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Numerical Simulation of Two-Dimensional Forced Convective Film Boiling Flow over a Horizontal Flat Surface

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Abstract

In this numerical study, a Coupled Level Set and Volume of Fluid (CLSVOF) interface capturing method with multi-directional advection algorithm is used to investigate the flow and heat transfer characteristics of a two-dimensional, incompressible, laminar, forced convective saturated film boiling over an isothermal horizontal flat plate without using the boundary layer approximations. In this study effects of both wall superheats and liquid free stream velocity are considered for water and refrigerant R-134a. Both heat transfer coefficient and wall shear stress are strongly dependent on the wall superheat and inflow free stream velocity and these effects are analysed. In the numerical simulation of forced convection saturated film boiling flows, the wall shear stress and the heat transfer coefficient results are validated with Cess and Sparrow[1] correlations developed based on the boundary layer approximations. The numerical results show good agreement with the correlations developed based on the boundary layer theory.

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Keywords: Laminar forced convection, film boiling, level set and volume of fluid, numerical simulation, flat plate.

1. Introduction

Forced convective film boiling over superheated horizontal flat surfaces is common in many heat transfer applications. Due to forced convective film boiling process, the heated wall surface experiences a significant

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reduction of the surface drag force and enhanced the heat transfer rate over the flat surface. Because of the interfacial phase change phenomenon, the surface of the heated object completely covers with the vapour phase and it avoids the direct contact of the liquid phase with the heated wall.

Cess and Sparrow[1] carried out an analytical study of forced convective saturated film boiling over the horizontal flat plate based on the laminar boundary layer approximations using the similarity solutions. They provided the correlations for both Nusselt number and the wall shear stress over the heated flat surface. Bromely[2] carried out an analytical study of laminar film boiling on vertical flat plate by considering a linear temperature variation across the thin vapour film layer region and neglected the inertia force of the vapor film. Cess and Sparrow[3] studied the subcooled forced convection film boiling over the horizontal flat plate using laminar boundary layer theory. They found that substantial variations in the flow and the heat transfer characteristics between saturated and subcooled film boiling processes over the heated wall surface were observed.

Koh[4] investigated the forced laminar flow film condensation of saturated vapour phase over the horizontal cold flat plate using the boundary layer approximations. Koh[5] studied laminar film boiling flow over a vertical flat surface by accounting the shear stress and vapour velocity at the liquid-vapour interface. They found that in contrast to the saturated film condensation problem where heat transfer is independent of $\rho\mu$-ratio and negligible interfacial shear stress for high Prandtl numbers, in the film boiling problem, the $\rho\mu$-ratio is an important dependent parameter for both the heat transfer and the interfacial shear stress[5]. Bui and Dhir[6] investigated natural convection saturated film boiling over an isothermal vertical flat surface both experimentally and theoretically. Ito and Nishikawa[7] carried out the two-dimensional forced convection saturated film boiling flow over a horizontal flat plate using the boundary layer approximation. They found that the heat transfer and skin friction characteristics correlations strongly depend on $\rho\mu$-ratio and weakly depend on the wall superheats. Nakayama and Koyama[8] studied forced convection subcooled film boiling on a flat plate using an integral method. Dhir and Purohit[9] carried out both experimental and theoretical investigations of subcooled film boiling heat transfer from spheres.

From the literature it is found that no detailed numerical study of forced convective film boiling flows is investigated without boundary layer approximations. Therefore, in this numerical study, 2-D forced convective saturated film boiling is investigated using the CLSVOF method with multidirectional advection algorithms without considering boundary layer approximations.

Nomenclature

C_{pf}	the specific heat at constant pressure, *J/kgK*
F_{st}	surface tension force, $\sigma\kappa\nabla H(\phi)$ *N/m³*
g	acceleration due to gravity, m/s^2
h	heat transfer coefficient, W/m^2K
h_{lg}	latent heat of vaporization, *J/kgK*
$H(\phi)$	smoothed Heaviside function
k_f	thermal conductivity of fluid, *W/mK*
$\dot{m}_I$	interfacial mass transfer, kg/m^2
$\dot{m}_{fs}$	rate of void fraction as $F^n V_I S_I / \Delta v$, s^{-1}
Nu	local Nusselt number
Pr	Prandtl number
Re_x	local Reynolds number, $\rho_l U_\infty x/\mu_l$
S_c	control volume of interface cell Δv, m^3
S_I	surface area of the interface, m^2
T_I	interfacial saturation temperature T_{Sat}, *K*
U_∞	liquid free stream velocity, *m/s*
Greek symbols	
$\delta(x)$	thickness of the vapour film, *m*
κ	mean curvature, m^{-1}
μ	dynamic viscosity, Ns/m^2
ρ	density, kg/m^3
σ	surface tension coefficient, *N/m*

