REVIEW OF NUMERICAL STUDIES OF BOILING TWO-PHASE FLOW IN SMOOTH TUBES

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ABSTRACT
The rapid and continuous depletion in the available energy resources nowadays has resulted in the quest for alternative sources of energy and more energy efficient processes. This review article present a broad and critical review of past researches on flow boiling and condensation, basically to provide a comprehensive understanding of both boiling and condensation heat transfer (HT) process as they are important phenomena that find application in various areas in the thermal science field and industry which includes; power generation, refrigeration and air conditioning, nuclear reactors, chemical engineering, high-power electronics component cooling and so on. An in-depth understanding of this phenomenon will help to know how to attain very high heat transfer rates with small variations in the surface temperature. Consequentially, this results in better energy efficient process with huge reduction in system size, volume and energy consumption, hence significant fall in the heat energy required to undertake the process. Fundamental parameters affecting these phenomena such as the classification of channel and their applications based on their properties and advantages, flow patterns and heat transfer mechanisms, heat transfer coefficient and critical heat flux are fully discussed. Finally, recommendation are made to provide clue for future researches in this area.

Keywords: Boiling and condensation heat transfer; Heat transfer mechanisms, Heat transfer coefficient; Critical heat flux; Flow patterns; Numerical modelling; Phase change.

INTRODUCTION
The rapid and continuous depletion in the available energy resources nowadays has resulted in the quest for alternative sources of energy and more energy efficient processes. The International Energy Agency (IEA) predicts that about thirty percent (30%) of the global energy utilization will be provided from solar energy by 2060 under favourable circumstances. Solar-based renewable energy has the potential to provide electricity and hot water (which can be used for heating and ventilating air conditioning systems) simultaneously depending on the requirements and technologies deployed [1]. Typical solar power system consists of concentrator, absorber and thermal storage which is based on the concentration of sunlight. There are three different concentration solar power systems: parabolic trough systems; solar power tower; parabolic dish technology using a Stirling motor. A large area of sunlight can be captured and focused onto a smaller area (absorber) using the concentrating collector (for better performance). The captured solar radiation is reflected to the absorber located at the focal point of the collector (solar power tower, parabolic trough or dish, linear Fresnel, etc) [2]. The stored heat energy vapourizes the thermal fluid passing through the evaporator. The evaporated fluid is expanded in the turbine for electricity generation. The low pressure vapour coming out from the turbine flows to the condenser. A new cycle begins as the condensed working fluid is pumped back to the evaporator (Figure 1).
Thermal storage in the system homogenizes the unsteady temperature of the fluid before entering the turbine, thus making the fluid's temperature evenly distributed. The stored heat energy makes the system operational at all times depending on its capacity even when there is no sun [2, 5].

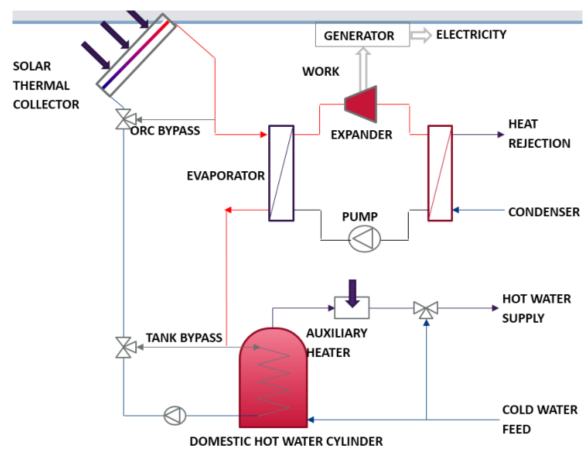

Figure 1: Schematic diagram showing the Thermodynamic phase change heat-engine cycle for distributed power generation using concentrated solar power plant [4].

The working fluid with appropriate thermal properties, the solar insolation, field surface and the heat storage capacity have been identified as important parameters for higher solar system efficiency.
For working fluid; water has found wide application in steam engine but it is only appropriate for high temperature application [4]. This brought about the consideration, development and utilization of organic fluid in organic Rankine cycle (ORC) for power generator most especially at lower critical temperature and pressure hence lesser heat required for evaporation compared to water, the more reason for its usage in renewable or waste energy systems [5]. Its evaporation takes place within the vapour region, hence no need for superheating, low freezing point and high temperature stability, low environmental impact;

ozone depletion potential (ODP) ≤ 0, no global warming, non-corrosive to blades, and non-toxic, adequately available and cheap, low pressure drop, only slight temperature difference between the condenser and the evaporator making a simple single stage turbines practicable [4].

