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A condensation experiment in accumulated conditions of noncondensable gas inside a vertical tube

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1. Introduction

It has been well known that even a small amount of noncondensable gas can reduce condensation heat transfer considerably. In the condenser tube the condensate flows as an annular liquid film adjacent to the tube wall and steam-noncondensable gas mixture flows in the core region. Then the reduction is mainly caused by a noncondensable gas layer formed adjacent to the liquid-gas interface.

Recently, several researchers have been performed for the condensation in the presence of noncondensable gases taken place inside a vertical tube. Siddique et al.[1], Araki et al.[2], Kuhn et al.[3], and Park and No[4] performed the condensation experiments under the presence of air or helium with the secondary jacket cooling method. On the other hand, Kim and No[5] and Oh and Revankar[6] conducted steam-air condensation experiments with the secondary pool boiling condition.

To calculate the local heat flux, Siddique[1] measured the local coolant bulk temperatures at the midpoint of the annular gap, and a small amount of air was injected through the annulus with the coolant to enhance turbulence and mixing. Kuhn[3] and Park[4] calculated them numerically by measuring temperatures at the inner and outer walls of the annulus. Araki[2] and Kim[5] determined the local heat flux from the measurement of the temperature gradient of the tube wall.

This research has been performed to study the heat transfer characteristics for PRHRS condensation heat exchanger in a SMART. The remaining heat is removed from the core passively by PRHRS in a period of serious accidence of the SMART.

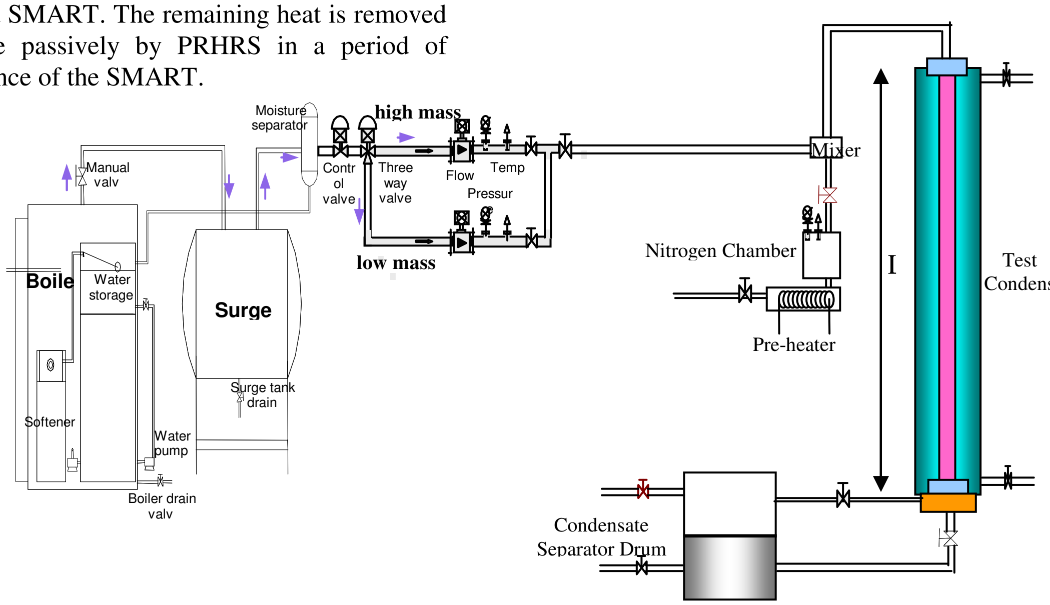
The experimental conditions of this study are similar with those of previous researches, except the noncondensable gas is accumulated and remained inside the vertical tube because the condensate fills the bottom of tube. In the previous researches, the noncondensable gas was flowing with constant flow rate. Therefore the heat transfer characteristics in the accumulated condition of noncondensable gas should be studied newly.

2. Experiment

2.1. Experimental facilties

The schematic of the test apparatus is shown in Figure 1. The experimental facilities consisted of a steam generator part, a steam flow rate control part, a steam-nitrogen mixing part, a test section, and data acquisition system. The inner condensing tube of the test section is made of type 304 stainless steel pipe with 13 mm inside diameter, 2.5 mm thickness, and 3 m length. A 40 mm inside diameter concentric acryl jacket surrounden the test condenser. At 13 different axial locations, each K-type thermocouple is silver welded onto the outer surface of condensing tube to measure the outer wall temperatures, and welded through the tube to measure the bulk temperatures. The local coolant temperatures are measured at 11 different axial locations with bubbling small amount of air because of the same reason with Siddique's[1].

