MODELING OF NUCLEATE BOILING HEAT TRANSFER OF A STAGNATION-POINT FLOW IMPINGING ON A HOT SURFACE

by

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A model using modified superposition approach is developed to predict the rate of heat transfer in the stagnation region of a planar jet impingement boiling on a hot flat surface. The total heat flux in this model is based on the combination of the single phase forced convection and nucleate pool boiling components. The single-phase component is calculated by using similarity solution approach. The applicability of the model is investigated on the boiling curves under conditions of single-phase, partial, and fully developed nucleate boiling. The effect of main parameters of water jet, i. e. jet sub-cooling, jet velocity, and nozzle to plate distance on the heat flux, is of concerned. A comparison of the obtained results of the model is made with various published experimental data and good agreement is reported.

Key words*: stagnation-point flow, jet impingement boiling, modeling, nucleate boiling, similarity solution, onset of nucleate boiling*

Introduction

Cooling a hot surface by water jet impingement boiling (JIB) is widely used in many industrial and engineering applications such as: cooling processes of hot rolling steel strip, arrays of coplanar computer chips and microelectronic circuits, and nuclear power plants. Because of the high effectiveness of cooling by water impinging jets, numerous studies have been performed in both single- and two-phase situations. Single-phase impinging jets have been studied experimentally and numerically extensively by various researchers, *i. e*. [1-5], but the two-phase state in which the impinging liquid is allowed to boil on the hot surface is rather less explored, numerically. Cooling by JIB has been studied under two conditions: steady-state and transient (quenching) states. In steady-state conditions either of the wall heat flux or the wall temperature is controlled during the cooling, whereas in transient conditions the surface is first heated to a desired temperature and then is cooled by a liquid jet. One of the pioneering studies in this field was done by Monde [6] who studied nucleate boiling of impinging jets of water and freon-l13 experimentally to determine critical (maximum) heat flux (CHF) and correlated it with the jet parameters. Miyasaka and Inada [7] investigated JIB at the stagnation-point and in the parallel flow, experimentally and also obtained a correlation for the onset of nucleate boiling heat flux in a part of the stagnation zone of a planar free-surface sub-cooled water jet. Vader *et al*. [8] investigated the boiling heat transfer during planar free-surface jet impingement of water under steady-state heat transfer condition. A comprehensive review of the JIB has been published by Wolf *et al*. [9]. They compiled various available correlations of boiling curve from

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steady-state measurements. Robidou *et al.* [10] studied jet impingement heat transfer under steady-state experiments along the entire boiling curve for hot plate temperatures. Hauksson [11] studied boiling heat transfer during water jet impingement on a hot steel plate, experimentally. Also, he used superposition model based on Chen [12] correlation and developed suppression factor of the model by fitting it to his experimental data. Then Summerfield [13] used modified Chen's correlation by fitting it to two sets of experimental data including Hauksson's data. Liu and Winterton [14] applied superposition method to estimate flow boiling in tubes and annuli and modified Chen's model suppression factor by some fitting parameters.

A thorough review of the literature reveals lots of experimental works and relatively fewer numerical and mechanistic works on JIB. However, because of the complex nature of the boiling, it will be more difficult to develop a comprehensive model or correlation when boiling is along with forced convection like what happens during jet impingement boiling phenomena. It may lead to the pessimistic view that mechanistic prediction of nucleate boiling is a hopeless task [15]. The exiting experimental studies are limited to obtaining a small range of data and also for a small range of variations of main parameters. On the other hand, the existing numerical models based on CFD simulations of flow boiling are very costly in solving details of phase change with not much progress in giving accurate results. Furthermore, most existing mechanistic models based on superposition approach contain empirical constants or fitting parameters in their validating process which are with not much of physical/analytical conceptions. So, these models and fitting parameters would be limited to certain data.

In the present study, a mechanistic model based on superposition of the single-phase forced convection and nucleate pool boiling is developed to predict the surface heat flux and capture boiling curve under steady-state conditions in stagnation region of a planar water jet impingement. No empirical constant or fitting parameter is used in this model. However, one can readily fit a composite model to experimental data by varying the fitting parameters without a physical conception. This proposed model uses a suppression factor by computing it under physical and analytical conceptions. A comparison of the obtained results by using the model with some available published experimental data [7, 10, 16] and for a wide range of experimental conditions and variations of nozzle jet parameters is included.

