



Nuclear Science and Technology

Journal homepage: <https://jnst.vn/index.php/nst>

Assessment of steam condensation model with the presence of non-condensable gas in a vertical tube using RELAP5 Mod 3.2 code and MIT exp. Data

V.T., Nguyen*, H.T., Trinh

School of Nuclear Engineering and Environmental Physics, Hanoi University of Science and Technology

**e-mail: thai.nguyenvan@hust.edu.vn*

Abstract: The non-condensable gas effect is a primary concern in some passive systems used in advanced design concepts, such as the Passive Residual Heat Removal System (PRHRS) of AP1000, APR1400, AES-2006, the Passive Containment Cooling System (PCCS) of AP1000 design, and Isolation Condensation System (ICS) of ESBWR design. The accumulation of the non-condensable gas inside the condensing tubes can significantly reduce the level of heat transfer which affects the heat removal capacity in accident condition and impacts plant safety. The objective of the present work is to assess the analysis capability of two wall film condensation models of RELAP5/Mod3.2 with the presence of non-condensable gas in a vertical tube on condensation experiments performed at MIT, USA. The results of the simulations and experimental data show the similar tendencies that the heat transfer coefficients increase as the inlet steam-non condensable gas mixture flow rate increases, the inlet steam-non-condensable gas mass fraction decrease, and the inlet saturated steam temperature decrease.

Keywords: *Steam condensation, non-condensable gas, heat transfer, passive system*

I. INTRODUCTION

Condensation heat transfer is a primary concern in passive systems used in advanced plants to increase the inherent safety such as the Passive Containment Cooling System (PCCS) of AP1000 design, the Isolation Condensation System (ICS) of ESBWR design, and the Passive Residual Heat Removal System in AP1000, APR1400, AES-2006 (Fig. 1 and 2)[1-2]. Even a small amount of non-condensable gas can significantly reduce the level of heat transfer. When condensation occurs at the interface of a liquid film on the wall of a vertical tube, a non-condensable gas will accumulate and form a non-condensable gas layer. This increases the non-condensable gas concentration at the interface between the liquid film and gas, which in turn reduces the condensation heat transfer rate. In these systems, condensation heat transfer in vertical

tubes is the main heat transfer mechanism, and non-condensable gases can be present. A lower condensation heat transfer rate causes the performance of the heat exchanger to deteriorate, which affects the heat removal capacity in accident conditions and impacts plant safety. It can be also important in a Pressurized Water Reactor (PWR). For example, in a Small Break Loss Of Coolant Accident (SBLOCA), the steam produced in the core can condense in the steam generator tubes through the secondary system cooling. The heat transfer rate in this situation can alter the accident progression and cause reflux condensation and re-criticality [3].

Several experimental studies have been performed to examine condensation in the presence of a non-condensable gas in a vertical tube. The research background has been used to support the design of a passive system.

Summary of experimental facilities and test matrix was presented by Bang et al. (2009) [3] and reproduced in Table I. The performance of condensation heat transfer models in presence of air has been investigated by Park and No (1999) and Bang et al. (2009) [3-4]. The results

shown that the default model used in MARS and RELAP5 codes did not predict accurately heat transfer coefficient. The alternative models are then proposed considering the interfacial shear stress and flow condition determination criterion.

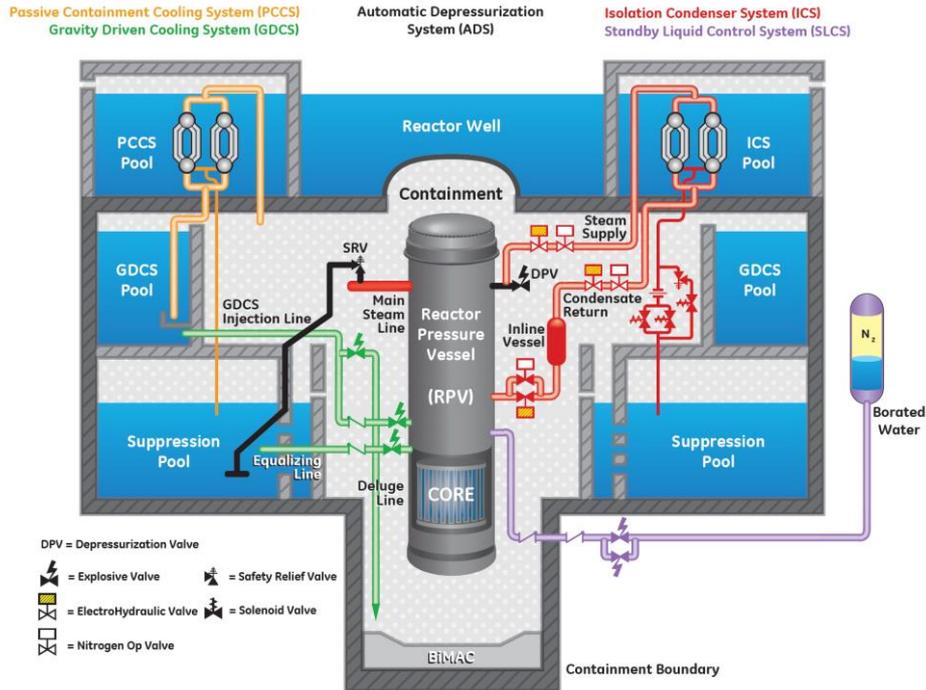


Fig. 1. Passive Safety System of ESBWR Design [1]

