Condensation Heat Transfer Modeling with Non-Condensable Gas in the TASS/SMR-S Code

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1. Introduction

The TASS/SMR-S code is a thermal-hydraulic system code for integral reactors, comprising conservation equations for mixture mass, liquid mass, non-condensable gas mass, mixture momentum, mixture energy, and the enthalpy of vapor and non-condensable gases. It has been employed for safety analysis of small modular reactors such as SMART-100. Upon extending its applicability to systems using non-condensable gas as a pressurant, reliable prediction of condensation heat transfer has become increasingly important for assessing primary system pressure. In system codes including RELAP5 and SPACE, condensation at stratified vertical interfaces is typically modeled by the McAdams correlation which was developed for natural convection over open-ended free plates. However, in pressurizers, condensation occurs mainly on the liquid surface inside the vessel. This study implements alternative correlations for level-surface condensation and extends wall condensation models to account for non-condensable gas effects. The applicability of these models is evaluated through comparison with representative benchmark problems, aiming to enhance the predictive capability and reliability of TASS/SMR-S for SMR pressurization systems.

2. Condensation at level surface

The correlations used in thermal-hydraulic system codes to predict heat transfer at vertically stratified interfaces are based on experiments of natural convection over horizontal plates subjected to a temperature difference with the fluid. The flow patterns differ depending on whether the hot plate faces upward (heated floor or cold roof) or downward (heated roof or cold floor). When the hot plate is at the bottom, the lower fluid is heated first, causing flow instability and earlier transition to turbulence; in contrast, when the hot plate is at the top, the flow remains stable even for large temperature differences. Table 1 lists representative models. The Rayleigh number exponent ranges from 1/4 to 1/3 for heated floor models and 1/5 to 1/4 for heated roof models. Appropriate models should be selected according to the flow conditions. Although these natural convection correlations are based on experiments and their strict applicability is limited, the log(Ra)–log(Nu) relationship is approximately linear, allowing system codes to apply them over a wide range of conditions.

Table 1: Natural convection heat transfer correlations

Models	Case	Correlation	Range	Remarks
McAdmas [1]	Heated floor	$Nu=0.54Ra^{1/4}$	$10^5<Ra<2x10^7$	Laminar
		$Nu=0.14Ra^{1/3}$	$2x10^7<Ra<3x10^{10}$	Turbulent
	Heated roof	$Nu=0.27Ra^{1/4}$	$3x10^5<Ra<3x10^{10}$	Laminar
Fujii-Imura [2]	Heated floor	$Nu=0.13Ra^{1/3}$	$5x10^8<Ra$	Turbulent
	Heated roof	$Nu=0.58Ra^{1/5}$	$10^6<Ra<10^{11}$	Laminar
Min *et al.* [3]	Heated floor	$Nu=0.33Ra^{0.31}$	$Ra\sim10^{11}$	**Enclosed room**
	Heated roof	$Nu=0.07Ra^{0.255}$	$Ra\sim10^{11}$	

The heat transfer coefficient in Min *et al.*'s heated roof model is significantly lower than in other models. While many researchers, including McAdams and Fujii–Imura, conducted experiments on freely supported plates with open ends, Min *et al.*'s experiments were performed in a fully enclosed room. In a confined space, flow recirculation and interactions reduce heat transfer compared to unbounded flow [4]. Therefore, models developed in enclosed conditions are expected to be more appropriate for interface heat transfer in plant coolant systems. In this study, the models listed in Table 1 were tested and the most suitable model was selected.

3. Condensation near wall

When condensation occurs on a cold wall in the presence of non-condensable gases, the formation of a liquid film increases the local partial pressure of the gas, lowering the vapor saturation temperature and reducing the temperature difference with the wall. The non-condensable gas boundary layer further inhibits vapor transport, decreasing the convective heat transfer coefficient.

Various methods exist to account for the effects of non-condensable gases in condensation heat transfer calculations. Directly solving the conservation equations for the liquid film and gas boundary layer to include diffusion is not practical for system codes analyzing the entire plant. This study applies and compares widely used experimental degradation factors [5,6] and diffusion boundary layer heat balance model [7].

4. Computational results

The applicability of these models is evaluated through comparison with representative MIT pressurizer benchmark results [8]; additional benchmark cases will be included in the presentation.

7.1. Steam pressurizer

The initial water level in the tank was 0.35 m, with a pressure of 0.696 MPa and a temperature of 437.89 K. Subcooled water at 297.04 K was injected for approximately 64 seconds, and the pressure response was observed. As no non-condensable gas was present, the differences in results due to the interface heat transfer models could be clearly seen. The system was simulated using 10 nodes.

