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Effect of inclination on the convective boiling performance of a microchannel heat sink using HFE-7100

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ABSTRACT

The effect of inclination on the convective boiling heat transfer characteristics of the dielectric fluid HFE-7100 within a multiport microchannel heat sink having a hydraulic diameter of 825 μm is studied. The inclinations spans from −90° (vertical downward) to 90° (vertical upward), and a flow visualization is also conducted in this study. It is found that the heat transfer coefficient for the vertical upward and horizontal is comparable, and the heat transfer coefficient for 45° upward considerably exceeds other configurations. On the other hand, the enhancement of heat transfer coefficient under inclined arrangement is reduced with the rise of mass flux. The results also indicate that downward arrangements always impair the heat transfer performance, and more than 50% deterioration of heat transfer coefficient is encountered for $G = 100\ \mathrm{kg\ m^{-2}\ s^{-1}}$ relative to that of 45° arrangement. The flow visualization confirms that the heat transfer augmentation for upward inclination is due to the rise of slug velocity.

1. Introduction

Flow boiling in microchannels is very effective in thermal management of high-flux modern electronics. To avoid electric hazards of electronic equipment or IC component, the dielectric fluids such as fluorocarbon fluids featuring excellent electrical and chemical properties, are the leading candidates for such applications. Since the working fluid in the microchannels may be in direct contact with the electronics, the use of dielectric liquids has drawn recent attention for these applications. There had been some studies associated with the connective boiling performance for dielectric fluids in microchannels [1–5], and these studies were focused upon the FC series dielectric fluids.

However, FC series coolants are Fluorinert liquids which have high global warming potentials (GWP) and long atmospheric lifetimes. As a result, a family of low-global warming materials (HFE series, segregated hydrofluoroethers) designed to balance performance with favorable environmental and worker safety properties were then developed (3M [6]). In this regard, it is the purpose of this study to examine the associated heat transfer performance of HFE-7100 within multi-channel microchannel heat sink. In addition, in practice the multiport microchannel heat sink may be placed vertically, horizontally, or even inclined at a specific angle. However, the majority of the published data concerning the influence of inclination were mainly for macro size channel, the only publications relevant to the influence of inclinations on the performance of microchannel heat sink were carried out by Kandlikar and Balasubramanian [7] and Zhang [8]. Kandlikar and Balasubramanian [7] performed two-phase flow visualization as well as heat transfer measurements in a 1054 × 197 μm parallel microchannel (hydraulic diameter = 332 μm) using water. Their results showed that the heat transfer coefficients for horizontal and upward vertical are the same but the heat transfer coefficient for vertical downward arrangement is impaired by 30–40%. Zhang [8] conducts two-phase boiling experiments of FC-72 flow in a microchannel heat sink with the channel dimensions of 0.2 mm (W) × 2 mm (H) × 15 mm (L), and presented results for three different orientations: vertical upflow (VU), vertical downflow (VD) and horizontal flow with horizontal facing (HH). Their results showed that comparable thermal resistance amid vertical downflow and horizontal arrangement, and the thermal resistance of vertical upflow is slightly lower (∼5%).

As seen, the existing results about the influence of inclination are not conclusive. Moreover, the published results were applicable only to horizontal and vertical arrangements, there were no systematic information about the influence of inclination on the two-phase convective boiling in microchannels. In this sense, it is

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Nomenclature

A	surface area (m^2)
c_p	specific heat (J kg^{-1} K^{-1})
D_h	hydraulic diameter (m)
Fr	Froude number
G	mass flux (kg m^{-2} s^{-1})
h	heat transfer coefficient (W m^{-2} K^{-1})
I	current (A)
i_{fg}	latent heat of HFE-7100 (J kg^{-1})
k_s	thermal conductivity of cold plate (W m^{-1} K^{-1})
$\dot{m}$	mass flow rate (kg s^{-1})
q	heat flux (W m^{-2})
Q'	heat transfer into the cold plate (W)
Q_{input}	supplied input power (W)
Q_{loss}	heat loss across backlite (W)
T_s	saturation temperature (K)
$\overline{T}_{wall}$	average surface temperature of cold plate (K)
$\overline{T}_{b,wall}$	average surface temperature of the thermocouple measurements (K)
V	voltage (V)
x	average vapor quality of inlet and outlet
x_{in}	vapor quality of inlet

Greek symbols

δ_w	thickness of the cold plate (m)
θ	inclination angle from horizon (°)
ΔT_m	effective mean temperature difference (K)
ΔT_{sub}	inlet subcooling of HFE-7100 (K)

the objective of this study to examine the associated influence of orientation on the overall performance of a microchannel heat sink.

