Enhancement of critical heat flux under subcooled boiling of deionized water in a curved flow channel

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Abstract - The numerical calculations are performed to predict critical heat flux (CHF) during subcooled boiling of deionized water through 90° curved flow channel with square cross-section. Two types of flow regimes have been observed at low and high flow velocities. In case of low flow velocities, the continuous accumulation of vapors formed a vapor blanket on heating surface which consequently reduced heat dissipation out of the surface and triggered CHF due to drying out of liquid between vapor blanket and heating wall. On the other hand, coolant circulation at higher velocities hindered the formation of vapor blanket and allowed subcooled liquid for rewetting. The consistent rewetting led the formation of small and separate vapor blankets. The enhancement in CHF has been observed with increased degree of subcooling and by increasing flow velocity in present study.

1. Introduction

The understanding of subcooled boiling heat transfer with respect to the flow rate is crucial to examine the phenomena of CHF in a channel. The convection boiling in the channel under subcooled diabatic condition has been used in search of effective cooling in worldwide applications especially in high power electronic devices and in nuclear reactors. Various researchers have proposed correlations for the determination of CHF on channel’s wall by varying flowrates. Authors observed that the flow velocity plays a significant role in affecting the start of boiling superheat and the nucleate boiling heat transfer till CHF. Therefore, it is necessary to investigate the accurate measurement of subcooled boiling heat transfer in view of fluid velocity to clarify CHF mechanisms\cite{1, 2}. According to Kutateladze\cite{3} and Zuber\cite{4} the occurrence of CHF is the result of hydrodynamic instability. The Katto et al.\cite{5} investigated that the boiling in a small heated flow channel is somehow same as that of the external flow. They proposed a CHF correlation for external flow by modifying Kutateladze equation in eq. (1) as,

\[
\frac{q_m}{\rho v h_{\text{int}}} = \left[ \frac{\rho_v \mu_u^2 d}{\sigma}, \frac{g(\rho_l - \rho_v) d}{\rho_u \mu_u^2}, \frac{\mu_v \rho_{\text{win}} d}{\mu_l} \right]
\]  

The d and U are the characteristic length and velocity of flow boiling condition. It had been reported in literature that the physical peculiar of CHF at low flow velocities is closer to the pool
boiling [6-12]. By ignoring the viscosity and gravity for high velocity case, the aforementioned equation had been suggested for low flow rates and high rates in eq. (2) and eq. (3) respectively.

$$\frac{q_m}{\rho \rho'_{l_0}} \left[ \frac{g \rho (\rho - \rho'_{v})}{\rho'_{v}} \right]^{1/4} = f \left( \frac{\rho'_{v}}{\rho'_{l_0}} d', u' \right)$$  \hspace{1cm} (2)

$$\frac{q_m}{\rho'_{v l_0}} = f \left( \frac{\rho'_{v}}{\rho'_{l_0}} \sigma^2 d' \right)$$  \hspace{1cm} (3)

Whereas,

$$d' = d \left[ \frac{g (\rho - \rho'_{v})}{\sigma} \right]^{1/2}; u' = u \left[ \frac{g (\rho - \rho'_{v})}{\rho'_{v}} \right]^{1/4}$$

The study of CHF for saturated flow boiling conducted by Katto and Kurata[13] with water and R-113 as working fluid and by considering fluid velocity from 1.25-10ms⁻¹ and from 0.5-12.5ms⁻¹ as presented in eq. (4) and eq. (5) respectively,

$$\frac{q_m}{\rho'_{vl_0}} = 0.1869 \left( \frac{\rho'_{v}}{\rho'_{l_0}} \right)^{0.559} \left( \frac{\rho'_{l_0}}{G_{l_0}} \right)^{0.264}$$  \hspace{1cm} (4)

$$\frac{q_m}{\rho'_{vl_0}} = 0.66 \left( \frac{\rho'_{v}}{\rho'_{l_0}} \right)^{0.604} \left( \frac{\rho'_{l_0}}{G_{l_0}} \right)^{0.415}$$  \hspace{1cm} (5)

For pool boiling and low fluid velocity it has been reported that the CHF is directly proportional to the degree of subcooling[14, 15] whereas for flow boiling with higher velocity the nonlinear relationship with CHF has been observed[6, 16].

