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A simulation of condensation heat transfer coefficient of a vertical single tube in down-scaled experiment model using GOTHIC code

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1. Introduction

After Fukushima nuclear accident, importance for passive cooling system which operates without any AC power is rising. There are several overseas nuclear power plants such as AP1000, VVER1200, etc. which adopts a passive cooling system. iPOWER which has been developed in domestic is planning to adopt passive containment cooling system (PCCS). When DBA comes, PCCS releases energy to atmosphere through the heat exchangers (Fig.1), and the containment can be maintained in a safe condition at least 72 hours.

PCCS heat exchanger, which consists of a number of tubes and upper/lower headers, mostly removes energy through the tubes, so the condensation heat transfer phenomenon through the tube is important. As summarized in table I, many researchers measured thermal performance mainly represented by heat transfer coefficient (HTC) on a vertical single tube by experiments[1]~[5]. However, there are large deviations between the experiment results because experimental equipment specifications such as tube diameter, size of a pressure vessel and the range of wall-subcooling are somewhat different. Thus, KHNP designed a single tube experiment equipment which is considered prototype design of PCCS and performed the tests in an expected operating conditions.

Meanwhile, Ha et al.[6] conducted numerical analysis to determine optimal design of PCCS using GOTHIC code. Heat transfer models (Uchida, DLM-FM) were used for conducting numerical analysis. Diffusion Layer Model with Film roughening and Mist formation (DLM-FM) has been used widely in the condensation problem with non-condensible gas. In this paper, a numerical analysis is performed to validate the DLM-FM model using KHNP's test data.

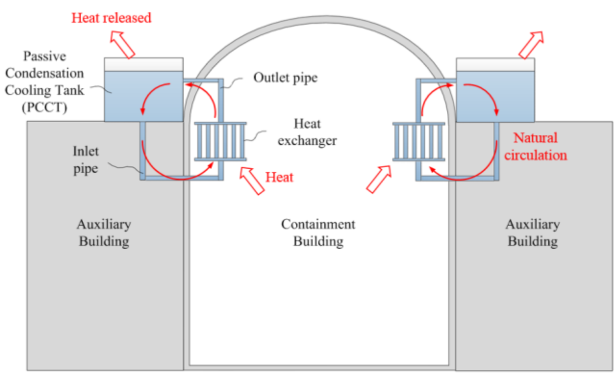


Fig. 1. A concept of PCCS

Table I. Specification of test apparatus on several studies

	Tube outer diameter	Height of pressure vessel	Shape of pressure vessel
Dehbi[1]	38.0 mm	5.0 m	Cylinder type
Kawakubo[2]	10.0 mm (I.D.)	2.7 m	
Su[3]	38.0 mm	3.5 m	
Y. G. Lee[4]	40.0 mm	2.0 m	
J. Lee[5]	31.8 mm	3.9 m	Dome type

2. Experimental Apparatus

2.1 Description of test facility

A schematic diagram of experiment facility is shown in Fig 2. It is consist of a pressure vessel, a steam supply system, a pure water generation system, a pressure tank and a single tube, etc. The scale ratios of a pressure vessel compared to the actual containment is one-twentieth in length scale and one-eight thousandth in volume scale. The length of a tube is one meter. The

