Pool boiling heat transfer enhancement on porous surface tube

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Abstract The passive residual heat removal exchanger (PRHR HX), which is a key equipment of the passive residual heat removal system, is installed in an elevated pool. Its heat transfer performance affects security and economics of the reactor, and boiling heat transfer in the liquid surrounding the exchanger occurs when the liquid saturation temperature exceeded. The smooth tubes, which are widely used as heat transfer tubes in PRHR HX, can be replaced by some enhanced tubes to improve the boiling heat transfer capability. In this paper, the pool boiling heat transfer characteristics of smooth tube and a machined porous surface tube are investigated by using high-pressure steam condensing inside tube as heating source. Compared with smooth tube, the porous surface tube considerably enhances the boiling heat transfer, and shortens the time significantly before reaching the liquid saturation temperature. Its boiling heat transfer coefficient increases from 68% to 75%, and the wall superheat decreases by 1.5ºC. Combining effect of condensation inside tube with boiling outside tube, the axial wall temperatures of heat transfer tube are neither uniform nor linear distribution. Based on these investigations, enhance mechanism of the porous surface tube is analyzed.

Key words Pool boiling, Porous surface tube, Enhanced heat transfer

1 Introduction

In some advanced nuclear power plants, a passive residual heat removal system (PRHR) is used to remove core decay heat in the event of a station blackout. The PRHR system is independent of external mechanical or electrical power, signals or forces, such as electric pumps[1]. It is designed with sufficient simplicity, with decreased human interaction and reduced operating cost. The passive residual heat removal heat exchanger (PRHR HX), a key equipment of PRHR, is supported and submerged inside the in-containment refueling water storage tank (IRWST) in the advanced passive plant of AP1000 [1], because its heat transfer performance directly affects the reactor reliability and safety, thus acting as the heat sink for the heat exchanger.

The PRHR HX, connected to the reactor coolant system (RCS) in a natural circulation loop, is isolated by two parallel air-operated valves that are normally closed. When the PRHR HX works, the temperature and its elevation difference between the hot inlet water and cold outlet water drives the natural circulation, and the water in IRWST absorbs the heat transferred to the PRHR HX, thus escalating its water temperature. When the water temperature in the IRWST exceeds its saturation point after operating for several hours, nucleate boiling occurs to transfer the heat in the core to the pool[2], then the steam condenses on the containment walls, and a safety-grade guttering system back to the IRWST directs the condensate to continue the cycle. So pool boiling heat transfer in PRHR HX is a long-time operating condition, and plays an important role in the PRHR operation.

The smooth tubes are widely used as the heat transfer tubes in PRHR HX to enhance the boiling performance by ameliorating the heating surface due to limitation of their common boiling heat transfer capability. This can be performed by creating

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Received: 2010-07-18

rough/geometrical surfaces specially treated and coated[3], e.g. the porous structure surface with sample and effective boiling performance.

Gao *et al.*[4] studied the nucleate pool boiling performance of R134a and R142b outside a horizontal bank of twisted tubes with machined porous surfaces, indicating that the enhancement ratios of R134a and R142b could be up to 1.4–1.7 and 4.4. Kim *et al.*[5] conducted the pool boiling heat transfers on structured and enhanced pore tubes with connecting gaps. They tested the R11, R123 and R134a with the pore sizes of 0.20, 0.23 and 0.27 mm, and verified that their coefficient varied with the pore size and refrigerant variety, being 6.5, 6.0 and 5.0 times larger than those of the smooth tubes. Min *et al.*[6] reported that a hot-powder compaction method could be used to make 2-D and 3-D modulated coatings in the enhanced pool boiling performance, indicating that their maximum measured critical heat flux were 3.3 and 2.0 times higher than those of the plain surfaces, and strongly depended on the modulation pitch with little effects of particle diameter and porosity.