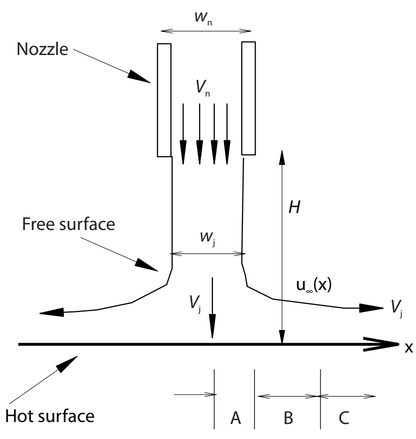

Figure 1. Nozzle configuration and velocity field of a planar free surface jet, along with the respective flow regions

Formulation of the nucleate flow boiling model

The jet impingement configurations are categorized in five different configurations: free-surface jet, plunging jet, submerged jet, confined jet, and wall jet [9]. This study focuses on free surface jets. The details of the nozzle configuration, impinging jet configuration and the velocity field of a planar free jet impinging on a flat plate is illustrated in fig. 1. As the water jet impinges on the heated surface, the flow of the liquid can be divided into stagnation (A), acceleration (B), and parallel flow (C) regions. Combination of the stagnation and the acceleration regions together are referred to as the impingement region, [9].

The water jet is assumed to divert symmetrically around the stagnation line and extends to the surface in a parallel manner. The stagnation region is within

$x/w_j \leq 0.5$, and the stream-wise velocity increases nearly linear with distance from the stagnation point. Within the acceleration region ($0.5 \leq x/w_j \leq 2$) the flow continues to accelerate until it approaches the jet velocity. For $x/w_j \geq 2$ (parallel-flow region), the stream-wise velocity is essentially equal to the jet velocity [9].

The nozzle jet velocity and its width are corrected for the effects of gravitational acceleration between the nozzle discharge and the impingement surface by expressions:

$$V_\mathrm{j} = \sqrt{V_\mathrm{n}^2 + 2\mathrm{g}H} \quad \text{and} \quad w_\mathrm{j} = w_\mathrm{n}\left(\frac{V_\mathrm{n}}{V_\mathrm{j}}\right)$$

The pressure is maximum at the stagnation point and decreases away from it. The pressure on the surface at the stagnation point can be obtained by the Bernoulli equation:

$$p = p_0 + \frac{1}{2}V_\mathrm{j}^2 \tag{1}$$

In different surface temperatures, various heat transfer regimes are observed. Within the surface temperature below the temperature required for vapor bubble nucleation, *i. e.* T_s, p, T_sonb single-phase cooling regime occurs. When the surface temperature is well above the saturation temperature of the liquid, isolated vapor bubbles begin to nucleate and grow on the surface. This temperature is shown as T_onb (onset of nucleate boiling) in fig. 2. At this point, partial nucleate boiling regime begins. Partial nucleate boiling is characterized by hydrodynamic of bulk flow, and by dynamic formation, growth, and collapse of isolated vapor bubbles on the surface. Hence, the heat flux on the surface is increased (comparing to single-phase regime) due to isolated vapor generation and bubble activity at the wall. With a further increase in the surface temperature, much more bubbles are formed and their growth lead to the boiling regime transition from partial nucleate boiling to fully developed nucleate boiling where the heat transfer mechanism is dominated by evaporation and bubbles motion and agitation. The results of investigations into rates of heat transfer in boiling are usually plotted on a graph of surface heat flux, q'', against wall superheat, ΔT_sat, that is called the boiling curve. The trend of the pool and flow boiling curves schematically in forced convection (single-phase) and nucleate (partial and fully developed) boiling (two-phase) regions which are under study in this paper is shown in fig. 2.

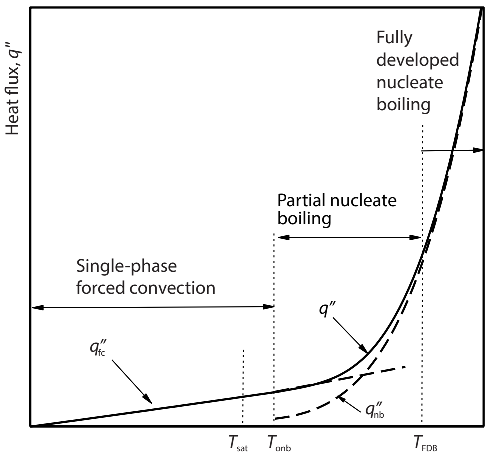

Figure 2. Schematic of the boiling curves; solid line denotes the flow boiling and dashed line denotes the pool boiling curve

