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Extending ORNL HPDM Capabilities for Design and Optimization of New Refrigerant Blends – FY19 2nd Quarter (Regular) Milestone Report: Literature Review of Heat Transfer and Pressure Drop Correlations for HFO blends in Water Heating application

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Extending ORNL HPDM Capabilities for Design and Optimization of New Refrigerant Blends – FY19 2nd Quarterly Milestone Report: Literature Review of Heat Transfer and Pressure Drop Correlations for HFO Blends in Water Heating Application

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Literature review of heat transfer and pressure drop correlations for HFO blends in water heating application
(Regular Milestone)

Executive Summary

Based on a comprehensive literature review, the two-phase heat transfer and pressure drop correlations, best suitable for modelling the next generation HFO blends in brazed plate heat exchangers and fluted tube-in-tube heat exchangers, were identified. The selected correlations have been incorporated to the segment-to-segment (finite volume) heat exchanger models in the HPDM library. Details are described below.

Two-Phase Heat Transfer in Brazed Plate Heat Exchangers

<u>Boiling</u>

Giovanni et al. (2015a) presented a new model for refrigerant flow boiling inside Brazed Plate Heat Exchangers (BPHEs) based on a set of 251 experimental data. The data covered HFC refrigerants (HFC236a, HFC134a, HFC410A), HC refrigerants (HC600a-Isobutane, HC290-Propane, HC1270-Propylene), and also a new low Global Warming Potential (GWP) HFO refrigerant (HFO1234yf). The new model works for nucleate and convective boiling. A set of 505 experimental data by different laboratories, were used to validate the new model, including HFC134a, HFC410A, HFC507A and HCFC22 with different plate geometries.

Two-phase evaporation of BPHEs comprises two mechanisms, i.e. nucleate boiling and convective boiling, where nucleate boiling is dominated by heat flux and convective boiling is controlled by mass flux and quality. The Thonon et al. (1997) criterion is used to separate the two boiling regimes, as given below:

$$Bo = q/(G * \Delta h_{LV}) \qquad \text{(Eq.1)}$$

$$X_{tt} = \left[\frac{1 - X_m}{X_m}\right]^{0.9} (\rho_G/\rho_L)^{0.5} (\mu_G/\mu_L)^{0.1} \qquad \text{(Eq.2)}$$

Where Bo is dimensionless boiling number. q is heat flux. G is mass flux. Δh_{LV} is refrigerant vaporization heat. X_{tt} is dimensionless Martinelli number. X_m is vapor quality. ρ_G and ρ_L are refrigerant vapor and liquid density, respectively. μ_G and μ_L are refrigerant vapor and liquid viscosity, respectively.

When $BoX_{tt} > 0.15e-3$, it is dominated by nucleate boiling. When $BoX_{tt} < 0.15e-3$, it is dominated by convective boiling.

In convective boiling regime, the heat transfer coefficient is calculated as:

$$h_{cb} = 0.122 * \emptyset * (k_l/d_h) Re_{eq}^{0.8} Pr_L^{1/3} \qquad \text{(Eq.3)}$$

Where $\emptyset$ is brazed plate surface enlargement factor, k_l is the liquid thermal conductivity, d_h is the channel hydraulic diameter, Pr_L is the liquid Prandtl number, Re_{eq} is equivalent Reynolds number.

$$Re_{eq} = G\left[(1 - x) + x\left(\frac{\rho_L}{\rho_G}\right)^{\frac{1}{2}}\right] d_h/\mu_L \qquad \text{(Eq.4)}$$

In the pool boiling dominated regime, the heat tranfser coefficient considers impacts of heat flux, reduced pressure and plate surface roughness.

$$h_{nb} = 0.58 * \emptyset * h_0 * C_{Ra} * F(P^*) * (q/q^0)^{0.467} \qquad \text{(Eq.5)}$$

Where C_{Ra} represents the effect of plate surface roughness $C_{Ra} = (R_a/0.4)^{0.1333}$, R_a is the surface roughness [μm]. $(q/q^0)^{0.467}$ represents the heat flux term, $q^0 = 20{,}000$. $F(P^*)$ represents the effect of reduced pressure.

