Transactions of the Korean Nuclear Society Autumn Meeting
Gyeongju, Korea, October 29-30, 2015

Evaluation of Advanced Models for PAFS Condensation Heat Transfer in SPACE Code

Byoung-Uhn Bae[a]*, Seok Kim[a], Yu-Sun Park[a], Kyung Ho Kang[a], Tae-Hwan Ahn[b], Byong-Jo Yun[b]
[a]*Thermal Hydraulics Safety Research Division, Korea Atomic Energy Research Institute*
Daedeok-daero 989-111, Yuseong-gu, Daejeon, 305-353, Republic of Korea
[b]*School of Mechanical Engineering, Pusan National University*
63-gil, Busandaehak-ro, Geumjeong-gu, Busan 607-735, Republic of Korea
** Corresponding author :bubae@kaeri.re.kr*

1. Introduction

The APR+ (Advanced Power Reactor) adopts several new safety features and the PAFS (Passive Auxiliary Feedwater System) is one of the advanced safety features which can cool down the nuclear reactor without any external power supply in case of accidents. The PAFS is operated by the natural circulation to remove the core decay heat through the PCHX (Passive Condensation Heat Exchanger) which is composed of the nearly horizontal tubes.[1]

For validation of the cooling and operational performance of the PAFS, PASCAL (PAFS Condensing Heat Removal Assessment Loop) facility was constructed [2] and the condensation heat transfer and natural convection phenomena in the PAFS was experimentally investigated at KAERI (Korea Atomic Energy Research Institute). From the PASCAL experimental result, it was found that conventional system analysis code underestimated the condensation heat transfer.[3] Therefore, improvement of the prediction capability of the code with the condensation heat transfer model applicable to the PCHX of PAFS is required.

In this study, advanced condensation heat transfer models which can treat the heat transfer mechanisms with the different flow regimes in the nearly horizontal heat exchanger tube were analyzed. The models were implemented in a thermal hydraulic safety analysis code, SPACE (Safety and Performance Analysis Code for Nuclear Power Plant), and it was evaluated with the PASCAL experimental data.

2. Review on Condensation Heat Transfer Model

2.1 Conventional Condensation Model

During condensation process inside horizontal tubes, the two-phase flow pattern may be dominated by vapor shear or gravity force. While annular flow pattern is associated with high vapor shear, stratified flow appears when gravity is the dominant force. In the fully developed annular flow pattern, there is a thin uniform condensate film on the entire tube wall, while the gas phase flows in the central core, and heat transfer is governed by vapor shear and turbulence. In the stratified flow regime, a certain thickness of the condensate layer forms at the bottom of the tube and a thin liquid film settles on the wall in the upper portion of the tube. In this condition, the heat transfer through the thin film is generally analyzed by the classical Nusselt theory.[4] According to this theory, the condensation heat transfer coefficient (h_c) could be determined as following relations.

$$h_c = k_l / \delta \tag{1}$$

$$\delta = \left[\frac{3\mu_l^2 \operatorname{Re}_{fm}}{4g\rho_l\Delta\rho}\right]^{1/3} = 0.9086 \cdot \left[\frac{\mu_l^2 \operatorname{Re}_{fm}}{g\rho_l\Delta\rho}\right]^{1/3} \tag{2}$$

Shah (1979) proposed a simple dimensionless correlation [5] for predicting heat transfer coefficients during film condensation inside plane tubes as shown in Eqs. (3), (4), and (5). It was verified by comparison with various experimental data. It was in agreement with data for water, refrigerants, and organics covering a wide range of experimental conditions in horizontal, vertical, and inclined tubes. However, in a later paper, Shah stated that his correlation had weakness for simulating the experimental conditions under very low flow rates and highly reduced pressure. He found that the deviations were found to be related to the viscosity ratio of phases and reduced pressure.