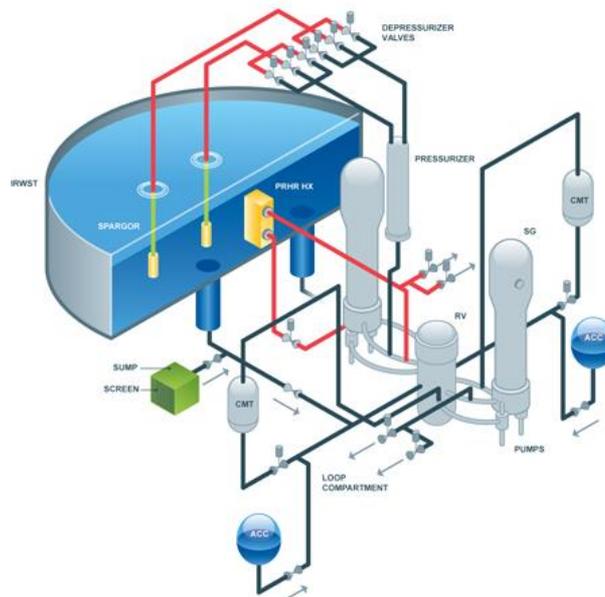


Fig. 2. Passive Safety System of AP1000 Design [2]

Unfortunately, no investigation has been reported to assess the effect of hydrogen on steam condensation models implemented in system analysis code like MARS, RELAP5. In case of a severe accident in a Light Water Reactor, significant amounts of hydrogen may be generated due to the metal water reaction in the reactor core. This actually happened in the nuclear accident at Three Mile Island. The hydrogen subsequently collected in the steam condenser and along with air severely inhibited the condensation process. However, it is much safer to handle helium compared to hydrogen

because hydrogen's potential for combustion. Helium is then used to indicate the effects of hydrogen because of the similarities in the thermo-physical properties of two gases. Due to that reasons, steam condensation experimental data of MIT in the presence of helium was selected for assessing the steam condensation models implemented in RELAP5 Mod3.2. The capability and applicability of the RELAP5 code for predicting the condensation heat transfer in a vertical tube with a non-condensable gas (Helium) were investigated in this studies.

Table I. Condensation Experiments in a Vertical Tube with Non-condensable Gases

| | Goodykoontz (1964) [6] | Siddique (1993) [7] | Kuhn (1997) [8] | Park (1999) [4,5] | Lee (2008) [9] |
|-----------------------------|------------------------|---------------------|-------------------|-------------------|-------------------|
| | NASA | MIT | UCB | KAIST | POSTECH |
| Tube length (m) | 2.15 | 2.54 | 2.4 | 2.4 | 2.8 |
| Tube ID (mm) | 15.875 | 46 | 47.5 | 47.5 | 13 |
| Tube Thickness (mm) | 1.58 | 2.4 | 1.65 | 1.65 | 2.5 |
| Secondary jacket ID (mm) | 50.8 | 62.7 | 76.2 | 100 | 40 |
| Non-condensable gas | - | Air, Helium | Air, Helium | Air | Nitrogen |
| Secondary cooling | Forced Convection | Forced Convection | Forced Convection | Forced Convection | Forced Convection |
| Steam flow (g/s) | 4.2 - 14.7 | 2.4-8.9 | 8.2-17 | 2-11 | 1.8-7.8 |
| Inlet air mass fraction (%) | - | 10-35 | 0-40 | 10-70 | 0-40 |
| Pressure (MPa) | 0.1 | 0.1-0.5 | 0.1-0.5 | 0.17-0.5 | 0.1-0.13 |
| HTC (W/m ² K) | 4485-11867 | 100-25000 | 500-13000 | 100-7000 | 300-27900 |

II. CONDENSATION HEAT TRANSFER MODEL IN VERTICAL TUBE

Condensation is defined as the removal of heat from a system in such a manner that vapor is converted into liquid. This may happen when vapor is cooled sufficiently below the saturation temperature to induce the nucleation of droplets. Such nucleation may occur homogeneously within the vapor or

heterogeneously on entrained particulate matter. Heterogeneous nucleation may also occur on the walls of the system, particularly if these are cooled as in the case of a surface condenser. In this latter case there are two forms of heterogeneous condensation, drop-wise and film-wise, corresponding to the analogous cases in evaporation, of nucleate boiling and film boiling. Film-wise condensation occurs on a cooled surface which

is easily wetted. On non-wetted surfaces the vapor condenses in drops which grow by further condensation and coalescence and then roll over the surface [11]. The tube inside wall is at a prescribed varying temperature T_w , lower than the saturation temperature of steam, and therefore condensation take place. In general, during forced in-tube condensation of a vapor, the condensed liquid flows as an

annular film adjacent to the cooled tube wall while the uncondensed vapor flows through the tube core. The high density difference between the condensed liquid and the gaseous core lead to a very low liquid volumetric fraction and together with the shear force of the gaseous core an annular flow pattern is maintained over most of the condensing tube length.