$(\rho\mu)_g/(\rho\mu)_l$	$\rho\mu$-ratio of two-phase flow

2. Problem formulation

2.1 Physical model

The physical model considered in the present numerical study is shown in Fig. 1. A horizontal flat plate is kept at constant wall temperature, T_W. The saturated liquid flows over the flat plat from the left side of the plate with uniform free stream velocity, U_∞ *m/s*. At the bottom boundary, a no-slip wall with constant temperature condition is considered. Zero gradient (or open) boundary condition is imposed at both top and right side boundaries. A thin layer of vapour film of thickness, $\delta(x)$ is formed over the horizontal flat plate. The constant thermo-physical properties of saturated liquid and its vapour phases are evaluated at its saturation temperature, T_{Sat} and pressure P_{Sat} for both water and refrigerant R-134a. The effects of gravity, inertia force and radiation heat transfer are not included in the similarity solutions of Cess and Sparrow[1]. However, in the present numerical study, only the effect of radiative heat transfer was neglected.

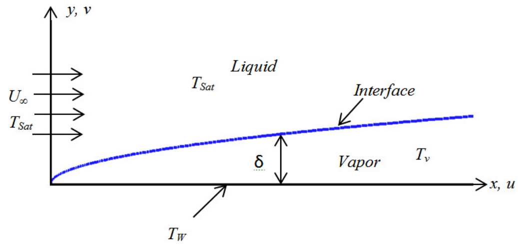


Fig. 1 Physical model and boundary conditions of the forced convective film boiling over the flat plate

2.2 Numerical governing equations

The numerical simulation of forced convective saturated film boiling two phase flow is carried out by solving the governing equations conservation of mass, momentum and energy. In the present study, no boundary layer approximations are made and 2-D, unsteady coupled governing equations are solved to obtain the steady state solutions. For unsteady, incompressible flow, the continuity equation can be written as, $\nabla\cdot V = 0$. It can be modified at the interface to include the effect of mass transfer as

$$\int_{S_c} \vec{V}\cdot\vec{n}\,ds + \int_{S_I(t)} \left(\frac{1}{\rho_l}-\frac{1}{\rho_g}\right)\frac{\left\|q_I''\right\|\cdot\vec{n}}{h_{lg}}\,ds = 0 \tag{1}$$

The modified momentum transport equation for the two phase flows can be written as

$$\rho_f\left(\frac{\partial V}{\partial t}+V\cdot\nabla V\right) = -\nabla P + \nabla\cdot\mu_f\left[\nabla V + (\nabla V)^T\right] + \rho_f g + F_{st} \tag{2}$$

where, the superscript T indicates the transpose operation of diffusion terms. F_{st} is the interfacial volumetric surface tension force and is non-zero only at the liquid-vapour interface cell region. The unit normal vector, mean curvature and the surface tension force across the interface is computed using the smoothed LS function, ϕ as

$$\vec{n} = \frac{\nabla\phi}{|\nabla\phi|}\ ;\ \kappa = -\nabla\cdot\left(\frac{\nabla\phi}{|\nabla\phi|}\right) \text{ and } F_{st} = \sigma\kappa\vec{n}\delta_s \tag{3}$$

In saturated film boiling, both the liquid and the interface are at saturation temperature. The energy equation for the superheated vapour phase region can be written as

$$\rho_f C_{Pf}\left(\frac{\partial T}{\partial t}+V\cdot\nabla T\right) = \nabla\cdot k_f(\nabla T) \tag{4}$$

For boiling flow problems, the thermo-physical properties of fluids are discontinuous at the interface leads to numerical instability. To overcome this problem, the fluids properties such as density, viscosity, specific heat and thermal conductivity are evaluated based on a smoothed Heaviside function, $H(\phi)$ as

$$\gamma_f = \gamma_g + \left(\gamma_l - \gamma_g\right) H\left(\phi\right) \tag{5}$$

