VI. Condensation Inside of Horizontal Tubes
Chapter 8 (in Databook III)

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Topics to Be Covered in Chapter:

• Flow patterns during intube condensation.
• Introductory comments about intube condensation.
• Description of older condensation design methods.
• More recent condensation methods.
• Condensation of zeotropic mixtures.
Condensation in horizontal tubes may involve partial or total condensation of the vapor.

Depending on the application, the inlet vapor may be superheated, equal to 1.0 or below 1.0.

Hence, the condensation process path may first begin with a dry wall desuperheating zone, followed by a wet wall desuperheating zone, then a saturated condensing zone and finally a liquid subcooling zone.

The condensing heat transfer coefficient is a strong function of local vapor quality, decreasing as the vapor quality decreases.

The condensing heat transfer coefficient is also a strong function of mass velocity, increasing as the mass velocity increases.

Opposed to external condensation, intube condensation is independent of the wall temperature difference ($T_{sat}-T_w$) for most operating conditions, i.e. except at low mass flow rates in stratified types of flows.
There is a great similarity between condensing flow regimes and those for adiabatic two-phase flows. Here, however, condensate forms all around the tube perimeter even in stratified flows. As illustrated in Figure 8.2, the fully stratified flow regime with all the liquid normally in the lower portion of the tube for adiabatic flow will have a thin layer of condensate around the upper perimeter.

As shown in Figure 8.3, at low flow rates the flow is stratified and a film of condensate forms by film-wise condensation and drains from the top of the tube towards the bottom under the force of gravity. The film flow is laminar and primarily downwards when the vapor core velocity is low. If the vapor shear is sufficient and the onset to turbulence has been surpassed, then a turbulent film is formed whose predominant flow direction is axial.

For low vapor shear conditions, the condensation process on the inside perimeter around the top and sides of the tube is very similar to that on the outside of a horizontal tube. Thus, Nusselt falling film analysis may be applied to the upper zone of the tube, which has been done first by Chaddock (1957) and then by Chato (1962).
Chato (1962). The cross-sectional area of the stratified liquid layer at the bottom of the tube can be established from the local void fraction $\varepsilon$. Then, the stratified liquid angle $\theta_{\text{strat}}$ can be determined from geometry. The local heat transfer coefficient at this vapor quality $x$ is obtained by proration of the respective heat transfer coefficients with respect to the fraction of the perimeter they occupy as

$$\alpha(x) = \frac{\theta_{\text{strat}}}{\pi} \alpha_f + \frac{\pi - \theta_{\text{strat}}}{\pi} \alpha_{\text{strat}}$$  \[8.1.1\]

$\theta_{\text{strat}}$ is the angle from the top of the tube to the stratified layer and is thus equal to $\pi$ when there is no stratified layer present. $\theta_{\text{strat}}$ is expressed in radians. $\alpha_f$ is the mean heat transfer coefficient for the film obtained by integration of $[7.5.11]$ from 0 to $(\pi - \theta_{\text{strat}})/2$. The heat transfer coefficient for the stratified flow in the bottom of the tube is $\alpha_{\text{strat}}$.

Assuming that $\alpha_{\text{strat}}$ is negligible compared to $\alpha_f$, the second term can be neglected while $\alpha_f$ is determined as:

$$\alpha_f = \frac{\rho_f (\rho_v - \rho_f) \beta \kappa_L h_L}{\mu_f d (T_v - T_f)}^{0.75}$$  \[8.1.2\]

The value of $\Omega$ is a geometric function of $\theta_{\text{strat}}$ where $\Omega = \beta \theta_{\text{strat}}/\pi$ and $k_L$ is the liquid thermal conductivity. Jaster and Kosky (1976) have shown that the value of $\Omega$ is related to the vapor void fraction $\varepsilon$ as $\Omega = 0.728 \varepsilon$. They used the Zivi (1964) void fraction equation, which is a function of vapor quality $x$ and the vapor and liquid densities:

$$\varepsilon = \frac{1}{1 + \left[ (1-x)/x \right] \rho_v / \rho_f}$$  \[8.1.3\]
Akers, Deans and Crosser (1959) proposed a modified version of the Dittus-Boelter single-phase turbulent tube flow correlation, developed with a database for several refrigerants and organic fluids. Their local condensing coefficient is

\[
\frac{\alpha(x) d_l}{k_l} = C \text{Re}_{L}^{n} \text{Pr}_{L}^{1/3} \tag{8.1.4}
\]

