A preliminary study on the condensation in the presence of air with various subcooling degrees

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1. Introduction

After the Fukushima Daiichi accident, the concept of the Passive Containment Cooling System (PCCS) was introduced to enhance the safety of Nuclear Power Plants (NPPs) in Station Black Out (SBO) conditions. The PCCS consists of a series of heat exchangers, natural circulation loops, valves, and ultimate heat sink tanks, which is shown in Fig. 1.

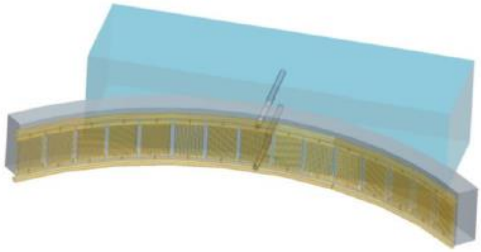

Fig. 1. Schematic of the Passive Containment Cooling System

Several studies were conducted on the condensation heat transfer on various thermal-hydraulic conditions. In particular, many studies focused on how the wall subcooling affects the condensation heat transfer. Dehbi [1], Kawakubo [2], Su [3, 4], Lee [5], and Liu [6] experimentally researches the subcooling effect on condensation heat transfer. Dehbi, Su and lee reported the condensation heat transfer would decrease as the wall subcooling increases. However, the sensitivity of condensation heat transfer on the wall subcooling was evaluated differently. Kawakubo reported that the effect of the wall subcooling on the condensation heat transfer inflected near 15K. Liu reported that the condensation heat transfer coefficient increases as the wall subcooling increases, which contradicts other researches. However, previous studies except Kawakubo and Liu did not measure under wall subcooling 5K conditions. Also, at the low subcooling conditions verify the inflection point is an important part of the subcooling effect. For these reasons we studied the various subcooling degrees to analyze the condensation heat transfer.

In this paper, we experimentally investigate the effect of the wall subcooling on condensation heat transfer. As the results of the experimental data, the condensation heat transfer tended to decrease with increasing subcooling.

2. Experimental system and methods

2.1 Experimental facility

The schematic diagram of the experimental facility used for analysis is shown in Fig. 2. The facility consists of a pressure vessel and a vertical tube. The pressure vessel 2500mm in an axial direction and 447.2mm in a radial direction. The outer diameter is 38.1mm, the thickness is 1.2mm. The total length of the Tubes is 1.1m.

K-type thermocouples were used to measure the temperature. The location of the installed thermocouples on vertical tubes is shown in Fig. 3. Total of 28 thermocouples were installed to measure the wall temperature and the coolant temperature. Two bulk thermos couples were installed at the pressure vessel to measure the bulk temperature. Three wall thermocouples were installed at 120 degrees in the azimuthal direction to measure the wall temperature.

Two pressure gauge was used to measure the pressure vessel and the coolant side. Electromagnetic flowmeter was used to measure the flow rate of the coolant.

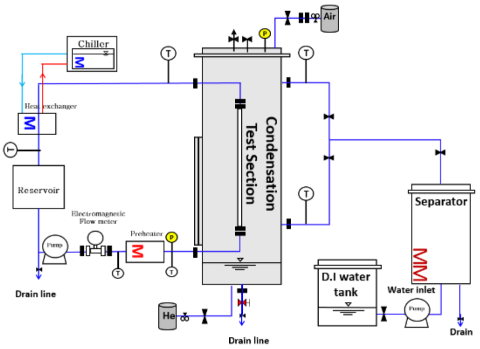

Fig. 2. Schematic diagram of the experimental facility

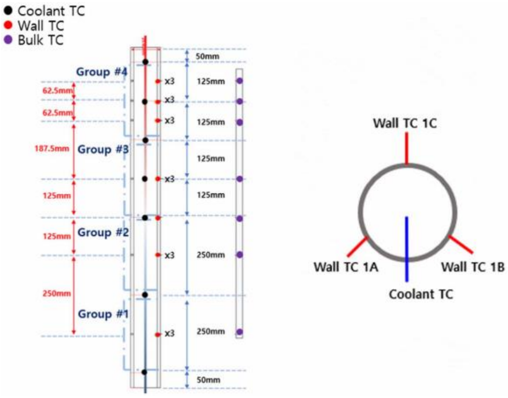

Fig. 3 Condenser tube and measurement point

2.2 Data Reduction

Table I: Experimental Conditions

	Range
Pressure(bar)	2~3bar
Wall subcooling(K)	5~45K
Air mass fraction(-)	0.30,0.56,0.75

Table I shows the experimental conditions considered for the investigation. The temperature difference of the coolant was constant as 4~6K so that the error can be minimized. All the measurements were conducted once in a 5 seconds, and a total of 1000 steps were measured in the steady-state.

The energy balance at the tube can be expressed as;

$$Q = \dot{m}C_p(T_{c,o} - T_{c,i}) = \mathrm{h}2\pi\mathrm{rL}(T_b - T_w) \qquad (1)$$

where $\dot{m}, T_{c,o}, T_{c,i}, T_b$ and T_w represent the coolant mass flow rate, coolant outlet temperature, coolant inlet temperature, bulk temperature and wall temperature. The average heat transfer coefficient is obtained by the arithmetic average of the local heat transfer coefficient of each group. Accordingly, the most important condition in analyzing the average heat transfer coefficients should be that the wall temperature increases or decreases linearly. A bulk temperature gradient should not occur depending on height. If the Bulk Temperature changes on height mean the air mass fraction changes as the axial directions.

