Experimental Observation of Thermal-hydraulic Behavior in PCCS Horizontal Heat Exchanger

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A series of thermal-hydraulic experiments have been performed using a prototypical-scale experimental facility simulating a horizontal heat exchanger of a Passive Containment Cooling System (PCCS) for next generation BWRs. The influences of multi-dimensional boiling flow in secondary water pool on primary flow in parallel tubes are investigated. The experimental results at postulated accident conditions; 0.7 MPa, steam flow rate equivalent to 1% core power with 1% non-condensable gas, show that steam condensation completes in almost the same heat transfer length in all the instrumented tubes. The secondary heat transfer coefficient is relatively small at the lower portion in the tube bundle when the flow regime is bubbly flow, and increases with elevation as the flow regime turns into churn-like flow. The primary steam flow distribution among tubes is rather insensitive to such a variation in the secondary heat transfer coefficient, since the contribution of the secondary heat transfer to the local heat resistance is 30% or less at postulated accident conditions. The influence of steam flow rate is insensitive too, while the contribution of the secondary heat transfer coefficient increases at low pressure conditions.

**KEYWORDS:** Passive Containment Cooling System (PCCS), horizontal heat exchanger, homogeneity, heat transfer, heat removal, thermal resistance

I. Introduction

A passive containment cooling system (PCCS) removes the decay heat inside containment for a long-term cooling during a severe accident (SA) when a containment spray system is in failure. The heat removal suppresses the containment pressure increase, and prevents containment from breakage. This paper presents the study on the thermal-hydraulic characteristics of a PCCS horizontal heat exchanger.

PCCS of interest is a once-through system for a next generation BWR, which directly connects dry well (DW) to wet well (WW) as shown in Fig. 1. This system is expected to work during a postulated SA ex-vessel transient when ECCS including containment spray system is in failure. Core melt that dropped down to lower DW generates steam and non-condensable gas in a water pool formed by AM coolant injection from WW suppression pool (S/P). It should be noted that steam is condensed well by containment spray system when it works as designed. Steam and gas generated in lower DW are driven to the PCCS heat exchanger by pressure difference between DW and WW. Steam is condensed in the heat exchanger, and condensate and non-condensable gas drain into WW S/P respectively by gravity and the pressure difference.

The driving force for the PCCS operation is limited by the plant geometrical configuration; \(\Delta P\), Fig. 1. Consequently, the differential pressure across the heat exchanger should be less than a limiting value; i.e. \(~2\) kPa. Steam-gas mixture in DW enters WW through main vent line during such transient as LOCA blow-down phase or at the incipience of lower head failure by high-temperature melt when the steam generation rate in DW is large, causing the DW-WW
pressure difference far exceeds the limiting value. Heat input to S/P through the main vent will cease when the DW-WW pressure difference becomes lower than ΔP as the steam generation rate in DW decreases. Since the containment pressure depends on steam pressure in WW, which is in terms of S/P water surface temperature, the heat removal by PCCS contributes to suppress the containment pressure increase and is suitable for long-term cooling of containment.

A vertical in-tube heat exchanger has been proposed for the next generation BWR such as SBWR [1-3]. Conventional exchanger composed of horizontal U-tube heat, however, has not been considered as a candidate primarily because of lack in the experiences about the in-tube steam condensation at the PCCS operation condition, while the horizontal heat exchanger has advantages in seismic responses, manufacture and maintenance. Therefore, the Japan Atomic Energy Research Institute (JAERI) started single horizontal U-tube experiments in 1999 to confirm the adequacy of the condensation heat removal capacity and smooth drain of condensate and non-condensable gas. The experimental results showed that the horizontal condenser tube may satisfy the required performance in respect to heat transfer, condensate drainage and gas exhaust at small pressure loss across the condenser tube [4-6]. Based on the results above, performance confirmation experiment was started in 2002 using a prototypical-scale heat exchanger model composed of tube bundle. One of objectives of this experiment is to observe the influences of the multi-dimensional boiling two-phase flow in the secondary side onto primary in-tube thermal-hydraulic behavior in a tube bundle. This paper summarizes the experimental results especially focusing on the influences of variation in the secondary side heat transfer coefficient onto the primary flow behavior.