The equivalent Reynolds number for two-phase flow \( \text{Re}_e \) is determined from an equivalent mass velocity, which in turn is obtained by applying a multiplying factor to the total mass velocity:

\[
\bar{m}_e = \bar{m} \left( 1 - x \right) + x \left( \frac{\rho_1}{\rho_0} \right)^{1/2} \tag{8.1.5}
\]

The total mass flow rate of liquid plus vapor is used for the total mass velocity. The empirical parameters \( C \) and \( n \) to use in [8.1.4] are:

- \( C = 0.0265 \) and \( n = 0.8 \) for \( \text{Re}_e > 50,000 \)
- \( C = 5.03 \) and \( n = 1/3 \) for \( \text{Re}_e < 50,000 \)

With the Dittus-Boelter correlation as a starting point, Shah (1979) proposed an alternative multiplier (see brackets) acting on the liquid Reynolds number as

\[
\frac{\alpha(x) d_l}{k_l} = 0.023 \text{Re}_{L}^{0.8} \text{Pr}_{L}^{0.4} \left[ (1-x)^{0.8} + \frac{3.8(1-x)^{0.76}}{p_r^{0.38}} \right] \tag{8.1.6}
\]

He used the reduced pressure \( p_r = p_{sat}/p_{crit} \) where \( p_{sat} \) is the saturation pressure and \( p_{crit} \) is the critical pressure of the fluid rather than the density ratio and a database for condensation of steam, refrigerants and organic fluids. \( \text{Re}_L \) is the tubular liquid Reynolds number determined with the total mass flow rate of liquid plus vapor.

Thome (1994, 1998) recommended using the Shah correlation when mass velocities are greater than 200 kg/m²s and that of Akers, Deans and Crosser (1959) when they are lower, based on comparisons to local test data in the literature for R-134a, R-22 and others.
Dobson and Chato (1998) have proposed a vast improvement of the Chato (1962) method that includes both a stratified-wavy flow method with film condensation from the top towards the bottom of the tube and an annular flow correlation. Their annular flow correlation is

\[
\begin{align*}
\text{Nu}(x) &= 0.023 \text{Re}_{Ls}^{0.8} \text{Pr}_L^{0.4} \left[ 1 + \frac{2.22 \times 10^{-3}}{X_{tt}^{0.07}} \right] \\
\text{where the local Nusselt number Nu}(x) is} & \quad \text{Nu}(x) = \frac{\alpha(x)d}{k_L} \quad \text{8.1.8}
\end{align*}
\]

Their superficial liquid Reynolds number \( \text{Re}_{Ls} \) is

\[
\text{Re}_{Ls} = \frac{\overline{md}_L(1-x)}{\mu_L} \quad \text{8.1.9}
\]

The Martinelli parameter for turbulent flow in both phases, \( X_{tt} \), is

\[
X_{tt} = \left( \frac{1-x}{x} \right)^{0.5} \left( \frac{\rho_G}{\rho_L} \right)^{0.5} \left( \frac{\mu_L}{\mu_G} \right)^{0.1} \quad \text{8.1.10}
\]

To implement the method for stratified-wavy flow, first the void fraction \( \varepsilon \) is calculated using the Zivi void fraction equation. Assuming all the liquid is stratified in the bottom of the tube (neglecting condensate formed on the walls), the angle from the top of the tube to the stratified liquid layer in the bottom \( \theta_{strat} \) is then determined

\[
1 - \frac{\theta_{strat}}{\pi} \approx \frac{\arccos(2\varepsilon - 1)}{\pi} \quad \text{8.1.11}
\]

The stratified-wavy heat transfer coefficient is obtained by a proration between the film condensation coefficient on the top perimeter of the tube (left term) and the forced convective heat transfer coefficient on the stratified perimeter (right term) as

\[
\text{Nu}(x) = \frac{0.23 \text{Re}_{G0}^{0.12} \left( \frac{G\alpha g L}{J\alpha L} \right)^{0.25} \left( 1 - \frac{\theta_{strat}}{\pi} \right) \text{Nu}_{strat}}{1 + 1.11X_{tt}^{0.5}}}
\]