$$h_c = h_{sf} \cdot \left(1 + \frac{3.8}{Z^{0.95}}\right) \tag{3}$$

$$h_{sf} = h_l \cdot \left(1 - x_e\right)^{0.8} \tag{4}$$

$$Z = \left(\frac{1}{x_e} - 1\right)^{0.8} \cdot p_{red}^{0.4} \tag{5}$$

The model of Shah (1979) was improved and extended to a wider range of parameters suggesting a correction factor which was developed through data analysis [6], as shown in Eqs. (6), (7), (8), and (9). This result was found to give good agreement with data obtained at the experimental conditions under higher flow rates for both horizontal and vertical tubes.

$$h_{TP} = h_I \text{ (for regime I)} \tag{6}$$

$$h_{TP} = h_I + h_{Nu} \text{ (for regime II)} \tag{7}$$

$$h_I = h_{LT} \left(\frac{\mu_f}{14\mu_g}\right)^n \left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{p_r^{0.38}}\right] \tag{8}$$

$$h_{Nu} = 1.32\,\mathrm{Re}_{LS}^{-1/3}\left[\frac{\rho_l\left(\rho_l-\rho_g\right)gk_f^3}{\mu_f^2}\right]^{1/3} \quad (9)$$

2.2 Condensation Model Package

To estimate accurately the condensation heat transfer rate in the nearly horizontal tube, a new condensation heat transfer model package has been developed by PNU (Pusan National University).[7] It consists of a one-dimensional separated flow model for a void fraction, a flow regime identification model, and the condensation heat transfer correlations. The model package takes into consideration the inclination angle of the condensing tube. When flow condition is given by an input variable including the total mass flux and the quality, the void fraction can be obtained from solving momentum equation for each phase.

Film condensation heat transfer correlation was based on the Nusselt theory which formulates liquid film flowing down along the vertical plate. The Nusselt-type correlation could not take into account the interfacial shear effects arisen by high convective vapor flow, since it was developed under the laminar flow conditions. To overcome this drawback, a new film condensation correlation which could take into consideration the interfacial shear stress effect was proposed by adding the vapor phase Reynolds number to the Nusselt-type correlation as follows.

$$h_f = 0.729\left(1+8.9\times10^{-4}\,\mathrm{Re}_g^{\,0.57}\right)\times \left(\frac{g\rho_l\left(\rho_l-\rho_g\right)h_{fg}k_l^{\,3}}{\mu_l D\left(T_{sat}-T_w\right)}\right)^{0.25} \quad (10)$$

On the other hand, Dittus and Boelter (1930) correlation for turbulent single-phase heat transfer [8] was applied to convective heat transfer of condensate flowing along the bottom of the tube. The actual Reynolds number and hydraulic diameter of liquid phase were used in the correlation as follows.

$$h_c = 0.023\,\mathrm{Re}_l^{\,0.8}\,\mathrm{Pr}_l^{\,0.3}\left(\frac{k_l}{D_{hl}}\right) \quad (11)$$

2.3. Other advanced condensation models

Kim (2015) improved Shah (2009) model by modifying the heat transfer coefficient in the regime I [9] as shown in Eq. (12). It included the effect of slip ratio (S) between two phases and the interface roughness factor. This model was validated with the PASCAL and HOCO experimental data.[9]

$$h_I = h_{LT}\left(\frac{\rho_f}{\rho_g}\right)^{1/3} 0.52S^{-0.173}\times \left[\left(1-x\right)^{0.8}+\frac{3.8x^{0.76}\left(1-x\right)^{0.04}}{p_r^{0.38}}\right] \quad (12)$$

In RELAP5/MOD3.3kc code, the original condensation heat transfer model was modified with adopting Jaster-Kosky correlation as shown in Eq. (13). This model has effect of replacing a constant coefficient in Chato model [10] by a function of the void fraction. Also, this modification included extended application of the horizontal condensation heat transfer to a channel with 45 degrees inclination.

$$h_{c,\,jaster-kosky} = 0{,}725\alpha^{3/4}\left(\frac{g\rho_f\Delta\rho h_{fgb}k_f^3}{D_h\mu_f\left(T_{sppb}-T_w\right)}\right)^{\frac{1}{4}} \quad (13)$$

3. Validation of SPACE Code

3.1 Implementation of condensation heat transfer model

In order to evaluate the developed model package for the condensation heat transfer coefficient, SPACE was utilized. The SPACE code has been developed for a thermal hydraulic safety analysis of a nuclear power plant, dealing with the various fluid flow features. It adopts a three-dimensional three-field model, which treats the transport of a continuous liquid, vapor, and droplet field separately.

