Internal Annular Flow-boiling and Flow-condensation: Context, Results and Recommendations

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Abstract

In-tube boiling and condensing flows, a subset of the area of phase-change heat transfer, are reviewed here with emphasis on their annular regime realizations in innovative devices. The review is also relevant because of recently reported experimental and technological approaches that make it feasible for annular/stratified regimes to cover the entire lengths of millimeter scale flow-boilers and flow-condensers. The review article’s content also relates to new ways of addressing challenges that have emerged in the area of high heat-flux (500-1000 W/cm² or greater) cooling of data centers, super computers, laser weapons, and other devices. The article summarizes current knowledge base and provides a design example each for steady annular operations of flow-boilers and flow-condensers. Furthermore, the article deals with some recent breakthroughs in the simulation of annular boiling/condensing flows and how such approaches can be extended – in conjunction with well-planned and parallel experiments – to further develop the science and innovative applications associated with steady or pulsatile operations of annular flow-boilers and flow-condensers.
1. Introduction

In-tube boiling and condensing flows, a subset of the area of phase-change heat transfer, has been a subject of scientific studies – involving experiments, modeling, computations, and analysis – since the early 1900s. As engineering support for boiler and condenser operations in macro-scale applications (as in power generations sector, heating ventilation and air conditioning refrigeration, waste heat recovery, steam generation, etc.) have matured, new challenges have emerged in the area of high heat flux (500-1000 W/cm² or greater) cooling for data center, super computers, laser weapons, and other devices. One of these challenges is smallness of available space for device cooling (that must employ flow-boilers) and occasionally, also for heat rejection (for devices that must employ flow-condensers). The space restrictions and low pumping power needs often limit hydraulic diameter $D_h$ of the boiler/condenser tubes to be small but not too small (e.g. $100 \mu m \leq D_h \leq 8$ mm) whereas safety and cost effectiveness issues restrict pure fluids to available refrigerants and/or water.

Another challenge is a restriction on allowed boiling surface and/or condensing-surface temperatures. For example, typically, 75-85 °C is the maximum allowed temperature for the boiler tube surfaces connected to cold plates used in electronic cooling (Kandlikar and Hayner 2009; also see chapter: Single and Multiphase Flow for Electronic Cooling). Besides high heat removal capabilities at high heat-fluxes, there are demands on effectiveness of such millimeter-scale boilers and condensers. For example, effectiveness often requires low pumping power consumption through manageable pressure drops – and, in case of high heat-flux flow-boilers, it may also mean an ability to recover large amounts of mechanical power available (associated with higher exiting vapor speeds from mm-scale ducts) at the exit.

Often effectiveness also requires avoidance of vapor compressibility related choking effects (Ghiaasiaan 2007) and this, in turn, may require modest mass-fluxes ($G$) in tubes of modest hydraulic diameters ($D_h \equiv 4A/P$, where $A$ and $P$ respectively represent the cross-sectional area and perimeter of the tube). Based on available experimental studies and experiences, modest $G$ ($10 \leq G \leq 500$ kg/m².s) means avoiding high $G >500$ kg/m².s and modest $D_h$ ($100 \mu m \leq D_h \leq 8$ mm) means tube diameters that are neither too small (e.g., $D_h \geq 100 \mu m$) nor too large (e.g., $D_h \leq 8$ mm). This would mean, according to some popular naming conventions (Ghiaasiaan 2007; Kandlikar and Grande 2003; Mehendale et al. 2000), that effectiveness considerations may restrict
one to mini-channels ($\sim 100 \, \mu m \leq D_h \leq 1 \, mm$) and a subset of macro-channels ($\sim 1 \, mm \leq D_h \leq 8 \, mm$) – excluding micro-channels ($\sim 10 \, \mu m \leq D_h \leq 100 \, \mu m$) as well as very large diameter ($D_h \geq 8 \, mm$) macro channels.

Fig 1: (a) Typical flow-regimes encountered in traditional operation of a horizontal flow-boiler (b) For certain $G$ values and heating conditions, expected qualitative variation in local values of heat transfer coefficient $h_x$ over $0 \leq x \leq L$ are shown.

Unfortunately, established technologies, such as an array of small diameter heat pipes or their Capillary and Looped heat pipe variations (Faghri 1995) are not appropriate solutions for the increasing high heat-flux and heat-load needs of these applications (Agostini et al. 2007; Ball 2012; also see chapter: *Heat Pipes and Thermosyphons*). This is because heat pipes do not work well,
for these applications, under the limitations of: necessarily large vapor speeds involved, wicking limits, and the need for large distances between boilers and condensers.

Innovative and effective operations (Kivisalu 2015; Kivisalu et al. 2014) may also require a change from traditional flow-boiler and flow-condenser operations associated with multiple flow-regimes (see, e.g., Figs. 1-2 for depiction of different flow-regimes in a horizontal duct, and Carey 1992 and Ghiaasiaan 2007 for discussions on the different flow-regimes) to operations that control the flow-regimes through control of inlet quality – and attain annular flow-regime (see Figs. 3 a-b) over all of a suitably chosen device length. Furthermore, to address high heat-flux needs (500-1000 W/cm² or greater), as discussed near the end of this review, resonant pulsations may be needed to create vapor acoustics-enabled large amplitude standing waves on the interface of highly stabilized thin liquid film annular flows realized through flow-controls shown in Figs 3a-b. Such wavy annular flow realizations are also needed to impose pulsations (Kivisalu 2015; Kivisalu et al. 2014) and take advantage of high heat-flux contact-line flow-physics normally associated with pool boiling during an ebullition cycle. For discussions on pool boiling contact-line flow-physics, see Gerardi et al. (2010), Kunkelmann et al. (2012), McHale and Garimella (2010) and Raghupathi and Kandlikar (2016) and the chapter: Nucleate Pool Boiling.

Fig 2: Typical flow-regimes encountered in traditional operation of a horizontal flow-condenser

By traditional operations of flow-boilers and flow-condensers (Figs. 1-2), it is meant here that a flow-boiler sees an all liquid flow at the inlet with the inlet temperature close to saturation temperature (i.e., only a slight sub-cooling) and a flow-condenser sees an all vapor flow at the inlet with the inlet temperature close to saturation temperature (i.e., only a slight super heating). The near inlet downstream flow-regime in Fig. 1a often consists of a nucleate boiling flow-regime which may be more (at certain low mass-flux $G$ values, making the scenario similar to pool boiling) or less (at higher moderate mass-flux $G$ values) of a contributing factor – relative to the convective
contributions to the total heat transfer rates – further downstream of the inlet (see Fig. 1b). The horizontal configuration flow-regimes in Figs. 1-2 are different than flow-regimes associated with downward or upward inclined (with respect to the gravity vector) tubes (Carey 1992; Ghiaasiaan 2007). For modest to high mass-flux values of $G$ and millimeter-scale hydraulic diameters $D_h$, interfacial shear and pressure gradients may dominate effects of gravity and, therefore, even tubes at different inclinations may exhibit flow-regimes that are qualitatively close to the ones associated with the horizontal configuration (as shown in Figs. 1-2). This is because axial and transverse gravity components $g_x$ and $g_y$ may become effectively negligible compared to inertia forces (see definitions of Froude numbers, introduced later on in this paper). It is also well known that there are some qualitative similarities between the boiling and condensing flow-regimes in Figs. 1-2 with regimes that are encountered in adiabatic flows (Carey 1992; Ghiaasiaan 2007). Despite this, even if saturated isothermal liquid and saturated isothermal vapor (constituting adiabatic flows) flows at different uniform (for all duct locations) qualities $X$ (where $X \equiv \dot{M}_V/\dot{M}_{in}$, where $\dot{M}_V$ represents the vapor mass flow rate and $\dot{M}_{in} \equiv \dot{M}_V + \dot{M}_L$ represents the total mass flow rate, including the liquid mass flow rate $\dot{M}_L$) are compared with saturated boiling/condensing flows of Figs. 1-2 at a particular location “$x$” where the quality $X$ is the same for the same tube diameter $D_h$ and mass-flux $G$; there are bound to be quantitative differences in the quality-based flow-regime transition boundaries (i.e., differences in critical values of $X$, denoted as $X_{cr}$, that typically need to be ascertained through experimental identification of such boundaries) – particularly for annular flows involving high heat-fluxes and high interfacial mass-fluxes (which are zero for the adiabatic flows).

Consider low to modest mass-flux steady boiling/condensing flow-regimes in Figs. 1-2. Clearly the spatial location where a quality $X(x)$ is realized depends on the knowledge with regard to “method of heating/cooling” (i.e. whether wall heat-flux or wall temperature values are known and uniform at all “$x$” locations or are non-uniform in a certain known way) and the “level” of heating/cooling (which is, either the average heat-flux $\bar{q}_w^w$ values or the average values of imposed temperature difference $\Delta T \equiv |T_w - T_{sat}(p_0)|$). The average spatial heat-flux values are associated with the steady wall heat-flux $q_w^w(x)$ values and the average spatial values of imposed temperature difference $\Delta T$ is associated with locally varying $\Delta T(x) \equiv |T_w(x) - T_{sat}(p_0)|$ values over $0 \leq x \leq L$, where $T_w(x)$ represents steady wall temperature variations, $\bar{T}_w \equiv \frac{1}{L}\int_0^L T_w(x).dx$ represents
average wall temperature, and $L$ is the device length. The realization of various steady flow-boiling regimes depend on the stability of the various time-averaged values of flow variables – which also depends on avoidance of phenomena associated with critical heat flux (CHF), as discussed in literature (Carey 1992; Ghiaasiaan 2007; Thome 2004). For characterizing macro-scale convection dominated phase-change flows, or the convective component of these flows, standard continuum-level fluid properties and associated non-dimensional numbers for flow-boiling and flow-condensation are sufficient (Naik and Narain 2016; Naik et al. 2016; Narain et al. 2015; Ranga Prasad et al. 2016).

Fig 3: (a) Innovative flow-boiler operations (b) Innovative flow-condenser operations
If one has to consider flow-boiling regimes which involve significant amount of nucleate boiling – then added dependences on fluid properties and other variables that characterize micro-/nano-layer liquid flows near contact-lines associated with nucleating bubbles (see Carey 1992; Ghiaasiaan 2007; Raghupathi and Kandlikar 2016, etc.), surface-texture measures that relate to nucleation site density, and parameters affecting bubble growth and departure must also be considered.

Similar to nucleate boiling (pool or flow), the analogue for condensation is nucleation of vapor in the form of dropwise condensation on hydrophilic/super-hydrophilic or hydrophobic/super-hydrophobic surfaces – with or without discussions of condensate removal mechanisms (inclined or vertical plates, moving vapor, etc.). Such condensation has high heat-flux capabilities and are of great current fluid-physics interests (see Rykaczewski et al. 2013, and also chapter on: Film and Dropwise Condensation). For reasons related to durability of mechanisms and development of applications of interest, the traditional (Fig. 2) and innovative (Fig. 3b) mm-scale annular flow-condensers remain the primary interest of this review.

For innovative steady operations (without the imposition of pulsations) depicted in Figs. 3 a-b, the primary focus of this review is on: realization of steady annular flows, its quantitative characterization, and flow-control based avoidance of neighboring non-annular flow-regimes (typically thermally inefficient plug-slug regimes in Figs. 1-2). Innovative operations, such as those in Fig. 3, involve: controlled recirculation rate for the passive vapor flow through proper setting of the inlet quality for a given device length L, mass-flux G, and a range of heating/cooling conditions. Characterization of the heat transfer efficiency for the flow in Fig. 3 involves quantitative assessment of the local values of heat transfer coefficient \( h_x \) appearing in the defining relationship: \( q_w^* (x) \equiv h_x \cdot \Delta T \). Typically, for the annular flows in Fig. 3, one seeks a correlation of non-dimensional values of \( h_x \) in terms of suitable non-dimensional values of overall flow variables, namely mass-flux G, quality \( X(x) \), heating levels (average heat flux \( \bar{q}_{w} \) representing \( q_w^*(x) \) variation or average temperature difference \( \Delta T \) representing \( \Delta T(x) \) variation), heating methods characterized by fixed spatial dependence of ratios such as \( q_w^*(x)/\bar{q}_w \) (or \( \Delta T(x)/\Delta T \)), and relevant fluid properties. Furthermore, clearly for the annular flows in Fig 3, the intermediate flow variables such an local film thickness \( \delta(x) \), interfacial shear, ratio of average gas and liquid phase velocities (\( U_V \) and \( U_L \)) being determined through their dependence on fluid
properties, quality $X$, and the void-fraction $\epsilon$ ($\equiv A_G/A$ where $A_G$ is the mean cross-sectional area occupied by the gas phase at location “x”, etc.) play a significant role in determining the spatial variations in the overall flow variables (such as $X(x)$). For a more general definition of the void-fraction ($\epsilon$), see Carey (1992), Ghaasiaan (2007), Thome (2004), etc. Another key intermediate flow variable that is characterized by interfacial shear is the local pressure gradient $dp/dx (x)$ which relates to pressure-drop and pumping power associated with the device. All these intermediate flow variables are very important in defining and obtaining suitably characterized relationships for the overall variables and the local heat transfer coefficient, $h_x$.

The above discussions necessitate a brief overview of relevant knowledge that exist with regard to correlations for:

- Heat transfer coefficient $h_x$ or its non-dimensional form termed Nusselt number $Nu_x$.
- Flow-regime maps and associated critical values of quality $X_{cr}$ that help identify flow-regime transitions with distance “x” while other non-dimensional parameters characterizing the flows’ realization (see Figs. 1-2) are well-defined.
- Void-fraction $\epsilon$ correlation and its dependence on quality $X$ and fluid properties.
- Pressure-gradient and pressure-drop correlations.
- An understanding of relevant critical heat-flux (CHF) mechanisms and available correlations.

Such an overview is undertaken in the next two sections.

2. Basic variables and correlations of scientific and engineering interest – and their relationship to one-dimensional modeling of flow-boiling and flow-condensation

2.1 Basic variables and their correlations

In this review, the primary focus is on the shear and pressure-drop driven annular flows inside tubes of circular (diameter $D$) or rectangular (height $h$ and width $w$) cross-sections, including channels (i.e. $h/w \ll 1$ rectangular cross-sections; also see chapters on: Flow Boiling in Tubes, and Boiling and Two Phase flow in Narrow Channels). Both traditional saturated flow-boiling and flow-condensation (Figs. 1-2) and innovative annular flow operations (Figs. 3a-b) are of interest. Unless otherwise stated, the characteristic length scale used in the literature is the hydraulic diameter:
\[ D_h \equiv \frac{4A}{P} \]  

where, \( A \) is the tube cross-sectional area and \( P \) is the heated/cooled perimeter. Furthermore, for present purposes, commercially smooth and hydrophilic heating/cooling surfaces (i.e. heat-exchange surfaces) are assumed for the flows in Figs. 1-3. This does not mean that other cross-sections (elliptical, etc.), mini and macro-fins, twisted tape inserts, special hydrophilic or hydrophobic micro and nano-structured surfaces (see, Kim et al. 2006, Mogaji et al. 2013, Thome 2004, Wu et al. 2013 etc.) aren’t actively being investigated – or relevant here (when discussed) – for performance enhancement purposes (also see chapters on: Boiling On Enhanced Surfaces and Boiling with Nano Particles).

As mentioned earlier, commonly available pure fluids (refrigerants, fluorinert electronic cooling fluids, water, etc.) are the primary focus of this article – although the literature also shows a significant amount of interest in phase change flows with azeotropic and non-azeotropic fluid mixtures, nano-fluids, and phase change in the presence of non-condensable gases (see, Dalkilic and Wongwises 2009, Kim et al. 2007, Shah 2015b, etc.; also see chapter on: Mixture Boiling).

The liquid and vapor phases in the flows of Figs. 1-3 are denoted with subscripts \( I = "L" \) and \( I = "V" \) respectively. Both phases are modeled as incompressible (i.e. vapor Mach numbers are low). The properties (density \( \rho \), viscosity \( \mu \), specific heat \( C_p \), and thermal conductivity \( k \)) are denoted with subscript “I”. The properties are to take their representative constant values for each phase (\( I = "L" \) or "V").

Let the temperature, pressure and velocity fields over the two phases – in the steady-in-the-mean flows depicted in the Figs. 1-3 – be respectively denoted as \( T_I, p_I \) and \( \vec{v}_I = u_I\hat{i} + v_I\hat{j} \). Let \( p_0 \) be the mean inlet pressure (or inlet pressure at a designated inlet location), \( h_{fg} \) be the heat of vaporization at a local interfacial pressure \( p \) or associated saturation temperature \( T_{sat}(p) \), \( T_w \) be the mean heat-exchange surface temperature associated with steady but spatially varying wall temperature \( T_w(x) \), \( \Delta T(\equiv |T_{sat}(p_0) - T_w|) \) be a representative controlling temperature difference between the fluid and the heat exchange surface, and \( \dot{M}_{in} \) be the total steady mass flow rate (kg/s) through the tube. Furthermore, let \( \dot{M}_{in} \) consist of liquid mass flow rate \( \dot{M}_L(x) \) and vapor mass flow rate \( \dot{M}_V(x) \) at any distance "x" from the inlet – i.e. \( \dot{M}_{in} = \dot{M}_L(x) + \dot{M}_V(x) \). Let \( X(x) (\equiv \dot{M}_V(x)/\dot{M}_{in}) \) be the local quality, and \( G (\equiv \dot{M}_{in}/A) \) be the mass-flux \((\text{kg/m}^2\cdot\text{s})\), and let the characteristic speed be \( U (\equiv \ldots \)
For specifying the heating/cooling method for the flows in Figs. 1-3, one may know the temperature when using temperature-controlled heating/cooling approach (typically arranged by a single-phase hot or cold fluid flowing around the ducts) or know the heat-flux values which is natural when using a heat-flux controlled heating/cooling approach (e.g. those arranged by electric heating of the ducts for flow-boiling investigations). Regardless of whether temperature or heat-flux or both are known in an arrangement, the "method of heating/cooling" or "thermal boundary condition" for the heat exchange surface in contact with the phase-change flows in Figs. 1-3, is characterized by known (or assumed) values of the wall temperature $T_w(x)$ or by known (or assumed) values of the wall heat-flux $q_w^*(x)$. It is common to replace wall temperature variations $T_w(x)$ by the characterizing temperature difference $\Delta T(x)$.

Next it is important to define the "level" of heating/cooling. Over the heating/cooling length “$L$” in Figs. 1-3, these "levels" are characterized by either the average temperature difference $\Delta T \equiv |\bar{T}_w - T_{sat}(p_0)|$ between the saturation temperature $T_{sat}(p_0)$ and the axially averaged mean wall temperature $\bar{T}_w \equiv \frac{1}{L} \int_0^L T_w(x). dx$ or the axially averaged mean wall heat-flux $\bar{q}_w^* \equiv \frac{1}{L} \int_0^L q_w^*(x). dx$. Furthermore, a specific non-uniform "method of heating/cooling" is defined as $\theta_w(x)$ or $\Psi_q(x)$ through the relations:

$$\Delta T(x) \equiv |T_w(x) - T_{sat}(p_0)| \equiv \Delta T. \theta_w(x) \quad (2)$$

and

$$q_w^*(x) = q_w^*(x). \Psi_q(x) \quad (3)$$

For non-uniform temperature-controlled heating/cooling, a specific $\theta_w(x) \neq 1$ over $0 \leq x \leq L$ defines a specific "method of heating/cooling" – whereas for uniform temperature heating/cooling that specific function is $\theta_w(x) = 1$ over $0 \leq x \leq L$. Similarly for a non-uniform wall heat-flux controlled heating/cooling, a specific $\Psi_q(x) \neq 1$ over $0 \leq x \leq L$ defines a specific "method of heating/cooling" – whereas for uniform heat-flux heating/cooling that specific function is $\Psi_q(x) = 1$ over $0 \leq x \leq L$.