In this paper, we investigate the pool boiling heat transfers of the smooth and machined porous surface tubes. Electric heating is mostly used as heating source and the heat transfer condition as uniform heat flux. In actual operation of the PRHR HX, heat transfer process to the pool is combined by convection or condensation to the inside surface of the heat exchanger tubes, heat conduction through the tube walls and natural convection /nucleate boiling in the pool. Heat transfer condition is neither uniform wall temperature nor uniform heat flux. Therefore, experiments were carried out with high-pressure saturated steam condensation inside tubes to simulate the actual working condition in PRHR HX.

2 Experimental

2.1 The apparatus

To investigate a series of pool boiling heat transfers of tube submerged in a water pool, a small scale experimental facility was set up, as shown in Fig.1. It includes electric heating boiler, boiling tank, boiling heat transfer tube, condensate measure tank, condenser, data acquisition system, valves, and connecting pipeline.

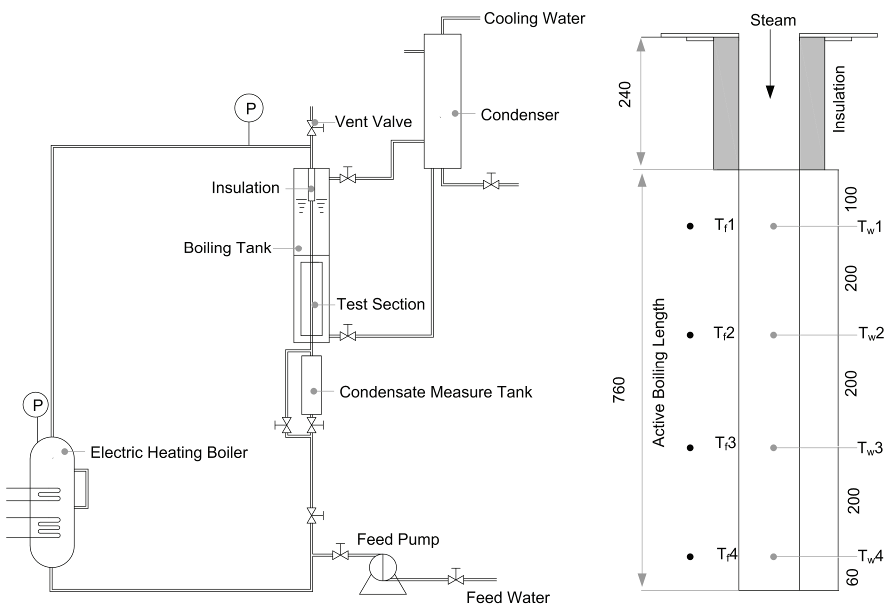

Fig.1 Schematic diagram of experimental apparatus and test section.

The smooth or porous heat transfer tube was installed vertically in the cylindrical stainless steel tank of *Φ*100-mm ID and 1000-mm height. A glass window of 80 mm×500 mm on the tank was used to view the tube and take photographs. The smooth tube with 16.0-mm outer diameter and 2.0-mm thickness was made of B30 copper/nickel alloy; while the porous surface tube with 16.3-mm diameter was formed by numerous micro-cavities on the outer surface of smooth tube, as shown in Fig.2.

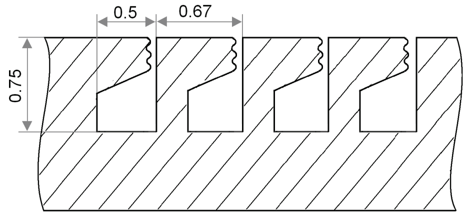

Fig.2 Scheme of porous surface tube (in mm).

The two tubes are of 1000-mm length. The initial water level in the tank is 800 mm. To avoid the water level fluctuation induced by boiling, the upper 240 mm of the two tubes were coated with an insulation layer, leaving an effective heating length of 760 mm. Four thermocouples were installed outside the tubes, at 100, 300, 500 and 700 mm from the inlet, to measure the temperature distribution of tube surface. Four more thermocouples were deployed in the tank to measure the water temperature (Fig.1). Two sheathed thermocouples were installed in the steam pipe and condensate tank to measure temperature of the steam and condensate. The thermocouples were calibrated prior to their installation, and were connected to a data acquisition system with an accuracy of 0.1ºC. Coated with thermal insulation material, the heat losses of the apparatus are neglected in the analysis.

