

## Discussion on Superheated Steam Condensation and Re-vaporization

Soon-Joon HONG<sup>a\*</sup>, Je-Hee LEE<sup>a</sup>

<sup>a</sup>FNC Tech. Co. Ltd., Floor 32, Heungdeok IT Valley, 13 Heungdeok 1-ro, Giheung-gu, Yongin-si, Gyeonggi-do, 16954, Korea

\*Corresponding author: sjhong90@fnctech.com

### 1. Introduction

Wall condensation of water vapor (steam) widely happens in nuclear system. Some case is for pure steam, and some for steam/air mixture. The steam in steam/air mixture is usually saturated in case of loss-of-coolant-accident (LOCA), while usually superheated in case of main steam line break (MSLB) accident.

The commonly accepted concept for wall condensation mechanism has been proposed by Nusselt [1,2]. The condensed steam makes condensate film on the wall. And the film velocity is obtained from the force balance of gravity and shear, and the film thickness is calculated from condensate flow based on this velocity by equating mass transfer by condensive heat transfer. During that manipulation the film temperature profile is assumed linear between wall temperature and interface temperature, thus the heat flux is calculated easily by the temperature gradient.

Many of the nuclear thermal hydraulic codes such as MARS-KS [3] consider the wall condensation and the steam is assumed saturated (or the state is not so seriously considered, and the steam-superheat is not so seriously treated). However, United State Nuclear Regulatory Commission (US NRC) suggested a concept of 're-vaporization', which means some of the condensate may be assumed to remain in the vapor region without changing into liquid film[4]. It is realized in CONTEMPT-LT code [5]. This conceptual phenomenon yields the lower atmosphere temperature in case of MSLB, because the more mass exist in the atmosphere even though the same heat is removed to liquid. However, the fraction of remaining condensate is input by user, whose maximum is limited as 8%.

Since the Nusselt derivation, several studies have been conducted on the superheated steam condensation [6-12]. However any of these literature mentioned the 're-vaporization'.

This study provides several discussion issues on the superheat steam condensation

### 2. Literature Survey

#### 2.1 Measurement of Heat Transfer Coefficient

In case of vertical plate wall condensation, the heat flux that is dissipated out of the wall is calculated using the wall temperature gradient. That is, the condensing wall surface temperature and the other temperature inside the wall are measured and these two temperatures can

make the heat flux using the wall thermal conductivity and the measurement distance between the two points. This heat flux is assumed the same to the condensive heat flux between gas bulk and interface. The short-come of this method is large uncertainties arising from the fact the distance is very small, just a few millimeters. But the benefit is that the local heat transfer coefficient (HTC) can be obtained.

The other method is to measure the coolant temperature rise outside the condensing plate. When using coolant for the removal of wall heat from the steam condensation, the coolant temperature rise is corresponding to the heat removal, and the average heat flux is easily calculated. The shortage is that the local HTC cannot be obtained and uniform wall temperature or uniform heat flux is hardly made.

Regardless of the steam type, saturated or superheated, one of these methods has been used. Thus, the wall heat flux is believed condensive flux in most of the HTC models.

#### 2.2 Mechanism of Condensation: Re-vaporization

The concept of re-vaporization is shown in NUREG-0588, and does rarely appear in the other literature. Fig. 1 shows the conceptual schematics of re-vaporization in NUREG-0588.

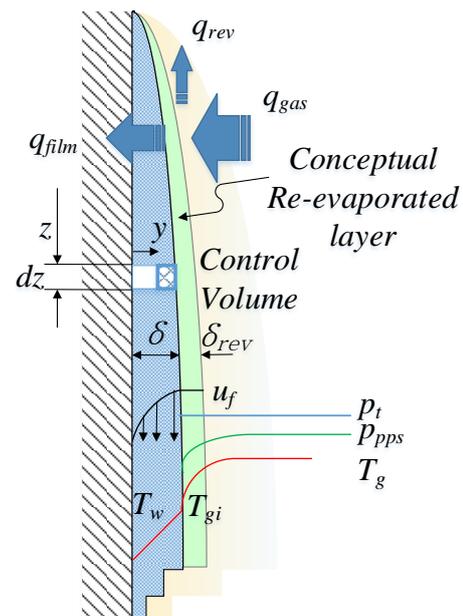


Fig.1 Flow of a laminar film on a surface

As mentioned in section 2.1, the heat flux that is measurable is only the wall heat flux, and it is the same to the lump-summed heat flux in gas phase. The heat flux in gas phase is affected by several factors. Usually the film temperature profile and gas phase boundary layer, together with their calculation methods affect the heat flux. And some other factors are the consideration of sensible heat and its treatment.

