

Experimental Study of Condensation Heat Transfer in the Presence of Noncondensable Gas on the Vertical Tube

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ABSTRACT

After the Fukushima nuclear power plant accident, passive safety systems have drawn much attention as a strategy to mitigate the accident of nuclear reactors. Among them, the Passive Containment Cooling System (PCCS) removes the energy released to the containment building through the condensation heat transfer phenomenon in the event of the loss of coolant accident (LOCA) or the main steam line break (MSLB). The PCCS will be deployed to prevent overpressurization failure of the containment in an advanced PWR. The thermal performance of the PCCS is governed by the condensation heat transfer rate of the steam mixed with the noncondensable gas in the containment atmosphere. In this study, an experimental investigation of the steam-air condensation on an outer surface of a tube is conducted to obtain heat transfer data for the PCCS in which the heat exchanger assemblies consist of multiple vertical tubes. The condensation heat transfer coefficient is measured on a vertical tube of 40 mm in outer diameter and 1.0 m in length. In the experiments, nearly isothermal wall conditions are maintained and the stratification of gaseous mixture is also investigated. Experiments were carried out at the pressure ranging from 2 to 4 bar and the noncondensable gas mass fraction from 0.1 to 0.8. The measured heat transfer coefficients were compared with existing correlations. Based on the experimental results, a new empirical correlation is proposed in terms of Grashof number, Jacob number and Schmidt number.

KEYWORDS

Passive Containment Cooling System (PCCS), external condensation, vertical tube, noncondensable gas, empirical correlation

1. INTRODUCTION

The integrity of containment is lost during the Fukushima accident. As a result of the accident, a large amount of radioisotope released to environment. After the accident, an importance of containment which prevents radioisotope from releasing to environment has been more emphasized.

The Passive Containment Cooling System (PCCS) that will be introduced in the Korea next generation nuclear power plant removes released energy to containment through the condensation heat transfer phenomenon in the event of the loss of coolant accident (LOCA) or main steam line break (MSLB). The

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released steam becomes gas mixtures with air and it is condensed on the tube bundle's outer surface of the PCCS.

In order to conduct the thermal-hydraulic analysis of the PCCS heat exchanger, it is required to have experimental data and an analysis model for condensation heat transfer in the presence of noncondensable gas. If noncondensable gas is mixed with steam, they will accumulate with high fraction near a condenser tube degrade considerably the heat transfer rates. As result of those processes, condensation heat transfer rate dramatically decreases. A couple of experimental studies were focused on condensation phenomenon occurred outside of vertical tube.

Uchida et al. [1]'s and Tagami [2]'s experimental studies are well known to investigate the condensation phenomenon in the containment during the LOCA. The correlations on heat transfer coefficient proposed by Uchida and Tagami function of mass fraction of noncondensable gas only. Dehbi [3] conducted condensation experiment on the vertical tube evaluated the pressure behavior in the containment. Dehbi set up an experimental facility which has vertical tube of 3.5 m in height, 38 mm in outer diameter and measured to obtain the heat transfer coefficients at 1.5, 3.0, 4.5 atm. The correlation proposed by Dehbi takes account of noncondensable mass fraction, total pressure, and wall subcooling. Liu et al. [4] performed the single tube condensation experiment for the PCCS heat exchanger of the Internal Evaporator-Only (IEO) type. Liu measured wall temperature along axial direction to allow boiling of coolant. Kawakubo et al. [5] is obtained the total heat transfer coefficient from measured the gas mixture's temperature and the inlet and outlet of coolant temperature. The condensation heat transfer coefficient is conjectured through calculating the heat transfer coefficient of inside of condensing tube, indirectly. Kawakubo conducted a series of experiments to find effect of the wall subcooling, quantitatively.

In this study, an experimental study is performed to investigate the condensation heat transfer in the presence of noncondensable gas on vertical tube. Experiments were conducted using a tube with 1000 mm in height, 40 mm in outer diameter. The experimental data are obtained at pressure of 2, 3, 4 bar and the air mass fraction varied from 0.1 to 0.8. In order to measure the exquisite condensation heat transfer coefficients, no stratification in the chamber and nearly uniform tube's outer surface temperature maintained. The measured heat transfer coefficients were compared with existing models of Uchida, Tagami and Dehbi's correlations. Based on the experimental results, a new empirical correlation is proposed in terms of Grashof number, Jacob number and Schmidt number.