II. Experimental Facility

The prototypical-scale tube bundle experimental facility shown in Fig. 2 is composed of U-shaped horizontal condenser tubes, and located at the top of the ROSA-V/LSTF [7] of JAERI. Steam from two LSTF steam generators is provided to the facility inlet plenum via a mist separator and flow control valves that provide a slight superheating in the inlet steam. Non-condensable gas such as air is mixed with steam after temperature control by a gas heater. Flow rates of steam-gas mixture and non-condensable gas are measured by electromagnetic flowmeters. The inlet plenum in this facility equips an internal structure to flatten flow velocity distribution at the tube inlet.

The tube bundle is a half cut model and installed in a rectangular secondary water pool so that one of the pool inner wall forms the vertical and symmetrical plane of the tube bundle model. The outer diameter of the condenser tube is ~32 mm, and the averaged length is ~8 m, following the results of single-tube experiments.

Condensate and non-condensable gas drain first into the outlet plenum and flow down to a WW simulation tank through a drain piping. The condensate in the tank is returned to the LSTF steam generators with feedwater pumps, and the gas is released to atmosphere through a pressure control valve.

The secondary water pool is 4.5 m in length, 1.5 m in width, 3.0 m in height being covered by thick heat insulator. The secondary coolant is saturated because of significant boiling in the tube bundle and subject to decreased because of evaporation. Demineralized water at about 60 °C is continuously supplied to the pool through multiple inlets in a manner such that it is well mixed with the convecting secondary coolant. The generated steam is emitted to atmosphere through large-diameter pipes connected to the pool ceiling. The pool side walls have a number of windows to observe boiling and convecting flow behavior.

In six selected condenser tubes, primary fluid temperatures and primary and secondary wall surface temperatures are measured at 4 locations for 2 tubes and 7 locations for 4 tubes. As for the primary side, three sheathed thermocouples (T/Cs) are inserted from top, side and bottom of tube on the same plane normal to the flow axis at each measurement location. Center T/C to measure fluid temperature is inserted such that the tip is located on the flow axis. Two other T/Cs are inserted into the tube by ~5 mm from the wall. To measure the tube wall surface temperatures, two sheathed T/Cs (0.5 mm o.d.) are embedded such that the sheath surface is flush to wall surface, which faces to coolant. The measurement of secondary fluid temperature is performed at the five planes perpendicular to the direction of the heat exchanger axis. At each plane, coolant temperature is measured at circumference and inside of the heat exchanger by 12 – 14 T/Cs. All T/Cs are calibrated by comparison with each other for long-term still water temperature after the end of experiment.

Void fraction meters made of optical glass fibers are installed in the PCCS pool. Each void fraction meter
discriminates bubbles by the difference in the laser light reflection intensity at the tip of optical fiber. Two in the void fraction meters are traversed from the center elevation to the upper end elevation of the bundle to measure the void fraction distribution in the bundle at two locations of $L/D=11$ and 117 ($L$ is the distance from tube inlet).

**III. Experimental Result at Nominal Condition**

A series of tests were performed parametrically focusing on heat exchanger behavior around nominal conditions required for PCCS; the pressure of 0.7 MPa, the tube inlet steam-gas mixture flow rate equivalent to 1% decay core power removal and the non-condensable gas partial pressure of 1%. The tube inlet gas mixture velocity is ~20 m/s at this nominal conditions. The steam-gas mixture at the tube inlet is dry and almost saturated with slight superheating.

The primary fluid temperatures along the flow direction at the nominal condition are compared for six instrumented tubes in Fig. 3. The temperatures indicate the averaged value from three T/Cs at top, center and bottom at each measurement location. The fluid temperature decrease rate increased at ~4 m from the tube inlet, since steam condensation had almost completed at this location. The secondary-side flow observation show that this location corresponds to the terminus of similar steam bubble generation region on the tube surface. Temperature response of all the six instrumented tubes are almost the same, suggesting the similar inlet flow rate and steam condensation behavior among these tubes, irrespective of U-bend radius and thus the tube position within the bundle.

