"Advances in Condensation Technology: Numerical Simulations to Improve Condensation Heat Transfer and Pressure Drops in Circular, Non-Circular and Inclined Tubes"

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Description of the LTCM Lab

- 400m² two-phase flow and heat transfer lab specializing in macro- and micro-scale flows.
- LTCM lab has 20+ post-docs and Ph.D. students.
- Prof. Thome is author of 4 books, including *Engng. Databook III*, free e-book at [www.wlv.com](http://www.wlv.com) and 15 years as full-time consultant from 1984-1998.
- Research work covers: two different inhouse 3D numerical codes, inhouse modified Fluent code, 1D unified set of methods for annular flows, flow-pattern based heat transfer and pressure drop methods, extensive experimental and flow visualization work.
Better Simulations for Advanced Condensers

• More energy efficient condenser designs can be attained using better pressure drop models for steam-side...they give the local $T_{\text{sat}}$ for the incremental LMTD calculations, so just as important as $U_o$ when going for close temperature approaches!

• SHAPE COUNTS! Non-circular channels can improve heat transfer without increase in pressure drop...but non-condensable gas (air) effects need to be investigated.

• Designs should now be looking at true-local htc’s...those around perimeter with air-cooled fins on outside...use of perimeter $U_o$ values is more accurate in handling fin efficiency calculations since htc’s are very non-uniform.
Surface Tension in Non-Circular Channels

Local laminar flow heat transfer coefficient:

Heat transfer coefficient = \frac{(\text{thermal conductivity of liquid})}{(\text{film thickness})}
Description of the LTCM Numerical Model

- Developed first a 2-D model for simulation with surface tension effects but **without** axial vapor shear: *use for fast screening of potential geometries.*
- Developed a detailed 3-D model including axial vapor shear, surface tension, conjugate, etc. effects: *use for detailed modeling of local heat transfer and pressure drops in non-circular channels.*
- Validated model vs. experimental data from four different labs so far (various fluids and shapes).
- Code particularly useful to: develop new geometries/enhance heat transfer, simulate effect of tolerances, simulate oil effects, etc.
3D Model: Major Assumptions

Major hypothesis: PhD. Thesis of Dr. S. Nebuloni / new thesis by N. Antonsen
- all the inner perimeter of the test section is wetted by the liquid
- laminar condensate film
- Newtonian fluids
- vapor is at saturation conditions at each section
- the thermal inertia of the condensate is neglected
- wall and saturation temperature are constant in time and space
- fluid properties ($\sigma$, $\rho_L$, $\rho_V$) are constant in the domain
- inlet boundary conditions are constant in time (fixed inlet vapor quality and mass flow rate)

In the development of the Model, the following effects are taken into consideration:
- axial and cross-sectional curvature of the liquid-vapor interface
- interfacial shear stresses at the L-V interface
- gravity vector variable in direction ($x$-$y$-$z$) and time

The vapor domain (mass balance, momentum equations) is treated as mono-dimensional (vapor mean velocity, pressure, ... are function of time and axial location)

The interfacial vapor shear is calculated using the approach suggested by Mickley et al. and the friction factor is obtained by Churchill correlation
3D Model: Implementation & Control Volume

Equations are implemented with finite volume method

Mass conservation:

\[
\frac{d}{dt} \iint_{V_{j,k}} dV + \iiint_{\partial V_{j,k}} \left( \bar{u} + \bar{v} \right) \cdot \bar{n} dS = \frac{1}{\rho_l} \iint_{\partial V_{j,k}} \frac{\partial \dot{m}_c}{\partial S} dS
\]

Momentum balance projected on the curvilinear coordinate \( s(w) \):

\[
\dot{s}_{j,k} \cdot \rho_l \frac{d}{dt} \iint_{V_{j,k}} \bar{u} dV + \dot{s}_{j,k} \cdot \rho_l \iiint_{\partial V_{j,k}} \bar{u} \cdot \bar{n}_s dS + \dot{s}_{j,k} \cdot \rho_l \iiint_{\partial V_{j,k}} \bar{u} \cdot \bar{n}_z dS =
\]