Also, the quality of the heat added process (Evaporation/boiling) and the heat rejection process (condensation) determines the efficiency of the organic Rankine cycle. Flow boiling has been carefully selected for investigation in renewable energy power transmission because of its rapid rise in the heat transfer coefficient with increasing heat flux in nucleate boiling which does not depend on the mass flux [6], higher steam specific volume and rapid heat dissipation. It finds wide applications in thermal power generation, food preservation, nuclear reactors, and high-power electronics component cooling and so on.

A critical look is hereby given to flow boiling and condensation principally to optimize the phenomenon in order to attain very high heat transfer rates, huge reduction in system size and energy consumption in undertaking the process. To attain this, efforts must be made to enhance the performance of the heat transfer devices (evaporator and condenser) in the system. Hence, the need to conduct fundamental research using advanced numerical methods on flow with phase-change (boiling and condensation) heat transfer in the presence of unsteadiness (solar radiation variations) and to use the data to develop system models that can describe transient plant performance.

Evaporation process in a heat exchanger

Buongiorno describes boiling as a heat transfer process that involves phase change from liquid to vapour state at the solid liquid boundary at constant temperature. It is characterized by rapid formation of bubbles at the solid surface with temperature T_s higher than the fluid saturation temperature T_{sat} [7]. The formed bubbles at certain sizes detach from the solid surface and condense after dissipating the acquired heat energy to other liquid particles above the surface.

Boiling can be classified as pool boiling and flow boiling, subcooled boiling and saturated boiling depending on the bulk fluid motion. Further classification includes nucleate boiling, natural convection boiling, Film boiling and Transition boiling. Nucleate boiling is the commonest however, because of the wide industrial application of film boiling in areas like cryogenic, steel mills, etc it has become the focus of many boiling studies [8].

Subcooled flow boiling occurs in many process and engineering applications such as in nuclear reactors, steam generators, refrigeration and air-conditioning systems, etc where large heat transfer coefficients for effective heat exchange is required. Subcooled flow boiling is the evaporation of subcooled liquid near a heated surface at a bulk flow temperature little lesser than the local saturation temperature. The analysis of this process is complex and challenging basically due to the thermodynamic non-equilibrium between the two fluids phases [9].

Evaporation as a heat transfer process differs from boiling in that the phase change happens in the liquid-vapours interface. It takes place when the liquid's saturation pressure exceeds the vapour pressure at a constant temperature with the addition of large amount of heat (latent heat of vapourization) [7].

Numerical Simulation in Flow Boiling

Despite the numerous applications of flow boiling, there is yet no established numerical method for solving it. There are varying degree of accuracy and slight inconsistences in results obtained from different researchers using different methods. Constitutive models that explains the gas-liquid phase interaction are used to qualify CFD codes for two-phase flows [1]. For flow boiling involving bubbly flow, the forces causing bubble formation, coalescence and disintegration are given full consideration coupled with the heat transfer acting along the liquid-gas interface. For a two fluid approach, the drag forces describes the momentum exchange in flow direction while the non-drag forces acting perpendicular to the flow direction determines the flow structure [10].

New numerical methods was first introduced by Son and Dhir [11] and Juric and Trygvason [12] to compute flow boiling using Front Tracking algorithm and Level Set Method for the interface description. Volume of Fluid (VoF) numerical methods was developed in 2000 by Welch and Wilson [13]. Jamet *et. al.* [14] applied Second Gradient Method to simulate bubbles formation in flow boiling. Esmaeeli and Tryggvason utilized a general methodology for 3D computations in complex geometries [9]. Level set method was used with Volume of Fluid method by Tomar *et. al.* [15] to analyze flows boiling. Impacting boiling droplets on hot walls was estimated using the boundary layer subgrid scale treatment proposed by Ge and Fan [16, 17]. Gibou *et. al.* [18] developed sharp interface method using Ghost Fluid Method for film boiling. Tanguy *et. al.* [19, 20] developed a sharp interface method for droplets visualization using the Ghost Fluid Method. Sato and Niceno [21] developed sharp interface numerical methods using Front Tracking method. 3D computations of the nucleate boiling on a horizontal surface and saturated film boiling on a horizontal cylinder was presented by Son and Dhir [22, 23]. Many of these numerical work shows similarities with Legendre and Magnaudet [24] analytical work. However, the evaluation of the accuracy and performances of these numerical methods are still very difficult. Tanguy *et. al.* [20] demonstrated that the way the mass flow rate for flow boiling is computed has a significant effect on the global accuracy. Phase change process is also a concern in the computation of expansion flow. He also established that the Continuous Surface Force approach is inadequate to accurately determine the bubble expansion rate as a result of the velocity field spreading out at the phase interface where the interfacial source terms are smoothed across interface. Hence, a sharp interface method using the Ghost Fluid Method was examined and it was found to be successful in computing the bubble expansion rate.

