Development of Fluid to Fluid Scaling Criteria for Modeling Condensation in Horizontal Tubes

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1. Introduction

Following the accident in the Fukushima nuclear power plants, more focus has been directed to design and adapt new passive safety systems. The APR+ (Advanced Power Reactor Plus), which is currently being developed in South Korea, has adopted a Passive Auxiliary Feedwater System (PAFS), which removes the decay heat through the steam generator replacing the conventional active auxiliary feedwater system. The main part of the PAFS, in which the steam condenses, is the Passive Condensation Heat Exchanger (PCHX). The steam condenses under high pressure (7.4 MPa, 290 C) in 8.4m long horizontal tube, with 3 degrees inclination, with inner diameter of 44.8mm and wall thickness of 3mm. the PCHX consists of 4 bundles, each bundle has 60 condensation tubes.

A separate effect test facility using the PASCAL (PAFS Condensing Heat Removal Assessment Loop) experimental facility to verify the cooling and operational performance of the PAFS had been conducted. Evaluation of the one-dimensional safety analysis code MARS (Multidimensional Analysis of Reactor Safety) against the experimental data showed that the MARS code under predict the local heat transfer coefficient [1]. Several models were proposed to predict the condensation heat transfer in horizontal heat exchangers. However, due to the limited number of studies and experimental data, the prediction of the heat transfer and the heat capacity of such type of heat exchangers are still not precise. Therefore, experiments regarding condensation mechanism and heat transfer in horizontal tubes are meaningful and the resulting data are highly valuable.

Due to the high construction cost of a full size high pressure condensation experimental facility, a reduced-size reduced-pressure condensation experiment is suggested to evaluate the heat removal performance for a horizontal heat exchanger. This study attempts to develop a scaling criteria for modeling a high pressure water condensation system with low pressure Freon system.

2. Scaling Criteria Development

In accordance with the general scaling procedure, scaling of physical phenomena is based on a set of dimensionless groups, which can be derived from dimensionless forms of the governing equations or by the application of the Buckingham Pi theorem. In the current paper, the Buckingham Pi theorem was selected as the way to establish the scaling criteria for condensation in horizontal tubes. Later on, developing the nondimensional parameters based on the governing equations approach will be conducted.

For applying the Buckingham Pi theorem, the independent variables that determine the behavior of a particular dependent variable of interested should be firstly defined. For the local condensation heat transfer coefficient (α) in horizontal tubes it was found that it is dominated by the following 12 variables; the mass flux (G), tube diameter (d), liquid density (ρ_L), vapor density (ρ_V), liquid viscosity (μ_L), vapor viscosity (μ_V), surface tension (σ), liquid thermal conductivity (λ_L), liquid specific heat (C_{pL}), latent heat of vaporization (h_{LV}), gravity acceleration (g) and vapor quality (x).

The relation between the dependent parameter (α) and the independent variables can be expressed as:

$$g(\alpha, G, d, \rho_L, \rho_V, \mu_L, \mu_V, \sigma, \lambda_L, C_{pL}, h_{LV}, g, x) = 0$$

Based on the listed parameters, a dimensional matrix has been constructed. Since the problem includes heat transfer it is recommended to include the following fundamental dimensions in the dimensional matrix: mass (M), length (L), time (T), temperature (θ) and heat (H). The dimensional matrix is shown below:

$$\begin{array}{c|ccccccccccccc}
 & \alpha & d & \rho_L & \rho_v & G & \mu_L & \mu_v & \sigma & \lambda_L & Cp_L & h_{LV} & g & \chi \\
\hline
M & 0 & 0 & 1 & 1 & 1 & 1 & 1 & 1 & 0 & -1 & -1 & 0 & 0 \\
L & -2 & 1 & -3 & -3 & -2 & -1 & -1 & 0 & -1 & 0 & 0 & 1 & 0 \\
T & -1 & 0 & 0 & 0 & -1 & -1 & -1 & -2 & -1 & 0 & 0 & -2 & 0 \\
\theta & -1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -1 & -1 & 0 & 0 & 0 \\
H & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 1 & 1 & 0 & 0
\end{array}$$

It can be shown that the fifth order determinant of this matrix is not zero, hence the rank of this matrix is 5. Accordingly, $G, d, \rho_L, h_{LV}, and\ C_{pL}$ have been selected as the repeating parameters, so that 8 non-dimensional numbers were generated as shown in the table below.