2. Experimental setup

The schematic of the experimental apparatus and test section is shown in Fig. 1a. The experimental setup comprises of the following main loop, namely, a dielectric fluid loop, a HFE-7100 degas device, a water loop for pre-heater, a sub-cooler and a condenser, along with measurement units and data acquisition system. Among these loops, the preheater loop is for controlling the inlet quality into the test section whereas the sub-cooler loop is to ensure fully liquid state before flowing into the preheater loop for easier calculation of the heat into the preheater.

Under ambient condition, HFE7100 contains 53% of air by volume; implying that a unit of HFE-7100 liquid will contain 0.53 unit of air at standard pressure and temperature, which is equivalent to a concentration of about 366 ppm. For reference, the air content in water under similar condition is only 8.5 ppm. Hence, it is necessary to build up a degas device for the test fluid. A separate degas device is used in this study, and its schematic and operation details can be found from [9]. Before conducting the experiments, the HFE-7100 is first circulated into a leak-free tank below which a uniform Kapton heater is placed. The HFE-7100 is then boiled up into vapor, it then carries along the non-condensable gas toward the graham condenser where the vapor HFE-7100 is condensed again and return to the tank whereas the non-condensable is relieved to the ambient. The degas process continues for about one hour to ensure the deviation between the vapor pressure and its corresponding temperature of this measured vapor pressure to be within ±0.2 °C.

The microchannel is made of copper via precise machining. The dimension of the micro channel heat sinks is 25.4 mm × 25.4 mm with the corresponding rectangular microchannels with a hydrodynamic diameter of 825 μm. A total of five inclinations for the heat sink are conducted during the experiments, with the inclination angle (θ) of −90° (vertical downward), −45°, 0° (horizontal), 45°, and 90° (vertical upward) as shown in Fig. 1b. The detailed dimension along with inlet location of the heat sink is shown in Fig. 2a. Thermocouples are used to measure the surface and fluid temperature. A total of nine T-type thermocouples are placed beneath the cold plate for measurement of the average surface temperature whereas two thermocouples are used to record the inlet and outlet temperature of HFE-7100 across the cold plate. The locations of the thermocouples are distributed below the surface of cold plate as schematically shown in Fig. 2b. The thermocouples were pre-calibrated with an accuracy of 0.1 °C. The test cold plate is located above a well-fitted bakelite. A transparent glass is placed on top of the test section. Observations of flow patterns are obtained from images produced by a high speed camera of Redlake Motionscope PCI 8000 s. The maximum camera shutter speed is 1/8000 s. The high-speed camera can be placed at any position above the square microchannel. To minimize the effect of maldistribution caused by the inlet, an inlet plenum is made at the entrance of the test section whereas the inlet is placed at the bottom of the plenum as seen in Fig. 2c. With this design, the working fluid enters into the plenum and rises gradually before it distributes quite evenly into the multiport microchannels. Similarly, a downstream plenum having a similar configuration is exploited to reduce the effect from the downstream. A 10 mm thick rubber insulation is wrapped around the cold plate and bakelite to minimize the heat loss to the surrounding. At the inlet and outlet of the cold plate, a precise differential pressure transducer having a full scale accuracy of 0.1% is used to measure the pressure drop across the cold plate.

3. Data reduction

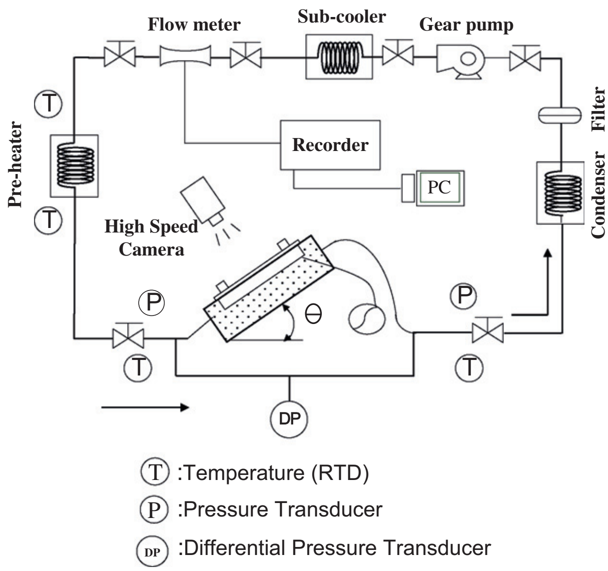
The measured temperatures at the nine locations underneath the microchannels were first corrected to obtain the corresponding wall temperature at the inner wall surface via one-dimensional conduction equation. i.e.