The heat transfer to or from heat-exchange surfaces for the phase-change flows in Figs. 1-3 are characterized by the value of local heat transfer coefficient $h_x$ defined as:

$$q_w(x) \equiv h_x. \Delta T(x) = h_x. |T_w(x) - T_{sat}(p_0)| \quad (4)$$
where, local heat transfer coefficient $h_x$ depends on the overall flow-specifying geometry and boundary conditions. These amount to the variables of: distance "x" from the inlet, tube diameter $D_h$, inlet mass flow rate $\dot{M}_{in}$, inlet (at $x = 0$) quality $X_{in}$ (which is zero for saturated flow-boiling in Fig. 1, unity for saturated flow-condensation in Fig. 2, and $X_{in}$ where $0 < X_{in} < 1$, for flows in Figs. 3a-b), relevant fluid properties, controlling thermodynamic variables for phase-change, the "level" and – for non-uniform "method of heating/cooling" – the functions $\theta_w(x)$ or $\Psi_q(x)$ at the tube walls, and tube-orientation that yield the values of gravity components $g_x$ and $g_y$ (note $g_x \equiv 0$ for the horizontal flow configurations of interest in Figs. 1-3). For known wall temperatures $T_w(x)$, or known $\Delta T(x)$ ($\equiv \Delta T. \theta_w(x)$), this means that – for traditional flows in Figs. 1-2 and $x$ (the distance from inlet) lying in the convective plug/slug or annular regimes – one can assume that the convective component of heat transfer coefficient $h_{x\mid cb}$ (in case of nucleation being important for annular flow-boiling) or total heat transfer coefficient $h_x$ (for flow-condensation and suppressed nucleation flow-boiling for which $h_x = h_{x\mid cb}$) has the dependence:

$$h_x \equiv h_x \left(x, G, D_h, \Delta T, \theta_w(x), \rho_L, \rho_V, \mu_L, \mu_V, C_{pl}, k_L, h_{fg}, \sigma, |g_x|, |g_y| \right)$$ (5)

Similarly, for known wall heat-flux "method of heating/cooling." Eq. (5) for the flows in Figs. 1-2 is replaced by:

$$h_x \equiv h_x \left(x, G, D_h, \overline{q}_w, \Psi_q(x), \rho_L, \rho_V, \mu_L, \mu_V, C_{pl}, k_L, h_{fg}, \sigma, |g_x|, |g_y| \right)$$ (6)

Heat transfer coefficient (just the convective component $h_{x\mid cb}$ or total $h_x$) is typically non-dimensionalized as Nusselt number $Nu_x$ and is given below:

$$Nu_x \equiv h_x \frac{D_h}{k_L}$$ (7)

At times, for channel flows, the channel height “h” is used as the characteristic length scale (Naik and Narain 2016; Naik et al. 2016; Ranga Prasad et al. 2016) instead of the hydraulic diameter $D_h$. Therefore, under Eq. (1), for channel flows with only bottom-wall heating/cooling – as in Naik et al. (2016) and Ranga Prasad et al. (2016) – $D_h = 4h$.

Using the $Nu_x$ definition in Eq. (7), it is easily seen that the functional dependencies in Eqs. (5) – (6) can be non-dimensionalized, with the help of Pi-Theorem (White 2003), in the following forms:

$$Nu_x \equiv Nu_{x\mid conv} \left( \hat{x}, Re_T, Ja, Pr_L, We, Fr_x^{-2}, Fr_y^{-2}, \frac{\rho_V}{\rho_L}, \frac{\mu_V}{\mu_L}, \theta_w(x) \right)$$ (8)
and

\[ \text{Nu}_x \equiv \text{Nu}_{x|\text{conv}} \left( \hat{x}, \text{Re}_T, \text{Bl}, \text{Pr}_L, \text{We}, \text{Fr}_x^{-2}, \text{Fr}_y^{-2}, \frac{\rho_v}{\rho_L}, \frac{\mu_v}{\mu_L}, \Psi_q(x) \right) \]  \quad (9)

In Eqs. (8) and (9), \( \hat{x} \equiv x/D_h \), \( \text{Re}_T \equiv GD_h/\mu_v \), \( \text{Ja} \equiv C_pL\Delta T/h_{fg} \), \( \text{Pr}_L \equiv \mu_L C_pL/k_L \), \( \text{Fr}_x^{-2} \equiv |g_x|D_h/U^2 \), \( \text{Fr}_y^{-2} \equiv |g_y|D_h/U^2 \), \( \text{We} \equiv \rho_L U^2 D_h/\sigma \) and \( \text{Bl} \equiv \tilde{q}_w/G_{fg} \). The total Reynolds number \( \text{Re}_T \) simply represents non-dimensional value of mass-flux \( G \). Non-dimensional surface tension parameter Weber number (\( \text{We} \)) may also influence steady annular flows in Fig. 3, particularly if tube diameters are small.

Furthermore, for annular flows in Figs. 3a-b, the argument list on the right sides of Eqs. (8) and (9) will have direct additional dependence on inlet quality \( X_{in} \) (Ranga Prasad et al. 2016). Also, it should be noted that for annular flow-boiling with nucleation, Eqs. (8) and (9) only represent the structure for convective boiling component of the Nusselt number \( \text{Nu}_{x|\text{cb}} \) (Ranga Prasad et al. 2016).

The condensing/boiling flows of interest typically have a one-to-one correspondence because of monotonically decreasing/increasing values of quality \( X \) with distance \( x \) (or non-dimensional distance \( \hat{x} \)). It is therefore possible and common to replace the non-dimensional distance \( \hat{x} \equiv x/D_h \) in Eqs. (8) and (9) by the local quality \( X(\hat{x}) \) defined as:

\[ X(\hat{x}) = \frac{\dot{M}_v(\hat{x})}{\dot{M}_{in}} \]  \quad (10)

One of several reasons for replacing distance \( \hat{x} \) with quality \( X \) is the expectation that its use, in place of "\( \hat{x} \)", will allow more convenient and meaningful characterization of flow-regime transition boundaries because of similarities found among different flow-regimes encountered in phase-change flows (see Figs. 1-2) and those encountered for different realizations of adiabatic flows (Ghiaasiaan 2007; Thome 2004), where a uniform quality \( X(x) = X \) over \( 0 \leq x \leq L \) retains a value between zero and one. Another advantage of using \( X \) in place of \( \hat{x} \) – for developing the \( \text{Nu}_x \) correlations in Eqs. (8) - (9) – is that it is likely to significantly weaken the influence of different functions \( \theta_w(\hat{x}) \) or \( \Psi_q(\hat{x}) \) that characterize the effects of spatially non-uniform methods of heating/cooling. For a certain class of annular flow-boiling, this fact has been verified by Ranga Prasad et al. (2016). As a result of the above, it is a common practice to characterize heat-transfer
for phase-change flow processes in horizontal tube configurations of Figs. 1-2 (where $g_x = 0$ and effects of $g_y$ are often negligible within the plug-slug or annular regimes of interest but may affect the flow-regime transition boundaries) by seeking $N_u_x$ correlation for known wall temperature heating/cooling cases – using experiments or computations or their synthesis – in the functional dependence structure of the form (or its equivalent):

$$N_u_x \equiv N_u_x \left( X, J_a, R_e_T, Pr_L, We, \frac{\rho_v}{\rho_L}, \frac{\mu_v}{\mu_L} \right)$$  \hspace{1cm} (11)

For a known heat-flux specifying the “method of heating/cooling” of horizontal tubes, the Nusselt number dependence is of the form (or its equivalent):

$$N_u_x \equiv N_u_x \left( X, Bl, R_e_T, Pr_L, We, \frac{\rho_v}{\rho_L}, \frac{\mu_v}{\mu_L} \right)$$  \hspace{1cm} (12)

The non-dimensional arguments in Eqs. (8) – (9) or Eqs. (11) – (12) simply represent a broad structure. While developing correlation(s), one may choose to simplify the dependences further or optimize for different equivalent combinations. For example, one may choose to limit mass-flux ($G$ or speed $U \equiv G/\rho_v$) effects to Reynolds number $R_e_T$ and replace Weber number $We$ and Froude numbers ($F_r^{-2}$ and $F_r^{-2}$) respectively by Suratman number $Su \equiv \sigma \rho_v D_h \mu_v^2$ and a pair of non-dimensional gravity numbers ($g_{nd-x}$ and $g_{nd-y}$ where $g_{nd-x(y)} \equiv g_x(y) \rho_v^2 D_h^3 \mu_v^3$).

Furthermore, for thin film annular flows of Fig. 3, as established by several analyses (Narain et al. 2015; Ranga Prasad et al. 2016), independent dependences on $J_a$ and $Pr_L$ on the right sides of Eqs (11) - (12) can be simplified by a single parameter dependence on “$J_a/Pr_L$.” Thus, for thin steady annular flows of Fig. 3 restricted to mm-scale channels (curvature of the interface is negligible), effect of the surface tension parameters ($We$ or $Su$) can be assumed to be negligible and one can replace Eqs. (11) – (12) by their respective and further simplified forms – with the addition of inlet quality $X_{in}$ (which is a variable only for innovative flow-boiling, as it equals 1 for innovative flow-condensation) – given below:

$$N_u_x = N_u_x \left( X, J_a, R_e_T, \frac{J_a}{Pr_L}, \frac{\rho_v}{\rho_L}, \frac{\mu_v}{\mu_L} \right)$$  \hspace{1cm} (13)

and

$$N_u_x = N_u_x \left( X, X_{in}, Bl, J_a, R_e_T, Pr_L, \frac{\rho_v}{\rho_L}, \frac{\mu_v}{\mu_L} \right)$$  \hspace{1cm} (14)
Again for annular flow-boiling with nucleation, Eq. (13) – (14) only specify the structure for \( \text{Nu}_{x|cb} \) – i.e., convective component of HTC (see Ranga Prasad et al. 2016). The goal is to carefully obtain correlations of the forms (or their equivalent) given in Eqs. (8) – (9) or (11) – (12) or (13) – (14) over a well-defined parameter space by: experiments, or accurate modeling/simulation techniques, or a synthesis of the two.

If \( \text{Nu}_x \) correlations in the above form (or equivalent) could be reliably obtained from experimental data on flow-condensers and, by incorporating a few additional nucleation specifying parameters in case of flow-boiling with significant nucleation, such experimental correlations would already be consistent – at least approximately – with all the relevant laws of nature (mass, momentum, energy, etc.).

### 2.2 Other indirect variables and their influence on aforementioned key variables

**Laminar or turbulent nature of flows:** The total Reynolds number \( \text{Re}_T \) cannot shed much light on the expected laminar or turbulent nature of the different phases in annular flows. Clearly laminar or turbulent nature of the flows impact heat transfer correlations for \( \text{Nu}_x \) and most other variables of interest. For this assessment, one often uses "local" values of liquid and vapor Reynolds numbers \( \text{Re}_L(x) \) and \( \text{Re}_V(x) \) defined as:

\[
\text{Re}_L \equiv \frac{G \cdot (1 - X(x)) \cdot D_h}{\mu_L} \tag{15}
\]

and

\[
\text{Re}_V \equiv \frac{G \cdot X(x) \cdot D_h}{\mu_v} \tag{16}
\]

For separated annular flows, \( \text{Re}_L \) (or \( \text{Re}_V \)) < 2000 continue to indicate laminar nature of the flow in that phase and \( \text{Re}_L \) (or \( \text{Re}_V \)) \( \gg \) 2000 continue to indicate turbulent nature of flow in that phase. In fact for thin film flows, for which \( \Delta \ll h \), \( \text{Re}_\Delta (\equiv G \cdot (1 - X(x)) \cdot 4\Delta/\mu_L) < 1800 \) is the more appropriate thin film flow laminarity criteria (Bergman et al. 2011) which is automatically satisfied whenever \( \text{Re}_L < 2000 \).

**Flow-regime transition criteria:** The instabilities that lead to transition between one flow-regime to another often need to be characterized with the help of intermediate variables which leads to a "critical" value, or a range of critical values of quality \( X \) (i.e. \( X_{c_{ri} \rightarrow (i \pm 1)} \) or \( X_{c_{r-Li} \rightarrow (i \pm 1)} \leq X_{cr} \leq X_{c_{r-Hi} \rightarrow (i \pm 1)} \)) that mark a transition between two adjacent regimes – viz. "i" to "i±1" – as quality
X sufficiently increases or decreases in Figs. 1 and 2. Such transitions clearly influence the nature of the \( \text{Nu}_x \) correlation function and most variables of interest.

For the annular thin-film flows in Fig. 3, the significant non-dimensional parameters being the ones listed on the right-sides of Eqs. (13) – (14), one expects that the transition criteria between "regime-\( i \equiv \text{regime - annular} \)" and "regime - \( |i\pm1| \equiv \text{regime - plug/slug} \)" needs characterizations for known temperature or known heat-flux ways of specifying the "methods of heating/cooling."

This can, in principle, be obtained in the following forms:

\[
X_{cr|i\rightarrow(i\pm1)} = X_{cr|i\rightarrow(i\pm1)} \left( \frac{Ja}{Pr_L}, Re_T, \frac{\rho_v}{\rho_L}, \frac{\mu_v}{\mu_L} \right) \tag{17}
\]

and

\[
X_{cr|i\rightarrow(i\pm1)} = X_{cr|i\rightarrow(i\pm1)} \left( Bl, Re_T, \frac{\rho_v}{\rho_L}, \frac{\mu_v}{\mu_L} \right) \tag{18}
\]

Clearly characterizations as in Eqs. (17) – (18) will also depend on the laminar or turbulent nature of the flows associated with the separate phases in the annular regime.

**Film thickness and void fraction correlations:** Heat transfer rates obtained from \( \text{Nu}_x \) correlations should, clearly, also depend on the thinness of the liquid films adjacent to the heat-exchange surfaces (e.g. the liquid film thickness \( \Delta(x) \) in the annular flows of Fig. 3 or liquid thickness near the heat-exchange surface in the plug-slug regimes of Figs. 1-2). For most steady annular regimes of interest here, an estimate of film thickness can be obtained from measurements based experimental correlations for void fraction \( \epsilon \), whose definition and expected dependences for flows of interest are of the type:

\[
\epsilon \equiv \frac{A_G}{A} = \epsilon \left( X, \frac{\rho_v}{\rho_L}, \frac{\mu_v}{\mu_L}, Re_T, \text{other parameters} \right) \tag{19}
\]

**Pressure-drop and friction-factors:** Clearly, for annular flows in Fig. 3 and plug-slug flows in Figs. 1-2, representative overall pressure gradients \( \left( -\frac{\partial p}{\partial x}\right|_{\text{total}} \) are significantly coupled to the flow-regimes and are also related to the local flow quality \( X(x) \). For annular flows in Fig. 3, the pressure gradients are also directly related to interfacial shear and liquid film thickness \( \Delta(x) \) – which, in turn, also control heat transfer rates being sought through \( \text{Nu}_x \) correlations of the type given by
Eqs. (13) – (14). The overview in the next section outlines the importance of such correlations to many Nu\textsubscript{x} correlation in the literature (Ghiaasiaan 2007; Thome 2004).

**CHF Instabilities:** There are different kinds of critical heat-flux (CHF) instability mechanisms (Das et al. 2012; Qu and Mudawar 2004) for flow-boiling. All mechanisms correspond to the fact that it is possible to raise the "level" of heating to such sustained high values that a stable, steady realization of flow boiling – as depicted in Fig. 1 or Fig. 3a – is no longer possible. At such levels, different instability mechanisms are triggered, all leading to a situation where the heated-surface of the tube/channel starts experiencing expanding dry patches (vapor exposed directly to adsorbed layers – see discussions in section 3) that would, in time, lead to unsteady bulk flows and runaway rise in wall temperatures $T_w(x, t)$. The runaway rise in wall temperatures may lead to eventual melting and other disastrous consequences. Clearly one needs to seek Nu\textsubscript{x} correlations of the type given by Eqs. (13) – (14) provided that the associated wall heat-flux $q_w(x)$ is below a pertinent threshold value, denoted as $q_{CHF|relevant}$.

### 2.3 Segmented flow-regime dependent Nu\textsubscript{x} correlations

The above discussions (in section 2.2) on other indirect variables influencing key variables of Nu\textsubscript{x} correlations suggest that they indirectly influence the very form of the Nu\textsubscript{x} function – in ways that the forms differ from one flow-regime to another. This suggests that it may be more convenient to obtain the sought-for and more accurate Nu\textsubscript{x} correlations in Eqs. (8) – (9) by an approach that restricts the correlations to one regime at a time. That is, if “regime - i” denotes a particular flow regime (such as bubbly flow regime, plug-slug regime, annular regime, etc.), and its neighboring upstream and downstream regimes in Figs. 1-2 are respectively marked as "regime - (i – 1)" and "regime - (i+1)" – with corresponding flow regime transition criteria in terms of quality (see Eqs. (17) - (18)) denoted as $X_{cr|(i-1)\rightarrow i}$ and $X_{cr|i\rightarrow (i+1)}$ – one could seek correlations of the type:

\[
\text{Nu}_x = \text{Nu}_{x|\text{regime-}i}(X, \text{Re}_T, \frac{\text{Ja}}{\text{Pr}_L}, \frac{\rho_v}{\rho_L}, \frac{\mu_v}{\mu_L}, ...)
\]

or,

\[
\text{Nu}_x = \text{Nu}_{x|\text{regime-}i}(X, BI, \text{Re}_T, \frac{\rho_v}{\rho_L}, \frac{\mu_v}{\mu_L}, ...)
\]

where, $X(\bar{x})$ is in the range

\[
X_{cr|(i-1)\rightarrow i} \leq X(\bar{x}) \leq X_{cr|i\rightarrow (i+1)}
\]
2.4 Underlying **one-dimensional modeling** approach to obtain spatial $x$–variations of flow-variables that are known or correlated in terms of quality $X$ and other parameters

For the flow-boiling and flow-condensation realizations in Figs. 1-3, **one-dimensional Energy Balance** can be applied to the control volume between any two arbitrary locations "$x$" and "$x + \Delta x$" (see Fig. 4).

