Assessment of the MELCOR 1.8.6 condensation heat transfer model under the presence of noncondensable gases

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1. Introduction

Condensation heat transfer under the presence of noncondensable gases (NCGs) is an important issue in nuclear safety because the presence of even a small quantity of NC gases in the vapor largely reduces the condensation rate. To understand this phenomenon, a lot of theoretical and experimental studies have been performed[1~8]. The extensive assessment of the condensation model of the safety analysis codes has been also performed.

The main objective of the present study is the assessment of the condensation heat transfer model of the severe accident code MELCOR 1.8.6 under the presence of NCGs.

2. Condensation model of the MELCOR 1.8.6

When NCGs are present, the condensation phenomenon is largely reduced by accumulated NCGs near the condensing surface. Since the total pressure remains constant, the partial pressure of vapor at the liquid-vapor interface is lower than that in the bulk mixture, providing the driving force for vapor diffusion towards the liquid-vapor interface[11]. This situation is illustrated in Fig. 1.

The MELCOR 1.8.6 employs the molar-based stagnant film model (SFM) to predict this phenomenon[9,10]. In the SFM, the condensation mass flux is given by

$$\dot{m}_c = h_D \rho_v ln\left(\frac{P_{tot} - P_{srf}}{P_{tot} - P_{stm}}\right), \tag{1}$$

where ρ_v is the density of vapor at $T_{sat}(P_{tot})$, P_{tot} is the total volume pressure, P_{srf} is the saturation pressure of steam at the surface temperature and P_{stm} is the steam partial pressure in the volume. Eq. (1) is derived from the Fick's laws of diffusion which is based on the mass transfer theory. The mass transfer coefficient, $h_D=Sh \cdot D/L_c$, is a function of the Sherwood number, $Sh=NuSc^{1/3}Pr^{-1/3}$, which is drawn from the theory of the heat and mass transfer analogy.

In the MELCOR 1.8.6, the film tracking model is used to calculate the liquid film thickness as follows:.

$$\delta_f = \begin{cases} 0.909\delta^* Re_f^{1/3}, \text{ if } Re_f < 1000 \text{ (Laminar)} \\ 0.115\delta^* Re_f^{0.6}, \text{ if } Re_f > 3000 \text{ (Turbulent)} \end{cases}, \tag{2}$$

where

$$Re_f = \frac{2(\dot{m}_{in} + \dot{m}_{out})}{w\mu_f}, \tag{3}$$

$$\delta^* = \left(\frac{\mu_f^2}{\rho_f g sin\theta}\right)^{1/3}, \tag{4}$$

The film Reynolds number, Eq. (3), includes the mass flow rate into the heat structure surface from film drainage from an adjacent heat structure surface ($\dot{m}_{in}$) and the mass flow rate out of this surface ($\dot{m}_{out}$). w, μ_f and ρ_f in Eqs. (3) and (4) are the width of the surface, the viscosity of the film and the density of the film, respectively.

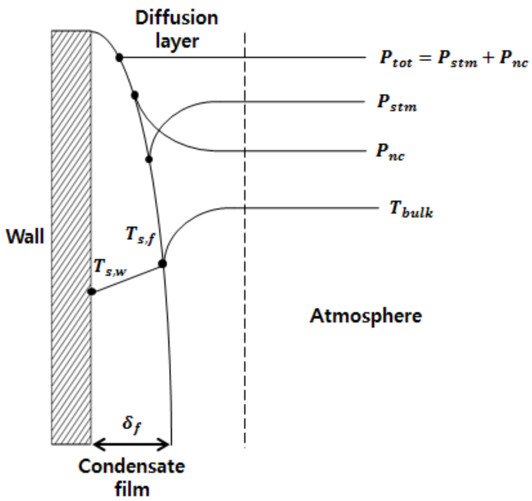

Fig. 1. Concept of the condensation under the presence of NCGs

3. Simulation results

Condensation experiments performed under the thermal-hydraulic conditions similar to those inside a reactor building during accidents are collected and categorized into 4 types: vertical flat plates, outer surface of vertical pipes, inner surface of vertical pipes, inner surface of horizontal pipes. The test conditions of condensation experiments[1~8] are presented in Table I.

Figures 2 through 5 summarize the results of the assessment. Fig.2 shows that the MELCOR under-predicts the condensation heat transfer on the vertical flat plate. Particularly, in high-speed inlet conditions (above 2.5m/s) the error increases. Fig. 3 indicates that the calculated condensation on outer surface of vertical pipe is good agreements with measured value. Fig.4 presents a comparison of condensation heat flux

