Condensation In Plain Horizontal Tubes: Recent Advances In Modelling of Heat Transfer to Pure Fluids and Mixtures

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Introduction

Collaborative research on intube condensation at the EPFL (J.R. Thome, J. El Hajal) has been established with the University of Padova (A. Cavallini, D. Del Col) and the University of Pretoria (J. Meyer, L. Liebenberg, F.J. Smit) to advance our understanding of condensation of pure fluids and mixtures inside plain horizontal tubes. To date, this has resulted in:

- New mixture condensation heat transfer data by Smit, Thome and Meyer [1];
- A new two-phase flow pattern map for condensation by El Hajal, Thome and Cavallini [2];
- Additional condensation heat transfer data and flow pattern observations by Liebenberg, Thome and Meyer [3];
- A new flow pattern, flow structure condensation heat transfer model by Thome, El Hajal and Cavallini [4];
- An updated version of the Silver-Bell-Ghaly condensation model for mixtures by Del Col, Cavallini and Thome [5].

The goal of this collaboration is to advance physically-based models to predict local heat transfer as a function of local flow patterns and attempt to achieve significant advances in accuracy and reliability. The idea is to try to capture the main phenomena and thermal mechanisms in simple, geometrical analytical models that are then adjusted by a minimum of empirical constants (unavoidable in turbulent flows), which not only are statistically accurate but are also shown to faithfully predict the trends in the data, something purely empirical methods often do not. Achieving these goals using a minimum number of new empirical constants to fit the data is taken as a qualitative proof that the underlying thermal model assumed captures the important features of the flow, as opposed to empirical approaches that sometimes require 20 or more such constants.

Many methods have been proposed in the last half-century for gravity-controlled...
a small decrease in vapor quality, which is not observed experimentally and hence incorrectly represents the experimental data.

In this light, rather than reformulating new methods based on old ideas, there is significant potential for progress by creating new heat transfer models that include more physical description of the actual flow structure. To do this, one must first predict (identify) the local two-phase flow pattern based on the local flow conditions, which requires a reliable two-phase flow pattern map. Secondly, some simplified but realistic geometrical formulation of the flow structure must be assumed for physically describing the flow. In order to quantify an annular flow structure, assumed to be an annular ring for instance, at the minimum a void fraction model is required to predict the relative cross-sectional areas occupied by the two-phases. The heat transfer or pressure drop model can also be formulated to represent the appropriate heat transfer and frictional mechanism(s) occurring around the perimeter of the tube, which may be locally wet or dry depending on the flow regime. Thus, a stratified-wavy regime or a more complex flow pattern is required to represent these two respective perimeters. The interaction between the two-phases may also be important, such as the effect of interfacial waves on condensation. This flow pattern type of approach is not new but in the past was primarily implemented with just one flow pattern in mind and has resulted in a patchwork of methods with conflicting transition criteria and step changes in predicted values from one flow pattern to another.