![Fig 4: A schematic of a control-volume between "$x$" and "$x + \Delta x$". The heat-flux arrows, as shown, are positive for boiling. The reversed direction negative values are for flow-condensation.](image)

It is easy to see that, in the limit of $\Delta x \to 0$, the energy balance yields:

$$\frac{dX}{dx} \approx \pm \frac{|q_w(x)|}{M_{in} h_{fg}(p_0)} = \pm \frac{h_x |T_w(x) - T_{sat}(p_0)|}{M_{in} h_{fg}}$$  \hspace{1cm} (22)

Using the definitions given earlier for the relevant non-dimensional variables and numbers, Eq. (22) can be non-dimensionalized as:

$$\frac{dX(\hat{x})}{d\hat{x}} = \pm 4. \frac{Nu_x}{Pr_L} \frac{Ja}{Re_T} \frac{1}{\mu_v} \mu_L \theta_w(x)$$  \hspace{1cm} (23)

for known wall temperatures specifying the "method of heating/cooling." For the known heat-flux values specifying "method of heating/cooling," Eq. (22) is non-dimensionalized as:

$$\frac{dX(\hat{x})}{d\hat{x}} = \pm 4. Bl. \Psi_q(x)$$  \hspace{1cm} (24)

The "+" and "−" signs in Eqs. (22) – (24) are, respectively, for flow-boiling and flow-condensation.

For known wall temperatures specifying the "method of heating/cooling," the non-linear *ordinary differential equation* (ODE) in Eq. (23) may be solved over $0 \leq \hat{x} \leq L/D_h$, in conjunction with a
reliable $\text{Nu}_x$ correlation in its overall form (covering all flow-regimes of saturated flow-boiling or flow-condensation in Figs. 1-2) given in the form of Eq. 11 (or its equivalent) and subject to initial condition.

$$X(0) = \begin{cases} 
0, & \text{for saturated flow – boiling} \\
1, & \text{for saturated flow – condensation}
\end{cases} \quad (25)$$

Alternatively, for the known wall temperature "method of heating/cooling" cases, the related non-linear ODE in Eq. (23) may be solved over $x_i^* \leq x \leq x_{i+1}^*$ alone, distances associated with flow-regime "i" and the flow-regime specific $\text{Nu}_x$ correlation as discussed through Eqs. (20) – (21). The initial conditions for these flow-regime specific solutions of $X(x)$, for a guessed value of $x_i^*$ then becomes:

$$X(x_i^*) = X_{cr(i−1)→i} \quad (26)$$

where the $X_{cr(i−1)→i}$ correlation is available from Eq. (18) and the guessed value of $x_i^*$ remains unknown until $X(x)$ is known from $x = 0$, though prior flow-regimes up to flow conditions at $x = x_i^*$.

Current knowledge of segmented and flow-regime specific $\text{Nu}_x$ correlations may be approximate and reasonable for annular flow realizations but the knowledge of the flow-regime boundaries, as sought in non-dimensional forms (such as Eqs. (17) – (18)), is poor for the flows in Fig. 3. The flow-regime transition boundaries, which occur for flows in Figs. 1-2, are also poorly known.

For a known heat-flux value specifying the "method of heating/cooling," the ODE in Eq. (24) yields linear $X(\hat{x})$ variation if it is solved for a uniform heat-flux prescription (i.e. for $\Psi_q(x) \approx 1$) – over $0 \leq \hat{x} \leq L/D_h$ subject to the initial conditions in Eq. (25). If the overall $\text{Nu}_x$ correlation in Eq. (12) (or its equivalent) is known, it allows evaluation of $h_x$ under given values of $\overline{q}_w^*$, Bl, and $X(\hat{x})$ variation. Use of this $h_x$ in the defining relationship of Eq. (4) then yields the temperatures $T_w(x)$ over $0 \leq x \leq L$.

Alternatively for known heat-flux values specifying the "method of heating/cooling" cases and available knowledge of $\text{Nu}_x$ being a flow-regime specific correlation (as in Eq. (20)), the solution of the ODE in Eq. (24) over $x_i^* \leq x \leq x_{i+1}^*$ may be obtained – and $X(\hat{x})$ will remain linear in $\hat{x}$ for $\Psi_q(x) \approx 1$. With the help of the initial condition in Eq. (25) and "regime - i" specific $\text{Nu}_x$ correlation in Eq. (20), $h_x$ can be evaluated for given values of $\overline{q}_w^*$, Bl, and $X(\hat{x})$ variation. Use of
this \( h_x \) in the defining relationship of Eq. (4) then yields temperatures \( T_w(x) \) over the \( x \)-locations \( x^*_i \leq x \leq x^*_i + 1 \) for regime-\( i \). The physical location of this regime from \( x = 0 \) is not possible until one also knows more about the flow conditions at \( x = x^*_i \) or over \( 0 \leq x \leq x^*_i \).

### 3. Overview of available correlation for direct and indirect variables of interest

The correlations that are relevant to the focus of this review – annular flow-boiling and flow-condensation – correspond typically to laminar liquid film flows, laminar or turbulent vapor flows, and negligible entrainment rates. These correlations (say for heat transfer coefficient and pressure-drop) are typically related to key variables (dimensional or non-dimensional) that define a specific flow realization, namely: hydraulic diameter \( D_h \), mass-flux \( G \), relevant fluid properties, imposed heat-flux \( \bar{q}_w \) or wall temperature \( \bar{T}_w \), length \( L \), quality \( X \) and inlet quality \( X_{in} \) (for annular flow-boiling only) in the horizontal tube configuration of the flows in Fig. 3. Though horizontal tube considerations are sufficient for most inner diameters \( D_h \) and mass-flux \( G \) values of interest to this review, downward tilted \( (g_x > 0) \) flow-boilers and flow-condensers for some macro-scale diameters \( (5 \text{ mm} \leq D_h \leq 15 \text{ mm}) \) and low mean mass-flux \( (G = 10 \text{ to } 30 \text{ kg/m}^2 \cdot \text{s}) \) may also be of interest – and are briefly discussed here.

Effects of non-uniform heating/cooling methods have been discussed by Naik et al. (2016), Naik and Narain (2016), and Ranga Prasad et al. (2016) from a theoretical point of view. Regardless of whether the correlations assume knowledge of wall temperature \( T_w(x) \) or heat flux \( q_w''(x) \) prescriptions, one can – for convective boiling and flow-condensation – establish (with the help of CFD simulations) equivalence between the two (see Naik et al. 2016 and Ranga Prasad et al. 2016). Next relevant correlations for heat transfer coefficient \( h_x \), flow-regime maps, pressure-drop, and void-fraction are reviewed. Furthermore, relevant CHF issues (for flow-boiling) related to the annular flows in Fig. 3 are discussed.

### 3.1 Local heat transfer coefficient \( h_x \) from Nusselt Number \( \text{Nu}_x \) correlations

#### 3.1.1 Flow-boiling

Known results from the flow-boiling experiments that are cited here, typically use the fact that the mean wall heat-flux \( \bar{q}_w'' \) and key flow defining variables are known and \( \text{Nu}_x \), for traditional boiling operations, can be correlated in the structural form (or its equivalent) as indicated in Eq. (12).
Several local heat transfer coefficient correlations covering a range of experiments (in mini/micro-channels for both single and multi-channel configurations, as given in Table 1) have been recently considered by Kim and Mudawar (2013c) and an order of magnitude curve fit for flow-boiling – covering all the flow-regimes in Fig. 1 – has been proposed. These experiments include the annular regime results, which are of particular interest to this review.

Table 1: Previous saturated flow-boiling heat transfer correlations considered by Kim and Mudawar (2013c)

<table>
<thead>
<tr>
<th>Authors</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Recommended for macro-channels</strong></td>
<td></td>
</tr>
<tr>
<td>Cooper (1984)a</td>
<td>6000 data points for nucleate pool boiling</td>
</tr>
<tr>
<td>Gungor and Winterton (1986)</td>
<td>(D = 2.95-32 \text{ mm, water, R11, R12, R113, R114, R22, ethylene glycol, 4300 data points})</td>
</tr>
<tr>
<td>Shah (1982)</td>
<td>Experiments are mostly in horizontal tubes of circular or rectangular cross-sections. (D = 6-25.4 \text{ mm, water, R11, R12, R113, cyclohexane, 780 data points})</td>
</tr>
<tr>
<td><strong>Recommended for mini/micro-channels</strong></td>
<td></td>
</tr>
<tr>
<td>Agostini and Bontemps (2005)</td>
<td>(D_h = 2.01 \text{ mm, 11 parallel channels, R134a})</td>
</tr>
<tr>
<td>Bertsch et al. (2009)</td>
<td>(D_h = 0.16 – 3.1 \text{ m, water, refrigerants, FC-77, Nitrogen, 3899 data points})</td>
</tr>
<tr>
<td>Ducoulombier et al. (2011)</td>
<td>(D = 0.529 \text{ mm, CO}_2)</td>
</tr>
<tr>
<td>Lazarek and Black (1982)</td>
<td>(D = 3.15 \text{ mm, R113, nucleate boiling dominant})</td>
</tr>
<tr>
<td>Li and Wu (2010)</td>
<td>(D_h = 0.16 – 3.1 \text{ m, water, refrigerants, FC-72, ethanol, propane, CO}_2, 3744 data points)</td>
</tr>
<tr>
<td>Oh and Son (2011)</td>
<td>(D = 1.77, 3.36, 5.35 \text{ mm, R134a, R22})</td>
</tr>
<tr>
<td>Tran et al. (1996)</td>
<td>(D = 2.46, 2.92 \text{ mm, } D_h = 2.40 \text{ mm, R12, R113, nucleate boiling dominant})</td>
</tr>
<tr>
<td>Warrier et al. (2002)</td>
<td>(D_h = 0.75 \text{ mm, 5 parallel channels, FC-84})</td>
</tr>
<tr>
<td>Yu et al. (2002)</td>
<td>(D = 2.98 \text{ mm, water, ethylene glycol, nucleate boiling dominant})</td>
</tr>
</tbody>
</table>

aThe Cooper (1984) correlation was developed for nucleate pool boiling
The “more general” HTC correlation of Kim and Mudawar (2013c) – given in Eqs. (27) – (30) below – has a built in, but ad hoc, breakup of total HTC into its nucleate and convective components.

\[ h_x = \left( h_{x|nb}^2 + h_{x|cb}^2 \right)^{0.5} \]  

where

\[ h_{x|nb} \equiv 2345 \left( \frac{Bl}{Pr} \right)^{0.7} P_{Re}^{0.38} (1 - X)^{0.51} \left( 0.023 Re_L^{0.8} Pr_L^{0.4} \frac{k_L}{D_h} \right) \]  

and

\[ h_{x|cb} \equiv 5.2 \left( \frac{Bl}{Pr} \right)^{0.08} We_L^{0.54} + 3.5 \left( \frac{1}{X_{tt}} \right)^{0.94} \left( \frac{V}{\mathcal{P}_L} \right)^{0.25} \left( 0.023 Re_L^{0.8} Pr_L^{0.4} \frac{k_L}{D_h} \right) \]

The parameters in the above definitions of \( h_x \) are:

\[ Bl \equiv \frac{q_w}{G \cdot h_{fg}}, \quad Pr = \frac{p_o}{p_{cr}}, \quad Re_L = \frac{G (1 - X) D_h}{\mu_L}, \quad Re_{L0} = \frac{GD_h}{\mu_I}, \quad Pr_L = \frac{\rho_L C_P L}{k_L}, \quad We_L = \frac{G^2 D_h}{\rho_L \sigma}, \quad X_{tt} \equiv \left( \frac{\mu_L}{\mu_V} \right)^{0.1} \left( \frac{1 - X}{X} \right)^{0.9} \left( \frac{\rho_V}{\rho_L} \right)^{0.5} \]

where, \( P_F \) is the wetted perimeter (in case a rectangular cross-section channel is not wetted on all its periphery) and \( P_H \) is the heated perimeter (in case a rectangular cross-section is not heated on all its periphery). This is also the perimeter where \( q_w''(x) \) is replaced by \( q_w''(x) \).

The correlation in Eqs. (27) – (30) covers all the saturated flow-boiling regimes depicted in Fig. 1 – i.e. as the quality \( X \) increases and the flow goes from nucleate boiling (nb) dominated to the convective boiling (cb) dominated regimes. Use of the Martinelli parameter \( X_{tt} \) in Eq. (29) is not indicative of both phases being turbulent – it is simply a correlation parameter. The uncertainty in the predictions from a correlation, such as the one in Eqs. (27) – (30), can be high as it carries many experimental and conceptual uncertainties. Some of the significant sources of uncertainty are associated with:

(i) Replacing \( X(x) = \dot{M}_v(x)/\dot{M}_{in} \) inferred from the expression of \( X(x) \equiv X_{th}(x) \) where \( X_{th}(x) \) is a thermodynamic vapor quality obtained from: experimentally assessing (with some uncertainty) an “\( x = 0 \)” location where the fluid is at close to saturation conditions and then applying energy balance to the flow-boiling control-volume, between the identified “\( x = 0 \)”
location and one at an “x > 0” location, with knowledge of the experimentally measured values of heat input ($q_{W\text{ }}(x) = \int_{0}^{x} q_{W}(x) \text{ d}x$) between the locations.

(ii) Uncertainties in measuring $\Delta T(\equiv |T_{W} - T_{\text{sat}(p_{0})}|)$ and $q_{W}(x)$ (due to thermocouple and other instruments’ accuracy limitations – more so with older experiments with less accurate sensors) and then evaluating $h_{x} \equiv q_{W}(x)/\Delta T$.

(iii) Uncertainties associated with the current practice of combining data obtained for different cross-sectional geometries – e.g. from channel flows (i.e. high aspect ratio rectangular cross-sections) not influenced by curvature effects and surface-tension with those obtained from mm-scale tubes where flows are affected by curvatures and surface-tension.

(iv) Inherent inaccuracies associated with developing a single correlation covering all flow-regimes (as in Eqs. (11) – (12)) as opposed to developing segmented correlations specific to different flow-regimes (as in Eq. (20)).

A rather ad hoc splitting of total HTC $h_{x}$ in its convective ($h_{x|cb}$) and nucleate ($h_{x|nb}$) boiling parts (see Ranga Prasad et al. 2016).

Despite the aforementioned uncertainties and weaknesses, correlations – such as the one in Eqs. (27) – (30) – can provide useful order of magnitude estimates until more accurate flow-regime specific correlations are developed in appropriate ranges of parameter space of interest to the user. For design purposes, as in the sample example of section 4, one can use a range of estimates based on the results in Eqs. (27) – (30). A crude estimate of uncertainty of the correlation in Eqs. (27) – (30) is:

$$0.5h_{x|\text{Table}-2} \leq h_{x} \leq 2h_{x|\text{Table}-2}$$

Correlation of Kim and Mudawar (2013c) in Eqs. (27) – (30) also covers annular flows – involving both laminar and turbulent flows of the vapor and the liquid phases. Direct numerical simulation (DNS) approach (see section 5 for further discussions) have been recently used by Ranga Prasad et al. (2016) to propose local heat transfer coefficient ($h_{x|cb}$) correlations for annular boiling cases (Fig. 3a) for sufficiently thin liquid film flows that remain laminar. Such $h_{x|cb}$ values can better be combined with experimental $h_{x}$ values for annular flow-boiling to estimate nucleate boiling contribution $h_{x|nb}$ (when they are present).
A sample correlation for low imposed heat-flux $q_w(x)$ and mass-flux $G$ values is obtained by CFD solutions and their correlation is presented in Ranga Prasad et al. (2016) for channel flows (of gap "h" and hydraulic diameter $D_h = 4h$) – with CFD employing laminar liquid and laminar vapor assumptions, but the resulting heat transfer correlations (given in Eq. (32) below) continuing to apply to moderately turbulent vapor phases as well ($Re_V < 40000$). The correlation provided in Ranga Prasad et al. (2016) is for known wall temperature specification of "method of heating" with considered/specific parameter-space corresponding to flow situation in Table 2.

$$\frac{h_{x|cb} \cdot D_h}{k_L} = 0.413 \cdot X^{1.61} \cdot X_{in}^{0.128} \cdot Re_T^{-0.0384} \cdot \left( \frac{Ja}{Pr_L} \right)^{0.0595} \cdot \left( \frac{\rho_L}{\rho_V} \right)^{-0.3999} \cdot \left( \frac{\mu_L}{\mu_V} \right)^{0.454}$$

(32)

where, $0.5 \leq X_{in} \leq 0.86$, $2466 \leq Re_T \leq 39524$, $0.0048 \leq Ja/Pr_L \leq 0.0424$, $0.0046 \leq \rho_V/\rho_L \leq 0.0097$, $0.0216 \leq \mu_V/\mu_L \leq 0.0295$.

<table>
<thead>
<tr>
<th>Table 2: Ranges of raw variables and fluid flow conditions considered for the development of the correlation given in Eq. (32)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Working fluids</td>
</tr>
<tr>
<td>Inlet pressure, $p_0$ (kPa)</td>
</tr>
<tr>
<td>Channel Height, $h$ (mm)</td>
</tr>
<tr>
<td>Mass flux, $G$ ($\equiv \rho_U$) (kg/m$^2$/s)</td>
</tr>
<tr>
<td>Transverse Gravity, $g_y$ (m/s$^2$)</td>
</tr>
<tr>
<td>Average Inlet Vapor Speed, $U$ (m/s)</td>
</tr>
<tr>
<td>Temperature Difference, $\Delta T$ (°C)</td>
</tr>
</tbody>
</table>

Correlations such as Eq. (32) can also be developed with the help of simulations for low heat-flux $q_w(x)$ and mass-flux $G$ values for known heat-flux specification of “method of heating.” These issues, along with methodologies for obtaining $h_{x|cb}$ for a higher $q_w(x)$ and $G$ values – typically associated with laminar liquid and turbulent vapor flows in Fig. 3 – are also discussed in Ranga Prasad et al. (2016). Although vapor flows can be laminar or turbulent and liquid film flows laminar, the CFD/DNS approach to yield $h_{x|cb}$ for a higher range of $G$ and other parameters can be supplemented with experimental measure of total HTC – in the presence of significant nucleate
boiling contributions $h_{x|nb}$ through relations as in Eq. (33). Thus, following the superposition approach in Eq. (33), one can also propose annular flow-boiling correlations in the presence of nucleate boiling. Such proposals are being studied for compatibility with experiments dealing with the flow in Fig. 3a (Gorigtrattanagul 2017; Sepahyar 2018) and are expected to be in one of the following approximate forms:

$$h_x = \left(\left(h_{x|nb}\right)^n + \left(h_{x|cb}\right)^n\right)^{1/n}, n = 1, 2, 3, ... \quad (33)$$

where $n = 1$ is preferred if $h_{x|cb}$ is given by the approaches similar to the one leading to Eq. (32).

In this approach $h_{x|nb}$ (with a suppression factor $S < 1$) may be similar to the one used in Chen (1966), Kenning and Cooper (1989), and Gungor and Winterton (1986) correlations for vertical up, down, and horizontal tube flow-boiling. These authors’ use of $h_{x|nb}$ is related to nucleate boiling correlation of Cooper (1984) – which replaces the earlier Forster and Zuber (1955) correlation – and is given by:

$$h_{x|nb} = 55 \ast P_R^{0.12} \ast (-\log P_R)^{-0.55} M^{-0.5} \left(\frac{q_w}{q^*}\right)^{0.67} \quad (34)$$

where, $P_R$ is the reduced pressure as defined in Eq. (30) and $M$ is the molecular weight of the working fluid.