The objective of present study is to measure subcooled pool boiling heat transfer by varying fluid velocity and to predict CHF in the curved flow channel numerically by using CFD two-phase (liquid-vapor) boiling model. The constant heat flux is supplied perpendicular to the convex wall of the channel while deionized water as a coolant is allowed to flow through the channel. The specifications of flow channel and boundary conditions are enlisted in Table 1.

| Specification                  | Value          | Unit       |
|-------------------------------|----------------|------------|
| Inlet gauge pressure, \( P_1 \) | 24.50          | [kPa]      |
| Outlet gauge pressure, \( P_2 \) | 21.16          | [kPa]      |
| Inlet velocity, \( v_i \)      | 0.001, 0.005, 0.010 | [ms⁻¹]    |
| Inlet temperature, \( T_1 \)   | 293, 353       | [K]        |
| Heat flux, \( Q \)             | 1500           | [kWm⁻²]    |
| Hydraulic diameter, \( d_h \)  | 0.040          | [m]        |
| Bend angle of channel, \( \theta \) | 90°             | [degrees]  |
| Radius of curvature           | 0.368          | [m]        |

2. Mathematical modeling for numerical solution

There are several different mechanisms in which heat is transferred from hot body to the surrounding fluid i.e. in boiling. At the part of the heating surface where no bubble inhabits, the heat flows to the subcooled liquid directly like a single-phase flow. The part of the heating surface where the generation of vapors occurs at the nucleation sites extracts heat out of the surface. With the escaping of bubbles, the liquid mixing mechanism exists. The cold liquid from the bulk comes into contact with the heating surface as a result of recirculation around the detaching bubble, this phenomenon conducts additional cooling. In terms of two sets of conservation equations, governing the balance of mass, momentum, and energy of each phase the two-phase subcooled boiling model has been established. In governing equations, the interaction terms couple the interface transfer of mass, momentum, and energy. The water-liquid is taken as continuous phase whereas water-vapor is illustrated as a dispersed or secondary phase. The k-\( \varepsilon \) turbulence model[17] in a square cross section with hydraulic diameter of 0.040m and radius of curvature of 0.368m has been numerically solved for boiling water with heat flux of 1500kWm⁻² by incorporating CFD code.

Conservation of mass,
\[ \nabla (\rho_m u_m) = 0 \]  
\( (6) \)

Mass average velocity \((u_m)\),
\[ u_m = \sum_{k=1}^{2} a_k \frac{\rho_k u_k}{\rho_m} \]  
\( (7) \)

Mixture density \((\rho_m)\),
\[ \rho_m = \sum_{k=1}^{2} \alpha_k \rho_k \]  
\( (8) \)

Conservation of momentum\([18]\),
\[ \nabla (\rho_m u_m u_m) = -\nabla p + \nabla [\rho_m (\nabla u_m + \nabla u_m^T)] + \rho_m g + \nabla (\sum_{k=1}^{2} \alpha_k \rho_k u_{dr,k} u_{dr,k}) \]  
\( (9) \)

The viscosity of mixture \((\mu_m)\),
\[ \mu_m = \sum_{k=1}^{2} \alpha_k \mu_k \]  
\( (10) \)

The drift velocity \((u_{dr,k})\) for \(k\)-phase,
\[ u_{dr,k} = u_k - u_m \]  
\( (11) \)

The slip velocity \((u_{sl})\),
\[ u_{sl} = u_v - u_t \]  
\( (12) \)

The relative velocity consisting of diffusion term as a result of dispersion by the turbulent\([19]\):
\[ u_{rel} = \frac{(\rho_v - \rho_m) \alpha_d^2 n}{\rho_m} \frac{\nu_m}{\alpha_D} \nabla \alpha_t \]  
\( (14) \)

