

# Modeling and code implementation of condensation in presence of non-condensable gases inside vertical tubes

J.C. de la Rosa<sup>a</sup>, J.L. Muñoz-Cobo<sup>a</sup>, A. Escrivá<sup>a</sup>

<sup>a</sup>Universidad Politécnica de Valencia, Spain.

**Abstract.** The Polytechnic University of Valencia (UPV) has elaborated a non-iterative model for the condensation in the presence of non-condensable gases inside vertical tubes. The total average relative error of this model gives a much better approach than the empirical-correlation UPV based model. On the other hand, the empirical-correlation model has been implemented in TRAC-BF1 code. The obtained results are similar to those obtained when the model is separately implemented in a FORTRAN code, giving a good approximation when comparing to the internal condenser wall temperature, total heat transfer coefficient, and steam mass flow rate from Vierow experimental data.

## 1. INTRODUCTION

During the second Research Coordination Meeting on the CRP on Natural Circulation Phenomena, Modeling, and Reliability of Passive Safety Systems that Utilize Natural Circulation, the Polytechnic University of Valencia (UPV) together with the Research Centre for Energy, Environment and Technology (CIEMAT) of Spain, has presented a paper concerning condensation with noncondensable-gases modeling.

One of the two models presented dealt with the Passive Containment Cooling Condensers (PCCC) of the ESBWR and SBWR reactors. A tube drains the steam plus non-condensable mixture from the containment and drives it to a distributor that transport the mixture by natural circulation to the upper headers of the PCCCs, as it is displayed in figure 1. The upper header is a large cylindrical deposit with the vertical tubes connected to it. This header distributes the gas mixture among the vertical tubes connected to it. Once the gas enters into the tubes, the steam condenses on the walls of the tubes, and the condensation heat is transferred to the PCCC pool. The condensate is drained by gravity to the lower header, were accumulates at the lower part of the lower drum where it is removed by gravity through the drain line which is directly connected to the reactor vessel. The non-condensed steam plus the non-condensable gases are discharged through the vent line into the wetwell pool.

Although the presented model has not a high average relative error, this is an empirical-correlation based model, taken from Vierow and Schrock experiments results. The UPV has performed a theoretical model in order to account for all the phenomena occurred. Furthermore, the model has also avoided the weakest point of the mechanistic models, that is, the iterative nature of the model, which makes a slowdown if the model had to be implemented in a code. In the proposed model, the presence of non-condensable gases is always accounted for, even for the liquid film thermal resistance computation. The obtained results are better than the empirical-correlation based model results.

On the other hand, the UPV has implemented the empirical-correlation based model, presented in last RCM, in the TRAC-BF1 code. The results are equals to those obtained with the model implemented separately in an in-house FORTRAN code. Here, these results have been compared with the wall temperature, heat transfer coefficient, and steam mass flow rate from Vierow experimental data, in order to check the adequacy of the model implementation.

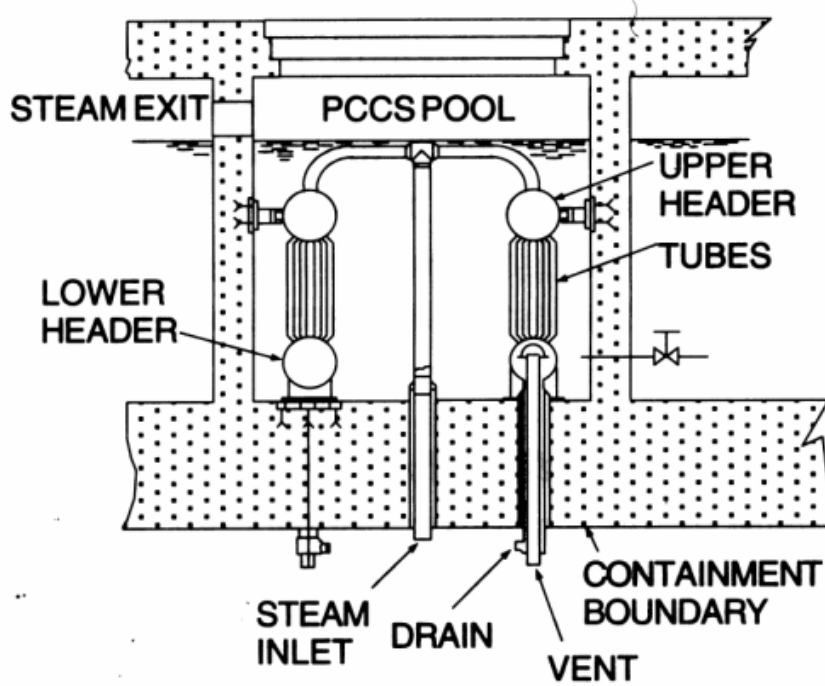


Fig. 1. Passive Containment Cooling Condensers of the ESBWR. The steam plus non-condensables mixture enters through the steam inlet connected to the drywell. In the lower drum the condensed steam is discharged in the reactor through the drain line and the uncondensed steam plus the non-condensable gases are discharged in the wetwell pool through the vent line.

## 2. UPV EMPIRICAL-CORRELATION BASED MODEL

The local heat transfer coefficient,  $h$ , is expressed as the product of the local heat transfer coefficient,  $k_l/\delta$ , without NC effects, and a degradation factor,  $f$ , due to NC and other effects, where  $k_l$  stands for the liquid film thermal conductivity, and  $\delta$  for the liquid film thickness. The  $f$  factor accounts for the degradation in the heat transfer due to the presence of NC gases in the gas mixture and also for the model defects to predict correctly the condensate layer thickness,  $\delta$ . In this way, the heat transfer coefficient is expressed as:

$$h = \frac{k_l}{\delta} \cdot f \quad (1)$$

For the thickness calculation, Muñoz-Cobo et al expression [1] is used:

$$\delta = \frac{1.259 \delta_N^{4/3}}{\left[ \delta_p \sum_{j=1}^2 a_j x_p^{j-1} + l_i \sum_{j=1}^3 b_j x_p^{j-1} + m_i \delta_p (c_1 + c_2 x_p) \right]^{1/3}} \quad (2)$$

where  $\delta_N$  stands for the condensate layer thickness for condensation over a plane plate (taken by Nusselt) [2], and can be calculated through two different formulations. The first one is the classical Nusselt formulation:

$$\delta_N = \left( \frac{4k_l \cdot z \cdot \mu_l \cdot (T_i - T_w)}{g \cdot h_{fg} \cdot \rho_l \cdot (\rho_l - \rho_m)} \right)^{1/4} \quad (3)$$

Where  $z$  is the axial distance from the top of the tube,  $\mu_l$  is the liquid viscosity,  $\rho_l$  and  $\rho_m$  are respectively the liquid and gas-vapor density,  $g$  is the gravity acceleration, and  $T_i$ ,  $T_w$  are respectively the interface and wall temperature.