As it is seen, in a fixed wall superheat, ΔT_sat, the heat transfer rate of flow boiling is more than pool boiling because of the forced convection effects. According to the figure, total heat flux can be expressed as a combination of forced convection and pure pool boiling heat flux. This idea of additive contributions was first introduced by Rohsenow [17]. Then Bergles and Rohsenow [18] developed this idea by considering this point that just prior to incipient boiling, heat flux still can be expressed by forced convection heat flux. So, they suggested the following equation for flow boiling in heated tubes:

$$q'' = \left[q_\mathrm{fc}''^2 + \left(q''_\mathrm{nb} - q''_\mathrm{onb}\right)^2\right]^{1/2} \tag{2}$$

In our study, we develop this model for JIB. For this purpose, the related correlations of the jet impingement have to be applied. Also, a comprehensive study has been done to calculate heat transfer parameters in incipience of boiling, ΔT_{onb} and q''_{onb}, which are essential parameters in the present model. Equation (2) is a superposition of q''_{fc} and q''_{onb}, modified by subtracting q''_{onb}, in order to make $q'' = q''_{\text{fc}}$ at the incipience of boiling. It can be rewritten in the following form:

$$q'' = \left\{ q''^{2}_{\text{fc}} + q''^{2}_{\text{nb}} \left[1 - \left(\frac{q''_{\text{onb}}}{q''_{\text{nb}}} \right) \right]^2 \right\}^{1/2} \tag{3}$$

To determine relation between q''_{onb} and q''_{nb} we use the following approximation that the heat flux q'' in boiling curve is almost proportional to the third power of the wall superheat temperature, [19], eq. (18). Therefore, eq. (3) may be rewritten:

$$q'' = \left\{ q''^{2}_{\text{fc}} + q''^{2}_{\text{nb}} \left[1 - \left(\frac{\Delta T_{\text{sat}_{\text{onb}}}}{\Delta T_{\text{sat}}} \right)^3 \right]^2 \right\}^{1/2} \tag{4}$$

The term inside the inner parenthesis can be introduced as modified suppression factor of Chen's model [12]:

$$S = 1 - \left(\frac{\Delta T_{\text{sat}_{\text{onb}}}}{\Delta T_{\text{sat}}} \right)^3 \tag{5}$$

So, eq. (2) can be rewritten in the final form of:

$$q'' = \left[q''^{2}_{\text{fc}} + \left(S q''_{\text{nb}} \right)^2 \right]^{1/2} \tag{6}$$

The suppression factor, S, shows the effects of forced convection on pool boiling in flow boiling regime. As it can be seen from eq. (5), the suppression factor, S, is zero in incipience of boiling, increases by increasing wall temperature and approaches unity at high superheat temperature close to temperature respect to CHF. As expected from the physics of the problem and eq. (5) and fig. 2, by increasing the surface temperature and therefore increasing active nucleate boiling sites, S is increased as a result of increase in the pool boiling contribution opposed to the single-phase in the two-phase model. This is explained better by illustrating of S graphs *vs.* superheat temperature in the results section. By knowing appropriate equations to calculate the rate of the forced convective heat transfer q''_{fc}, the nucleate pool boiling q''_{nb}, and the suppression factor S, the model is complete and so the total heat flux q'' can be obtained by using eq. (6). These terms are estimated in next sections.

Sing-phase forced convection heat transfer

The first term is forced convective heat flux. This term is defined by Newton's law of cooling:

$$q''_{\text{fc}} = h\left(T_{\text{s}} - T_{\text{f}}\right) = h\left(\Delta T_{\text{f}}\right) \tag{5}$$

So, it is needed to know the correlations of the heat transfer coefficient, h. This coefficient can be obtained by solving the boundary-layer equations. The simplified form of 2-D laminar boundary-layer equations (by employing the Bernoulli's equation in the potent $-\partial P/\rho\partial x = \mathrm{d}U/\mathrm{d}x$) are presented:

Continuity equation:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \tag{8}$$

Momentum equation:

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=U\frac{\mathrm{d}U}{\mathrm{d}x}+\nu\frac{\partial^2 u}{\partial y^2} \tag{9}$$

Energy equation:

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\frac{\partial^2 T}{\partial y^2} \tag{10}$$

The free stream velocity component of the classical potential flow solution is as $U = Cx$. The C parameter introduces the velocity gradient that is expressed in terms of the jet velocity and the jet width as $C=\bar{C}v_j/w$, [1] where the value of $\bar{C}=\pi/4$ [1, 2].