Where \(\nu_{rel}\) is the turbulent viscosity of mixture, \(\sigma_D\) is the Prandtl dispersion coefficient. The drag force \(f_{drag}\), is considered as\([20]\):
\[ f_{drag} = \begin{cases} 1 + 0.15 R e_v^{0.687} & R e_v \leq 1000 \\ 0.0183 R e_v & R e_v > 1000 \end{cases} \]  
\( (15) \)

The Reynolds number,
\[ R e_v = \frac{u_m d_p}{\nu_m} \]  
\( (16) \)

The acceleration \((a)\),
\[ a = g - (u_m \nabla) u_m \]  
\( (17) \)

The energy equation for the mixture is as follows;
\[ \nabla \left[ \sum_{k=1}^{2} (\alpha_k \rho_k c_h) \right] = \nabla \left( k_{eff} \nabla T \right) + Q_w \]  
\( (18) \)

\[ k_{eff} = \left[ \sum_{k=1}^{2} \alpha_k (k_h + k_v) \right] \]  
\( (19) \)

The effective conductivity \((k_{eff})\) and the turbulent thermal conductivity \((k_t)\) are related as;
\[ k_t = \frac{C_p \mu_t}{\rho_t} \]  
\( (20) \)

The heat source i.e. the wall heat flux \(Q_w\) which is equivalent to the latent heat required for phase change. \(C_p\) is specific heat capacity. By incorporating continuity equation for the phases, the void fraction for dispersed phase i.e. vapors is computed as;
\[ \nabla (\alpha_v \rho_v \nu_m) = -\nabla (\alpha_v \rho_v \nu_{dr,v}) + \sum_{k=1}^{2} (m_{lv} - m_{vl}) \]  
\( (21) \)

\(\nu\), is the mass transfer through evaporation and condensation. The turbulence model \((k-\varepsilon)\) has been used in this investigation due to complex modeling in the multiphase simulation. It is mainly effectual for turbulent core flows. The equations for turbulent kinetic energy and dissipation rate are represented as;
\[ \nabla (\rho_m u_m k) = \nabla \left( \frac{\mu_{m,k}}{\sigma_k} \right) + G_{k,m} - \rho_m \varepsilon \]  
\( (22) \)

\[ \nabla (\rho_m u_m \varepsilon) = \nabla \left( \frac{\mu_{m,\varepsilon}}{\sigma_\varepsilon} \right) - C_2 \rho_m \varepsilon \left( C_{1e} G_{k,m} - C_{2e} \rho_m \varepsilon \right) \]  
\( (23) \)

The turbulent viscosity and turbulent kinetic viscosity are taken as\([21]\);
\[ \mu_t = \frac{C_\mu \rho_m k^2}{\tau} \]  
\( (24) \)

\[ G_{k,m} = \mu_t m (\nabla u_m + \nabla u_m^T) : \nabla u_m \]  
\( (25) \)
\( \sigma_k, \sigma_e, C_{ke}, C_{2e}, C_\mu \) equal to 1, 1.3, 1.44, 1.92, 0.09 respectively are the empirical constants\[22\]. The effective viscosity is given as:

\[
\mu_{\text{eff}} = \mu_{\text{eff}} + \mu_{\text{m}}
\]  

(26)

The wall heat flux \((Q_w)\) is constituted into three parts:

\[
Q_w = Q_c + Q_e + Q_q
\]  

(27)

The first part is convective heat flux \((Q_c)\):

\[
Q_c = A_1 St \rho_l C_{pr} u_l (T_w - T_l); \quad A_1 = 1 - A_2
\]  

(28)

Where \(A_1\) and \(A_2\) are the fraction of the wall surface under the influence of liquid and vapor formed on the wall surface respectively. \(T_l\) and \(u_l\) are the liquid temperature and velocity at the cell closed to the wall. \(St\) is the Stanton number.