### 3.1.2 Flow-condensation

Known results from flow-condensation experiments that are cited here assume that the mean wall temperature $T_w$ along with some key problem defining variables are known and $Nu_x$ values can be correlated in the structural form (or its equivalent) indicated in Eq. (11).

Table 3 shows several "local" heat transfer coefficient $h_x$ correlations (that included consideration of annular regime in the flow-condensation experiments from which data were used ) considered by Kim and Mudawar (2013a) before they proposed their own order of magnitude curve-fit correlation given below in Eqs. (35) – (38).

$$Nu_x \equiv \frac{h_x D_h}{k_L} = 0.048 Re_L^{0.69} Pr_L^{0.34} \left(\frac{\Phi_g}{\bar{X}_{tt}}\right) \quad (35)$$

where,

$\bar{X}_{tt}$, turbulent - turbulent Lockhart - Martinelli parameter is used merely as a known function and defined as:
The parameter $\Phi_g$ in Eq. (35) is the two-phase multiplier defined as:

$$\Phi_g^2 = 1 + CX_m + X_m^2$$  \hspace{1cm} (37)

where $C$ is a Lockhart – Martinelli coefficient defined in Kim and Mudawar (2013a) and also in Table 5 of this review, and $X_m$ is a Lockhart – Martinelli parameter defined as:

$$X_m = \left(\frac{(dp/dx)_L}{(dp/dx)_V}\right)^{1/2}$$

where, $(dp/dx)_L$ and $(dp/dx)_V$ represent the frictional pressure gradients of liquid and vapor phases respectively flowing alone in the pipe and are computed using the following equations:

$$\frac{(dp)}{(dx)}_L \equiv -2f_L G^2(1-X)^2 \quad \text{Re}_L \equiv \frac{G(1-X)D}{\mu_L}$$

$$\frac{(dp)}{(dx)}_V \equiv -2f_V G^2(X)^2 \quad \text{Re}_V \equiv \frac{G(X)D}{\mu_V}$$

$$f_L = B\text{Re}_L^{-n} \quad G \equiv \frac{4M}{\pi D_h^2}$$

$$f_V = B\text{Re}_V^{-n}$$

The friction factor $f_L$ and $f_V$ are defined as above for vapor and liquid phases, with laminar flows’ ($\text{Re}_{L/V} < 2000$) values of $B = 16$ and $n = 1$ and turbulent flows’ ($\text{Re}_{L/V} > 2000$) values of $B = 0.0079$ and $n = 0.25$.

**Table 3:** Previous annular flow-condensation heat transfer correlations considered by Kim and Mudawar (2013a) involving experiments employing macro and mini/micro-channels

<table>
<thead>
<tr>
<th>Author(s)</th>
<th>Remarks*</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Recommended for macro-channels</strong></td>
<td></td>
</tr>
<tr>
<td>Akers and Rosson (1960)</td>
<td>$D = 19.05$ mm R12, propane</td>
</tr>
<tr>
<td>Cavallini and Zecchin (1974)</td>
<td>$R12, R22, R113, 7000 \leq \text{Re}_{LO} \leq 53000$</td>
</tr>
<tr>
<td>Dobson and Chato (1998)</td>
<td>$D = 3.14 – 7.04$ mm; R12, R22, R134a, R32/R125</td>
</tr>
<tr>
<td>Haraguchi et al. (1994)</td>
<td>$D = 8.4$ mm; R22, R123, R134a</td>
</tr>
<tr>
<td>Moser et al. (1998)</td>
<td>$D = 3.14 – 20$ mm; R11, R12, R125, R22, R134a, R410a</td>
</tr>
</tbody>
</table>
Shah (1982)  
D = 7 – 40 mm; water, R11, R12, R22, R113, methanol, ethanol, benzene, toluene, trichloroethylene

**Recommended for micro-channels**

Bohdal et al. (2011)  
D = 0.31 – 3.30 mm; R134a, R404a

Huang et al. (2010)  
D = 1.6, 4.18 mm; R410a, R410a/oil

Koyama et al. (2003)  
D_h = 1.46 mm; R134a; multi-channel

Park et al. (2011)  
D_h = 1.45 mm; R134a, R236fa, R1234ze (E); multi-channel

Wang et al. (2002)  
D_h = 1.46 mm; R134a; multi-channel

* Experiments deal with mostly tubes and annular flows are the dominant flow-regimes. Both circular and rectangular cross-section ducts have been considered

**Computational Fluid Dynamics (CFD)** which becomes a **Direct Numerical Simulation (DNS)** approach for laminar/laminar case has been recently used by Narain et al. (2015), Naik et al. (2016), and Naik and Narain (2016) (see section 5 and for further discussions) to propose local heat transfer coefficient \( h_x \) correlations for annular flow-condensation cases (Fig. 3b) involving sufficiently thin laminar liquid flows and low values of heat-flux \( q_w(x) \) and mass-flux \( G \).

A sample \( \text{Nu}_x \) correlation in Eq. (39) is for channel flows (of gap "h" and hydraulic diameter \( D_h = 4h \)) and applies to reasonably turbulent vapor phases \( (\text{Re}_V < 40000) \) as well – provided the known temperature specification of "method of cooling" falls within the parameter-space restrictions given for Eq. (39).

\[
\frac{h_x D_h}{k_L} \equiv \text{Nu}_x |_{D_h} = 0.02 \times (1 - X)^{-0.59} \text{Re}_T^{0.122} \left( \frac{Ja}{Pr_L} \right)^{0.3} \left( \frac{\rho_V}{\rho_L} \right)^{-0.73} \left( \frac{\mu_V}{\mu_L} \right)^{0.069} \tag{39}
\]

where, \( 3200 \leq \text{Re}_\text{in} \leq 92000, \ 0.0058 \leq Ja/Pr_L \leq 0.021, \ 0.0013 \leq \rho_V/\rho_L \leq 0.011, \ 0.012 \leq \mu_V/\mu_L \leq 0.034. \)

**Table 4**: Range of raw fluid variables and flow conditions considered for the development of the correlation given in Eq. (39)

<table>
<thead>
<tr>
<th>Working fluids</th>
<th>FC 72</th>
<th>R113</th>
<th>R113</th>
<th>R134a</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet pressure, (kPa)</td>
<td>100</td>
<td>25</td>
<td>225</td>
<td>150</td>
</tr>
<tr>
<td>Saturation Temperature (°C)</td>
<td>55.94</td>
<td>11.1</td>
<td>73.86</td>
<td>-17.15</td>
</tr>
</tbody>
</table>
Hydraulic Diameter (h = 0.001 – 0.003 m in Fig. 3b)

<table>
<thead>
<tr>
<th>Transverse Gravity, $g_y$ (m/s²)</th>
<th>4h</th>
<th>4h</th>
<th>4h</th>
<th>4h</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass flux, G (kg/m²s)</td>
<td>4.55 – 127.4</td>
<td>4.2 – 115.3</td>
<td>5.51 – 154.1</td>
<td>4.1 – 113.5</td>
</tr>
<tr>
<td>Temperature Difference, $\Delta T$ (°C)</td>
<td>2.93 – 12.30</td>
<td>8.69 – 36.50</td>
<td>2.25 – 9.45</td>
<td>3.45 – 14.45</td>
</tr>
</tbody>
</table>

It was shown by Narain et al. (2015) that, for heat-flux $q_w$ and mass-flux G cases, the channel flow DNS yields $h_x$ values much higher than the corresponding $D_h \equiv 4h$ predictions from the Kim and Mudawar correlation in Eqs. (35) – (38). The in-tube (as opposed to channel) condensing flow predictions for $h_x$ from Naik et al. (2016), Naik and Narain (2016) yield values (also see Narain et al. 2015) closer to the one in Kim and Mudawar (2013a) whereas the channel predictions are further off. This suggests that high aspect ratio associated with in-tube flows may degrade the heat transfer performance.

As discussed in section 5 of this review, the CFD/DNS approach can be extended to yield reliable $h_x$ values for a higher range of parameters (particularly mass-flux G, etc.) than those indicated in Table 4.

### 3.2 Flow-regime maps/correlations

Flow-regime maps are needed and recommendations exist (Ghiaasiaan 2007; Kim and Mudawar 2013a) for identifying flow-regimes for traditional operations of boiling (Fig. 1) and condensing flows (Fig. 2).

#### 3.2.1 Annular adiabatic cases

Flow-regimes maps also exist for adiabatic flows (Carey 1992; Ghiaasiaan 2007) which may also be useful in the inlet region, just past the splitter plates in the innovative annular operations shown in Fig. 3. This is mainly because the flow will approach adiabatic behavior before it senses, significant amounts of imposed heating/cooling (as shown in Fig. 3).

Precise non-dimensional transition maps, as suggested by correlations of the type indicated in Eqs. (17) – (18), currently do not exist – even for adiabatic two-phase flows. For the adiabatic case, following certain earlier studies (Baker 1953; Hewitt and Roberts 1969), a map employing raw variables proposed by Mandhane et al. (1974) – which uses superficial velocities of vapor and
liquid phases, but covers a significant range of the parameter-space (see Ghiaasiaan 2007) which includes conditions of interest (i.e. horizontal tube co-current cases) to this review.

### 3.2.2 Annular flow-boiling

This regime is of interest here (Fig. 3a) and it also occurs at locations further downstream of the plug-slug regime as quality $X$ increases in the traditional saturated flow-boiling cases (Fig. 1). For known uniform heat-flux values specifying the "method of heating," Harirchian and Garimella (2012) recommended the following criteria for micron-scale $D_h$:

$$\frac{\text{Bo}^{-0.5}}{\text{Re}} > 160$$

and

$$\text{Bl} \cdot \frac{X \cdot \frac{P_H}{D^2} \cdot \frac{\rho_L - \rho_V}{\rho_V}}{> 96.65 \left(\frac{\text{Bo}^{0.5} \text{Re}}{\text{Re}}\right)^{-0.258}}$$

(40)

where, $P_H$ = heated perimeter, $D \equiv \sqrt{A}$, $\text{Bo} \equiv g(\rho_L - \rho_V)D^2/\sigma$, $\text{Bl}$ is as defined in Eqs. (8) – (9), and $\text{Re} \equiv G \cdot D/\mu_L$. This means annular flows are expected over downstream distances “$x$” satisfying:

$$x > x_A \equiv 96.65 \left(\frac{\text{Bo}^{0.5} \text{Re}}{\text{Re}}\right)^{-0.258} \text{Bl}^{-1} \frac{\rho_V}{\rho_L - \rho_V} \frac{A}{P_H}$$

(41)

or quality $X$ satisfying:

$$X > X_{cr} = X(x = x_A)$$

(42)

where $X (x)$ has been obtained from the approach described in section 2.4 along with use of an appropriate $\text{Nu}_x$ correlation, such as the one in Eqs. (27) – (30), and estimates being approximate (as in Eq. (31)). The parameter ranges over which Eq. (41) is valid is given in Harirchian and Garimella (2012) but it does not include the larger $D_h$ values of interest to this review. For identifying critical transition quality for annular flow-boiling Kim and Mudawar (2013a) also recommend another criterion. This criterion is the similar to the one for flow-condensation and, therefore, is described instead in section 3.2.3. The effectiveness of these empirical correlations are limited and may provide only order of magnitude estimates. More accurate, non-linear stability based transition quality (plug-slug to annular) correlations for Eqs. (17) – (18) can also be obtained by a synthesis of CFD/DNS approach (see Ranga Prasad et al. 2016) with new experiments – but such estimates are currently limited to flow-boiling in channels and convective boiling or suppressed nucleation cases (as in Eq. (32)).
3.2.3 Annular flow-condensation

This regime is of interest here (Fig. 3b) and it also occurs upstream of the plug-slug regime, as quality \( x \) increases, in traditional flow-condenser operations (Fig. 2). For known wall temperature cases specifying "method of cooling," Kim and Mudawar (2013a) recommend the following criteria:

\[
We^* > 7.\bar{X}_{tt}^{0.2}
\]  

(43)

where \( \bar{X}_{tt} \) is as defined in Eq. (30) (or Eq. (36))

\[
We^* \equiv \frac{2.45 Re_V(x)^{0.64}}{Su_{V0}^{0.3}(1 + 1.09\bar{X}_{tt})^{0.9}} \text{ for } Re_L(x) \leq 1250
\]  

(44)

or

\[
We^* \equiv \frac{0.85 Re_V\bar{X}_{tt}^{0.157}}{Su_{V0}^{0.3}(1 + 1.09\bar{X}_{tt})^{0.09}} \left[ \frac{\mu_V}{\mu_L} \right]^{0.4} \left[ \frac{\rho_L}{\rho_V} \right]^{0.084} \text{ for } Re_L(x) > 1250
\]  

(45)

Note that in Eqs. (43) – (45), \( Su_{V0} \equiv \rho_V \sigma D_h / \mu_V^2 \) and the range of parameter space for these correlations is as given in Kim and Mudawar (2013a). By plotting the above criteria on an \( X - Re_T \) plane, the results in Eqs. (43) – (45) can be presented in the form of Eq. (17). Again, effectiveness of the above described type of correlations are expected to be limited and, at best, are meant only to provide order of magnitude estimates for tubes and rectangular cross-sections of aspect ratio near unity. For specific fluids and parameter ranges (inlet pressure, etc.), the correlations in Eqs. (43) – (45) can be compared with flow-regime maps of Coleman and Garimella (2003), etc. For low mass-fluxes \( (G) \) and channel flow-condensation in Fig. 3b (aspect ratio \( \sim 0 \)) the correlation in Eq. (43) – (45) can even be compared with accurate non-linear stability based transition quality (annular to plug-slug) correlation given by Naik et al. (2016) and Naik and Narain (2016).

The correlation proposed by Naik et al. (2016) and Naik and Narain (2016) for flow-condensation in a channel yields the distance from \( x = 0 \) (where \( X(0) = 1 \)) to the onset of plug-slug regime \( (x = x_A |_{lg}) \) in the horizontal channel flow configurations of Fig. 3b. Using \( D_h = 4h \) (instead of "h" used as characteristic length in Naik et al. (2016) and Naik and Narain (2016)), it is recalled here that (see Narain et al. (2015)): 

30
\[ \delta_{A|lg} \cong 0.735 (Re_T)^{0.85} \left( \frac{ja}{Pr_L} \right)^{-2.17} \cdot \left( \frac{\rho_V}{\rho_L} \right)^{1.03} \cdot \left( \frac{\mu_V}{\mu_L} \right)^{1.64} \]  

(46)

and

\[ X_{cr} = X(\bar{x})|_{\delta = \delta_{A|lg}} \cong 1 - 0.813 \cdot \delta_{A|lg}^{0.73} (Re_T)^{-0.64} \left( \frac{ja}{Pr_L} \right)^{1.62} \cdot \left( \frac{\rho_V}{\rho_L} \right)^{-0.33} \cdot \left( \frac{\mu_V}{\mu_L} \right)^{-0.86} \]  

(47)

The parameter ranges over which Eqs. (46) – (47) are valid are the same as the ones given for Eq. (39). These low heat and mass-flux parameter ranges are more limited than the ones given in Kim and Mudawar (2013a) for Eq. (43). There is an order of magnitude agreement in the flow-regime transition boundaries obtained from Eq. (43) and Eqs. (46) – (47), (see Narain et al. 2015) for 16000 \( \leq Re_T \leq 32000 \). Note that there is no surface tension dependence in Eqs. (46) – (47) for modest mass-flux thin-film annular flows (unlike Suratman number \( Su_{VO} \) dependence in Eqs. (43) – (45)) and this is consistent with the physics of channel flows. Eq. (43), being a curve-fit, loses accuracy but gains in the parameter-space ranges over which order of magnitude estimates are valid.

### 3.3 Void-fraction (\( \epsilon \)) and quality (\( X \)) correlations

Void-fraction (\( \epsilon \)) can be defined "locally" (Carey 1992; Ghiaasiaan 2007), i.e., as a variable whose value depends on the location of a point and the instant of time in a given two-phase flow field. However for the "steady in-the-mean" in-tube flows of interest (such as the ones in Figs. 1-3), void fraction \( \epsilon \) definition simplifies (Ghiaasiaan 2007) to:

\[ \epsilon = \frac{A_V(x)}{A} \]  

(48)

where \( A_V(x) \) is the cross-sectional area occupied by the gas-phase at any location of "x" (see Figs. 1-3) for two-phase flows in a tube of cross-sectional area "A."

It is expected that dependence of void-fraction \( \epsilon \), on quality \( X(x)(\equiv M_V(x)/M_{in}) \), density ratio \( \rho_V/\rho_L \), viscosity ratio \( \mu_V/\mu_L \), etc. need to be correlated. This is important in assessing the importance of mean gas-phase speed \( U_V(x) \) and the mean liquid-phase speed \( U_L(x) \). This is because

\[ U_V(x) \equiv \frac{M_V(x)}{(\rho_V * A_V(x))} = G \cdot X(x)/(\rho_V \cdot \epsilon(x)) \quad \text{and} \quad U_L(x) \equiv \frac{M_L(x)}{(\rho_L * A_L(x))} = G \cdot (1 - X(x))/(\rho_L - (1 - \epsilon(x))) \]  

As a result of this importance, several such correlations have been developed and are used in developing correlations for: total pressure gradient \( \left(- \frac{\partial p}{\partial x}\right)_T \) – particularly its accelerational component, which is very important in high heat-
flux boiling where $U_V(x)$ rapidly increases with $x$ (Kim and Mudawar 2014; Thome 2004); interfacial shear correlations; film thickness correlations; and heat transfer rate correlations for high mass-flux annular flows (Cavallini et al. 2006; Kosky and Staub 1971; Thome 2004).

Most of these void-fraction correlations have been developed by considering adiabatic flows and using various homogeneous, separated, and drift-flux modeling hypotheses (Ghiaasiaan 2007) – suitably correlated for agreement with adiabatic flow experiments in limited contexts. These correlations are not to be taken, however, as ones that model the "physics" of the phase-change flows. For example, for low mass flux laminar/laminar flows in a channel, $\epsilon - X$ relationships can be obtained by a combination of analysis and DNS for annular adiabatic, certain annular flow-boiling (see Ranga Prasad et al. 2016) and for annular flows-condensing cases (Naik et al. 2016, Naik and Narain 2016 and Narain et al. 2015). The comparisons of exact results for a representative situation (involving $\rho_V/\rho_L = 0.086$, $\mu_V/\mu_L = 0.0235$, $Ja/Pr_L = 0.034$, $Re_T = 4.0$, $Re_h = 9616$) is plotted in Fig. 5 – along with results from some correlations in literature – on an $\epsilon - X$ plane. Clearly laminar liquid/laminar vapor nearly exact $\epsilon - X$ relationship of adiabatic flows do not match similar exact CFD/DNS-based accurately modeled "flow-physics" results obtained for flow-condensation and flow-boiling.