The non-condensable gas in the steam loop can cause a resistance to heat and mass transfer inside the tube. Prior to the experiments, the steam mixed with non-condensable gas was vented to atmosphere via the venting valve. After removing non-condensable gas completely, we closed valve to initiate the experiment. Natural circulation could be easily established by the elevation difference between the boiling tank and electric heating boiler. The high-pressure saturated steam in the boiler flows from the top of tube and condenses in the tube. The condensed water flows downward along the tube and gathers in the measurement tank. The condensation rate was measured before the water returns to the boiler. The heat was transferred to the outer tube surface, which heated the water in the tank. On reaching the saturated water temperature, the heat was transferred to tank due to nucleate boiling. The steam in the boiling tank was condensed by the condenser and returned to the tank from the inlet in the tank bottom. The saturated water was approximately maintained by controlling the low sub-cooling and making good insulation to the boiling tank during the boiling experiment.

2.2 Data reduction

The average boiling heat transfer coefficient for smooth and porous surface tubes are defined as,

$$h_0 = Q/(A_0 \Delta T) \tag{1}$$

where Q is the heat transfer power, A_0 is the area of boiling surface, and ΔT is the average wall superheat measured by the experiments.

Because the heating source is the high-pressure saturated steam condensed inside the tubes, the Q is calculated by the condensing heat per unit time,

$$Q = V(t) h_{fg} / (v\, t) \tag{2}$$

where $V(t)$ is condensate volume at the measurement time of t, h_{fg} is the latent heat, and v is the specific condensate volume.

3 Results and Discussion

During the experiments, the steam pressure in tube was maintained at 0.3 MPa, and the water temperature in tank was increased gradually from room temperature to saturation temperature, because the steam condensing in tube and the water in the tank experienced natural convection and sub-cooled boiling, and reached saturated nucleate boiling.

Figure 3 shows that the time depends on water temperatures at different elevations of the smooth and porous surface tubes, and Fig.4 shows the vertical temperature distribution versus the heating time. Before reaching water saturation temperature, the two tubes have similar heat transfers (Fig.3). During the heating process, temperature of the water in the tank rose gradually because the heat transfer tube is placed vertically in the boiling tank, leading to a strong thermal stratification in vertical direction. The heated water adjacent to the tube moved up by the buoyancy force due to its reduced density. Hot water reaching the top surface tends to flowing downward. However, the hot water moved towards the tank wall along the surface due to its low density. Over a period of time, strong thermal stratification occurred in the tank water, and the tube-to-tank heat transfer became significantly low under thermal stratification conditions[7].