The smoothed Heaviside function, $H(\phi)$ is computed using continuous LS function, ϕ as

$$H\left(\phi\right) = \begin{cases} 1 & \textit{Liquid phase}, \quad \textit{if } \phi > \varepsilon \\ 0.5 + \left(\dfrac{\phi}{2\varepsilon}\right) + \dfrac{1}{2\pi}\sin\left(\dfrac{\pi\phi}{\varepsilon}\right) & \textit{Interface}, \quad \textit{if } \left|\phi\right| \le \varepsilon \\ 0 & \textit{Vapor phase}, \quad \textit{if } \phi < -\varepsilon \end{cases} \tag{6}$$

where, ε is the width of interface region and computed using the grid size as $(3/2)\Delta x$. The transient motion of the two-phase interfaces is computed using the CLSVOF method by solving the advection equations of both LS and VOF functions. The advection equation of the VOF function, F for phase change process can be written as

$$\frac{\partial F}{\partial t} + (V \cdot \nabla)F = \ddot{m}_{fs} \tag{7}$$

The advection equation of the LS function, ϕ with phase change problem can be written as

$$\frac{\partial \phi}{\partial \mathrm{t}} + (V_{mt} \cdot \nabla)\phi = 0 \tag{8}$$

where, V_{mt} is the total cell velocity components and is taken as the sum of the cell velocity and the interface velocity due to mass transfer across the interface.

2.3 Numerical solution procedure

In this numerical study, a collocated grid based finite volume method is used. For velocity and pressure coupling, the SIMPLE algorithm has been used. A CLSVOF method with multi-directional advection algorithm developed by Ningegowda and Premachandran[10,11] for two-phase flows is used in the present numerical study. The Continuum Surface tension Force (CSF) model is used to compute the interfacial surface tension force. In the solutions of momentum and energy equations, the diffusion terms are discretized using the second order accurate CDS scheme and the convective terms are discretized using the third order accurate QUICK schemes. In the first order explicit time step scheme, stability constraints based on inertia, viscous, gravity and surface tension force are considered. For the VOF method, the interface position vector is geometrically reconstructed from the Piecewise Linear Interface Calculation (PLIC) method. The second order accurate Essentially Non-Oscillatory (ENO) scheme is used to solve the convective terms of the LS advection equation. For re-initialization of the LS function, ϕ, the PLIC based VOF geometric reconstruction algorithm have been considered.

3. Results and discussion

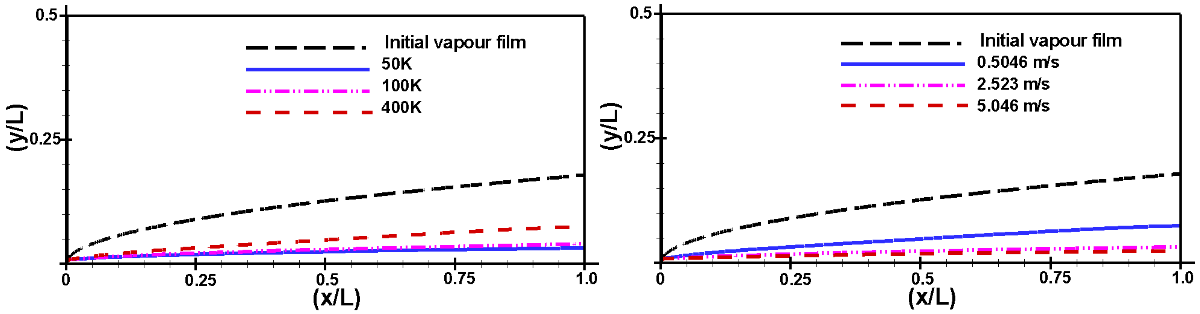
In this study, a detailed parametric study was carried out by varying the wall superheat and Reynolds number to understand the characteristics of the heat transfer and the fluid flow. In order to validate the present numerical model, the results obtained from the present numerical study are compared with those of Cess and Sparrow[1]. In this study, water at saturation conditions of *646.15K* and *21.9MPa* and refrigerant R-134a at saturation conditions of *370.16K* and *3.763MPa* are considered. Constant thermo-physical properties are used at the saturated conditions[11]. In the film boiling flows, the formation of additional vapour phase occurs across the liquid-vapour interface due to the interfacial mass transfer. This generated vapour phase avoids the direct contact of saturated liquid phase with the superheated wall surface. The inflow of saturated liquid pushes the vapour phase out of the domain through the right side boundary. In this numerical study, only the steady state solutions are considered.