Numerical Simulation in Condensing Flow

Margolin *et. al.* [25] first used VOF method for the numerical simulation for condensation. From then, several researches employed the VOF method to model condensation flow. Together with the VOF multiphase flow model, De Schepper *et. al.* [26] used a piecewise linear interface calculation (PLIC) to reconstruct the interface of air-water flow in each computational cell. Their simulated horizontal flow regimes showed good agreement with the Baker chart, [27]. Lee *et. al.* [28] utilized the in-built VOF method of FLUENT to compute the conservation equations of both liquid and vapour phase while considering the mass transfer between the two phases. By applying a numerical backward finite difference scheme, Saffari and Naziri [29] predicted condensation heat transfer coefficients during stratified flow in an inclined tube. To simulate the bubbly flow regime and model source terms in VOF governing equations, Jeon *et. al.* [30] also used the VOF model of FLUENT together with a user-defined function. Both Liu *et. al.* [31] and Aghanajafi and Hesampour [32] studied the filmwise condensation numerically. The former study employed the PLIC method to reconstruct a sharp interface while taking into account the surface tension. Based on a finite volume approach, Groff *et. al.* [33] presented a two-phase model for film condensation from a downward turbulent flow of vapour-gas mixtures in a vertical tube. Da Riva and Del Col [34, 35] presented a three-dimensional VOF simulation of R134a condensing flow. Based on finite volume formulation of the Navier-Stokes and energy equations, Nebuloni and Thome [36] presented a model for laminar annular film condensation in mini- and micro-channels. Meier *et. al.* [37] presented a technique for including surface tension in PLIC-VOF methods by using an estimator function. The VOF method has been used to model condensation heat transfer and fluid flow characteristics in micro-channels by Ganapathy *et. al.* [38]. The methods of both Gibou *et. al.* [18] and Jamet *et. al.* [14] gave excellent qualitative results with respect to boiling flows. However, their use for condensing flows are yet to be explored.

The heat transfer rate in boiling and condensation process has been found to strongly depend on the channel diameter, heated length, inclination, degree of roughness and materials using different working fluids of varying mass velocity, heat fluxes, saturation temperature and pressure. The following are measured consequentially to understand the flow and its properties; flow patterns, heat transfer coefficient, pressure drop, Critical Heat Flux, liquid entrainment and Void fraction.

Flow patterns

The distribution of individual fluid phases describe the configuration of the flow pattern of a multiphase flow which is being determined by the interaction of gravity, surface tension, evaporation, inertia, shear and bubble nucleation forces in liquid-vapour interface [39]. The understanding of the flow patterns is crucial; it affects design parameters and determines the heat transfer coefficient, critical heat flux and pressure drop in a system. Some certain flow pattern conditions can damage the equipment [40, 41]. Stratified or churn flow pattern is observed in a macrochannel during two-phase flow in evaporators at lower total mass fluxes. As the diameter reduces in horizontal channel, gravitational effects become negligible, while the inertia force becomes significant to form annular flow pattern. Heat flux also becomes more significant in two-phase flow pattern at microscale [42]. Figure 2 [43] shows a schematic of evaporative flow patterns in a horizontal channel. The reverse flow patterns are applicable to the condensation process in a horizontal channel. Below is the description of different flow patterns experienced during evaporation.

