Transactions of the Korean Nuclear Society Spring Meeting
Jeju, Korea, May 23-24, 2019

The Experimental Study of Condensation with Non-Condensable Gas for an Inclined Tube

Seung-Min Ohk *, Sang-Gyu Lim, Jae-Min Lee, Jong Cheon, Sang-Won Lee
KHNP Central Research Institute, 1312 70-gil Yuseong-daero, Yuseong-gu, Deajeon 34101, Korea
**Corresponding author: dopeseungmin@khnp.co.kr*

1. Introduction

The condensation is an effective heat transfer process as the vapor is converting into the liquid. However, the excessive condensation would impair the heat transfer due to the condensate film as the thermal resistance [1].

The advanced NPP such as AP1000, AES-2600 and CAP 1400 are considering the passive containment cooling system (PCCS) to maintain the integrity of the NPP coping with severe accident based on SBO. KHNP is planning to adopt the PCCS. As shown in Fig. 1, the HX is consist of several tube arrangements connected to the PCCT [2]. When the postulated accident occurred such as LOCA and MSLB, the facility relieves excessive temperature and pressure in containment by releasing thermal energy into the PCCT or atmosphere through the heat exchange on the relatively cold HX tube exterior surface. Thus, estimation of condensation heat transfer is important to design PCCS in accordance with the heat removal performance. In addition, as the condensation heat transfer enhanced with increase in inclination of the tube [1], the tilted heat exchanger are the considerable issue.

This study aimed to investigate the condensation heat transfer for a single tube according to the inclination. The tube inclinations corresponded to 60° and 90° with vertical. The pressures varied from 2bars to 4bars. The air mass fractions varied from 0.2 to 0.6. The wall subcoolings corresponded to 10°C, 30°C and 50°C.

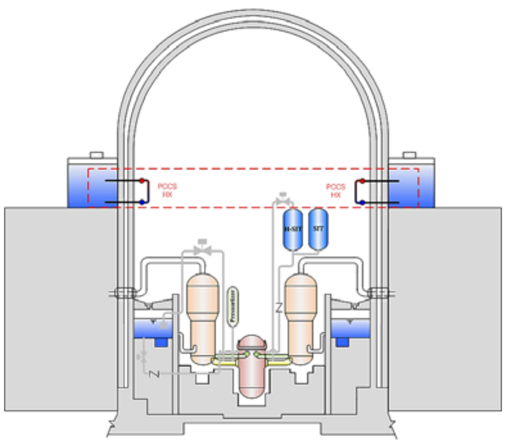


Fig. 1. Concept of PCCS HX in containment [2].

2. Experiments

2.1 Test matrix

Table 1 presents the test matrix. The length and outer diameter of the tube corresponded to 1 m and 0.032 m, respectively. The inclinations of tube were 60° and 90°. The pressures varied from 2bar to 4bar. The wall subcoolings were 10°C, 30°C and 50°C. The air mass fractions varied from 0.2 to 0.6.

Table I. Experimental test matrix for the single tube.

Conditions	Notes
Pressure	2, 3, 4 bar
Air mass fraction	0.2, 0.3, 0.4, 0.5, 0.6
Wall subcooling	10, 30, 50 °C
Tube inclination	60°, 90°(Vertical)

2.2. Test facility

The experiment was carried out utilizing a small test loop for reactor containment natural convection and condensation (ATRON) in KHNP-CRI. Fig. 2 shows a schematic view of the ATRON. The facility is consisted of the containment, steam supply system and cooling system. The containment is scale downed 1/20 in length and 1/8000 in volume comparing to the actual containment. The steam is suppled into the bottom of the containment and injected into the existing water in order to mitigate the velocity of injected steam. In order to maintain a linearity of temperature, the thermal entry length was secured before the test section. In addition, the coolant passed through the honeycomb, which is located at the bottom of the tube, and circulated to the top of tube with forced convective flow to form a uniform flow.