2. Experiment

2.1. Experimental apparatus

As described in Fig. 1, experimental loop consists of mainly two sections: condensation section and cooling section. The condensation section includes chamber which installed tube inside, steam generator, condensation water tank and recirculation pump. The cooling section has water storage tank and pump.

The diameter of chamber is 609 mm and the height is 1950 mm. A vertical tube with 40 mm in outer diameter, 3 mm in thickness and 1000 mm in length installed inside of the chamber. Both of them are made of SUS-304. The steam is generated submerging heaters which is maximum 120 kW ($30 \text{ kW} \times 4$) in the steam generator and it goes to chamber. The steam is mixed with air in the chamber and condensed on the tube. The condensate flows to the condensate tank and it is sent to steam generator using the recirculation pump to maintain the water inventory. Fig. 2 describes configuration of the experimental apparatus of this study.

Fig. 3 describes the location of thermocouples (K-type) in the tube and the chamber. In order to find distribution of gas mixtures in the chamber, two sets of thermocouples are installed inside the chamber.

Each set consists of 7 thermocouples arranged vertically along the tube height, and these two sets are installed with a polar angle of 90 degrees. For measurement of wall temperature on the tube, 6 thermocouples (K-type) are installed along the axial direction. The thermocouples on the tube wall are fixed with silver soldering.

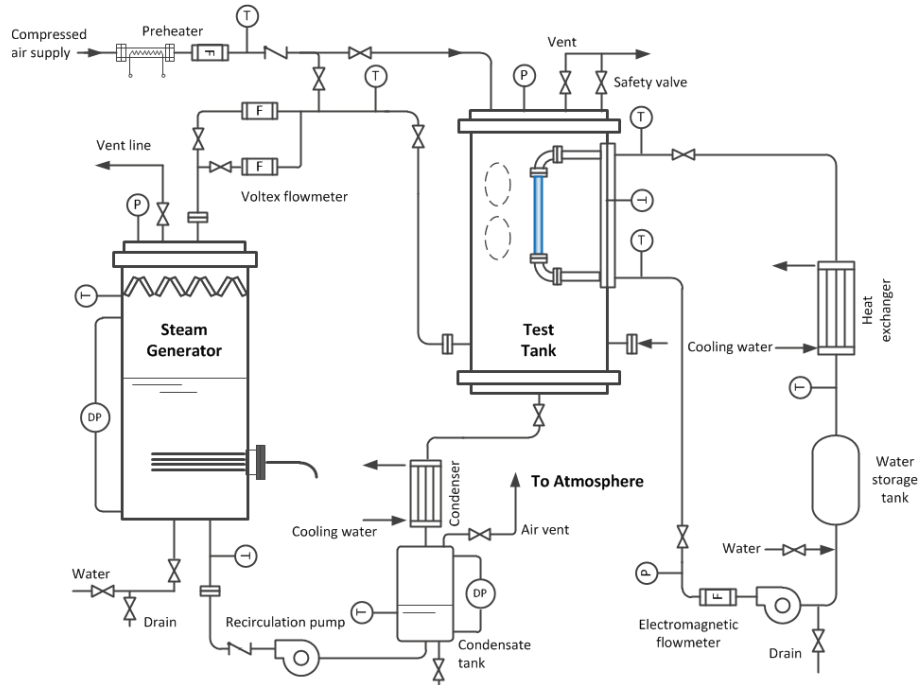


Figure 1. Condensation experimental facility.

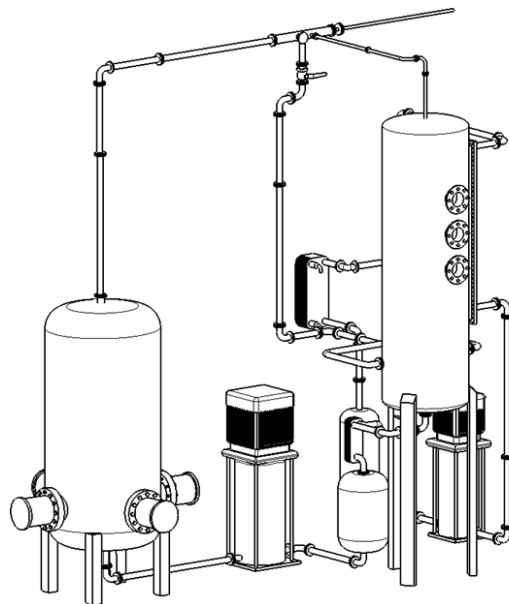


Figure 2. Configuration of experimental facility.