Bubbly flow

This is characterized by random motion of the gas bubbles in the bulk flow [39] detached from the channel wall due to wall superheat during evaporation process. Bubble flow soon become elongated bubbles as the channel's diameter reduces for adiabatic flow. At very low fluid quality, bubbly flow were seen in the subcooled region in rectangular channels of D_h = 0.48 mm and 1.86 mm [44]. At low mass fluxes and channel diameter of D_h = 0.5 mm using R-134a and R-245fa, single streams of bubbles grew in size [45]. Chen *et. al.* [46] observed experimentally the significance of channel diameter and mass fluxes over vapour qualities and two-phase flow pattern for flow regime transition boundaries using R134a. Figure 3 shows the flow pattern obtained for 1.10 mm and 4.26 mm respectively. However, the significance of types and properties of working fluid to flow boiling and condensation becomes obvious as Chen *et. al.* [46] work using R134a shows full contradiction to the condensation study of Coleman *et. al.* [47] using R236fa in 0.5mm channel and evaporation study of Revellin *et. al.* [48] using R245fa in 0.8 mm channel. However, at larger diameter, D_h = 100 μm, bubbles disappears for air-water mixture [49]. As heat fluxes increase with very short slug lengths, bubbly to slug flow transition was experienced [50]. At a higher channel diameter of 400 μm and above, the bubbly flow effect fades out as the diameter reduces below 100 μm for microchannels [51].

Explosive bubble growth

This is caused by continuous and large bulk liquid superheat in flow boiling in microchannel [52, 53, 54, 55]. This results into flow instabilities and reversal. These problems are solved by inserting nucleation cavities of radii satisfying criteria for nucleation and also by placing restrictors at flow inlet.

Elongated bubble/slug flow

This is prevalent in microchannels and it often appears after explosive bubble growth with continuous wall superheat. It is characterized by an intermittently stretched merged bubbles bounded by threadlike liquid film attached to the channel wall. It fades out as heat fluxes increases resulting in annular flow regime formation [56]. Since the annular flow is replaced by slug flow along a condensation length, the condensation heat transfer coefficient decreases gradually [57].

Injection flow

Injection flow, unique to condensation flow in micro-channels has been studied by Quan *et. al.* [58]. Quan described the injection flow as an expansion stage, where the vapour ligament attached downstream of the annular flow grows radially and a detachment stage, where the vapour ligament breaks and Taylor bubbles begin to appear.

Annular flow

This is characterized by a continuous liquid film attached to the wall forming an annulus around the lighter fluid as a result of continuous wall superheated causing the formed downstream slug to break off [59]. Park *et. al.* [60] introduced the annular-wavy with and without gravity influence. In a condensing flow, the annular flow seems to deliver the highest heat transfer coefficient.

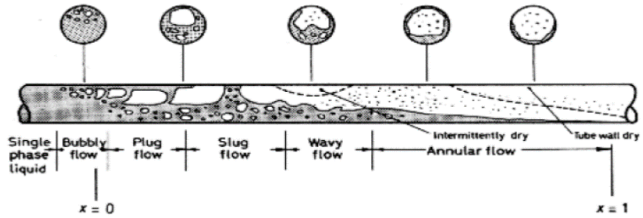

Figure 2: Illustration of the sequence of two-phase flow patterns during evaporation **[43]**

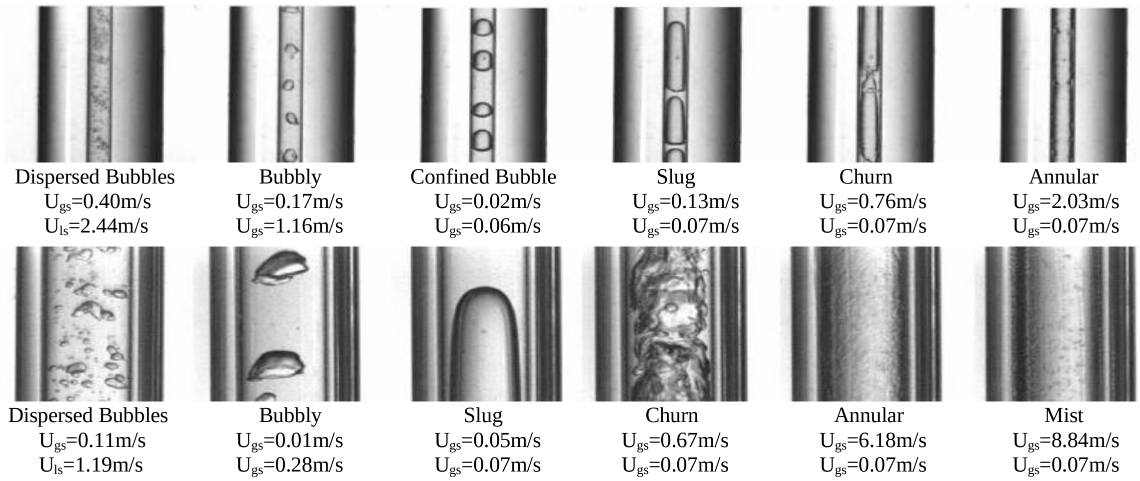

Figure 3: Vertical two-phase flow regimes from Chen *et. al.* [46] at 10 bar for the (a) 1.10mm and (b) 4.26 mm channel.

