CONJUGATE HEAT TRANSFER IN TWO-PHASE FLOW IN MICROTUBE

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ABSTRACT

Flow boiling in microchannel is a promising way for removal of high heat flux. This work presents a systematic two-dimensional numerical study of the conjugate effect of heat transfer in two-phase flow in a microtube subjected to constant heat flux. The fluid is assumed to be water which enters in the microtube with a preheated temperature of 373 K. It is intended to obtain the enhancement in heat transfer by taking into account the flow boiling effect. Furthermore, the effect of wall to fluid conductivity ratio ($k_{sf}$), tube wall thickness to tube inner radius ratio ($\delta/L$), and inlet Reynolds number in an unsteady flow boiling of water in a microtube under constant heat flux are studied. The result indicates that the mixing of fluid layer is improved due to presence of vapor bubbles along the length of the microtube that results in improvement in overall heat transfer coefficient. The growth and motion of vapor bubble due to evaporation are observed.

NOMENCLATURE

- $c_p$: Specific heat of fluid, J/kgK
- $h_z$: Local heat transfer coefficient, W/m²K
- $k_s$: Solid thermal conductivity, W/mK
- $k_f$: Fluid thermal conductivity, W/mK
- $k_{sf}$: Ratio of $k_s$ and $k_f$
- $k_{eff}$: Effective thermal conductivity, W/mK
- $L$: Total length of tube, m
- $q'$: Heat flux experienced on the outer surface of the microtube, W/m²
- $R$: Inner radius of microtube, m
- $r$: Radial coordinate, m
- $T_f$: Bulk fluid Temperature, K
- $T_w$: Wall temperature, K
- $u$: Velocity in the axial direction, m/s
- $\bar{u}$: Average velocity at inlet, m/s
- $z$: Axial coordinate, m
- $z^*$: Non-dimensional axial coordinate, z/L

Greek symbols
\[
\begin{align*}
\delta_t & \quad \text{Inner radius of the tube, m} \\
\delta_i & \quad \text{Thickness of the tube wall, m} \\
\delta_f & \quad \text{Ratio of } \delta_i \text{ and } \delta_t \text{ (-)} \\
\nabla & \quad \text{Differential parameter} \\
\mu & \quad \text{Dynamic viscosity, Pa-s} \\
\rho & \quad \text{Density, kg/m}^3 \\
\text{Subscripts} & \\
\text{f} & \quad \text{Fluid} \\
\text{i} & \quad \text{Inner surface of tube} \\
\text{o} & \quad \text{Outer surface of tube} \\
\text{s} & \quad \text{Solid} \\
\text{w} & \quad \text{Solid-fluid Interface at the inner surface of the wall} \\
\end{align*}
\]

INTRODUCTION

The International Technology Roadmap of Semiconductors (ITRS 2015) predicted that the average heat flux values in the high performance electronic systems could reach to 2.5 – 4.5 MW/m² by 2029. Therefore, cooling of such high performance electronic devices can be achieved through microfluidic cooling (single-phase / two-phase) only. In the field of microchannel systems, flow boiling can be an effective solution for the removal of high heat flux utilizing latent heat of evaporation. Moreover, in microchannel, the flow is mostly laminar due to smaller hydraulic diameter along with smaller flow Re. The small hydraulic dimensions of microchannel flow passages present a large frictional pressure drop in flows. Hence, to improve the overall thermal performance, the two-phase flow boiling heat transfer is gaining greater attention in recent years. The fundamental issues of flow boiling in small hydraulic diameter channels has been discussed in [1-3]. They stated that flow pattern, heat transfer coefficient, frictional pressure drop and bubble nucleation time and temperature are the important parameters that controls the overall thermal performance of a two-phase flow microchannel. For example, Karayiannis & Mahmoud [4] presented different mechanism of boiling in microchannel as shown schematically in Fig. 1. The nucleate and convective boiling are dominant flow boiling mechanisms. They also reported that the dominance of the heat flux was not only parameter that causes flow boiling. The surface roughness and the flow regime are also important. To observe the growth rate of bubble and liquid-vapor phase change, Gong and Cheng [5] used a lattice Boltzmann model.

Similarly, many authors [6-8] had reported that the nucleate boiling and convective boiling are effective mechanisms for improvement in overall performance of the microtube/ microchannel. The mixing of boundary layer is an appropriate option. For this purpose, flow boiling in microchannel plays the dominant role. Therefore, with the presence of bubble the mixing of layers improves and increases the overall thermal performance. Wang et al. [9] observed the flow instabilities in a parallel microchannel. The high degree of wall superheat required to initiate nucleation in microchannel leads to rapid evaporation and flow instabilities, often resulting in flow reversal. Effect of bubble contact angle at the surface is also an important parameter that requires effective technique for the separation of bubble and mixing in the fluid. It is also reported in the literature [10-12] that the contact angle hysteresis is strengthened with decreasing the viscosity ratio, and the surface force strongly influenced by the channel material.

