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Pure steam condensation in a vertical tube submerged at a water pool in a passive safety system

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1. Introduction

Condensation heat exchangers cooled by a water pool have drawn attention in nuclear power plants because they can cool down the core heat energy without any power sources. It would be very helpful for station blackout situations. Thus, in this study, the average and local condensation heat transfer were experimentally investigated in a vertical tube which is submerged at a water pool to design a new condensation heat exchanger in a passive safety system.

2. Experiments

2.1 Experimental loop

The experimental loop briefly consisted of two parts: a steam loop, and a cooling loop. (Fig. 1) In the steam loop, pressurized steam was generated from a steam generator and moved into the test section. The steam generator of maximum 150 kW heating capacity has a pressure relief valve, as well as an automatic isolation valve. After passing though the test section, the residual steam and condensate were condensed and cooled down enough (around 20 °C) by two condensers. The flow rate of this subcooled water was precisely regulated by a flow rate control valve and it was stored in the reservoir. Finally, a main pump resupplied the water into the steam generator with controlling flow rate. In cooling loop, coolant water, which was stored in a large water tank, was supplied to the each two condensers individually in counter flow by two pumps. And then, coolant water returned to the water tank again via heat exchangers where cold water was circulating from cooling tower. Also, water was supplied to the secondary jacket of the test section. The temperature of coolant water was maintained at a certain level (T = 20 °C) by a cooler.

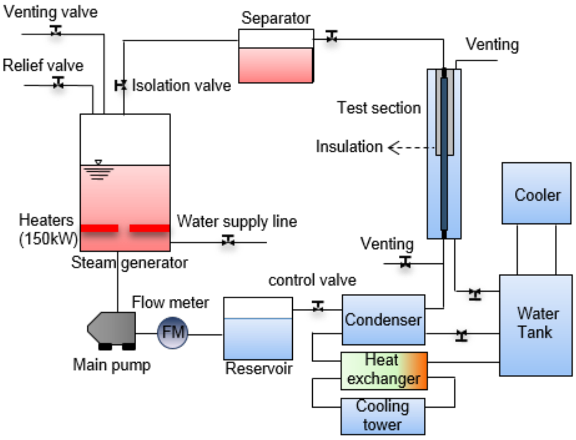

Fig. 1. Experimental loop.

The test section, total length of 2020 mm, was designed to evaluate the behavior of steam condensation in a vertical pipe by measuring local heat transfer coefficients at certain locations and an average heat transfer coefficient. (Fig. 2) Steam flowed downwards inside the vertical pipe made of stainless steel 316L, and this pipe was surrounded by an annular secondary jacket. The first 910 mm of this pipe was insulated with cerak wool. After the insulated region, calibrated K-type thermocouples were inserted to measure the inner/outer pipe wall temperatures (T'_{wi}, $T_{wo,}$ diameter of 1 mm) at 5 local points and bulk steam temperature (T_{sb}, diameter of 3.2 mm) at 1 point. At each local point, two sets of T'_{wi} and T_{wo} were mounted at two circumferential positions to ensure the measurement results. (Fig. 3) T'_{wi} was inserted in a hole of 5.2 mm depth and T_{wo} was soldered in a small groove on the outer wall surface. T_{sb} radially penetrated the pipe and located in the center of the pipe.

The Secondary jacket was initially filled with coolant water supplied from the water tank. As pressurized high temperature steam flowed in the vertical pipe, water became the saturation condition at atmospheric pressure. The evaporation rate of coolant water was obtained by water pressure changes from hydrostatic pressure changes of water in secondary jacket using a differential pressure transducer with respect to the time.

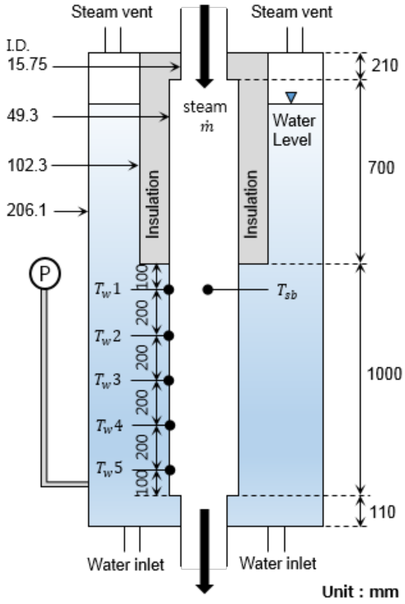

Fig. 2. Test section.

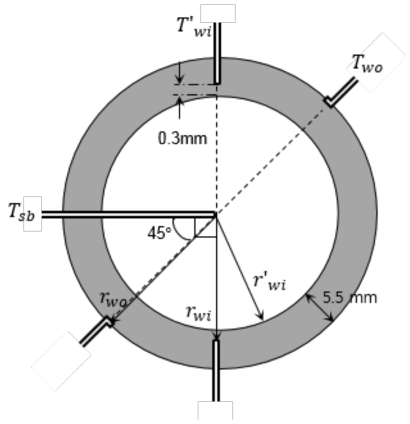

Fig. 3. Cross section of the test section and the geometry of thermocouples.