$$\overline{T}_{wall} = \overline{T}_{b,wall} - \frac{Q'\delta_w}{k_s A} \tag{1}$$

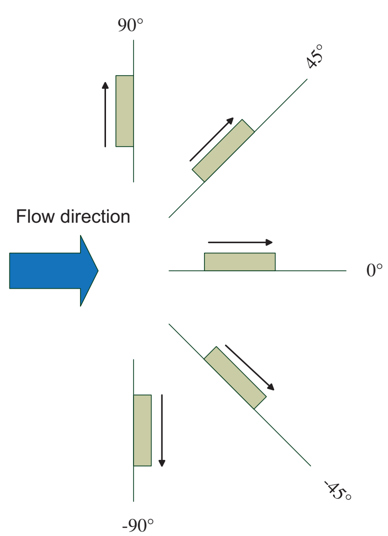
where $\overline{T}_{wall}$ is the average surface temperature of cold plate whereas $\overline{T}_{b,wall}$ is the average surface temperature beneath the cold plate and δ_w the thickness of the cold plate. A is the total surface of the microchannel cold plate above the heater. k_s is the corresponding thermal conductivity of cold plate (copper). Q' is the heat transfer into the cold plate, and is obtained by subtracting the heat loss from the input power:

$$Q' = Q_{input} - Q_{loss} \tag{2}$$

where Q_{input} ($I \times V$) is the supplied input power and Q_{loss} is the heat lost across bakelite which can be estimated from the one dimensional heat conduction equation from the measured temperature difference of the thermocouples that were placed above and below the bakelite. Note that the thermocouples were pre-calibrated with a precision of 0.1 °C. The average heat transfer coefficient can be obtained as follows:

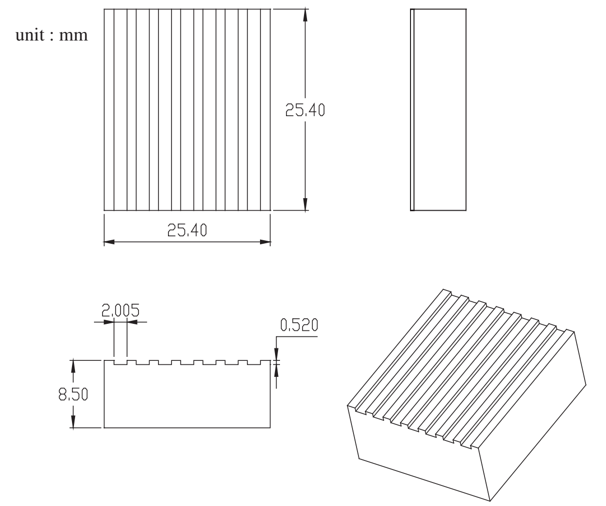


(a) Test Facility

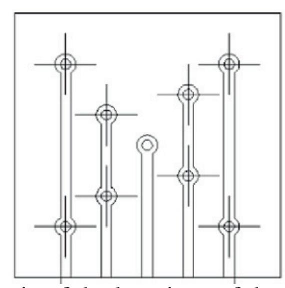


(b) Inclinations

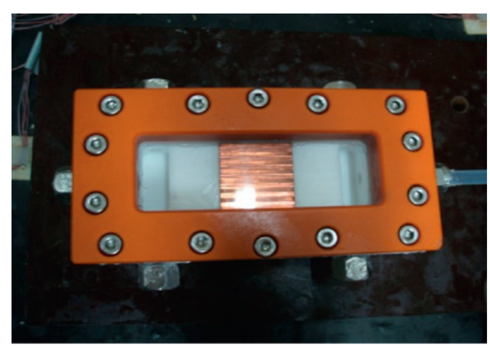
Fig. 1. Schematic of the test loop and tested inclinations.



(a) Detailed dimension of the test section with D_h = 825 μm.



(b) Schematic of the locations of the thermocouples.



(c) Photo of the inlet/outlet plenum design of test section

Fig. 2. Configuration of the microchannels heat sink test section.

$$h = \frac{Q'}{A\Delta T_m} \tag{3}$$

where A is total surface area and ΔT_m is the effective mean temperature difference, and is calculated from the following:

$$\Delta T_m = \overline{T}_{wall} - T_s \tag{4}$$

Notice $\overline{T}_{wall}$ is the mean surface temperature on the bottom of microchannel heat sink and T_s is the saturate temperature of HFE-7100 based on the inlet pressure. $\overline{T}_{wall}$ can be easily calculated from the average of nine thermocouples measurements using Eq. (1).