Nomenclature

\[ A = \text{cross-sectional area of flow channel, m}^2 \]
\[ A_{\text{c}} = \text{cross-sectional area occupied by liquid-phase, m}^2 \]
\[ A_{\text{v}} = \text{cross-sectional area occupied by vapor-phase, m}^2 \]
\[ c = \text{empirical constant} \]
\[ C = \text{empirical constant} \]
\[ c_{\text{h}} = \text{specific heat of the liquid, J/kg s} \]
\[ c_{\text{v}} = \text{specific heat of the vapor, J/kg s} \]
\[ d = \text{tube diameter, m} \]
\[ f_i = \text{interfacial roughness correction factor} \]
\[ F_{\text{c}} = \text{non-equilibrium mixture factor} \]
\[ G = \text{mass velocity of liquid plus vapor, kg/m}^2 s \]
\[ G_{\text{eq}} = \text{equivalent mass velocity, kg/m}^2 s \]
\[ G_{\text{strat}} = \text{transition mass velocity into fully stratified flow, kg/m}^2 s \]
\[ G_{\text{wavy}} = \text{transition mass velocity into stratified-wavy flow, kg/m}^2 s \]
\[ g = \text{gravitational acceleration, 9.81 m/s}^2 \]
\[ h_{\text{vap}} = \text{laminar heat of vaporization, J/kg} \]
\[ \Delta h_{\text{m}} = \text{enthalpy change of mixture, J/kg} \]
\[ m = \text{exponent} \]
\[ n = \text{exponent} \]
\[ Pr_{\text{L}} = \text{liquid Prandtl number} \]
\[ Pr_{\text{v}} = \text{vapor Prandtl number} \]
\[ q = \text{heat flux, W/m}^2 \]
\[ r = \text{radius of tube, m} \]
\[ R_{\text{f}} = \text{vapor-phase heat transfer resistance on roughened interface, m}^2 K/W \]
\[ R_{\text{v}} = \text{vapor-phase heat transfer resistance on falling film, m}^2 K/W \]
\[ Re_{\text{c}} = \text{equivalent Reynolds number} \]
\[ Re_{\text{L}} = \text{liquid Reynolds number} \]
\[ Re_{\text{v}} = \text{vapor Reynolds number} \]
\[ T_{\text{sat}} = \text{saturation temperature of vapor, K} \]
\[ T_{\text{w}} = \text{wall temperature of tube, K} \]
\[ \Delta T_{\text{w}} = \text{temperature glide of mixture, K} \]
\[ u_{\text{L}} = \text{mean velocity of liquid, m/s} \]
\[ u_{\text{v}} = \text{mean velocity of vapor, m/s} \]

Flow Pattern Map for Condensation

The two-phase flow pattern map for condensation proposed by El Hajal, Thome and Cavallini [2] is a slightly modified version of Kattan, Thome and Favrat [7] map for evaporation and adiabatic flows in small diameter horizontal tubes. Thome and El Hajal [8] have simplified implementation of that map by bringing a void fraction equation into the method to eliminate its iterative solution scheme. It is this last version that is the starting point for the condensation flow map. Presently, flow patterns are classified as follows: fully-stratified flow (S), stratified-wavy flow (SW), intermittent flow (I), annular flow (A), mist flow (MF) and bubbly flow (B). Intermittent flow refers to both the plug and slug flow regimes (it is essentially a stratified-wavy flow pattern with large amplitude waves that wash the top of the tube). Also, stratified-wavy flow is often referred to in the literature as simply wavy flow. For a detailed definition of the flow patterns used here, refer to those in Collier and Thome [9].

The flow pattern map for evaporation is shown in Figure 1 for R-134a in an 8.0 mm tube evaluated at 40°C. The transition boundary between annular flow (A) and stratified-wavy (SW) flow at high vapor quality represents the onset of dryout of the annular film and is thus a function of heat flux. For condensation, saturated vapor enters a condenser tube and forms either a thin liquid film around the perimeter of the tube (as an annular flow) or a liquid layer in the bottom of the tube and a gravity-controlled condensing film around the upper perimeter (as a stratified or stratified-wavy flow). Since dryout does not actually occur for condensation, the transition curve labeled G_{strat} is presumed to reach its minimum value and then continue horizontally to the vapor quality of 1.0, as shown in Figure 1. This means that a saturated vapor enters at x = 1.0 and goes directly into either the annular flow regime, the stratified-wavy flow regime or the stratified regime, depending on whether G is greater or less than G_{strat}. The other boundaries remain the same, assuming the gravity-controlled condensing film around the upper perimeter does not affect them. The bubbly flow...
regime occurs at mass velocities higher than those shown on the present map and is also beyond our current database. In a mist flow, it can be envisioned that the layer of condensate will be sheared from the wall and that a new condensate layer will immediately begin to grow again in its place.

To cover a broader range of reduced pressures from 0.02 to 0.80, the second change to the condensation map relative to the prior evaporation map is its method for calculating void fraction, where a simple logarithmic mean void fraction $\varepsilon$ is introduced and calculated as

$$\varepsilon = \frac{\varepsilon_h - \varepsilon_{ri}}{\ln(\varepsilon_h/\varepsilon_{ri})}$$  \hspace{1cm} (4)