The secondary fluid temperature distribution in the PCCS pool was homogeneous at saturation temperature during the test. The maximum temperature difference among 113 T/Cs for secondary fluid temperature measurement was ~2 K (including measuremental error), while water at 60 °C was furnished to replenish evaporation. Thus, the test results are not affected by any thermal stratification in the secondary side.

Local heat removal rate $Q$ and local heat transfer coefficients $\alpha_i$ and $\alpha_o$ at both primary and secondary sides are estimated from measured wall temperatures in consideration of the embedding depth $\delta$ of T/Cs for wall temperature measurement. The local heat removal rate and local heat transfer coefficients are given as

$$Q = \frac{2\pi l \lambda (t_{iw} - t_{ow})}{\ln(r_o/r_i) - \delta \left(\frac{1}{r_i} + \frac{1}{r_o}\right)}$$

(1)

$$\alpha_i = \frac{Q}{2\pi r_i l (t_i - t_{iw}) - \frac{\delta Q}{\lambda}}$$

(2)

$$\alpha_o = \frac{Q}{2\pi r_o l (t_{ow} - t_o) - \frac{\delta Q}{\lambda}}$$

(3)

The embedding depth of T/Cs is defined as the distance between the T/C center and the tube wall surface, assuming that the T/Cs measure the temperature at the center of T/C junction and the thermal conductivity between the T/C junction and the tube wall surface is almost equal to that of SUS. A pair of T/Cs, thus six T/Cs on the wall, three T/Cs in the primary side and one T/C in the secondary side at each measurement location, is used to obtain $\alpha_i$, $\alpha_o$ and $Q$ at each of three orientations; top, side and bottom, of the tube cross-section.

![Fig. 3 Primary fluid temperature (0.7 MPa, tube inlet velocity: ~20 m/s, non-condensable gas: 1%)](image)

![Fig. 4 Heat removal rate (0.7 MPa, tube inlet velocity: ~20 m/s, non-condensable gas: 1%)](image)
Averaged local heat removal rate, which is obtained from the average of three orientation values, is shown in Fig. 4. The averaged local heat removal rate decreased with the distance from the tube inlet and the decrease rate increased at ~4 m from the tube inlet where the completion of steam condensation has been suggested from the primary fluid temperatures in all the instrumented tubes. This comparison shows that the influence of the tube location in tube bundle on the local heat removal behavior is little. This result supports the previous suggestion based on fluid temperature such that the tube inlet flow rate would be almost uniformly distributed among tubes. The heat removal rate in the downstream leg decreases in rather exponential manner according to the decrease in the temperature difference between the primary and secondary sides.

The averaged local heat transfer coefficient at the primary side is shown in Fig. 5. Three local heat transfer coefficients obtained at tube top, side and bottom are averaged. The heat transfer coefficient decreased rather monotonically with the distance from the tube inlet, insensitive to the tube location in tube bundle, while there was a small deviation in the condensation heat transfer in a few tubes. This result also suggests the uniform distribution of tube inlet flow rate.

It is interesting to note that the heat transfer rate at the tube bottom surface (not shown) increased locally at the U-bend outlet: ~4.5 m from the tube inlet, apart from the monotonical decrease manner in the averaged value shown in Fig. 5. This result suggests that flow agitation because of flow acceleration in the U-bend and transition into supercritical flow at the U-bend exit enhanced both the steam condensation on the agitated condensate flow and the wall heat transfer from the condensate. Such an enhancement of heat transfer would have occurred in the U-bend too, while there is no temperature measurement to confirm it.