Similarity Solution

To convert PDE (8) and (10) into a set of ODE, the following dimensionless similarity variables are introduced [20]:

$$\eta=\sqrt{\frac{C}{\nu}}y,\ u=Cxf'(\eta),\ v=-Cf(\eta),\ \theta(\eta)=\frac{T-T_\mathrm{f}}{T_\mathrm{s}-T_\mathrm{f}} \tag{11}$$

Substituting these transformations into momentum and energy eqs. (9) and (10) respectively, yields the following non-linear ODE:

$$f'''+ff''-f'^2+1=0 \tag{12}$$

$$\theta''+\mathrm{Pr}\,f\theta'=0 \tag{13}$$

The boundary conditions in eqs. (12) and (13) are:

$$\eta=0\ :\ \begin{cases}u=0,\ \ v=0,\ T=T_\mathrm{s}\\ f'=f=0,\ \theta=1\end{cases} \tag{14}$$

$$\eta\to\infty\ :\ \begin{cases}u=U,\ T=T_\mathrm{f}\\ f'=1,\ \theta=0\end{cases} \tag{15}$$

The heat transfer coefficient, h, can be calculated by the following relation:

$$h=-\frac{K\left.\dfrac{\partial T}{\partial y}\right|_{y=0}}{\Delta T_\mathrm{f}}=-\rho c_p\,\mathrm{Pr}^{-1}\sqrt{C\nu}\,\theta'(0) \tag{16}$$

To solve the set of eqs. (12) and (13), we applied the fourth-order Runge-Kutta numerical scheme with a shooting method and iterated the solution process till satisfaction of the initial boundary conditions. By solving of eqs. (12) and (13), the coefficient of $\theta'(0)$ and hence the heat transfer coefficient is obtained by eq. (16). Previous computations have been done for laminar water jet. A water jet is turbulent if the Reynolds number is above about 4000, [21]. Miyasaka and Inada [7] obtained stagnation heat transfer coefficient of a turbulent water jet experimentally:

$$h=0.909\,\mathrm{Re}_j^{\frac{1}{2}}\,\mathrm{Pr}^{\frac{1}{3}}\frac{k_\mathrm{f}}{w_j} \tag{17}$$

The present study uses eqs. (16) and (17) for laminar and turbulent jet, respectively. In all the previous variation of thermophysical properties with temperature has been taken into account by evaluating them at the film temperature defined by $(T_s + T_f)/2$. The correlation equations for the thermophysical properties of water and other coolants in gas and liquid states are presented as a function of temperature in VDI-Heat Atlas [22].

Nucleate pool boiling heat transfer

The next term in eq. (6) to be estimated is pool boiling heat flux q''_{nb}. Pool boiling is defined as boiling on a heated surface submerged in a large volume of stagnant liquid. This liquid may be at its saturation boiling point in which case the term saturated pool boiling is employed or below its saturation boiling point in which case the term subcooled pool boiling is used [23]. The first complete correlation for pool boiling was proposed by Rohsenow [19] that for water is given:

$$q'' = \mu_f h_{fg} \left(g \frac{\rho_f - \rho_g}{\sigma} \right)^{1/2} \left(\frac{c_{p_f}}{c_{sf} h_{fg} \mathrm{Pr_f}} \Delta T \right)^3 \tag{18}$$

The value of c_{sf} varies with the liquid/surface combination. Subsequently, several predictive correlations for boiling heat transfer coefficient for pure liquids were presented. These correlations are generally empirical or semi-empirical including the correlation proposed by Forster and Zuber [24], Stephan and Abdelsalam [25], Cooper [26], Gorenflo and Kenning [27]. Among these, Gorenflo's correlation is the newest and has the major distinction in comparison to other exiting correlations that includes main groups of variables, like properties of the liquids, the nature of the heated surfaces, and the operating parameters (namely, the heat flux q'' and pressure p). It is reported in the literature and verified by us that the Gorenflo model is validated as a model which compares best with experiment and has the best performance in comparison to the other mentioned models [28]. Therefore, Gorenflo model is used to calculate q''_{nb} in our study. This model is based on a reference heat transfer coefficient, h_0, at the following standard conditions: reduced pressure $p_0^* = 0.1$, heat flux $q''_0 = 20$ kW/m^2 and Cu surface roughness with an intermediate value $R_{a0} = 0.4$ μm of the arithmetic mean roughness height of the surface, which lies within the common range for heater surfaces manufactured in practice, [23, 27]. The reference heat transfer coefficient for water is, $h_0 = 5600$ W/m^{2}K. To obtain heat transfer coefficient at other conditions, the following expression is used, [27]:

$$\frac{h}{h_0} = F_q F_p F_w \tag{19}$$

where the F quantities are independent non-dimensional functions applicable to all fluids (with a very small number of exceptions), representing the relative influences on h of the heat flux q'', the reduced pressure $p^* = p/p_c$, and wall properties. The functions are defined:

$$F_q = \left(\frac{q''}{q''_0} \right)^n, \; F_p = F\left(p^*\right), \; F_w = \left(\frac{R_a}{R_{a0}} \right)^{2/15} \tag{20}$$