Table 1: List of non-dimensional numbers generated

Pi term	Non-dimension number name
$\pi_1 = \alpha/GC_{pL}$	*(Not defined previously in the literature)*
$\pi_2 = \rho_v/\rho_L$	Vapor to quality density ratio
$\pi_3 = \mu_L/dG$	Liquid phase Reynold's number
$\pi_4 = \mu_v/dG$	vapor phase Reynold's number
$\pi_5 = \rho_L\sigma/dG^2$	Weber number
$\pi_6 = \lambda_L/dGC_{pL}$	*(Not defined previously in the literature)*
$\pi_7 = gd\rho_L^2/G^2$	Froude number
$\pi_8 = x$	Quality

After investigating the previous pi-terms and based on previous research of F.Xing [2] and Dobson [3] it was noticed that for large dimeter tubes the surface tension has less effect on condensation heat transfer, comparatively, the inertia force and gravity force become more important. So that the Weber number (π_5) was disregarded, from the previous list, as one of the factors affecting the condensation phenomena in the PAFS.

π_1 has been chosen as the dependent dimensionless variable, therefore, π_1 can be evaluated as accurately as the local condensation heat transfer coefficient (α). So the general local heat transfer coefficient prediction equation can be written as:

$$\pi_1 = F(\pi_2, \pi_3, \pi_4, \pi_6, \pi_7, \pi_8)$$

It follows that in order to design a model for predicting the local heat transfer coefficient the value of each Pi term of the prototype and the model should be equal, that is:

$$(\pi_2)_{prototype} = (\pi_2)_{model}$$
$$:: \qquad\qquad ::$$
$$(\pi_8)_{prototype} = (\pi_8)_{model}$$

However, satisfying the above condition is still not practical and the perfect similarity between the prototype and the model cannot be satisfied, that is $(\pi_1)_{prototype} \neq (\pi_1)_{model}$, and as it is stated in Ahmad's paper [4] that in fluid flow problems, the distorted models are the rule rather than the exception. In order to solve this distortion between the prototype and the model, Ahmad has used the so-called compensated distortion technique.

2.1 Compensated distortion technique

Ahmad [4] has developed a fluid-to-fluid modeling of critical heat flux from classical dimensional analysis and theory of models. Where the problem of multiple distortion was solved by introducing a modeling parameter. The technique of compensated distortion was employed in the development of the modeling parameter. A similar procedure is followed in this study to solve for the distortion in the scaling criteria developed for condensation in horizontal tube.

The idea of the compensated distortion technique is to compensate for an inequality in one or more dimensionless groups by introducing a controlled distortion in another. In order to apply this technique, the Pi terms that can be conveniently controlled in experimentation should be isolated. The terms π_2 and π_7 have been selected as the non-distorted parameters, since they can be controlled by controlling the mass flux, steam temperature and pressure, and the pipe geometry.

Since pressure should be fixed in order to control the liquid to vapor density ratio (π_2), the other Pi terms that include other physical properties, like viscosity (μ), thermal conductivity (λ) and specific heat (C_p), will be distorted. That is, those parameter (π_3, π_4, π_6 and π_8) were selected as the distorted parameters. Based on the distortion on the Pi terms, the modeling parameter (ψ_α) is defined as:

$$\psi_\alpha = F(\pi_3, \pi_4, \pi_6, \pi_8)$$

For the ease of the analysis of the previous equation, and since both π_3 and π_4 are form of the liquid phase and vapor phase Reynold's number, their ratio has been used (liquid-to-vapor viscosity ratio) and a new dimensionless number was formed, as follow:

$$\pi_{3,4} = \frac{\mu_L}{\mu_V}$$

$$\psi_\alpha = F(\pi_8, \pi_{3,4}, \pi_6) \qquad (1)$$

Equation (1) can be expressed by a product of power function over the range of variables of interest, as:

$$\psi_\alpha = x \times (\mu_L/\mu_V)^{n1} \times \left(\lambda_L/(dGC_{pL})\right)^{n2}$$

Following Ahmad's procedure, the values of n_1 and n_2 are determined from experimental data of condensation in horizontal pipes for two fluids over a series of vapor qualities and fixed mass flow rate. The Cavallini data [5] for condensation of R134a and R410A in horizontal 8mm diameter smooth tube have been used. It was found that the values of n_1 and n_2 are equal to 0.9657 and 0.9511; respectively.

