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Evaluation of the Natural Circulation Flow Loop with Inclined Downward Heating Channel

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1. Introduction

Various safety systems are designed and adapted in nuclear power plants to prevent postulated accidents, to enhance the life time and economic benefit, and to increase a public acceptance of the plants. Postulated severe core damage accidents have a high threat risk for the safety of human health and jeopardize the environment. Versatile measures have been suggested and applied to mitigate severe accidents in nuclear power plants as recently presented by Rempe *et al*. [1]. To improve the thermal margin for the severe accident measures in high-power reactors, engineered corium cooling systems involving boiling-induced two-phase natural circulation have been proposed for decay heat removal [2]. The boiling-induced natural circulation flow is generated in a coolant path between a hot vessel wall and cold coolant reservoir. In general, an increase in the natural circulation mass flow rate of the coolant leads to an increase in the critical heat flux (CHF) on the hot wall, thus enhancing the thermal margin [3].

An ex-vessel core catcher under consideration, which is one of the engineered corium cooling systems, is a passive system consisting of an inclined engineered cooling channel made of a single channel between the body of the core catcher and the inside wall of the reactor cavity [4]. Under severe accident conditions, water is supplied from the IRWST to the engineered cooling channel. The water in the inclined channel absorbs the decay heat transferred from the corium through the carbon steel structure of the core catcher body and boils off as steam. The latter is subsequently released into the free volume of the containment above the corium spreading compartment. Water continues to flow from the IRWST to the cooling channel as a result of buoyancy-driven natural circulation. The engineered cooling channel is designed to provide effective long-term cooling and stabilization of the corium mixture in the core catcher body while facilitating steam venting. To maintain the integrity of the ex-vessel core catcher, however, it is necessary that the water coolant be circulated at a sufficiently high rate through the inclined cooling channel for decay heat removal by downward facing boiling of the water circulated from the IRWST. KAERI performed the experimental study to evaluate the cooling performance of ex-vessel core catcher system with inclined downward facing heating surface.

Boiling-induced natural circulation flow experiments in the cooling channels of the ex-vessel core catcher are introduced. The natural circulation flow are also evaluated analytically by considering the mass, momentum, and energy equations

2. Natural circulation flow experiments

Boiling-induced natural circulation flow experiments in the cooling channels of the ex-vessel core catcher are investigated. A scaling analysis is applied to design the test facility compared with the prototypic core catcher cooling system. As the geometry and heating wall heat flux of the heating channel of the test facility will be the same as those of the prototypic core catcher cooling system except for a reduced width of the heating channel, assume that the axial distributions of the coolant quality (or void fraction) between the prototype and model facility are expected to resemble each other. Thus, using this fact, the down-comer piping design characteristics of the modeled experimental facility can be determined from the relationship derived from the scaling analysis.

Figure 1 shows a schematic drawing of the experimental facility for the natural circulation flow simulation of the prototypic ex-vessel core catcher. As shown in Fig. 1, the horizontal length of the experimental facility is 3 m, that is, the half section of the core catcher is simulated. The gap size of the cooling channel is 0.1m as is the gap between the prototypic core catcher body and concrete body. Seven short columnar structures, called studs, each with a dimension of 0.07m×0.1m×0.1m(width× length×height), are placed in the cooling channel gap, and the diameter are scaled to coincide with the flow blockage ratio of the prototypic ex-vessel core catcher. The gap of the cooling channel is determined appropriately such that the cooling channel also has an inclination angle of 10 degrees as is the actual core catcher system to facilitate the steam venting. A water tank is also installed to supply static pressure to the cooling channel.

The measuring parameters during the test are the input power using a power-meter, the total circulation mass flow rate of the coolant using a flow meter, the local coolant temperature using T-type thermocouples, the local heated wall temperature using K-type thermocouples, the local pressure using pressure transducers, the water level in the coolant tank using a level transmitter, the estimated void fractions using vertical differential pressure transmitters, and a visualization of the flow pattern in the coolant channel using a camera.