![Fig. 5: Exact solutions’ (analytical and computational) comparisons for laminar/laminar annular flow – adiabatic flows, flow-condensation ($Ja/Pr_L = 0.034$), and flow-boiling ($Ja/Pr_L = 0.034$). For order of magnitude]
comparison purposes, Zivi (1964) and Rouhani and Axelsson (1970) correlations are also plotted for the same parameters ($\rho_V/\rho_L = 0.086$, $\mu_V/\mu_L = 0.0235$, $Re_T = 4$, $Re_h = 9616$).

Despite the "physics" issues associated with using $\epsilon - X$ adiabatic correlations for phase-change flows, two such popularly used engineering correlations are also plotted for laminar/laminar situations in Fig. 5. These popular correlations are as follows:

- **Zivi (1964):**
  \[
  \epsilon = \frac{1}{1 + \frac{1 - X}{X} \left(\frac{\rho_V}{\rho_L}\right)^{2/3}}
  \] (49)

- **Rouhani and Axelsson (1970) (with Steiner 1993 modifications for horizontal tubes):**
  \[
  \epsilon = \frac{X}{\rho_V} \left\{ \left[ 1 + 0.12(1 - X) \right] \left( \frac{X}{\rho_V} + \frac{1 - X}{\rho_L} \right) + \frac{1.18}{G} \left[ \frac{g \sigma (\rho_L - \rho_V)}{\rho_L^2} \right]^{1/4} (1 - X) \right\}^{-1}
  \] (50)

Besides quality $X$, adiabatic flow $\epsilon - X$ curve depends on $\rho_V/\rho_L$, $\mu_V/\mu_L$, and $Re_T$. Condensing and boiling flows additionally depend on $Ja/Pr_L$ (or $Bl$).

The main value and use of adiabatic flow correlations, such as the ones in Eqs. (49) – (50), for flow-boiling and flow-condensation is not its ability to capture flow-physics. It lies in the fact that, their order of magnitude correctness allows over a large range of parameters and flow-physics ($G$ values, liquid–vapor Reynolds numbers, etc.), development of other "curve-fit" correlations for HTC and pressure-drop that cover a large range of experimental data – such as those in Kim and Mudawar (2013c) and Kim and Mudawar (2013d).

### 3.4 Pressure-drop correlations

The engineering approach (as opposed to CFD-simulations approach) is to *a priori* decompose the total pressure gradient $\left( -\frac{\partial p}{\partial x} \right)_T$ for two-phase flows into three parts (frictional, gravitational, and accelerational/momentum) as:

\[
\left( -\frac{\partial p}{\partial x} \right)_T = \left( -\frac{\partial p}{\partial x} \right)_{fric} + \left( -\frac{\partial p}{\partial x} \right)_g + \left( -\frac{\partial p}{\partial x} \right)_{acc}
\] (51)

The subsequent step consists of modeling these three parts separately and then assembling the three modeled equations over the tube-length ($0 \leq x \leq L$) of interest. Denoting the total pressure-
drop, or rise, as \( \Delta p_T (\equiv p_{in} - p_{out} = p(0) - p(L)) \) – and allowing \( \Delta p_T \) to be negative (as pressure-rise is possible for some condensing flow cases) – one obtains:

\[
\Delta p_T \equiv \int_0^L \left(-\frac{\partial p}{\partial x}\right)_{\text{fric}} \, dx + \int_0^L \left(-\frac{\partial p}{\partial x}\right)_{g} \, dx + \int_0^L \left(-\frac{\partial p}{\partial x}\right)_{\text{acc}} \, dx \equiv (\Delta p)_{\text{fric}} + (\Delta p)_g + (\Delta p)_{\text{acc}} \tag{52}
\]

For each of the three pressure-gradients, there are several modeling approaches (Ghiaasiaan 2007; Kim and Mudawar 2014; Thome 2004). The following definitions arise from adding 1-D form of vapor and liquid momentum balance equations:

\[
\left( -\frac{\partial p}{\partial x} \right)_{\text{acc}} = G^2 \frac{d}{dx} \left[ \frac{X(x)^2}{\rho_v \epsilon} + \frac{1}{\rho_l} \frac{(1 - X(x))^2}{1 - \epsilon} \right] \tag{53}
\]

\[
\left( -\frac{\partial p}{\partial x} \right)_g = [\epsilon \rho_v + (1 - \epsilon) \rho_l] g \sin \phi \tag{54}
\]

In obtaining Eqs. (53) – (54), mass-balance equations and definitions of: void-fraction \( \epsilon \), mass-flux \( G \), and quality \( X \) are also used. Clearly, in Eq. (54), \( \left( -\frac{\partial p}{\partial x} \right)_g \approx 0 \) for horizontal tubes, because \( \phi = 0 \). This is because "\( \phi \)" measures the angle between the tube-axis and the horizontal.

For integrating Eq. (53) one may choose a void-fraction model – such as Zivi’s in Eq. (49) or the one in Eq. (50). For the frictional pressure gradient in Eq. (52), one may choose any one of several adiabatic pressure-gradient calculating models by replacing "\( X(x) = \text{constant} \)" situation for the adiabatic cases with genuine \( X(x) \) variations with \( x \) – as obtained by integrating the ODEs (Eqs. (23) – (24) of section 2.3) under use of appropriate flow-boiling or flow-condensation \( \text{Nu} \) correlations. Samples of such \( \text{Nu} \) correlations are given in Eqs. (27) – (30) of section 3.1.1 or in Eqs. (35) – (38) of section 3.1.2.

Two well-known frictional pressure gradient models for Eqs. (51) – (52) are:

**Lockhart-Martinelli Model:**

\[
\left( -\frac{\partial p}{\partial x} \right)_{\text{fric}} = \left( -\frac{\partial p}{\partial x} \right)_L \Phi_L^2 \tag{55}
\]

where \( \Phi_L \) is one of the several two-phase multipliers which depends on a certain Martinelli factor \( \tilde{X}_{MF} \) through the following relationships:

\[
\left( -\frac{\partial p}{\partial x} \right)_L \equiv \frac{2f_L G^2 (1 - X(x))^2}{\rho_L D_h} \tag{56}
\]
\[
\left(- \frac{\partial p}{\partial x}\right) \equiv \frac{2f_V G^2 X(x)^2}{\rho_V D_h} \quad \varnothing_L^2 = 1 + \frac{C}{X_{MF}} + \frac{1}{X_{MF}} \\
\bar{X}_{MF}^2 \equiv \frac{(\partial p/\partial x)_L}{(\partial p/\partial x)_V}
\]

The following quantities, with subscript "k" – where "k = L" or "k = V", are needed for evaluating the terms in Eqs. (55) – (56). These quantities include: \( \text{Re}_L(x) \equiv G(1 - X(x))D_h/\mu_L \) and \( \text{Re}_V(x) \equiv G. X(x)D_h/\mu_V \).

For phase "k" to be laminar, \( \text{Re}_k < 2000 \); and for it to be turbulent \( \text{Re}_k > 2000 \). The constant \( C \) in Eq. (56) is given in Table 5.

Table 5: Values of C for Lockhart-Martinelli model

<table>
<thead>
<tr>
<th>Liquid</th>
<th>Gas</th>
<th>C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbulent</td>
<td>Turbulent</td>
<td>20</td>
</tr>
<tr>
<td>Laminar</td>
<td>Turbulent</td>
<td>12</td>
</tr>
<tr>
<td>Turbulent</td>
<td>Laminar</td>
<td>10</td>
</tr>
<tr>
<td>Laminar</td>
<td>Laminar</td>
<td>5</td>
</tr>
</tbody>
</table>

The friction factors in Eq. (56) are:

\[
f_k = 16\text{Re}_k^{-1} \quad \text{for} \quad \text{Re}_k < 2000,
\]
\[
f_k = 0.079\text{Re}_k^{-0.25} \quad \text{for} \quad 2000 \leq \text{Re}_k \leq 20,000 , \quad \text{and}
\]
\[
f_k = 0.046\text{Re}_k^{-0.2} \quad \text{for} \quad \text{Re}_k \geq 20,000
\]

Friedel Model:

\[
\left(- \frac{\partial p}{\partial x}\right)_{\text{fric}} = \left(- \frac{\partial p}{\partial x}\right)_L \varnothing_{\text{Friedel}}^2
\]

where, while retaining the remaining definitions in Eq. (56), \( \varnothing_{\text{Friedel}}^2 \) is defined (see Friedel 1979 and Thome 2004) as:

\[
\varnothing_{\text{Friedel}}^2 \equiv E + \frac{3.24 \text{ FH}}{F_l^{0.045} + \text{We}_L^{0.035}}
\]

The terms in Eqs. (56) are:
\[ E = (1 - X)^2 + X^2 \frac{\rho_L}{\rho_G} \frac{f_G}{f_L} \]

\[ F = (1 - X)^{0.224} + X^{0.78} \]

\[ H = \left( \frac{\rho_L}{\rho_G} \right)^{0.91} \left( \frac{\mu_G}{\mu_L} \right)^{0.19} \left( 1 - \frac{\mu_G}{\mu_L} \right)^{0.7} \]

\[ \text{Fr}_H = \frac{G^2}{gD\rho_H^2}, \quad (59) \]

\[ \rho_H = \left[ \frac{X}{\rho_G} + \frac{1-X}{\rho_L} \right]^{-1}, \quad \text{and} \]

\[ \text{We}_L = \frac{G^2D_h}{\sigma \rho_H} \]

Kim and Mudawar (2014) report poor comparisons of experimental pressure drop (considering a large set of data) for flow-boiling and flow-condensation with those obtained from the above described procedures employing correlations by Lockhart and Martinelli (1949) and Friedel (1979).

Kim and Mudawar (2013d), after introducing, \( \text{Re}_{LO}(\equiv GD/\mu_L) \) & \( \text{Su}_{VO}(\equiv \rho_V\sigma D_h/\mu_V^2) \), recommend continued use of Lockhart and Martinelli model in Eqs. (55) – (56) with replacements for the constant \( C \) (appearing in Eq. (25)) in Table 5 being for adiabatic and condensing flows given in Table 6a below. For boiling flows, calculation of constant \( C = C_{\text{boiling}} \) involves use of \( C_{\text{non-boiling}} \) (calculated for adiabatic and condensing flows) in Table 6a through relationships given in Table 6b.

**Table 6:** Values of \( C \) for (a) adiabatic and condensing flows; (b) boiling flows

<table>
<thead>
<tr>
<th>Liquid</th>
<th>Gas</th>
<th>( C_{\text{non-boiling}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbulent</td>
<td>Turbulent</td>
<td>( 0.39\text{Re}<em>{LO}^{0.03}\text{Su}</em>{VO}^{0.1}(\rho_L/\rho_V)^{0.35} )</td>
</tr>
<tr>
<td>Laminar</td>
<td>Turbulent</td>
<td>( 0.0015\text{Re}<em>{LO}^{0.59}\text{Su}</em>{VO}^{0.19}(\rho_L/\rho_V)^{0.36} )</td>
</tr>
<tr>
<td>Turbulent</td>
<td>Laminar</td>
<td>( 8.7 \times 10^{-4}\text{Re}<em>{LO}^{0.17}\text{Su}</em>{VO}^{0.5}(\rho_L/\rho_V)^{0.29} )</td>
</tr>
<tr>
<td>Laminar</td>
<td>Laminar</td>
<td>( 3.5 \times 10^{-5}\text{Re}<em>{LO}^{0.44}\text{Su}</em>{VO}^{0.5}(\rho_L/\rho_V)^{0.48} )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>(b)</th>
<th>( C_{\text{boiling}} )</th>
</tr>
</thead>
</table>

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\[
\begin{align*}
Re_L \geq 2000 & \quad C_{\text{non-boiling}} \left[ 1 + 60We_{LO}^{0.32} \left( \frac{Bo}{P_H/P_F} \right)^{0.78} \right] \\
Re_L < 2000 & \quad C_{\text{non-boiling}} \left[ 1 + 530We_{LO}^{0.52} \left( \frac{Bo}{P_H/P_F} \right)^{1.09} \right]
\end{align*}
\]

**Gronnerud Model:** This popular pressure-gradient model, available in Grönnerud (1972), is used here – but not reviewed for brevity.

### 3.5 CHF considerations

As discussed in section 2.2, crossing certain threshold values of critical heat-flux in the most conservative "method of heating" – a uniform heat-flux imposition to on the flow-boilers in Fig. 1a or Fig. 3a – often leads to initial intermittent vapor blanketing of parts of the boiler-surface followed by sustained unsteady instabilities that lead to sustained vapor blanketing responsible for run-away rise in the temperature of the boiling-surfaces. There are several mechanisms for flow-boiling instabilities (Dhir and Liaw 1989; Haramura and Katto 1983; Katto 1994; Linehard and Dhir 1973; Zuber 1959) that have been summarized in the review article of Das et al. (2012) and their relationships to the mechanisms associated with the ones in flow-boiling have been discussed. The articles by Qu and Mudawar (2004) and Kim and Mudawar (2013b) also highlight CHF mechanisms for flow-boiling, CHF correlations and dry-out quality correlations.

For the traditional flow-boiling in Fig. 1a, Kumar and Kadam (2016) claim that CHF could manifest through one of the two following mechanisms. If the mas-flux is low, at low qualities, it manifests through departure from nucleate boiling (DNB) and results in vapor blanketing. This leads to inverted annular flows (Qu and Mudawar 2004) further downstream of DNB – where, unlike the typical plug-slug flows, the vapor flows along the boiling-surface and the core has liquid inside. This further leads to unsteady rise in boiling-surface temperatures in the inverted annular regions. However, if the mass-flux is high, CHF typically manifests at higher qualities through dry-out instability of annular liquid films, usually in the presence of some entrainment effects. The second mechanism associated with higher mass-flux values is generally considered to be the CHF associated with dry-out. It should be noted that mere dry-out (when one reaches dry-out quality, as correlated in Kim and Mudawar 2013b) for annular flows does not necessarily imply onset of unsteady instabilities. For these instabilities to occur, a threshold \( \tilde{q}_{\text{CHF}} \) value has to be crossed.

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If the mass-flux is of moderate value, CHF could manifest through either of the two mechanisms discussed above (Shah 2015a). This dual realization pathway could be because other non-dimensional parameters might also influence the transition between the two CHF mechanisms, and these other non-dimensional numbers are yet to be characterized or incorporated in the correlation. For moderate mass-fluxes in micro-channel, Kumar and Kadam (2016) also point out yet another instability caused by vapor bubbles growing in size to become comparable to $D_h$ of tube and then choking (creating pressure-drop instabilities) the tube flow.

Qu and Mudawar (2004) point out yet another instability mechanism for micrometer-scale hydraulic diameter flow-boilers. In this mechanism, inlet flow conditions of a saturated liquid as it approaches the inlet of a micro-tube may cause, in case of very high heat-flux steady pre-heating of the tube, backflow (into the feeding reservoir) and dry-out instabilities at the inlet location itself.

For the flow in Fig. 3a, dry-out instability is most relevant – particularly if the start-up procedure (see Kivisalu et al. 2014) is such that it first realizes annular flows at modest heat-flux levels and higher heat-flux levels are only gradually and subsequently imposed.

As discussed by Das et al. (2012), CHF is enabled by a macro-scale instability which works in tandem with a micro-scale instability phenomenon that does not allow replenishment of fluid in the micro-layer (see Raghupathi and Kandlikar 2016 and section 6) present on the wetting surface. With the macro- and micro-layer instabilities working together, a dynamic process then ensues resulting in a situation where only an adsorbed layer of fluid in the vapor-blanketed domain is allowed – and the area associated with this vapor-blanketing becomes large and sustained. The dry-out instability for the flow in Fig. 3a is also indicated by a large surplus in mechanical energy supplied to the film over which energy is dissipated within the film – see Ranga Prasad et al. (2016) – and this surplus of mechanical energy, typically, has to go to the vapor flow near the dry-out zone (see streamline patterns in Ranga Prasad et al. 2016). At high heat-flux values, this surplus energy can no longer be absorbed by the vapor or the wetting surface. Therefore, dry-out instabilities can often be delayed (i.e. CHF values increased) by enhancing the wetting characteristics in the exit zone. This requires sustaining the liquid flow into the micro-layer over the adsorbed layer by: super-hydrophilicity of the surface, or use of a porous exit zone boiling-surface with additional and independent liquid supply, or active wettability enhancement by using a dielectric boiling-surface near the exit and electric field imposition, etc.
For design purposes of flow-boilers in Fig. 3a (as discussed in the next section), one can tentatively use a relevant and existing CHF threshold value estimates (as discussed in Das et al. 2012) and/or simply ensure – by flow-control (also see section 6 as this is possible through proper design of innovative annular flow-boilers) – that the exiting liquid layer in sufficiently thick ( > 10-20 µm or so).

4. Aforementioned review of two-phase flow correlations and their use in design problems for annular and steady innovative flow-boiler and flow-condenser operations

The correlations discussed earlier (sections 2-3) were developed separately or by a combination of experimental or as modeling/simulations data obtained from CFD. These were aided by non-dimensionalization process and some understanding of the physics underlying phase-change heat transfer. These correlations are needed for the design of new experiments or new systems (for a given working fluid) which, in turn, require preliminary but reliable “order of magnitude” estimates of: liquid and vapor flow rates, heat transfer coefficient values, wall heat-flux or wall temperatures, inlet pressure, length of the device, etc. To begin with, until better correlations or experimentally obtained refinements for the existing ones become available, correlations that are presently available need to be judiciously used to define experimental conditions and instrumentation needs.

Need and use of existing correlations are shown here with the help of two specific examples. The first example is that of a preliminary design of a millimeter-scale flow-boiler operating in a steady annular/stratified regime with thin liquid film flows (Fig. 3a) either in the presence of nucleation (at micron and sub-micron scale bubble diameters) or under suppressed nucleation conditions (involving low heat-fluxes). The second example is that of a preliminary design of a millimeter-scale flow-condenser for a steady annular/stratified film-wise condensation on a hydrophilic surface (see Fig 3b). In both examples, it is desired that the duct be of rectangular cross-section of high aspect ratio (adequately modeled as a channel) and the heat-exchange surface be the bottom plate.

4.1 Design of millimeter-scale annular flow-boilers

The next sub-section discusses some of the desired specifications and constraints for the design of an innovative flow-boiler operating in the annular flow-regime.
4.1.1 Desired specifications, constraints, and information/knowledge needed for specifying them

(i) A pure working fluid should be chosen such that it has a saturation temperature in the 30-90 °C range for operating inlet pressures in the 100-110 kPa range.

(ii) The inlet quality (at \( x = 0 \) in Fig 3a) should be higher than the critical quality of transition from non-annular to annular flow-regimes – ensuring annular flow realization. For this, it is important that the quality at which flow regime transition occurs be approximately known with estimates coming from different flow regime transition maps whose required scientific structures have been discussed in section 2 and the status of available knowledge has been discussed in section 3. These estimates are sought to be within a reasonable range of values that would suffice with trial-and-error experimental adjustment, but in future, could be further improved (i.e., made more accurate) through a proper synthesis of experiments and modeling.

