Condensation of steam with high CO₂ concentration on a vertical plate

Minghui Ge^{a,b}, Shixue Wang^b, Jun Zhao^b, Yulong Zhao^b, Liansheng Liu^a

^a School of Energy and Environmental Engineering, Hebei University of Technology, Tianjin, 300401, China

^bKey Laboratory of Efficient Utilization of Low and Medium Grade Energy (Tianjin University), Ministry of Education of China, Tianjin, 300072, China
Tel: +862227402567; E-mail address: wangshixue_64@tju.edu.cn

Abstract: The effect of a non-condensable gas on steam condensation has been studied. From experimental and theoretical studies to date, clear evidence of significant heat transfer deterioration by the non-condensable gas can be seen. The deterioration varies with the concentration of the non-condensable gas, geometric parameters, system pressure, etc. However, these data and correlations may be unreliable due to the high concentration of CO₂ in the vapor mixture and lack of verification through experiments. In this paper, a systematic investigation has been conducted for film-wise condensation using a vapor mixture of steam and a high concentration of CO₂. Some data were obtained on a vertical plate, with the average vapor velocity of 1.2 m/s, CO₂ mass fraction of 20%—94% and a pressure of 1 atm. The effects of CO₂ concentration and surface sub-cooling on heat transfer characteristics have been investigated. Moreover, the relevant parameters such as gas/liquid film resistance were included in the description of the condensation phenomenon. The developed correlation for steam and a high concentration of CO₂, based on experimental results, has a standard deviation of less than $\pm 20\%$.

Keywords: Non-condensable gas; CO₂; Condensation heat transfer characteristics; Vertical plate; High concentration

1. Introduction

It is well established that the presence of even tiny amounts of a non-condensable gas produce a large reduction in steam condensation characteristics. Theoretical and experimental explanations for the impeding effects of the non-condensable gas have been given by many researchers to date. Most of the investigations have involved the use of a non-condensable gas such as air or nitrogen^[1]. In case of CO₂ capture and storage processes (namely CCS technology), however, the CO₂ gas serves as a non-condensable gas present in steam, for either the amine absorption or calcium-looping methods^[2]. The CO₂ must be purified for deep burial, and the steam must return to the system for recycle. Therefore, the research of steam condensation heat transfer with a high fractional CO₂ gas has important application value for CO₂ purification.

Since steam condensation with a non-condensable gas was quantitatively researched by Othmer^[3] first in 1929, interesting as this phenomenon may be, it remains at present largely of academic interest even though there are possible other

applications. In particular, with the development of seawater desalination and the nuclear power industry, the condensation heat transfer with a non-condensable gas is becoming a research focus. Earlier results^[3-31] studied few possibly trace amounts, of non-condensable gases^[4,8], geometries^[16-21,26-28], and flow patterns^[6,29], while only a few experiments involved the condensation heat transfer process with a high fractional non-condensable gas^[7,15,30-34]. The general trend the research follows is: attaining a more reasonable experimental system design, and then increased reliability of the measurement of variable parameters, especially the temperature of the condensing plate, temperature of the bulk vapor mixture, and gas-liquid interface temperature.

For condensation of steam/non-condensable gas mixtures, several experimental investigations have been conducted on a vertical plate by Al-Diwanni and Rose^[5], Desesquelles and Foch^[6], Huhtiniemi and Corradini^[7], and Wei^[8]. Al-Diwanni and Rose^[5] found that the heat transfer coefficient increased with an increase of total pressure under the conditions of 0—25% non-condensable gas and 5°C—80°C surface sub-cooling. The experiments and simulation results made by Desesquelles and Foch^[6] over a flat plate have good consistency for both a laminar and turbulent boundary layer. Referring to the report from Huhtiniemi and Corradini^[7], which is schematically drawn experimental data for condensation with a steam-air mixture on a plate varied from 0 to 90°, the heat transfer coefficient decreased 15—25% depending on the concentration of the non-condensable gas. Wei^[8] observed the appearance of the condensate film on a vertical plate (50 × 50 mm) using air-R113 mixtures with various air mass fractions between 0 and 2.6%.