In this expression, the value of $\varepsilon_{ri}$ from the Rouhani and Axelsson [10] drift flux expression:

$$\varepsilon_{ri} = \frac{x}{\rho_l} \left[ 1 + 0.12 \left( 1 - \frac{x}{\rho_l} \right) \right] \left[ \frac{x}{\rho_l} + \frac{1-x}{\rho_v} \right]^{-1}$$  \hspace{1cm} (5)

and the homogeneous void fraction $\varepsilon_h$ is calculated as:

$$\varepsilon_h = 1 + \left( 1 - \frac{x}{\rho_l} \right) \frac{\rho_v}{\rho_L}$$  \hspace{1cm} (6)

For the complete set of equations for the condensation flow pattern map, refer to El Hajal, Thome and Cavallini [2]. Figure 2 depicts a comparison of this map to flow pattern observations for R-407C obtained by Liebenberg, Thome and Meyer [3], in which all observations at the test conditions indicated were correctly identified.

**Heat Transfer Model for Pure Vapors and Azeotropic Mixtures**

The objective here was to develop a new flow pattern/flow structure based condensation heat transfer model analogous to that proposed by Kattan, Thome and Favrat [11] for evaporation inside horizontal tubes. The condensation heat transfer model proposed by Thome, El Hajal and Cavallini [4] uses the same flow pattern map as for evaporation but with the new modifications noted above. The new condensation model assumes that two types of heat transfer occur around the perimeter of the tube: convective condensation and film condensation. Convective condensation refers to the axial flow of the condensate along the channel due to the imposed pressure gradient while film condensation refers to the flow of condensate from the top of the tube towards the bottom due to gravity. Previous condensation models typically have subdivided the process into only two flow regimes: stratified flow and unstratified flow. Instead, here it is divided into the traditional flow regimes: annular flow, stratified-wavy flow, fully stratified flow, intermittent flow, mist flow and bubbly flow. Only the first five are addressed here as few data are available for bubbly flows while intermittent and mist flows are treated as an annular flow for simplicity’s sake. The three basic two-phase flow structures assumed are shown in Figure 3.

**Figure 3. Flow structures for annular, stratified-wavy and fully stratified flows (left to right in bottom three diagrams) and for fully stratified flow and its film flow equivalent (top two diagrams).**

The above two heat transfer mechanisms are applied to their respective heat transfer surface areas as shown in Figure 4. The convective condensation heat transfer coefficient $\alpha_c$ is applied to the perimeter wetted by the axial flow of liquid film, which refers to the entire perimeter in annular, intermittent and mist flows but only part of the perimeter in stratified-wavy and fully stratified flows. The axial film flow is assumed to be turbulent. The film condensation heat transfer coefficient $\alpha_f$ is applied to the perimeter that would otherwise be dry in an adiabatic two-phase flow and hence is the...
upper perimeter of the tube for stratified-wavy and fully stratified flows. \( \alpha_L \) is obtained by applying the Nusselt [12] falling film theory to the inside of the horizontal tube, which assumes the falling film is laminar and falls downward without any axial velocity component. Heat transfer coefficients for stratified types of flow are known experimentally to be a function of the wall temperature difference and this effect is included through the Nusselt falling film heat transfer equation in the present model.

The general expression for the local condensing heat transfer coefficient \( \alpha_{\text{op}} \) is:

\[
\alpha_{\text{op}} = \frac{\alpha_f r \theta + (2\pi - \theta) r \alpha_L}{2\pi}
\]  

(7)

In this expression, \( r \) is the internal radius of the tube and \( \theta \) is the falling film angle around the top perimeter of the tube, which occurs on the upper perimeter that would otherwise be dry in an adiabatic falling film angle around the top perimeter of the tube, which occurs for stratified-wavy or fully stratified flow, \( \theta = 0, \alpha_L \) is equal to \( \alpha_L \). The falling film angle \( \theta_{\text{strat}} \) is calculated from the following implicit geometric equation:

\[
A_L = \frac{d^2}{8} \left[ (2\pi - \theta_{\text{strat}}) - \sin(2\pi - \theta_{\text{strat}}) \right]
\]  

(8)

where the cross-sectional area occupied by the liquid phase \( A_i \) is

\[
A_i = (1 - \varepsilon)A
\]  

(9)

and the cross-sectional area occupied by the vapor is

\[
A_V = \varepsilon A = 1 - A_L
\]  

(10)

\( A \) is the total cross-sectional area of the tube and \( \varepsilon \) is the local vapor void fraction, which is determined using the logarithmic mean void fraction (LMVe) using the Rouhani and Axelsson drift flux model and the homogeneous model in order to cover the range from low to high reduced pressures.