Fig. 4, in the case of the measured wall temperature, a linear graph showing a constant increase for each height was shown. Therefore, it is feasible to use Eq. (2) to analyze the average heat transfer coefficient throughout the experiment.

$$\bar{h} = \frac{\dot{m}C_p(T_{c,o}-T_{c,i})}{2\pi\mathrm{rL}(\overline{T_b}-\overline{T_w})} \qquad (2)$$

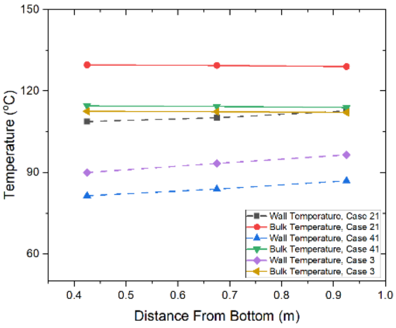

Fig. 4. The influence of Air on mixture temperature

3. Experimental results

3.1 The influence of the wall subcooling effect

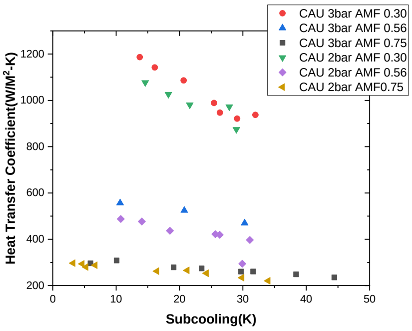

Fig. 5. Effect of wall subcooling on the heat transfer coefficient

Fig. 5 shows the variation of the condensation heat transfer coefficient according to the wall subcooling. Experiment results show that the condensation heat transfer coefficient tends to increase as wall subcooling decreases. On the other hand, inflection point around, which is 15K mentioned by Kawakubo was not observed in our test results. The experiments also showed that the heat transfer coefficient increased as pressure increased for 2bar to 3bar.

3.2 The influence of the Air mass fraction effect

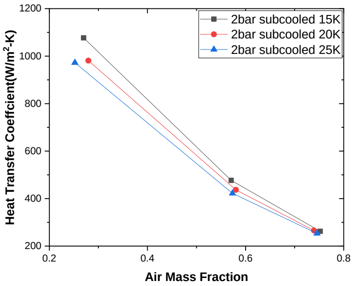

Fig. 6. Effect of air mass fraction on heat transfer coefficient

Fig. 6 shows the variation of the condensation heat transfer coefficient according to the air mass fraction difference. It shows that the condensation heat transfer coefficients tend to decrease as air mass fraction increases.

4. Results and discussions

4.1 Comparison result by Dehbi's correlation

$$h = \frac{L^{0.05}[(3.7+27.8P)-(2438+458.3P)\log(W_a)]}{(T_b-T_w)^{0.25}} \qquad (3)$$

Eq. (3) shows the Dehbi's correlation. To validate the experimental result with precedent researches, we compared the experiment result with other different correlations. Note that, the correlation of Dehbi is multiplied by 1.25, which is a curvature correction proposed by Dehbi (1991) for 38mm outer diameter tube.

Which showed in Fig. 7 an average 20% deviation with the experiment. However, as shown in Fig.8, a single consent subcooling index cannot properly represent the subcooling effect. In particular, at a low wall subcooling region the effect of subcooling on condensation is somewhat different from that in the high subcooling region. Note that, there indices, -0.25, -0.35 and -0.6 which can be found on precedent researches of Dehbi and Su were used for analysis.

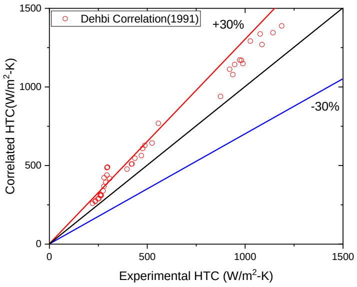

Fig. 7. Comparison of Dehhi correlation against experimental data

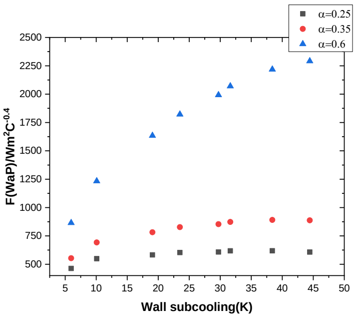

Fig. 8. Experimental results compare with subcooling index

It is known that the condensation heat transfer coefficient has a form of;

$$h = (T_b - T_w)^{\alpha} F(W_a, P) \qquad (3)$$

Where α, W_a, P represent the subcooling index, the air mass fraction and the pressure.

Until now, the subcooling index is considered as one therefore, a subcooling index function. Which can represent subcooling effect in all range of subcooling degrees, should be developed.

5. Conclusions

In this study, the condensation heat transfer phenomenon for a wide range of subcooling. As a result of the analysis, a decrease in the heat transfer coefficient was observed as the degree of subcooling increased, and which is considered to be due to the accumulation of non-condensable gas near the condensing tube.

The results of this experiment were consistent with the results of Dehbi and Su but contradict the results of Liu and Kawakubo.

Until now, the subcooling index was expressed as a constant value. The result of our test indicates that a single constant subcooling index cannot represent subcooling effect in all ranges. Thus, a function investigated on subcooling effect should be conducted.

The results of this experiment could be used for the PCCS for nuclear power plants.

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