Different from other thermal physical phenomena, the modeling of condensation with a non-condensable gas on a vertical wall has been investigated in theory more than experimentally. However, the use of a semi-empirical approach in models of the condensation process has attracted increasing attention, due to its advantage for stronger generality. The two first published correlations are from Uchida^[9] and Tagami^[10]. Uchida's correlation is in good agreement with the condensation experimental results obtained by Sagawa^[11]. All of these experiments were conducted on a steel surface with an area of 140 mm × 300 mm and a system pressure close to atmospheric pressure. Dehbi^[12] investigated the condensation of a steam-air mixture and steam-helium-air mixture with a velocity between 1 m/s and 3 m/s. For the steam-air experiments, the variation of the heat transfer rate with the non-condensable fraction is logarithmic. For a steam-helium-air mixture, the heat transfer coefficient deteriorated as a function of an increasing helium fraction, especially in the region of low air fractions. Kataoka et al^[13] proposed a correlation depending only on η based on a facility with a 4.2-m high vertical wall. The condensation behavior under the presence of a non-condensable gas on the condensing wall height of 0.9 m—4.2 m was experimentally studied by Murase^[14]. An empirical formula for the steam/air mass ratio was used to calculate the heat transfer coefficient under the conditions of $q \cdot L \leq 40$ kW/m (q is the heat flux and L is the height of the heat transfer area), and concluded that the effect of the falling liquid film is negligibly small. Steam condensation performances on a vertical smooth tube have been researched by Liu^[15].

With an air mass fraction of 30%—65%, an empirical correlation for the heat transfer coefficient takes the steam mole fraction, total pressure, and surface sub-cooling as variables.

In addition to the plate form mentioned above, some researchers have also selected other condensing forms. Ivashchenko et al.^[16] found that the heat transfer coefficient decreases so dramatically in the range of 0—5% compared with other non-condensable concentrations for a steam-nitrogen mixture condensation on a vertical tube. The effect of an interfacial shear for a condensation in a vertical tube has also been studied by Lee and Kim^[17]. Oh and Revankar^[18] conducted experiments for a steam-air mixture condensation with secondary pool cooling, in place of the previous jacket cooling. Tadeusz^[19] showed the dependence of air concentration on the R404A in a tubular channel. With a decrease in pressure of 30%, the heat transfer coefficient for 2.5% mixtures decreased 20% to 50% more than those for a pure refrigeration medium. Dalkilic and Wongwises^[20] presented experiments for film condensation in a tube with an inner diameter of 8.1mm. Data obtained under different non-condensable concentrations, condensation temperature, and mass flux conditions were summarized into correlations to analyze the dependence of a pressure drop on the heat transfer coefficient. An approximate approach to an analysis of film condensation in a vertical tube was proposed by Berrichon et al.^[21]. The model was in good agreement with their experimental data for pure steam and steam-air condensation in a vertical tube under high pressure. Measurements with low pressure, on the other hand, yield a heat transfer coefficient decrease of 50% when the air concentration was 4%.

The steam condensation with a high fractional non-condensable gas has been performed based on different application backgrounds. Experiments with non-condensable gases ranging from 30% to 65% were conducted on a vertically mounted tube to evaluate the heat removal capacity of a passive cooling unit in a post-accident containment by Liu^[15]. Tong^[30] used a pin-fin tube in the condensation of pure steam and steam-air. He found that the enhancement ratio would maximally reach 1.8 with an air mass fraction larger than 90%. Zhu^[31] designed a shell and tube condenser to evaluate the performance of the steam condensation process with an air mass fraction from 34% to 81%. The overall heat transfer coefficient varied from 350 to 3000 W/(m²·°C), which has promising applications to the dewvaporation desalination process in the presence of a high fractional non-condensable gas, and satisfies applicable requirements in industry. Caruso et al.^[32,33] performed a study on the average heat transfer coefficient for condensation inside horizontal and inclined tubes, at atmospheric pressure and in a wide range of non-condensable gases concentrations. They showed that condensation in the presence of non-condensable gases is not sensibly affected by inclination, especially at high gas concentrations.