The local heat transfer coefficient at the secondary side is shown in Fig. 6. Small temperature difference between the tube outer wall and the secondary coolant made it difficult to properly obtain the secondary heat transfer coefficient in the downstream leg after the primary steam condensation completed. The obtained result clearly show that the heat transfer coefficients of two tubes located at the lowest level in the bundle, namely tube d and f, are smaller than that of the other tubes. This difference seems to be attributed to the difference in the secondary-side boiling behavior in the upper and lower parts of tube bundle. Figure 7 shows photos of the secondary-side boiling behavior taken at ~1 m from the tube inlet by digital high-speed video system at a rate of 4500 picture/s. At the bottom tube, there was an incipience of subcool boiling in the bottom surface. The boiling condition changed to bulk boiling around the top surface of the bottom tube. The flow pattern turned from bubbly flow at the lower part of the tube bundle to churn-like flow at the upper part of the bundle. The increase in the void fraction and flow velocity resulted in such a flow regime transition. Figure 8 shows the vertical void fraction distribution in the tube bundle from the heat exchanger center to the top of the bundle, measured by a traversing optical fiber void meter. It takes about 20 minutes to obtain the whole profile of the vertical void fraction distribution. The void fraction at around the bottom tube is ~0.3, and increases to ~0.8 at the bundle top. The void fraction in subchannel increases probably because of the relatively low mixture velocity, while it decreases at narrow gap between tubes as the mixture velocity increases. Considering that the flow regime
transition from bubbly to churn flow in vertical pipes and rod bundles takes place when void fraction exceeds ~0.5, the observed flow behavior may roughly agrees with this transition condition, though the flow of interest is the vertical cross-flow through horizontal rod bundle. Consequently, secondary flow is highly accelerated and agitated causing an enhancement of boiling heat transfer in the upper portion of rod bundle, while the flow around the bottom tubes especially is too mild to enhance the heat transfer.

The local heat removal rate is not significantly influenced by the difference in the secondary-side heat transfer coefficients as shown in Fig. 4. The reason of this response is attributed to the thermal resistance across the tube wall as follows.

The primary-to-secondary local heat removal rate can be described as

\[ Q = \frac{(t_i - t_o)2\pi L}{\alpha_{s_t} + \frac{\ln(r_i/r_o)}{\lambda} + \frac{1}{\alpha_{s_t}r_o}}. \]  

(4)

The total thermal resistance is thus

\[ \frac{1}{\alpha_{s_t}r_o} + \frac{\ln(r_i/r_o)}{\lambda} + \frac{1}{\alpha_{s_t}r_o} \frac{2\pi L}{\lambda}. \]  

(5)

The first and third terms correspond to the contributions of the primary and secondary heat transfer to the total thermal resistance.

Figure 9 shows the contribution of the primary and secondary heat transfer to the total thermal resistance. The contribution of the secondary heat transfer was less than 30%. The secondary heat transfer coefficients of the tubes d
and f are ~30% smaller than those of the other tubes at the upstream leg because of the multi-dimensional boiling flow as has been shown in Fig. 6. The influence of this difference in the secondary heat transfer coefficients is mitigated to less than ~10% in the local heat removal rate, since no difference was observed in the primary heat transfer and the thermal conductivity. The decrease in the influences of the multi-dimensional boiling two-phase flow resulted in the uniform flow distribution among tubes.

IV. Effects of Primary Pressure and Inlet Steam Flow Rate

The experiments are performed at the primary pressure of 0.2, 0.4 and 0.7 MPa to study the influences of the primary pressure to the heat exchanger performance. The tube length for the condensation heat transfer depends strongly on the temperature difference between the primary and secondary fluids; both in saturated condition. The tube length for the completion of steam condensation increases as the primary pressure decreases such that ~4 m for 0.7 MPa becomes ~7 m for 0.4 MPa when 1% equivalent decay heat is removed by steam condensation. A part of steam is uncondensed and exits from the tube end when the primary pressure decreases to 0.2 MPa.