The second one is the following:

$$\delta_N = 0.909 \text{ Re}_1^{1/3} l_e \quad \text{for } \text{Re}_1 < 30 \quad (4)$$

$$\delta_N = 1.2165 \text{ Re}_1^{0.22} l_e \quad \text{for } \text{Re}_1 > 30 \quad (5)$$

These two expressions came from a simple force balance, where Nusselt number has been taken as a function of Reynolds number according to Mills [3]:

$$\text{Nu} = 1.1 \text{ Re}_1^{1/3} \quad \text{for } \text{Re}_1 < 30 \quad (6)$$

$$\text{Nu} = 0.822 \text{ Re}_1^{0.22} \quad \text{for } \text{Re}_1 > 30 \quad (7)$$

With respect to the denominator:

$$\begin{aligned} a_1 &= 2; a_2 = -\frac{28}{15}; a_3 = -\frac{1}{3} \\ b_1 &= \frac{4}{3}; b_2 = -2; b_3 = \frac{8}{15}; b_4 = \frac{1}{3} \\ c_1 &= \frac{1}{2}; c_2 = -\frac{8}{15} \end{aligned} \quad (8)$$

$\delta_p$  as the condensate film thickness inside a vertical pipe, neglecting interfacial shear forces. This thickness is given by the expression [1]:

$$\delta_p = \frac{1.189 \delta_N}{\left(1 + a'_1 x_N + a'_2 x_N^2\right)^{1/4}} \quad (9)$$

$x_p$  is the ratio of  $\delta_p$ , and the inner tube radius  $R$ :

$$x_p = \frac{\delta_p}{R} \quad (10)$$

where  $x_N$  in equation, is the ratio of the condensate thickness and the tube radius:

$$x_N = \frac{\delta_N}{R} \quad (11)$$

and

$$a'_1 = -\frac{4}{5}; a'_2 = -\frac{1}{3} \quad (12)$$

Finally, the coefficients  $l_i$  and  $m_i$  depend on the interfacial shear stress, and the friction factor of the steam/Non Condensable (NC) mixture respectively, and are given by:

$$l_i = \frac{2 \tau_{if}}{(\rho_f - \rho_g^*) g} \quad (13)$$

and

$$m_i = \frac{180 f_g \rho_g (u_g - u_i)^2}{(\rho_l - \rho_g^*) g R} \quad (14)$$

Where  $\tau_{if}$  is the interfacial shear stress acting on the condensate layer,  $f_g$  is the friction factor of the steam/NC mixture,  $\rho_g^*$  is the fictitious density of the steam/NC mixture [1], and  $u_g$  is the steam/NC mixture velocity respectively,  $\rho_g$  is the mixture density. Finally,  $u_i$  is the condensate layer velocity at the interface.

With respect to the degradation factor,  $f$ , this has been obtained through a non-linear regression procedure using the MINUIT code [4]. The degradation factor has been defined as the ratio between the real heat transfer coefficient ( $h_{nc}$ ) and the theoretical condensation heat transfer coefficient without the boundary layer thermal resistance ( $h_{th}$ ):

$$f = \frac{h_{nc}}{h_{th}} = h_{exp} \cdot \frac{\delta}{k_l} \quad (15)$$

Because the theoretical heat transfer coefficient has accounted both for the thinning effect of the film layer due to the shear stress produced by the gas mixture, and for the wavy effects of the condensate layer, the degradation factor only should include the non-condensable gases effect:

$$f = 1 - P_1 M_{NC}^{P_2} + P_3 e^{-(P_4 M_{NC})} \quad (16)$$

Where the fitted parameters are:

$$P_1 = 1.0266 ; P_2 = 0.12791 ; P_3 = 0.78074 ; P_4 = 29.122 \quad (17)$$

The average relative error for a total of 282 points in 23 gas-vapor experiments<sup>1</sup> from Vierow and Schrock data base is 45%.

For its simplicity, this will be the first implemented model in TRAC-BF1 thermal-hydraulic code, as it was mentioned above.

### 3. UPV NON-ITERATIVE MODEL

Heat transfer from the bulk gas to the wall tube will consist of two thermal resistances coupled in series:

$$1/h_t = 1/h_l + 1/h_g \quad (18)$$

where  $h_t$  is the total heat transfer coefficient,  $h_l$  is the liquid film heat transfer coefficient, and  $h_g$  is the gas-vapor boundary layer heat transfer coefficient.

---

<sup>1</sup> In fact, the gas-vapor experiments carried out by Vierow were 25, but two of them are lacking in interest in order to validate a model because their results were anomalous, as Vierow herself verified.

The liquid film thermal resistance has been calculated as in the empirical-correlation based model:

$$h_l = \frac{k_l}{\delta} \quad (19)$$

where  $\delta$  is supposed to be small enough to neglect the convective liquid transport. The thickness is computed as before. How this term is computed will be explained below.

With respect to the gas-vapor boundary layer, it must account for both the vapor phase change energy transfer, and the convection due to the temperature jump between bulk and interface:

$$h_g = h_c + h_s \quad (20)$$

where  $h_s$  and  $h_c$  are respectively the sensible and condensation heat contribution, both calculated from a Stanton correlation:

$$St = \frac{h_s}{\rho_g u_g C_{pg}} \quad St_m = \frac{h_m}{\rho_g u_g} \quad (21)$$

Where  $\rho_g$ ,  $u_g$ ,  $C_{pg}$  are respectively the gas mixture density, velocity, and specific heat. Relative to the numerator,  $h_m$  is the mass transfer coefficient, formulated from heat and mass transfer analogy, where the condensation mass flux,  $m''_{cond}$ , has been taken as the result of a convective and a diffusion contribution (Fick's law):

$$m''_{cond} = [-\rho D \nabla W_v]_i + (W_v m''_{tot})_i = -h_m \frac{W_{v,i} - W_{v,b}}{1 - W_{v,i}} \quad (22)$$

$W_v$  is the steam mass fraction computed at the interface ( $i$  subscript) or at the bulk ( $b$  subscript),  $m''_{tot}$  is the gas-vapor flux moving to the interface, and  $D$  is the diffusion coefficient of the vapor in the air. Then,  $h_m$  will be:

$$h_m = \frac{\rho D}{\delta_{diff}} \quad (23)$$

Being  $\delta$  the gas-vapor diffusion boundary layer thickness.

With respect to the condensation heat transfer,  $h_c$ , this can be defined as follows:

$$h_c = \frac{m''_{cond} h_{pfg}}{(T_b - T_i)} \quad (24)$$

Where  $T_b$  is the bulk temperature;  $h_{pfg}$  accounts for the phase change enthalpy plus the subcooling term, developed by Rohsenow [5]:

$$h_{pfg} = h_{fg} \left[ 1 + 0.68 \left( \frac{C_{pl}(T_i - T_w)}{h_{fg}} \right) \right] \quad (25)$$

The iterative nature of the model lies in that the interface temperature is implicit, on the one hand, in the liquid film thermal-hydraulic properties calculation, as well as in the friction factor and in the condensation mass flux, and on the other hand, in the Nusselt liquid film thickness.

To avoid the condensation term iteration, No and Park [6], from the Korea Advanced Institute of Science and Technology, have developed a condensation heat transfer formulation in which the interface temperature does not take part in the calculations.

From a simple heat balance it is possible to obtain the following equation:

$$h_l(T_i - T_w) = (h_c + h_s)(T_b - T_i) \Rightarrow (T_b - T_i) = \frac{h_l}{h_l + h_c + h_s}(T_b - T_w) \quad (26)$$

Expanding the vapor interface mass fraction from Taylor series:

$$\begin{aligned} W_{v,i} &\approx W_{v,b} + \left. \frac{\partial W_v}{\partial T} \right|_b (T_i - T_b) \rightarrow 1 - W_{v,i} = 1 - W_{v,b} - \left. \frac{\partial W_v}{\partial T} \right|_b (T_i - T_b) = \\ &= 1 - W_{v,b} + \frac{h_l}{h_l + h_c + h_s}(T_b - T_w) \left. \frac{\partial W_v}{\partial T} \right|_b \end{aligned} \quad (27)$$

Returning now to the condensation mass flux equation, we obtain an expression where the interface temperature is avoided:

$$h_c = g i_{fg} \frac{\left. \frac{\partial W_v}{\partial T} \right|_b}{\left( 1 - W_{v,b} + \frac{h_l}{h_l + h_c + h_s}(T_b - T_w) \left. \frac{\partial W_v}{\partial T} \right|_b \right)} \quad (28)$$

Using now the Clausius-Clapeyron equation, it is possible to expand the partial derivative:

$$\frac{\partial W_v}{\partial T} = \frac{\partial W_v}{\partial P_v} \frac{\partial P_v}{\partial T} = \frac{1}{P} N_A \frac{\partial P_v}{\partial T} \approx \frac{i_{fg} \rho_v}{PT} N_A \quad (29)$$