For water, relationships $n(p^*)$ and $F(p^*)$ are according to:

$$n\left(p^*\right) = 0.9 - 0.3 p^{*^{0.15}}, \; F\left(p^*\right) = 1.73 p^{*^{0.27}} + 6.1 p^{*^2} + 0.68 \frac{p^{*^2}}{1 - p^{*^2}} \tag{21}$$

Onset of nucleate boiling

With known correlations of q''_{fc} and q''_{nb} the remaining main problem is finding S or on the other hand the wall superheat with respect to onset of nucleate boiling, ΔT_{onb}. Most studies in the literature have used Hsu's model to estimate a relation between q''_{onb} and ΔT_{onb}. For uniform surface heat flux, Hsu [29] established the following expression between the superheat and the incipience heat flux:

$$\left(\Delta T_{\mathrm{sat}}\right)_{\mathrm{onb}} = \left(\frac{8\sigma q''_{\mathrm{onb}} T_{\mathrm{sat}}}{h_{fg} k_{\mathrm{f}} \rho_{\mathrm{g}}}\right)^{1/2} \tag{22}$$

where ΔT_{onb} is a function of q''_{onb}, while itself is unknown explicitly. Moreover, the eq. (22) is for saturation pool boiling that does not include sub-cooling boiling and flow velocity (forced convection) effects what are important in the jet impingement boiling. To obtain an explicit equation to estimate ΔT_{onb} as a function of the known parameters along with considering the flow and sub-cooling influences on incipience of boiling, Newton's law of cooling at the point of incipience can be applied:

$$q''_{\mathrm{onb}} = h\left(T_{\mathrm{onb}} - T_{\mathrm{f}}\right) = h\left[\left(\Delta T_{\mathrm{sat}}\right)_{\mathrm{onb}} + \Delta T_{\mathrm{sub}}\right] \tag{23}$$

The heat flux in eq. (22) can be replaced by eq. (23), and then the wall superheat may be expressed explicitly in terms of hand ΔT_{sub}:

$$\left(\Delta T_{\mathrm{sat}}\right)_{\mathrm{onb}} = \frac{1 + \left(1 + 4\lambda \Delta T_{\mathrm{sub}}\right)^{1/2}}{2\lambda} \tag{24}$$

where:

$$\lambda = \frac{k_{\mathrm{f}} h_{fg}}{8\sigma v_{\mathrm{g}} T_{\mathrm{sat}} h} \tag{25}$$

Substituting eq. (24) into eq. (5) yields the following expression for suppression factor as an explicit function of thermophysical properties of liquid jet and surface temperature as:

$$S = 1 - \left(\frac{1 + \left(1 + 4\lambda \Delta T_{\mathrm{sub}}\right)^{\frac{1}{2}}}{2\lambda \Delta T_{\mathrm{sat}}}\right)^{3} \tag{26}$$

As it can be seen, S considers effects of jet velocity and liquid temperature (sub-cooling), properly under the physical conception of the problem. Now, with knowing all terms, total heat flux q'' can be calculated by using eq. (6). Some main numerical results are presented in next section.

Results and discussion

Boiling curve and comparison of the model predictions with experiments

This section presents obtained data by the proposed model for stagnation region including boiling curves, effect of nozzle and jet configuration on boiling curves and supersession factor and a correlation to calculate onset nucleate heat flux as a function of wide range of nozzle jet velocity, v_{n}, and liquid sub-cooling, T_{f}, for a planar water jet. To validate the proposed model, the obtained results are compared with some available experimental data in literature in figs. 3-5.

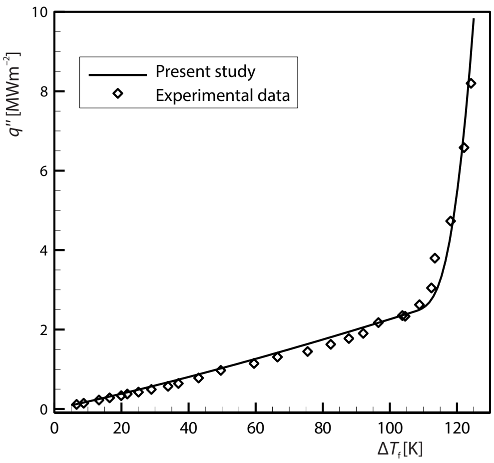

Figure 3. Comparison of JIB curve result with data of [7], in the case of $v_n = 3.2$ m/s, $H = 6$ mm, $w_n = 10$ mm, and $T_f = 15$ °C

As it can be seen, a good agreement is depicted between results of the model and the experimental data of various studies at different conditions of nozzle configuration, jet velocity and sub-cooling. Also, the transition from single-phase convection to nucleate boiling is identified with the same trend in experimental results.