Figure 1. Schematic diagram of experimental facility



2.2. Data anaysis

The local heat flux is obtained from the slope of the coolant temperature profile.

$$q''(x) = -\frac{\dot{m}_{cw} C_p}{\pi d_i} \frac{dT_{cw}(x)}{dx} \quad (1)$$

The local inner wall temperature is calculated using the measured outer wall temperature and the obtained heat flux.

$$T_{w,i}(x) = T_{w,o}(x) + q''(x) \cdot \frac{r_i \ln(r_o / r_i)}{k_{sus}} \quad (2)$$

Then the local heat transfer coefficient is given by the local heat flux and local temperatures.

$$h(x) = \frac{q''(x)}{(T_b - T_{w,i})} \quad (3)$$

3. Result and discussion

At first pure steam tests were run at flow rate from 7 to 19 kg/h. Figure 2 shows the temperature distribution at flow rate 9.7 kg/h, which is the expected operating flow rate of PRHRS. It shows that the condensation was completed between 1.15 m and 1.4 m and there existed a boundary between two-phase region and single liquid phase region. In this case, only the data upstream of the boundary were used in fitting the coolant bulk temperature profile for calculation of the heat flux. Figure 3 shows the local heat transfer coefficients increase as the inlet steam flow rates increase.

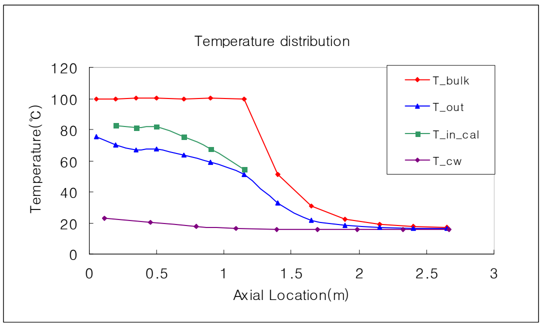


Figure 2. Temperature distribution for 9.7 kg/h

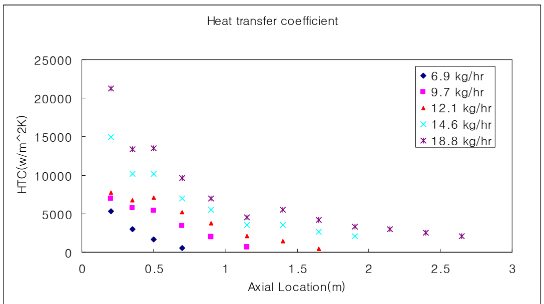


Figure 3. Heat transfer coefficients of pure steam tests

In order to study the heat transfer characteristics in the accumulated condition of noncondensable gas, the nitrogen gas about 10% tube volume was added after the two- or single-phase boundary was made inside a vertical tube at steam flow rate 9.5 kg/h approximately. After about 30 min., the data was obtained. This process was repeated 12 times. So the added total volume of nitrogen was 120% tube volume. However, there is no special difference between pure steam and nitrogen accumulated cases, as shown in Figure 4. The reason is the solubility of nitrogen. The liquid film has large surface area, so the nitrogen could be soluble in water easily. The amount of dissolved nitrogen in condensate during 30 min. is almost same with it of added nitrogen gas. Figure 4 also shows the results in the similar test condition with previous researches. To study more clearly, the transient tests during 30 min. are being performed with various steam flow rates and added nitrogen volumes.

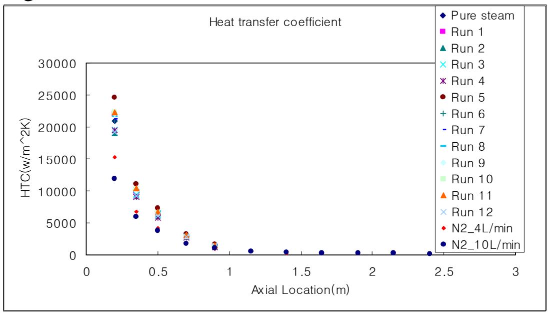


Figure 4. Heat transfer coefficients at accumulated condition

4. Conclusion

The research has been performed to study the heat transfer characteristics for PRHRS condensation heat exchanger in the SMART. In the study of accumulated conditions of noncondensable gas, the solubility of noncondensable gas should be considered.

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