\[
S_t = \frac{Nu}{Re P r l}
\]  

(29)

The active nucleation density is correlated as:

\[
n = [210 (T_w - T_{sat})]^{1.805}
\]  

(30)

The second part is evaporation heat flux \((Q_e)\) for the generation of vapor bubbles:

\[
Q_e = n f \left(\frac{\pi d_{\text{Bw}}^2}{6}\right) \rho_v h_{\text{vt}}
\]  

(31)

The bubble frequency \((f)\) is driven from the correlation:

\[
f = \sqrt{\frac{4 g (\rho_{\text{liq}} - \rho_v)}{3 \pi d_{\text{Bw}} \rho_{\text{eff}}}}
\]  

(32)

As depicted that the frequency is dependent on bubble diameter and phase density. The average of multiple sizes of bubbles is usually considered as a single size bubble in literature. The bubble departure diameter is computed as:

\[
d_{\text{Bw}} = \min \left[1.4, 0.6, \exp \left(-\frac{\Delta T_{\text{sat}}}{45}\right)\right]
\]  

(33)

The third part is quenching heat flux \((Q_q)\) representing heat flux from heating wall to the bulk liquid by departing or collapsing bubbles during the bubble eruption cycle. The \(Q_q\) is calculated from the temperature difference of wall and the bulk liquid multiplied by the heat transfer coefficient. Whereas, \(K=2\)[23]

\[
Q_q = \left(\frac{\pi}{4} K_{\text{eff}} \rho_{\text{eff}} C_{\text{pleff}} T\right) A_2 (T_w - T_l)
\]  

(34)

\[
A_2 = n K \left(\frac{\pi d_{\text{Bw}}^2}{4}\right)
\]  

(35)

By accounting the mass transfer from the liquid to the vapor phase on the heating wall is:

\[
m_{\text{lw}} = \frac{n \rho_e}{v_{\text{vt}} + C_{\text{pleff}} (T_{\text{sat}} - T_l)}
\]  

(36)

The mass transfer rate from liquid to the vapor (evaporation) and from vapor to the liquid (condensation) inside the fluid is evaluated by using the following equations:

\[
m_{\text{lv}} = \begin{cases} 
\frac{r_{\text{lv}} T_{\text{sat}}}{T_{\text{sat}} - T_l}; & T_l \geq T_{\text{sat}} \\
0; & T_l < T_{\text{sat}}
\end{cases}
\]  

(37)

\[
m_{\text{vl}} = \begin{cases} 
\frac{r_{\text{vl}} T_{\text{sat}}}{T_{\text{sat}} - T_v}; & T_v \geq T_{\text{sat}} \\
0; & T_v < T_{\text{sat}}
\end{cases}
\]  

(38)

Phase volume fraction is represented as \(\alpha\), density is \(\rho\) whereas \(r_{\text{lv}}\) and \(r_{\text{vl}}\) are mass transfer time parameters with unit ms\(^{-1}\)[24]. The aforementioned boiling model is coupled with the CFD simulation where the wall boundary condition is the wall heat flux taken as 1500kWm\(^{-2}\) and the reservoir temperature is considered as 20°C and 80°C. By considering the velocity, void fraction and heating wall temperature distribution under sub cooling effect, the heat transfer from the heating wall to the flowing fluid is examined and the HTC is predicted.
3. Results and discussion