As seen, for a surface temperature, less than, T_{onb}, flow is in single-phase and heat transfer regime is forced convection. So, slope of the boiling curve is almost constant (Newton's law of cooling). Upon reaching temperature to T_{onb}, the first nucleation sites are activated and the first bubbles are formed and grow. So, the evaporation is initiated and discrete bubbles begin to attach on the surface, leading to enhancing heat transfer. In this level, boiling regime is partial nucleate boiling where vapor generation is limited to a small population of bubbles and the bulk flow continuing to strongly influence convection heat transfer from the surface. As a result of latent heat and bubble formation effects, the rate of heat transfer exceeds the single-phase case. With further increase in surface temperature, more bubbles form (increase of bubble density) and bubble departure from the surface increases. So, the heat flux increases abruptly in such a way that the slope of the boiling curve has a significant growth in fully developed boiling regime.

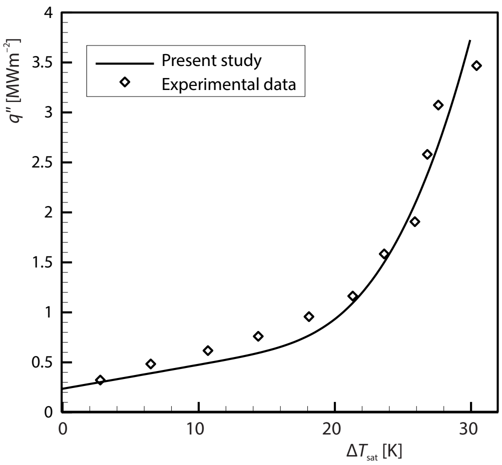

Figure 4. Comparison of JIB curve result with data of [10], the case of $v_n = 0.68$ m/s, $H = 6$ mm, $w_n = 1$ mm, and $\Delta T_{sub} = 10$ °C

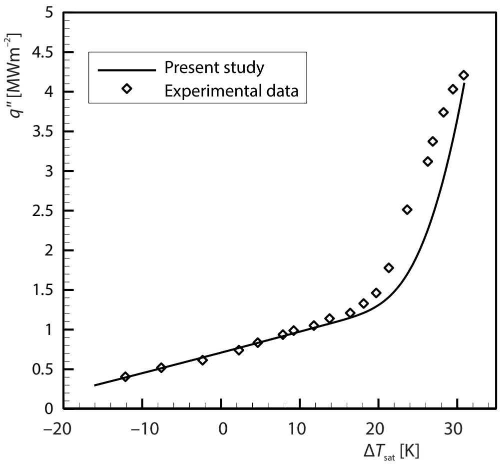

Figure 5. Comparison of JIB curve result with data of [16], in the case of $v_n = 0.75$ m/s, $H = 10$ mm, $w_n = 1$ mm, and $\Delta T_{sub} = 28$ °C

The previous comprehensive comparison show a well performance of the present mode for different range of sub-cooling, ΔT_{sub}, jet velocity, v_n, and jet nozzle configurations, H and w_n.

Effect of main parameters on boiling curve

The effect of jet velocity on the boiling curve at the stagnation region is illustrated is fig. 6. The heat flux is plotted against the superheat temperature in the single-phase and nucleate boiling regimes. The curves were measured with a free surface jet, at a sub-cooling of 15 K, a nozzle width 1 mm, and a nozzle to plate spacing of 10 mm. For small ranges of nozzle jet velocity, this parameter does not have a strong influence on the boiling curve. While with increasing jet velocity, its effect becomes significant on single and partial nucleation boiling. But almost no significant effect of the jet velocity in the fully developed nucleate boiling regime can be observed. In the fully developed boiling regime at higher wall superheats, the boiling curves converge towards the pool boiling curve indicating that the hydrodynamic of bulk flow becomes insignificant once the wall superheat is sufficiently high. This is in agreement with the measurements by [7-9].