For annular, intermittent and mist flows, \( \theta = 0 \). For fully stratified flow, \( \theta > 0 \). The stratified angle \( \theta_{\text{strat}} \) is obtained by assuming a quadratic interpolation between its maximum value of \( \theta_{\text{strat}} \) at \( G_{\text{strat}} \) and its minimum value of 0 at \( G_{\text{wavy}} \):

\[
\theta = \theta_{\text{strat}} - \left[ \frac{G_{\text{wavy}} - G}{G_{\text{strat}} - G_{\text{strat}}} \right]^{0.5}
\]  

(11)

The values of \( G_{\text{strat}} \) and \( G_{\text{wavy}} \) at the vapor quality in question are determined from their respective transition equations in the flow pattern map. The convective condensation heat transfer coefficient \( \alpha_c \) is obtained from the following turbulent film equation:

\[
\alpha_c = c \left. R_e \right|_L Pr_L \frac{\lambda_i}{\delta} f_i
\]  

(12)

where the liquid film Reynolds number \( R_e \) is based on the mean liquid velocity of the liquid in \( A_L \), as:

\[
R_e = \frac{4G(1 - \varepsilon)\delta}{(1 - \varepsilon) \mu_L}
\]  

(13)

and \( \mu_L \) is the liquid Prandtl number defined as:

\[
Pr_L = \frac{c \left. \rho_i \mu_i \right| L}{\lambda_i}
\]  

(14)

In these expressions \( c, n \) and \( m \) are empirical constants determined from the heat transfer database and \( \delta \) is the thickness of the liquid film. The best value of the exponent \( m \) on \( Pr_L \) was determined to be \( m = 0.5 \) while the best values of \( c \) and \( n \) for Eq. (12) were found statistically to be \( c = 0.003 \) and \( n = 0.74 \). The liquid film thickness \( \delta \) is obtained from solving the following geometrical expression:

\[
A_L = \frac{(2\pi - \theta) \left( \frac{d}{2} \right)^2 - (d - 2\delta)^2}{8}
\]  

(15)

where \( d \) is the internal diameter of the tube. When the liquid occupies more than one-half of the cross-section of the tube in a stratified-wavy or fully stratified flow at low vapor quality, this expression will yield a value of \( \delta > d/2 \), which is not geometrically realistic. Hence, whenever \( \delta > d/2 \), \( \delta \) is set equal to \( d/2 \).

Analysis of the data demonstrated that an additional factor influenced convective condensation. After looking at various possibilities, the interfacial surface roughness was identified as the most influential based on the following reasoning. First of all, the shear of the high speed vapor is transmitted to the liquid film across the interface and hence increases the magnitude and number of the waves generated at the interface, which in turn increases the available surface area for condensation, tending to increase heat transfer. Secondly, the interfacial waves are non-sinusoidal and thus tend to reduce the mean thickness of the film, again increasing heat transfer. An interfacial roughness correction factor \( f_i \) was introduced to act on \( \alpha_L \) in Eq. (12) as follows:

\[
f_i = 1 + \left( \frac{u_V}{u_L} \right)^{1/2} \left( \frac{\rho_L - \rho_V}{\rho_V} \right) \frac{\delta^2}{\sigma} G^{1/4}
\]  

(16)

where \( u_V \) and \( u_L \) are the mean velocities of the phases in their respective cross-sectional areas \( A_V \) and \( A_i \):

\[
u_L = \frac{G(1 - \varepsilon)}{\rho_i \varepsilon}
\]  

(17)

\[
u_V = \frac{Gx}{\rho_V (1 - \varepsilon)}
\]  

(18)

The interfacial roughness correction factor \( f_i \) tends towards a value of 1.0 as the film becomes very thin (roughness must be proportional to film thickness) but \( f_i \) tends to increase as the slip ratio \( u_V/u_L \) increases. Finally, \( f_i \) tends to decrease as \( \sigma \) increases, since surface tension acts to smooth out the waves. For fully stratified flow, interfacial waves are damped out and hence the above expression becomes

\[
f_i = 1 + \left( \frac{u_V}{u_L} \right)^{1/2} \left( \frac{\rho_L - \rho_V}{\rho_V} \right) \frac{\delta^2}{\sigma} \left( \frac{G}{G_{\text{strat}}} \right)^{1/4}
\]  