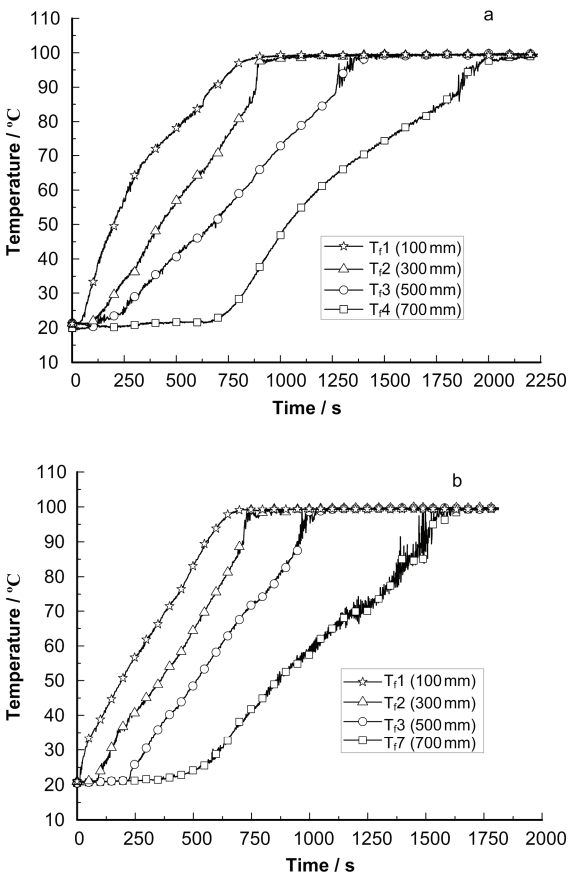

Fig.3 Water temperature versus heating time. Smooth tube (a), and porous surface tube (b).

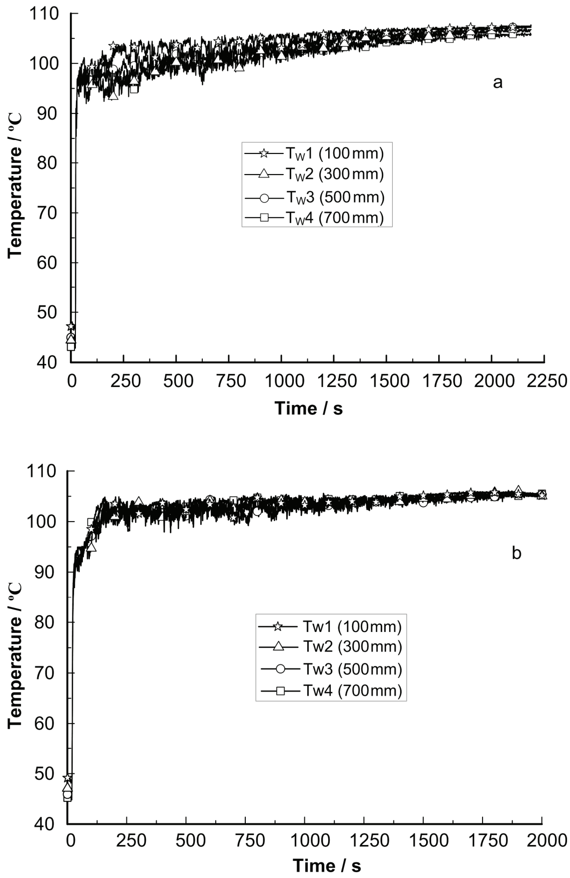

Fig.4 Wall temperature versus heating time. Smooth tube (a), and porous surface tube (b).

Differently, the wall temperature for the four different elevations increased rapidly as the high temperature steam flew into the tube, and rose slowly in similar trend (Fig.4). The total thermal resistances in the experiments include the condensation inside tube, the conduction on tube wall, and the natural convection between the tank water and tube surface. The latter is larger, so the wall temperature is affected insensitively by that of tank water.

Before the bulk water temperature becomes saturated, the temperature rising in the porous surface tube is at about 1700 s; and in the smooth tube, at about 2000 s. So the heat was transferred to the tank by natural convection and sub-cooled boiling. As the heat transfer coefficient of natural convection is lower than that of boiling, a strong thermal stratification occurred in the tank during the heating up, thus shortening the water operation outside the heat transfer tube in natural convection region and improving the PRHR HX performance.

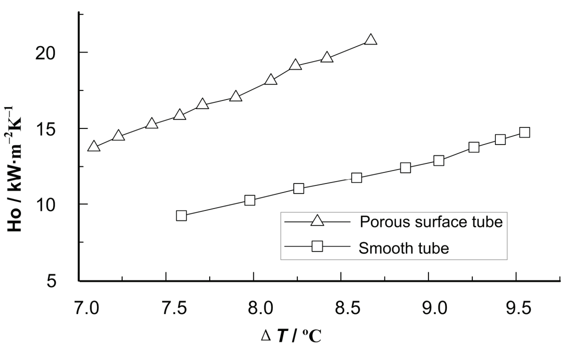

Fig.5 Relations of boiling heat transfer coefficient with superheat.