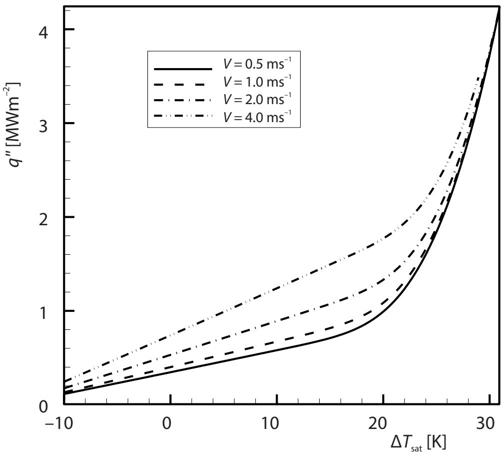

Figure 6. Effect of nozzle jet velocity on the heat flux, when H = 10 mm, w_n = 1 mm, and ΔT_{sub} = 15 °C

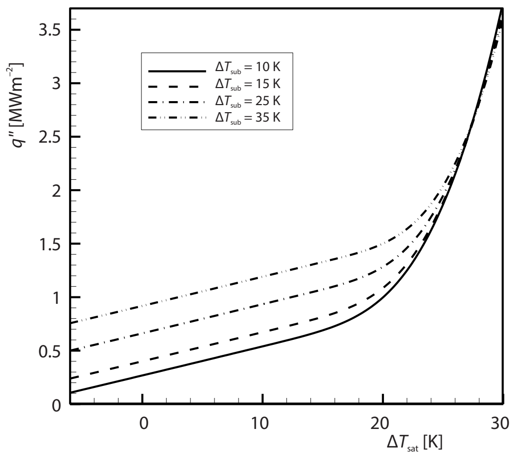

Figure 7. Effect of degree of sub-cooling on the heat flux, when v_n = 1 m/s, H = 10 mm, and w_n = 1 mm

Boiling incipience depends strongly on forced convection. By increasing jet velocity, the onset of nucleate boiling is shifted to higher wall superheat temperatures as seen in fig. 6. This is because with increasing jet velocity the thermal boundary-layer becomes thinner and temperature gradient within it is not sufficient to allow bubble formation and thus nucleation is delayed. This effect will be observed better in the next section where the suppression factor and onset of nucleate boiling temperature and heat flux are investigated. Also, the effect of the nozzle to plate distance is the same as the jet velocity effect on the heat flux. On the other hand, an increase of the nozzle to plate distance (thereby an increase in the velocity at the stagnation region) increases heat transfer rate. In the fully developed nucleate boiling regime, the jet velocity (and also the distance) has no significant influence on the heat transfer.

The sub-cooling has a strong influence on the heat transfer along the entire boiling curve. Figure 7 depicts the effect of sub-cooling of liquid jet on the boiling curve for a jet velocity of 1 m/s and a nozzle to plate distance of 10 mm. As seen, increasing the degree of sub-cooling temperature from 10 to 35 K results in a significant increase in the rate of heat transfer, especially in single-phase and partial boiling regions, and a delay in the onset of nucleate boiling.

In all of the previous results, figs. 6 and 7, jet velocity, nozzle to plate distance, and sub-cooling parameters, does not affect the heat flux in the fully developed nucleate boiling regime noticeably and the fully developed boiling data collapse to a single curve.

Effect of main parameters on suppression factor

The suppression factor is a factor of pool boiling contribution in the presented model for jet flow boiling which is the result of the bulk flow hydrodynamic. For surface temperatures, less than T_{onb} flow is non-boiling and suppression factor is zero. With activation of the first nucleate sites, boiling is started and by increasing wall temperature the suppression factor will increase from zero to a maximum amount which is almost unity for fully developed boiling region. Figures 8 and 9 illustrate effects of nozzle jet velocity and sub-cooling on suppression factor. According to eq. (1), with increasing jet velocity the stagnation pressure will increase and thus the local saturation temperature will increase as well. So, as seen in the fig. 8, the graphs shift to a higher wall superheat at starting point. Also, as expected, with increasing the jet velocity, the forced convection effect becomes larger. So, the suppression factor decreases when jet velocity increases.