The computational domain of height to length ratio, *H/L* of *0.5* is discretized into uniform control volumes. In order to ensure that the results obtained from the numerical simulations are independent of the grid size, a grid independence study was carried out. For the grid independence study, the grid sizes of *120×60*, *240×120*, *360×180* and *480×240* were selected. Water was considered as the working fluid. The wall superheat of 100K and the Re = 1×10^5 were selected. For the time step convergence study, CFL number, $C = U_\infty \Delta t/\Delta x$ of *0.1, 0.05, 0.01* and *0.005* were used. From the grid independence study and time step convergence study, negligible variations in both vapour film thickness and the wall Nusselt number were observed for the grid sizes of *360×180* and *480×240* and for the CFL numbers of *0.01* and *0.005*. To avoid additional computational power, the grid size of *360×180* and the CFL number of 0.01 were considered for further numerical simulations.

In this study, the effect of wall superheat and the free stream velocity are varied to understand fluid flow and heat transfer characteristics of film boiling as follows: (a)For both working fluids, wall superheats $\Delta T_{Sup} = (T_W - T_{Sat})$ are varied from *50K* to *400K* at uniform free stream velocity, U_∞ m/s corresponds to Reynolds number $Re_L=(\rho_l U_\infty L/\mu_l)$ of 1×10^4 and (b) To study the effect of liquid free stream velocities U_∞, the Reynolds number, Re_L is varied from 1×10^4 to 1×10^5 at ΔT_{Sup} of *400K*.

3.1 Heat transfer analysis

The steady state vapor film thickness obtained for water at different wall superheats and different free stream velocity conditions are shown in Fig. 2 (a) and (b), respectively. At constant free stream velocity, U_∞ m/s corresponds to $Re_L = 1\times10^4$, increase in the wall superheats increases the thickness of the vapour layer due to higher generation of vapour across the interface. The variation of local vapour thickness for various wall superheats is shown in Fig. 2 (a). At the wall superheat of 400K, the numerical results of film boiling flow show that increase in the liquid free stream velocity reduces the thickness of the vapor film as shown in Fig. 2 (b).



(a) (b)

Fig. 2 Steady state solutions for water vapour film thickness at time, t=0.4s (a) effects of wall superheats and (b) effects of free stream velocities.

As mentioned earlier, in the present numerical study, numerical model was developed without boundary layer approximation and the results obtained from the numerical simulations are compared with those of Cess and Sparrow[1]. The local Nusselt number expression given by Cess and Sparrow[1] as

$$\frac{Nu}{\sqrt{\mathrm{Re}}}\left(\frac{\mu_g}{\mu_l}\right)\left[1+\sqrt{\pi}\left(\frac{Nu}{\sqrt{\mathrm{Re}}}\right)\left(\frac{\mu_g}{\mu_l}\right)\right]^{\frac{1}{2}} = \frac{1}{2}\left(\frac{(\rho\mu)_l}{(\rho\mu)_g}\frac{Cp_g\Delta T}{h_{lg}\,\mathrm{Pr}}\right)^{-\frac{1}{2}} \tag{9}$$