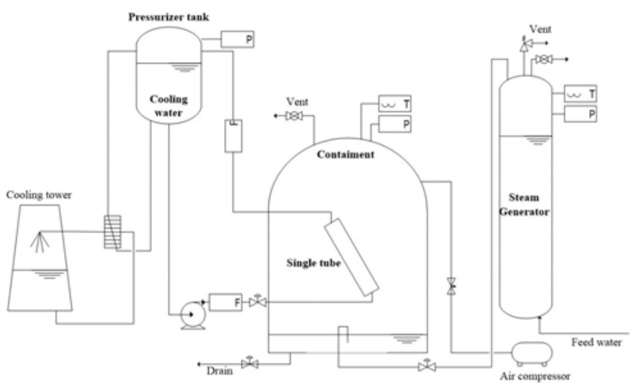
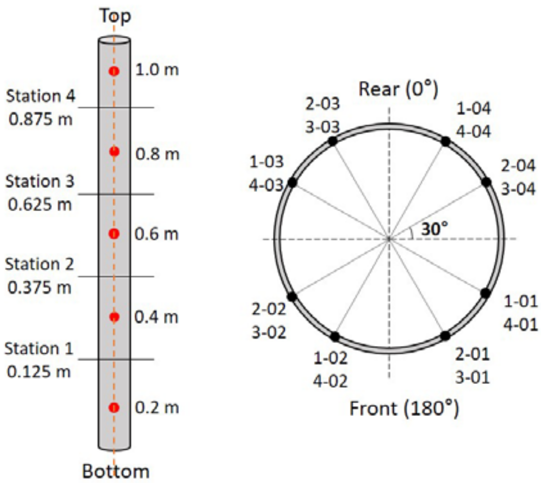
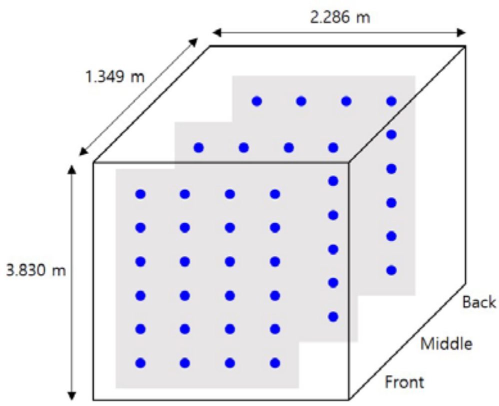


Fig. 2. Schematic diagram of test facility.

Figure 3 presents the location of thermocouples at the tube and containment. Temperature measurement is important as the temperature difference in tube and wall subcooling determines the HTC. Fig. 3(a) shows that the thermocouples are located in tube centerline for each elevation to measure the coolant temperature in axial direction. In order to measure the local wall temperature in circumferential direction for each elevation, thermocouples are flush-mounted on tube wall. Fig. 3(b) shows the location of 72 thermocouples in containment. Average temperatures and containment pressures used to calculate the air mass fraction in containment.



(a) Location of thermocouple for the tube



(b) Location of thermocouple in containment

Fig. 3. Illustration to the location of thermocouples

3. Results and Discussion

3.1 The heat transfer rate of single tube

The heat transfer rate of single tube can be evaluated by heat transfer coefficient, which are calculated as below,

$$h_{HTC} = mc_p \Delta T_{tube}/A \Delta T_{sub} \quad (1)$$

Where,

$$\Delta T_{tube} = T_{outlet} - T_{inlet} \quad (2)$$

$$\Delta T_{sub} = T_{bulk} - T_{wall} \quad (3)$$

Instrument measurement error was considered for the final dependent variable of the heat transfer coefficient. The calculated maximum uncertainty is about 11% which is the most affected by ΔT_{tube}.

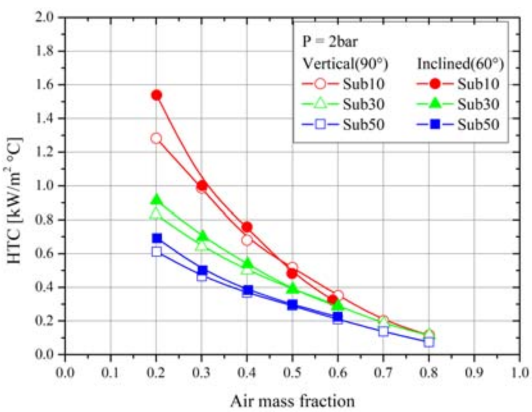
3.2 Effect of non-condensable gas

Figure 4(a)-(c) present tube heat transfer coefficient (HTC) of inclined and vertical tube according to the air mass fractions and wall subcoolings for each pressure. The closed symbols and open symbols are the HTCs of inclined tube and vertical tube, respectively. Both the HTCs of inclined and vertical tube decreased as the air mass fraction decreased. The absolute value of HTCs decreased as the wall subcooling increased.

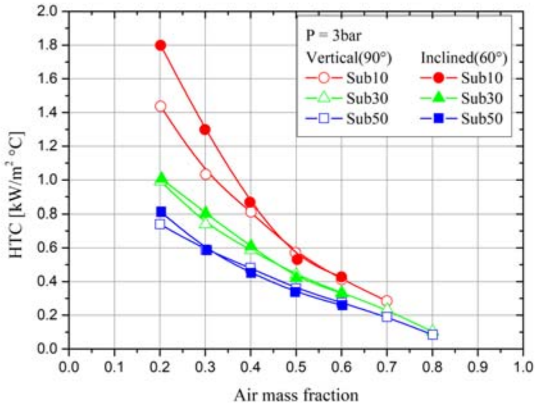
For the same wall subcooling, HTC decreased as the air mass fraction increased because the condensation heat transfer impaired due to the increase of presence of steam near the tube wall.