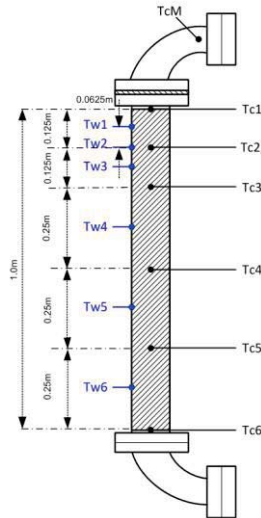


Figure 3. Scheme of temperature measurement in the chamber.

2.2. Experimental procedure

After filling water up in the steam generator for condensation section and the water storage tank for cooling section, the heaters are turned on for pressurization and generation of steam. The chamber was ventilated for degassing when temperature and pressure of systems approach a proper level. After the degassing, noncondensable gas is injected into the chamber and the system is pressurized to prescribed condition. If the pressure reaches to the setting level, steady state should be confirmed. Then, the temperature of the gas mixtures, tube's outer wall, inlet and outlet coolant are measured to obtain the heat transfer coefficient.

Table I represents test matrix of this study. Generally, the design pressure of the containment is about 4 bar. Therefore, heat transfer coefficient is measured at 2, 3, 4 bar to reflect the transient status of the containment in this experimental study. The air mass fraction is adjusted from 0.1 to 0.8 to investigate the effect of noncondensable gas on the condensation. The coolant flow rate is controlled in a range from 0.030 to 0.263 kg/s.

Table I. Test matrix

2 bar			3 bar			4 bar		
W_a	ΔT_w [°C]	\dot{m} [kg/s]	W_a	ΔT_w [°C]	\dot{m} [kg/s]	W_a	ΔT_w [°C]	\dot{m} [kg/s]
0.096	33.161	0.263	0.109	32.430	0.257	0.110	36.746	0.240
0.207	33.942	0.177	0.205	40.195	0.228	0.205	36.229	0.229
0.306	41.640	0.194	0.310	32.625	0.130	0.303	43.506	0.229
0.408	42.645	0.132	0.410	36.927	0.084	0.407	40.347	0.094
0.510	40.333	0.088	0.498	38.406	0.095	0.508	40.140	0.062
0.600	39.620	0.049	0.608	40.342	0.066	0.611	39.094	0.062
0.709	40.262	0.049	0.706	38.546	0.067	0.704	43.512	0.068
0.805	40.553	0.037	0.797	38.966	0.050	0.802	44.429	0.030

(W_a : air mass fraction, ΔT_w : wall subcooling)

2.3. Data reduction and Measurement uncertainties

To obtain the condensation heat transfer coefficient, it is necessary to get temperature information from tube inlet and outlet. The heat transfer rate can be calculated using the inlet and outlet temperature of the coolant in the tubs as:

$$Q = \dot{m}c_p(T_o - T_i) \quad (1)$$

where, \dot{m} , c_p , T_o and T_i represent the mass flow rate of coolant, the specific heat, the outlet and inlet temperature of coolant, respectively. Utilizing the Newton's cooling law, Eq. (1) can be used to express the average condensation heat transfer coefficient of the condensing surface as:

$$\bar{h} = \frac{\dot{m}c_p(T_o - T_i)}{A(T_b - T_w)} \quad (2)$$

where, A , T_b and T_w represent the total heat transfer area, the temperature of steam and noncondensable mixtures and the temperature of outer surface of tube, respectively.