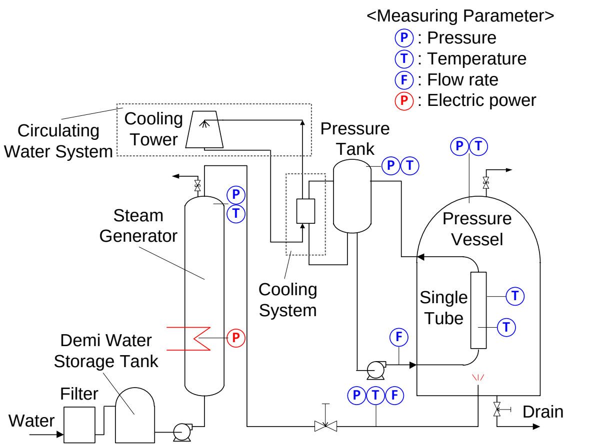


Fig. 2. Schematic view of the experimental apparatus

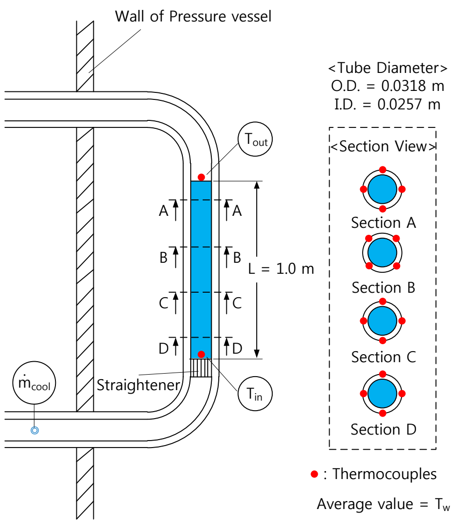
Table II. Details of the experimental apparatus

	Item	Unit	Value
Pressure vessel	Design pressure	bar	7.0
	Material	-	Carbon steel (SS304-coated)
	width / length / height	m	2.286 / 1.349 / 3.830
Single tube	Length	m	1.0
	Diameter (outer/inner)	m	0.0318 / 0.0257
	Material	-	Stainless steel

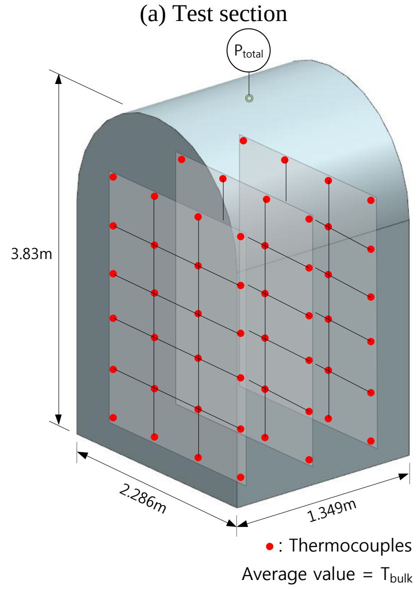
outer diameter and the inner diameter are 0.0318 meter and 0.0257 meter, respectively. Steam is injected into the pressure vessel by sparging method to make a steam flow uniform. A flow control valve is installed on the steam supply loop. Forced circulating flow condition inside a circulation loop is maintained by a pump. Heat absorbed in the circulating loop passes through the heat exchanger and is removed into the circulating water system. Finally, heat is released to atmosphere. Specifications of main experimental equipment are indicated in table II.

2.2 Measurements and test matrix

Major measurement variables are pressure, temperature and mass flow rate inside of the tube. The locations of measuring points are shown in Fig. 3. Total pressure was measured on the top of the pressure vessel outside and flow rate of cooling water was measured at the entrance of the tube. Temperatures on the wall, in



(a) Test section



(b) Locations of thermocouples inside of the pressure vessel

Fig. 3. Details about main test apparatus

Table III. Test matrix

Condition	Value
Air mass fraction (W_a)	0.2 ~ 0.8
Wall-subcooling ($\Delta T_{w,sub}$)	5℃ ~ 60℃
Total pressure (P)	2 bar

the single tube and the pressure vessel were measured locally, as indicated in the figures. Space-averaged value of all local values were used as a representative and the values were used to calculate heat transfer coefficient.

Table III indicates the test matrix. Tests were conducted in accordance with increasing wall-subcooling. After that, the same test group were carried out in various air mass fraction from 0.2 to 0.8. Wall-subcooling, which means the difference between tube wall temperature and bulk temperature in the pressure vessel, was changed by controlling the flow rate of coolant inside of the single tube. Also, total pressure in the vessel was controlled by flow control valve which regulates the amount of steam injection rate by the method of PID (Proportional Integral Differential) control.

3. Experiment result

3.1 Thermal performance

Heat transfer coefficient represents thermal performance in this study. It is calculated by following equations.

$$HTC = \frac{\dot{m}_c \times c_p \times (T_{out} - T_{in})}{A_s \times \Delta T_{w,sub}} \qquad \text{Eq. (1)}$$

$$(\text{where } \Delta T_{w,sub} = T_{bulk} - T_w)$$

Each variable used in Eq. (1) is a time-average value during 10 minutes in the steady state condition. As shown in Fig. 3, wall temperature (T_w) and bulk temperature (T_{bulk}) are average values of 16 and 72 local points.