Achieving the saturation in boiling tank, the saturated boiling occurred around the entire tube surface. As shown in Fig.5, the boiling heat transfer coefficient is about 1.75 times higher in the porous surface tube than in the smooth tube at 7.5ºC superheat; and 1.68 times higher at 8.5ºC. Furthermore, the porous surface tube has a steeper boiling curve, hence the enhanced boiling heat transfer. This shows that the boiling heat transfer can be intensified by increasing the active nucleation sites of vapor generation and the heat supply to vapor bubbles. The porous surface tube has more cavities over a wide range of radius, so the nucleate boiling heat transfer capability, strongly dependent on the active nucleation sites density, was enhanced with the superheat. Parts of cavities could just be activated with a certain wall superheat. Also, some inactive cavities at low superheat could be activated with the superheat increasing on the porous surface.

Figure 6 shows the relationships of average wall temperature with heating steam pressure in the two tubes. The porous surface tube maintaining boiling is 1.5ºC lower than the smooth tube under the same steam condensation pressure, indicating that parts of activated cavities on porous surface stably generate bubbles at low wall superheat. So pool boiling on the porous surface tube occurred earlier, and the outer wall temperature decreased because of the heat transfer coupling of boiling and condensation outside and inside tube. Then, the condensation temperature difference inside the porous surface tube is larger than the smooth tube, because the total heat transfer temperature difference maintained constant under a steam condensation pressure.

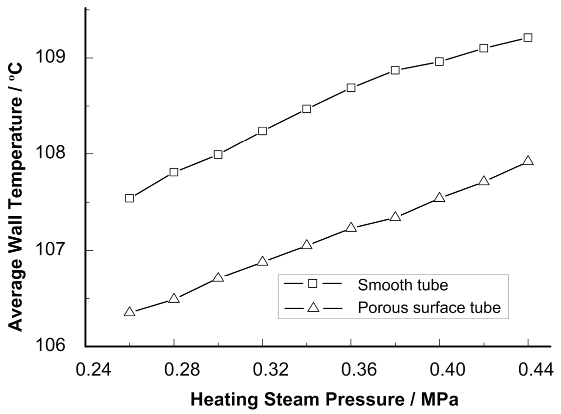

Fig.6 Relations of average wall temperatures with heating steam pressure.

The uniform electric heating with convenient control has been wildly used in boiling experiments. In operating the PRHR HX, we used the steam condensation inside tube as heating source to simulate the heat transfer mechanisms, including convection or condensation inside tubes, thermal conductivity through tube wall, and nucleate boiling outside tubes. Because the condensate film thickness increases along the flow direction, the heating condition is far more complex than electric heating without uniform heat flux and wall temperature. The four thermocouples mounted on the smooth or porous surface tube were used to analyze the local heat transfer by the wall temperature distribution. The steam condensing inside vertical tube, which is similar to quiescent vapor onto a vertical plate, was affected by the vapor shear stress at the vapor/liquid interface. The large interfacial shear stress cpuld modify the film condensation at high vapor rates, accelerating the drainage of condensate, thinning the condensate film, and enhancing heat transfer at down flow of vapor.

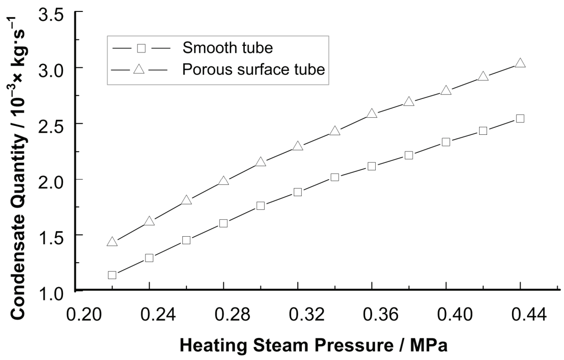

Fig.7 Condensate quantity change with heating steam pressure.