Momentum balance projected on the axial coordinate \( z \):

\[
\dot{z}_{j,k} \cdot \rho_l \frac{d}{dt} \iint_{V_{j,k}} \bar{v} dV + \dot{z}_{j,k} \cdot \rho_l \iiint_{\partial V_{j,k}} \bar{v} \cdot \bar{n}_z dS + \dot{z}_{j,k} \cdot \rho_l \iiint_{\partial V_{j,k}} \bar{v} \cdot \bar{n}_s dS =
\]

Pressure variation across the interface: \( \Delta p_{L-V} = \sigma (\kappa_z + \kappa_w) \)

For each control volume, both the liquid film velocities and geometrical surface (which depends on the local film thickness) are changing with time; therefore each time derivative splits into two parts:

\[
\frac{d}{dt} \iiint_{V_{j,k}} \bar{u}(t, w, z) dV = \frac{d}{dt} \left[ \dot{u}_m(t, w, z) \cdot \Delta V_{j,k}(t, w, z) \right] = \frac{\partial \dot{u}_m}{\partial t} \Delta V_{j,k} + \dot{u}_m \frac{\partial \Delta V_{j,k}}{\partial t}
\]

Comparison of Our Model to Wang-Rose Model

Above: comparison of simulated mean (over the perimeter of the channel) heat transfer coefficients of R-134a obtained with the model by Wang and Rose (circles in black), by the reduced actual model (dashed line, in blue) and the complete model (continuous line, in red); these simulations have been obtained for a square channel having 1 mm hydraulic diameter and for the following parameters: \( G = 300 \text{ kg/(m}^2\text{s}) \), \( \Delta T = 6 \text{K} \), \( T_{\text{sat}} = 50 \text{ C} \) and normal gravity
Comparison of Our Model to Experimental Data

\[ \frac{h_{\text{mod}}}{h_{\text{exp}}} \]

- \( D_h = 2 \text{ mm} \)
- \( D_h = 0.691 \text{ mm} \)
- \( D_h = 0.493 \text{ mm} \)

\( X_{\text{exp}} - \text{average vapor quality} \)
This version of the model includes the coupling between the thin film fluid dynamics, the heat transfer in the condensing fluid and the heat conduction in the channel wall. *Comparison vs. Univ. of Padova data*
2D Condensation Model: Elliptical Shape

A parametric study can be done to see the influence of the geometry on the performance of different shapes:

\[ x(\theta) = a \cos(\theta) \]
\[ y(\theta) = b \sin(\theta) \]

Varying the aspect ratio \( r \) while keeping constant the hydraulic diameter, the resulting \( h-e \) profiles can be obtained:

Fluid is FC 7100, \( T_{\text{sat}} = 70^\circ\text{C} \),
\( D_h = 1\text{ mm} \),
\( g_y = -9.81 \text{ m/s}^2 \)

Local heat transfer coefficient distribution (red line)
Steady state condition:
Mass flow rate: \( G = 50 \frac{kg}{s m^2} \)
Inlet vapor quality: \( X_{in} = 0.99 \)

Fluid: FC-7100
Saturation temperature: \( T_{sat} = 70°C \)
Wall temperature: \( T_{wall} = 69.5°C \)
Gravity field: \(|g| = 9.81 \text{ m/s}^2\)
3D Condensation Model: Elliptical Shape

Steady state condition:
Mass flow rate: \( G = 50 \frac{kg}{sm^2} \)
Inlet vapor quality: \( X_{in} = 0.99 \)

Fluid: FC-7100
Saturation temperature: \( T_{sat} = 70^\circ C \)
Wall temperature: \( T_{wall} = 69.5^\circ C \)
Gravity field: \( |g| = 9.81 \text{ m/s}^2 \)