For proper numerical analysis of multiphase flow, a prior knowledge of the flow regime peculiar to the flow of interest based on the flow condition is important. Several flow pattern prediction maps has been developed for some working fluids under both diabatic and adiabatic conditions in different channels using numerical simulation tools. Adiabatic flow pattern maps was proposed for macrochannel [61, 62, 63] while diabatic flow pattern maps was proposed [64, 65, 66] using different flow properties. However, there is still need to develop more flow pattern maps for simple and complex geometries in order to enhance understanding of heat transfer, critical heat flux, pressure drop and void fraction in a unified global manner [67]. The significant effect of inertia and surface tension forces over buoyancy for transition flow under microgravity condition was observed in adiabatic two-phase flow [68, 69, 70]. Hence, Weber number was proposed in a model to serve as correlation for the flow regime transition. The proposed flow pattern map were divided into three distinct zones (Figure 4); annular flow regime (a region dominated by the inertia force), bubbly and slug regimes (a region dominated by surface tension force), and transition zone (a region of equal dominance of both surface tension and the inertia forces) [71].

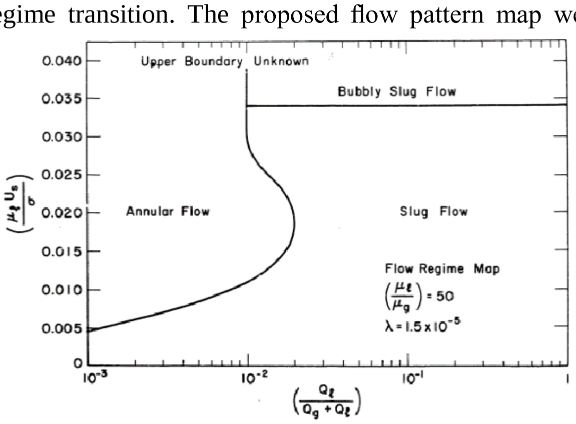

Figure 4: Two-Phase Adiabatic Flow Pattern Map [71].

Further to this work, a flow pattern map subdivided into four region was developed (Figure 5); that is surface tension dominated (bubby, plug/slug), Inertia dominated region 1 (annular), Inertia dominated region 2 (dispersed) and transition boundary region [55].

Park *et. al.* [60] also constructed two flow regime maps on condensation experimental data based on dimensionless parameters (Weber number, dimensionless superficial vapour velocity and Martinelli parameter) taken into account the influence of gravity (Figure 6 & 7), hence predicting the flow transition region more accurately.

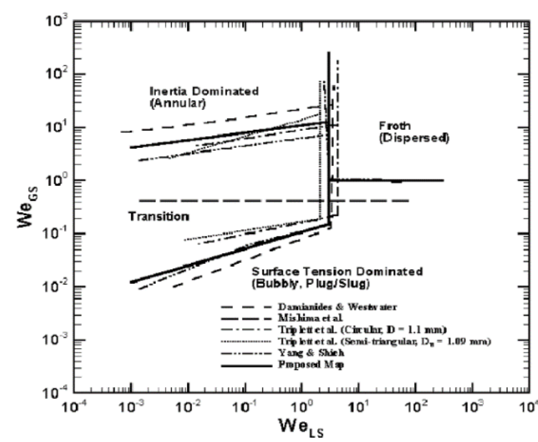

Figure 5: Flow pattern map comparison for circular and near-circular channels; $D_h \leq 1$mm [72].

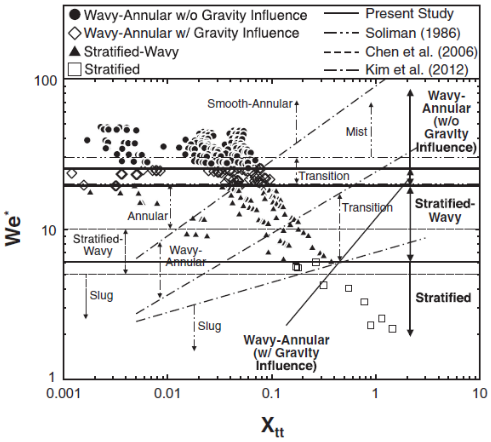

Figure 6: Flow patterns boundaries based on modified Weber number and the Martinelli parameter [60].