The numerical calculations have been performed by taking deionized water as a coolant through one sided heated curved channel at gauge pressure of 24.50kPa covering flow velocity of 0.001ms⁻¹, 0.005ms⁻¹, and 0.010ms⁻¹ along with subcooling range of 20-80°C (293-353K). The boiling process prior and after CHF under the influence of varying flow conditions has been discussed. The CHF mechanism is associated with inlet velocity and subcooling. The effect of inlet subcooling and flow velocity on CHF is illustrated in Figure 1. The substantial enhancement in CHF can be seen in Figure 1 with increasing flow velocity from 0.001 to 0.010ms⁻¹ and with increased degree of inlet subcooling from 293K to 353K. The subcooling caused generation of small vapor bubbles which rarely spread to the neighboring nucleation sites as compared to the larger bubbles. The CHF trend in view of inlet velocity with subcooling temperature of 353K (i.e. when inlet temperature was 20°C) is found higher than the subcooling temperature of 293K (i.e. when inlet temperature was 80°C). High flow velocity imposed shear and drag forces on independent vapor growing on heated wall, as a result rapid departure of bubbles out of the wall occurred with high frequency followed nucleation. In case of high velocity, the entrained bubbles used to be smaller than that of fluid flowing with low velocity. The breakdown of vapor blanket into smaller portions at high velocity is due to Helmholtz instability which occurs at the interface of liquid core-vapor blanket parallel to the heating wall due to large difference in velocity between liquid core and vapor blanket.

![Figure 1](image.png)

**Figure 1.** The effect of inlet fluid velocity and subcooling on CHF

| Sub-cooling [K] | Inlet velocity[ms⁻¹] | CHF [Wm⁻²] |
|-----------------|----------------------|------------|
| 353             | 0.001                | 1.150e7    |
| 353             | 0.005                | 1.240e7    |
| 353             | 0.010                | 1.351e7    |
| 293             | 0.001                | 1.122e7    |
| 293             | 0.005                | 1.203e7    |
| 293             | 0.010                | 1.220e7    |

Table 2. Outcomes

The predicted values of CHF limit at three different velocities and two subcooling temperatures are tabulated in Table 2. The flow velocity and subcooling have strong influence on boiling. Figure 2. and
Figure 3. depict velocity distribution and temperature distribution at maximum CHF limit of 1.351e7Wm⁻² with inlet velocity of 0.010ms⁻¹ and subcooling temperature of 353K. The bubble boundary thickness reduced with increasing velocity. The low velocity of flowing fluid leads dry out following development of continuous vapor blanket whereas CHF in the range of high velocity triggered by drying out of vapor blankets which screened smaller area of the heated surface. The intense vapor generation and two-phase turbulence raised temperature of heating wall.

Figure 2. Velocity distribution at $v_{in}=0.010\text{ms}^{-1}$; $\Delta T_{sat}=353K$

Figure 3. Temperature distribution $v_{in}=0.010\text{ms}^{-1}$; $\Delta T_{sat}=353K$

Figure 4. depicts distribution of heat flux where maximum critical heat flux is achieved near the upstream due to the accumulation of vapor bubbles on upper wall i.e. on the heating wall under the influence of buoyancy. CHF is characterized by the formation of vapor blanket on heating surface which appeared thick near upstream. The large bundles of vapors seized heat dissipation due to its low thermal conductivity by disconnecting heating surface from rewetting. The occurrence of thick vapor blanket induced thermal resistance which consequently became a cause of boiling crisis.

Figure 4. Heat flux distribution $v_{in}=0.010\text{ms}^{-1}$; $\Delta T_{sat}=353K$
4. Conclusions
The CFD based two-phase (liquid-vapor) numerical simulations have been performed to determine the effects of flow velocity and subcooling on CHF from flowing fluid (DI water) through the 90° curved flow channel with square cross section having 40mm of hydraulic diameter and 368mm of radius of curvature while a constant heat flux is applied perpendicular to the convex wall of the channel. It has been concluded that the higher limit of CHF is achieved with higher flow velocities. The augmenting velocity enlarged the contact of liquid to the heating surface by reducing the thickness of vapor layer. The enhancement of CHF is reported by varying inlet fluid velocity from 0.001-0.010ms⁻¹. Moreover, the CHF is observed to increase with the increased degree of subcooling from 293-352K. The maximum value of CHF limit is achieved at 0.010ms⁻¹ with inlet subcooling temperature of 353K.