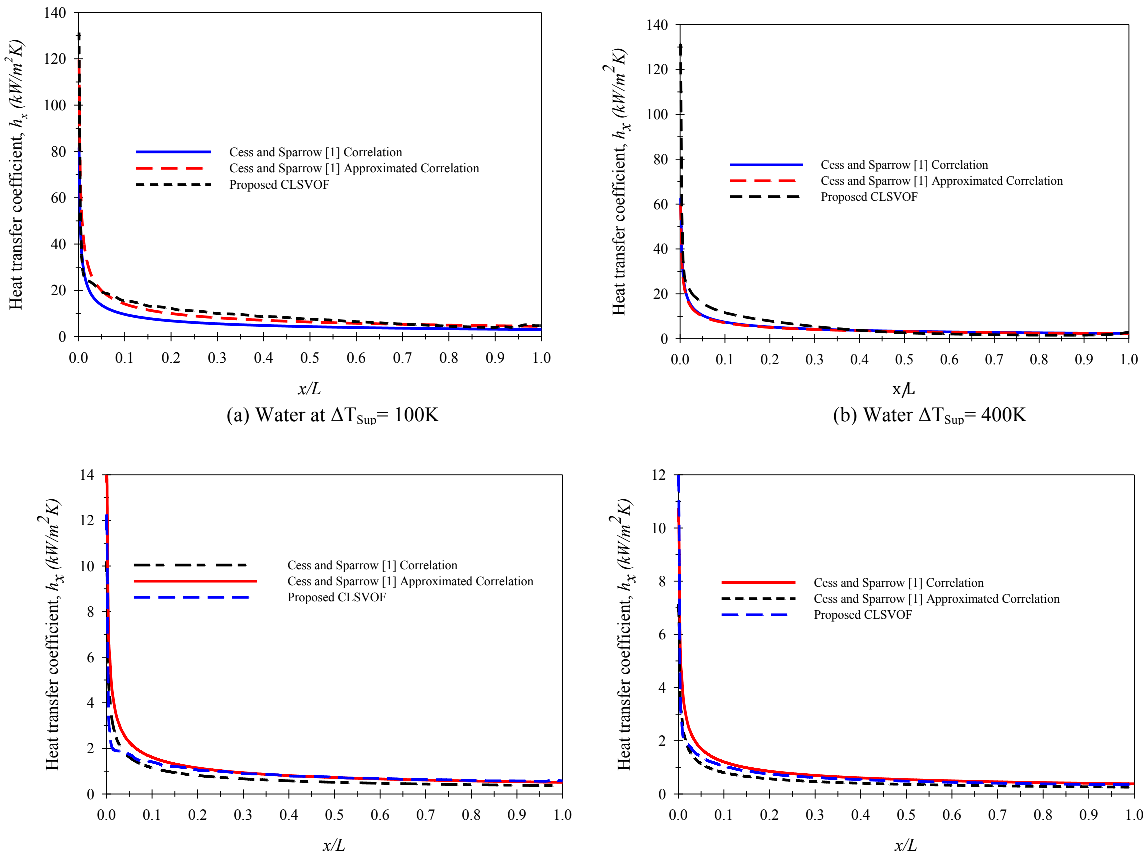
To overcome the difficulty arise in the iterative solution of eqn. (9), Cess and Sparrow[1] approximated eqn. (9) as

$$\frac{Nu}{\sqrt{\mathrm{Re}}}\left(\frac{\mu_g}{\mu_l}\right) = \frac{1}{2}\left(\frac{(\rho\mu)_l}{(\rho\mu)_g}\frac{Cp_g\Delta T}{h_{lg}\,\mathrm{Pr}}\right)^{-\frac{1}{2}} \tag{10}$$

In the numerical study, the local wall Nusselt number distribution is computed as

$$Nu = \frac{h_x\, x}{k_f};\ \text{where}\ h_x = \frac{q_w}{(T_w - T_{sat})};\ q_w = -k_f\left(\frac{\partial T}{\partial y}\right)_{y=0}\ \text{and}\ \ \mathrm{Re}_x = \frac{\mu_l\, U_\infty}{\rho_l}x \tag{11}$$

For both water and refrigerant R-134a, the local wall heat transfer coefficient, h_x obtained from the numerical study are compared with the values obtained using eqns. (9) and (10) for two wall superheats of *100K* and *400K* at uniform free stream velocity corresponds to $Re_L = 1\times10^4$ are shown in Fig. 3 (a)-(d). The local heat transfer coefficient obtained from the present numerical study for both fluids show good agreement with the value obtained using eqn.(10) as compared to the exact eqn. (9). As the wall superheat increases, the heat transfer rate from the heater surface decreases due to decrease in the temperature gradient near the heated wall.