Forced convection condensation in the stratified liquid is correlated as

\[
\text{Nu}_{strat} = 0.0195 \text{Re}_L^{0.8} \text{Pr}_L^{0.4} \left( 1.376 + \frac{C_L}{X_{tt}^{0.5}} \right)^{1/2} \quad \text{8.1.13}
\]
The value of 1.376 makes this expression match the Dittus-Boelter correlation when \( x = 0 \). The liquid Galileo number \( Ga_L \) for the tube is
\[
Ga_L = \frac{g \rho_L (\rho_L - \rho_\infty) d^3}{\mu_L} \quad [8.1.14]
\]
while the vapor only Reynolds number \( Re_{\infty} \) is
\[
Re_{\infty} = \frac{\rho_\infty d \nu_0}{\mu_0} \quad [8.1.15]
\]
The liquid Jakob number \( Ja_L \) defined as
\[
Ja_L = \frac{c_{\nu_L} (T_{sat} - T_\infty)}{h_{LG}} \quad [8.1.16]
\]
and the liquid Froude number \( Fr_L \) is
\[
Fr_L = \frac{\dot{m}_L^2}{\rho_L g d} \quad [8.1.17]
\]
The empirical constants \( c_1 \) and \( c_2 \) are obtained as a function of \( Fr_L \) as follows:

For \( 0 < Fr_L \leq 0.7 \):
\[
\begin{align*}
c_1 &= 4.172 + 5.48 \cdot Fr_L - 1.564 \cdot Fr_L^2 \quad [8.1.18a] \\
c_2 &= 1.773 - 0.169 \cdot Fr_L \quad [8.1.18b]
\end{align*}
\]
For \( Fr_L > 0.7 \):
\[
\begin{align*}
c_1 &= 7.242 \quad [8.1.19a] \\
c_2 &= 1.655 \quad [8.1.19b]
\end{align*}
\]
The Soliman (1982) transition criterion for predicting the transition from annular flow to stratified-wavy flow was used to distinguish which heat transfer regime to apply. His method is based on a Froude transition number $Fr_{so}$ given as

$$Fr_{so} = 0.025 \frac{Re_{Ls}^{1.09}}{Ga^1} \left( 1 + \frac{1.09 X_{Ls}^{1.09}}{X_g} \right)^{1.5} \frac{1}{Ga^{0.15}}$$ \[8.1.20\]

for $Re_{Ls} \leq 1250$ and for $Re_{Ls} > 1250$ it is

$$Fr_{so} = 1.26 \frac{Re_{Ls}^{1.04}}{Ga^1} \left( 1 + \frac{1.09 X_{Ls}^{0.09}}{X_g} \right)^{1.5} \frac{1}{Ga^{0.15}}$$ \[8.1.21\]

While Soliman set the transition from annular flow to wavy flow at $Fr_{so} = 7$, Dobson and Chato noted that a transition value of 20 fit their heat transfer data better and this is the value they used.

The Dobson-Chato method is implemented as follows:

- For mass velocities greater than 500 kg/m$^2$s (367,896 lb/h ft$^2$), the annular flow correlation is always utilized;
- For mass velocities less than 500 kg/m$^2$s (367,896 lb/h ft$^2$), the annular flow correlation is used when $Fr_{so} > 20$;
- For mass velocities less than 500 kg/m$^2$s (367,896 lb/h ft$^2$) and for $Fr_{so} < 20$, the stratified-wavy correlation is used.

This method does not have a smooth transition in the heat transfer coefficient from annular flow to stratified-wavy flow; instead, it gives a significant step change in value that is not observed experimentally.

Other than this inconvenience, their method appears to be the most accurate design method currently available according to Cavallini et al. (1995), who compared it to independent test data. The discontinuity in the heat transfer coefficient may be resolved for now by implementing a simple linear proration based on $Fr_{so}$ between the corresponding calculated heat transfer coefficients at say $Fr_{so} = 7$ with the stratified-wavy correlation and at $Fr_{so} = 20$ with the annular flow correlation.
Tang (1997) has also proposed a simple correlation that is an extension of the Shah (1979) approach and covers reduced pressures from 0.2 to 0.53 and mass velocities from 300 to 810 kg/m\(^2\)s. His correlation is applicable to annular flow only and is:

\[
\alpha(x) \frac{d_i}{k_L} = 0.023 \text{Re}_i^{0.8} \text{Pr}_L^{0.64} \left[ 1 + 4.86 \left( \frac{-x \ln p_r}{1-x} \right)^{0.85} \right] \tag{8.1.22}
\]

Cavallini et al. (2001): test data for intube condensation for an 8 mm (0.315 in.) tube for a wide range of pressures (0.246 to 3.15 MPa, 35.7 to 456.8 psia) have been measured for five fluids: R-134a, R-125, R-32, R-410a and R-236ea. They covered mass velocities from 100 to 750 kg/m\(^2\)s (73,579 to 551,844 lb/h ft\(^2\)) and vapor qualities from 0.15 to 0.85 in quasi-local type of tests.