Where  $N_A$  is a constant depending on the vapor and gas molar weights and mass fractions:

$$N_A = \frac{M_v M_g}{[M_g(1 - X_v) + M_v X_v]^2} \quad (30)$$

Finally, a second degree equation in  $h_c$  with constant coefficients -so calculated solutions are exact- is obtained:

$$h_c = \frac{-B \pm |B| \sqrt{1 - 4AC/B^2}}{2A} \quad (31)$$

Where:

$$\begin{aligned} A &= 1 - W_{v,b} & H_1 &= (h_c + h_s) & B_{2T} &= \frac{i_{fg} \rho_v}{PT} (T_b - T_w) N_A \\ B &= H_1 A + H_2 B_{2T} - B_{3T} & H_2 &= h_l & & \\ C &= -H_1 B_{3T} & B_{3T} &= \frac{g i_{fg}^2 \rho_v}{PT} N_A & & \end{aligned} \quad (32)$$

As A is always positive, and C is always negative, then we have only one solution:

$$h_c = \frac{-B + |B| \sqrt{1 - 4AC/B^2}}{2A} \quad (33)$$

Owing to the non-iterative nature of the model of No and Park, the heat transfer coefficient for the liquid film was computed considering the interface temperature as the saturated vapor temperature at its vapor partial pressure. In reality, the interface temperature drops to a value that already does not depend only on the vapor partial pressure in the bulk, but above all depends on the mass diffusion process through the gas boundary layer, which causes a vapor concentration gradient in a way that the vapor mass fraction (and so, the vapor partial pressure) is lower than in the case without the mass diffusion process.

This means that the influence of the noncondensable gases over the liquid film thermal resistance was not considered.

The UPV has developed a way to take into account the influence of the noncondensable gases over the liquid thermal resistance. Let us consider again equation number (26). If this is called the real heat transfer,  $q_r''$  (expressed in terms of heat flux), and the heat transfer for vapor only is called  $q_t''$  (theoretical heat transfer), then:

$$q_r'' = q_t'' \frac{q_r''}{q_t''} = h_{lt} (T_{b,sat} - T_w) \frac{h_{lr}}{h_{lt}} \frac{(T_i - T_w)}{(T_{b,sat} - T_w)} \quad (34)$$

Where  $h_{lr}$ ,  $h_{lt}$ , are respectively the liquid film heat transfer coefficients for the real and theoretical case.

Expanding now the ratio between both heat transfer coefficients:

$$\frac{h_{lr}}{h_{lt}} = \frac{k(T_{li})}{k(T_{lb})} \frac{\delta(T_{lb})}{\delta(T_{li})} = \text{ratk} * \text{rat}\delta \quad (35)$$

Here,  $T_{li}$ ,  $T_{lb}$ , are respectively the average liquid temperature considering the real interface temperature and the saturated bulk temperature.

Expanding now the film thickness ratio,  $\text{rat}\delta$ , using expression (2):

$$\begin{aligned} \text{rat}\delta &= \frac{\delta(T_{lb})}{\delta(T_{li})} = \frac{1.259(\delta_N^*(T_{lb}))^{4/3}}{\left[ \delta_p(T_{lb})(a_1 + a_2x + a_3x^2) + l_i(T_{lb})(b_1 + b_2x + b_3x^2 + b_4x^3) + m_i(T_{lb})\delta_p(T_{lb})(c_1 + c_2x) \right]^{1/3}} \\ &= \frac{\left[ \delta_p(T_{li})(a_1 + a_2x + a_3x^2) + l_i(T_{li})(b_1 + b_2x + b_3x^2 + b_4x^3) + m_i(T_{li})\delta_p(T_{li})(c_1 + c_2x) \right]^{1/3}}{1.259(\delta_N^*(T_{li}))^{4/3}} = \quad (36) \\ &= \frac{\delta_N^{*4/3}(T_{lb}) \left[ 2\delta_p(T_{li}) + 4/3l_i(T_{li}) + 1/2m_i(T_{li})\delta_p(T_{li}) \right]^{1/3}}{\delta_N^{*4/3}(T_{li}) \left[ 2\delta_p(T_{lb}) + 4/3l_i(T_{lb}) + 1/2m_i(T_{lb})\delta_p(T_{lb}) \right]^{1/3}} = \\ &= \text{rat}\delta_N^{*4/3} \frac{\left[ 2\delta_p(T_{li}) + 4/3l_i(T_{li}) + 1/2m_i(T_{li})\delta_p(T_{li}) \right]^{1/3}}{\left[ 2\delta_p(T_{lb}) + 4/3l_i(T_{lb}) + 1/2m_i(T_{lb})\delta_p(T_{lb}) \right]^{1/3}} \end{aligned}$$

Since  $x$  is a very small value, it is possible to simplify the denominator polynomials getting only their constant coefficient, and having already subsituted these coefficients  $a_1$ ,  $b_1$ , and  $c_1$ , by their respective value.

If we consider that  $\text{rat}\delta_N^* \approx \text{rat}\delta_p$ , then:

$$\begin{aligned}
 \text{rat}\delta &= \text{rat}\delta_N^{*4/3} \cdot \frac{[2\delta_p(T_{li}) + 4/3l_i(T_{li}) + 1/2m_i(T_{li})\delta_p(T_{li})]^{1/3}}{[2\delta_p(T_{lb}) + 4/3l_i(T_{lb}) + 1/2m_i(T_{lb})\delta_p(T_{lb})]^{1/3}} = \\
 &= \text{rat}\delta_N^{*4/3} \cdot \frac{[2\delta_p(T_{li}) + 4/3l_i(T_{li}) + 1/2m_i(T_{li})\delta_p(T_{li})]^{1/3}}{[\delta_p(T_{lb})(2 + 4/3l_i(T_{lb})/\delta_p(T_{lb}) + 1/2m_i(T_{lb}))]^{1/3}} = \\
 &= \text{rat}\delta_N^{*4/3} \cdot \frac{[2\text{rat}\delta_p^{-1} + 4/3l_i(T_{li})/\delta_p(T_{lb}) + 1/2m_i(T_{li})\text{rat}\delta_p^{-1}]^{1/3}}{[(2 + 4/3l_i(T_{lb})/\delta_p(T_{lb}) + 1/2m_i(T_{lb}))]^{1/3}} \approx \\
 &\approx \text{rat}\delta_N^{*4/3} \cdot \frac{[(2 + 1/2m_i(T_{li})) + 4/3\text{rat}\delta_N^{*4/3}l_i(T_{li})/\delta_p(T_{lb})]^{1/3}}{[(2 + 4/3l_i(T_{lb})/\delta_p(T_{lb}) + 1/2m_i(T_{lb}))]^{1/3}}
 \end{aligned} \tag{37}$$

From the classical expression of Nusselt film thickness:

$$\text{rat}\delta_N^* = \left[ \frac{4k_l(T_{lb})(T_{\text{sat},b} - T_w)z\mu(T_{lb})}{gh_{fg}(T_{lb})\rho_l(T_{lb})^2} \cdot \frac{gh_{fg}(T_{li})\rho_l(T_{li})^2}{4k_l(T_{li})(T_i - T_w)z\mu(T_{li})} \right]^{1/4} \approx \left[ \text{rat}\mu \cdot \frac{(T_{\text{sat},b} - T_w)}{(T_i - T_w)} \right]^{1/4} \tag{38}$$