Much higher \( h \) than circular channel!
2D Condensation Model: Flower Shape

HFE7100 (video):
- $D_h = 1 \text{ mm}$
- $n=4$
- $\Delta R = 100 \mu m$
- $g_y = -9.81 \text{ m/s}^2$
- $\Delta T = 0.5 \text{ K}$
3D Condensation Model: Flower Shape

Steady state condition
Mass flow rate: \( G = 50 \frac{kg}{sm^2} \)
Inlet vapor quality: \( X_{in} = 0.99 \)

Fluid: FC-7100
Saturation temperature: \( T_{sat} = 70^\circ C \)
Wall temperature: \( T_{wall} = 69.5^\circ C \)
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3D Condensation Model: Flower Shape
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Cross sectional plots of liquid vapor interface at different axial locations
Shape Effects on HTC’s and Pressure Drops

Fig. 4. Nusselt numbers plotted versus the dimensionless axial location for elliptical shape channels (E) with different aspect ratios $r_E$ and Bond numbers for the following conditions: $Sn = 30, Re_s = 0.4, We_v = 8, \Gamma = 0.05, Re_v = 15,000$. $\Delta X$ is the change in vapor quality from inlet to outlet and $\Delta P/\Delta X$ is the ratio between the dimensionless vapor phase total pressure drop and the vapor quality variation.

Fig. 5. Nusselt numbers plotted versus the dimensionless axial location for flattened shape channels (F) with different aspect ratios $r_F$ and Bond numbers for the following conditions: $Sn = 30, Re_s = 0.4, We_v = 8, \Gamma = 0.05, Re_v = 15,000$. 

$r_E = 1.0, Bo_x = 2: \Delta X = 0.439, \Delta P/\Delta X = 4.91$
$r_E = 1.2, Bo_x = 2: \Delta X = 0.434, \Delta P/\Delta X = 4.96$
$r_E = 1.2, Bo_y = 2: \Delta X = 0.415, \Delta P/\Delta X = 5.22$
$r_E = 1.4, Bo_x = 2: \Delta X = 0.507, \Delta P/\Delta X = 3.77$
$r_E = 1.4, Bo_y = 2: \Delta X = 0.46, \Delta P/\Delta X = 4.4$

$r_F = 0.0, Bo_y = 2: \Delta X = 0.439, \Delta P/\Delta X = 4.91$
$r_F = 0.5, Bo_y = 2: \Delta X = 0.513, \Delta P/\Delta X = 3.6$
$r_F = 1.0, Bo_y = 2: \Delta X = 0.565, \Delta P/\Delta X = 3.19$
$r_F = 1.5, Bo_y = 2: \Delta X = 0.553, \Delta P/\Delta X = 3.4$
$r_F = 0.5, Bo_y = 8: \Delta X = 0.590, \Delta P/\Delta X = 2.96$
$r_F = 1.0, Bo_y = 8: \Delta X = 0.556, \Delta P/\Delta X = 3.28$
$r_F = 1.5, Bo_y = 8: \Delta X = 0.542, \Delta P/\Delta X = 3.43$
Recent & Future Steps for Advanced Condensers

- 3D model: recent improvements include transient effects of gravity, uniform heat flux & convective b.c.’s; 1D annular model: includes entrainment, film thickness, etc.
- Our models compare very well vs. experimental data.
- Energy efficient condenser designs can be attained using better pressure drop models...they give the local $T_{\text{sat}}$ for incremental LMTD calculations, so just as important as $U_0$!
- SHAPE COUNTS! Non-circular channels can improve heat transfer without increase in pressure drop...but non-condensable gas (air) effects need to be investigated.
- Designs should now be looking at true-local h.t.c.’s...those around internal perimeter with air-cooled fins on outside.
- Numerical 2-phase heat transfer DOES have value once it is validated experimentally! Do NOT use non-validated codes.