The influence of primary pressure is equally imposed as saturation temperature to the primary steam flow in all the tubes. However, the primary pressure is found to cause a counter effect through the secondary-side boiling behavior, especially at low pressures. Figure 10 compares the secondary side heat transfer coefficients for six instrumented tubes at the pressure of 0.2 MPa with the inlet steam flow rate corresponding to 0.5% decay heat. The tube inlet volumetric flow rate increases when the primary pressure decreases, because gas density decreases while the change in the latent heat is small. The obtained heat transfer coefficients increase with the elevation from the center of tube bundle. Such a heat transfer distribution is caused by the relative low heat removal rate. The local heat removal rate at the condition is shown in Fig. 11. Since the local heat removal rate at the pressure of 0.2 MPa is far smaller than in 0.7 MPa cases, the steam generation rate on the tube wall surface at 0.2 MPa is mild. This small steam generation rate results in a gradual increase in the vertical void fraction distribution. The flow regime transition from bubbly to churn-like flow takes place over a large vertical span in the tube bundle.

Figure 12 compares the averaged contribution ratio of the secondary-side heat transfer in the total local heat transfer rate in terms of primary pressure at four locations; 0.5, 1.0, 2.0 and 3.5 m from the tube inlet. When pressure is decreased, the contribution ratio of the secondary-side heat transfer increases, and reaches to 40% at 0.2 MPa; twice as much as that at 0.7 MPa. Some flow distribution may appear among tubes in low pressures, because of the tube-to-tube non-homogeneous heat transfer rate such as larger heat transfer rate in the tube located at the upper part of the bundle than smaller heat transfer in the tubes located at the lower part.

Another series of parameter experiments are performed by changing tube inlet steam flow rate in the range that equivalent to 0.25 - 1.25% decay power. Every test was performed at pressure of 0.7 MPa and at the non-condensable gas partial pressure of 1%. Figure 13 shows the primary fluid temperatures along the flow direction. Tube length to complete condensation increased monotonically with the steam flow rate such that ~1 m for 0.25 % becomes ~4 m for 1.25%. The contribution ratio of
the average secondary heat transfer rate in the total tube wall heat resistance is shown in Fig. 14. The contribution ratio decreased only slightly with the tube inlet flow rate. Therefore, following the discussion in the previous section, the tube inlet steam flow rate is not influential to the flow distribution among heat exchanger tubes.

V. Conclusion

A series of experiments have been performed using a prototypical-scale experimental facility simulating a horizontal heat exchanger of Passive Containment Cooling System (PCCS) for next-generation BWRs. Thermal-hydraulic phenomena are observed in six instrumented tubes and secondary side water pool. Obtained results are summarized as follows.

1) Inhomogeneous flow distribution among condenser tubes does not appear in postulated accident conditions; 0.7 MPa, steam flow rate equivalent to 1% decay heat removal with 1% non-condensable gas concentration. Temperature measurement revealed that steam condensation completes in almost the same tube length and no significant deviations exist in the primary side heat transfer coefficients among the six instrumented tubes.

2) The secondary side boiling heat transfer is influenced by two-phase flow regimes in subchannel; bubbly and churn-like flows. The secondary heat transfer coefficient is relatively small at bottom row tubes which are in bubbly flow, and increases at upper rows which are in churn-like flows, in the postulated accident conditions.

3) Contribution of the secondary heat transfer on the tube thermal resistance is 30% or less in the postulated accident conditions. Local heat removal rate is thus insensitive to a small difference in the secondary heat transfer coefficient.

4) Variation in the secondary heat transfer coefficient among tubes becomes significant as the primary pressure decreases. Since the contribution of the secondary heat transfer increases at low pressures, inhomogeneous heat removal distribution among the tubes may happen at low pressure conditions.

5) The influence of the steam flow rate on the secondary heat transfer variation is almost negligible.
Nomenclature

- \( l \) unit length
- \( Q \) heat removal
- \( r_i \) inner radiuses of condenser tube
- \( r_o \) outer radiuses of condenser tube
- \( t_i \) primary fluid temperature
- \( t_o \) secondary fluid temperature
- \( t_{pw} \) primary wall temperature
- \( t_{sw} \) secondary wall temperature
- \( \alpha_i \) primary heat transfer coefficient
- \( \alpha_o \) secondary heat transfer coefficient
- \( \delta \) embedding depth of T/C \( (\delta = 2.75 \times 10^{-4} \text{ m})\)
- \( \lambda \) thermal conductivity of SUS \( (\lambda = 16.5 \text{ W/mK})\)

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References