If we now cube the heat balance equation, (26), and being:

$$\begin{aligned}
 b &= 2 + m_i/2 \\
 c &= \frac{4}{3} \frac{l_i}{\delta_p(T_{lb})}
 \end{aligned} \tag{39}$$

where  $m_i$  and  $l_i$  have been evaluated to the bulk liquid temperature,  $T_{lb}$ , it remains:

$$\begin{aligned}
 h_{\text{diff}}^3 (T_b - T_i)^3 &= q_t''' \frac{(T_i - T_w)^3}{(T_b - T_w)^3} \text{rat}\delta_N^{*3} \frac{(b + \text{crat}\delta_N^*)}{\text{den}^3} = q_t''' \frac{(T_i - T_w)^3}{(T_b - T_w)^3} \frac{(b\text{rat}\delta_N^{*3} + \text{crat}\delta_N^{*4})}{\text{den}^3} = \\
 &= \frac{q_t''' (T_i - T_w)^3}{\text{den}^3 (T_b - T_w)^3} \left( b \left[ \text{rat}\mu \cdot \frac{(T_{\text{sat},b} - T_w)}{(T_i - T_w)} \right]^{3/4} + c \left[ \text{rat}\mu \cdot \frac{(T_{\text{sat},b} - T_w)}{(T_i - T_w)} \right]^{4/4} \right) = \\
 &= \frac{q_t''' (T_i - T_w)^3}{\text{den}^3 (T_b - T_w)^3} \text{rat}\mu \cdot \frac{(T_{\text{sat},b} - T_w)}{(T_i - T_w)} \left( b \left( 1/\text{rat}\mu \cdot \frac{(T_i - T_w)}{(T_{\text{sat},b} - T_w)} \right)^{1/4} + c \right)
 \end{aligned} \tag{40}$$

Since  $\left( 1/\text{rat}\mu \cdot \frac{(T_i - T_w)}{(T_{\text{sat},b} - T_w)} \right)^{1/4} \approx 1$ , and calling  $A = \left[ \frac{h_{\text{diff}} \text{den} (T_b - T_w)}{q_t'''} \right]^3 \frac{1}{(T_{\text{sat},b} - T_w)}$ ,

and  $D=A/(b+c)$ , finally we get a cubic degree equation in  $T_i$ :

$$\{-D/\text{rat}\mu\}T_i^3 + \{3T_b D/\text{rat}\mu - 1\}T_i^2 + \{-3T_b^2 D/\text{rat}\mu + 2T_w\}T_i + \{T_b^3 D/\text{rat}\mu - T_w^2\} = 0 \tag{41}$$

As the equation discriminant is always positive, there is only one real solution.

Once the interface temperature is known, is enough to take the gas-boundary layer heat transfer and substitute both the sensible and condensation heat transfer coefficients.

With respect to the viscosity ratio, two alternatives have been taken. The direct way of resolution is through a numerical approach, knowing that the temperature range is limited by  $T_{lb}$  and  $T_{li}$ . In this

way, we could implement an approximated ratio. Other way of resolution would be to expand the liquid viscosity at  $T_{li}$  by means of a Taylor series:

$$\mu(T_{li}) = \mu(T_{lb}) + \left. \frac{\partial \mu}{\partial T} \right|_{lb} (T_{li} - T_{lb}) + \left. \frac{\partial^2 \mu}{\partial T^2} \right|_{lb} (T_{li} - T_{lb})^2 + \dots \approx \mu(T_{lb}) + \left. \frac{\partial \mu}{\partial T} \right|_{lb} (T_{li} - T_{lb}) \quad (42)$$

Where the simplification is valid since the temperature range is small. If the liquid temperature is computed as follows:

$$T_l = T_w + F \cdot (T_i - T_w) \quad (43)$$

Where F has been taken as 0.25 -the experimental value obtained by Drew [7]-, then:

$$(T_{li} - T_{lb}) = -(T_{sat,b} - T_i) \quad (44)$$

so we can put the viscosity as a temperature function:

$$\mu(T_{li}) \approx \mu(T_{lb}) - \left. \frac{\partial \mu}{\partial T} \right|_{lb} (T_{sat,b} - T_i) \quad (45)$$

If this equation is implemented, then we will get a quartic equation. This method is already being performed by means of the Euler algorithm for the quartic equation. The first results seem to indicate that the quartic equation alternative give a better approach than the cubic equation alternative.

### 3.1. Empirical correlations and adjustments

We have only used a simple correlation that has been adapted for the condensation heat transfer in terms of the heat and mass transfer. The used correlation is the Reynolds analogy, which holds that:

$$St = \frac{C_f}{2} Pr^{-1} \quad (46)$$

For the friction factor, we have taken the Wallis' expression [8]:

$$f_i = f_m \left( 1 + 360 \frac{\delta}{d} \right) \quad (47)$$

where  $f_m$  is the fanning friction factor.

Finally, we have considered the influence of the flow development by means of the Siddique formulation [9]:

$$Nu_{dvp} = Nu \left[ 1 + \frac{0.8(1 + 70000 Re_m^{-1.5})}{z/d} \right] \quad (48)$$

Although the model is not *frozen* in relation with other possible improvements (i.e., film waviness, suction factors), it already gives good results.

### 3.2. Results

Next, a comparison is made of the average relative error for the Vierow gas experiments. We have compared the UPV iterative model, the Korean model<sup>2</sup> implemented with our correlations, and the new UPV cubic equation non-iterative model. It is possible to see that the latter is the model which gives the better results, although the difference with the Korean model is not very high.

run number	UPV iterative case error	No and Park case error	UPV non-iterative case error
4	0.113259966	0.1585464	0.3562436
5	0.164837833	0.1274604	0.2583651
6	0.53701899	0.1679732	0.2762653
7	0.289708611	0.1829378	0.3098646
8	0.2875168	0.1283495	0.3081870
9	0.256264077	0.1459634	0.1197055
11	0.236515593	0.5907647	0.4582885
12	0.271798997	0.5326898	0.3106946
13	0.26344855	0.4123574	0.2422042
20	0.340850822	0.7198645	0.6369835
21	0.183734786	0.3892861	0.2236620
22	0.424756133	0.2508551	0.3986355
23	0.355369277	0.2402395	0.1739054
24	0.456424327	0.4713369	0.4778577
26	0.542696636	0.2003823	0.1714238
27	2.137466531	1.0929710	0.4203402
28	0.293599615	0.0639672	0.1973645
30	0.456330778	0.1094049	0.1653917
31	0.287597429	0.2246998	0.2451365
32	0.780690985	0.1603985	0.1697962
34		0.4304019	0.2452696
35		1.1756401	0.4617317
36		0.1607725	0.1086726
	<b>Total average error</b> 0.433994337	<b>Total average error</b> 0.35379404	<b>Total average error</b> 0.29286910

## 4. MODEL IMPLEMENTATION IN TRAC-BF1/MOD2 VERSION 2001.1

As it was mentioned before, we have implemented the UPV empirical-correlation based model, mainly due to its simplicity. Some main features of TRAC-BF1 are the followings:

- It is a best-estimate code with capacity for three-dimensional components.
- It has a two-fluid formulation, with the possibility of including noncondensable gases (only air) in the gas phase. The air is supposed to be perfectly mixed with the vapor, forming a Gibbs-Dalton mixture.
- It is based on a six conservation equation set (an additional conservation mass equation for the air is included).

---

<sup>2</sup> The original Korean model using Dipprey's Stanton correlation, when comparing with Vierow databank, gives worse results (a total average error of 57%).

#### 4.1. General structure of TRAC-BF1/MOD2 related with the condensation model

The variables involved in the condensation process are:

1. The wall heat transfer coefficients, both with the liquid and gas phase ( $h_v, h_l$ ).
2. The interfacial heat transfer coefficients, both with the liquid and gas phase ( $h_{ig}, h_{il}$ ) and the interfacial area ( $A_i$ ).
3. The mass-related terms.