(iii) The inlet film thickness \( \Delta_{\text{in}} \), the liquid film thickness at the splitter plate in Fig. 3a, is desired to be around 300 µm (neither too thin nor too thick) for subsequent and possible transitioning of the steady operations to pulsatile operations – this is to take advantage of the operation principles and underlying physics hypotheses discussed in section 6 (see experimental results given by Kivisalu et al. 2014). This controlled inlet thickness of the liquid film is essential because, otherwise, as the liquid exits the splitter plate in Fig. 3a, it may change its thickness abruptly to an undesirable range. Thickness at or below 300 µm are needed to keep the films very stable – with or without tiny nucleating bubbles (micron/sub-micron bubble diameters) – even to the imposition of large amplitude standing waves on its interface. For such a design, a correlation for inlet film thickness with dependence on inlet quality is required as input. Since this is a stratified/annular flow in a rectangular channel, with nearly adiabatic self-seeking free-surface locations immediately after the inlet splitter plate (see Fig 3a), a set of void-fraction correlations for adiabatic flows (discussed in Section 3) could be used to obtain a good range of appropriate inlet quality and associated inlet film thickness values.

(iv) To avoid dry-out related CHF instabilities at or near the exit, the exit quality should be less than 1 and the heat-flux at the exit should be less than the available order of magnitude
estimates for CHF associated with dry-out instability \((q^*_{\text{w|exit}} < q^*_{\text{CHF|dry-out}})\). For this, an estimate of a dry-out related CHF, obtained from correlation(s), such as the ones presented by Qu and Mudawar (2004), may be used. However, instead, the following more conservative constraint is recommended. As a safety measure, it is required that the film thickness at the exit, estimated in different ways, be equal to or greater than one-fifth the inlet film thickness (i.e., \(\Delta_{\text{out}} \geq \Delta_{\text{in}}/5 \geq 0 \text{ (50 µm)}\)).

(v) Some of the important information that are required for this design are the Reynolds numbers: Reynolds numbers based on total mass-flux \(\text{Re}_T \equiv \frac{G h}{\mu_V}\), liquid Reynolds number \(\text{Re}_L \equiv \frac{G(1-X) h}{\mu_L}\), and vapor Reynolds number \(\text{Re}_V \equiv \frac{GX h}{\mu_V}\). The first one is typically used in evaluating Nusselt number correlations for saturated flow-boiling covering annular regimes (over appropriate range of qualities) while the second and third Reynolds numbers are often needed for assessing whether the liquid and vapor flows are laminar or turbulent and also for selecting constants in sub-correlations (either directly for Nusselt number correlations or indirectly for related pressure drop correlations). The knowledge of the laminar/turbulent nature of liquid and vapor flows are also useful in choosing appropriate correlation(s) that are available for different void-fraction models.

(vi) Since there are separate inlet channels for liquid and vapor, and a splitter plate is used (as in the experiments of Kivisalu et al. 2014) to ensure that the liquid and vapor do not mix before they enter the test-section (as shown in the Fig 3a), the cross-sectional area for vapor flow just before the inlet (locations \(x < 0\)) is different than the cross-sectional area for the vapor flow within the test-section (locations \(x \geq 0\)). Furthermore, because of non-zero vapor inflow rate at the inlet and its subsequent acceleration associated with flow-boiling, the vapor speed at the exit of the test-section would be higher (much higher for higher heat-fluxes) than it is at the inlet of the test section. To avoid compressibility related choking effects (see Ghiaasiaan 2007), it is required that the vapor speeds at both the inlet and the exit be, approximately, less than \(1/3\text{rd}\) of the speed of sound for saturated vapor operating at pressures that are at or below the inlet pressure value.

(vii) The inlet pressure \(p_{\text{in}}\) is required to be above atmospheric pressure such that, despite the pressure drop along the length of the channel, the exit pressure is also higher than the atmospheric pressure (i.e., \(p_{\text{out}} > p_{\text{atm}}\)). This requirement is to make the design simpler by
not requiring the system to exhibit stringent and prolonged tolerance to vacuum pressures. For systems operating below atmospheric pressure, even small leaks of air can, over time, lead to substantial buildup of non-condensable air into the system and system-design based on pure vapor and pure liquid correlation and flow-physics, as proposed, will fail.

(viii) Further, despite the pressure-drop between the inlet and the exit of the channel, mechanical power in the vapor at the exit is likely to be much more than the incoming power at the inlet – i.e., for exiting vapor speeds relative to the inlet, it is expected that \( P_{V,\text{out}} (\equiv p_{\text{out}} \cdot v_{V,\text{out}} \cdot A_{\text{out}}) > P_{V,\text{in}} (\equiv p_{\text{in}} \cdot v_{V,\text{in}} \cdot A_{\text{in}}) \). The net power out \( (P_{V,\text{out}} - P_{V,\text{in}}) \) has to be maximized for high heat-flux cases. This is to ensure that only minimal additional power is needed for the vapor-phase compressor (and pulsator, in Fig 3a). Note that the liquid pump and associated pulsator (discussed in section 6) already consume insignificant power.

(ix) For a known uniform heat-flux specifying the "method of heating," Eq. (24) yields linear quality variations and, subsequently, order of magnitude estimate of the heat transfer coefficient \( h_x \) can be obtained from a Nusselt number correlation – which is only needed for assessing wall temperature \( T_w(x) \) variations associated with the boiling-surface. It is required that the design be such that the mean wall temperature \( \overline{T}_w \) not be too high and remain below a certain threshold value.

(x) Some of the parameters whose desired ranges need to be recommended or chosen to propose a design that satisfies the above conditions are: mass-flux \( G \), length of the channel \( L \), height of the channel \( h \), inlet pressure \( p_{\text{in}} \), and inlet quality \( X_{\text{in}} \).

4.1.2 Implementation of a sample design methodology meeting the requirements in section 4.1.1 – and results for annular flow-boilers

A range of inlet pressures \( p_{\text{in}} \), total mass fluxes \( G \) and mean heat-flux \( \overline{q}_w \) (or mean wall temperatures \( \overline{T}_w \) depending on "method of heating") are initially chosen and considered. They are then optimized to satisfy most of the constraints mentioned in the above described design requirements. Subsequently, the length \( L \) of the flow-boiler may also be adjusted to satisfy the remaining constraints. This is reasonable because sequential series and parallel stacking of innovative boilers is a feasible solution for a new system covering a designated area of design interest.
Towards this, one particular combination of inlet pressure, total mass-flux, and mean heat-flux (or mean wall temperature, depending on the choice for the "method of heating") is defined as a specific initial operating condition for a given channel height $h$ and a single flow-boiler design. For this single flow-boiler design requirement, the channel height is chosen to be 5 mm, the working fluid is R-123, and the remaining operating conditions are as follows: inlet pressure $p_{in} = 120$ kPa; total mass-flux $G = 300$ kg/m$^2$s; and mean heat flux $\bar{q}_w = 50$ W/cm$^2 = 500$ kW/m$^2$ for steady operations. By superposing pulsatile operations, as discussed in section 6, the steady design's $\bar{q}_w$ handling capability can be increased by a factor of 5-15.

(i) For any chosen operating condition, the first step is to find the critical quality for transition from non-annular to annular flow-regimes ($X_{cr|NA-A}$). This is accomplished by using approximate flow-regime transition maps. A few appropriate flow-regime transitions maps should be chosen, based on hydraulic diameter $D_h$, fluid, orientation (horizontal, in this case) of the channel/tube, etc. Since the flow-regime transition maps are semi-empirical in nature (correlated using data from either existing experiments or modeling/simulations), as a conservative measure, it is recommended that the critical quality for transition from non-annular to annular flow-regime $X_{cr|NA-A}$ be obtained from at least three different flow-regime transition maps and the highest of those qualities be used. For the flow-boiler design under consideration, flow-regime transition maps proposed by Harirchian and Garimella (2012), Kim and Mudawar (2013a), and Mandhane et al. (1974) were considered to be approximate ones for further evaluations. Even though the map by Mandhane et al. (1974) was developed for adiabatic flows, its use in the present scenario is justified – this is because the flow will be approximately adiabatic immediately after it exits the splitter plate at the inlet (Fig. 3a). The transition qualities obtained using the correlations are given in Table 7. It can be noticed that the critical transition qualities obtained from correlations proposed by Kim and Mudawar (2013a) and Harirchian and Garimella (2012) are quite low and less likely to be applicable to the present design estimates. It was further noticed that, for low values of mass-flux ($G < 30$ kg/m$^2$s), Kim and Mudawar (2013a) correlation provided higher values of critical transition quality while the critical transition qualities obtained from correlation by Harirchian and Garimella (2012) and flow-regime map by Mandhane et al. (1974) were quite low ($< 0.1$). For the initial conservative estimate, it is then decided to set $X_{in} > 0.3$ (as implied by flow-regime map of Mandhane et al. 1974).
Table 7: Critical transition qualities from non-annular to annular flow-regime obtained using correlations for the channel height, \((h = 5 \text{ mm}, \text{Fluid } – \text{R-123} \& \text{operating conditions: } p_{\text{in}} = 120 \text{ kPa}; G = 300 \text{ kg/m}^2\text{s}; q_{\text{wall}} = 50 \text{ W/cm}^2)\)

| Flow regime transition correlations       | Transition quality, \(X_{\text{cr|NA-A}}\) |
|------------------------------------------|------------------------------------------|
| Harirchian and Garimella (2012)          | 0.028                                    |
| Kim and Mudawar (2013a)                  | 0.0283                                   |
| Mandhane et al. (1974)                   | 0.307                                    |

(ii) For approximately realizing the assumed inlet film thickness \(\Delta_{\text{in}}\) of around 300 \(\mu\text{m}\), first the vapor Reynolds number \(\text{Re}_V(\equiv G\chi h/\mu_2)\) is calculated with \(X_{\text{cr|NA-A}}\) as initial guess for quality \(X (= X_{\text{in}})\) to check if the vapor flow is laminar or turbulent. If the vapor flow is turbulent (and it is ensured that the liquid flows are still laminar), the adiabatic void fraction correlations proposed by Zivi (1964) and Rouhani and Axelsson (1970) (with modifications for horizontal flows proposed by Steiner 1993) are used to calculate film thickness (note that \(\epsilon_{\text{in|correlation}} = (h - \Delta_{\text{in}})/h\)). If the vapor flow is also laminar, void-fraction correlation proposed by the authors (Ranga Prasad et al. 2016) may be used along with the ones mentioned earlier. Although all these above mentioned void-fraction correlations were originally proposed for adiabatic flows, their use in this case is, once again, justified since the flow is approximately adiabatic immediately after the splitter plate at the inlet in Fig. 3a. Using \(X_{\text{cr|NA-A}}\) as the first guess, the film thicknesses are calculated using these void-fraction correlations. Next the inlet quality value is typically increased to values higher than \(X_{\text{cr|NA-A}}\) until the chosen correlations give a maximum liquid film-thickness value of around 300 \(\mu\text{m}\) or less. However, since these are semi-empirical correlations with large uncertainties, the mean of the different film thicknesses obtained from the three different correlations above is actually considered to be the final film thickness value. The final inlet film thickness value, for this example, therefore is estimated to be 215.3 \(\mu\text{m}\) and the corresponding inlet quality is found to be 0.53 (> \(X_{\text{cr|NA-A}}\)).

(iii) For this inlet quality, using the space available for the vapor flow above the splitter plate (3.7 mm for the case in Fig. 3a), the vapor velocity and its ratio relative to the speed of sound is
calculated. For the current case, it is found to be 0.22. If the ratio of inlet vapor to the speed of sound is more than 0.28 (conservatively chosen to be less than 0.33), it is suggested that, initially, the mass-flux $G$ be reduced (which essentially means changing the initially guessed operating condition) and steps (i) to (ii) be repeated. If that is not feasible, the height of the channel may also be increased so that the ratio of the vapor speed to sound speed is reduced.

(iv) If the compressibility effect (Mach number $< 0.28$) constraint is satisfied at the inlet, assuming an arbitrarily high exit quality (say, 0.98) for a sufficiently long boiler, and then using Nusselt number correlation(s), the heat transfer coefficient values for the range of qualities are calculated, along with their nucleate boiling ($h_{nb}$) and convective boiling ($h_{cb}$) components. For the current design problem, the correlation proposed by Kim and Mudawar (2013c), as discussed in Section 3, is used. The total heat-transfer coefficient $h_x$ and quality $X(x)$ variations (see Fig. 6) are then computed by integration of Eq. (24) and as per discussions in Section 2. The film thickness values for these qualities can be estimated using the formula:

$$\Delta(x) \geq \Delta_{CFD}(x) \cong \frac{k_L}{h_{x|cb}(x)}$$  \hspace{1cm} (60)

provided $h_{x|cb}$ values in Eq. (29) can be trusted (see Ranga Prasad et al. 2016 for better estimation of $h_{x|cb}$). However, it should be noted that, this correlation (in fact most correlations) is semi-empirical in nature and obtained from curve-fitting available data for the total $h_x$; so, while the total heat transfer coefficient $h_x$ value might be approximately correct to within a certain percentage, the nucleate boiling and convective boiling term decomposition making up the total is arbitrary and (as discussed in Ranga Prasad et al. 2016) in all likelihood not correct. See Fig. 7a for such an initial (and likely incorrect) decomposition provided by correlation from Kim and Mudawar (2013c). Therefore, a correction factor – defined to be the ratio of inlet film thickness calculated using void-fraction models (in step (ii)) and the inlet film thickness calculated using convective boiling heat transfer coefficient term $h_{x|cb}$ – is introduced. This is done because it is assumed that the void-fraction models provide more reliable estimate of film thickness value at the inlet – because of adiabatic type conditions there. It is then assumed that this can be used to better evaluate the convective component of heat transfer coefficient (with $h_{x|cb, in} \cong k_L/\Delta_{in}(x)$) at the inlet because this is a more reliable estimate. Therefore, the convective boiling term $h_{x|cb}$ in Kim and Mudawar (2013c) is not trusted and is corrected by dividing it by the
correction factor (which was found to be 5.48 for the present case) – this is to achieve compatibility with \( h_{x|\text{cb,in}} \) above. The nucleate boiling contribution term is calculated assuming that the total heat transfer coefficient, which is experimentally known to be of reasonable accuracy, through relations such as: 

\[
h_{x|\text{nb}} = h_x - h_{x|\text{cb}} \quad \text{or} \quad h_{x|\text{nb}} = \sqrt{h_x^2 - h_{x|\text{cb}}^2}
\]

depending on how \( h_x \) was originally decomposed between \( h_{x|\text{nb}} \) and \( h_{x|\text{cb}} \).

(v) Using the new corrected convective boiling \( h_{x|\text{cb}} \) term as a function of calculated quality \( X(x) \), the corresponding film thickness \( \Delta(x) \) values are calculated (using Eq. (60) given in step (iv)) as a function of quality \( X \). Then using energy balance formula in Eq. (24), variation of quality with distance is calculated (Fig. 6). Further, using the vapor height above the exit film thickness \( (\Delta_{\text{exit}} = \Delta(L)) \), the vapor speed and its ratio with respect to the speed of sound are calculated for the exit location \( x = L \). Now with the knowledge of film thickness as a function of quality and quality variation along the length of the channel, this step is iterated until an exit quality and a corresponding length \( L \) is found such that, the ratio of exit vapor speed to the speed of sound is equal to or less than 0.28 and the film thickness is greater than or equal to \( \frac{1}{5} \ast \Delta_{\text{in}} \).

This quality is considered as maximum exit quality and the corresponding channel length, is considered the maximum possible length \( L_{\text{max}} \). The exit quality and maximum length thus calculated for the present design problem are 0.88 and 0.18 m respectively. The heat transfer coefficient values, for the range of qualities between the inlet and the exit, along with their corrected component terms \( (h_{x|\text{nb}} \text{ and } h_{x|\text{cb}}) \) are plotted in Fig. 7b. The film thickness values along the length of the channel (for the same quality range), calculated using convective boiling heat transfer coefficient term, are plotted in Fig. 8. Using total heat transfer coefficient values, wall temperature values are calculated using Eq. (4) and are plotted in Fig. 9.
Fig 6: Variation of quality $X$ along the length of the channel.
Fig 7: Variation of heat transfer coefficients with quality – calculated using correlation proposed by Kim and Mudawar (2013c) (a) along with the decomposition of $h_x$ into $h_{x|nb}$ and $h_{x|cb}$ using the correlation; (b) with decomposition of $h_{x|nb}$ and $h_{x|cb}$ using the correction factor discussed in step (iv).

Fig 8: Variation of liquid film thickness $\Delta$ along the length of the channel – calculated from corrected convective boiling term of heat transfer coefficient $h_{x|cb}$ using Eq. (60)
Fig 9: Variation of wall temperature $T_w(x)$ along the length of the channel – calculated from total heat transfer coefficient using Eq. (4)

A few important points to note are as follows:

(a) It is first assumed that the pressure drop is quite low compared to the inlet pressure and therefore the sound speed at the exit is first calculated using the inlet pressure.

(b) The length calculated here is the maximum permissible length that would satisfy the constraints. However, as a conservative measure, it is suggested that the length of the channel could be a fraction (say, 0.7) of the maximum length $L_{\text{max}}$ calculated earlier and that fraction should be further reduced with increasing mean heat-flux ($\overline{q}_{w}^{*}$) values.

(vi) For the calculated/identified quality variations between the inlet and the exit, both acceleration and frictional pressure drops are calculated along the length of the channel. Acceleration pressure drop is calculated using Eq. (53) and frictional pressure drop is calculated using three different correlations viz. Friedel (1979), Grönnerud (1972) and Lockhart and Martinelli (1949). This is done to get a range of possible pressure drop values. However, for further calculations, as a conservative measure, the highest of those pressure-drop values is chosen. The pressure drop values along the length of channel are shown in Fig. 10 and the final exit pressure (including momentum/acceleration pressure drop) is, for this example, 114.05 kPa.
Fig. 10: (a) Variation of frictional pressure gradient along the length of the channel – calculated from Gronnerud, Friedel and Lockhart & Martinelli pressure-drop correlations. (b) Variation of net pressure difference between inlet and a location x downstream of the inlet. The three curves from the three models are closer together than in part (a). This is because of dominance of accelerational pressure-drop in Eq. (52).

(vii) Using the chosen inlet pressure and the just calculated exit pressure, along with vapor velocities (calculated in steps (iii) and (v) for the inlet and the exit respectively), and the
corresponding cross-sectional areas; net output power per unit channel width for the test section (as discussed in constraint (vi) of section 4.1.1) is calculated and is found to be 7.5 kW/m. The vapor compressor power per unit width of the section is conservatively estimated as \(1.2 \times (p_{in} - p_{out}) \times v_{in} \times h \cong 1.03\) kW/m. Since net mechanical power input from the compressor is less than the mechanical power output, the flow-boiler design operation is considered energy efficient.

4.2 **Design of millimeter-scale annular flow-condensers**

Design of the steady annular innovative operation of condenser shown in Fig. 3b is discussed in this section. Because of factors such as absence of CHF, inlet quality of vapor flows being \(X \cong 1\), etc., the number of constraints in the design of annular flow-condensers is relatively less – as compared to that of annular flow-boilers.