In order to quantify the measurement uncertainties, uncertainty analysis is conducted about the total heat transfer rates and the average heat transfer coefficients. The bias uncertainty of measurement instruments are summarized in Table. II and uncertainty of condensation heat transfer coefficient is calculated as following:

$$U_{\bar{h}}^2 = \left(\frac{\partial \bar{h}}{\partial \dot{m}} U_{\dot{m}} + \frac{\partial \bar{h}}{\partial \Delta T_c} U_{\Delta T_c} + \frac{\partial \bar{h}}{\partial \Delta T_w} U_{\Delta T_w} \right) \quad (3)$$

where, ΔT_c and ΔT_w represent inlet and outlet temperature difference of coolant and wall subcooling, respectively. Accordingly, relative uncertainty of condensation heat transfer coefficient is calculated as following:

$$\left(\frac{U_{\bar{h}}^2}{\bar{h}} \right)^2 = \left(\frac{U_{\dot{m}}}{\dot{m}} \right)^2 + \left(\frac{U_{\Delta T_c}}{\Delta T_c} \right)^2 + \left(\frac{U_{\Delta T_w}}{\Delta T_w} \right)^2 \quad (4)$$

where, total uncertainties of ΔT_c and ΔT_w are defined as following:

$$(U_{\Delta T_c})^2 = (U_{T_{c,o}})^2 + (U_{T_{c,i}})^2 \quad (5)$$

$$(U_{\Delta T_w})^2 = (U_{T_b})^2 + (U_{T_w})^2 \quad (6)$$

Average uncertainty of the heat transfer coefficient is 12 % from analysis results. Most of it comes from coolant mass flow measurement uncertainty. The measurement uncertainty of the wall subcooling is found to be relatively negligible.

Table II. Bias error of measurement instruments

Parameter	Measurement instrument	Bias error
Fluid Temperature	Thermocouple (K-type)	0.2 °C
Wall Temperature	Thermocouple (K-type)	1.1 °C
Coolant flow rate	Electrical flow meter	0.1 % of span
Chamber pressure	Pressure transmitter	0.075 % of span

3. Result and Discussion

3.1. Condensation phenomena

Condensation is a process in which the removal of heat from a system causes a vapor to convert into liquid [6]. Presence of a small amount of noncondensable could affect decreasing the condensation heat transfer coefficient because the steam is changed into liquid film due to condensate on the tube wall and the noncondensable gas is accumulated near the liquid film. This accumulated layer plays a role of barrier on the diffusivity of the steam.

Fig. 4 shows that concentration distribution of gas mixture in the presence of noncondensable gas on the tube. This boundary layer interrupts the heat transfer between the steam and the tube. Therefore, the condensation heat transfer rate is reduced by noncondensable gas. Fig. 5 shows the condensate on the outer surface of the tube using CCD camera. Filmwise and dropwise condensation is observed on the surface. The filmwise condensation is observed condensate forms a continuous liquid film on the surface of the tube and the dropwise condensation is observed condensate forms droplet with variety diameter. Other phenomena with respect to condensation are suction effect, film waviness and mist formation. The suction effect is condensation surface pulling the steam to convert the phase. The film waviness is the enhancement of heat transfer. It caused by the ripple shape on the condensate surface. Mist formation is observed experimentally. Enhancement of its heat transfer is not so much but it may be the one of the effect to increase condensation phenomena.

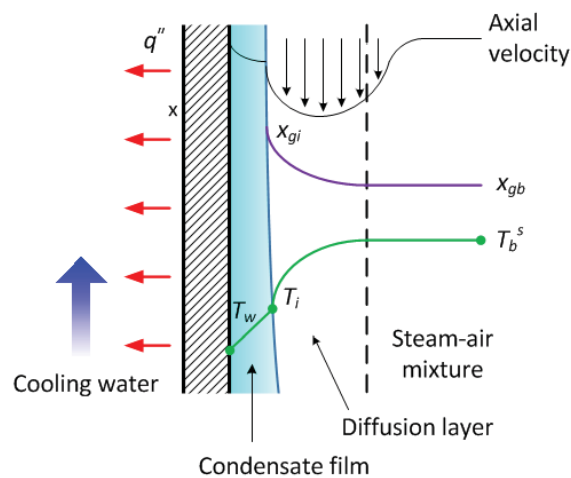


Figure 4. Schematic diagram of distribution of gas mixture in the presence of noncondensable gas on the tube.