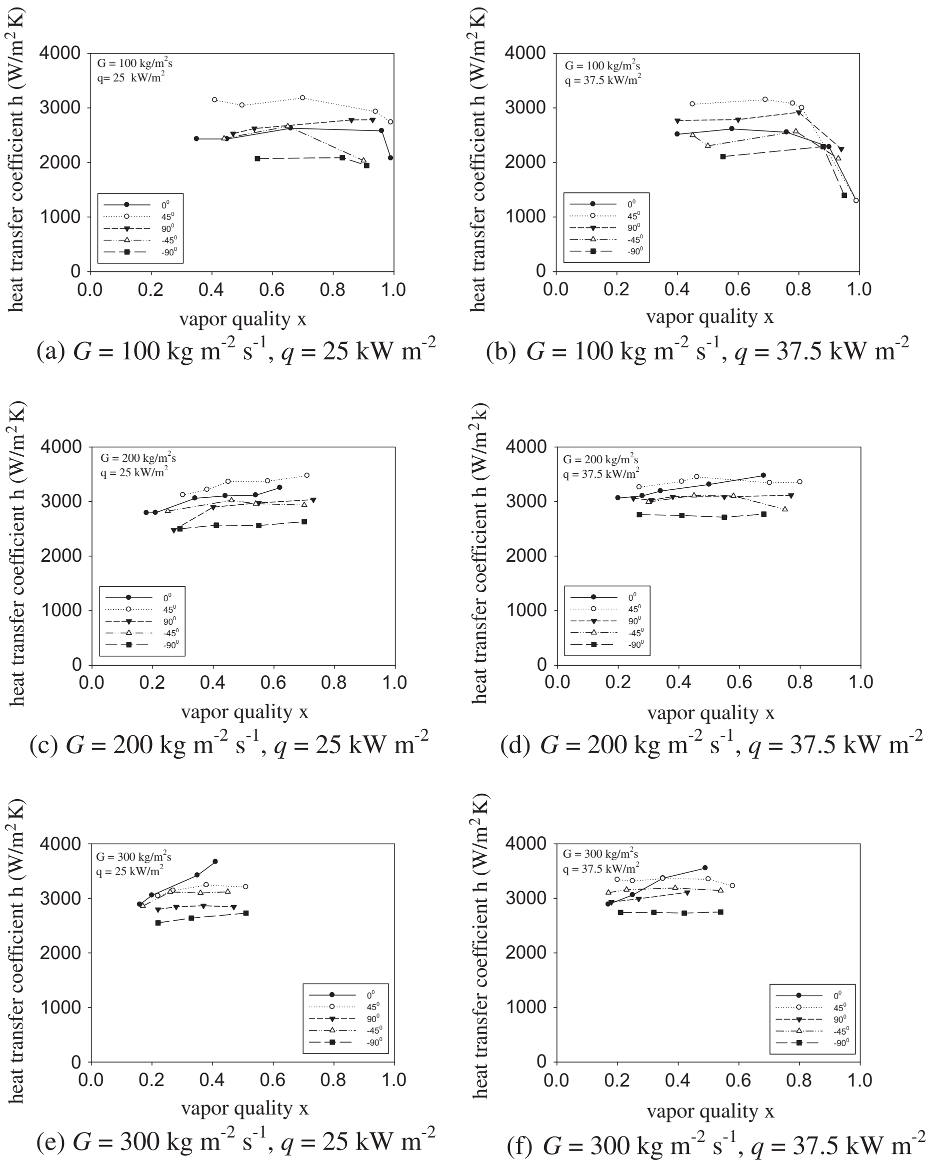
During the two-phase experiment, the inlet vapor quality is controlled by a double-pipe heat exchanger which is circulated with controlled water temperature by a thermostat. Note that the HFE-7100 is initially subcooled before entering the preheater. Hence the corresponding thermodynamic quality can be estimated from the simple energy balance from the preheater:

$$x_{in} = \frac{Q_{water} - \dot{m}c_{p,\mathrm{HFE7100}}\Delta T_{sub}}{\dot{m}i_{fg}} \tag{5}$$

Notice that ΔT_{sub} is the inlet subcooling of HFE7100, i_{fg} the latent heat of HFE-7100. The test conditions within the test sample are all at saturated state. The maximum uncertainty of the derived heat transfer coefficient calculated from the measured heat transfer rate, mean temperature in Eq. (3) is estimated to be within 3.2%.