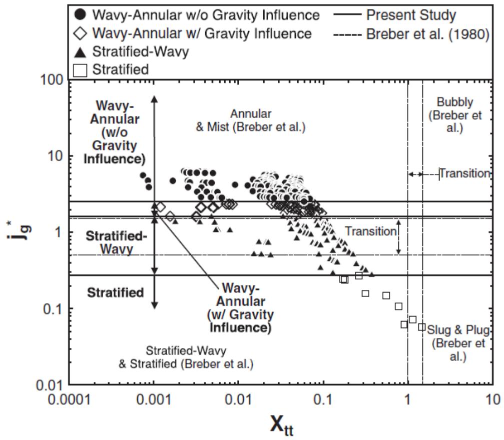

Figure 7: Flow patterns boundaries based on dimensionless superficial vapour velocity and the Martinelli parameter [60].

Pressure Drop

Frictional pressure drop experiments for both diabatic and adiabatic conditions in small diameter channels have been carried out by various researchers using a different type of working fluid (organic and inorganic) with circular and rectangular channels for single and multi-channel configurations as low as 100 x 100 μm cross-sectional [42]. Similar pressure drop trends were observed for both micro- and macro-channels. Pressure gradient attains a peak value as vapour quality increases, at decreasing mass velocity and increasing saturation temperature [73] whereby the maximum (turning) point attained is equivalent to the transition point from annular to mist flow [74] (Figure 8). It also varies with the working fluid properties based on relative liquid-vapour phase density difference and the fluids' viscosities. As the channel length increases at lower vapour quality, the decrease in the heat transfer coefficient becomes steeper [75]. Though this result may be misleading for longer channel consequential to the linear pressure drop profile adopted [42].

Correlations developed for pressure drop in macroscale channel cannot predict for microscale channel basically due to the difference in fluid behaviours consequential to the forces acting on the flow. Tibirica and Ribatski [40] highlights the reasons for the differences in the results on pressure drop by different authors as inadequate information on the roughness ratio of the inner surface of the channels, inadequate knowledge of the experimental conditions, no experimental result validation using single-phase flow pressure drop and energy balance, poor, inconsistent and non-uniform assumptions during evaluation. Frictional pressure drop profile is obtained by deducting acceleration (momentum) component from the total experimental pressure drop. The accelerational pressure drop for a macro-scale predictive methods is a function of the superficial void fraction but it is relevant even under adiabatic conditions owing to the flashing effect in microscale channel.

Cioncolini *et. al.* [76] analysed series of available data from different researchers on adiabatic pressure drop for channel radius ranging from 0.252mm to 15.85mm. Based on this, a mechanistic model that includes the Weber number of gas core droplet, the gas core velocity effects, average thickness of the liquid film and the liquid entrained fraction was proposed to estimate frictional pressure drop for both macro- and micro-scale conditions in annular flow. This shows good correlations with predictive methods and experimental data from previous authors. It was also found correct for multi microchannel data of Costa-Patry *et. al.* [77].

Pressure drop in micro-channels was also studied experimentally by Kim *et. al.* [57]. Comparing with existing experimental data, they discover that two-phase homogeneous models predicted data with more accuracy than the separated flow models. Authors concluded that condensation in mini/micro-channels is much closer to adiabatic flows than boiling flows. This is due to the significant droplet entrainment taking place in flow boiling and not in condensation and adiabatic flows. Based on experimental results, Lips and Meyer [78] showed that pressure drop depends on tube orientation, vapour quality and mass flux.

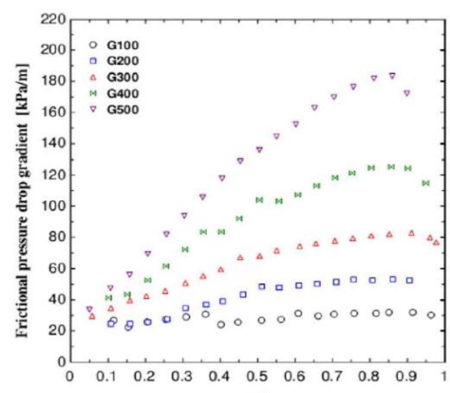

Figure 8: Frictional pressure drop vs vapor quality; the effect of mass velocity using R245fa, 1.1 mm I.D. channel and T_{sat}=41°C. [74]

Critical Heat Flux (CHF)

This is also known as boiling crisis, burn-out heat flux, departure from nucleate boiling, or dryout depending on the circumstance at which it occurs [79]. It gives the upper thermal limit for safe operation of any boiling device. At CHF, surface/wall temperature increases rapidly consequential to sharp decrease in the heat transfer rate and efficiency hence a localized over-heating and an irrecoverable damages of the device. This problem makes accurate analysis of flow boiling CHF very crucial for reliable designs of heat transfer systems and its safety.