The condensation heat transfer model in Section 2 has been implemented in the wall condensation model of the SPACE code, so that the calculation result was compared to that of the default condensation model in the SPACE code. Implemented wall condensation model for the PAFS in the SPACE code is activated when the wall condensation option (Data 6 of Hxxx-xx-6001~6009 card) is equal to 3, while the default model could be used with setting this option as zero. The advanced condensation models can be applied only at the left surface of a cylindrical heat structure.

3.2 Validation Test Matrix

The improved models for the condensation heat transfer in the SPACE code were validated by comparing to the PASCAL experimental result. Since the major objective of the PASCAL test is to validate heat removal capability of the PCHX in the APR+ PAFS, the U-tube condensation heat exchanger in the PASCAL facility was designed to simulate equivalent phenomena anticipated inside the PCHX of the PAFS as shown in Fig. 1.[2] It has the same length, inclination angle, thickness, diameter and material with the prototype PCHX. In order to conserve the natural circulation phenomenon outside the PCHX, the U-tube heat exchanger was submerged in the PCCT (Passive Condensation Cooling Tank) pool.

In this study, the experimental result for the steady-state condition in the PASCAL test facility was used to evaluate the effect of improved models in the SPACE code. During the test, the data for the heat transfer coefficient was acquired after a steady state condition for a constant pressure, temperature, and flow rate was achieved at a given thermal power.

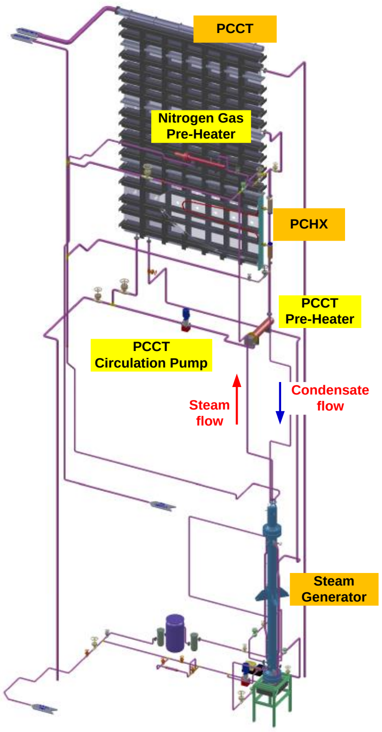

Fig. 1 Overview of PASCAL facility [2]

3.3 Validation Result

As described in Section 2, six cases were evaluated for the test matrix of the PASCAL facility, where the condensation model package in Section 2.2 was evaluated with two cases for a full model package version and a replacement of condensation heat transfer coefficient. Figure 2 compares the calculation result for the steam pressure in SS-540-P1 test of the PASCAL. In this experimental case, 540 kW of the thermal power in the steam generator was supplied. As shown in the figure, the calculation result with the default wall condensation model of the SPACE code predicted the steam pressure and temperature higher than the experimental result. It means that the condensation heat transfer coefficient was lower than the measured value as shown in Fig. 3, and a steady state condition calculated by the SPACE code was achieved at the higher steam pressure and temperature. While Shah (2009) model and Kim (2015) model revealed improved prediction for the condensation heat transfer, the calculation result with the condensation heat transfer model package with the correlation version showed the best performance. It points out that the mechanistic modeling for the film condensation in the steam phase and the convection in the condensate liquid separately contributed to enhance prediction capability of the heat transfer coefficient in the wall condensation model of the SPACE code. In case of adopting the full model package with the void fraction and flow regime models, the steam pressure presented an over-prediction due to a lower heat transfer coefficient inside the tube as shown in Fig. 2 and 3.