4. Results and discussion

Typical test results of two-phase convective heat transfer coefficient subject to the influence orientation for G = 100 kg m^{-2} s^{-1}, 200 kg m^{-2} s^{-1}, and 300 kg m^{-2} s^{-1} are shown in Fig. 3. The saturation pressure is fixed at 110 kPa. As seen, for a prescribed heat flux



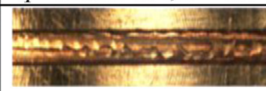
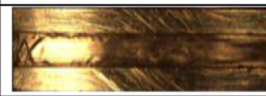
(a) $G = 100$ kg m^{-2} s^{-1}, $q = 25$ kW m^{-2}
(b) $G = 100$ kg m^{-2} s^{-1}, $q = 37.5$ kW m^{-2}
(c) $G = 200$ kg m^{-2} s^{-1}, $q = 25$ kW m^{-2}
(d) $G = 200$ kg m^{-2} s^{-1}, $q = 37.5$ kW m^{-2}
(e) $G = 300$ kg m^{-2} s^{-1}, $q = 25$ kW m^{-2}
(f) $G = 300$ kg m^{-2} s^{-1}, $q = 37.5$ kW m^{-2}

Fig. 3. Heat transfer coefficient vs. vapor quality subject to inclinations.

of 25 kW m^{-2} or 37.5 kW m^{-2}, the boiling heat transfer coefficient remains roughly unchanged with respect to vapor quality except at the very high quality region where a significant drop is encountered due to partially dry out [9]. For the influence of inclination on the heat transfer coefficient for G = 100 kg m^{-2} s^{-1}, the results reveal a very interesting behavior. Relative to that of horizontal configuration, the convective boiling heat transfer coefficient is increased by approximately 30% for an inclination angle of 45° but a further increase of inclination angle to 90° leads to a considerable drop of heat transfer coefficient. The test results show that the heat transfer coefficient amid the horizontal and vertical arrangement is about the same. By contrast, a downward inclination angle always results in a decrease of heat transfer coefficient, and it shows a substantial 50% lower heat transfer coefficient for the vertical downward arrangement relative to that of 45° upward arrangement. The results are applicable when the heat flux is raised to 37.5 kW m^{-2}. Analogous results are shown for G = 200 kg m^{-2} s^{-1} and G = 300 kg m^{-2} s^{-1} but the difference caused by inclination is less pronounced as that in G = 100 kg m^{-2} s^{-1}.

The effect of inclination on the convective heat transfer coefficients for macro channels had been reported by several researches. Hetsroni et al. [10] measured the local heat transfer coefficient using infrared thermography in a 49.2 mm diameter tube having upward inclined arrangements, and concluded that there was a tremendous rise in heat transfer coefficient subject to a slight increase of inclination angle. Local heat transfer coefficient for air–water flow in a 27.9 mm diameter pipe with horizontal and slightly upward inclinations (2°, 5°, and 7°) were reported by Ghajar and Tang [11], the heat transfer coefficients can be increased by more than 40% depending on the superficial gas and liquid Reynolds number. Cho et al. [12] examined the influence of upward flow in a 5 mm tube for evaporative heat transfer coefficient for CO_2 and CO_2/propane mixture subject to inclination angles of 45° and 90°, respectively. They found that the inclinations always augments heat transfer but the heat transfer enhancement is more pronounced for an inclination angle of 45° than for vertical arrangement. The foregoing test results were applicable for macro size diameter tube. As cited in the introduction, for microchannel configurations, Kandlikar and Balasubramanian [7] showed that the heat transfer coefficients for horizontal and upward vertical are the same but the heat transfer coefficient for vertical downward arrangement is impaired by 30–40%. The results are in line

Table 1
Representative flow pattern associated with inclination.

θ	$G = 100$ kg m^{-2} s^{-1}, $q = 25$ kW m^{-2}, $x = 0.45$	$G = 200$ kg m^{-2} s^{-1}, $q = 25$ kW m^{-2}, $x = 0.4$	$G = 300$ kg m^{-2} s^{-1}, $q = 37.5$ kW m^{-2}, $x = 0.2$
0°			
45°			
90°			
-45 °			
-90 °			

with the present results to some extent, the major difference is that a considerable rise of heat transfer performance for an inclination angle of 45°. The considerable rise of heat transfer performance for inclined arrangement is attributed to some possible mechanisms.