Figure 1 shows the pressure response inside the steam pressurizer. During the subcooled water injection, the pressure change was primarily driven by steam compression resulting from the rising water level, and thus differences among the models were minimal. After the injection was stopped, the pressure decrease depended on the condensation rate at the liquid-vapor interface for each model. As the injected subcooled water lowered the liquid temperature below that of the vapor, a heated-roof natural convection condition develops inside the tank. For comparison, a case where the heated-floor correlation was enforced is also presented.

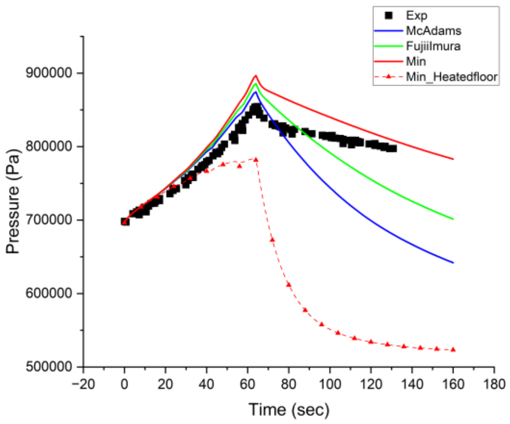

Fig. 1. Pressure response inside steam pressurizer

7.2. Steam-Nitrogen pressurizer

The interface heat transfer models were also compared for an experiment in which both steam and nitrogen were present. The initial water level was 0.5318 m, with a pressure of 0.5309 MPa and a liquid temperature of 427.15 K, and the system contained 10% nitrogen. Subcooled water at 294.26 K was injected for approximately 35 seconds.

Figure 2 shows the comparison of interface condensation models without considering the effects of non-condensable gas on the wall condensation. The differences with experimental data were larger than in the steam pressurizer case, suggesting that the influence of non-condensable gases should also be in calculations. When Min *et al.*'s heated-floor correlation was enforced, the pressure continued to drop after the subcooled water injection was stopped, failing to stabilize.

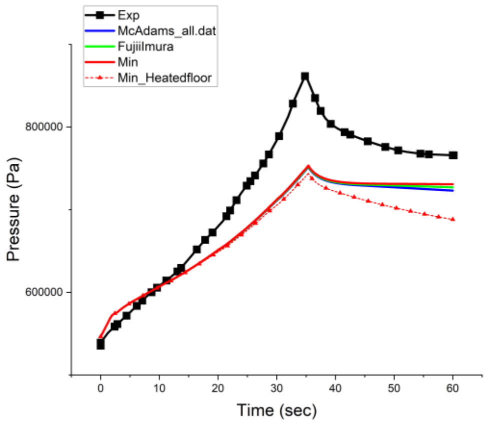

Fig. 2. Pressure response inside steam-N2 pressurizer without non-condensable gas models

Condensation at the stratified interface was fixed using Min *et al.*'s model, while the influence of non-condensable gas on wall condensation was evaluated with different models. Without accounting for the non-condensable gas effect (Min-WSLAB 0), the condensation rate was higher than in the cases where experimentally derived degradation factors were applied (Min-WSLAB 1 and 2), although the difference was minor, as shown in Figure 3.

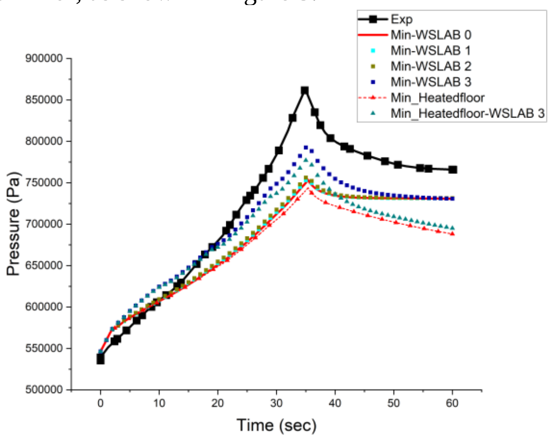

Fig. 3. Pressure response inside steam-N2 pressurizer with non-condensable gas models

In contrast, when the diffusion model was applied (Min-WSLAB 3), the suppression of wall condensation by non-condensable gas was effectively captured, leading to a pressure rise that closely matched the experimentally measured peak. Even when the heated-floor correlation was enforced at the interface, applying the diffusion model for wall condensation still produced a significant reduction in condensation rate; however, its predictive accuracy was lower than that obtained with the physically appropriate heated-roof correlation.

5. Conclusion

Representative benchmark problems were solved with the TASS/SMR-S code by applying natural convection correlations more suitable than the conventional McAdams model. In addition, the influence of wall condensation models was examined in the presence of non-condensable gas. These results demonstrate the need to consider both interfacial correlations and non-condensable gas effects in condensation simulations. Additional validation results not included in the abstract will be presented during the talk.

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