$$F(P^*) = 1.2 * P^{*\,0.27} + \left[2.5 + \frac{1}{1 - P^*}\right] P^* \qquad \text{(Eq.6)}$$

P^* is the reduced pressure, i.e. the refrigerant local pressure divided by its critial pressure. h_0 is a base pool boiling coefficient specific to each individual refrigerant type. It can be calculated using Cooper correlation.

$$h_0 = 55 P_0^{*\,(0.12 - 0.2 log10(0.4))} (-log10(P_o^*))^{-0.55} q_0^{0.67} M^{-0.5} \qquad \text{(Eq.7)}$$

Where $P_o^* = 0.1$, q_0=20000, M is the refrigerant molar weight [g/mole].

Condensation

Based on the study of Giovanni et al. (2015b), condension in brazed plate heat exchangers are dominated by two mechnisms, including gravity-controlled region and forced convection region. The criterion to separate the two regions is the two phase Renolds number as given in Equation 4. When $Re_{eq} < 1600$, it is the forced convection region; $Re_{eq} > 1600$, it is the gravity-controlled region.

In a gravity-controlled region, the average condensation coefficien along the flow channel is:

$$h_{grav,ave} = 0.943 * \emptyset * [(k_L^3 \mu_L^2 g h_{LV})/(\mu_L \Delta T L)]^{1/4} \qquad \text{(Eq.8)}$$

Where ΔT is the temperatue difference between the plate surface and the refrigerant saturation. L is the plate flow channel length. Pr_L is the liquid phase Prandtl number.

In a forced convection region,

$$h_{fc} = 1.875 * \emptyset * (k_l/d_h) Re_{eq}^{0.445} Pr_L^{1/3} \qquad \text{(Eq.9)}$$

Heat Transfer and Pressure Drop in Fluted Tube-in-Tube Heat Exchangers

Rousseau et al. (2003) modelled and validated a tube-in-tube condenser with complex fluted geometries. They proposed a method to calcualte two-phase heat transfer respectively at the inner tube and annualar sides, which is to scale the heat transfer of single-phase heat transer in a smooth tube using two ratios, one acccounts for geometry factor i.e. friction pressure drop or single-phase heat transfer in a fluted tube versus a smooth tube and, the other account for two-phase heat transfer versus single-phase heat transfer in a smooth tube.

The geometrical factors below are specific to fluted tubes:

$$D_{vi} = \sqrt{\frac{4Vol}{\pi L}} \qquad \text{(Eq.10)}$$

Where D_{vi} is the tube side volume-based diameter, Vol is the tube side volume, L is the coiled tube length. The volume based outside diameter of the fluted tube is calculated as:

$$D_{vo} = D_{vi} + 2t \qquad \text{(Eq.11)}$$

t is the wall thickness.

The flute angle is calculated as:

$$\theta = \arctan\left(\frac{\pi D_{vo}}{Np}\right) \qquad \text{(Eq.12)}$$

Where N is the number of flutes, p is the flute pitch.

Condensation

Annular side:

The single-phase friction coefficient of helical coil is calculated using Das (1993).

$$f_{helical} = 4[0.079Re_v^{-0.25} + 0.075\left(\frac{\frac{D_{ho}}{D_{vo}}}{\sin(\theta)}\right)^{0.5} + 17.5782Re_v^{-0.3137}\left(\frac{\frac{D_{ho}}{D_{vo}}}{\sin(\theta)}\right)^{0.3621}(e/D_{ho})^{0.6885}]$$

(Eq.13)

Where Re_v is the vapor phase Reynolds number at the total mass flow rate based and annualar side hydraulic diameter. $D_{ho} = D_{o,i} - D_{vo}$ is the annualar hydraulic diameter, given by the difference between the inside diameter of the outer tube and the outside volumetric diameter. e is the flute depth.