The open literature already revealed that in conjugate heat transfer scenario, the axial wall conduction is the dominant factor and it reduces the overall heat transfer coefficient [13-16]. Though many studies of two-phase flow boiling exist in the literature, none have reported the effect of conjugate heat transfer in two-phase flow microchannel.

In this numerical work, parametric variations considered to explore the effect of conjugate heat transfer in two phase flow boiling in a microtube subjected to constant heat flux boundary condition. The parameters considered in this study are: Flow Re = 100, solid to fluid conductivity ratio \((k_{sf}) \sim 2.26 \cdot 646\) and tube wall thickness to tube inner radius ratio \((\delta_t)/\delta_i \sim 1-5\).
fraction of the vapor phase is set to be 0 where the temperature of water at inlet is 373 K. No slip boundary condition is used at the solid-fluid interface wall of the microtube. Pressure boundary is used at the outlet of the microtube which is equal to atmospheric pressure. Walls exposed to the ambient are adiabatic and constant heat flux boundary condition is applied on the outer surface of the microtube. The governing equations are as follow:

\[
 \nabla \bar{u} = 0
\]

\[
 \frac{\partial}{\partial \bar{t}} (\rho \bar{u}) + \nabla \cdot (\rho \bar{u} \bar{u}) = -\nabla p + \nabla \cdot \left[ \mu (\nabla \bar{u} + \nabla \bar{u}^T) \right] + \rho g + F - \rho C_p \left( \frac{\partial T}{\partial \bar{t}} + \bar{u} \nabla \cdot T \right) = \nabla \cdot (k_{sf} \nabla T)
\]

Commercially available ANSYS-FLOUT used for this simulation. For pressure transclusion, the PRESTO (pressure staggering option) strategy was utilized and for the energy and momentum equations, QUICK (Quadratic Interpolation for Convective Kinetics) was chosen. The PISO (pressure-implicit with splitting of operators) strategy was selected for pressure-velocity coupling. The ratio of the time step to the resistance of the cell time defined by the Courant number is adjusted to be 0.25 for the calculation of volume fraction.

**RESULT AND DISCUSSION**

A microtube of inner diameter 0.4 mm and length 60 mm are considered for the computational domain with varying wall thickness as shown in Fig. 2. A uniform heat source imposed at the outer surface of the tube, and walls exposed to ambient as insulated. In conjugate heat transfer, the actual solid-fluid interface is away from the outer surface. Hence, the actual thermal boundary condition is different from the ideal condition. We study the heat transfer behavior of two-phase flow along the tube, for constant heat flux, with axial wall transfer in the tube wall.

Fig. 3 and Fig. 4, shows the growth of bubbles (different time steps) at the solid-fluid interface and their motion away from the surface. The tube thickness considered in the simulation where the thickness ratio \((\delta_{sf})\) varied from 1 – 5. The solid portion is not shown in Fig. 3 & 4 for better visualization of the fluid region. The results demonstrated different mechanism during the heat transfer process. The mechanisms are: (a) The liquid slug flow (see Fig. 3a and Fig. 4a), where the heat transfer process shows the single-phase flow behavior. (b) As the phase change process going the vapor bubble form at the interface wall (Fig. 3c) and from here, convective flow boiling process takes place. (c) Thin film evaporation, at the interface wall is formed, which is clearly observed in Fig. 4 (b, c). This results in small fluctuation of heat flux and the fluid reaches to partial dryout state. Hence, for the completion of the overall phase change process, above flow boiling mechanism is used.

The axial variation of wall temperature, bulk- fluid temperature, and vapor fraction of liquid is reported in Fig. 5. Ideally, as the working fluid (water) gains the heat, and the water temperature goes on increasing until temperature reaches to the saturation temperature (373.15 at 1 atm). The vaporization process takes place at constant temperature (saturation temperature) after that point along the tube length. After completion of vaporization process, the vapor temperature goes on increasing depending on the heat source. This ideal axial temperature variation is shows in Fig. 5 by the dash line. Here water enters the tube at 373 K. Thus, the water reaches to its saturation temperature and vaporization process starts with very small heat input. This also saves lot of computational time.

Here in Fig. 5, two cases of tube thickness \((\delta_{sf} = 1, 5)\) are reported. For \(\delta_{sf} = 1\), the variation of fluid temperature deviates from the ideal behavior in the liquid-vapor region. Initially the temperature decreases, and then it increases. This happens due to formation vapor layer (see Fig. 3), which act as barrier to heat transfer from the wall to the fluid due to low specific heat of vapor compared to the liquid.

