurring in the condensing heat exchanger can be examined by dividing it into several stages. In the present study, five stages are divided, which are marked along the gas flow path as A, B, C, D and E respectively. Shown in Fig. 7 are the heat transfer coefficients of staged heat exchangers when the volumetric gas flow rates are 0.08, 0.0774 and 0.074 m^3/s respectively, the volumetric fraction of water vapor at inlet is 0.128, the volumetric flow rate of cooling water is 0.6 m^3/h. The entering gas temperature is 250 °C. As the condensation proceeds, the water vapor fraction is continuously decreased, the corresponding saturation temperature $T_{sat(pv)}$ is also decreased, and the gas temperature T_g is decreased more rapidly, which gives rise to higher Ch. As such, the staged convective-condensation heat transfer coefficient of the heat exchangers downstream along the gas flow path gets higher.

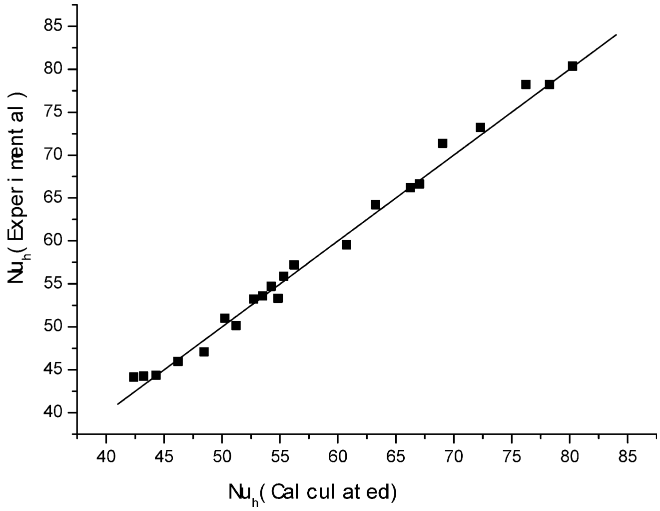

Fig. 5 Calculated and Experimental Nu_h

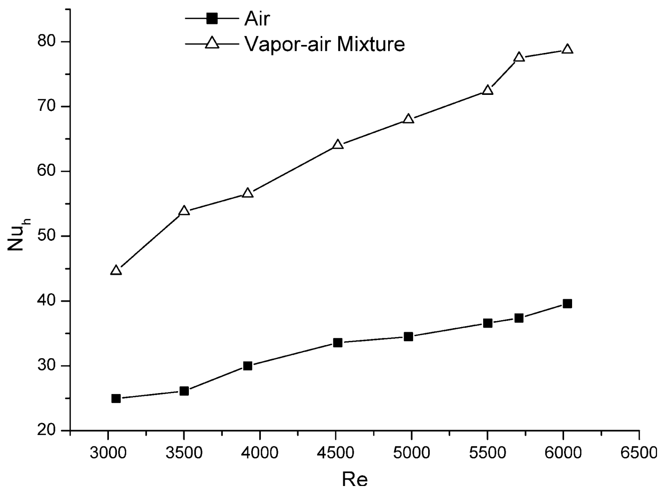

Fig. 6 Comparison of Heat Transfer Characteristic of Air and Vapor-Air Mixture

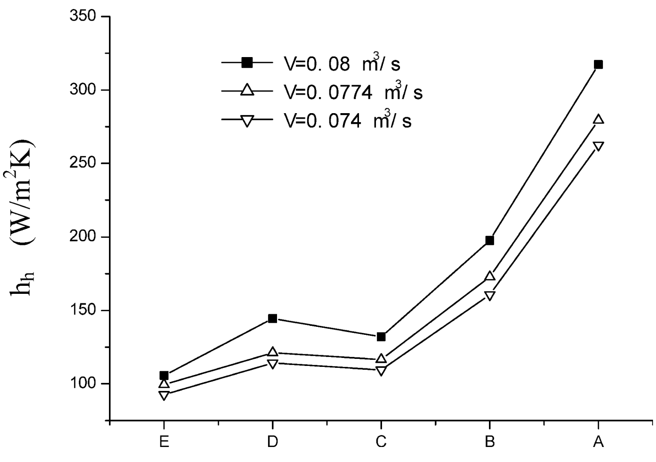

Fig. 7 Staged Heat Transfer Coefficient along Gas Flow Path

Conclusions

Colburn-Hougen method can be used to analyze the heat and mass transfer process when moisture laden gas flows across a heat exchanger and the water vapor condenses into water on the tubes. The influential factors of convective-condensation heat transfer coefficient include the partial pressure of the vapor p_v, the temperature of the outer tube wall T_w, the mixture temperature T_g, Re and Pr. A new dimensionless number Ch, which is called condensation factor, has been put forward. Based on the experimental data, the nondimensionalized equation for convention-condensation heat transfer can be expressed as $Nu = 0.1823\ \mathrm{Re}^{0.7707}\ \mathrm{Pr}^{1/3}\ Ch^{0.2615}$ in the experimental range. The convection-condensation heat transfer coefficient is $1.5 \backsim 2$ times higher than that of the forced convection without condensation. The staged convection-condensation heat transfer coefficient gets higher downstream along the gas flow path.

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