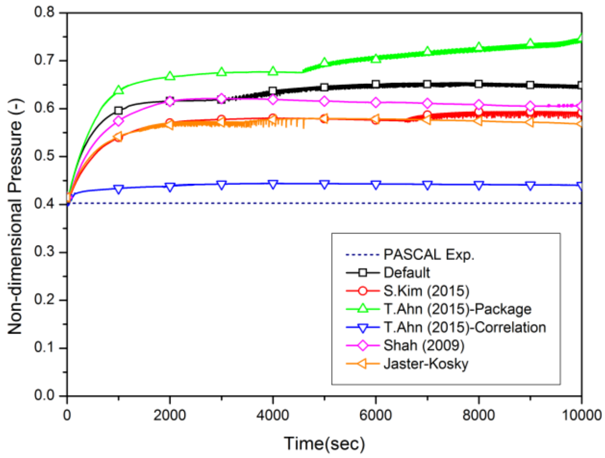

Fig. 2 Steam pressure in simulation of SS-540-P1 test

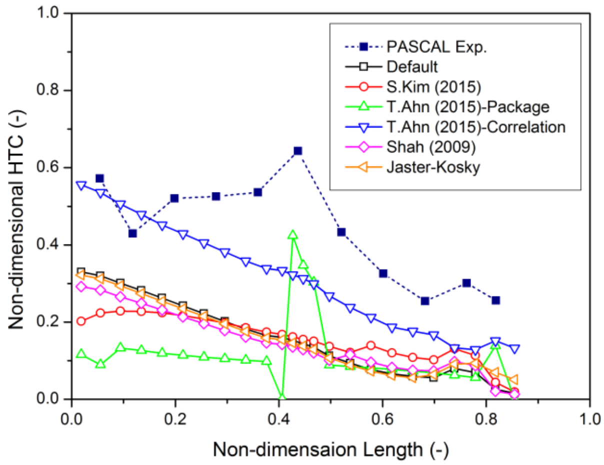

Fig. 3 Condensation heat transfer coefficient in simulation of SS-540-P1 test

Figures 4 and 5 compare the steam pressure and temperature at the steady state condition in all test matrix, respectively. As observed in the calculation result of SS-540-P1 test, the mechanistic modeling for the condensation heat transfer coefficient contributed to enhance the prediction capability of the SPACE code in simulation of all test conditions with varying the thermal power in the steam generator. Therefore, it is concluded that the enhanced condensation heat transfer model could reasonably reduce conservatism in the wall condensation model of the SPACE code. On the other hand, all models presented under-prediction of the condensation heat transfer coefficient in higher thermal power condition, so that it induced the over-estimation of the steam pressure at the quasi-steady state condition. Therefore, as a further study, improvement of the present mechanistic model is required especially for a larger steam flow and higher pressure conditions.