The friction coefficient of a straight tube is adopted from Swamee and Jain (1976).

$$f_{straight} = 0.25/(log10(\frac{r/D_{ho}}{3.7} + 5.74/{Re_v}^{0.9}))^{2.0} \quad \text{(Eq.14)}$$

Where r is the tube surface roughness. Thus, the heat transfe enhancement due to the fluted geometry is

$$r_h = f_{helical}/f_{straight} \quad \text{(Eq.15)}$$

According to the Shah correlation, the two-phase condensation enhancement factor is calcualted as:

$$r_{tp} = (1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{Pr^{0.38}} \quad \text{(Eq.16)}$$

If use Dittus-Boelter correlation to calculate single-phase heat transfer in a stright tube, the two-phase condensation heat transfer at the annular side is given as:

$$h_{tp} = 0.023Re_l^{0.8}Pr_l^{0.4}k_l/D_{ho}r_h r_{tp} \quad \text{(Eq.17)}$$

Where Re_v is the vapor phase Reynolds number at the total mass flow rate based and annualar side hydraulic diameter.

Tube side:

Based on the work by Arnold and Christensen et al. (1993), the helical tube side single-phase heat transfer (Nusselts number) is:

When $Re_l \leq 5000$

$$Nu_{flute} = 0.014Re_l^{0.842}(e^*)^{-0.067}(p^*)^{-0.293}(\theta^*)^{-0.705}Pr^{0.4} \quad \text{(Eq.18)}$$

When $Re_l > 5000$

$$Nu_{flute} = 0.064Re_l^{0.773}(e^*)^{-0.242}(p^*)^{-0.108}(\theta^*)^{-0.599}Pr^{0.4} \quad \text{(Eq.19)}$$

Where $e^* = flute\ depth/D_{vi}$, $p^* = flute\ pitch/D_{vi}$, $\theta^* = \theta/(\pi/2)$.

At tube side, the single-phase heat transfer enhancement factor due to the fluted geometry is $r_h = Nu_{flute}/Nu_{straight}$. And $Nu_{straight}$ can be calculated using the Dittus-Boelter corrlation.

Evaporation

Huang et al. (2014) modelled and validated a fluted tube-in-tube evaporator. They correlated single-phase heat transfer as below:

$$Nu_{lo} = \frac{f_{lo}/Re_{lo}Pr}{1 + 9.77\left(P_r^{\frac{2}{3}} - 1\right)\sqrt{\frac{f_{lo}}{8}}}{Re_{lo}}^{-0.2}(e^*)^{-0.32}(p^*)^{-0.28}(r^*)^{-1.64} \quad \text{(Eq.20)}$$

When $Re_{lo} \leq 800$, the liquid friction pressure drop coefficient is calculated as:

$$f_{lo} = 96(r^*)^{0.035}/Re_{lo}(1 + 101.7)\ {Re_{lo}}^{0.52}(e^*)^{1.65 + 2\theta^*}(r^*)^{5.77} \quad \text{(Eq.21)}$$

When $Re_{lo} \leq 40000$

$$f_{lo} = 4\left[1.7372\ln\left(\frac{Re_{lo}}{1.964\ln(Re_{lo}) - 3.8215}\right)\right]^{-2}(1 + 0.0925r^*)e_l \quad \text{(Eq.22)}$$

$$e_l = 1 + 222\, {Re_{lo}}^{0.09}(e^*)^{2.4}(p^*)^{-0.49}(\theta^*)^{-0.38}(r^*)^{2.22} \text{(Eq.23)}$$

Where $r^* = D_{vo}/D_{o,i}$.

They used Kandlikar (1991) to calcualte two-phase evaporation heat transfer.

The nucleate boiling term:

$$h_{nb} = 0.6683Co^{-0.2}(1-x)^{0.8}h_{lo} + 1058Bo^{0.7}(1-x)^{0.8}F_{fl}h_{lo} \quad \text{(Eq.24)}$$

The convective boiling term:

$$h_{cb} = 1.136Co^{-0.9}(1-x)^{0.8}h_{lo} + 667.2Bo^{0.7}(1-x)^{0.8}F_{fl}h_{lo} \quad \text{(Eq.25)}$$

Where $Co = (1-x)^{0.8}(\frac{\rho_g}{\rho_l})^{0.5}$ is the convective dimension less number.

F_{fl} is empirial parameter specific to each individual refrigerant type. If not known, $F_{fl} = 1.0$.

The overall boiling coefficient is the larger value between the nucleate boiling and convective boiling.

$$h_{evap} = max(h_{nb}, h_{cb}) \quad \text{(Eq.26)}$$

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