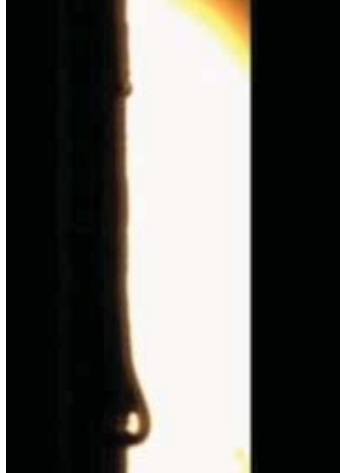


Figure 5. Phenomenon of condensation on the outer surface of tube.

3.2. Condensation heat transfer coefficient

Fig. 6 shows temperature distribution during the experiment at 4 bar. There is no stratification (Temperature difference between max. and min. is 0.9 K). The tube wall temperatures are well controlled because temperature difference is 4.11 K. The condensate affected the wall temperature distribution when the condensate flows along the tube's outer surface. Fig. 7 shows the air mass fraction distribution along the height at each pressure. The result shows the air mass fraction also maintained well during the experiment. It means the behavior of the noncondensable and steam in the chamber is mixed well caused by natural convection circulation.

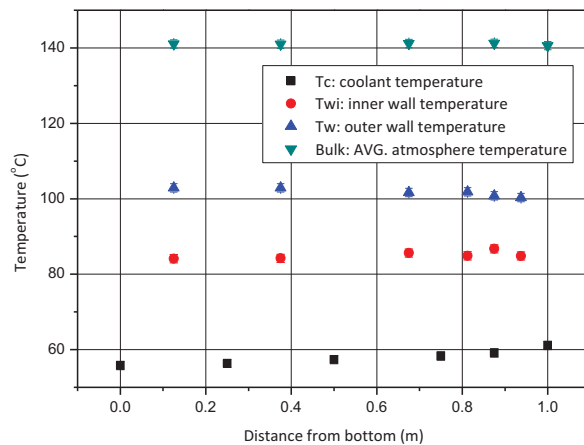


Figure 6. Temperature distribution air mass fraction 0.1 at 4 bar.

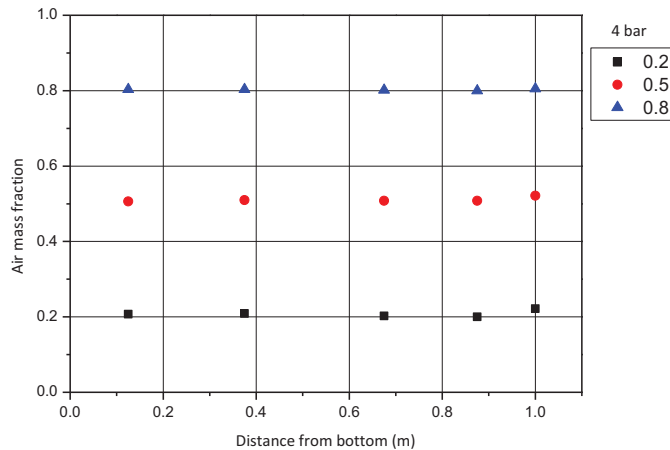


Figure 7. Comparison between bulk temperature distribution in the air mass fraction 0.2, 0.5, 0.8 at 4 bar.

Fig. 8 shows condensation heat transfer rate depends on the air mass fraction at each pressure. Fig. 9 shows that the condensation heat transfer coefficients measured at 2 to 4 bar. The condensation heat transfer coefficient dramatically reduces with an increase of the noncondensable mass fraction because noncondensable disturb the condensation phenomena. Through experiment results, it is confirmed that if gas mixtures density is high, then the condensation heat transfer coefficient has a decrease tendency. The physical distribution of the steam and the noncondensable gas determines the condensation heat transfer performance.

The experimental results show that the heat transfer coefficients increase with increasing pressure. It can be explained by Kang et al. [6]. The Kang's results showed that increasing gas pressure induces increment of the gas density. When gas density increases, the heat transfer coefficient increases due to increase of contact efficiency between gas particles and condensing tube.

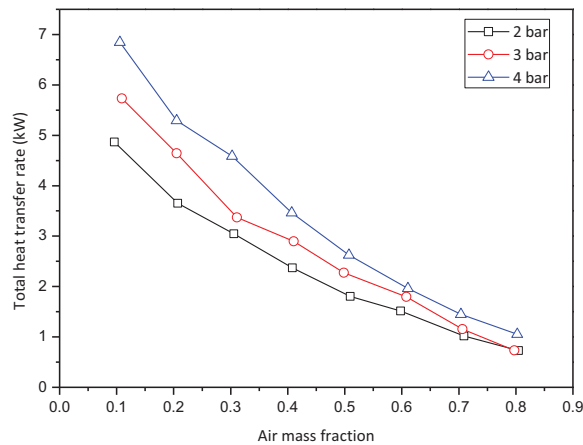


Figure 8. Condensation heat transfer rate at different pressure condition.