8.3 Condensation of Condensable Mixtures in Horizontal Tubes

The Silver-Bell-Ghaly method [Silver (1947) and Bell and Ghaly (1973)] is successfully used to predict condensation of miscible mixtures where all components are condensable but no non-condensable gases are present. When condensing a mixture, the vapor phase must be cooled as the dew point temperature of the mixture falls along the tube, in addition to removing the latent heat. Hence, the process is controlled by condensation and by single-phase cooling of the vapor. This approach assumes two things with respect to cooling of the vapor:

- Mass transfer has no effect on the single-phase heat transfer process in the vapor;
- The vapor occupies the entire tube cross section in determining the vapor phase heat transfer coefficient.

The error in ignoring the first assumption becomes significant for mixtures with large condensing temperature ranges, so their method is reliable for mixtures with small to medium condensing ranges (say smaller than 30 K). The second assumption is conservative since interfacial waves in annular flows augment the vapor phase heat transfer coefficient.

**Note:** New results suggest that 10-15K is upper limit of this method.
The effective condensing heat transfer coefficient $\alpha_{\text{eff}}$ for a mixture is

$$\frac{1}{\alpha_{\text{eff}}} = \frac{1}{\alpha(x)} + \frac{Z_G}{\alpha_G} \tag{8.3.1}$$

To implement this expression, the condensation heat transfer coefficient $\alpha(x)$ is obtained with an intube correlation for pure fluids in the previous section but inputting the local physical properties of the mixture. The single-phase heat transfer coefficient of the vapor $\alpha_G$ is calculated with the Dittus-Boelter turbulent flow correlation using the vapor fraction of the flow in calculating the vapor Reynolds number. The parameter $Z_G$ is the ratio of the sensible cooling of the vapor to the total cooling rate:

$$Z_G = x c_p \frac{dT_{\text{dew}}}{dh} \tag{8.3.2}$$

where $x$ is the local vapor quality, $c_p$ is the specific heat of the vapor and $dT_{\text{dew}}/dh$ is the slope of the dew point temperature curve with respect to the enthalpy of the mixture as it condenses, i.e. the slope of the condensation curve. This method has been applied to hydrocarbon mixtures and more recently to binary and ternary zeotropic refrigerant blends by Cavallini et al. (1995) and to binary refrigerant mixtures by Smit, Thome and Meyer (2001).

**New, more general approach that includes the additional effects of interfacial waves and non-equilibrium effects has been developed by Del Col, Thome and Cavallini (currently in review at JHT)**

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**Example Calculation:** Propane is condensing inside a horizontal, plain tube whose internal diameter is 15 mm. The refrigerant enters at its saturation temperature of 2°C (5.07 bar) as a saturated vapor and leaves as a saturated liquid. The flow rate of vapor entering is 0.03534 kg/s and the tube wall has a mean internal temperature of -10°C. Determine the following values: the local condensing heat transfer coefficient using the Akers, Shah, and Dobson-Chato methods at a vapor quality of 0.5. Next, assuming a hydrocarbon mixture with the same physical properties as propane but with a linear temperature glide during condensation from 2°C to -3°C, determine the local condensing heat transfer coefficient using the Dobson-Chato method at a vapor quality of 0.5. The physical properties of propane at 2°C are:

- $\rho_L = 528 \text{ kg/m}^3$;
- $\rho_G = 11.0 \text{ kg/m}^3$;
- $\mu_L = 0.0001345 \text{ Ns/m}^2$;
- $\mu_G = 0.0000075 \text{ Ns/m}^2$;
- $h_{LG} = 373100 \text{ J/kg}$;
- $k_L = 0.108 \text{ W/m K}$;
- $c_{pL} = 2470 \text{ J/kg K}$;
- $pc_{\text{rit}} = 4264 \text{ kPa}$;
- $Pr_L = \frac{c_{pL}}{\mu_L/k_L} = 3.08$;
- $k_G = 0.0159 \text{ W/m K}$;
- $c_{pG} = 1880 \text{ J/kg K}$ so that $Pr_G = 0.887$.