2.2 Procedure

The behavior of steam was evaluated with respect to the mass flux and pressure. The experimental cases were pressure of 6, 10 and 14 bar, and mass flux of 8.73, 17.46 and 26.19 kg/m^2/s. At first, test section was isolated and pressurized to the target pressure. After the test section reached the target pressure, the power applied to the heaters in steam generator was controlled to satisfy the target mass flux. Simultaneously, the flow rate control valve was opened and main pump was operated. With controlling the flow rate of main pump, the flow rate control valve was adjusted until the water level of the reservoir stayed constant. If the temperature of water in the secondary jacket reaches the 100 °C uniformly, experimental data were collected by a data acquisition system (Agilent 34980A). Experiment was valid only the water level in secondary jacket was not lower than the insulation region.

2.3 Data reduction

The local heat flux at the inside surface of the steam pipe in test section was determined by conduction equation in cylindrical coordinate

$$q = \frac{2\pi kL(T'_{wi} - T_{wo})}{\ln(r_{wo} / r'_{wi})}, \tag{1}$$

where q, k, L, r_{wo}, and r'_{wi} are heat (kW), thermal conductivity (W/m/K) of stainless steel 316l, length (m) of each local region, distance (m) from the center of the test section to the location of T_{wo} (K) and T'_{wi} (K), respectively. Heat flux, q'' (kW/m^2), was easily obtained by dividing q with the condensation area. Then, heat transfer coefficient, h_{local} (kW/m^2/K), was calculated by

$$h_{local} = \frac{q''}{T_{sb} - T_{wi}}, \tag{2}$$

where T_{wi} is the temperature (K) of the condensation surface, which was calculated from

$$T_{wi} = \frac{T'_{wi} - T_{wo}}{\ln(r'_{wi} / r_{wo})} \ln(r_{wi} / r'_{wi}) + T'_{wi}, \tag{3}$$

where T_{wi} and r_{wi} are temperature (K) of inner wall surface and the distance between center to the location of T_{wi}.

The condensation rate, $\dot{m}$ (kg/s), at each local region was simply,

$$\dot{m} = \frac{q}{h_{lv}}, \tag{4}$$

where h_{lv} is latent heat (kJ/kg) of saturated steam. Then, the local mass flow rate of liquid film, $\dot{m}_{film,n}$ (kg/s), was the sum of the condensation rates in whole upper region,

$$\dot{m}_{film,n} = \sum_{1}^{n} \dot{m}_n \tag{5}$$

Thus, the local quality (x_n) was obtained as,

$$x_n = \frac{\dot{m}_{total} - \dot{m}_{film,n}}{\dot{m}_{total}}. \tag{6}$$

With measured evaporation rate in secondary jacket, total heat, q_{total} (kW), was given by,

$$q_{total} = \frac{evaporation\,rate}{h_{lv}}. \tag{7}$$

However, at the end of the test section, pipe diameter was changed to 3/4″ tube, and it was taken into account for q_{test} (kW). Then, the average heat transfer coefficient, $\bar{h}$ (kW/m^2/K) was obtained as,

$$\bar{h} = \frac{\bar{q}''}{T_{sb} - \bar{T}_{wi}}, \tag{8}$$

where $\bar{q}''$ is the average heat flux (kW/m^2) and $\bar{T}_{wi}$ is the average inner wall temperature (K) that is,

$$\bar{T}_{wi} = q_{test} \frac{\ln(r_{wo} / r_{wi})}{2\pi kL} + T_{wo}. \tag{9}$$

Lastly, outlet quality was obtained in the same manner of calculating local quality.

The average uncertainty of average heat transfer coefficient was 2.7%, and the maximum uncertainty was 5.7%.

3. Results

All experiments were repeated two times to check the repeatability of the data. Average repeatability error of the heat transfer coefficients was 5.6% and the maximum error was 9.5%. The repeatability error of the heat transfer coefficients was very small, as shown in fig. 4. The experiments were repeated three times for only pressure of 6.5 bar and mass flux of 18.44 kg/m^2/s case.

The local heat transfer coefficients decreased with vapor quality because liquid film which is the major thermal resistance was thickened with vapor condensing.