### 2.3 Driving Potential for Condensation Heat Transfer and Mass Transfer Relation

The driving potential for the condensation is usually known to be wall superheat ( $T^s - T_w$ ). However, when considering sensible heat transfer, some authors such as Lee et al. (1991) in Webb (1998) uses temperature difference itself ( $T_g - T_w$ ). But it looks just a mathematical convenience.

For the condensation from saturated steam to saturated water the condensation mass rate is calculated by

$$\dot{m}_{cond} = \frac{q}{h_{fg}} \quad (1)$$

If the film gets subcooled, the condensation mass rate is given by

$$\dot{m}_{cond} = \frac{q}{h_{fg}^*} \quad (2)$$

$$h_{fg}^* = h_{fg} \left[ 1 + \frac{3}{8} \frac{c_{pf}(T^s - T_w)}{h_{fg}} \right] \quad (3a)$$

Or by Rohsenow in Collier et al. (1994).

$$h_{fg}^* = h_{fg} \left[ 1 + 0.68 \frac{c_{pf}(T^s - T_w)}{h_{fg}} \right] \quad (3b)$$

From Eqs. (2) and (3a) or (3b) following relation is obtained.

$$h_{fg} \dot{m}_{cond} = q - \frac{3}{8} c_{pf} (T^s - T_w) \dot{m}_{cond} \quad (4)$$

Second term in right hand side corresponds to the enthalpy rise of film from wall temperature to saturation temperature. The coefficient,  $3/8$ , appears surely because of the film temperature profile. Some the other coefficient can be derived according to the film temperature profile. Thus, this equation means that only the pure heat that excludes the cooldown heat of the film is concerned for the condensate mass. Let's substitute  $h_{cooldown}$  for  $\frac{3}{8} c_{pf} (T^s - T_w) \dot{m}_{cond}$  in order to express it using enthalpy addition (subtraction). Then following relation is obtained.

$$\dot{m}_{cond} = \frac{q}{h_g^s - (h_f^s - h_{cooldown})} \approx \frac{q}{h_g^s - h_f} \quad (5)$$

The denominator is phasic enthalpy difference.

For superheat steam a similar suggestion is found. Webb (1998) proposed following relation.

$$h_{fg}^* = h_{fg} \left[ 1 + \frac{c_{pv}(T_b - T^s)}{h_{fg}} \right] \quad (6)$$

It is very interesting that the equation form is same with that of film subcooling. Similar interpretation can be done; only the pure heat that excludes the cooldown heat of the superheated steam to saturation is concerned for the condensate mass. Of course,  $c_{pv}(T_b - T^s)$  is corresponding to bulk superheat amount, and  $h_g^s + c_{pv}(T_b - T^s) \approx h_g$ , gas phasic enthalpy. Then, the mass transfer rate can be obtain using similar manipulation.

$$\dot{m}_{cond} \approx \frac{q}{h_g - h_f^s} \quad (7)$$

In some thermal hydraulic codes following relation is optionally used for the mass transfer

$$\dot{m}_{cond} = \frac{q}{h_g - h_f} \quad (8)$$

The enthalpy in above equation is phasic specific enthalpy and they are not necessarily the saturated enthalpy. This relation may be derived under the assumption of conceptual re-vaporization. If the condensation of superheat steam is subsequently made by the cooldown of superheated steam, phase change from saturated steam to saturated water, and finally cooldown of film to subcooled state, then the condensing mass rate should account for the phase change heat as follow (i.e. the cooldown heats should be subtracted!).

$$\dot{m}_{cond} = \frac{q - \dot{m}_{cond}(h_g - h_g^s) - \dot{m}_{cond}(h_f^s - h_f)}{h_{fg}} \quad (9)$$

By some manipulation we get,

$$\dot{m}_{cond} = \frac{q}{h_g - h_f} \quad (10)$$

Here it should be noted that the denominator is not  $h_{fg}$  but  $h_g - h_f$ , phasic enthalpy difference.