Table I. The classification of experiments and the test conditions

Experiments	Geometry	No. of test	Pressure (bar)	Air mass fraction	Velocity (m/s)	Mass flow rate (kg/s)
COPAIN[1]	Vertical flat plates	4	1.0	0.77~0.86	0.3~3.0	-
CONAN[2]		10	1.0	0.13~0.72	2.5~2.6	-
Dehbi[3]	Outer surface of vertical pipes	9	1.5 ~ 4.5	0.33 ~ 0.89	-	-
Pan[4]		2	2.0, 4.0	0.45, 0.95	-	-
Kuhn[5]	Inner surface of vertical pipes	24	1.0 ~ 5.0	0.20 ~ 0.40	-	0.010~0.027
Park[6]		5	1.1 ~ 4.7	0.20 ~ 0.30	-	0.005~0.011
Siddique[7]		26	1.1 ~ 4.9	0.14 ~ 0.35	-	0.003~0.013
Wu[8]	Inner surface of horizontal pipes	34	1.0 ~ 4.0	0.05 ~ 0.20	-	0.006~0.053

predictions with experimental data in the case of inner surface of vertical pipe. Kuhn's experiment conducted under relatively large mass flow rate is well predicted but Park's and Siddique's experimental data are under-predicted.

In the condensation experiment on inner surface of horizontal pipe conducted by Wu, temperatures are measured at top and bottom side. However, calculated value is averaged one. Thus one to one comparison between the calculation and the experimental data is not reasonable. Thus, local heat fluxes are compared as shown in Fig. 5. It illustrates that the MELCOR code under-predicts the experiment. The results of other conditions are also similar to those in Fig. 5.

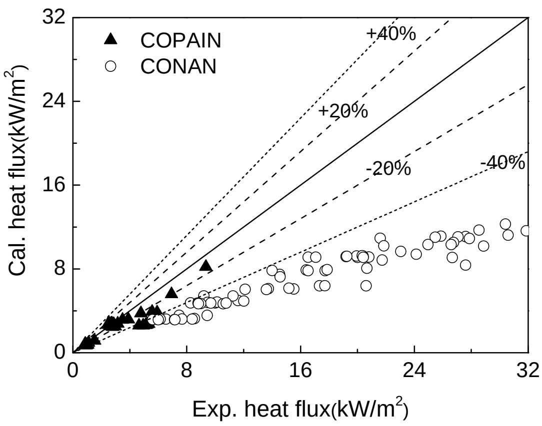

Fig. 2. Condensation on vertical flat plate

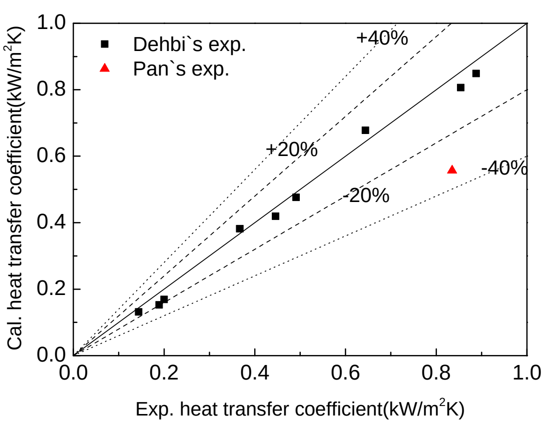

Fig. 3. Condensation on outer surface of vertical pipe

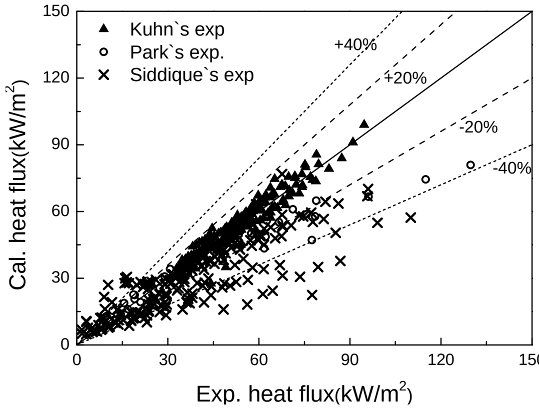

Fig. 4. Condensation on inner surface of vertical pipe

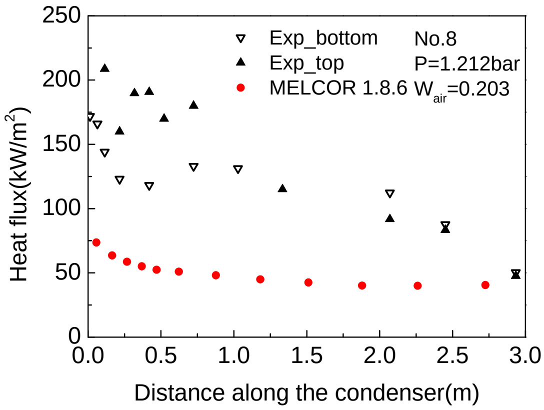

Fig. 5. Condensation on inner surface of horizontal pipe

4. Conclusions

In this study, the condensation heat transfer model of the MELCOR 1.8.6 is assessed using various experiments which have 4 different types of geometry. Through the comparison of the results, it was shown that the MELCOR code generally under-predicts the condensation heat transfer except the condensation on outer surface of vertical pipes and improvement is needed for other geometries. Especially, considering the major applications of MELCOR, the condensation heat transfer model on a vertical plate, which is equivalent to the inner wall of a reactor building, should be improved first.

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