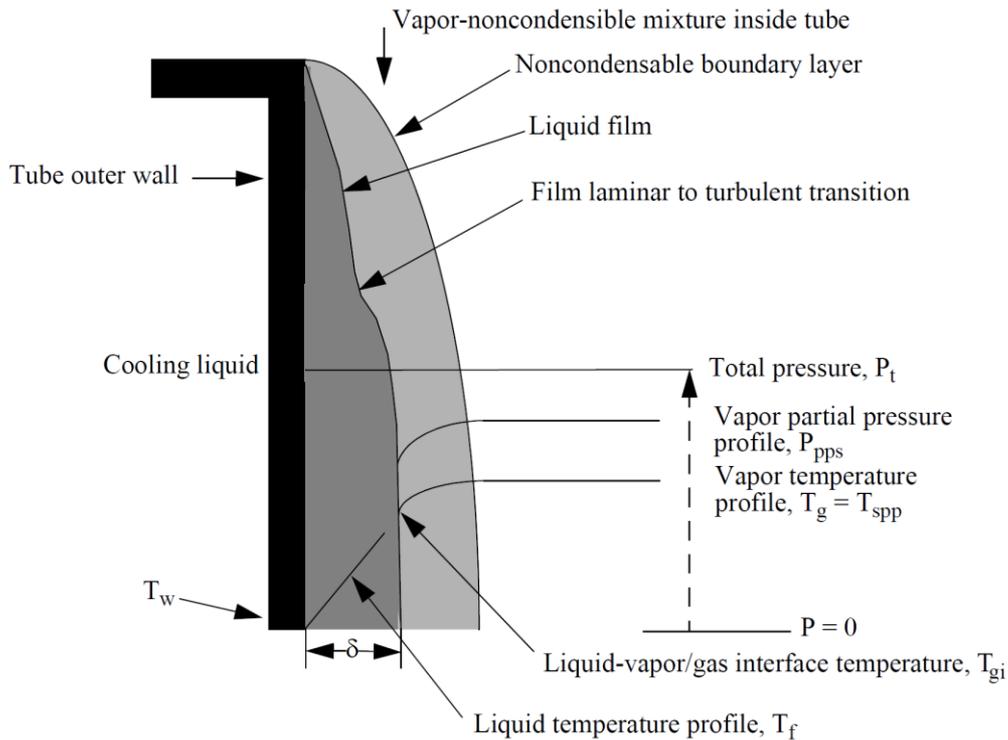


Fig. 3. Film-wise Condensation Model [10]

The presence of even a small quantity of non-condensable gas in the condensing vapor has a profound influence on the resistance to heat transfer in the region of the liquid-vapor interface. The non-condensable gas carried with the vapor towards the interface where it accumulates. The partial pressure of gas at the interface increases above that in the bulk of the mixture, producing a driving force for gas diffusion away from the surface. This motion is exactly counterbalanced by the motion of the vapor-gas mixture towards the interface. This

situation is illustrated in Fig.3 which also shows the variation of temperature in the region of the interface. It is usual to assume that the temperature at the interface corresponds to the saturation temperature equivalent to the partial pressure of vapor at the interface. For the theoretical analysis of the heat and mass transfer during condensation of a vapor in the presence of a noncondensable gas, either the boundary layer analysis or the heat and mass transfer analogy methods are generally employed. In both of these methods

the condensation process is viewed as occurring in two interacting boundary layers - the vapor/gas and condensate boundary layers. The boundary layer solutions currently available deal primarily with the flat plate configuration and stagnant atmospheric conditions. Extension to a tube geometry with turbulent flow is not straight forward. The heat and mass transfer analogy models follows the general methodology of Colburn and Hougen, who were the first to develop a stepwise iterative solution method for predicting the condensation heat transfer rate from a vapor/noncondensable gas mixture. Their main equation was based on a heat balance at the liquid/gas interface, where the heat transferred from the gas/vapor boundary layer is equated to the heat transferred through the condensate film. The heat transfer from the gas phase was viewed as made up of the sensible cooling of the uncondensed gas and the contribution due to mass transfer, that is, the latent heat due to condensation of the diffused vapor at the interface. Separate models for the sensible and latent heat fluxes were used. Mass transfer coefficients were obtained from an analogy with heat transfer. The unknown interface temperature was solved iteratively. This procedure was stepwise applied down the condenser length and the results were integrated to determine the total required tube length [7]. The Colburn-Hougen diffusion calculation involves an iterative process to solve for the temperature at the interface between the vapor/gas and liquid film was adapted in RELAP5 Mod3.2.

The heat flux due to condensation of vapor mass flux, j_v , flowing toward the liquid-vapor/gas interface is

$$q_v'' = j_v \cdot h_{fgb} \quad (1)$$

where h_{fgb} ($= h_{fgsat}(P_{vb})$) is vapor minus liquid

saturation specific enthalpy based on the vapor partial pressure in the bulk and P_{vb} is vapor partial pressure in the bulk.

The mass flux is given by:

$$j_v = h_m \rho_{vb} \ln \left(\frac{1 - \frac{P_{vi}}{P}}{1 - \frac{P_{vb}}{P}} \right) \quad (2)$$

where P is total pressure, P_{vi} is vapor partial pressure at the liquid-vapor/gas interface, h_m is mass transfer coefficient, ρ_{vb} ($= (1 - X_n) \rho_{mb}$) is saturation vapor density at P_{vb} , and ρ_{mb} is combined vapor and gas density in the bulk vapor/gas temperature.

The heat flux due to mass flux, q_v'' , then, is:

$$q_v'' = h_{fgb} h_m \rho_{vb} \ln \left(\frac{1 - \frac{P_{vi}}{P}}{1 - \frac{P_{vb}}{P}} \right) \quad (3)$$

The heat flux from the liquid film to the wall is calculated by

$$q_l'' = h_c \cdot (T_{vi} - T_w) \quad (4)$$

where T_{vi} is interface $T_{sat}(P_{vi})$ saturation corresponding to the interface vapor pressure.