2.2 The modeling design criteria

Based on the previous analysis, the condensation scaling which provide a relation between a prototype and model is defined as:

$$(\psi\alpha)_R = \left[x \times \left(\frac{\mu_L}{\mu_v}\right)^{0.9657} \times \left(\frac{\lambda_L}{dGC_{pL}}\right)^{0.9511}\right]_R = 1$$

$$\left[\frac{\rho_L}{\rho_v}\right]_R = 1$$

$$\left[\frac{gd\rho_L^2}{G^2}\right]_R = 1$$

Where (R) is the prototype to model ratio of the non-dimensional quantity.

Since π_1 has been chosen as the dependent dimensionless variable and in order to fully satisfy the similarity between the prototype and the model it was found, by analyzing different prototypes and models, that a factor $\left[\frac{\lambda_P}{\lambda_M}\frac{C_{pM}}{C_{pP}}\frac{G_P}{G_M}\right]$ should be used to satisfy the local heat transfer similarity between the prototype and the model. In other terms, the following relation should be satisfied.

$$(\pi_1)_{prototype} = \left[\frac{\lambda_P}{\lambda_M}\frac{C_{pM}}{C_{pP}}\frac{G_P}{G_M}\right](\pi_2)_{model}$$

$$\frac{\alpha_P}{G_P C_{pP}} = \left[\frac{\lambda_P}{\lambda_M}\frac{C_{pM}}{C_{pP}}\frac{G_P}{G_M}\right]\frac{\alpha_M}{G_M C_{pM}}$$

Since the aim of this scaling criteria is to predict prototype's local condensation heat transfer coefficient based on the model's local condensation heat transfer coefficient, the first is linked to the later, based on the previous equation, via the following relation:

$$\alpha_P = \frac{\lambda_{LP}}{\lambda_{LM}}\frac{{G_P}^2}{G_M^2}\alpha_M \qquad (2)$$

Where P and M refer to prototype and model

3. Scaling Criteria Validation

The developed scaling criteria have been validated by applying it to different benchmark problems, for each problem the heat transfer coefficient of the prototype and the model were calculated using different condensation model; namely, Thome model [6], Cavallini model [7] and modified Shah model [8]. Those three condensation models are widely used to design condensation heat exchangers.

The scaling criteria have been applied to 8 benchmark problems. In table 2 each prototypes' parameters are listed. And for each problem, the equivalent model parameters are listed in table 3. The scaling criteria have been validated on mass flux range [200-750] kg/m^{2}s, tube diameter from 8mm to 44.8mm, and steam quality of [0.1-0.9].

For each prototype-model pair; first, the prototype local heat transfer coefficient over the applied quality range has been calculated using one of the condensation models. Then the model's local heat transfer coefficient was calculated using the same condensation model, and then based on the model's local heat transfer coefficient the equivalent prototype condensation heat transfer coefficient (equation 2) was calculated. The ratio between the predicated local heat transfer coefficient (equation 2) and the prototype local heat transfer coefficient was plotted verses the steam quality. The results of the scaling criteria validation while applied on the previously mentioned three condensation models are presented in figures 1, 2 and 3.

Table 2: Benchmark problems' prototypes list.

Case no.	Fluid	Pressure (MPa)	Mass flux (Kg/m^{2}s)	Diameter (mm)
1	Water	7.4	250	44.8
2	Water	5	464	38.7
3	Water	1.5	221	26.6
4	Water	3.78	400	26.6
5	R134a	1.016	300	8
6	R134a	1.016	750	8
7	R134a	1.016	750	8
8	R125	2	750	8

Table 3: Models' parameters based on the scaling criteria

Case no.	Fluid	Pressure (MPa)	Mass flux (Kg/m^{2}s)	Diameter (mm)
1	R134a	1.2	348	37
2	R134a	0.79	662	34
3	R134a	0.235	338	24
4	R134a	0.593	578	24
5	Water	6.36	214	9.5
6	Water	6.36	533	9.4
7	R125	0.88	840	8.1
8	Water	13.5	497	10.5