Reviewing Eqs (5), (7), and (10), it can be found that the condensate mass is consistently calculated using phasic enthalpy difference.

### 2.4 Heat Transfer Rate

The higher the superheat, the more the heat transfer. It is consistent trend in most of the authors. The main reason for such a trend is the thinner film (Mitrovic, 2000). Wall subcooling does rarely affect the heat transfer. But when the wall subcooling is small, the larger wall subcooling induces less heat transfer (Minkowycz et al., 1966). The non-condensable gas also affect a negative effect in case of superheat steam. The seam superheat has more effect on HTC in case of air/steam mixture rather than pure steam. Fig.2 shows

the typical trend and it is taken from Minkowycz et al.(1969).

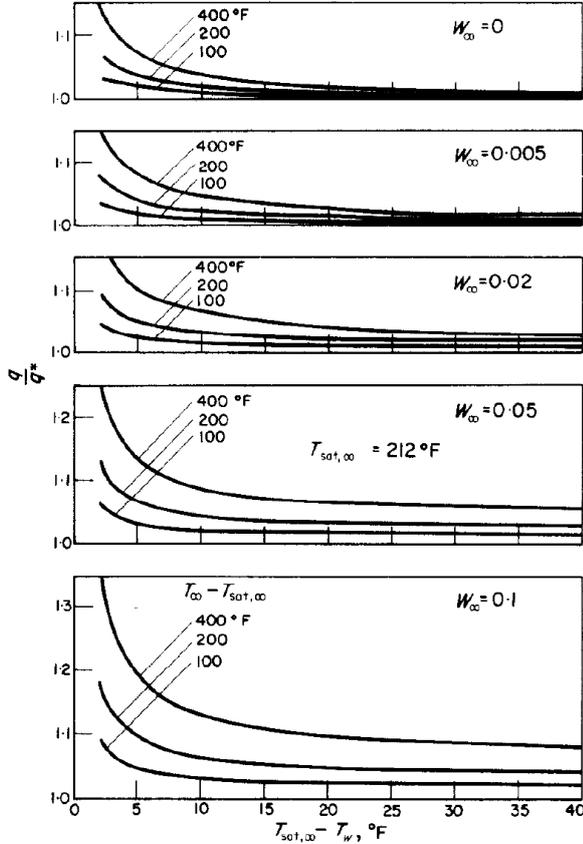


Fig.2 Effect of noncondensable and steam superheat on heat transfer (Minkowycz et al. (1969),  $q^*$ : the case of no superheating.)

All these theoretical approach model looks lack of experimental verification. Only Lee et al. (1991) in Webb (1998) shows the verification

### 3. Further Theoretical Exploration of Re-vaporization

Now let's check the details of Nusselt's approach for the better understanding of superheated steam condensation. The schematic for the film-wise condensation is shown in Fig.1. From the force balance, following equation can be setup.

$$(\delta - y)dzw(\rho_f - \rho_g)g = \mu_f \left( \frac{du(y)}{dy} \right) dzw \quad (8)$$

,where  $w$  is the width of vertical plate. Integrating with the boundary condition  $u = 0$  at  $y = 0$ , then

$$u(y) = \frac{(\rho_f - \rho_g)g}{\mu_f} \left[ y\delta - \frac{y^2}{2} \right] \quad (9)$$

And velocity profile in conceptual re-evaporated layer is assumed same to the velocity at  $y = \delta$  (See Fig. 1 for easier understanding)

$$u(y)|_{y=\delta} = \frac{(\rho_f - \rho_g)g \delta^2}{\mu_f} \quad (10)$$

Mass flowrate in actual film is given by

$$\dot{m}_{film} = \rho_f \int_0^\delta u_y dy w = \frac{\rho_f(\rho_f - \rho_g)g \delta^3}{3\mu_f} w \quad (11)$$

And mass flowrate in the conceptual re-evaporated layer is given by

$$\dot{m}_{rev} = \rho_f \int_\delta^{\delta+\delta_{rev}} u_y dy = \frac{\rho_f(\rho_f - \rho_g)g \delta^2 \delta_{rev}}{2\mu_f} w \quad (12)$$

If applying the concept of re-vaporization, the heat balance can be setup as following (Total heat is the sum of film conduction heat and re-vaporization heat, See Fig.1).

$$q_{gas} = q_{film} + q_{rev} \quad (13)$$

$$\begin{aligned} w \frac{k_f}{\delta} (T_{gi} - T_w) dz + \dot{m}_{rev} (h_g - h_f^s) \\ = w \frac{k_f}{\delta} (T_{gi} - T_w) dz \\ + \frac{\rho_f(\rho_f - \rho_g)g \delta^2 \delta_{rev}}{2\mu_f} w (h_g - h_f^s) \end{aligned} \quad (14)$$

The first term in right hand side is the same to Nusselt derivation. Here, in order to further evolution of above equation we need some method to obtain the virtual re-vaporized layer thickness ( $\delta_{rev}$ ), which may be a further work.