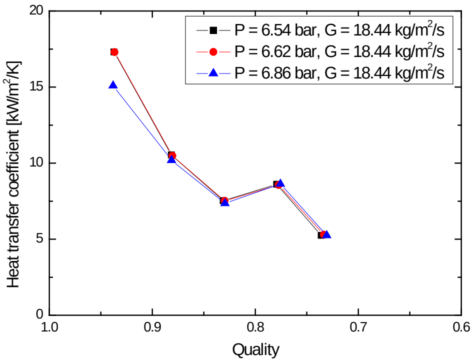

Fig. 4. The local heat transfer coefficients with vapor quality at the same condition (P=6.5 bar & G=18.4 kg/m^2/s)

Convection is the major factor influencing the condensation heat transfer. Therefore, mass flux is the main parameter for condensation. In this study, the mass flux range was 9.8 to 27.4 kg/m^2/s and it is relatively narrow range. Thus, the increment was 15% and it was not high enough to show significant enhancement of the heat transfer coefficients, as shown in fig. 5.

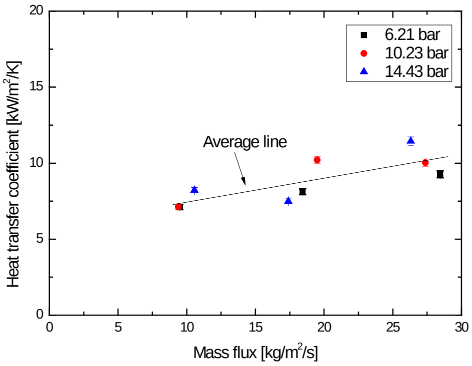

Fig. 5. The average heat transfer coefficients with mass flux at given pressures.

Pressure might have influences on condensation heat transfer because fluid properties related to the thermal and hydraulic dynamics and the saturation temperature vary with the system pressure of working fluid. In this pressure range of 6.2 to 14.4 bar, the heat transfer coefficients were enhanced about 11% as shown in fig. 6, but this increment was not enough to expect to be continued beyond the pressure of 14 bar. Thus, the effect of pressure on condensation should be investigated more.

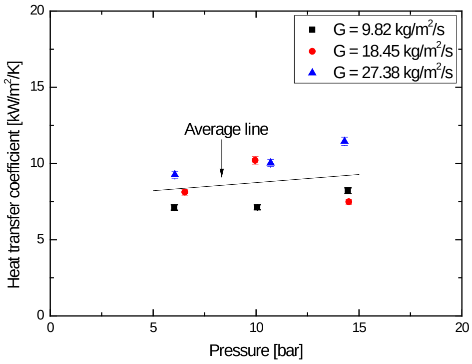

Fig. 6. The average heat transfer coefficients with pressure at given mass fluxes.

The average heat transfer coefficients and local heat transfer coefficients were estimated using Nusselt correlation[1], Shah correlation[2] and Chen correlation[3]. The mean absolute deviation (MAD) was used to compare prediction accuracy.

$$MAD = \frac{1}{N}\sum_{i=0}^{N}\left|\frac{y(i)_{pred} - y(i)_{\exp}}{y(i)_{\exp}}\right| \qquad (10)$$

Average	Nusselt	Shah	Chen
MAD	60.1%	30.1%	18.0%

Table 1. MAD of the average heat transfer coefficients with correlations

Local	Nusselt	Shah	Chen
MAD	56.6%	31.5%	23.9%

Table 2. MAD of the local heat transfer coefficients with correlations

As shown in table 1 and 2, Chen correlation accurately predicted the average and local condensation heat transfer coefficients in this experiment with MAD of 18% and 23.9% for the average and local values, respectively. Chen correlation mostly estimated the heat transfer coefficients within 20% error range, as shown in fig. 7.

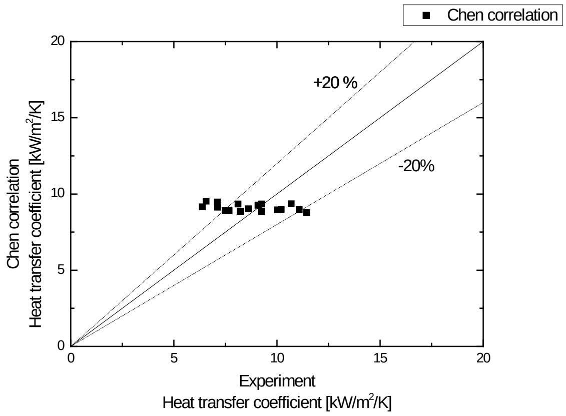

Fig. 7. Comparison between measured heat transfer coefficients and predicted values from Chen correlation.

4. Conclusion

Condensation heat transfer in a vertical tube which is submerged in a water pool was experimentally investigated and the average and local condensation heat transfer coefficients were measured at pressure of 6.2 to 14.4 bar and mass flux of 9.8 to 27.4 kg/m^2/s. The average heat transfer coefficients increased 15% with mass flux and 10% with pressure in this study. The average and local heat transfer coefficients were accurately predicted with Chen correlation (MAD=18% for the average and 23.9% for the local values).

ACKNOWLEDGEMENT

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