Experiments of condensation heat transfer in a vertical tube with non-condensable gas

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1. Introduction

Vertical steam condensation heat exchanger was adopted in PRHRS (Passive Residual Heat Removal System) and ETC (Emergency Cooling Tank) of SMART (System-integrated Modular Advanced ReacTor) which is being developed by KAERI [1]. These systems operate in the high pressure steam conditions where a small amounts of non-condensable gases are possibly present in the abnormal operating condition. However, existing condensation experiments have been performed in the low steam mass flux and pressure condition, such as less than 20 kg/m^{2}s and 5 bar, respectively [2–6]. In this study, we investigated experimentally the condensation heat transfer coefficient in a vertical tube under the condition of an extended ranges of steam flow rate and pressure. In addition, the effect of non-condensable gas on the heat transfer coefficient was investigated.

2. Experimental setup

Fig. 1 shows a fluid system of condensation test facility [7] and vertical test section. The condensation test section was made of stainless steel 304L with an inner diameter of 40 mm, a length of 3 m and a thickness of 5 mm. The cooling water flows at a flow rate of 1.8 kg/s and temperature of 25℃ through an external annulus channel with a diameter of 80 mm.

Ten measuring positions were arranged at equal intervals along flowing direction of the test section. The heat transfer coefficient was obtained from the local heat flux and wall temperature measured by the wall temperature gradient method at each measuring location. Temperature distributions of the fluid were also measured at 15 points by traversing the thermocouple along the radial direction at each measuring plane. From this, the local air mass fraction distribution was obtained on the measurement plane in the steam and non-condensable gas condition.

Table 1 summarizes experimental conditions prepared in the present test.

Table I: Test matrix

Inlet parameters	value
Pressure, P (bar)	1, 2, 3, 4, 5, 6.5
Steam mass flux, G (kg/m^{2}s)	10, 20, 30, 40, 50
Air mass fraction, W_a (%)	0, 1, 5, 10, 25, 50

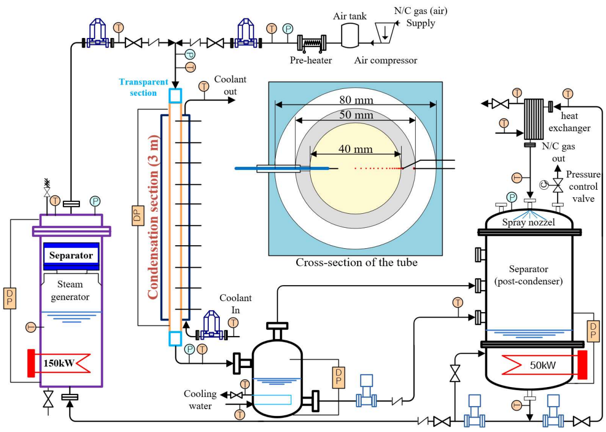

Fig. 1. Schematic diagram of test apparatus and test section.

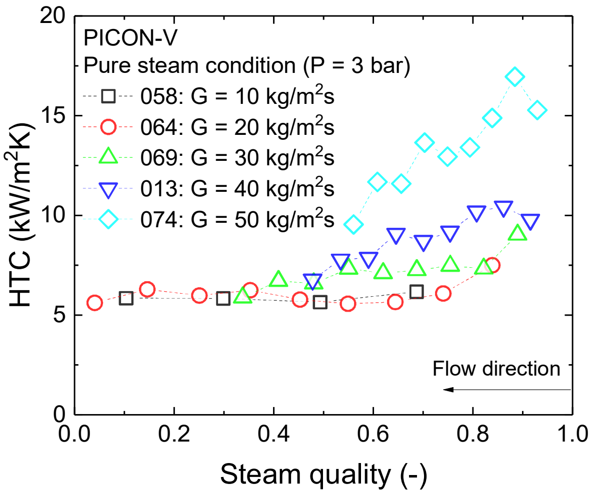

Fig. 2. Heat transfer coefficient distribution with respect to the inlet steam mass flux.

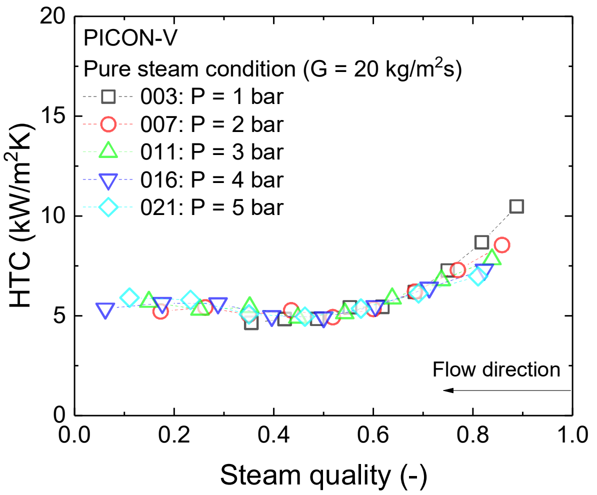

Fig. 3. Heat transfer coefficient distribution with respect to the pressure.

3. Experimental results

3.1 Heat transfer coefficient in pure steam condition

Figs. 2 and 3 show axial distributions of heat transfer coefficient with respect to the steam mass flux and pressure, respectively. As the mass flux increases, the heat transfer coefficient tends to increase. In contrast, the pressure has little effect on the heat transfer coefficient.