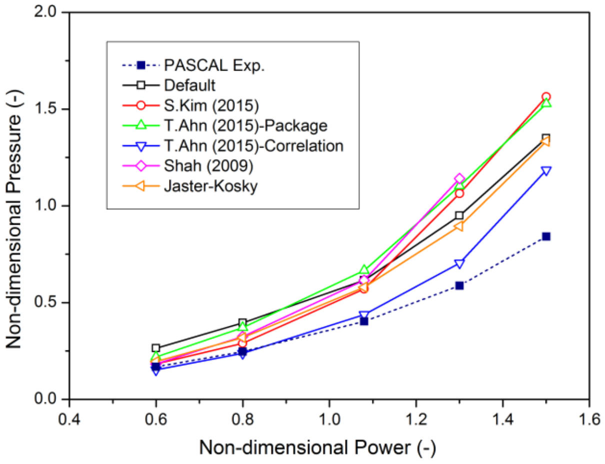

Fig. 4 Steam pressure in simulation of all test cases

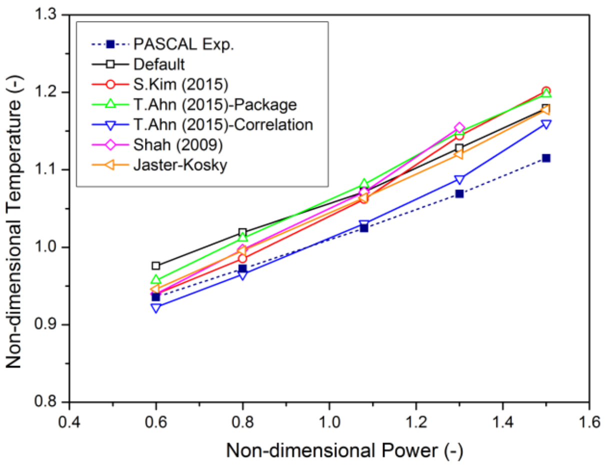

Fig. 5 Steam temperature in simulation of all test cases

4. Conclusions

With an aim of enhancing the prediction capability for the condensation phenomenon inside the PCHX tube of the PAFS, advanced models for the condensation heat transfer were implemented into the wall condensation model of the SPACE code, so that the PASCAL experimental result was utilized to validate the condensation models. Calculation results showed that the improved model for the condensation heat transfer coefficient enhanced the prediction capability of the SPACE code. Among the models, adopting the condensation heat transfer model package presented the best performance to predict the steam pressure and temperature at the steady state condition. This result confirms that the mechanistic modeling for the film condensation in the steam phase and the convection in the condensate liquid contributed to enhance the prediction capability of the wall condensation model of the SPACE code and reduce conservatism in prediction of condensation heat transfer.

REFERENCES

[1] J. Cheon et al., "The Development of a Passive Auxiliary Feedwater System in APR+ Track 1: Water-Cooled Reactor Programs & Issues", Proceedings of ICAPP'10, June 13-17, San Diego, CA, USA 2010.
[2] K. H. Kang et al., "Separate And Integral Effect Tests for Validation of Cooling And Operational Performance of the APR+ Passive Auxiliary Feedwater System", Nuclear Engineering and Technology, vol. 44, no. 6, pp. 597-610, 2012.
[3] Y. J. Cho et al., "Assessment of Condensation Heat Transfer Model to Evaluate Performance of the Passive Auxiliary Feedwater System", NUCLEAR ENGINEERING AND TECHNOLOGY, VOL.45 NO.6, pp. 759-766, 2013.
[4] W. Nusselt, Die oberflachenkondensation des wasserdampfes (The surface condensation of water vapor). Zeit. D. Ver. Deut. Ing. Frankfurt 60, 541–575, 1916.
[5] M. M. Shah, A general correlation for heat transfer during film condensation inside pipes. Int. J. Heat Mass Transfer 22, 547–556, 1979
[6] M. M. Shah, An Improved and Extended General Correlation for Heat Transfer during Condensation in Plain Tubes. HVAC&R RESEARCH, VOLUME 15, NUMBER 5. 889-913, 2009.
[7] T. H. Ahn et al., "Development of a new condensation model for the nearly-horizontal heat exchanger tube under the steam flowing conditions", International Journal of Heat and Mass Transfer 79, 876– 884, 2014.
[8] F. W. Dittus and L. M. K. Boelter, "Heat transfer in automobile radiator of the tube type", Publication in Engineering, University of California, Berkley, Vol. 2, p. 250, 1930.
[9] S. Kim, "Report on Basic Model Development of Condensation Heat Transfer Coefficient for Horizontal Condensation Heat Exchanger," KAERI/TR-5846, 2015.
[10] J. C. Chato, Laminar condensation inside horizontal and inclined tubes. ASHRAE J. 4 (2), 52–60, 1962.