Several correlations has been proposed by various researchers for CHF prediction in small to micro-diameter channel using empirical correlations based on dimensionless numbers obtained from fluid properties, channel diameter, mass velocity, heated length and subcooling at the channel inlet. Water and macroscale channel convective flow boiling databases has widely been used for distinct CHF predicting correlation for subcooled and saturated condition. Recently, data for other heat transfer fluids and smaller diameter channel was published [42].

For saturated boiling, CHF occurs at the outlet of the test section when thermodynamics vapour quality is higher than zero '0', at this point the liquid film experiences dryout. Thinner liquid film in microscale channel causes CHF even at low flow rates.

For subcooled boiling, CHF is attained at high degrees of sub-cooling at test section's inlet at high mass velocities and low heated length to channel diameter ratio [72].

Critical heat flux shares direct relationship with mass flux and inlet sub-cooling but inverse relationship with the heated length to diameter ratio and the saturation temperature. It varies also with the fluid properties. The decrease in CHF at higher saturation temperature is consequential to the decrease in the latent heat of vaporization [80, 81].

Karayiannis *et. al.* [75] compared correlations by different authors graphically using R134a, 1mm channel diameter and 150mm heated length in order to understand the influence of mass flux, diameter and pressure on CHF. It was observed that all correlations agreed that CHF increases with increase in mass flux however with different slopes. Qi *et. al.* [82] and Qu and Mudawar [83, 84, 85, 86] and work predicts values ten times higher than other researchers''

Inlet subcooling was found to be insignificant for multichannel configuration compared to single channel due to flow instability observed before the critical heat flux in the multichannel. This yields turbulent mixing of the vapour and the incoming liquid making the liquid temperature to rise towards saturation [75]. Except for the correlation of Kosar *et. al.* [87], all correlations predicted direct relationship between the CHF and the channel diameter. All correlations shows an inverse relationship between system pressure and the CHF. Fluid properties constitute to the influence of system pressure on the CHF. Pressure increase yields gas density increases, liquid density decreases, gas-liquid density ratio increases, latent heat decreases, surface tension decreases and gas-liquid viscosity ratio increases for R134a. This shows inverse relationship between CHF; and gas-liquid density and viscosity ratio and direct relationship with latent heat and surface tension [75].

Revellin and Thome [88] used the conservation of mass, momentum, energy, the Laplace–Young equation and a semi-empirical expression for the interfacial wave's height at stable and saturated conditions in an evenly heated circular microchannels to develop a theoretical model to determine CHF during annular flow.

Tibiriçá *et. al.* [82, 83] presented experimental results showing the effect of channel geometry (round and flat channels) on CHF. At constant equivalent length/overall heat transfer area, CHF was found not to be affected by the aspect ratio of the channel.

Heat Transfer Coefficient

A mechanistic model for obtaining time-averaged local heat transfer coefficient for specific period in a system was first presented by Thome *et. al.* [89]. This model covers the three heat transfer zones for unsteadiness in the local heat transfer coefficient in the evaporation of different flow pattern in flow boiling in microchannel. From this algebraic turbulence model, Cioncolini and Thome [90] developed a model for heat transfer coefficient evaluation in adiabatic and evaporating annular flows. Further to these works, Costa-Patry *et. al.* [91] developed a unified model for heat transfer coefficient evaluation in microscale channel by merging the three heat transfer zones model of Thome *et. al.* [89] with the algebraic turbulence annular flow model of Cioncolini and Thome [90]. This model worked relatively well for multiple channel without any change in the experimental constant.

Tibirica & Ribatski [40] gave a fine summary and schematics diagram of heat transfer coefficient behaviours against vapour quality in flow boiling for microchannel in terms of heat and mass flux, saturation temperature and internal diameter. He also stated that surface roughness affects heat transfer coefficients. It increases heat transfer coefficient until dryout, even at negligible variation in the mass velocity but increasing heat flux.

For condensation heat transfer coefficients, Thome *et. al.* [92] presented a prediction correlation for annular, intermittent, stratified-wavy, fully stratified and mist flow in a horizontal tube where he represented the fluid as a uniform truncated annular film with the same angle of stratification and liquid cross-sectional area in order to achieve a uniform film thickness in stratified-wavy and fully stratified flows.