The basic equations as implemented in TRAC (and with TRAC notation) related with these parameters are:

$$\begin{aligned}
 QEVAP &= h_{wl} A_{wl} (T_w - T_l) \\
 \Gamma_{wall} &= \frac{QEVAP}{h_{fg}} \\
 q_i &= h_{il} A_i (T_s - T_l) + h_{ig} A_i (T_s - T_v) \\
 \Gamma_{interface} &= \frac{q_i}{h_{fg}}
 \end{aligned} \tag{49}$$

Where  $T_s$  is the saturated temperature at vapor partial pressure.

Inside *Prepass* first stage, heat transfer coefficients, together with the wall energy term (QEVAP), related with the wall mass transfer rate ( $\Gamma_{wall}$ ), are computed.

After the *Prepass* stage, inside the calculation of the field equations, the interfacial heat terms (heat transfer coefficients and area) are computed.

The main subroutines that deal with the condensation calculation are the followings:

- HTCOR, which calculates the convective wall heat transfer coefficient and QEVAP. HTCOR is the subroutine which determines the wall heat transfer regime. Each regime has a flag called XIDREG.
- HTPIP, which calls HTCOR and multiplies QEVAP by the wall area. This subroutine is used for a PIPE component, as it is our case.
- TF1E. This subroutine solves the momentum equation, and it is called by the TF1D, which performs the solution to the 1-D hydrodynamic equations.
- TF1I. This subroutine solves the mass and energy equation, and it is also called by the TF1D subroutine. In these two equations, the heat and mass terms written above are involved.
- HEATIF, called by TF1D, computes liquid and vapor heat transfer coefficients.

#### 4.2. Default model for condensation in TRAC-BF1

The model implemented in TRAC computes the condensation in the presence of non-condensable gases based on a complex series of correlations, at no time conceived for the condensation inside vertical tubes.

If wall temperature is lower than saturation temperature at partial vapor pressure, and void fraction is higher than 0.5, then two-phase condensation mode is activated. First, a Nusselt horizontal tube equation is computed:

$$h_l^{\text{Nusselt}} = 0.725 \left[ \frac{\rho_f^2 \cdot g \cdot h_{fg} \cdot k_f^3}{\mu_f (T_v^{\text{sat}} - T_w) D_h} \right]^{0.25} \quad (50)$$

being  $h_v = 0$

If the quality is less than 0.71, these coefficients will be used unless adjusted for a high void fraction.

At higher qualities the liquid film becomes thin, and a lineal interpolation in [0.71,0.999] and  $[h_l^{\text{Nusselt}}, 0]$  is made to find out  $h_l$ . In the case that noncondensable gases exists, this coefficient will be multiplied by a correction factor due to Isachenko:

$$f_{\text{isachenko}} = 0.168 \left[ \frac{\alpha \cdot (\rho_v - \rho_a)^2}{(1 - \alpha) \rho_a \rho_v} \right]^{0.1} \quad (51)$$

For the vapor-wall heat transfer coefficient,  $h_v$ , this is obtained by interpolation between zero at  $X=0.71$  and the maximum of the Dittus-Boelter, McAdams, and modified Kays correlations at  $X=0.999$ :

$$h_v^{\text{Dittus-Boelter}} = 0.023 \frac{k_v}{D_h} \text{Re}_v^{0.8} \text{Pr}_v^{1/3} \quad (52)$$

$$h_{v,\text{nc}}^{\text{McAdams}} = 0.13 \frac{k_v}{D_h} \left( \frac{\text{Gr}_v}{\text{Pr}_v} \right)^{1/3} \quad (53)$$

$$h_{v,\text{lam}}^{\text{modified Kays}} = 4 \frac{k_v}{D_h} \quad (54)$$

Next, Chen correction is used to account for the two-phase effect in the forced convection correlation if void fraction is higher than 0.05, that is, if some vapor exists in the control volume, another coefficient has to be computed:

$$h_l^{\text{forc}} = \max(h_{l,\text{turb}}^{\text{Dittus-Boelter}}, h_{l,\text{nc}}^{\text{McAdams}}, h_{l,\text{lam}}^{\text{Kays modificado}}) \quad (55)$$

This time, McAdams and modified Kays formulations are calculated with the liquid properties, whereas Dittus-Boelter expression uses a different Reynolds number formulation:

$$F = 2.35 \cdot (\chi_{\text{TT}}^{-1} + 0.213)^{0.736} \quad (56)$$

$$\text{Re}_{\text{Dittus-Boelter}} = \text{Re}_l \cdot F$$

Where the Lockhart-Martinelli factor is first calculated as follows:

$$\chi_{\text{TT}}^{-1} = \left( \frac{X}{1-X} \right)^{0.9} \left( \frac{\rho_l}{\rho_g} \right)^{0.5} \left( \frac{\mu_g}{\mu_l} \right)^{0.1} \quad (57)$$

The highest of the three heat transfer coefficients is then chosen as  $H_{\text{forc}}$ . Finally:

$$h_l = (1 - \alpha) \cdot H_{\text{forc}} + \alpha \cdot h_{l,\text{old}} \quad \text{for } T_s - T_w \geq 5$$

$$h_l = \frac{1 - \alpha(T_s - T_w)}{5} H_{\text{forc}} + \frac{\alpha(T_s - T_w)}{5} H_{\text{forc}} \quad \text{for } T_s - T_w < 5 \quad (58)$$

The vapor heat transfer coefficient is unmodified.

Last modification is related with the void fraction value:

$$h_l^\alpha = \left( \frac{0.999 - \alpha}{0.009} \right) \cdot h_{l,\text{old}} \quad \text{for } 0.99 < \alpha \leq 0.999$$

$$h_l^\alpha = h_{l,\text{old}} \quad \text{for } 0.75 \leq \alpha < 0.99 \quad (59)$$

$$h_l^\alpha = \left( \frac{\alpha - 0.5}{0.25} \right) \cdot \left( \frac{0.999 - \alpha}{0.009} \right) \cdot h_{l,\text{old}} + \left( \frac{0.75 - \alpha}{0.25} \right) \cdot H_{\text{forc}} \quad \text{for } 0.50 < \alpha < 0.750$$

A later modification of the Penn State University corrected these values with new interpolations:

$$h_{\text{VCOND}} = 7.55 \frac{k_f}{D_h} (\text{Re}_m \cdot \text{Pr}_m)^{0.33333} \quad \text{if } \text{Re}_m < 76132$$

$$h_{\text{VCOND}} = 0.0398 \frac{k_f}{D_h} \text{Re}_m^{0.8} \cdot \text{Pr}_m^{0.33333} \quad \text{if } \text{Re}_m \geq 76132 \quad (60)$$

Where Reynolds number is defined as follows:

$$\text{Re}_m = \frac{u_m \cdot \rho_m \cdot D_h}{\mu_f} \cdot \left[ (1 - x_{\text{AK}}) + x_{\text{AK}} \sqrt{\frac{\rho_f}{\rho_v}} \right] \quad (61)$$

And the  $x_{\text{AK}}$  factor is:

$$x_{\text{AK}} = \frac{\alpha \cdot \rho_v}{\alpha \cdot \rho_v + (1 - \alpha) \cdot \rho_f} \quad (62)$$