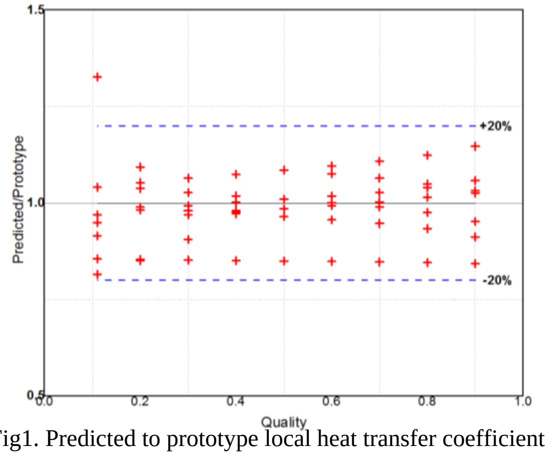

Fig1. Predicted to prototype local heat transfer coefficient ratio verses quality based on the Thome model.

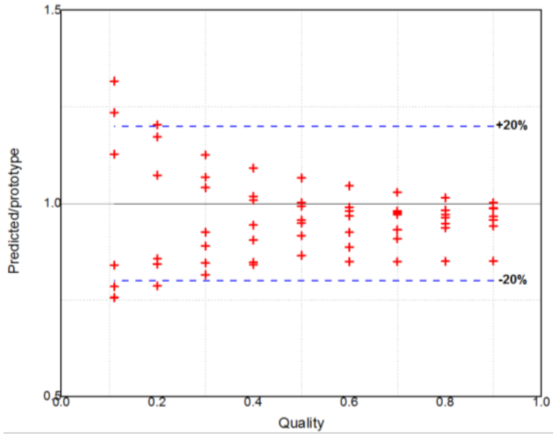

Fig2. Predicted to prototype local heat transfer coefficient ratio verses quality based on the Cavallini model.

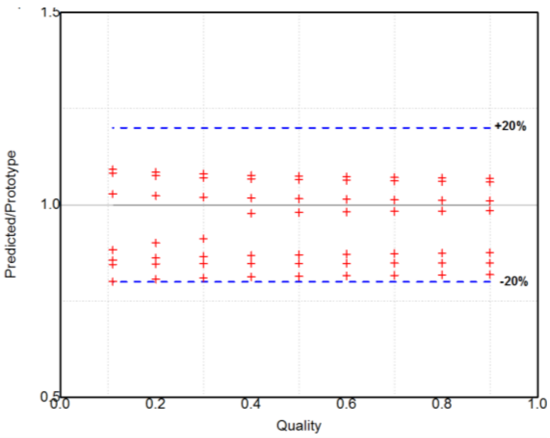

Fig3. Predicted to prototype local heat transfer coefficient ratio verses quality based on the modified Shah model.

The root mean square error (RMS), mean absolute deviation and average deviation between prototype heat transfer coefficient and the predicted heat transfer coefficient (equation 2) have been calculated for each condensation model as shown in table 4. Moreover, the flow patterns of prototypes and models based on each condensation model have been determined over the quality range and the percentage of matching the flow patterns between the prototype and the model has been calculated as shown in table 4 as well.

Table 4: Statistical analysis results

Model	RMS (%)	Av. Deviation (%)	Mean abs. deviation (%)	matching the flow patterns (%)
Thome	11.17	3.30	8.35	97.2
Cavallini	11.95	4.18	9.39	100
Modified Shah	11.35	5.08	6.69	100

3. Conclusions

In this study a scaling criteria have been developed to simulate a high pressure steam in horizontal tube condensation by a reduced pressure model using simulant fluid. The criteria have been developed based on the Buckingham Pi theorem and the compensated distortion technique used in Ahmad's paper [4]. The developed scaling criteria have been verified by comparing different prototypes and models using three different condensation models. The prototype-equivalent and the prototype local heat transfer coefficients showed good agreement with each other, based on the performed statistical analysis. As a future work, in order to verify the current scaling criteria, dimensional analysis based on the non- dimensionalization of the governing equations (continuity, momentum and energy) and the appropriate boundary conditions of each phase will be performed. The resulting scaling criteria will be compared with the scaling criteria proposed here and the necessary modifications will be applied, if required.

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