From experimental and theoretical studies to date, clear evidence of significant heat transfer deterioration by the non-condensable gas can be seen. The deterioration varies with the non-condensable gas concentration, geometric parameters, system pressure, etc. However, these data and their correlations may be unreliable due to the high concentration of CO₂ in the vapor mixture and lack of experimental verification.

Therefore, a more extensive systematic experimental investigation is needed. The present investigation provides new data for film condensation with high concentration of CO₂ on a vertical plate.

2. Experimental apparatus and measuring method

2.1 Experimental apparatus

The stainless-steel test apparatus, shown schematically in Fig.1, consisted of a loop, with test vapor (CO₂ gas and steam) generated in an electrically-heated boiler (maximum power is about 12kW). The vapor mixture was directed vertically upward through a calming section before flowing over the vertical, water-cooled, test condenser **block**. The condensing chamber was made from stainless steel with a volume of 140 mm × 20 mm × 300 mm, and the flow area was 140 mm × 20 mm. Reducing pipes were located at both inlet and exit of the test chamber to ensure uniform distribution of the gas mixture in the vertical direction. Excess vapor was condensed in the auxiliary condenser below the test section and all the condensate was returned to the boiler by a plunger pump. Cooling water was supplied via an electromagnetic flowmeter and sprayed to the cooling surface to ensure the condenser **block** maintained a specific surface sub-cooling, so that the vapor could condense and the heat could be transferred from the vapor to the **condensing plate**. A counter-current auxiliary condenser is adopted to condense the excess of vapor by cooling water. The test apparatus between the boiler and the section were well thermally insulated with polystyrene foam. The system maintained at a positive test section gauge pressure of 2 kPa to minimize the intrusion of any non-condensable gases.

Fig.1 Schematic diagram of experimental system

K-type thermocouples were used for temperature measurements. All thermocouples were calibrated in a high precision constant temperature bath against a

platinum resistance thermometer. A T-shaped brass condenser **block** of 20 mm $\times$ 20 mm was utilized in the experiment, with a cooling surface of 54 mm $\times$ 54 mm, as shown in Fig.2. Five equi-spaced slots, 0.6 mm square, were machined axially along the **condensing plate**. Thermocouples were inserted in the slots to measure the outer condensing **plate** temperature. The distortion of the isothermal in condensing block due to the heterogeneous heat flux on the condensing **plate** can be negligible according to the investigation of Kumagai et al^[34]. **As a result, the temperature distribution in the condenser block obtained by thermocouples during steam condensation can satisfy the one dimensional Fourier conduction law.** In addition to the condensation surface and cooling surface, the remaining surfaces were packaged with polytetrafluoroethylene (PTFE), so that the heat flux and temperature gradient only occurred in the direction of the block thickness.

Fig.2 Configuration of thermocouple sensors installed in the condensing plate

To ensure film-wise condensation, the following procedures were always completed before the installation of the condenser block. The block was first polished using P800 sandpaper and rinsed with distilled water. It was then cleaned by immersion for a few minutes in a 20 wt% sulfuric acid solution and then alcohol. Finally, the condenser block was rinsed with distilled water again and carefully installed in the test section. A standard syringe was used to supply an approximate volume of 1 μ L droplet onto the **condensing plate**. The average static contact angle was 35.4°, as shown in Fig. 3.

Fig.3 Contact angle measurements of **condensing plate**

Prior to measurements, the vapor mixture was vented to atmosphere for an hour to eliminate air in the test section. The reducing valve used in CO₂ supply system has the heating function. It can assure the temperature of CO₂ gas at almost 20°C when flow into the steam generator. The velocity of the vapor mixture was in advance estimated by adjusting the heating power of the steam generator and the CO₂ inlet mass flow, but finally was measured by a stopwatch and glass cylinder. Normally it took about ten minutes for conditions in the test section to become steady after varying the heating power of boiler and CO₂ mass flow. To obtain a uniform variation of surface sub-cooling, the cooling water temperature rise was controlled at 0.5°C per

minute by voltage regulator.