Then, a heat balance is made between the heat arriving to the wall from the liquid, and that arriving from the vapor to the interface:

$$q''_{\text{liquido-wall}} = h_{\text{VCOND}} \cdot (T_f - T_w) \quad (63)$$

$$q''_{\text{gas-liquido}} = h_v \cdot (T_s - T_s^{\text{sat}}) \quad (64)$$

If the heat arriving to the interface from the gas phase is lower than the heat transferred from the liquid to the wall, then the condensation heat is computed as follows:

$$h_l = h_{\text{VCOND}}$$

$$h_v = h_v \quad \alpha > 0.75 \quad (65)$$

$$q_{\text{evap}} = q''_{\text{cond}} = h_l \cdot (T_w - T_f)$$

$$\begin{aligned}
 h_l &= h_l^\alpha - \left( \frac{0.75 - \alpha}{0.25} \right) \cdot (h_l^\alpha - h_{VCOND}) \\
 h_v &= h_v & 0.75 > \alpha > 0.5 \\
 q_{evap} &= q''_{cond} = \left( \frac{\alpha - 0.5}{0.25} \right) \cdot h_{VCOND} \cdot (T_f - T_w)
 \end{aligned} \tag{66}$$

$$\begin{aligned}
 h_l &= h_l^\alpha - \left( \frac{0.75 - \alpha}{0.25} \right) \cdot (h_l^\alpha - h_{VCOND}) \\
 h_v &= 0.0 & \alpha < 0.5 \\
 q_{evap} &= q''_{cond} = \left( \frac{\alpha - 0.5}{0.25} \right) \cdot h_{VCOND} \cdot (T_f - T_w)
 \end{aligned} \tag{67}$$

If the heat balance has the contrary sign, then:

$$\begin{aligned}
 h_l &= 0.0 \\
 h_v &= h_v \cdot \frac{T_s - \max(T_s^{sat}, T_w)}{T_s - T_w} \\
 q_{evap} &= q''_{cond} = 0.0
 \end{aligned} \tag{68}$$

As it can be seen, the system is very complicated, although covering any kind of condensation. The correlations are clearly inadequate, first, because the main condensation correlation is based on horizontal tubes, and also because the correction factors are not designed for this type of configuration.

### 4.3. Modifications made by the UPV

As it was mentioned before, HTCOR subroutine determines the heat transfer regime. Each regime has a flag called XIDREG. To implement the UPV model, a new heat transfer regime has been added. The following changes have been made:

1. First, for choosing the new heat transfer regime, two criteria of selection were enforced:
  - a. A void fraction higher than 0.9 (since the implemented correlations are developed considering annular film flow).
  - b. The component in which condensation takes place must be vertical. It requires a 1.0 value of GRAVA variable (which measures the cosine of the angle with the gravity direction).
2. These criteria have been implemented at the end of the rest of the calculations, in a way that, if the above conditions are fulfilled, then the new calculations will be carried out, and if not, all will remain equal, and the new regime will be skipped.
3. The aim has been to modify the code as less as possible. For this reason, a new subroutine has been made. The lines added to HTCOR are:

```

Cupv
  IF(ALP .GT. .90D0 .AND. GRAVA .EQ. 1.00)THEN
    XIDREG=XIDREG+8.D0
    CALL HTCxx(REL,VISL,ROL,GC,REV,PA,P,CL,HVxx)
    HL = HVxx
    QEVAP=HL*(TW-TL)
  ENDIF

```

Cupv

Here we can see the inclusion of the new flag for this kind of regime, XIDREG=XIDREG+8.D0.

It is also interesting to comment some aspects about how TRAC-BF1/mod2 deals with condensation. As it can be seen above, the heat transfer coefficient calculated is the liquid-to-wall coefficient, since in this kind of regime an annular configuration is quickly established, and therefore, the walls will be wetted by the liquid. But there are also some discrepancies between how the code sees the inside tube condensation process, and how reality is. If TRAC model were based on a reality configuration, then the heat will all be transferred from the vapor to the liquid and, coupled in series, from the liquid to the vapor. The main inconvenience is that the condensation model was not implemented for this kind of configurations, and even if an annular flow configuration exists, the interfacial heat transfer coefficients are not those which have to be, that is, the boundary layer heat transfer coefficients. Due to the value of the interfacial heat coefficients already implemented in the code, the heat transfer from the vapor to the liquid is not equal to the reality transfer. In this way, because of the need of dealing with the heat transfer already implemented, to avoid the interfacial transfer was thought until a case was ran, and its low influence was checked. The interfacial heat (and mass) transfer contribution was almost negligible comparing to the wall transfer. It is even clearer since before the Penn State University contributions, a wall condensation mass generation was avoided. In fact, the name of the value that accounts for the condensation rate is called QEVAP, taken its name from the subcooled boiling regime, that is, the only regime where a mass generation was considered. So as the liquid temperature is almost the vapor temperature, the heat transfer can be computed in the way we did (and also, obviously, the condensate mass transfer, as this is the total heat transfer (computed by means of QEVAP) divided by the phase change enthalpy). As the interfacial contribution were negligible, the interfacial area was not very important. This factor also was different respect the reality. TRAC code has a minimum thickness in order to avoid a total annular configuration if the void fraction is higher than a fixed value. In our case, TRAC calculated some drop configuration, so not a total liquid wetted wall. But as it was said above, this factor was not important, as the interfacial factors were negligible comparing to those of the wall, so this was neither modified.

With respect to the wall heat contribution, the same occurs with the vapor-to-wall heat transfer. As the wetted area by the vapor is really small, and also because its heat transfer coefficient is much smaller than the liquid one, the vapor to wall contribution is negligible comparing to the liquid contribution. For this reason, it is not important to modify the wetted wall area in order to fix a total annular film. In the reality, the vapor drag force is enough high to shrink the liquid film even with a large void fraction, so violating the condition of maximum void fraction imposed by TRAC.

#### **4.4. Input deck. Vierow Experiments**

To validate the TRAC implemented model, Vierow experiments [10] carried out in the University of California at Berkeley (UCB) have been used.

The natural convection facility is composed by a 1.42 meter length and 10 centimeter diameter cylinder, which is the *lower plenum*. From one side of it, the *raiser* leaves out transporting the gaseous mixture to the higher part of the condenser. The condenser is a copper made 2.1 meters length, and has a 22 mm internal diameter tube and a 25.4 mm external diameter. The lower part of the condenser is welded to a very thin tube, the *downcomer*, which collects both the non-condensate and condensate steam, and the non-condensable gases, to the lower plenum. In the external part of the condenser a concentric tube 48.3 millimeters internal diameter acts like a heat sink, inside of which a counter-current cooler flows. The external tube is made by four welded sections, each one 50 centimeters length, and among them, between the external and internal tube, there is a small holed ring.

The experiments were performed as follows: the heater power was adjusted, and the valve which communicated the steam with the natural convection loop was open. A little later, when the steam has heated the loop and steady state conditions were achieved, the cooler loop was opened, taking place a transient scenario until the desired conditions were achieved. The vapor reached the lower plenum, where it get mixed with the non-condensate vapor that came from the downcomer; then, the vapor got into the riser going up to the condenser, closing the steady cycle. The non-condensable gases were added to the lower plenum in a fixed quantity, and then were dragged by the vapor and transported all

over the loop. The condensate level was controlled and kept constant in order to avoid oscillations or pressure changes in the system.

To implement it in the code, we have only focused on the condenser section, which can be isolated if the appropriated boundary conditions are applied. In this way, knowing the cooler flow rate, as well as the flow, composition, pressure and temperature of the gaseous mixture at the entrance of the condenser, it will be possible to model the test performance.

The condenser is modeled with two PIPE code components, each one connected to two boundary conditions: pressure, by means of a BREAK component, and velocity and composition, by means of a FILL component.

The only problem we have faced is about the holed rings between the condenser and the cooler tube. These rings create turbulent currents to homogenize the axial cooler temperature, but are impossible to simulate in the code. As the code has not the possibility to fix the wall (condenser) temperature as a boundary condition, to achieve equal axial distribution temperatures than the experimental was crucial. What we made was to modify the cooler PIPE cells cross section, in order to vary the forced convection coefficient of that section and achieve the appropriate axial temperatures distribution.