The experimental parameters are the heat flux from the heating block to the coolant, the inlet coolant subcooling, and the water level in the coolant tank.

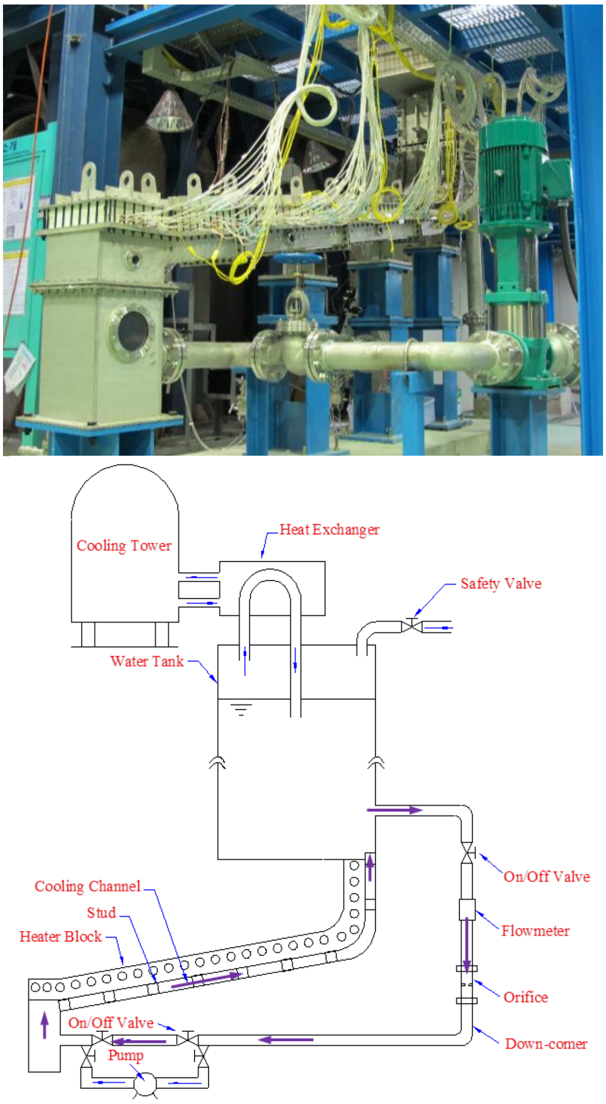

Fig.1. Schematic and photograph of natural circulation experimental facility.

3. Analysis of two-phase natural circulation flow

The two-phase flow is analyzed to predict the natural circulation mass flow rate occurring in the engineered corium cooling system [3]. Assuming the flow to be at a steady state in the coolant channel, the mass, momentum, and energy equations can readily be formulated. Since no mass is being added to the flow from outside the channel other than at the inlet, the overall mass flow rate is the sum of the liquid and vapor mass flow rate as given by equation (1).

$$\dot{m} = \rho_m u_m A = \dot{m}_f + \dot{m}_g \qquad (1)$$

The momentum equation is rearranged by using force balances, that is, the pressure difference along the vertical direction can be represented as the sum of the inertia force, gravitational force, wall friction loss induced by the flow, form loss by the geometric change of the flow path, and the flow loss due to a two phase retardation such as the velocity difference between the liquid and vapor phase as given by equation (2).

$$-\frac{dP}{dz} = \rho_m u_m \frac{du_m}{dz} + \rho_m g_z + \left(-\frac{dP}{dz}\right)_{fr} + \left(-\frac{dP}{dz}\right)_{fo} + \left(-\frac{dP}{dz}\right)_{tp} \qquad (2)$$

In this study, the flow loss due to two phase retardation is ignored since the two-phase pressure loss is usually much smaller than other loss terms [5]. If the energy losses are ignored through the flow channel, the energy equation can simply be represented by a balance between the flow enthalpy change and the heat input through the heated channel wall as given by equation (3).