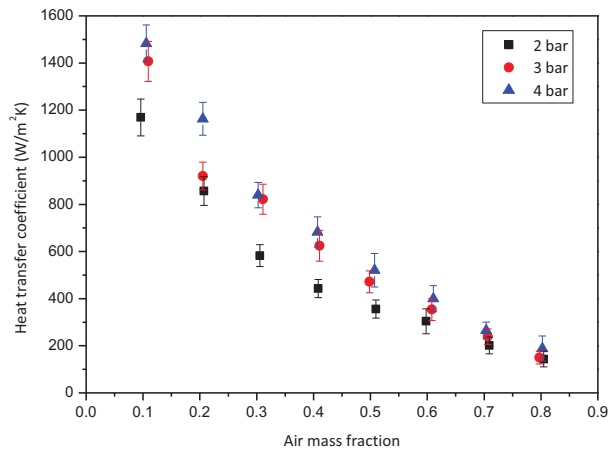


Figure 9. Condensation heat transfer coefficient at different pressure condition.

3.3. Comparison with existing correlation

The condensation heat transfer coefficients from the experiment are compared with existing correlations proposed by Dehbi, Uchida and Tagami. Figs. 10 to 12 show the comparison results between the experiment data and the predictions from correlations. Correlations of Uchida, Tagami and Dehbi are as following:

$$\bar{h}_U = 379 \left(\frac{W_a}{1 - W_a} \right)^{-0.707} \quad (7)$$

$$\bar{h}_T = 11.4 + 284 \left(\frac{1 - W_a}{W_a} \right) \quad (8)$$

$$\bar{h}_D = \frac{L^{0.05} [(3.7 + 28.7P) - (2438 + 458.3P) \log W_a]}{(T_b - T_w)} \quad (9)$$

From the comparison results, experimental results have a slight difference with Uchida's and Tagami's correlations. These correlations take account of noncondensable mass fraction, only and their prediction of the condensation heat transfer coefficient is known to be conservative. Generally, these correlations estimate conservative heat transfer coefficients. The Dehbi's correlation takes account of noncondensable mass fraction, total pressure and wall subcooling. In that respect, experimental results are similar with prediction of the Dehbi's correlation. But still the experimental results underestimate the Dehbi's heat transfer coefficients. The difference of the condensation heat transfer coefficients may affect the velocity of the gas mixtures circulation, primarily. The Dehbi's test chamber is narrower than chamber which is utilizing in this study. The diameter difference leads to increase the natural circulation velocity. Dehbi's experiment did not maintain the uniform gas mixtures state inside chamber and has significant wall temperature gradient along the tube height. Also, Dehbi applied the same assumption of the dependency on the wall subcooling as Nusselt [7]'s theory, but it is applicable only to pure steam state.

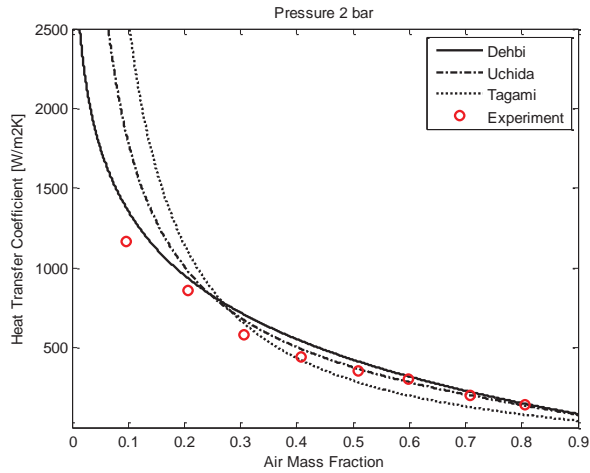


Figure 10. Comparison result at 2 bar.