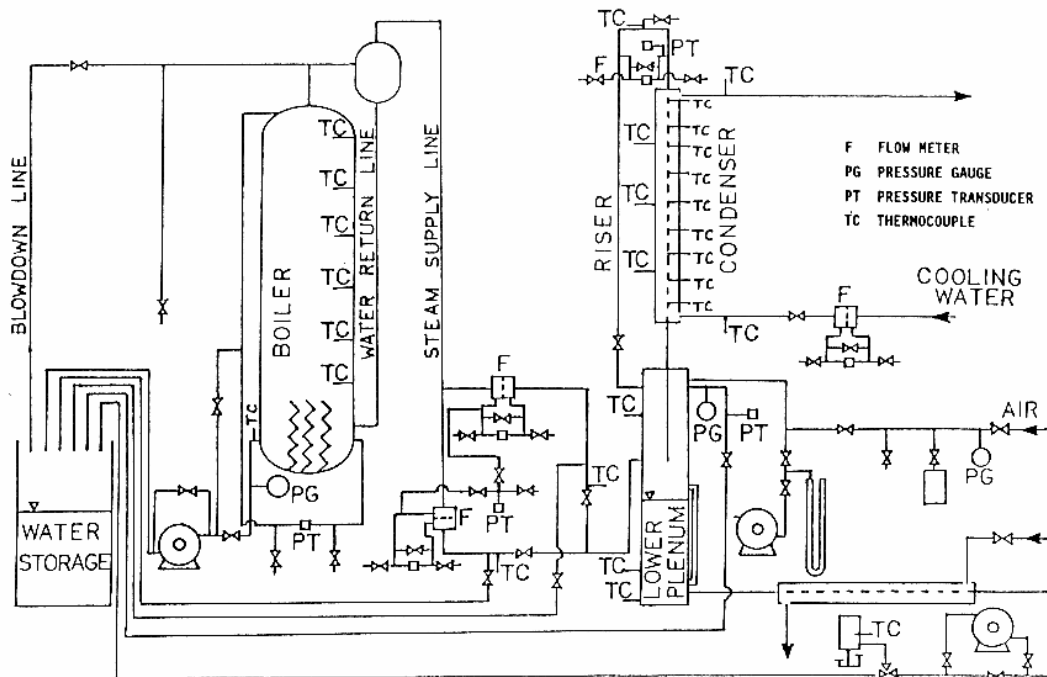


Fig. 2. Schematics of UCB experimental facility.

#### 4.5. Results and discussion

Due to the aim of reproduce the adequate cooler temperature profiles, the execution of one experiment became a very hard task. For this reason, the modeled experiments were limited up to six. Finally, we have implemented not only the UPV model, but the Vierow model too. The carried out tests were the number 4, 8, 9, 12, 13, and 28:

Table 1. Boundary conditions for the different implemented runs

Run	4	8	9	12	13	28
$\dot{m}''_{\text{steam+GNC}}$ (gr/s)	2.947	3.0302	1.8601	3.2411	2.02695	5.41986
$W_a^{\text{inlet}}$	0.00859	0.02429	0.04233	0.09122	0.12568	0.03730
Presión ( Pa )	105851.40	107780.29	168475.1	219284.5	178086.12	277388.53

Before showing the results for both implemented models, let us show a case running with the original condensation model developed in TRAC, in order to see how this condensation model works.

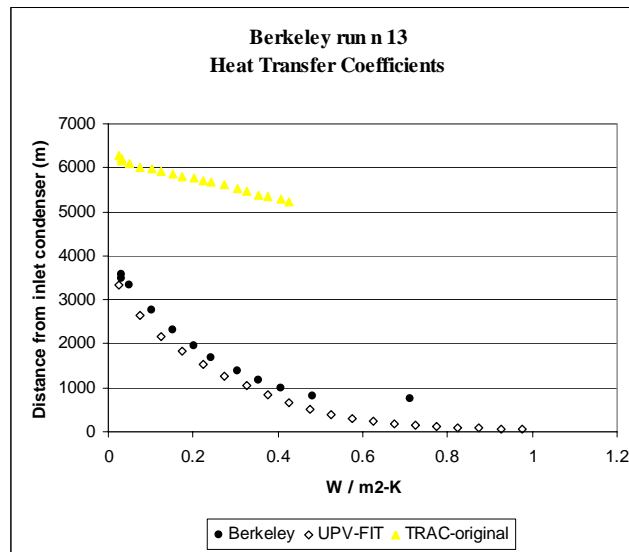


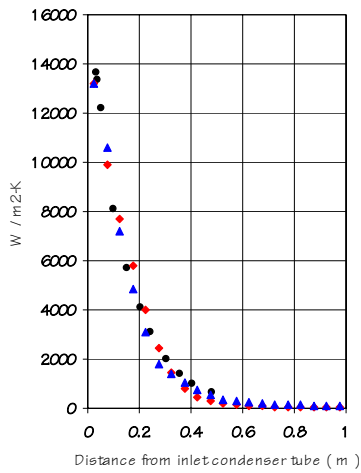
Fig. 3. HTC comparisons between TRAC original condensation model, experimental data, and UPV model.

From this figure it is possible to check that:

1. TRAC condensation without non-condensable gases implemented model yields a wrong solution: in the first values the code results are almost two times the experimental data.
2. It is well known that the evolution of the condensation with non-condensable gases follows a negative exponential tendency. This is true for the real and UPV model prediction, whereas TRAC model predicts a quasi-linear evolution.

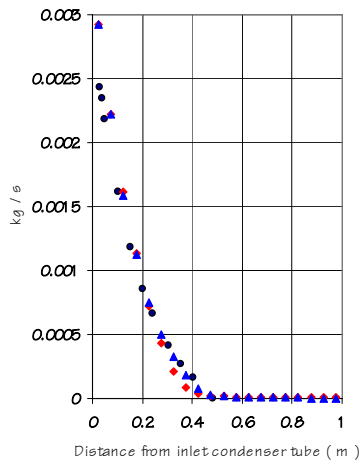
Next, the heat transfer coefficients, steam mass flow rate, and wall temperature figures are shown for different Vierow experimental facility runs. Although the tube is two meters length, we have only taken the first half part of the tube, since the non-condensable gases mass fraction at the middle is so high that heat transfer rates are negligible.