$$\rho_m u_m A \frac{dh_m}{dz} = \left(\frac{d\dot{Q}}{dz}\right)_{in} \qquad (3)$$

The wall friction loss induced by a one- or two- phase flow in a dynamic equilibrium state can be described by equation (4), assuming only the liquid or the vapour phase to be flowing in the original channel with their respective mass flow rates

$$\left(-\frac{dP}{dz}\right)_{fr} = \frac{f_v}{2D_h} \rho_m u_m^2 \qquad (4)$$

The friction factor, *f*, depends on the Reynolds number of each phase. The values of *C* and *m* for evaluating the friction factor given by equation (5) depend on the type of flow taking place inside the channel.

$$f_a = C\,\mathrm{Re}_a^{-m} = C\left(\frac{\rho_a u_a D_h}{\mu_a}\right)^{-m} \qquad (5)$$

The values of *C* and *m* are 64 and 1 for *Re* less than 2300, and 0.316 and 0.25 for *Re* greater than 2300.

The mixture quality is defined as the flow enthalpy change due to the wall heat input as given by equation (6).

$$x \equiv \frac{h - h_f}{h_{fg}} = \frac{(h_{inlet} + \Delta h) - h_f}{h_{fg}} = \frac{(h_{inlet} - h_f) + \frac{1}{\dot{m}}\int q''\xi dz}{h_{fg}} \qquad (6)$$

A method for predicting the void fraction is essential for predicting the acceleration and gravitational components of the pressure gradient in the two phase flow. Butterworth formula [6] has shown that several of the available void-fraction correlations can be cast in the general form given by equation (7).

$$\alpha = \left[1 + B_B \left(\frac{1-x}{x} \right)^{n_1} \left(\frac{\rho_g}{\rho_f} \right)^{n_2} \left(\frac{\mu_f}{\mu_g} \right)^{n_3} \right]^{-1} \qquad (7)$$

The values of the various constants in this relation corresponding to different correlations are suggested [7].

As the void fraction is calculated by equation(7), the mixture density can be also obtained by equation(8).

$$\rho_m = \alpha \rho_g + (1-\alpha)\rho_f \qquad (8)$$

If the momentum equation is integrated over the entire circulating flow loop from the inlet to the outlet and then back to the inlet, the result must be zero as shown in equation (9).

$$\oint \left(-\frac{dP}{dz} \right) dz = \oint \rho_m u_m \frac{du_m}{dz} dz + \oint \rho_m g_z \, dz + \oint \left(-\frac{dP}{dz} \right)_{fr} dz + \oint \left(-\frac{dP}{dz} \right)_{fo} dz = 0 \qquad (9)$$

If the form loss term in equation (9) is formulated by the mixture velocity, the only unknown in equation (9) is the mixture velocity with the assistance of equation (4) through (8). Each term in equation (9) is numerically integrated along with the natural-circulation flow loop by assuming the initial mixture velocity. A final mixture velocity that satisfies equation (9) can be obtained using a trial-and-error method. If the mixture velocity is found by solving equation (9), and the natural circulation mass flow rate can then be calculated by equation (1).

4. Discussion

The equation (9) can be solved numerically to get the circulation mass flow rates of the experimental facility. Figure 3 shows the assumed heat flux distribution imposed by the molten corium in the core catcher which was estimated considering the natural convection of the molten corium in the core catcher.

To calculate the circulation mass flow rate, the void fraction model as mentioned in equation (8) should be obtained. However, there is no void fraction model which can be applied to a rectangular cross-section with the downward facing heating surface. Therefore, the measured void fraction at the exit of the heating channel was correlated with the exit quality as shown in equation (10).

$$\alpha = \left[1 + B_B \left(\frac{1-x}{x} \right)^{n_1} \right]^{-1} \quad (1.29 \le P \le 1.73) \qquad (10)$$

The constants B_B and n_1 in the equations (10) can be determined with the experimentally obtained data of the void fractions and qualities at the exit of the heating channel by using the least-square-fitting method. The exit void fraction was predicted by the equation (10) with fitting values, $B_B = 0.5$, $n_1 = 0.29$ as shown in Fig. 4. Figure 5 shows that the errors of the estimated void fractions with the developed correlation are bounded within about 15 percent, and this therefore shows that the void fraction can be calculated based on the developed void fraction correlation.