During experiments, the ambient temperature and pressure, test section gauge pressure, cooling water flow rate, temperature and pressure of the vapor in the inlet of condenser chamber, cooling water inlet and exit temperature, and test plate wall temperature were measured. Visual observation of the condenser block was also made through a glass window in the test section to confirm film-wise condensation. When necessary, a mica strip heater was used on the outside surface to clear condensate from the window for observation. To avoid condensation on other walls of the chamber, other than controlling the heating power of the mica strip to ensure the temperature of glass surface a little higher than the bulk vapor mixture, the condensing chamber is thermally insulated with polystyrene foam to assure the condensation occur only on the surface of vertical plate.

2.2 Data reduction and analysis for system reliability

2.2.1 Mass concentration calculation of CO₂

On the basis of the Gibbs-Dalton relationship, the gas liquid equilibrium of CO₂ and vapor can be obtained at atmospheric pressure, as shown in Fig.4. The mass concentration of CO₂ gas under a corresponding temperature could be obtained using the phase equilibrium relation of vapor-CO₂ mixtures.

Fig.4 Gas-liquid equilibrium of CO₂ and vapor under standard atmospheric pressure

The vapor pressure and temperature are measured at the inlet of the condensing chamber. Three K-type thermocouples are placed there in order to improve the measurement accuracy, and the arithmetic mean value is considered to be the actual temperature of mixed vapor.

$$T_b = \frac{1}{3} \sum_{j=1}^3 T_{jb} \quad (1)$$

where T_{jb} is the temperature of the mixture measured with the j^{th} thermocouple. T_b is the average temperature of the mixed vapor.

2.2.2 Heat transfer coefficient

In this experiment, a linear fit to the temperature gradient of the condensation block has been made, obtaining the condensation surface temperature and heat transfer coefficient. The results are calculated below:

$$T(y) = ay + b \quad (2)$$

The heat flux can be defined as:

$$q = -\lambda_{brass} \cdot \frac{\partial T}{\partial y} = -\lambda_{brass} \cdot a \quad (3)$$

The surface sub-cooling ΔT is calculated as:

$$\Delta T = T_b - T_w = T_b - b \quad (4)$$

The heat transfer coefficient is given by:

$$h_{tot} = q / \Delta T = -\lambda_{brass} \cdot \frac{a}{T_b - b} \quad (5)$$

2.2.3 Condensate film thickness and interface temperature

The condensate film thickness δ_l and the interface temperature T_i were based on the mixed gas condensation model^[22,23] and the experimental heat flux q . The physical model for the numerical analysis is shown in Fig.5.

Fig.5 Physical model for the condensation of a vapor with a non-condensable gas

Both liquid film and vapor resistances to heat transmission have been accounted for by combining them in series to calculate the total heat flux from the bulk to the wall:

$$q = \lambda_l \frac{T_i - T_w}{\delta_l} = h_l (T_i - T_w) = (h_{conv} + h_{cond}) (T_b - T_i) \quad (6)$$

where T_b and T_w are the bulk and wall temperatures, respectively and h_l , h_{conv} and h_{cond} are the liquid film heat transfer coefficient, convection heat transfer coefficient and condensation heat transfer coefficient, respectively.

The film heat transfer coefficient is calculated by Nusselt equation:

$$h_l = \left[\frac{g \cdot \rho_l^2 \cdot r' \cdot \lambda_l^3}{4 \cdot \eta_l \cdot (T_i - T_w) \cdot x} \right]^{0.25} \quad (7)$$

where r' accounts for the condensate sub-cooling and the temperature jump across the film.

$$r' = r \left[1 + 0.68 \frac{c_{pl} \cdot (T_i - T_w)}{r} \right] \quad (8)$$

The film properties involved in the equations are evaluated at an intermediate temperature between that of the wall and the interface, so that the temperature jump across the condensate is accounted for:

$$T_l = T_w + 0.25(T_i - T_w) \quad (9)$$

From Eq. (7) the heat flux through the condensate film adjacent to a vertical wall can be written as:

$$q = \left[\frac{g \cdot \rho_l^2 \cdot r' \cdot \lambda_l^3}{4 \cdot \eta_l \cdot (T_i - T_w) \cdot x} \right]^{0.25} \cdot (T_i - T_w) \quad (10)$$

Consequently, the liquid film thickness is:

$$\delta_l = \lambda_l \cdot \frac{T_i - T_w}{q} = \left[\frac{4 \cdot \lambda_l \cdot \eta_l (T_i - T_w) \cdot x}{g \cdot \rho_l^2 \cdot r'} \right]^{0.25} \quad (11)$$

where x is the coordinate distance along the plate from the leading edge, with a value of 0.01m applied in the calculation process of this work.