Similarly, for \(\delta_{sf} = 5\), little fluctuation in fluid temperature is observed during the phase change process. This is due to axial wall conduction because of large wall thickness compared to inner radius of the tube. Therefore, less amount of heat reaches to the solid-fluid interface, thus less amount of water converted into vapor. The vapor density is also low in case of high thickness ratio (see Fig. 4).

The difference in axial wall temperature variation for two \(\delta_{sf}\) values can be observed clearly. At low tube thickness to inner radius ratio \((\delta_{sf} = 1)\), the trend of wall temperature is similar to the theory of constant heat flux boundary condition. But near the outlet, the temperature increases rapidly for both \(\delta_{sf} = 1\) and 5. This happened due to most of the liquid are converted into the vapor
phase. For the high tube thickness to inner radius ratio ($\delta_d = 5$), the trend of wall temperature is similar to the constant wall temperature boundary condition except near the outlet region.

The local heat transfer coefficient vs. local vapor quality for two different tube material ($k_d = 2.26, 646$) are reported in Fig. 6. For both cases, constant heat flux of 20 kW/m$^2$ is applied on the outer surface of microtube. At the solid fluid wall interface, the heat flux continuously adds up in the single-phase region. In addition, liquid gets converted into the vapor in the form of bubbles. However, due to formation of layer of vapor bubble at the interface wall, less amount of heat transfer takes place from the wall to the liquid. In this case, the basic mechanism of heat transfer is the amount of heat flux taken from single-phase region and adds it to the two-phase region in the axial direction. Due to axial flow of liquid, the vapor bubbles get separated from the interface, which can be clearly seen in Fig. 3(b) and Fig. 4(c). Therefore, the vapor bubbles get distributed over the full channel length. This results in non-uniform distribution of heat. This causes fluctuation in the values of local heat transfer coefficient along the tube length, which can be clearly seen in Fig. 6. For low thermal conductivity ratio $k_d = 2.26$ (see Fig. 6a), the local heat coefficient attains the lower values for all set of $\delta_d$. This happens because low conductive material offers higher thermal resistance and vapor bubbles (having low specific heat) also present at the interface wall. Their combined effect reduces the overall heat transfer coefficient. Again, for $\delta_d = 5$, the vapor quality is low as compared to $\delta_d = 1$. Similarly, for $k_d = 646$ and $\delta_d = 1$ (see Fig. 6b), the local heat transfer coefficient attains higher values because higher conductive material offers higher heat transfer rate. For higher $\delta_d (= 5)$, local heat transfer coefficient value decreases because more heat flow occurs in the axial direction in the solid wall. This shows the dominance of axial wall conduction.

CONCLUSION

Conjugate heat transfer in two phase flow microtube is studied numerically to find the basic difference in the conjugate heat transfer process experienced in a microtube subjected to constant heat flux imposed on its outer surface. Water is considered as the coolant, which flows through the microtube, which enters at 373K. The cross-sectional faces of the tube are considered to be adiabatic. The effect of wall thickness, wall material conductivity, and thermal and fluid-dynamic behavior of two-phase flow during evaporation are studied. The dimensionless variables considered in this study are (i) solid wall thickness to inner radius ratio ($\delta_d \sim 1$ to 5), and solid wall to working liquid conductivity ratio ($k_d \sim 2.2$ to 646). It is observed that with increasing the thickness of the microtube, the axial wall conduction dominants; thus, it distorts the heat flux at the solid-fluid interface results low values of local heat transfer coefficient. Similarly, the presence of vapor bubbles at the interface wall is also a dominant factor. In addition, induces axial wall conduction inside the tube wall.

REFERENCES


FIGURE 1. SCHEMATIC REPRESENTATION OF DIFFERENT FLOW BOILING MECHANISMS IN SMALL TO MICRO DIAMETER CHANNELS [4].

FIGURE 2. SCHEMATIC REPRESENTATION OF THE CONJUGATE MODEL SHOWING AXIAL WALL CONDUCTION AND BUBBLE FORMATION.
FIGURE 3. MOTION AND DENSITY OF VAPOR BUBBLES DUE TO EVAPORATION INSIDE THE TUBE AT DIFFERENT TIME STEP
{(a) 0.5 s, (b) 1 s, and (c) 1.5 s} FOR $\delta_{SF} = 1$
FIGURE 4. MOTION AND DENSITY OF VAPOR BUBBLES DUE TO EVAPORATION INSIDE THE TUBE AT DIFFERENT TIME STEP 

(a) 0.5 s, (b) 1 s, and (c) 1.5 s) FOR δSSF = 5

FIGURE 5. TEMPERATURE DISTRIBUTION OF THE FLUID (Tf) AND WALL (Tw) ALONG THE LENGTH OF THE TUBE (COPPER).
FIGURE 6. LOCAL HEAT TRANSFER COEFFICIENT VS. LOCAL VAPOR QUALITY