However, as the wall subcooling increased, the HTCs decreased though the higher wall subcooling is the expected condition that condensation heat transfer well. This is because the presence of non-condensable gas near the tube wall increased due to the steam is converted into the condensate. In accordance with the non-condensable gas act as thermal resistance, the condensation heat transfer was relatively impaired. In addition, the generated condensate acts as a thermal resistance either.

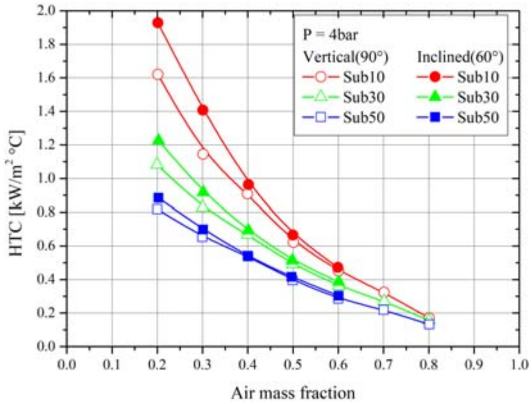
HTCs increased as the pressure increased due to the more presence of steam near the wall. However, it seems that the pressure effect is not significant for the air mass faction larger than 0.5.



(a) 2 bar



(b) 3 bar



(c) 4 bar

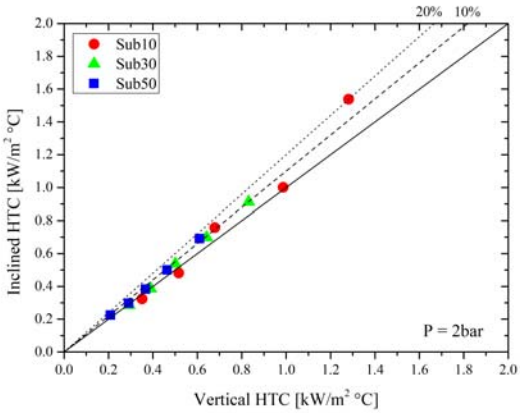
Fig. 4. HTC according to the air mass fraction.

3.3 Effect of inclination

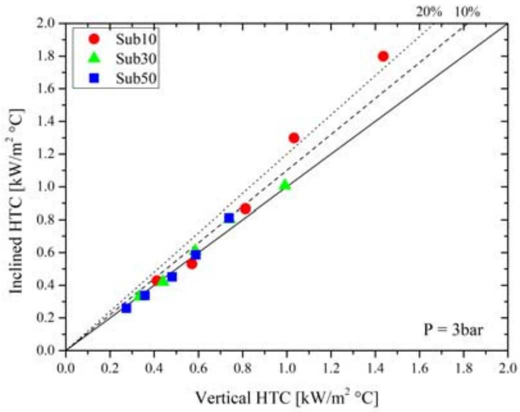
Fig. 5(a)-(c) present the comparison of heat transfer coefficients between inclined and vertical tube. For all pressures, the most results of the inclined tube were above to that of vertical tube. The maximum values of discrepancy were about 20% at the lowest air mass fraction and wall subcooling.

For the vertical tube, as the rivulets flow down to the bottom of tube due to the gravitation forces, dropwise condensation appeared at the top of tube and film wise condensation appeared at the bottom of tube.

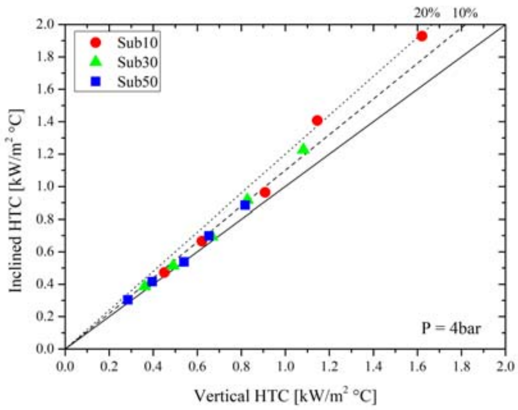
However, for the inclined tube, the rivulets travel in circumferential direction, and they are collected at the front side of tube. It is the shorter way that the rivulets sweep the surface of tube. Thus, it seems that as the dropwise condensation is more activated in the spot where the rivulet passes, the locally increased heat transfer affects the increase in ΔT_{tube}. The increase of ΔT_{tube} weakened as the wall subcooling increased because the increased condensate acts as a thermal resistance.