The film thermal resistance is calculated as:

$$R_l = \frac{T_i - T_w}{q} = \frac{1}{h_l} \quad (12)$$

From Eq. (6) the gas resistance is calculated as:

$$R_v = \frac{1}{h_{conv} + h_{cond}} \quad (13)$$

As a result, the ratio of gas/liquid film resistance can be written as:

$$\frac{R_v}{R_l} = \frac{h_l}{h_{conv} + h_{cond}} = \frac{T_b - T_i}{T_i - T_w} \quad (14)$$

2.2.4 Analysis for the uncertainty of the measurement

The error in q is largely determined by the temperature gradient. In the temperature range of interest (0°C—100°C), the K-type thermocouples fitted the calibration data within ± 0.1 K. The distance between the points of temperature measurement has ± 0.1 mm accuracy. The uncertainty of the measurement for experimental results on this basis would too large for small temperature gradients. For the vapor mixture condensation experiment, to ensure a higher reliability of the experimental results, we only used data with a surface sub-cooling higher than 3K in the concentration range of 23.8% to 74.8%. The corresponding uncertainty for the heat flux and the heat transfer coefficient were 5.2%—16.6% and 5.2%—19.2%, respectively. Data higher than 5K were selected for a concentration range of 84.0% to 93.6%, with an uncertainty of 7.0%—21.0% for the heat flux and 7.1%—22.5% for the heat transfer coefficient.

2.2.5 Analysis for system reliability and repeatability

Experiments using pure steam were conducted with a steam velocity of 0.8m/s only. Referring to Fig.6, similarity was seen between the present heat transfer coefficient and the Nusselt equation indicating system reliability. For inlet CO₂ concentrations of 45% and 84%, experiments were performed twice to confirm repeatability. The lines in Fig.7 indicate that the reproducibility is good.

Fig.6 Reliability of the experimental system

Fig.7 Repeatability and reproducibility of the system

3. Experimental results and discussion

3.1 Condensate heat transfer characteristics

Figure 8 shows the dependence of the heat transfer coefficient and heat flux respectively, plotted against surface sub-cooling. In contrast with the pure steam case, it is seen from the figure that presence of the CO₂ gas has significant influence on the heat transfer. From the single curve, it can be found that for the CO₂ mass fraction in the range of 23.3% to 74.8%, the effect of surface sub-cooling is thought to be higher. The effect of surface sub-cooling on the heat transfer coefficient is relatively small when the CO₂ mass fraction is higher than 74.8%. As a result, the condensation characteristics are changeless, i.e. the surface sub-cooling has a lesser effect on it. With the CO₂ mass fraction of 93.6%, the heat transfer coefficient is about 400 W/(m²·K). Moreover, the calculated convective heat transfer coefficient of pure CO₂ under the same mixture velocity is only 19.3 W/m²·K—19.8W/m²·K.

Fig.8 Condensation characteristics curves under an average vapor velocity of 1.2m/s

Figure 9 shows that the liquid film thickness was basically maintained below 0.07 mm, meanwhile, the increase of the CO₂ concentration has little effect on the thickness of liquid film. For example, the performance of the liquid film thickness reduces less than half for a fourfold increase in CO₂ concentration. Furthermore, a linear relationship between thickness and surface sub-cooling was found when the surface sub-cooling is higher than 10K. However, for an increase in surface sub-cooling below 10K, the thickness increase is greater than linear.