The wall condensation heat transfer coefficient for an inclined surface in RELAP5 Mod3.2 is the maximum of the Nusselt (1916, laminar) correlation and the Shah (1979, turbulent) correlation. The original work for laminar condensation was accomplished by Nusselt. The Nusselt expression for vertical surfaces uses the film thickness, δ , as the key parameter instead of the temperature difference, and it is given by

$$h_{Nusselt} = \frac{k_f}{\delta} \quad (5)$$

where the film thickness is:

$$\delta = \left[\frac{3\mu_f \Gamma}{g\rho_f(\rho_f - \rho_g)} \right]^{\frac{1}{3}} = \left[\frac{3\mu_f^2 \text{Re}_f}{4g\rho_f(\rho_f - \rho_g)} \right]^{\frac{1}{3}} \quad (6)$$

$$= 0.9086 \left[\frac{\mu_f^2 \text{Re}_f}{g\rho_f(\rho_f - \rho_g)} \right]^{\frac{1}{3}}$$

The Shah's correlation is given by

$$h_{Shah} = h_{sf} \left(1 + \frac{3.8}{Z^{0.95}} \right) \quad (7)$$

with

$$Z = \left(\frac{1}{X} - 1 \right)^{0.8} P_{red}^{0.4} \quad (8)$$

where X is static quality (= (mass vapor + mass noncondensable) / (mass vapor + mass noncondensable + mass liquid)), P_{red} is reduced bulk pressure ($P/P_{critical}$), and h_{sf} is superficial heat transfer coefficient.

$$h_{sf} = h_l (1 - X)^{0.8} \quad (9)$$

Dittus-Boelter coefficient assuming all fluid is liquid

$$h_l = 0.023 \left(\frac{k_l}{D_h} \right) \text{Re}_l^{0.8} P_{rl}^{0.4} \quad (10)$$

where the Reynolds number is given by:

$$\text{Re}_l = G_{total} \frac{D_h}{\mu_f} \quad (11)$$

Initially, the liquid-vapor interface partial pressure is assumed as the saturation pressure based on the wall temperature and so, the corresponding T_{vi} is known, and the energy balance equation can be checked by:

$$q_l'' = q_v''$$

$$\text{Or: } h_c \cdot (T_{vi} - T_w) = h_{fgb} h_m \rho_{vb} \ln \left(\frac{1 - \frac{P_{vi}}{P}}{1 - \frac{P_{vb}}{P}} \right) \quad (12)$$

The calculation is iterated until the convergence criterion is met. Total heat flux is calculated by:

$$q_{total}'' = h_c (T_w - T_{sppb}) \quad (13)$$

The liquid and the gas can both theoretically exchange energy with the wall since RELAP5 is a two-fluid code. Although film condensation is the only condensation mode considered, currently RELAP5 allows a heat flux both to liquid and to gas. The heat flux to liquid is:

$$q_l'' = h_c (T_w - T_f) \quad (14)$$

The gas to wall heat flux is the difference between the total heat flux and the liquid to wall heat flux, i.e:

$$q_g'' = h_{gas} (T_w - T_{sppb}) \quad (15)$$

Total heat transfer coefficient is calculated by:

$$h_{total} = h_c + h_{gas} \quad (16)$$

III. CONDENSATION EXPERIMENTS AT MIT

The experiment apparatus consisted of an open cooling water circuit and an open non-condensable gas/steam loop, as shown in the flow diagram of Fig. 4. The main component of the gas-steam loop were the boiler vessel and the cooled test section. Steam was generated by boiling water using four immersion type sheathed electrical heaters. Compressed air or helium was supplied to the base of the boiler vessel via a pressure regulating valve, a calibrated rotameter, and a flow control valve, respectively. This vessel also served as a mixing chamber, where the non-condensable gas while rising up attained thermal equilibrium with the steam as well as formed a homogeneous mixture with it. The test section consisted of an inner condenser tube and an outer cooling jacket. The vapor and non-condensable gas mixture was injected into the top of the vertical condensing tube and

cooling water was injected into the bottom of the cooling jacket placed outside the condensing tube. The injected gas mixture was cooled and condensed by heat transfer through the condensing tube wall. At different axial locations, thermocouples were welded: to the outer surface of the condensing tube to

measure the outer surface temperature, through the condensing tube to measure the mixture bulk temperatures, and to the outer side of the coolant jacket to measure the coolant temperatures. More details can be found in Sidique (1993) [7]. The experiment conditions used in this study was summarized in Table II.