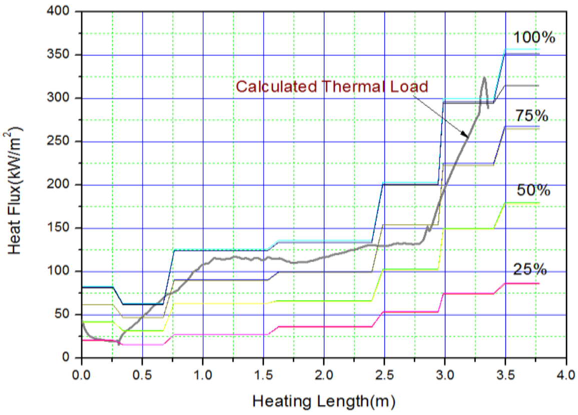

Fig.3. Heat flux distribution imposed on the cooling channel.

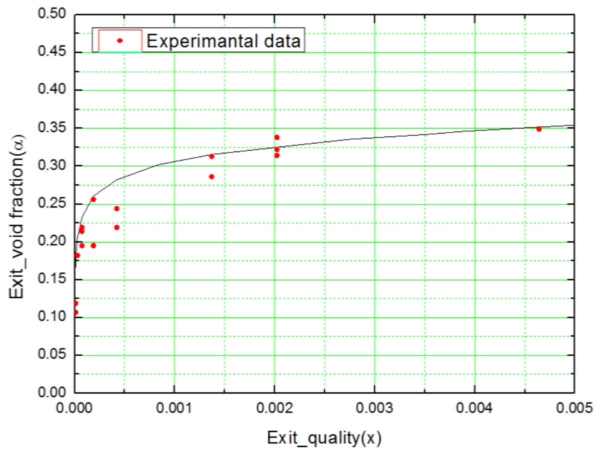

Fig.4. Variation of the exit void fraction predicted by analysis and experimental data.

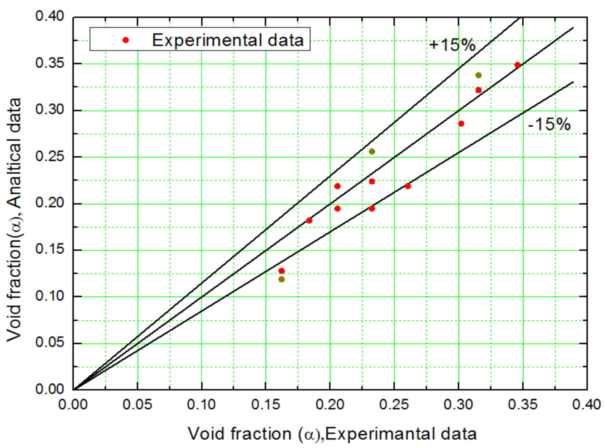

Fig.5. Error bounds of the predicted void fraction.

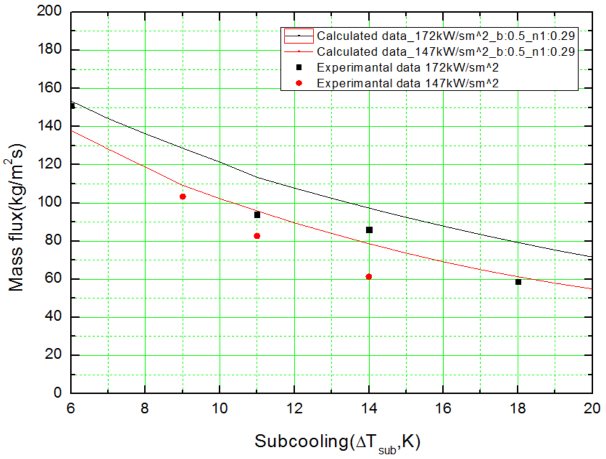

Fig.6. Comparison of the analysis circulation mass flux with experiments (water level 6.388m, 1.59bar).