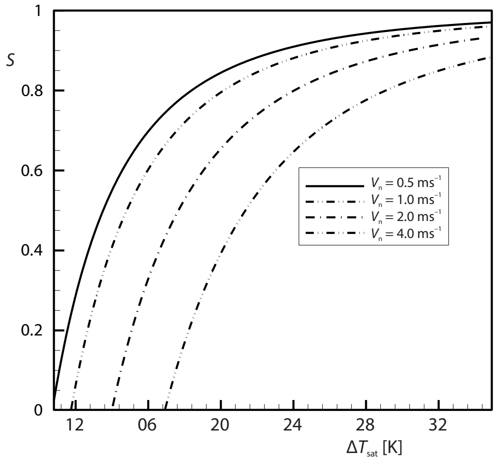

Figure 8. Effect of nozzle jet velocity on the suppression factor, when H = 10 mm, w_n = 1 mm, and ΔT_{sub} = 15 °C

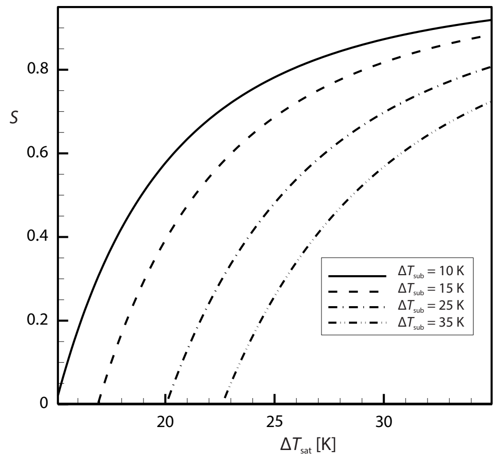

Figure 9. Effect of degree of sub-cooling on the suppression factor, when v_n = 1, H = 10 mm, w_n = 1 mm

A similar trend can be seen for different sub-cooling temperatures in fig. 9. When the temperature of the liquid jet increases, a higher wall temperature is needed for incipient boiling. So, S will decrease as a result of increasing of onset of nucleate boiling.

Conclusion

Modeling of the jet impingement boiling is a significant challenge in heat transfer problems. A model using modified superposition approach has been developed in this study to predict the rate of total heat flux in the stagnation region of a free planar JIB on a hot flat surface. It is based on the combination of the single-phase forced convection and nucleate pool boiling components as the total heat flux. A similarity solution has been presented to calculated single-phase forced convection and Gorenflo's correlation has been used to calculate nucleate pool boiling components. The proposed model uses a suppression factor by computing it under physical and analytical conception. This fairly simple model which avoids the complexity of two-phase flow consideration would lead to a significant reduction in the cost of flow boiling heat transfer simulations, while gives accurate results. The value of the model for jet impingement boiling problem is shown through good agreement of predictions with data obtained on various published experimental works which are for a wide range of conditions and variations of nozzle jet parameters. The effect of main parameters of liquid jet like velocity, nozzle to plate distance and the degree of sub-cooling on rate of heat transfer was investigated. These parameters were most pronounced in the single-phase and partial boiling regimes, where heat transfer mechanism is dominated by the temperature and hydrodynamics of the bulk flow, and not by evaporation or bubble motion. Within the fully developed boiling regime where heat transfer mechanism is dominated by evaporation and bubbles motion which slid and detach from the surface, jet parameters do not affect the heat flux noticeably.

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Nomenclature

c_p – specific heat, [Jkg^{-1}°C^{-1}]
c_{sf} – constant used in eq. [18]
F_p, F_q, F_w – function defined in eq. [20]
g – gravity acceleration, [ms^2]
h – heat transfer coefficient, [Wm^2°C^{-1}]
h_{fg} – latent heat of vaporization, [Jkg^{-1}]
H – nozzle to plate distance, [m]
K – conductivity [Jm^{-1}K^{-1}]
k – thermal conductivity, [Wm^{-1}°C^{-1}]
p – pressure, [Nm^2]
p_0 – atmosphere pressure
p^* – reduced pressure
Pr – Prandtl number
q'' – heat flux, [Wm^{-2}]
R_a – surface roughness
Re$_j$ – jet Reynolds number
S – suppression factor defined in eq. (5)
T – temperature, [°C] or [K]
ΔT – temperature difference, [°C] or [K]
u – component of fluid velocity in x-direction
V – jet velocity, [ms^{-1}]
w_j – jet width at impingement, [m]
w_n – nozzle width, [m]

Greek symbols

α – thermal difusivity, [m^{2}s^{-1}]
λ – parameter defined in eq. (25)
μ – dynamic viscosity, [kgm^{-1}s^{-1}]
ν – molecular kinematic diffusivity, [m^{2}s^{-1}]
ρ – density, [kgm^{-3}]
σ – surface tension, [Nm^{-1}]

Subscripts

fc – forced convection
f – fluid (liquid)
g – gas (vapor)
j – jet related value
n – nozzle related value
nb – nucleate pool boiling
onb – onset of nucleate boiling
s – surface
sat – saturation
sub – sub-cooled
0 – reference related value

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