Fig.9 Variation of liquid film thickness with surface sub-cooling

The variation characteristics of gas diffusion layer resistance and liquid film resistance are shown in Figure 10. For lower CO₂ concentration conditions, the

change of liquid film resistance is nonlinear. Since the two resistances work together and the ratio of the gas/liquid film resistance increases as surface sub-cooling increases, the heat transfer coefficient shows a sharper change. When the CO₂ concentration is greater than 84%, the gas diffusion layer resistance dominates at, almost forty times that of the liquid film resistance. In addition, the ratio of gas/liquid film resistance has a low surface sub-cooling dependence, which serves to explain the effect the surface sub-cooling has on the heat transfer coefficient, namely, the heat transfer coefficient decreases as the CO₂ concentration increases.

(a) Liquid film resistance

(b) Ratio of gas/liquid film resistance

Fig.10 Variation curves of the thermal resistance

Figure 11 shows the variation of the heat transfer coefficient plotted against the CO₂ concentration. With an increase in CO₂ concentration, the variation of the heat transfer coefficient is at first relatively high for a small CO₂ concentration, then begins to decrease slowly for higher CO₂ concentrations. This may be attributable to the fact that as the drop of interface vapor partial pressure increases, the CO₂ concentration increases, along with the variation of the interface temperature, at the same surface sub-cooling. It can also be seen from Fig.12, at a surface sub-cooling of 40K, the interface temperature decreases from 63.98°C to 57.79°C with the drop of 6.19°C, while the CO₂ concentration changes from 23.3% to 42.8%. Nevertheless, the

interface temperature decreases from 47.47°C to 32.19°C with the drop of 15.28°C under the variation of the CO₂ concentration from 62.7% to 84.0%. However, due to the variation of the wall temperature, the actual condensation driving force is relatively small compared to the increase of the CO₂ concentration. As shown in Fig.13, the condensation driving force ($T_i - T_w$) decreased by 2.91°C and 1.11°C in the above situations, respectively. Because the change of the liquid film resistance is relatively small, the condensation transfer coefficient can be represented using a concave curve.

Fig.11 Variation of the heat transfer coefficient with CO₂ mass fraction

Fig. 12 Gas-liquid interface temperature T_i

Fig. 13 Variation of the condensation driving force

3.2 Development of a new empirical correlation

Many researchers proposed semi-empirical correlations based on the experimental data and depended mainly on variables like non-condensable gas mass fraction, sub-cooling temperature and operating conditions. Comparisons with these correlations from Othmer[3], Meisenburg[4] and Dehbi[12] (presented in Table 1) are shown in Fig. 14. The relatively large deviation of more than 300% between the present data and those from Othmer may in part be due to difference in the kind of non-condensable gas, vapor velocity, and tube diameter. In the case of Meisenburg, the condensation is outside of a vertical pipe with an air volume concentration from 0.1% to 4%. The data calculated from Dehbi are significantly higher, 300%—1000%, due to the experimental conditions of a non-condensable gas mass fraction of 25%—90%, and a vapor mixture pressure of 1.5 atm—4.5 atm. Therefore, all above equations are not acceptable for steam condensation with high concentration CO₂.

Table 1 Description of existing correlations for condensation of vapor mixture

Authors	Equations
Othmer ^[3]	$\lg h_{tot} = (2.748 - 0.00471T_b) \cdot \lg \Delta T + (\log \Delta T / 3.439 - 0.926) \cdot [1.13 - 0.015T_b + \lg(100X + 0.505)]$ $T_b = 373 \sim 383 \text{ K}; X = 0 \sim 0.05$
Meisenburg ^[4]	$h_{tot} = 0.670 \cdot (\lambda^3 \rho^2 g r / L \eta \Delta T)^{0.25} (1/W)^{0.11}, W = 0.10 \sim 4.0$
Dehbi ^[12]	$h_{tot} = L^{0.05} [(3.7 + 28.7P) - (2438 + 458.3P) \log W] / (T_b - T_w)^{0.25}$ $(0.3 \text{ m} < L < 3.5 \text{ m}, 1.5 \text{ atm} < P < 4.5 \text{ atm}, 10^\circ \text{C} < T_b - T_w < 50^\circ \text{C})$