(19)

when \( G < G_{\text{strat}} \), which produces a smooth variation in \( \alpha_{\text{op}} \) across this flow pattern transition boundary just like for all the other transition boundaries and the ratio of \( G/G_{\text{strat}} \) acts to damp out the effect of interfacial roughness in stratified flow.
The film condensation heat transfer coefficient $\alpha_f$ is obtained from the theory of Nusselt [12] for laminar flow of a falling film on the internal perimeter of the tube, where $\alpha_f$ is the mean coefficient for this perimeter. Rather than integrating from the top of the tube to the stratified liquid layer at $\theta/2$ to obtain $\alpha_f$, which would be more theoretically satisfying, it was found sufficient to simply use the mean value for condensation around the perimeter from top to bottom with its analytical value of 0.728, and thus avoid a numerical integration to facilitate practical use of this method in designing condensers. Hence, $\alpha_f$ is:

$$\alpha_f = 0.728 \left[ \frac{\rho_L (\rho_L - \rho_v) g b_{LV} \lambda_L^3}{\mu_L d (T_{sat} - T_w)} \right]^{1/3}$$  \hspace{1cm} (20)

Since heat exchanger design codes are typically implemented assuming a heat flux in each incremental zone along the exchanger, it is more convenient to convert this expression to heat flux using Newton’s law of cooling, such that the heat flux version of the Nusselt equation where the local heat flux is $q$, is given by the expression:

$$\alpha_f = 0.655 \left[ \frac{\rho_L (\rho_L - \rho_v) g b_{LV} \lambda_L^3}{\mu_L d q} \right]^{1/3}$$  \hspace{1cm} (21)

where the leading constant 0.655 comes from $0.728^{1/3}$. The difference in the accuracy of the predictions whether using the first or second of these expressions for $\alpha_f$ is negligible.

To completely avoid any iterative calculations, the following expression of Biber [13] based on void fraction is used to very accurately (error \(\rho = 0.00005\) radians for $2\pi \geq \theta_{\text{sat}} > 0$) evaluate $\theta_{\text{sat}}$ instead of Eq. (8):

$$\theta_{\text{sat}} = 2\pi - 2 \left[ \pi (1 - \varepsilon) + \frac{3\pi}{2} \right]^{1/3}$$  \hspace{1cm} (22)

$$\quad \frac{1 - 2(1 - \varepsilon) + \left[ (1 - \varepsilon)^{1/3} - \varepsilon^{1/3} \right]}{\left( 1 - 2(1 - \varepsilon) + \left[ (1 - \varepsilon)^{1/3} - \varepsilon^{1/3} \right] \right)^{1/3}}$$

The above heat transfer prediction method cannot be evaluated at $\varepsilon = 1.0$ because of division by zero. Furthermore, experimental condensation heat transfer test data are reported with a measured error in vapor quality of at least $\pm 0.01$ and hence it does not make sense that test data can be evaluated for $x > 0.99$. Thus, the above condensation prediction method is applicable when $0.99 \geq x$; when $x > 0.99$, then $x$ should be reset to 0.99. Also, the lower limit of applicability is for vapor qualities $x \geq 0.01$. Our range of test data was from 0.97 > $x$ > 0.03. This method provides for a smooth variation in $\alpha_f$ across all the flow pattern transition boundaries without any jump in the value of $\alpha_f$.