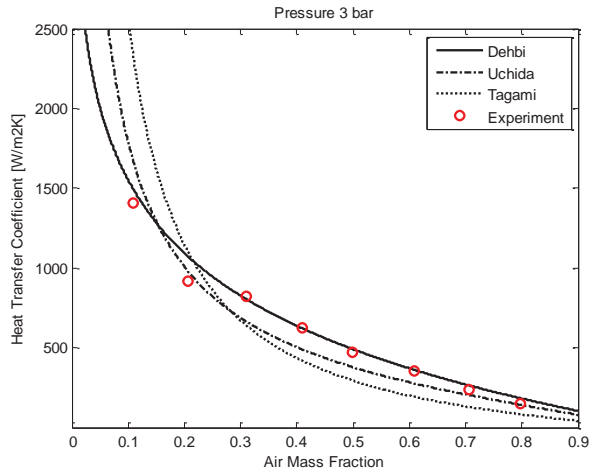


Figure 11. Comparison result at 3 bar.

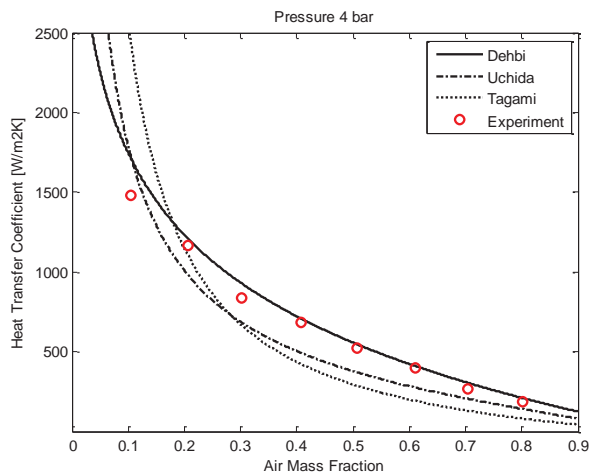


Figure 12. Comparison result at 4 bar.

3.4. New empirical correlation

The existing correlations are obtained by fitting the experimental results. It means correlations do not reflect the condensation physical phenomena, sufficiently. Even though, these correlations predict the condensation heat transfer coefficients, well. Therefore, the new empirical correlation proposed to reflects the condensation physical phenomena using dimensionless number such as Grashof number, Jacob number and Schmidt number.

The proposed correlation could derived from Minkowycz [11]'s governing equations. Eqs. (10) ~ (12) represent governing equations of momentum, energy, and concentration by Minkowycz.

$$\rho \left(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} \right) = g(\rho - \rho_\infty) + \frac{\partial}{\partial y} \left(u \frac{\partial u}{\partial y} \right) \quad (10)$$

$$\rho c_p \left(u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} \right) + (c_{pg} - c_{pv}) j_g \frac{\partial T}{\partial y} = \frac{\partial}{\partial y} \left(k \frac{\partial T}{\partial y} - \frac{\alpha R T M^2}{M_g M_v} j_g \right) \quad (11)$$

$$\rho \left(u \frac{\partial W_{nc}}{\partial x} + v \frac{\partial W_{nc}}{\partial y} \right) = \rho \frac{\partial}{\partial y} \left(D \frac{\partial W_{nc}}{\partial y} \right) \quad (12)$$

Each parameter is conducted normalization to obtain the dimensionless number from the governing equations. It represents the Table III.

Table III. Normalization of governing equations

Parameter	Normalization	Parameter	Normalization
x^*	$\frac{x}{L}$	T^*	$\frac{T_\infty - T}{T_\infty - T_w}$
y^*	$\frac{y}{L}$	h^*	$\frac{h_\infty - h}{h_{fg}}$
v^*	$\frac{v}{V}$	W_{nc}^*	$\frac{W_{nc,\infty} - W_{nc}}{W_{nc,\infty} - W_{nc,w}}$

After the normalization, governing equations were modified as follow. Eqs. (13) ~ (15) represent modified momentum, modified energy and modified concentration equation, respectively.