(a) 2 bar



(b) 3 bar



(c) 4 bar

Fig. 5. Comparison of HTC between inclined and vertical tube.

3.4 Visualization

Figure 6 shows the condensation on the tube when the condition of pressure is 4bar, air mass fraction is 0.2 and wall subcooling is 30°C. As the condensed water flows in circumferential direction due to the gravitation forces, filmwise condensation appears at the front side of tube and dropwise condensation appeared at the rear side of tube.

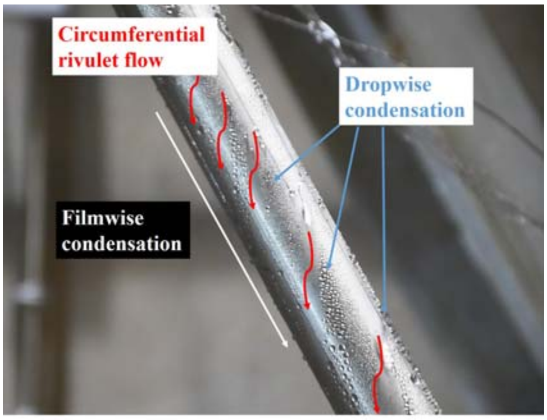
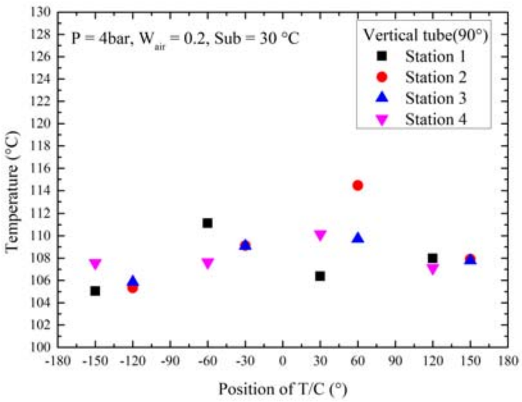
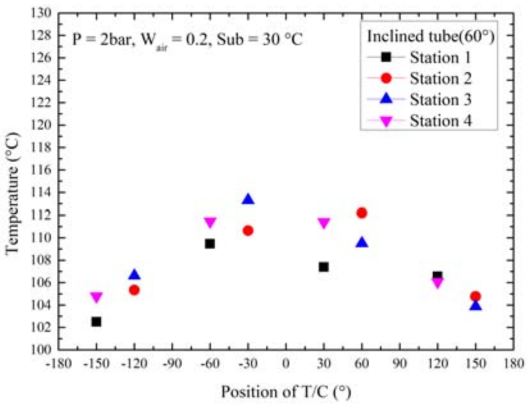


Fig. 6. Inclined tube condensation at
P = 4bar, Air mass fraction = 0.2, Subcooling = 30°C.

Figure 7(a) and (b) are the local tube exterior wall temperature of vertical tube and inclined tube, respectively. Inclination of 0° is the rear side of tube and 180° is the front side of tube. For the vertical tube, the local wall temperatures are distributed sporadically. It seems that the discrepancy between measured temperatures results from the unpredictable rivulet flows. However, for the inclined tube, the higher temperatures clearly appear near the rear side of tube and the lower temperatures appear near the front side of tube. This is because the collected condensate around the front side of tube acts as thermal resistance.



(a) Vertical tube



(b) Inclined tube

Fig. 7. Local tube exterior wall temperatures.

4. Conclusion

The experimental study of condensation heat transfer on the inclined and vertical tube were conducted with varying the pressure, air mass fraction and wall subcooling and the results were compared.

The HTCs decreased as the subcooling increased due to the lack of steam near the tube wall resulting from the activated condensation.

Regardless of pressure and wall subcooling, for the higher air mass fraction above 0.5, the HTCs of inclined tube were similar to that of vertical tube because the condensation heat transfer occurred not enough to manifest the inclination effect.

The lower air mass fraction, HTCs of inclined tube were higher than that of vertical tube and discrepancies were about 20% higher at the lowest air mass faction. It seemed that the increased dropwise condensation area due to the circumferential rivulet flow influenced on the increase in ΔT_{tube}.

ACKNOWLEDGMENT

This work was supported by the Nuclear Power Core Technology Development Program of the Korea Institute of Energy Technology Evaluation (KETEP), and granted financial resources by the Ministry of Trade, Industry and Energy, Republic of Korea (No. 20161510400120).

REFERENCES

[1] B. G. Jeon et al., Parametric experiments and CFD analysis on condensation heat transfer performance of externally condensing tubes, Nuclear Engineering and Design, Vol. 293, pp 447-457, 2015.

[2] S. W. Lee et al., The concept of the innovative power reactor, Nuclear Engineering and Design, Vol. 49, pp. 1431-1441, 2017.