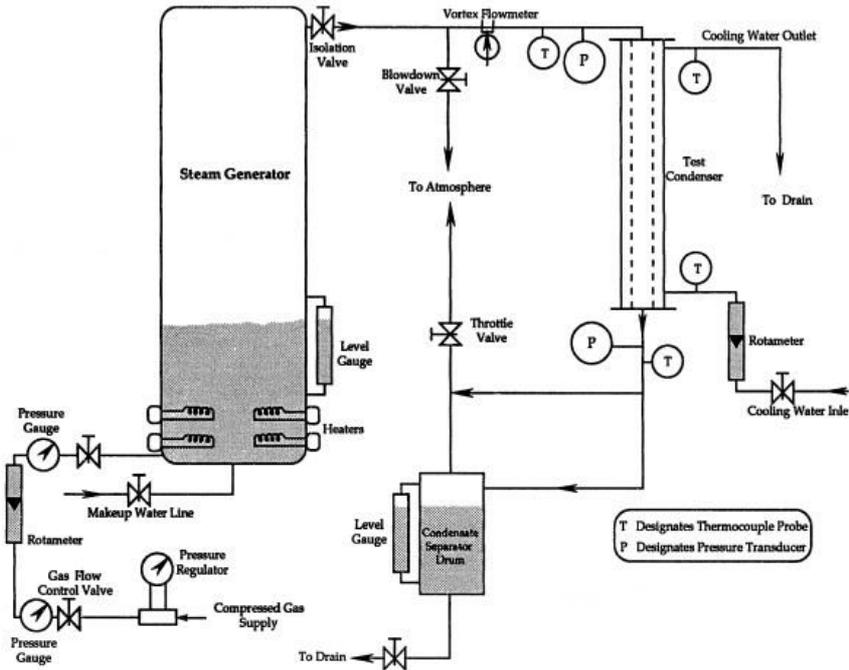


Fig. 4. MIT Steam Condensation Facility

Table II. MIT Experiment Conditions used in this study

| Run No. | Inlet Steam-Helium Mixture Flow rate (kg/s) | Inlet Helium Mass Fraction | Inlet Temperature (°C) |
|---------|---|----------------------------|------------------------|
| 53 | 0.00259 | 0.02 | 101.2 |
| 54 | 0.00265 | 0.04 | 101.2 |
| 55 | 0.00267 | 0.07 | 101.3 |
| 56 | 0.00275 | 0.10 | 100.8 |
| 57 | 0.00250 | 0.02 | 120.0 |
| 58 | 0.00255 | 0.05 | 120.0 |
| 59 | 0.00261 | 0.08 | 120.1 |
| 60 | 0.00268 | 0.11 | 119.7 |
| 61 | 0.00251 | 0.03 | 139.2 |
| 62 | 0.00251 | 0.04 | 139.6 |
| 63 | 0.00256 | 0.06 | 139.3 |

IV. RELAP5 NODALIZATION FOR MIT FACILITY

Fig. 5 shows the RELAP5/MOD3.2 code nodalization scheme for the condensation experiments. The RELAP5/MOD3.2 nodalization used for this simulation contained the following components: condensing pipe, connecting pipe, annulus, time dependent volume and junction, and heat structure. Condensing pipe component with 24 volumes, PIPE-150, was used to model the condenser tube. The time-dependent volumes acting as infinite mass and energy sources or sinks were used to represent the boundary conditions for steam and non-condensable gas flow inside the condensing tube.

The time-dependent volume, TDV-100, was used to provide the inlet flow of steam/non-condensable gas mixture. The pressure and temperature of this volume were determined using the measured bulk inlet pressure and temperature. The inlet steam was saturated. Therefore, the partial pressure of a non-condensable gas was determined by subtracting the saturated pressure of steam from the bulk inlet pressure after being

determined by the inlet temperature. The inlet flow rate of the steam/non-condensable gas mixture was controlled by the time-dependent junction, TDJ-105, and was used in cases in which the experimental data gives the mixture flow rate. When the steam flow rate and non-condensable gas flow rate were given separately, their sum was simply used as the flow rate at the time dependent junction, TDJ-105. The time-dependent volume, TDV-200, was used to provide the outlet boundary condition, and this condition was determined using the experimental data.

The heat structure, HX-150, was used to represent the heat transferred from the steam/non-condensable gas mixture to the coolant through the condensing tube. Heat transfer is a process consisting of condensation heat transfer in the condensing tube, conduction in the tube wall, and convective heat transfer in the cooling jacket AN-250. For simulation of the coolant jacket, two time dependent volumes TDV-300 and TDV-400 are connected to the annulus AN-250 with 22 volumes via a time dependent junction TDJ-305 and a single junction SJ-405.

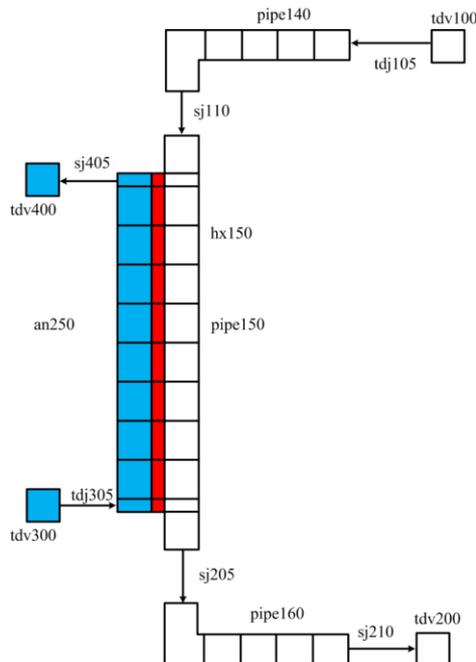


Fig. 5. RELAP5/MOD3.2 Nodalization for MIT Experiment Facility

V. RESULTS AND DISCUSSION

As listed in Table II, 11 cases are simulated by RELAP5/MOD3.2 and the following 4 input parameters are varied: the pressure at the inlet of the test section, P_{in} ; its temperature, T_{in} ; the inlet helium-air mixture flow rate, MF; the inlet helium mass fraction, HMF. Comparison results of coolant

temperature, inner surface temperature of condensing tube, centerline helium-steam mixture temperature, helium-steam mass fraction, and heat transfer coefficient between RELAP5/MOD3.2 simulation and experimental data along the test section length are shown in Figs. 7, 8, 9.