3.2 Local air mass fraction

Fig. 4 shows the local fluid temperature distributions along the axial and radial directions in the presence of non-condensable gas. Here, the local air mass fraction was calculated from the measured local fluid temperature as follows.

$$W_a = \rho_{air} / \left(\rho_{air} + \rho_{steam}\right), \qquad (1)$$

where the density of each phase is obtained from the property table with the measured temperature and partial pressure. The partial pressures of vapor and air was calculated by assuming that the steam is in a saturated state at the measured temperature.

The local air mass fraction distribution in Fig. 4 has a fairly high value near the wall where condensation occurs. The tendency becomes significant as it goes downstream of the condensation tube. This is because the turbulent mixing effect is diminished with decrease of the steam flow rate along the condensing tube.

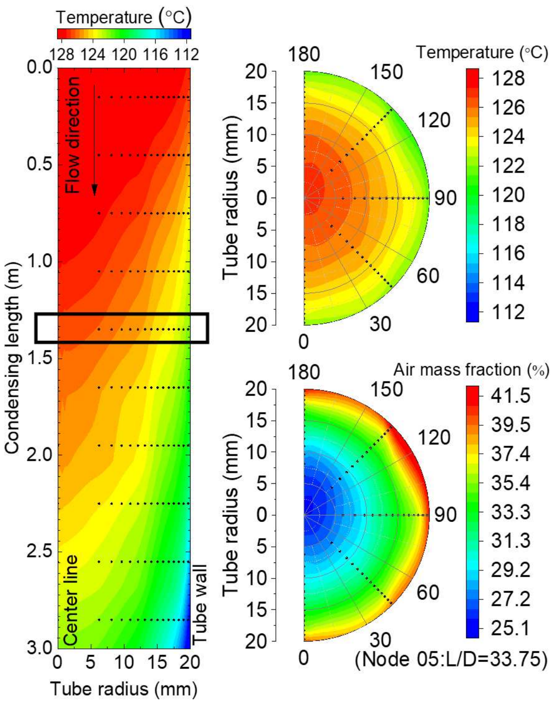

Fig. 4. Axial and radial distribution of air-steam mixture temperature and local air mass fraction. (P = 3 bar, G = 20 kg/m²s, W_a = 25%)

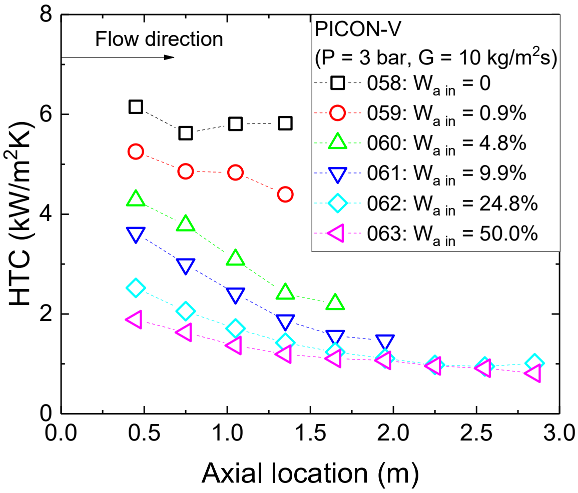

Fig. 5. Axial distribution of heat transfer coefficient in the air-steam mixture conditions.

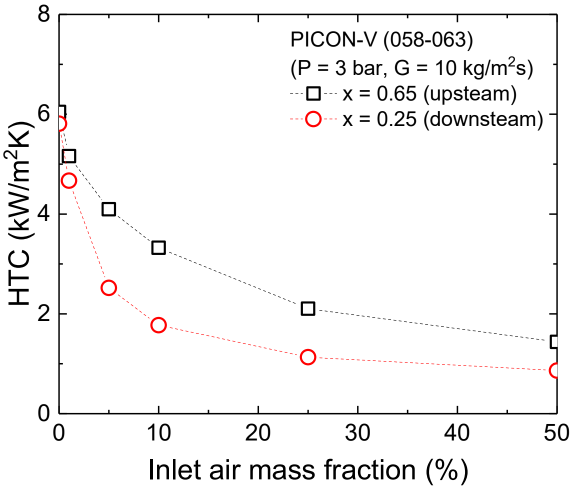

Fig. 6. Degradation effect of air mass fraction on the heat transfer coefficient.

3.3 Effect of non-condensable gas

Fig. 5 shows axial distributions of heat transfer coefficient in the steam and noncondensabe mixture condition. It shows that heat transfer coefficient decreases as the air mass fraction increases. Fig. 6 indicates that the heat transfer degradation by the non-condensable gas is larger in the exit area than in the inlet area. This is related to the distribution of non-condensable gas in the condensation tube, as shown in Fig. 4.

4. Conclusions

In this experimental study, condensation heat transfer coefficient and local fluid temperature distributions were measured in a vertical tube under the pure steam and non-condensable gas mixture conditions. In pure steam condition, It turns out that the heat transfer coefficient tended to increase with the steam mass flow rate, while the pressure has a little effect on the heat transfer coefficient. The local air mass fraction was higher near the wall than central region of the pipe in the steam with non-condensable gas condition. This distribution appears differently at the inlet and outlet of the condensation tube, which implies different degradation effect of the heat transfer coefficient. The experimental data obtained in this study will be used for the evaluation and improvement of the condensation model applicable to the condensation heat exchangers of passive safety systems.

ACKNOWLEDGMENTS

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