(a) Comparison with Othmer[3]

(b) Comparison with Meisenburg[4]

(c) Comparison with Dehbi[12]

Fig.14 Comparison of experimental data with values calculated from references

The heat transfer coefficient correlation for a CO₂-steam mixture, therefore, expressed in the form of the Nusselt pure steam condensation expression is given by:

$$\frac{h_{tot}}{h_p} = \frac{1.9864 - 0.4098 \ln(C + 16.2724)}{\Delta T^{0.18}} \quad (15)$$

where h_p is Nusselt theoretical solution for pure quiescent steam condensation **on a vertical plate, expressed in function of the surface sub-cooling**.

As seen in Fig.15, the new correlation fits the experimental data well, and its uncertainty is less than $\pm 20\%$. The ratio of h_{tot} and h_p decreases gradually as the surface sub-cooling increase. Especially, the heat transfer coefficient is inversely proportional to $\Delta T^{0.43}$, different from that of a pure steam condensation.

Fig.15 Comparison of experimental data with values calculated from correlation (15)

4. Conclusions

To provide new, accurate data for the condensation heat transfer of a steam-CO₂ mixture on a vertical plate, experiments were conducted for a wider range of CO₂ concentrations. The data show a large decrease in heat transfer performance with an increase of CO₂ concentration. The effect of surface sub-cooling is believed to be higher for CO₂ mass fraction in the range of 23.3% to 74.8%. The effect is found to be smaller when the CO₂ mass fraction is higher than 74.8%, and as a result the condensation characteristics change more abruptly. When the CO₂ mass fraction reaches 93.6%, the heat transfer coefficient is only 400W/(m²·K).

The liquid film thickness, liquid film resistance, ratio of gas/liquid film resistance, interface temperature, and condensation driving force were calculated on the basis of experimental measurements. The results show that the liquid film thickness is less than 0.07 mm. A linear relationship between thickness and surface sub-cooling was found when the surface sub-cooling is higher than 10K. However, the thickness increases sharply as the surface sub-cooling increases when the surface

sub-cooling is below 10K. When the CO₂ concentration is greater than 84%, the gas diffusion layer resistance is almost forty times that of the liquid film resistance. The interface temperature and condensation driving force change linearly with the surface sub-cooling. Thus it can be seen that conductive resistance of the liquid film for the vapor mixture condensation controlled by diffusion is relatively small. The condensation efficiency can't be significantly improved by just increasing the heat conductivity of the liquid film, but the most important is the improvement of the diffusion mass transfer process of vapor molecules. Therefore, the strengthening mechanism of vapor mixture condensation can come down to thinning the thickness of mass transfer boundary layer and promoting gas mixing in the mass transfer layer.

To compare with earlier experimental results, the present data are arranged in the form of Nusselt-type expression, with an uncertainty of less than $\pm 20\%$ for a condensation of the CO₂-steam mixture. The heat transfer coefficient is inversely proportional to $\Delta T^{0.43}$.

Nomenclature			
a	Constant in condensation surface temperature calculation	<i>Greek alphabet</i>	
		λ	Thermal conductivity, W/(m·K)
b	Constant in condensation surface temperature calculation	ρ	Density, kg/m ³
		η	Dynamic viscosity, Pa·s
C	CO ₂ mass fraction, %	ΔT	Surface sub-cooling, K
c_p	Specific heat	δ	Condensate film thickness, mm
g	Gravitational acceleration, m/s ²		
h	Heat transfer coefficient, W/(m ² ·K)	<i>Subscripts</i>	
L	Characteristic length, m	b	Bulk
P	Pressure, atm	cond	Condensation
q	Heat flux, kW/m ²	conv	Convection
r	Latent heat of condensation, J/kg	i	Vapor-liquid interface
R	Thermal resistance, m ² ·K/kW	j	Temperature measuring point
T	Temperature, K	l	Condensate
W	Mass fraction of non-condensable	tot	Total
X	Mole fraction of non-condensable	v	Vapor phase
x	Coordinate distance along the plate from the leading edge	w	Wall surface

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