The general expression used for local condensing heat transfer coefficient constitutes both the convective heat transfer coefficient in the axial flow and the falling film condensation heat transfer coefficient. A parameter representing the effects of interfacial roughness was also added to improve the accuracy of the model. In the overall, the model predicts 85% of the refrigerant data well to within ±20% and also the heat transfer coefficients for each flow regimes were accurately predicted. Compared to [92], Cavallini *et. al.* [93] developed two simple heat transfer coefficient equations based on a flow regime criterion between temperature difference dependent flow regime and temperature difference independent flow regimes. Hence, it cannot be used for a panoply of fluids.

Lyulin *et. al.* [94], Lips and Meyer [78, 95] and Meyer *et. al.* [96] demonstrated that the optimum inclination angle to the highest heat transfer coefficient is 15˚ to 30˚, downward flow. Irrespective of the tube orientation and inclination, Shah [97] presented a correlation based on a two-phase multiplier to determine the annular heat transfer coefficients.

Compared to the above-mentioned condensation heat transfer studies, only Kim *et. al.* [57] utilized a square-shape micro-channel to investigate condensation heat transfer coefficient where the general trend of increasing heat transfer coefficient with increasing mass velocity was observed. The heat transfer coefficient decreases gradually basically due to the thickening of the liquid film and the replacement of the annular flow by the transition or slug flow along the condensation length. The authors compared their experimental heat transfer data with existing correlations designed for macro-scale and mini/micro-channels. The macro-scale correlations provided better predictions of the condensation heat transfer coefficients than the mini/micro-channels correlations. In addition, they stated that the uniform liquid film thickness is not affected by the rectangular channel corners due to the low surface tension of FC-72.

Void Fraction

This is also a significant factor for predicting system's fluid inventory, the pressure drop, average liquid film thickness and the velocity of individual phases in two-phase analysis [40].

It is determined from flow pattern, channel geometry and orientation, flow instabilities and liquid entrainment fraction. Flow pattern is also significant in choosing the method of void fraction measurement. It can also be obtained from slug velocities, vapor phase superficial area and the total area of the two-phase flow ratio and bubble measurement from the analysis of captured images of two-phase flow. Linear relationship between mean superficial velocity for two-phase flow and the elongated bubble velocity has been graphically demonstrated.

Most studies based their void fraction measurements methods on drift flux model or curve fitting of data obtained from experiments. This method is limited in its application.

Only few publications exist for void fraction measurement in small diameter channel for flow boiling under adiabatic condition. A table summarizing previous studies on void fraction measurement for hydraulic radius ranging from 0.010 mm to 2.49 mm in single-port channel was presented by Tibirica and Ribatski [40] stating the working fluid, measurement technique and other experimental condition, though with the exception of works on multi-port channels.

Void fraction measurement incur significant error in annular flows basically due to extremely thin liquid film thickness and light refraction effects significant error.

In a well-cited paper by El Hajal *et. al.* [98], a new flow pattern map and a newly defined logarithmic mean void fraction method to calculate vapour void fraction for a wide range of pressure was presented. Later, Suliman *et. al.* [99] improved the stratified-wavy transition line of El Hajal *et. al.* [98] by increasing the Weber-to-Froude ratio exponent and the constant term in the latter transition line equation, resulting in a new transition line. With this slight improvement, the mean deviation shifted from 15% to 6% in the newly developed correlation, with all data points lying within ±25% deviation lines, thus predicting heat transfer coefficients of low mass fluxes very well.

Cycle Efficiency; Quality of Power Generation

The efficiency of an electric power plant is given as the ratio of useful electricity output from the generating system per unit time to the energy input/supplied from source within the same time. The efficiency of the ORC [100] is given as:

$$\eta_{th} = \frac{Net\ Workdone}{Q_{in}} = \frac{Q_{in} - Q_{out}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}}$$

This equation shows the huge dependency of the efficiency of the cycle on the performance of the evaporator (amount of heat been absorbed/acquired, Q_{in}) and the performance of the condenser (amount of heat rejected, Q_{out}). Hence, the need for an advanced numerical analysis of flow with phase-change heat transfer in the presence of unsteadiness (solar radiation variations) in order to improve/enhance the heat transfer processes for optimized and better heat transfer rates and to use the data to develop system models that can describe transient power plant performance.

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