The condensation heat transfer model is implemented as follows:

1. Determine the local void vapor fraction using the LMe method given by Eq. (6);
2. Determine the local flow pattern using the flow pattern map;
3. Identify the type of flow pattern (annular, intermittent, mist, stratified-wavy or stratified);
4. If the flow is annular or intermittent or mist, then $\theta = 0$ and $\alpha_f$ is determined with Eq. (12) and hence $\alpha_{fp} = \alpha_f$ in Eq. (7) where $\delta$ is obtained with Eq. (15) and $f_i$ is determined with Eq. (16).
5. If the flow is stratified-wavy, then $\theta_{\text{sat}}$ and $\theta$ are calculated using Eq. (8) or (22) and Eq. (11), then $\alpha_f$ and $\alpha_e$ are calculated using Eqs. (12) and (20) or (21), and finally $\alpha_{fp}$ is determined using Eq. (7) where again $\delta$ is obtained with Eq. (15) and $f_i$ is determined with Eq. (19).
6. If the flow is fully stratified, then $\theta_{\text{sat}}$ is calculated using Eq. (8) or (22) and $\theta_{\text{sat}}$ is set equal to $\theta$, then $\alpha_f$ and $\alpha_e$ are calculated using Eqs. (12) and (20) or (21), and finally $\alpha_{fp}$ is determined using Eq. (7) where $\delta$ is obtained with Eq. (15) and $f_i$ is determined with Eq. (19).

Figure 5 depicts a comparison of the new condensation heat transfer model to all the refrigerant database, representing eleven fluids with a total of 1850 data points taken by nine different research laboratories. Based on all the data points from these numerous different test facilities, 85% are predicted within $\pm 20\%$. Figure 6 depicts the distribution of errors by flow pattern, showing nearly uniform accuracy except for the fully stratified flows (where heat transfer is dominated by Nusselt film condensation and whose perimeter-averaged heat transfer coefficient may not be well represented by the measurement techniques).
diameter tube assuming a heat flux of 40 kW/m². The flow pattern map for three refrigerants and heat transfer coefficients (for R-410A) are shown in Figure 7. At the lowest flow rate, 30 kg/m²s, the flow is in the stratified regime from inlet to outlet and the heat transfer coefficient falls off slowly with decreasing vapor quality. At 200 kg/m²s, the flow enters in annular flow and then passes through intermittent and stratified-wavy flow. At 500 kg/m²s, the flow enters in the annular flow regime and converts to intermittent flow at about x = 0.55 and leaves in this same regime. The sharp decline in αtp at high vapor qualities results from the rapid growth of the annular film thickness. At 800 kg/m²s, the flow enters in the mist flow regime, goes into the annular flow regime and then leaves in the intermittent regime. As can be seen, the new heat transfer model predicts the variation in the local heat transfer coefficients across flow pattern transition boundaries without any discontinuity in the value of αtp.

Heat Transfer Model for Zeotropic Mixtures

The mixture effect on condensation heat transfer is illustrated by Figure 8 showing local experimental heat transfer coefficients for three refrigerant mixtures of R-125/R-236ea and its pure components obtained by Cavallini et al. [14]. Applying the new version of the Silver-Bell-Ghaly series resistance approach as follows:

$$\alpha_{Vt} = \frac{\alpha_{Vt}^f \theta + (2\pi - \theta)\alpha_{Vt}^m}{2\pi}$$  \hspace{1cm} (23)

where θ is the falling film angle around the top perimeter of the tube as already discussed. The convective condensation heat transfer coefficient of the mixture is obtained from the Silver-Bell-Ghaly series resistance approach as follows:

$$\alpha_{Vt}^m = \left[ \frac{1}{\alpha_c} + R_v \right]^{-1}$$  \hspace{1cm} (24)

where αc is computed as per the pure fluid model using mixture physical properties. The Silver-Bell-Ghaly vapor-phase heat transfer resistance Rv for cooling of the vapor to the dew point temperature is calculated as follows:

$$R_v = \frac{x c_p v \Delta T_{d}}{\Delta h_m \alpha_{Vt}^f}$$  \hspace{1cm} (25)

The resistance depends on vapor phase heat transfer coefficient referred to the vapor-liquid interface αVt. Silver-Bell-Ghaly assumed the value of αVt to be that of simple vapor flow in a plain tube without a liquid film while in fact there is a liquid film with an interfacial roughness affecting this heat transfer. Based on this reasoning, the same correction factor acting on αc should be applied to the vapor heat transfer coefficient that appears in the Silver-Bell-Ghaly resistance of the axial flow. Therefore the vapor coefficient can be written as:

$$\alpha_{Vt} = \alpha_V f_i$$  \hspace{1cm} (26)