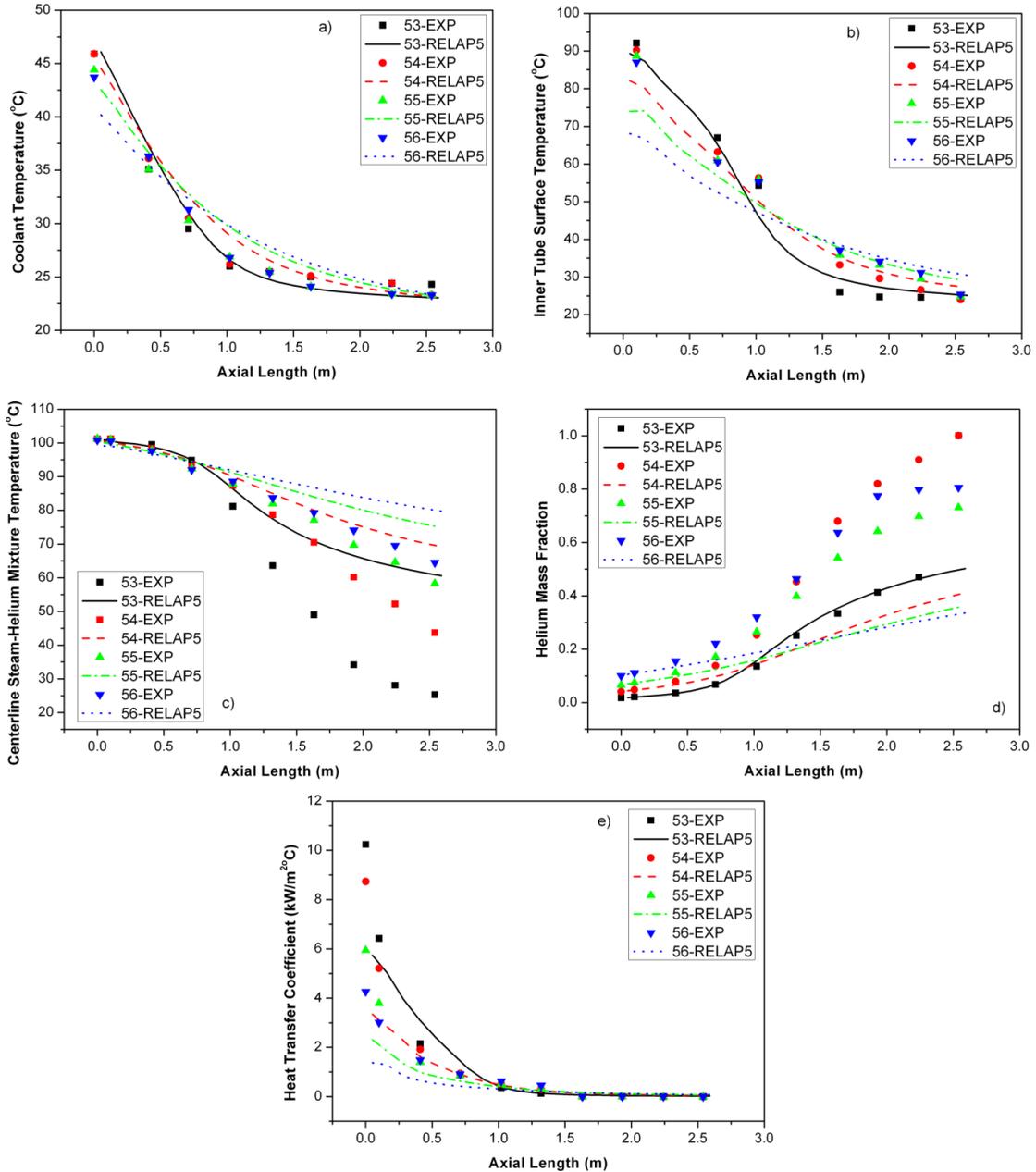


Fig. 6. Comparison Results of RELAP5/MOD3.2 Simulation and Experimental Data

It can be seen in these figures that the experimental data revealed some effects of non-condensable gas on the efficiency of steam condensation. The local heat transfer coefficient is much higher in the inlet of the test section with lower inlet helium mass fraction. It decreases more rapidly throughout the condensing tube than that with higher inlet helium mass fraction, and as a result it become similar in the outlet of the condensing tube (Figs. 7e, 8e, 9e). With lower helium mass fraction, more steam is removed from the

steam-helium mixture flow throughout the condensing tube. As the local mixture flow rate is decreased and the local helium mass fraction is increased, the heat transfer by condensation is reduced and the heat transfer by convection of mixture becomes dominant in the outlet of the test section. Consequently, the centerline steam-helium mixture temperature decrease and helium mass fraction increase more rapidly throughout the condensing tube than that with lower inlet helium mass fraction (Figs. 7c, 7d, 8c, 8d, 9c, 9d).