3.2 Effect of air mass fraction and wall-subcooling

Experimental results are indicated in fig. 4. As shown in Fig. 4, HTC decreases as air mass fraction (W_a) increases. It is explained that the increasing air concentration in the pressure vessel reduces heat transfer coefficient through the tube wall. That is, air obstructs a condensation phenomenon.

Meanwhile, HTC decreases as increasing the wall-subcooling in the same condition of air mass fraction. In high wall-subcooling case, condensation phenomenon is vigorous, and the air concentration in the boundary layer become higher than the opposite case. Hence, HTC becomes lower in the high wall-subcooling case than the low wall-subcooling case.

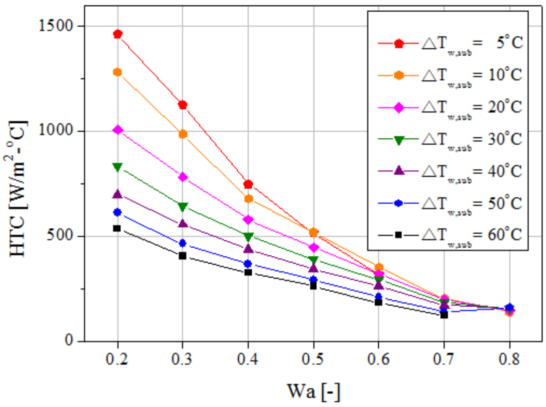


Fig. 4. Test data for HTCs in accordance with air mass fraction and wall-subcooling

As indicated in Fig. 4, the effect of wall-subcooling is weakened gradually as air mass fraction increases. Finally, almost same value of HTC was measured in 0.8 air mass fraction case. In the condition of very high air mass fraction, it is known that a small quantity of heat is transferred through the tube wall because air occupies large space in the pressure vessel.

4. Numerical analysis

4.1 Nodalization and analysis method

The containment vessel and single tube test section are modeled in the GOTHIC analysis. The containment is modeled as a lumped volume, and the outer surface of the single tube is modeled. Only test sections and main boundaries composes a calculation model to simplify the experimental system. The model is constructed by control volumes, flow paths, boundary conditions and a thermal conductor, as shown in Fig. 5. Each options are selected as follows because the code calculation is on a basic stage in this study.

<Assumptions>

A. Tube wall temperature is the same as the experiment result.

: to simplify the phenomenon

B. Liquid phase is excluded.

: amount of condensate rate is very small and liquid convection heat transfer takes little portion in the system.

<The calculation options>

A. Initial conditions are set to the steady state conditions in the experiment result.

B. DLM-FM is used as a condensation heat transfer model.

C. Steam injection rate is controlled by PI (Proportional Integral) control to make a pressure 2 bar which is a target condition.

D. The state of injected steam is saturated at the pressure of 2 bar. (The pressure of injected steam is the same as that of the pressure vessel because the amount of the steam injection rate is very small. Also the measured pressure and temperature just before the pressure vessel are almost equal to the saturation values.)

E. Calculation time is 100,000 seconds or more.

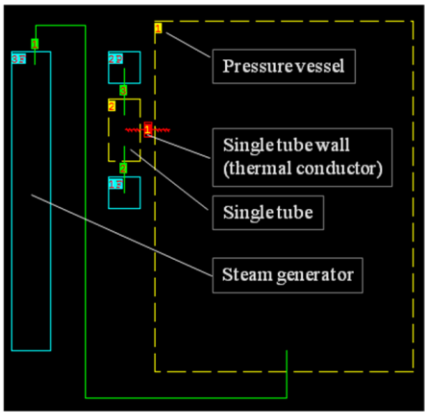
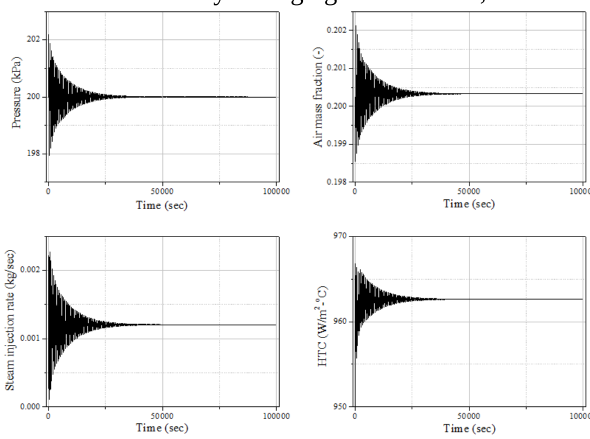