4.2.1 **Design specifications and constraints and types of information/knowledge needed to satisfy them**

This section discusses some of the constraints involved in the design of flow-condensers that may be required to operate in the thin liquid steady annular flow-regime. Such condensers can, in principle, be converted to high heat-flux compact devices by superposing a pulsatile flow-physics – described in section 6.

(i) It is important to know the transition quality from annular to non-annular flow-regimes so that the exit quality can be adjusted and kept above the transition quality, with the help of the passively recirculating vapor as in Fig. 3b. Using this strategy, non-annular flow regimes (associated with high pressure drop and poorer heat transfer rates) can be avoided – all the way up to the exit of the condenser in Fig. 3b. For this, some appropriate flow-regime maps, as discussed in section 3.2.2, are used.

(ii) The total inlet Reynolds number (\(\text{Re}_{in} = \text{Re}_T = G h/\mu V\)) needs to be computed for the chosen mass-flux \(G\). This number is directly or indirectly required to evaluate any of the suitable Nusselt number correlations for flow-condensers, and, obtain quality variations along the length of the channel for, say, a known wall temperature case for specifying "method of cooling." Further, for the known temperature specifying a "method of cooling," appropriate local Nusselt number correlation(s) would be required and should be used to
get an order of magnitude estimate of the local heat transfer coefficient \( h_x \), local heat flux \( q_{w}^*(x) \) and the quality \( X(x) \) variations along the length of the channel (obtained from using Eq. (23) in section 2).

(iii) Unlike the earlier case of flow-boiling, the mechanical power in the vapor flow at the exit is, typically, lower than the mechanical power available in the vapor flow at the inlet. This is because of reduction in velocity along the length of the channel. However, the design should be optimized such that the reduction in velocity is minimal, and, consequently the loss in mechanical power in vapor is also reduced – which is a relatively easy task for these millimeter scale steady annular operations.

(iv) The length \( L \) of the condenser should be chosen such that exit liquid film thickness \( \Delta(L) \) is less than say, 200 \( \mu \)m.

(v) Some of the parameters that are to be initially chosen and then varied to optimize the design are as follows: mass flux \( G \), height of the channel \( h \), length of the channel \( L \), inlet pressure \( p_{in} \), wall temperature \( T_{w} = \overline{T}_{w} \), etc.

4.2.2 Implementation of a sample design methodology meeting the requirements in section 4.2.1 – and results for annular flow-condensers

For a chosen working fluid, a range of inlet pressures \( p_{in} \), mass fluxes \( G \), and mean wall temperatures \( \overline{T}_{w} \) are to be initially considered. These can later be optimized, based on constraints mentioned above – or based on other specific and additional design requirements. However, similar to the flow-boiling design methodology, a particular combination of inlet pressure, total mass-flux and mean wall temperature is first chosen as a specific operating condition around which all variations are to be considered. For the present case, the following parameters are chosen: Fluid – R123; Channel height \( h = 5 \) mm; Inlet pressure \( p_{in} = 105.1 \) kPa; Total mass-flux \( G = 300 \) kg/m\(^2\)s; mean wall temperature \( \overline{T}_{w} = 382 \) K which corresponds to a cooling temperature difference \( \Delta T = 60 \) K.

(i) For the chosen operating condition, the critical quality of transition from annular to non-annular flow regimes \( (X_{cr\mid\text{NA-A}}) \) is calculated from available flow regime maps, some of which have been discussed in section 3.2.2. Flow regime transition maps should be appropriately chosen with the working fluid, the orientation (horizontal, vertical or inclined)
of the channel, and mm/μm-hydraulic diameter scale of the channel considered for selecting relevant maps in the literature. For the present case, flow-regime transition criterion proposed by Kim and Mudawar (2013a) and flow regime map proposed by Mandhane et al. (1974) have been used. The critical transition qualities obtained from these flow-regime criteria are 0.027 and 0.273 respectively. Again, as in the design of innovative flow-boilers (in section 4.1.2), the value obtained from the criterion proposed by Kim and Mudawar (2013a) is unacceptably low for this high mass-flux case even though the same correlation was found to have yielded reasonable estimates for the lower mass-fluxes \( G < 50 \text{ kg/m}^2\text{s} \). The highest of the qualities obtained from these two criteria is tentatively considered as the critical quality of transition from annular to non-annular flow-regime. Furthermore, as a conservative measure, since the exit quality should be more than the critical transition quality, an even higher value (say, by \( \Delta X = 0.05 \) to 0.1) than the estimated critical value of transition quality is chosen to be the exit quality (i.e., say \( X_{out} \approx X_{cr[NA-A]} + 0.1 \)). The chosen exit quality for the present case, at this step, is 0.373.

(ii) Assuming an inlet quality of 1 (or slightly less, say 0.99) and with the knowledge of the exit quality calculated/identified from step (i) above, the local heat transfer coefficient values for qualities between the inlet and exit qualities can be obtained by using Nusselt number correlation(s). For the current design problem, Nusselt number correlation proposed by Kim and Mudawar (2013a), and given in Eqs. (35) – (38), has been used. The variation of local heat transfer coefficient values (calculated using the chosen Nusselt number correlation) for the range of qualities between the chosen inlet quality and the identified exit quality have been plotted and shown in Fig 11. The kinks in Fig. 11 at low quality, which in reality will be a gradual smoothed curve, are due to change in the nature of vapor-phase flow from turbulent to laminar. The Nusselt number correlations, along with inlet and exit qualities can also be used to calculate the length of the channel \( L \), by obtaining \( X(x) \) variations using Eq. (23), which is, for the present case, found to be 0.798 m. However, it should be noted that this would be the maximum possible length of the channel and considerations should be given to the fact that it is to operate in annular flow-regime for the given operating conditions. Also, the test-section might have to operate for multiple operating conditions and all of the conditions should be taken into consideration before finally deciding the length of the channel. For the chosen operating condition, however, the variation of quality along
the length of the channel, as obtained from integrating Eq. (23), is shown in Fig. 12. The heat transfer coefficient values calculated above (see Fig. 11) can also be used to calculate the local wall heat-flux ($q_w(x)$) values, by using Eq. (4) – this is done and the results are plotted in Fig. 13.

**Fig 11:** Variation of heat transfer coefficients with quality – calculated using correlation proposed by Kim and Mudawar (2013a).

**Fig 12:** Variation of quality $X$ along the length of the channel calculated using Eq. (23)
(iii) Using the inlet and the exit qualities, the pressure-drop/rise can be calculated using appropriate correlation(s). For this design, to get an estimate of changes in pressure along the length of the channel, frictional pressure drop correlations proposed by Friedel (1979), Grönnerud (1972) and Lockhart and Martinelli (1949) were chosen and used. The accelerational pressure variation is calculated using Eq. (53). The net pressure variations along the length of the channel, calculated using the three chosen correlations are plotted in Fig. 14 and final exit pressure, for $L = 0.798 \text{ m}$, is found, by averaging the three pressures, to be approximately 115.35 kPa. Such rise in pressures (given $p_{in} = 105.1 \text{ kPa}$) are possible in condensing flow operations due to vapor deceleration effects. However, as mentioned in section 4.2.1, the design calculations should be done for different operating conditions and pressure-drop (or rise) may be optimized, if needed, to minimize the reduction in mechanical power in the vapor (along with any other specific design constraint(s) that may also need to be imposed).
Fig 14: (a) Variation of frictional pressure gradient along the length of the channel – calculated from Gronnerud, Friedel and Lockhart & Martinelli frictional pressure-drop correlations. (b) Variation of net pressure difference between a location $x$ downstream of the inlet and the inlet. This value rises with distance $x$. The three curves from the three models are closer together than in part (a). This is because of dominance of decelerational pressure-drop in Eq. (52).
5. **Improved CFD enabled correlations – current status and future trends – towards developing better design tools and improved understanding of flow-physics**

As discussed for the sample design methodologies presented in section 4, the approximate order of magnitude prediction capability of available correlations and their strong empirical nature – for both annular flow-boiling (because of the significance of nucleate boiling contribution to the annular boiling heat transfer coefficient) and flow-condensation – need to be improved for better understanding of flow-physics as well as improved accuracy of the correlations. The basic approach for such improvements is illustrated through the concept diagram in Fig. 15.

**Fig. 15:** Currently popular correlations employ, predominantly, the methodologies indicated in the experiments (blue-bordered) box. Modeling tool improvements require a bi-directional synthesis with the methodologies in the simulations (red-bordered) box.

Current status of correlations for annular flow-boiling and flow-condensation, as reviewed in Sections 2 and 4, is predominantly empirical (with limited exceptions) and are characterized by the methodologies indicated in the experiments (blue-bordered) box of Fig. 15. The sought-for improvements require development of first-principles (feasible for laminar/laminar annular flows) or minimal modeling assumptions based (for turbulent vapor and laminar liquid annular flows) computational simulations and accurate processing of such simulation results in the form of correlations – as indicated by the methodologies in the simulations (red-bordered) box of Fig. 15 – and their subsequent improvements with the help of proper synthesis with experiments. Therefore, such computational simulations need to be done in an environment where there is an active bi-directional information-sharing with appropriately planned experimental activities – for concurrently enabling both the methodologies/activities (for the experiments and the simulations boxes in Fig. 15) in a way that their results allow synthesis.

In the following sub-section, current status of simulation activities and their much needed evolutions, expected in the near future, are discussed separately for annular flow-boiling and annular flow-condensation.
5.1 Annular flow-boiling

As seen in estimates of heat transfer coefficient $h_x$ in section 4 (see empirical estimates for high heat and mass-flux cases as shown in Figs. 7a - b), experimental data underlying such correlation-based $h_x$ estimates (and its convective and nucleate boiling components) – supplemented by reasonable assumptions with regard to innovative annular flow-boiling approach in Fig. 3a – indicate dominance of nucleate-boiling over convective boiling ($h_{x|nb} \gg h_{x|cb}$), though the accuracy of estimates of $h_{x|cb}$ and $h_{x|nb}$ is at least as approximate as $h_x$ in Eq. (31). However, as discussed in section 4, the decomposition approach of $h_x$ into $h_{x|nb}$ or $h_{x|cb}$ – either through Eq. (33) in section 3 with $n = 2$ or other similar ones (e.g. Steiner and Taborek 1992, with $n = 3$) which use other values for “$n$” – is arbitrary and can be significantly improved by the superior definition of what is $h_{x|cb}$ and how it is to be obtained – as given in Ranga Prasad et al. (2016) and a sample reported here in Eq. (32). Since the film-thicknesses are small in Figs. 3a and 8 and the liquid film flows are laminar, these nucleate-boiling dominated high heat-flux shear-driven flows will typically have small bubble diameters (microns to sub-micron scales) whose population density and ebullition cycle (growth and departure) related frequencies – which increase with increased heat-flux, as in pool boiling (Ghiaasiaan 2007) – are, at present poorly understood because these mechanisms, unlike pool-boiling, are dictated by significant viscous and inertia forces in the presence of significantly weakened transverse gravity (or buoyancy) effects. Though some studies in related area exist (Zeng et al. 1993 etc.), it is quite likely that the existing pool boiling based knowledge (Egorov and Menter 2004; Gerardi et al. 2010; McHale and Garimella 2010) for mechanisms underlying the models for nucleation phenomena with regard to bubble-diameters, activated nucleation site densities, and ebullition cycle frequencies (which are typically 0.5 – 3 mm in bubble-diameters, $10^6 – 10^7$ activated sites/m², and 40-200 Hz for refrigerants experiencing pool-boiling under relevant operating conditions being similar to those considered for Figs. 6-10) are not entirely applicable to thin-film nucleate boiling. This is likely because changed bulk forces in the thin film liquid motion affect nucleation-controlling contact-line physics (Kandlikar and Grande 2003).

Furthermore, for low heat-flux and mass-flux cases ($\bar{q}_{lw} < 0.2$ W/cm², $G < 20$ kg/m².s, and refrigerants considered here), preliminary annular flow-boiling experiments and theoretical/computational $h_{x|nb}$ estimates (Gorgitrattanagul 2017; Narain 2012; Ranga Prasad et
(Ranga Prasad et al. 2016) indicate that the thinner downstream liquid flows do exhibit suppressed nucleation convective boiling (i.e. \( h_{\text{x|cb}} \gg h_{\text{x|nb}} \) if \( \Delta(x) \) is in 1 – 10 \( \mu \text{m} \) thickness range. Since total heat transfer coefficient \( h_x \) (in equations such as Eq. (33)) can be experimentally assessed through relations such as: \( q_{w|\text{Expt}}(x) \equiv h_x \times \Delta T(x) \), where \( \Delta T(x) \equiv T_w(x) - T_{\text{sat}}(p_{\text{in}}) \) and \( T_w(x) \), like wall heat-flux \( q_{w|\text{Expt}}(x) \), is also experimentally measured boiling-surface temperature, one can use computational simulations under the assumption of \textbf{suppressed nucleation} (as in Ranga Prasad et al. 2016) and obtain superior \( h_{\text{x|cb}} \) estimates for experimentally measured boiling-surface temperature \( T_w(x) \). With \( h_x \) and \( h_{\text{x|cb}} \) known, \( h_{\text{x|nb}} \) can be accurately estimated through the defining relationship given as:

\[
h_x \equiv h_{\text{x|nb}} + h_{\text{x|cb}}
\]

Therefore, in the \textbf{computationally obtained correlations} for \( h_{\text{x|cb}} \) for different wall temperature values in the structural format of Eq. (11) (see section 2), the "Ja/Pr_1" dependence in the temperature-prescribed version of \( \text{Nu}_x \) correlations for flow-boiling (under the assumption of suppressed-nucleation), one uses \( T_w(x) \) in \( \Delta T(x) \) for which actual experimental values of \( q_{w|\text{Expt}}(x) \) are known. Thus, as in Ranga Prasad et al. (2016), this recommendation for estimating \( h_{\text{x|cb}} \) does the following: (i) it defines what \( h_{\text{x|cb}} \) is for annular flow-boiling, and (ii) it realizes that though nucleation will actually impact convective boiling’s thermal boundary-layers - it puts all such effects (including direct nucleation effect) in the definition of \( h_{\text{x|nb}} \) through Eq. (61).

The above CFD enabled \( h_{\text{x|cb}} \) estimate has another advantage – such suppressed nucleation assumption CFD results when compared with experiments, where an actual nucleation suppression occurs (typically at low mass and heat-flux values and at thin film downstream locations), can help – and is being used (Gorgitrattanagul 2017; Sepahyar 2018) – in developing a criterion/correlation that can yield a-priori information on when nucleate boiling contributions to \( h_x \) can be considered to be actually absent.

\textbf{5.1.1 Low heat-flux and low mass-flux cases and} \( h_{\text{x|cb}} \) \textbf{estimates}

First-principles based accurate estimates of \( h_{\text{x|cb}} \) for the assumption of suppressed nucleation is fairly recent (Ranga Prasad et al. 2016) and is currently available only for limited low range of heat-flux and mass-flux values, as in Eq. (32). The development and implementation of CFD
algorithm in Ranga Prasad et al. (2016) is similar to the ones proposed for annular flow-condensation (Naik and Narain 2016; Naik et al. 2016). These include algorithm development which, in turn, are based on analytical approaches – covering both modeling and numerical methods (also see chapter on: Numerical Methods).

A sample annular flow-boiling of FC-72 (under the assumption of suppressed nucleation) on the bottom plate of a rectangular cross-section horizontal channel, as in Fig. 3a, is considered here for exhibiting its potential including prediction of \( h_{x|cb} \). The assumption of no nucleation is used not necessarily to say that the annular flow is actually occurring in the suppressed nucleation mode but is to accomplish the following:

(i) Obtain \( h_x \cong h_{x|cb} \) values as a function of quality \( X \) (see Fig. 16) for uniform temperature (or heat-flux) prescription at the wall over \( 0 \leq x \leq L \) (see Fig. 3a). Use such results, to propose correlations (see Eq. (32) in section 3).

(ii) Obtain key flow-physics details – such as film-thickness versus distance variation (Fig. 17a), liquid-vapor velocity profiles (Fig 17b), flow streamlines (Fig. 17e), etc.

(iii) Obtain and correlate, using non-linear stability approach (see, Naik et al. 2016 and Naik and Narain 2016), the criteria for transition from non-annular to annular flow-regimes as applied to suppressed nucleation flow-boiling (Ranga Prasad et al. 2016) – and seek its appropriate modifications by comparing the results with experimental results for nucleate-boiling dominant flows.

(iv) Compare \( h_{x|nb} \) results with low heat-flux and low mass-flux annular boiling experiments (Gorgitrattanagul 2017; Narain 2012) to obtain and correlate the estimates for locating the “point/zone” of onset of suppressed nucleation annular flows – which are likely to occur for a given fluid across a certain curve in the two parameter space of confinement number \( Co \equiv \sqrt{\sigma / (g(\rho_L - \rho_V)\Delta(x)^2)} \) and liquid-film Reynolds number \( Re_L(x) \equiv G(1 - X(x))/\mu_L \).

(v) By obtaining experimental values of \( h_x \) for annular flow-regimes along with theoretical/computational values of \( h_x = h_{x|cb} \), propose more reliable decomposition of \( h_x \) in the form of Eq. (61).
(vi) Use laminar liquid/laminar vapor or laminar liquid/turbulent vapor (discussed in the next section) simulations to obtain pressure-drop versus boiler-length (L) curves, compare the results with experiments, as well as with results from existing pressure-drop models (sections 3 and 4) to improve and propose new more accurate pressure-drop correlations for annular boiling.

**Fig. 16:** Plot of $h_x = h_{x,lb}$ versus quality $X$ as computationally obtained. The saturated annular flow-boiling in Fig. 3a is for a channel geometry $h = 2$ mm, $\Delta T = 10 \, ^\circ C$, fluid is FC-72, inlet quality $X_{in} \approx 0.65$, mass-flux $G = 13.98 \, \text{kg/m}^2\cdot\text{s}$, and $p_0 = 105.1 \, \text{kPa}$. 
(a) Film Thickness, $\Delta$ (m)

Distance along the length of the channel, $x$ (m)

(b) Distance from heated surface, $y$ (m)

Velocity, $U_1$ (m/s) @ $x = 0.02$ m (L = L or V)
Fig. 17: Flows conditions are same as in Fig. 16. (a) Plot of film-thickness versus distance (b) Liquid and vapor velocity profiles are indicated at \( x = 0.02 \) m (c) Plot of pressure profile at \( x = 0.02 \) m (d) Plot of temperature profile at \( x = 0.02 \) m (e) Streamline patterns. The mass flow rate between two adjacent (unequally spaced) streamlines are the same.