Figure 7 shows the steam condensation is higher inside porous surface tube than inside smooth tube because of intensive boiling heat transfer. As the steam pressure increased from 0.22 to 0.46 MPa, the condensate quantity in smooth tube increased from 1.14×10^{-3} to 2.63×10^{-3} $\mathrm{kg{\cdot}s^{-1}}$; and in the porous surface tube, from 1.43×10^{-3} to 3.14×10^{-3} $\mathrm{kg{\cdot}s^{-1}}$. According to the law of mass conservation, the steam velocity at inlet of the smooth tube is 8.17–9.43 $\mathrm{m{\cdot}s^{-1}}$; and at inlet of porous surface tube, 10.24–11.26 $\mathrm{m{\cdot}s^{-1}}$.

Figure 8 shows the local outside wall temperature distribution at the four thermocouple points (T_w1, T_w2, T_w3 and T_w4) for the smooth and porous surface tubes, under heating steam inlet pressures of 0.26, 0.34, and 0.42 MPa. For T_w1, at 100 mm from the tube top, growing bubbles went off quickly at lower parts of the tube because of intense liquid agitating. Besides, T_w1 is the nearest to the steam inlet, with the thinnest condensate film and the lowest condensation thermal resistance, the heat transfer is the best at the low wall temperature. So a porous surface tube with more active cavities has lower wall superheat maintaining boiling and wall temperature. The wall temperature at T_w2 rose for the two tubes, especially the porous surface tube. As thickness of the condensate film adhering on inner surface of tube increases gradually along steam flow direction, the condensation thermal resistance was higher than that at T_w1. The T_w4 at the tube bottom is close to the condensate inlet back to the boiling tank, hence a maximum condensation thermal resistance of the inner tube. The sub-cooling of condensate, which returns to the tank bottom, was about 0.3ºC because of the heat loss in the condenser and pipes, resulting in sub-cooled boiling. So the wall temperature at T_w4 is the highest. Because the water in the T_w3, above the T_w4, can generate the saturated pool boiling after heating up to saturation, its wall temperature at T_w3 is lower than at T_w4.

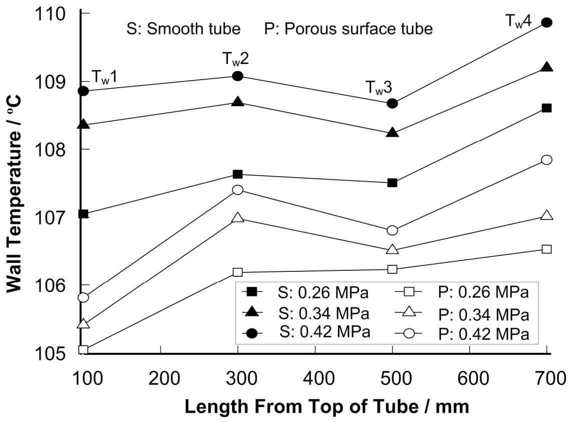

Fig.8 Local wall temperature distribution of smooth tube and porous surface tube.

4 Mechanism of enhanced boiling on porous surface tube

The numerous micro-cavities on the porous surface tube are interconnected each other by sub-tunnels, and the wall superheat decreases by increasing the bubble growth frequency, because the cavities and tunnels can trap some steam as activating the embryo nucleus of the next bubble.

The boiling on the porous surface tube reveals that the bubbles are mainly generated from the activated cavities at the intersection of sub-tunnels within a certain wall superheat, and number of the activated cavities increase with the wall superheat. But some cavities are inactivated at the high wall superheat. To understand the boiling heat transfer mechanism, two isolated bubbles generated on the porous wall at low heat flux were photographed (Fig.9). While the Bubble I was growing up, the Bubble II was grown up but did not detach from the wall yet. The brighter regions are the interface between their bottom and liquid micro-layer, and the bottom of brighter region of Bubble II is larger than that of Bubble I, indicating that the bubble grows up from the embryo nucleus. Also, the vapor/wall interface filter into the sub-tunnels was filled with liquid, and flew along the tunnel due to surface tension effect. When the bubble departs form the wall, as the steam in the tunnel does not resist the external liquid pressure, liquid flows into the tunnel, and the vapor/liquid interface of residual steam shrinks towards the active cavity. So the liquid reciprocating around the sub-tunnel is significant for boiling enhancement.