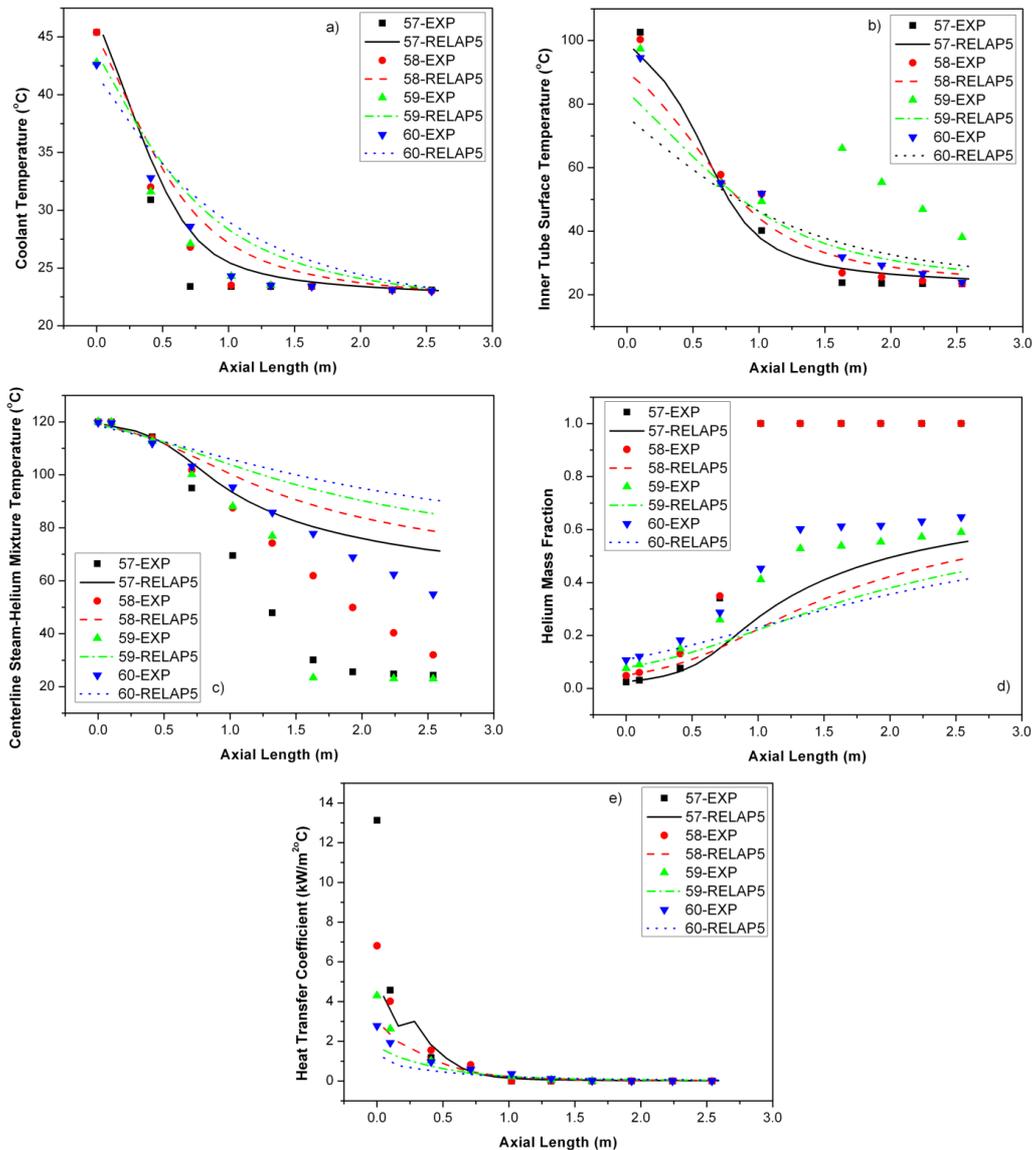


Fig. 7. Comparison Results of RELAP5/MOD3.2 Simulation and Experimental Data

The local heat transfer coefficient decreases more rapidly with higher inlet saturated steam temperature. It always keeps lower value throughout the condensing tube than that with low saturated steam temperature, except for the inlet of the test section, where it gives similar values regardless of the different inlet saturated steam temperature (Figs. 7e, 8e, 9e). The higher wall sub-cooling permits

higher heat flux due to the higher thermal driving force, or the temperature difference between the mixture bulk and the inner wall. However, when the heat flux is divided by the temperature difference to give the heat transfer coefficient, the local heat transfer coefficient is always lower with high inlet saturated steam temperature than that with lower one.