**Solution:** The mass velocity of the total flow of liquid plus vapor is:

$$m = \frac{M}{\pi D^2/4} = \frac{0.03534}{\pi(0.015)^2/4} = 200 \text{ kg/m}^2 \text{s}$$
From [5.8.5], the equivalent Reynolds number of Akers et al. (1959) is:

\[
\bar{m}_e = 200 \left(1 - 0.5 + 0.5 \left(\frac{528}{11}\right)^{1/2}\right) = 792.8 \text{ kg/m}^2\text{s}
\]

so that the equivalent Reynolds number is:

\[
Re_e = \frac{\bar{m}_e D}{\mu_L} = \frac{792.8(0.015)}{0.0001345} = 88416
\]

For \(Re_e > 50,000\), \(C = 0.0265\) and \(n = 0.8\). Applying [8.1.4], the local condensing heat transfer coefficient of Akers is:

\[
\bar{\alpha}(x) = \frac{0.0265(88416)(3.08)^{0.8}}{0.108} = \frac{2516 \text{ W/m}^2\text{K}}{}
\]

Turning to the Shah method, the reduced pressure is 0.1189 and the liquid Reynolds number is:

\[
Re_L = \frac{m D}{\mu_L} = \frac{200(0.015)}{0.0001345} = 22305
\]

Using [8.1.6], the local condensing heat transfer coefficient is:

\[
\bar{\alpha}(x) = \frac{0.023(22305)^{0.8}(3.08)^{0.8} (1 - 0.5)^{0.4} + \frac{3.8(0.5)^{0.7}(1 - 0.5)^{0.04}}{(0.1189)^{0.34}}}{0.108} = \frac{4283 \text{ W/m}^2\text{K}}{}
\]
The **Dobson and Chato** method is implemented as follows. The three dimensionless groups are calculated from \([8.1.9]\), \([8.1.10]\) and \([8.1.14]\):

\[
\begin{align*}
\text{Re}_{Ls} &= \frac{200(0.015)(1 – 0.5)}{0.0001345} = 11152 \\
\text{Fr}_{so} &= \frac{1-0.5}{0.5} \left(\frac{11}{528}\right)^{0.8} \left(\frac{0.0001345}{0.0000075}\right)^{0.1} = 0.1926 \\
\text{Ga}_{L} &= \frac{9.8(528)(528 – 11)(0.015)^3}{(0.0001345)^2} = 499600000
\end{align*}
\]

The transition criterion is obtained from \([8.1.21]\):

\[
\text{Fr}_{so} > 20, \quad \text{Nu}(x) = 0.023(11152)^{0.8}(3.08)^{0.4} \left[1 + \frac{2.22}{0.1926^{0.8}}\right] = 662.2
\]

From \([8.1.8]\), the local condensing coefficient is obtained:

\[
\text{Nu}(x) = \frac{\alpha(x)(0.015)}{0.108} = 662.2 \\
\alpha(x) = 4768 \text{ W/m}^2\text{K}
\]

Thus, the methods of Akers, Shah and Dobson-Chato give the respective values of 2516, 4283 and 4768 W/m² K.
The fall in the dew point temperature over the entire condensation range is 5°C (= \(dT_{\text{dew}}\)). The total enthalpy change is that of the latent heat plus sensible heat. The latter can be estimated as the mean of the liquid and vapor specific heats applied to the condensing temperature glide of 5°C. Thus, \(dh = (1/2)(2470+1880)(5) + 373100 = 10875 + 373100 = 383975 \text{ J/kg}\). Applying [8.3.2] gives:

\[
Z_{ii} = 0.5(1880)\frac{5}{383975} = 0.01224
\]

The Reynolds number of the vapor fraction is:

\[
Re_v = \frac{2000\times0.015\times0.51}{0.0000075} = 200000
\]

The convection heat transfer coefficient to the vapor is obtained with the Dittus-Boelter single-phase turbulent flow correlation:

\[
Nusselt = 0.023(200000)^{0.8} \times 0.887^{0.4}
\]

\(\alpha_v = 404.6 \text{ W/m}^2\text{K}\)

Applying [8.3.1] gives the condensing coefficient of the mixture as:

\[
\frac{1}{\alpha_{\text{eff}}} = \frac{1}{4768} + 0.01224
\]

\(\alpha_{\text{eff}} = 4160 \text{ W/m}^2\text{K}\)

The mass transfer effect reduces the condensing heat transfer coefficient by 13% for these conditions.
13: Propane is condensing inside a horizontal, plain tube whose internal diameter is 15 mm. The refrigerant enters at its saturation temperature of 2°C (5.07 bar) as a saturated vapor and leaves as a saturated liquid. The flow rate of vapor entering is 0.006 kg/s and the tube wall has a mean internal temperature of -10°C. Determine the following values: the local condensing heat transfer coefficient using (i) Akers et al., (ii) Dobson-Chato and (iii) Thome et al. methods at a vapor quality of 0.5. Use the properties in the Example Calculation. Surface tension = 0.02 N/m