5. Abbreviations
The following abbreviations are used in this manuscript:

- \( \rho \) Density \([\text{kgm}^{-3}]\)
- \( u \) Velocity \([\text{ms}^{-1}]\)
- \( \alpha \) Volume fraction of liquid or vapor phase
- \( \mu \) Kinematic viscosity \([\text{m}^2\text{s}^{-1}]\)
- \( v \) Dynamic viscosity \([\text{Pas}]\)
- \( u_{dr} \) Drift velocity \([\text{ms}^{-1}]\)
- \( u_{sl} \) Slip velocity \([\text{ms}^{-1}]\)
- \( \sigma \) Surface tension \([\text{Nm}^{-1}]\)
- \( f_{drag} \) Drag force \([\text{N}]\)
- \( d \) Diameter \([\text{m}]\)
- \( a \) Acceleration \([\text{ms}^{-2}]\)
- \( g \) Gravity \([\text{ms}^{-2}]\)
- \( h \) Enthalpy \([\text{Jkg}^{-1}]\)
- \( k \) Thermal conductivity \([\text{Wm}^{-1}\text{K}^{-1}]\)
- \( \alpha \) Volume fraction of vapor or liquid
- \( T \) Temperature \([\text{K}]\)
- \( \Delta T \) Sub cooled \([\text{K}]\)
- \( Q \) Heat flux \([\text{Wm}^{-2}]\)
- \( \dot{Q} \) Heat transfer rate \([\text{kW}]\)
- \( C_p \) Specific heat \([\text{Jkg}^{-1}\text{K}^{-1}]\)
- \( m \) Inter-phase mass transfer
- \( k - \epsilon \) Turbulence dissipation rate \([\text{Jkg}^{-1}\text{s}^{-1}]\)
- \( \text{Nu} \) Nusselt number
- \( \text{Pr} \) Prandtl number
- \( \text{St} \) Stanton number
- \( \text{Re} \) Reynolds number
- \( n \) Nucleation site density \([\text{m}^{-2}]\)
- \( r \) Mass transfer time \([\text{s}^{-1}]\)
- \( Q_{lw} \) Wall heat flux \([\text{Wm}^{-2}]\)
- \( Q_e \) Heat flux due to evaporative \([\text{Wm}^{-2}]\)
- \( Q_c \) Heat flux due to convection \([\text{Wm}^{-2}]\)
- \( A_s \) Surface area \([\text{m}^2]\)
- \( A_i \) Area fraction of wall under the influence of liquid \([\text{m}^2]\)
- \( A_2 \) Area fraction of wall under the influence of vapor \([\text{m}^2]\)
- \( f \) Bubble departure frequency \([\text{s}^{-1}]\)
- \( d_{BW} \) Bubble departure diameter \([\text{m}]\)
- \( h_{st} \) Latent heat of vaporization \([\text{Jkg}^{-1}]\)
- \( P_1 \) Inlet pressure of channel \([\text{Pa}]\)
- \( P_2 \) Outlet pressure of the channel \([\text{Pa}]\)
- \( V \) Voltage \([\text{V}]\)
- \( I \) Current \([\text{A}]\)
- \( \theta \) Bend angle of the channel \([\text{degree}]\)