The heat flux,  $q_{gas}$ , is different from the measure heat flux in wall, which is here the film heat flux,  $q_{film}$ . The HTC model for saturated steam may be corresponding to the gas heat flux,  $q_{gas}$ . If it is really such that, the HTC of saturated steam surely has some error, when applying to superheated steam.

Some more elaborate manipulation may make it possible to get the net HTC of superheated steam.

### 4. Review of System Code

The phasic energy equation in MARS-KS code has two terms that are related with wall heat transfer: One is direct heat transferred from wall ( $Q_{wg}$  in energy equation), and the other is heat addition by phase change term ( $\Gamma_w h'_g$ ). And the phase change term for condensation is calculated as

$$\Gamma_w = \frac{-Q_{ig}^w}{h'_g - h'_f} = \frac{-Q_{ig}^w}{h_g - h_f^s} \quad (17)$$

, where phasic enthalpies for condensation are defined as:  $h'_g = h_g$  and,  $h'_f = h_f^s$  respectively. Mass transfer of the condensate by superheated steam is treated in interfacial heat transfer. The wall condensation model in MARS-KS is all saturated steam condensation.

Conclusively, MARS-KS looks to consider unwillingly the re-vaporization.

Some the other codes such as CONTEMPT series adopts Tagami model and Uchida model for condensation, which are all for saturated steam in the presence of non-condensable gas. They use optionally  $h_g - h_f$  rather than  $h_{fg}$  when obtaining mass transfer rate from heat transfer rate. In addition they sometimes use re-vaporization fraction in their input.

### 5. Conclusions

Several relevant literatures were reviewed on the superheat steam condensation and the re-vaporization. They show consistent trends of superheated steam condensation phenomena regardless of their different approach. The heat transfer increases as the steam superheat get higher, because of the thinner film. The conventional codes do not fully reflect those so many study results on the superheated steam condensation. The codes usually use the saturation condensation models and the mass transfer of the ambient superheated gas is differently treated in the way of interfacial heat transfer. In some other code,  $h_g - h_f$  is used rather than  $h_{fg}$  in obtaining the condensation mass transfer, and in this case the condensation HTC model have to be one for saturated steam, with slight hesitation.

More accurate mechanism identification and models are necessary.

### NOMENCLATURES

$c_{pf}$	specific heat of film [J/kg-K]
$c_{pv}$	specific heat of superheated vapor [J/kg-K]
$g$	gravity [9.8m/s <sup>2</sup> ]
$h_c$	condensive heat transfer coefficient [W/m <sup>2</sup> -K]
$h_f$	liquid phase enthalpy [J/kg]
$h_{fg}$	latent heat [J/kg]
$h_g$	vapor phase enthalpy [J/kg]
$k_f$	thermal conductivity of liquidfilm [W/m-K]
$\dot{m}_{cond}$	condensed mass rate [kg/s]
$\dot{m}_{film}$	film mass flowrate [kg/s]
$\dot{m}_{rev}$	re-vaporized mass rate [kg/s]
$T_{ig}$	film surface temperature [K]
$T_w$	wall temperature [K]
$u(y)$	film velocity at y [m/s]
$q$	heat transfer rate [W]
$w$	wall depth [m]
$y$	horizontal coordinate across the film [m]
$z$	vertical directional coordinate [m]

#### Greek

$\delta$	film thickness [m]
$\delta_{rev}$	virtual re-vaporized water thickness [m]
$\rho$	density [kg/m <sup>3</sup> ]
$\mu_f$	film viscosity [N-s/m <sup>2</sup> ]

#### Superscript

$s$	saturation
$*$	no superheating

#### Subscript

$c$	condensation
$f$	liquid phase
$g$	gas phase
$i$	interface
$w$	wall

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