$$u^* \frac{\partial u^*}{\partial x^*} + v^* \frac{\partial u^*}{\partial y^*} = \frac{Gr}{Re_L^2} T^* + \frac{1}{Re_L} \frac{\partial^2 u^*}{\partial y^{*2}} \quad (13)$$

$$u^* \frac{\partial h^*}{\partial x^*} + v^* \frac{\partial h^*}{\partial y^*} = \frac{Ja}{Re_L Pr} \frac{\partial^2 T^*}{\partial y^{*2}} \quad (14)$$

$$u^* \frac{\partial W_{nc}^*}{\partial x^*} + v^* \frac{\partial W_{nc}^*}{\partial y^*} = \frac{1}{Re_L Sc} \frac{\partial^2 W_{nc}^*}{\partial y^{*2}} \quad (15)$$

The Grashof number, Jacob number and Schmidt number were obtained from Eqs. (13) to (15), respectively. Finally, new empirical correlation is proposed using those dimensionless numbers such as $Nu_D = aGr_D^b Sc^c Ja^d$. Formulation of the proposed correlation is using the Nusselt number which is heat transfer dimensionless number. Based on the experimental results, proposed correlation is as follow.

$$Nu_D = 1.59 \times 10^{-5} Gr_D^{0.291} Sc^{-19.0} Ja^{-0.744} \quad (16)$$

To verify the proposed model, it compared with the experimental result and Dehbi's data. Fig. 13 shows comparison between proposed correlation and experimental results. Overall, it shows a good agreement even though the low Nusselt numbers have little difference. These three points of low Nusselt number have low coolant flow rate (0.030 ~ 0.049 kg/s). It caused by coolant flow rate uncertainty. Based on the uncertainty analysis, the low coolant flow rate had maximum uncertainty. It may propagate Nusselt number. Fig. 14 shows the comparison result between proposed correlation and Dehbi's data. It shows a good agreement.

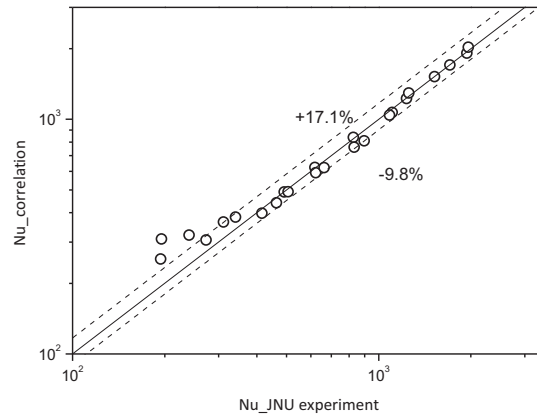


Figure 13. Comparison between proposed correlation and experimental results.

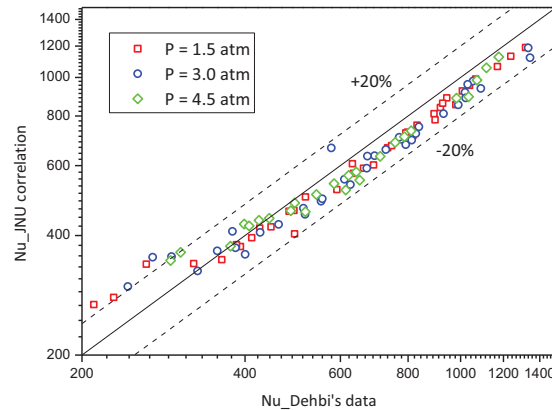


Figure 14. Comparison between proposed correlation and Dehbi's data.

4. CONCLUSIONS

In a vertical tube with 40 mm in O.D. and 1000 mm in height, three sets of experiments to measure the condensation heat transfer coefficient are performed at 2, 3, 4 bar. Experimental results show that the condensation heat transfer coefficient reduces with an increase of the noncondensable mass fraction. The results are compared with the prediction of existing correlations by Uchida, Tagami and Dehbi. The compared result shows that the experimental results underestimate these correlations. That's because Uchida's and Tagami's correlations do not reflect all the effects of primary physical parameters, Dehbi's experiment has the geometrical effect and did not maintain the uniform gas mixtures state inside chamber and has significant wall temperature gradient along the tube height. Based on the experimental results, a new empirical correlation is proposed in terms of Grashof number, Jacob number and Schmidt number. To verify the proposed correlation, experimental data and Dehbi's data was compared. The compared results are reasonably acceptance except the low Nusselt number ranges.

As further works, curvature effect will be a next step. In addition, the heat transfer enhancement by using a finned tube is to be experimentally investigated.

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