5.1.2 Moderate to high heat-flux and mass-flux cases and \( h_{x|cb} \) estimates

As per currently available estimates shown in Figs 7a-b, the \( h_{x|cb} \) estimates have additional uncertainty than those of \( h_x \) itself (Eq. (31)). The planned estimation of \( h_{x|cb} \) by CFD, under the assumptions of negligible/suppressed nucleation, laminar liquid, and turbulent vapor flows in the structure of Eq. (61) is expected to be a little bit more approximate as compared to laminar/laminar approach described in section 5.1.1. This is because of the need to use appropriate turbulence models for the vapor phase that is also consistent with the correct interfacial shear and vapor-side components of interfacial velocity. Even for assumed absence of nucleation and vapor streamlines (see Fig. 17e) suggesting existence of a “near interface” laminar layer in the vapor-phase that will always be adjacent to the thin laminar liquid flow, there is a possibility that - at high vapor-phase Reynolds numbers \( \text{Re}_V(x) \) – there will be an interaction between the typical indeterminate \( \text{interfacial} \) waves (or laminar turbulence, see Naik and Narain 2016; Naik et al. 2016; Narain et al. 2015) and \( \text{vapor-phase turbulence} \) that may change the interfacial shear values (from those obtained from laminar liquid flow and near-interface laminar vapor flow assumptions).
Therefore, for accurate estimation of $h_{x|cb}$ for such flows, it is proposed that, to begin with, it would be adequate to model interfacial shear estimates by retaining its values obtained from the laminar/laminar algorithm in Ranga Prasad et al. (2016) from preliminary considerations of laminar liquid flows and a near interface laminar vapor flow region. Then one could use these values for laminar-liquid and turbulent vapor simulations – with vapor turbulence modeled by an appropriate Reynolds Averaged Navier Stokes (RANS) or Large Eddy Simulation (LES) approach with user-provided and iteratively improved values of interfacial shear for the vapor-phase.

Besides the computational estimate for $h_{x|cb}$, recall that results from experiments continue to yield total HTC values through: $q_w|_{\text{Expt}}(x) \equiv h_x \cdot \Delta T(x)$, where $\Delta T(x) \equiv T_w(x) - T_{\text{sat}}(p_{\text{in}})$ and $T_w(x)$, like wall heat-flux $q_w|_{\text{Expt}}(x)$, is also experimentally measured boiling-surface temperature. With $h_x$ and $h_{x|cb}$ known, $h_{x|nb}$ can be estimated – though more approximately than in section 5.1.1 - through the defining relationship given in Eq. (61).

Also note that actual experimental values of film thickness $\Delta_{\text{Expt}}(x)$ is likely to be somewhat larger than the CFD predicted values of film-thickness $\Delta_{\text{CFD}}(x)$.

Pressure-drop predictions from such models, in conjunction with experimental measurements and existing pressure-drop correlations (see Sections 3 and 4), can also improve the existing pressure-drop correlations.

The above described approach for $h_{x|cb}$ estimates/correlations, along with further refined estimates for total $h_x$ obtained through the new experiments (and existing ones, as used in Kim and Mudawar 2013c) would again help elucidate heat transfer mechanisms through the simple model: $h_x = h_{x|nb} + h_{x|cb}$ – which is an $n = 1$ version of Eq. (33).

The proposed CFD would also help in providing improved understanding of the flow-physics such as the ones shown in the section 5.1.1 for laminar liquid and laminar vapor flows.

5.2 Annular flow-condensation

As discussed in sections 3 and 4, there are several empirical correlations for heat transfer coefficient $h_x$ as well as annular to non-annular flow-regime transition criteria. For the proposed innovative condenser and its design discussed in section 4, motion of the liquid film on a wetting (or hydrophilic) condensing-surface avoids the complexities of drop-wise condensation. Despite
this, correlation for $h_x$ and transition criteria from annular to non-annular flow regimes are currently lacking in quality and accuracy and need to be much more accurate for effective design calculations (such as the ones used in section 4). Again, as discussed earlier through Fig. 15, such improvements can be accomplished through CFD-and-experiments synthesis approach for improvements in $h_x$ and flow-regime maps. The status and need for such improvements are discussed next.

5.2.1 Low heat-flux and low mass-flux annular cases

First principles CFD-based accurate estimates (e.g. DNS for laminar condensate/laminar vapor cases in Fig. 3b) is now possible (Naik and Narain 2016; Naik et al. 2016; Narain et al. 2015) – at least for common refrigerants operating at $G < 20 \, \text{kg/m}^2\text{s}$ and $\dot{q}_{\text{w}} < 0.3 \, \text{W/m}^2$. The CFD/DNS for these laminar/laminar cases can accomplish the following:

(i) Obtain $h_x$ values as a function of distance $x$ and quality $X$ (see Fig. 18a-b).

(ii) Besides film-thickness versus distance variation with $x$ (Fig. 19a), obtain key flow-physics details for velocity, pressure, and temperature profiles (y-variation) of liquid and vapor phases at a representative $x$-location (Fig. 19b), annular to non-annular transition criteria by non-linear stability analysis (Naik et al. 2016 and Fig. 20), streamline patterns for shear-driven flow versus gravity driven flow (Fig 21), etc.

(iii) Use laminar liquid/laminar vapor pressure-drop predictions to improve pressure-drop prediction from those obtained from existing correlations – by comparison-based modification of the predicted results towards agreement with reliable experimental data.
Fig. 18: (a) Plot of the heat transfer coefficient vs non-dimensional distance (b) Plot of heat transfer coefficient vs quality (Run parameters: Fluid – FC-72, U = 1 m/s or G = 13.98 kg/m².s, p₀ = 105.1 kPa, ΔT = 5°C, channel height = 2 mm)
**Fig. 19:** (a) Plot of the two steady film thickness profiles shows negligible effects of transverse gravity. (b) Cross-sectional profiles for x-component of velocity \( U \), relative pressure \( p_I - p_0 \) and physical temperature \( T_I \) at \( x = 0.08 \) m location in Fig. 4a. The solid black line curve is for the absence of transverse gravity whereas the dashed grey line curve represents the presence of transverse gravity. (Run parameters: Fluid – FC-72, \( U = 1 \) m/s or \( G = 13.15 \) kg/m².s, \( p_0 = 101 \) kPa, \( \Delta T = 20^°C \), channel height, \( h = 2 \) mm)
Fig. 20: The plots show $\Delta(x^p, t \approx T)$ values for an unsteady simulation for a condensing flow ($g_y = -9.81$ m/s$^2$) for which $T = 0.15$ s for an initial film $\Delta(x^p, 0) = \Delta_{\text{steady}}(x^p) + \Delta'(x^p, 0)$ with $\Delta'(x^p, 0) \neq 0$. The solutions are shown for three different time-steps. The run parameters are same as in Fig. 19. The instability for $x^p > 0.06$ m corresponds to annular to non-annular transition any yields critical quality $X_{cr|A-NA}$ correlations as given in section 3.2.2.

Fig. 21: Streamline patterns of a steady flow for: (a) a horizontal channel, and (b) a 2° downward inclined channel for annular condensation. The color map shows the velocity magnitude distributions. The run parameters are same as in Fig. 19.
5.2.2 Moderate to high heat-flux and mass flux annular flow cases

Again CFD support for laminar liquid-turbulent vapor annular condensing flow predictions in section 4 is complicated for reasons similar to convective flow-boiling cases discussed in section 5.1.2.

For approximate estimation of $h_x$ for such flows, again it would be helpful to model interfacial shear, to begin with, by the same values that are associated with laminar liquid and laminar vapor (near interface regions only) assumption. It could subsequently be improved by multiplying the interfacial shear by a factor and then computationally adjusting the factor in a way that computational predictions of condensing-surface’s wall heat-fluxes (in case of temperature prescription) or wall temperatures (in case of heat-flux prescriptions) are in approximate agreement with experimental results for the flow-condenser – assuming experimentally measured/assessed values for both wall temperature and wall heat-flux will be available.

The above described approach, if further developed, can be used to obtain improved $h_x$ and pressure-drop correlations as well as improved understanding of the flow-physics.

6. High heat-flux advantages of superposing new time-varying "steady-in-the-mean" flow-physics on the steady innovative annular operation dealing with flow-boiling and flow-condensation

In recent experimental works dealing with innovative operations of flow-boiling and flow-condensers (see Fig. 3 and Kivisalu et al. 2014) at low mass-flux and heat-flux values, the thinness of the liquid films (see Figs. 3 and Fig 8) and their stability allowed superposition of large-amplitude standing waves on the interface – see Fig. 22c associated with pulsatile cases, as opposed to Figs. 22 a-b associated with steady non-pulsatile cases. The standing waves, as opposed to forward traveling waves, are formed in the experiments with help of introduction of pulsations in the incoming liquid and vapor flows and the enabling presence of vapor-phase acoustics resulting from the fact that the flow-boiler exit is closed in the flow direction (see Fig. 1). As a result of this standing wave’s superposition on steady solution, as troughs got closer to the wetting surface with increasing amplitude of the waves, local time-varying heat-flux measurements indicated very high heat-fluxes (see Figs. 23 a-b and Kivisalu et al. 2014). Consequently, both for annular flow-condensation and annular flow-boiling, the mean values of the measured heat-fluxes increased significantly (see Figs. 24 a-b).
Fig. 22: (a) Non-pulsatile steady innovative annular flow-boiling operations (Fig. 3a) at high heat and mass-fluxes (under use of common refrigerants) is dominated, as per section 4 estimates, by tiny nucleating bubbles (micron to sub-micron, represented by "dots") even for the steady annular regimes involving film-thicknesses in the range of 300 µm - 50 µm. (b) Non-pulsatile steady annular flow-boiling at low mass and heat-fluxes experience suppressed nucleation (arising from controlled recirculating vapor flow-rate at the inlet and associated low inlet film-thickness $\Delta_0 \sim 100$-300 µm. (c) The formation of long term (as opposed to short term forward moving waves formed by initial disturbances) and large amplitude standing waves on the interface is acoustically facilitated by the closed end exit-condition (because there are transverse openings in Figs. 3a-b) in the primary axial direction of the vapor. Large heat-flux enhancements are expected at the "troughs", see Figs. 23 a-b, because of enabling appearance of contact-line flow-physics – whether or not the flow is operating at low or high heat and mass fluxes.
Fig. 23: Time-varying heat-flux without (lower average) and with (higher average) imposed inlet liquid-vapor pulsations, measured by a flush-type heat flux meter. The dashed lines represent long-term (~ 40 minutes) averages.
Fig. 24: (a) Increase of average heat-flux (Kivisalu et al. 2014) with amplitude of imposed vapor fluctuations for annular flow-boiling (current and on-going experiments). (b) Increase of average heat-flux with amplitude of imposed vapor pulsations for annular condensing flows (Kivisalu et al. 2014).

The above described pulsatile flow realization approach consists of:

(i) Selecting proper inlet quality, exit quality, film-thickness range and length – as per design discussions in section 4.

(ii) Energy efficient ways of introducing inlet pulsations in the vapor and liquid flows as well as for recirculating the vapor. In case of high heat-flux boiling, the extra mechanical power available at the exit of the boiler is used to minimize the need for a recirculation aiding compressor.

(iii) Use of the closed end exit condition (Figs. 3 a-b) in the primary axial direction of the vapor flow – to generate acoustic signals in the vapor phase; so as to transform forward moving interfacial waves into standing wave-patterns shown in Fig. 22c, and as observed in experiments by Kivisalu et al. (2014).
(iv) Using the controlled thinness (50-300 μm) of liquid film to stabilize the film to impose pulsations as well as to reduce chances of liquid entrainment (even for high mass-flux cases).

Under aforementioned conditions of steady-in-the-mean pulsatile flow realizations, the heat-transfer enhancements shown in Figs. 24a-b are believed to result from the following hypothesis.

First, it is assumed that the flow-boiling enhancements in Fig. 24a may be but not significantly due to additional bubble (sub-micron diameters) nucleation at crests and reduced nucleation at troughs. This is because, if this was a primary reason, one would not observe high heat-flux enhanced values for the flow-condensation case in Fig. 24b. It is hypothesized that contact-line flow-physics and micro-scale film-flow physics combine to yield high heat-fluxes observed in Fig. 24. The hypothesis is depicted in Fig. 25 and supported by known similar results/hypothesis for pool-boiling, as shown in Fig. 26.

**Fig. 25:** (a) Near interface schematic of essential dynamics of the instantaneous spatial non-dimensional film thickness profile (δ ≡ Δ/h) associated with micro-meter scale wavy film flows (over a wetting surface) encountered in pulsatile flow-boiling (or flow condensation). The three (lower, central, and upper) intersecting circles in the figure respectively represent zones where nm-scale, μm-scale, and mm-scale phenomena interact with one another. Recirculating and through flows, at and near the troughs, at any given instant of time (b) Time-varying film thickness profile for location x = x* in Fig. 25a. The film-thickness time history at a trough location (x = x*) over an imposed time t_P = 1/f_P, where, f_P is the imposed pulsation frequency, is such that the crests and troughs “dwell” times weighted by instantaneous heat-flux values are dominated by the troughs (as in Fig. 23).

In Fig. 25:

- The three intersecting circles represent: effects of interaction zones of nm-scale, mm-scale, and macro-scale phenomena.
• Between the liquid-vapor interface and the solid-like adsorbed layer liquid-solid interface at the troughs, reduced pressure and slip-like reduced shear allow enabling micro-convective motion. The low-pressure at troughs is enabled both by the surface-tension and curvature effects near the vapor-liquid interface as well as by adsorbed (yellow) layer through negative tensile stresses (disjoining pressure) effects which occur near the lower solid-liquid interface. This low-pressure allows forward liquid motion and also enables micro-circulation and associated convective heat transfer enhancements. The low shear at troughs is similar to low shear expected at the rim of the pool-boiling bubbles in Fig. 26 — which enables realization of such large bubble departure frequencies (5 – 150 Hz), which further increases with increasing heat-flux despite the inertia of the stagnant liquid pool.

• Nucleation (of bubbles) may or may not be suppressed for thin annular boiling on wetting (or super-hydrophilic) surfaces – yet the contact-line physics advantages of nucleating bubbles (Kunkelmann et al. 2012), as shown in Fig. 26, is retained with the following differences. In Fig. 25: (i) the surface is always wetted with micro-layer (which is absent in Fig. 26 where the vapor is in contact with solid-like adsorbed layer), and (ii) there is continued forward flow through the “micro-layer” at the wave-troughs.

Fig. 26: Reduced pressure (disjoining pressure effects) and reduced shear (slip-like conditions) at the contact-line of nucleating bubbles enable high heat-flux values at the contact-line (due to reasonably high frequency, typically at about 5-200 Hz, alternating and vigorous recirculating motion at the nano-micro-macro layers of liquid in the presence of solid-liquid and liquid-vapor interfaces).
Fig. 27: The broad new flow-loop structure for innovative flow-boiling experiments using water.

The above described flow-physics for pulsatile operations of annular flow-boilers and flow-condensers – along with suitable design modifications for the non-pulsatile flows in section 4; and some hardware modifications for the flow-loop design (see Kivisalu et al. 2014. and Gorgitrattanagul 2017) – enable development of high heat-flux devices embedded in proper flow-loops equipped with suitable flow-controls (see, e.g., Fig. 27 for a flow-loop being used for science experiments (Sepahyar 2018) that are capable of removing heat (from the boiling-surface) at desired 500 - 1000 W/cm² levels.

7. Summary

This article, with the help of design examples in section 4, summarized the current knowledge base outlined in sections 2 & 3. Sections 5 & 6, summarized both the current status and recommendations for further development of the relevant science and engineering applications associated with realization of annular flow-boiling and flow-condensation.

Acknowledgement

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# Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Cross-sectional area, m²</td>
</tr>
<tr>
<td>Bl</td>
<td>Boiling Number ($\dot{q}<em>w/\dot{m}h</em>{fg}$)</td>
</tr>
<tr>
<td>Bo</td>
<td>Bond number</td>
</tr>
<tr>
<td>C</td>
<td>Lockhart-Martinelli constant</td>
</tr>
<tr>
<td>$C_p$</td>
<td>Specific Heat, J/kg-K</td>
</tr>
<tr>
<td>Dₜ</td>
<td>Hydraulic diameter, m</td>
</tr>
<tr>
<td>f</td>
<td>Friction factor</td>
</tr>
<tr>
<td>Frₓ</td>
<td>Froude Number in x-direction, ($</td>
</tr>
<tr>
<td>Frᵧ</td>
<td>Froude Number in y-direction ($</td>
</tr>
<tr>
<td>G</td>
<td>Mass-flux, kg/m²s</td>
</tr>
<tr>
<td>gₓ</td>
<td>Gravity component in x-direction, m/s²</td>
</tr>
<tr>
<td>gᵧ</td>
<td>Gravity component in y-direction, m/s²</td>
</tr>
<tr>
<td>h</td>
<td>Height of the channel, m</td>
</tr>
<tr>
<td>h_{fg}</td>
<td>Heat of vaporization, J/kg</td>
</tr>
<tr>
<td>Ja</td>
<td>Liquid Jakob Number, ($C_pL\Delta T/h_{fg}$)</td>
</tr>
<tr>
<td>k</td>
<td>Conductivity, W/m-K</td>
</tr>
<tr>
<td>L</td>
<td>Length of the channel or test-section, m</td>
</tr>
<tr>
<td>$\dot{M}_{in}$</td>
<td>Total mass flow rate, kg/m²s</td>
</tr>
<tr>
<td>$\dot{M}_L$</td>
<td>Liquid mass flow rate, kg/m²s</td>
</tr>
<tr>
<td>$\dot{M}_v$</td>
<td>Vapor mass flow rate, kg/m²s</td>
</tr>
<tr>
<td>p₀ or pᵢₜ</td>
<td>Steady inlet pressure (also pᵢₜ), kPa</td>
</tr>
<tr>
<td>P</td>
<td>Mechanical Power, W</td>
</tr>
<tr>
<td>Prₗ</td>
<td>Liquid Prandtl Number, ($\mu_LC_pL/k_L$)</td>
</tr>
<tr>
<td>$\dot{q}_w^-$</td>
<td>Mean wall heat flux, W/m²</td>
</tr>
<tr>
<td>Reₜ</td>
<td>Reynolds number representing non-dimensional ($GDₜ/\mu_v$)</td>
</tr>
</tbody>
</table>
\( S \) Suppression factor

\( \text{Su} \) Suratman number, \( (\sigma \rho_v D_h / \mu_v^2) \)

\( t \) Time

\( \bar{T}_w \) Mean wall temperature, °C

\( T_{\text{sat}}(p_0) \) Saturation Temperature at pressure \( p_0 \), °C

\( u \) Velocity, m/s

\( \text{We} \) Liquid Weber Number \( (\rho\text{L}U^2D_h/\sigma) \)

\( X \) Vapor quality

\( \hat{x} \) Non-dimensional distance \( (\equiv x/D_h) \)

\( x_A \) Non-dimensional length of the annular regime

\( X_{\text{tt}} \) Lockhart-Martinelli parameter

**Greek Symbols**

\( \mu \) Viscosity, kg/m-s

\( \rho \) Density, kg/m\(^3\)

\( \epsilon \) Void fraction

\( \Phi_g \) two-phase multiplier

\( \sigma \) Surface tension, kg/s\(^2\)

\( \Delta \) Physical value of liquid film thickness, m

\( \Psi_q \) Non-dimensional heat-flux

\( \theta_w \) Non-dimensional temperature

**Subscripts**

\( \text{cb} \) Represents “convective boiling”

\( \text{L} \) Represents liquid phase of the flow variable

\( \text{nb} \) Represents “nucleate boiling”

\( \text{V} \) Represents vapor phase of the flow variable
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