(a) Water at ΔT_{Sup}= 100K (b) Water ΔT_{Sup}= 400K

(a) R-134a at ΔT_{Sup}= 100K (b) R-134a at ΔT_{Sup}= 400K

Fig. 3 Effects of wall superheats based for the local wall heat transfer coefficients compared with Cess and Sparrow [1] correlations

3.2 Skin friction analysis

Based on the similarity solutions of film boiling flows, Cess and Sparrow[1] provided the following expression for the local wall shear stress over the heated flat surface as

$$\frac{\dfrac{\tau\sqrt{\mathrm{Re}}}{\rho_l U_\infty^2}}{\left[1-\sqrt{\pi}\,\dfrac{\tau\sqrt{\mathrm{Re}}}{\rho_l U_\infty^2}\right]^{\frac{3}{2}}}=\frac{1}{2}\left(\frac{(\rho\mu)_l}{(\rho\mu)_g}\frac{Cp_g\Delta T}{h_{\mathrm{lg}}\,\mathrm{Pr}}\right)^{-\frac{1}{2}} \tag{12}$$

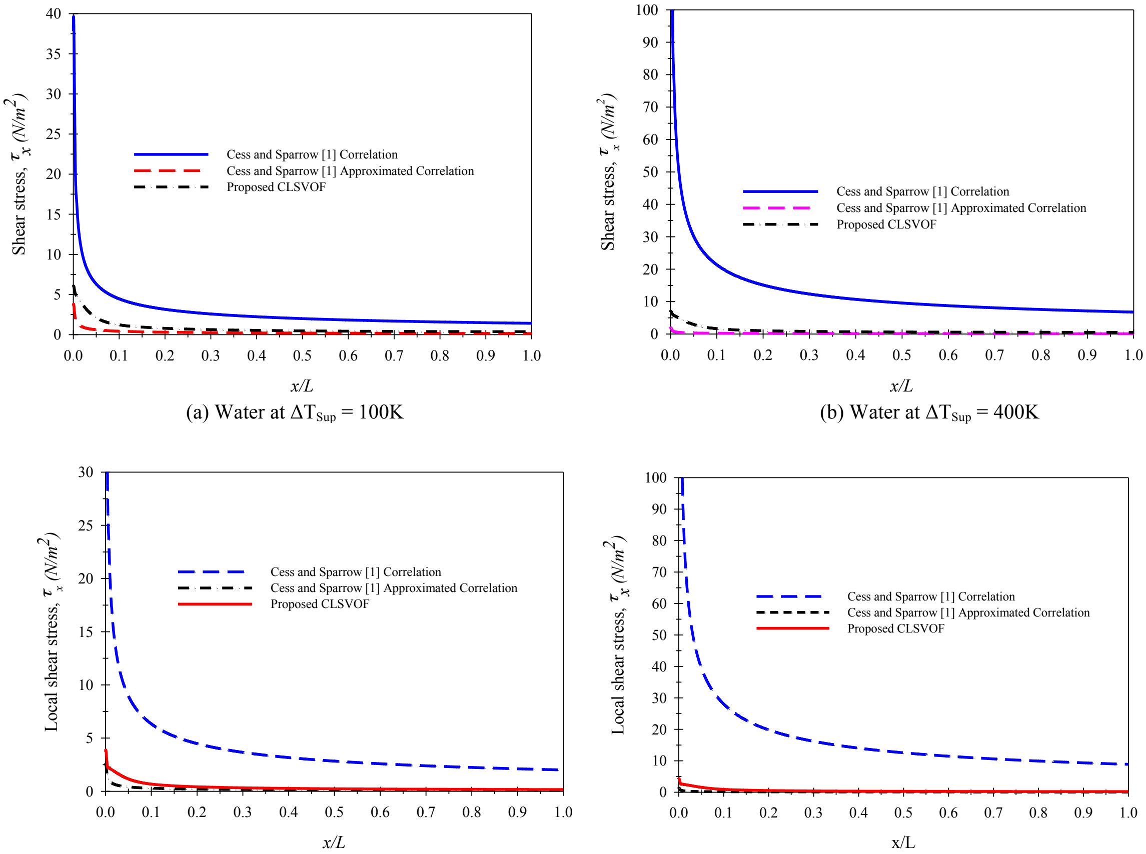
To overcome the iterative solution of eqn. (12), Cess and Sparrow[1] approximated the expression for wall shear stress as

$$\frac{\tau\sqrt{\mathrm{Re}}}{\rho_l U_\infty^2}=\frac{1}{2}\left(\frac{(\rho\mu)_l}{(\rho\mu)_g}\frac{Cp_g\Delta T}{h_{\mathrm{lg}}\,\mathrm{Pr}}\right)^{-\frac{1}{2}} \tag{13}$$

In the present numerical study, the local wall shear stress is computed as

$$\tau_x = \mu_f \left(\frac{\partial u}{\partial y} \right)_{y=0} \tag{14}$$

For two different working fluids at different wall superheat conditions, the local wall shear stress results obtained from the present numerical study are compared with the correlations of Cess and Sparrow[1] are shown in Fig. 4(a)-(d).