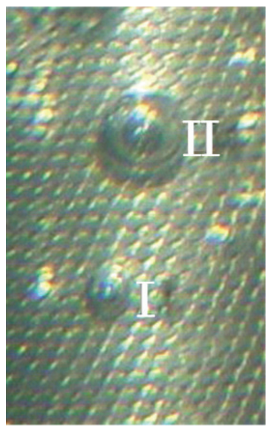

Fig.9 Growth of isolated bubble at low heat flux.

According to bubble generation during boiling heat transfer, the entire porous wall can be divided into three sections: the active cavities, the inactive cavities and their interconnected sub-tunnels. Inspired by cobweb, the porous geometries can be seen as a large porous network[8], as shown in Fig.10.

The black solid circles are porous cavities, and the circle diameters are its activation capacity. The dashed grid and solid rectangle are the sub-tunnels and their boundary. The activate cavities generate the bubbles at suitable wall superheat, and the liquid surrounds the other region of porous wall. The bubble bottom is filtered into the sub-tunnels by increasing its radius, some superheated microfilm evaporating on the bubble interface is extruded from the inactivate cavities to liquid pool, and the residual steam pressure captured by sub-tunnels decreases faster than that of the liquid pool[9]. The residual steam shrinking towards activate cavities causes the liquid to freely flow into or out of sub-tunnels through inactivate cavities, and the tunnel wall incessantly heats the microfilm. So, the activate cavities on porous wall are more sufficient due to the higher microfilm superheat. Also, the bubbles growing and departing at activate cavities cause pressure fluctuation and flow pulsation periodically in the sub-tunnels, and promotes the liquid exchange between sub-tunnels and liquid pool, to enhance the liquid-phase convection at the non-embryo region on porous wall due to flowing freely into or out of sub-tunnels.

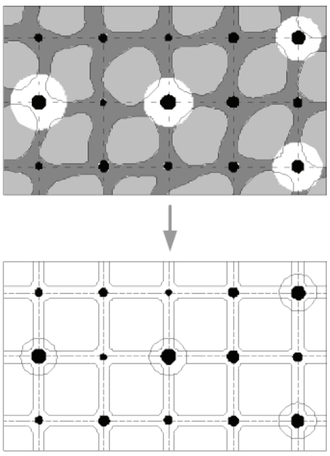

Fig.10 A schematic of boiling on porous wall.

5 Conclusions

The nucleate pool boiling characteristics on smooth or porous surface tube have been conducted. Before reaching saturation liquid temperature, the heat is mainly transferred to tank water by natural convection, and a strong thermal stratification with the different density is formed because the cold and hot liquid greatly restricts the heat transfer. The porous surface tube can enhance the heat transfer of natural convection and shorten the time, and the liquid saturation temperature is reached, thus improving the PRHR HX performance.

The pool boiling heat transfer is enhanced by the porous surface tube, and its coefficient at the heat flux of up to 1.68–1.75 times higher than that of smooth tube. The wall superheat decreases by 1.5ºC, thus keeping nucleate boiling.

Because of the coupling between the nucleate boiling and condensation outside and inside tube, the uniform and linear axial wall temperature along the heat transfer tube shows an undulated distribution.

The active or inactive cavities, and the sub-tunnels form a complete porous network. Bubbles are generated in the active cavities, and the liquid can freely flow into and out of the sub-tunnels through the inactivate cavities, heating the tunnel wall up to superheat. In the active cavities, the inactive cavities and the sub-tunnels, the porous surface tube can greatly enhance the boiling heat transfer.

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