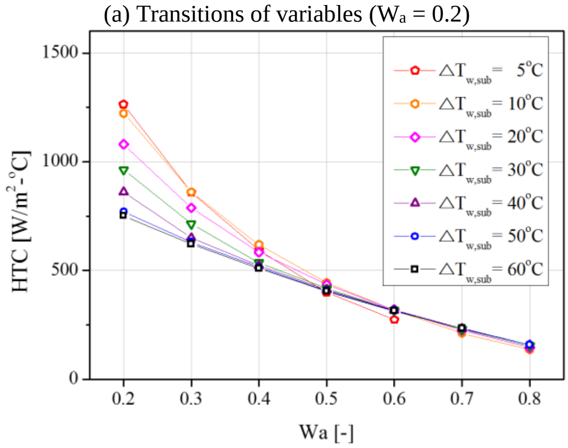
Fig. 5. The nodalization of simple model (GOTHIC code)

4.2 Calculation results

The calculation results in the same conditions with the experiments are indicated in Fig. 6. The major variables, such as pressure, air mass fraction, steam injection rate and HTC, reach steady state condition. HTC is calculated by averaging data from 96,400 to



(a) Transitions of variables (W_a = 0.2)



(b) The calculation results for HTCs in accordance with air mass fraction and wall-subcooling

Fig. 6. Calculation results

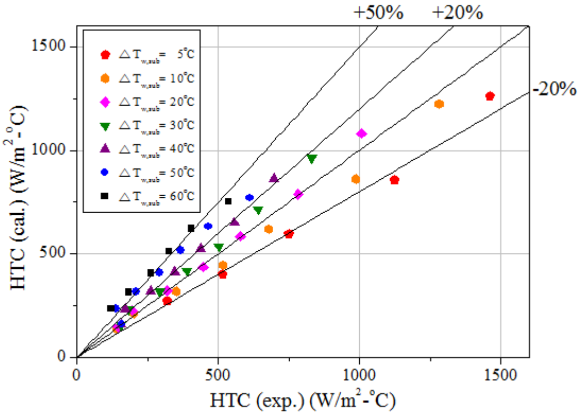


Fig. 7. Comparison of calculation results in contrast with experimental results

100,000 seconds. In Fig. 6(b), calculated HTC decreases as air mass fraction increases, and also HTC decreases as wall-subcooling increases. Hence, the calculation result shows the same tendency with the experiment result.

The ratio of calculated HTC to measured HTC are within ±20% in the conditions of low wall-subcooling (lower than 30°C), so the model predicts the test data well. In contrast, deviation between calculations and experiments becomes bigger in higher wall-subcooling conditions (higher than 50°C). In conclusion, DLM-FM overestimates HTCs in high wall-subcooling conditions.

5. Conclusions

In this study, the experiment results which include HTC in various conditions, air mass fraction and wall-subcooling, are presented. It is confirmed in the experiment results that HTC decreases as air mass fraction and wall-subcooling increases.

Also, HTC is calculated in the same conditions with the experiments by constructing a simplified model using GOTHIC code. It is known that calculation model predicts well in the range of 5 ~ 30°C of wall-subcooling. However, calculation model predicts higher than 20% compared to the experiment result in the range of 50 ~ 60°C. It is necessary to analyze the reason why DLM-FM overestimates in the high wall-subcooling conditions.

Sensitivity tests about diverse variables including heat transfer models (Uchida, DLM etc.), forms of the control volume, the effect of liquid and etc. are intended to examine, later. Finally, it is planned to build a methodology to analyze the HTC in PCCS.

Acknowledgement

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Nomenclature

A	area	m^2
c_p	Specific heat	kJ/kg-°C
HTC	Heat transfer coefficient	kW/m^2-°C
L	Length	m
$\dot{m}$	Mass flow rate	kg/s
P	Pressure	bar
T	Temperature	°C
W	Mass fraction	-

<Subscripts>

a	Air
bulk	average value of the specified space
cool	coolant (water)
in	inlet
out	outlet
s	surface
sub	subcooling
w	wall

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