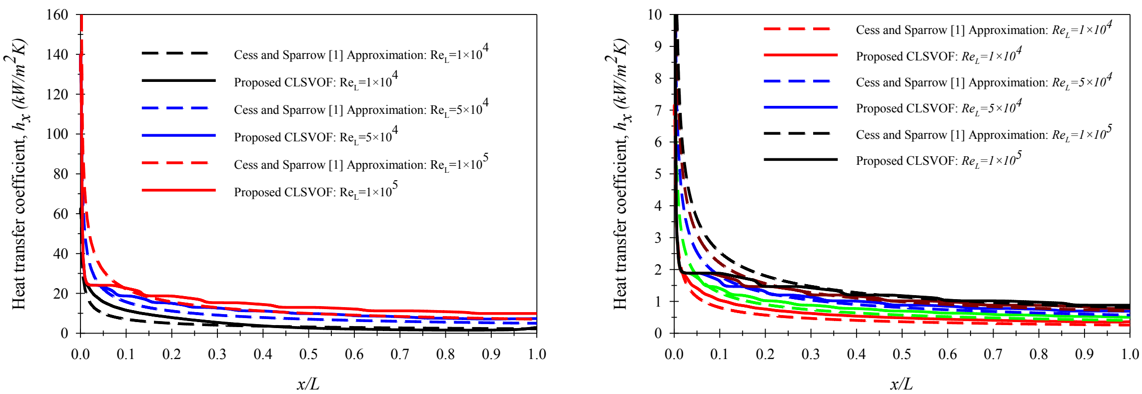


(a) Water at ΔT_{Sup} = 100K
(b) Water at ΔT_{Sup} = 400K
(c) R-134a at ΔT_{Sup} = 100K
(d) R-134a at ΔT_{Sup} = 400K

Fig. 4 Effects of wall superheats based on the local wall shear stress compared with Cess and Sparrow [1] correlations.

The numerical results obtained for both fluids show good agreement with Cess and Sparrow's approximated correlation (eqn.(13)) as compared to exact correlation (eqn.(12)). Both numerical results and the values obtained from the similarity solutions show that increase in the wall superheat increases the local wall shear stress due to the increase in the magnitude of velocity of the vapour film adjacent to the heated wall surface.

At different wall superheats conditions, the heat transfer and the wall shear stress obtained from the numerical result shows qualitatively good agreement with the boundary layer correlations for both the fluids. Similarly, the effect of free stream velocities corresponds to Reynolds numbers Re_L of 1×10^4, 5×10^4 and 1×10^5 at uniform superheats of *400K* are also studied. The results obtained for various free stream velocities are shown in Fig. 5. For both the fluids, as Reynolds number increases from 1×10^4 to 1×10^5 increases the heat transfer coefficients as shown in Fig. 5 (a) and (b) for water and refrigerant R-134a, respectively. Increase in Reynolds number increases the wall shear stress and heat transfer rate. However, as Reynolds number increases, the vapour film thickness over the heated plate decreases.



(a) Water (b) R-134a

Fig. 5 Effects of liquid free stream velocities based on the local heat transfer coefficients.

4. Conclusions

A numerical investigation of 2-D forced convective laminar saturated film boiling flows over the horizontal superheated flat plate has been carried out using a CLSVOF interface capturing method for different wall superheats and liquid free stream velocities. Water and R134a are considered as working fluids. The present numerical results of both heat transfer coefficients and the wall shear stress show good agreement with those of analytical solutions. At a fixed uniform liquid inflow condition, increase in the wall superheat increases both thickness of the vapour film and the wall shear stress and decreases the heat transfer coefficients from heated wall surface. At a constant wall superheat condition, increase in the liquid free stream velocity decreases the vapour film thickness and increases both heat transfer coefficient and the wall shear stress over the wall surface.

5. Acknowledgements

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