Berkeley run n°4  
Heattransfer coefficients



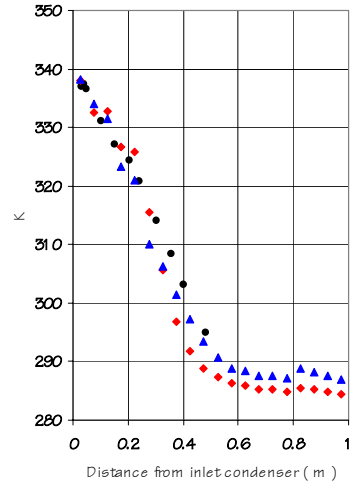
• Berkeley ♦ UPV-FIT ▲ GE

Berkeley run n°4  
Steam mass flow rate



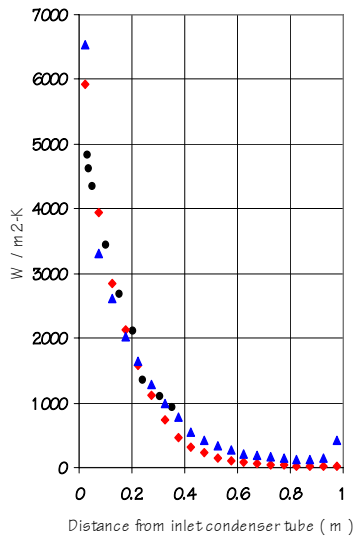
• Berkeley ♦ UPV-FIT ▲ GE

Berkeley run n°04  
Wall Temperature



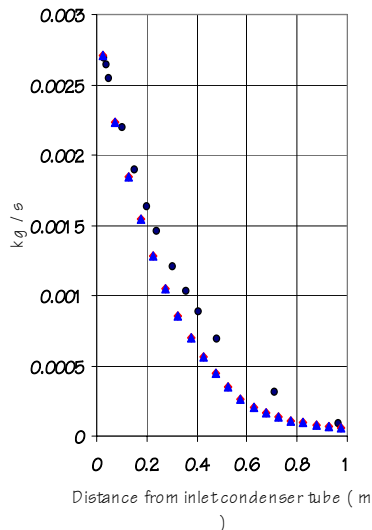
• Berkeley ♦ UPV-FIT ▲ GE

Berkeley run n°12  
Heattransfer coefficients



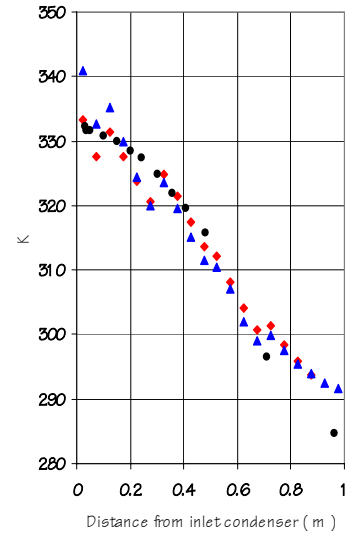
• Berkeley ♦ UPV-FIT ▲ GE

Berkeley run n°12  
Steam mass flow rate

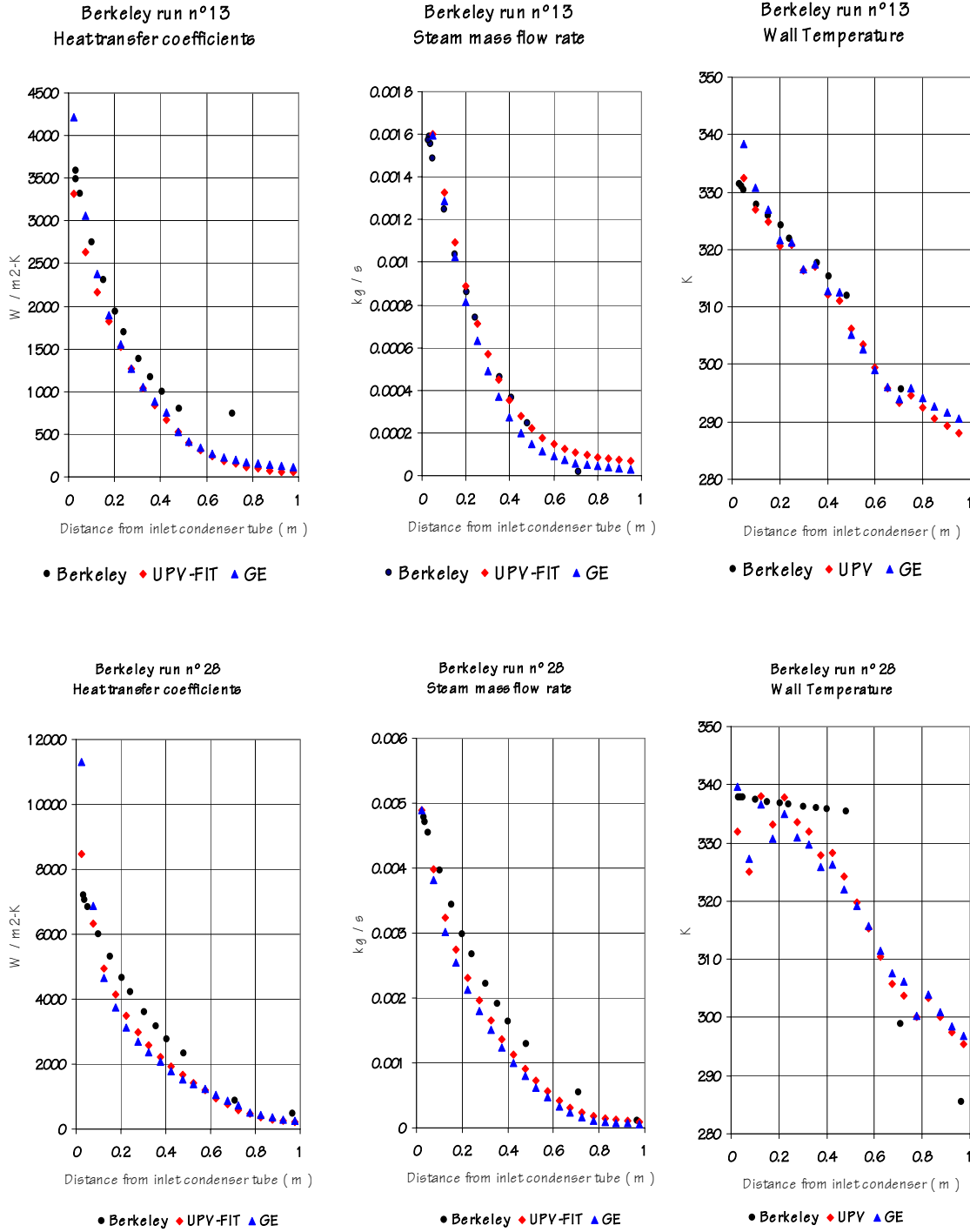


• Berkeley ♦ UPV-FIT ▲ GE

Berkeley run n°12  
Wall Temperature



♦ UPV-FIT ▲ GE • Berkeley



In general, UPV and Vierow models give good results. Some main features would be the followings:

1. Vierow and Schrock model shows worse results in the first centimeters, when condensation rates are the highest. We can see it, over all, in run number 12, 13, and 28, where the entrance velocity is very high.
2. Rather, in levels where vapor is scarce, so the latent heat is not the most important contribution to the interface heat transfer, UPV correlation tends to overpredict the gas boundary layer resistance. In these levels, the Vierow model gives little better results. We can

see it from levels lower than 20 centimeters from the top of the tube, over all, in run number 12, and 13.

3. In run number 28 it is shown what was mentioned above, that is, the difficulty to cope with the real cooler axial temperature profile. It is possible to see the rare profile that Vierow achieved in this run. The consequence of not achieving the temperatures adjustment caused a disagreement in the calculated values.

## **REFERENCES**

- [1] MUÑOZ-COBO, J.L., HERRANZ, L.E., SANCHO, J., TKACHENKO, I., VERDU G., "Turbulent vapor condensation with non-condensable gases inside vertical tubes", Int. Journal heat and Mass Transfer. Vol 39, No 15, pp 3249-320, 1996.
- [2] NUSSELT, W. "Die oberflächenkondensation des wasserdampfes" (the surface condensation of water vapor). Zeit. D. Ver. Deut. Ing. Frankfurt 60, 541–575. 1916.
- [3] MILLS, A.F., "Heat Transfer", Edited by Irwin , U.S.A., 1994.
- [4] JAMES, F., "MINUIT: Function minimization and error analysis", CERN Program Library. Writeup D506. Computing and Networks Division, 1994.
- [5] ROHSENOW, W. M. "Heat transfer and temperature distribution in laminar film condensation". Trans. ASME, 79, 1645-1648.
- [6] NO, H.C., PARK, H.S., "Non-iterative condensation modeling for steam condensation with non-condensable gas in a vertical tube", Int. Journal of Heat and Mass Transfer, Vol 45, pp 845-854, 2002.
- [7] DREW, T. B. "Heat Transmission", 3<sup>rd</sup> Edn, McGraw Hill, New York. 1954.
- [8] WALLIS, G. B. "One-dimensional two phase flow", McGraw Hill, New York. 1969.
- [9] SIDDIQUE, M., GOLAY, M.W., KAZIMI, M.S. Theoretical modeling of forced convection condensation of steam in a vertical tube in the presence of a noncondensable gas. Nucl. Tech. 106, 202–214.1994.
- [10] VIEROW, K.M., "Behaviour of steam-air systems condensing in cocurrent downflow", Master Thesis, University of California at Berkeley, 1990.