**Subscripts**

- \( v \) Vapor
- \( l \) Liquid
- \( m \) Mixture
- \( k \) Summation index
- \( sat \) Saturated
- \( eff \) Effective
- \( t \) Turbulent
- \( b \) Bulk
- \( d \) Drift
- \( e \) Evaporation
- \( c \) Convection
- \( q \) Quenching
6. References:
[1] Hata, K. and S. Masuzaki, Subcooled boiling heat transfer in a short vertical SUS304-tube at liquid Reynolds number range 5.19—104 to 7.43—105. Nuclear Engineering and Design, 2009. 239(12): p. 2885-2907.
[2] Hata, K. and S. Masuzaki. Subcooled Boiling Heat Transfer in a Short Vertical SUS304-Tube at High Liquid Reynolds Number. in 17th International Conference on Nuclear Engineering. 2009. American Society of Mechanical Engineers.
[3] Kutateladze, S.S., Heat transfer in condensation and boiling. AEC-tr-3770, 1959.
[4] Zuber, N., Hydrodynamic aspects of boiling heat transfer (thesis). 1959, Ramo-Wooldridge Corp., Los Angeles, CA (United States); Univ. of California, Los Angeles, CA (United States).
[5] Katto, Y., Critical heat flux, in Advances in heat transfer. 1985, Elsevier. p. 1-64.
[6] Vliet, G.C. and G. Leppert, Critical heat flux for nearly saturated water flowing normal to a cylinder. Journal of Heat Transfer, 1964. 86(1): p. 59-66.
[7] Cochran, T.H. and C.R. Andracchio, Forced-convection peak heat flux on cylindrical heaters in water and refrigerant 113. 1974.
[8] Lienhard, J.H. and R. Eichhorn, Peak boiling heat flux on cylinders in a cross flow. International Journal of Heat and Mass Transfer, 1976. 19(10): p. 1135-1142.
[9] Lienhard, J.H. and M.M. Hasan, On predicting boiling burnout with the mechanical energy stability criterion. Journal of Heat Transfer, 1979. 101(2): p. 276-279.
[10] Yilmaz, S. and J.W. Westwater, Effect of velocity on heat transfer to boiling Freon-113. Journal of Heat Transfer, 1980. 102(1): p. 26-31.
[11] Katto, Y., et al., Critical heat flux on a uniformly heated cylinder in a cross flow of saturated liquid over a very wide range of vapor-to-liquid density ratio. International Journal of Heat and Mass Transfer, 1987. 30(9): p. 1971-1977.
[12] Katto, Y. and Y. Haramura, Critical heat flux on a uniformly heated horizontal cylinder in an upward cross flow of saturated liquid. International Journal of Heat and Mass Transfer, 1983. 26(8): p. 1199-1205.
[13] Katto, Y. and C. Kurata, Critical heat flux of saturated convective boiling on uniformly heated plates in a parallel flow. International Journal of Multiphase Flow, 1980. 6(6): p. 575-582.
[14] Meyer, G., E.S. Gaddis, and A. Vogelpohl. Critical heat flux on a cylinder of large diameter in a cross flow. in Proc. 8th Int. Heat Transfer Conf. 1986.
[15] Kutateladze, S.S., Fundamentals of heat transfer. 1964.
[16] Vliet, G.C. and G. Leppert, Critical heat flux for subcooled water flowing normal to a cylinder. Journal of Heat Transfer, 1964. 86(1): p. 68-74.
[17] Jones, W.P. and B.E. Launder, The prediction of laminarization with a two-equation model of turbulence. International Journal of Heat and Mass Transfer, 1972. 15(2): p. 301-314.
[18] Fluent, A., Ansys Fluent theory guide. ANSYS Inc., USA. 15317: p. 724-746.
[19] Manninen, M., V. Taivassalo, and S. Kallio, On the mixture model for multiphase flow. 1996, Technical Research Centre of Finland Finland.
[20] Schiller, L., A drag coefficient correlation. Zeit. Ver. Deutsch. Ing., 1933. 77: p. 318-320.
[21] Lucas, D., E. Krepper, and H.M. Prasser, Use of models for lift, wall and turbulent dispersion forces acting on bubbles for poly-disperse flows. Chemical Engineering Science, 2007. 62(15): p. 4146-4157.
[22] Launder, B.E. and D.B. Spalding, The numerical computation of turbulent flows, in Numerical Prediction of Flow, Heat Transfer, Turbulence and Combustion. 1983, Elsevier. p. 96-116.
[23] Kenning, D.B.R., Fully-developed nucleate boiling: overlap of areas of influence and interference between bubble sites. International Journal of Heat and Mass Transfer, 1981. 24(6): p. 1025-1032.
[24] Abedini, E., et al., Numerical investigation of subcooled flow boiling of a nanofluid. International Journal of Thermal Sciences. 64: p. 232-239.