The interfacial roughness factor f_i is computed from the equations above. The vapor heat transfer coefficient αV can be computed with the Dittus and Boelter [17] correlation:

$$\alpha_V = \frac{\lambda V}{d} \cdot 0.023 \cdot Re^{0.8} \cdot Pr^{0.33}$$  \hspace{1cm} (27)
It is adapted to the present situation by defining the Reynolds number of the vapor phase $Re_v$ based on the mean vapor velocity of the cross-sectional area occupied by the vapor $A_f$:

$$Re_v = \frac{Gd_v}{\mu_v}$$  \hspace{1cm} (28)

$Pr_v$ is the vapor Prandtl number. The Silver-Bell-Ghaly procedure is applied to the film condensation component in the same way as for the convective term but without interfacial roughness on the laminar falling film, i.e. $f_i = 1.0$. One of the main assumptions of the Silver-Bell-Ghaly approach is that equilibrium exists between the liquid and vapor phases. In reality, the condensation process inside a tube departs from this equilibrium when the liquid condensate forms a stratified layer at the bottom of the tube and the film condensation component must be further corrected with a mixture factor $F_m$ acting to the resistance of the falling film:

$$\alpha_{fm} = F_m \left[ \frac{1}{\alpha_f} + R_f \right]^{-1}$$  \hspace{1cm} (29)

The heat transfer coefficient $\alpha_f$ is calculated from the pure vapor model using the properties of the mixture while the resistance $R_f$ can be determined in the form:

$$R_f = \frac{x \cdot c_p V}{\mu_v} \frac{\Delta T_{gl}}{1} \frac{1}{\Delta h_m} \frac{1}{\alpha_f}$$  \hspace{1cm} (30)

where no interfacial roughness factor applies to the vapor phase heat transfer coefficient because the falling film is supposed to be smooth. The new non-equilibrium mixture factor $F_m$ accounts for non-equilibrium effects in stratified flow regimes and has been correlated as a function of vapor quality, mass velocity, temperature glide and saturation to wall temperature difference as follows:

$$F_m = \exp \left[ -0.25 \cdot (1 - x) \left( \frac{G_{wavy}}{G} \right)^{0.5} \frac{\Delta T_{gl}}{T_{sat} - T_{wsat}} \right]$$  \hspace{1cm} (31)

The values of $F_m$ range between 0 and 1. It decreases when stratification is enhanced, that is at low mass velocity and low vapor quality. The mass transfer resistance depends on the temperature glide and that is why $F_m$ decreases when increasing $\Delta T_{gl}$. Finally, the effect of the saturation to wall temperature difference is the opposite of that in the Nusselt theory. Given the saturation temperature, when the wall temperature decreases, the vapor of the more volatile component accumulates near the interface and acts as an incondensable. When this temperature difference increases, the more volatile component begins to condense and this decreases the thermal resistance of the diffusion layer. This is the reason why the factor $F_m$ should increase with the saturation to wall temperature difference, leading to an increase in the heat transfer coefficient.

Figure 9 depicts the comparison of this method to test data from four publications. The database covers temperature glides up to 22 K and mass velocities from 57-755 kg/m’s. The method predicts 98% of the refrigerant heat transfer coefficients measured by Cavallini et al. [14, 18] to within ±20% and predicts 85% of the halogenated plus hydrocarbon refrigerant heat transfer coefficients measured by Lee [19] and Kim, Chang and Ro [20] to within ±20%.

Summary

Recent improvements in general thermal design methods for condensation inside plain, horizontal tubes made at the Laboratory of Heat and Mass Transfer at the EPFL in collaboration with the University of Padova and the University of Pretoria have been summarized. First, a new condensation two-phase flow pattern map has been proposed and partially verified by new flow pattern observations. Secondly, a new condensation heat transfer model has been developed including both flow pattern effects and interfacial roughness effects, which accurately predicts and emulates a large, diversified database. Finally, the well-known Silver-Bell-Ghaly model for predicting heat transfer in the condensation of miscible vapor mixtures has been adapted to this new heat transfer model, including the effects of interfacial flow structure and roughness on vapor phase heat transfer and non-equilibrium condensation, to more accurately predict condensation of mixtures with temperature glides up to 22 K.

References