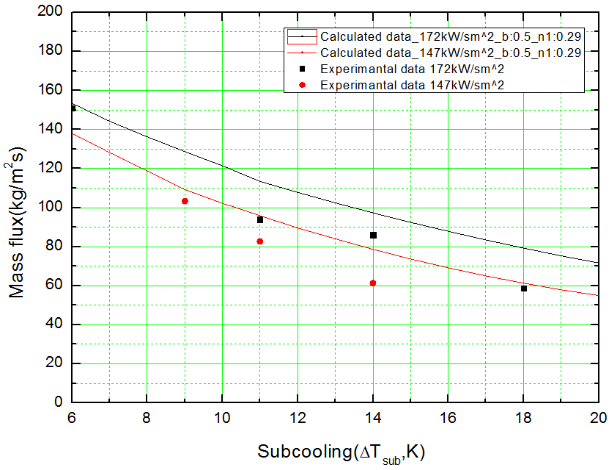

Fig.7. Comparison of the analysis circulation mass flux with experiments (water level 8.388m, 1.8bar).

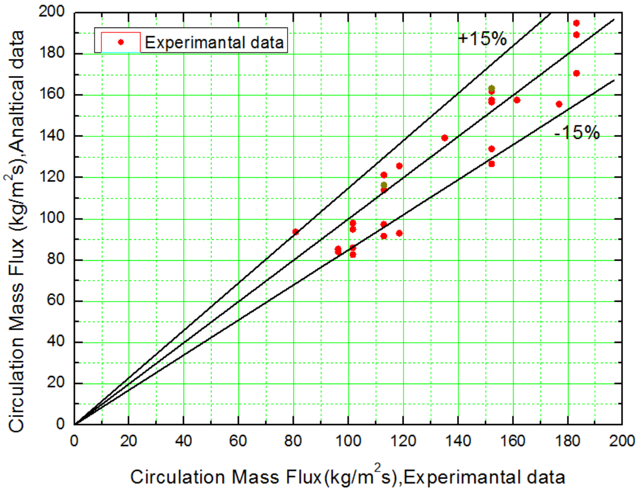

Fig.8. Error bounds of the predicted circulation mass flow rates.

Figure 6 and 7 shows the calculated circulation mass flux along with the coolant inlet subcooling and wall heat flux. As the coolant temperature and heat flux increase, the circulation mass fluxes also increase. In the high sub-cooling cases, the boiling point is retarded on the height, and the circulation mass flux then decreases. The induced natural circulation mass flux predicted by the present analytical model is also compared with the experimental data in Fig. 6 and in Fig. 7. Figure 8 shows that the errors of the predicted circulation mass fluxes compared with the experimental ones are bounded within about 15 percent, and this therefore shows that the circulation mass flux can be evaluated successfully using the suggested analysis method and void fraction model.

5. Conclusion

The boiling-induced natural circulation flow in the cooling channel of the core catcher has been evaluated experimentally and analytically. A scaling analysis is applied to design the test facility compared with the prototypic core catcher cooling system. The natural circulation flow experiments were performed along with the inlet subcooling, wall heat flux, and water level. As the coolant temperature and heat flux increase, the circulation mass fluxes also increase.

An analytical study was also performed to predict the natural circulation mass flux by considering the conservations of mass, momentum, and energy in the two-phase mixture, along with the two-phase friction drop and void fraction with inclined downward facing heating surface. The void fraction model with inclined downward facing heating channel was suggested by the least-square-fitting method using experimental data of void fractions and qualities at the exit of the heating channel. The naturally circulation mass flux in the cooling channel with the inclined downward facing heating surface could be evaluated successfully using the suggested analysis method and void fraction model within 15 percent error bound compared with experimental data.

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