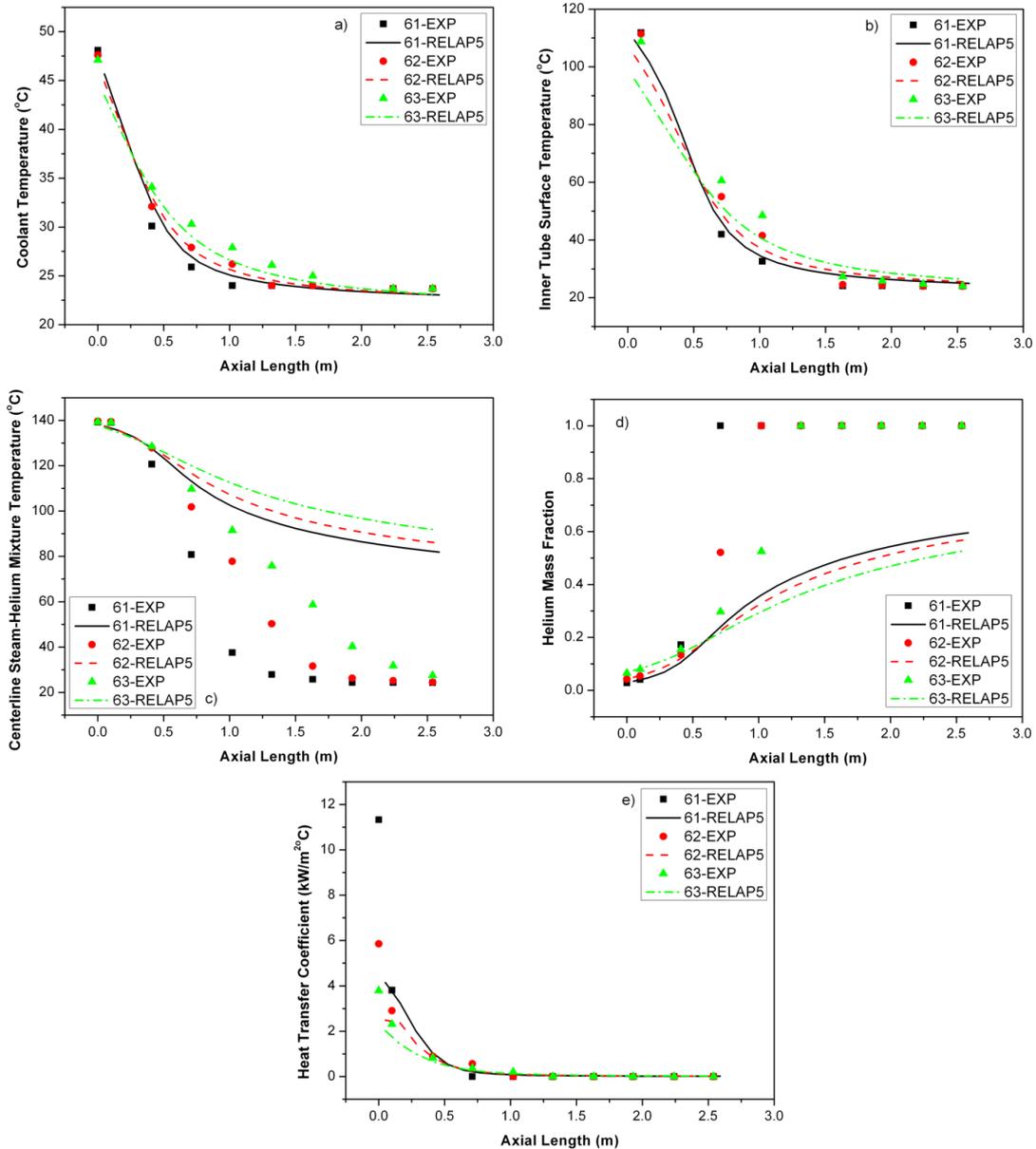


Fig. 8. Comparison Results of RELAP5/MOD3.2 Simulation and Experimental Data

The RELAP5/MOD3.2 simulation results the similar tendencies in all cases that the heat transfer coefficients increase as the inlet steam-non condensable gas mixture flow rate increases, the inlet steam-non-condensable gas mass fraction decrease, and the inlet saturated steam temperature decrease. However, throughout the condensing tube, the calculated heat transfer coefficient from the default model used in RELAP5/MOD3.2 is always much lower than the experimental data which lead to the underestimation of helium mass fraction and overestimation of the centerline steam-helium mixture temperature. This trend become more severe when the inlet mass fraction and steam saturated temperature are increased. Similar results with steam-air condensation were also obtained by Bang et al. [3] and Park and No [4] and have suggested that the effect of the interfacial shear stress was not sufficiently considered in previous correlations using the Reynolds number and they tried to modify the default model by using the degradation factor method to correlate with experimental data. Better results were obtained, however, more experimental and theoretical investigation are still needed.

V. CONCLUSIONS

The capability of the RELAP5/MOD3.2 code to model condensation heat transfer in a vertical tube with a non-condensable gas (helium) was assessed in this study. Overall, the RELAP5/MOD3.2 simulation results the similar tendencies in all cases that the heat transfer coefficients increase as the inlet steam-non condensable gas mixture flow rate increases, the inlet steam-non-condensable gas mass fraction decrease, and the inlet saturated steam temperature decrease. However, the current RELAP5/MOD3.2 code strongly underestimated the condensation heat transfer coefficient and helium steam mass fraction.

REFERENCES

- [1] www.ge-energy.com
- [2] www.ap1000.westinghousenuclear.com
- [3] Y.S., Bang, J.H., Chun, B.D., Chung, G.C., Park, "Improvement of condensation heat transfer models in MARS code for laminar flow in presence of non-condensable gas", Nuclear Engineering Technology, Vol. 41, No.8, pp.1015-1024, (2009).
- [4] H.S., Park and H.C. No, "Assessment of Two Wall Film Condensation Models of RELAP5/MOD3.2 in the Presence of Noncondensable Gas in a Vertical Tube", Nuclear Engineering Technology, Vol. 31, No.5, pp.465-475, (1999).
- [5] H.S., Park, "Steam condensation heat transfer in the presence of noncondensables in a vertical tube of passive containment cooling system", Ph.D. dissertation, Korea Advanced Institute of Science and Technology, Korea, (1999).
- [6] Jack H. Goodykoontz, and Dorsch, G., Robert "Local heat transfer coefficients for condensation of steam in vertical downflow within a 5/8-inch-diameter tube", NASA TN D-3326, National Aeronautics and Space Administration, (1966).
- [7] M.S., Siddique, "The effects of noncondensable gases on steam condensation under forced convection conditions", Ph.D. dissertation, Massachusetts Institute of Technology, (1992).
- [8] S.Z., Kuhn, "Investigation of heat transfer from condensing steam-gas mixtures and turbulent films flowing downward inside a vertical tube", Ph.D. dissertation, University of California, Berkeley, (1995).
- [9] K.Y., Lee, "The effects of noncondensable gas on steam condensation in a vertical tube of passive residual heat removal system", Ph.D. dissertation, Department of Mechanical Engineering, Pohang University of Science and Technology, Korea, (2007).
- [10] "RELAP5/MOD3.2 code manual volume IV: Models and Correlations", NUREG/CR-5535.
- [11] John G. Collier, John R. Thome, "Convective Boiling and Condensation", Oxford Science Publications, (1994).