Firstly, normally at a moderate mass flux and vapor quality region, the major two-phase flow pattern within microchannels is elongated bubble, this is also applicable in the present study as can be seen from Table 1. As pointed out Bonnecaze et al. [13], the bubble velocity is a combination of two components. The first component is due to the buoyancy force whereas the second component is associated with non-stationary liquid slug which is traveling with at the no-slip velocity. Their theoretical calculation indicated an increase of bubble rise with respect to inclination angle, and a peak value occurred at an inclination angle amid vertical and horizontal arrangement. One of the reasons postulated is because the rise of distance between the bubble nose and contact surface brings about the increase of velocity adjacent to the contact surface. This phenomenon may become more pronounced when the tube size is reduced for surface tension imposes more influence on the smaller diameter channel. In essence, the results implicate a rise of bubble velocity. In fact, the buoyancy plays a role to strengthen the length of vapor slug and increases the bubble velocity accordingly. On the other hand, the flow inertia for upward arrangement is concurrent with the buoyancy force, thereby reinforcing the enhancement effect. The results had been confirmed with some visual observations. For example, Cheng and Lin [14] found that the thickness of liquid film around the gas slug is not uniform in an inclined tube as is the case in a horizontal tube. A schematic modified from Cheng and Lin [14] showing the asymmetric/symmetric nature of the elongated bubble subject to inclination is shown in Fig. 4. As shown in Fig. 4, the bubble is relatively symmetric in both vertical and horizontal configurations but becomes asymmetric with inclinations. When the configuration is upward inclined, the asymmetric feature helps to slide bubble and increase the bubble translation velocity accordingly. By contrast, the flow inertia is against the buoyancy direction for downward inclination. Hence, the size of the bubble is shortened and the heat transfer is impaired. The foregoing discussions are substantiated by Bonnecaze et al. analysis [13]. Bendiksen [15]

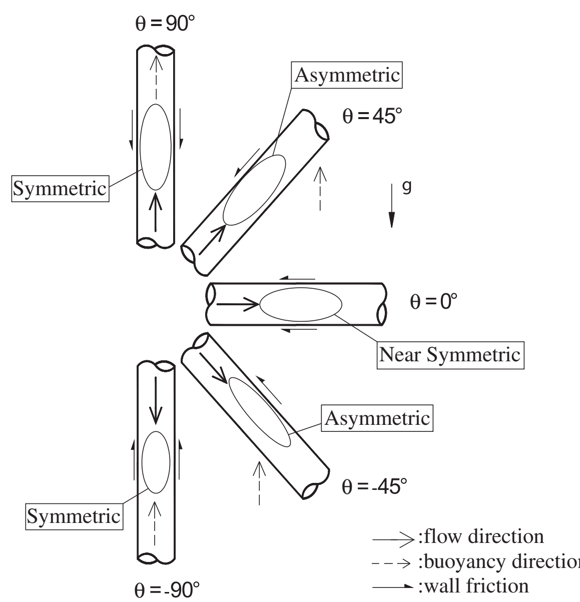


Fig. 4. Schematic of symmetric/assymetric elongated bubble and its size variation subject to inclinations.

proposed the gas slug is asymmetric (deviates from the axis of tube) and attached to the upper surface when Fr ≪ 1. When Fr ≫ 1, the gas slug is axially symmetric. In this study, it is found that the average Froude number spans between 27 and 103, suggesting some influence of inclination still exists. However, it should be mentioned that a rise of mass flux will reduce the effect of inclination. Secondly, Hetsroni et al. [10] also indicated that the turbulent dispersion plays an important role in increasing heat transfer, and they accounted for the heat transfer enhancement by inclination from two aspects. The first is due to the replacement of the liquid layer by an elongated bubble passing alongside the surfaces, and the other is subject to the additional turbulence caused by the bubble.

Table 1 shows the difference of the observed flow pattern subject to the influence of inclination for several mass flux and heat flux. It is interesting to know that there is a discriminate difference of the observed flow patterns vs. inclination. For $G = 100\ \text{kg m}^{-2}\ \text{s}^{-1}$, $x = 0.45$, and $q = 25\ \text{kW m}^{-2}$, the flow pattern for horizontal arrangement is slug flow. As the inclination angle is increased to 45° upward, the flow pattern becomes wispy-annular and it accompanies with flow some froth, implicating an increase in slug velocity. On the other hand, with a further increase of inclination, the vapor slug becomes more and more symmetric, resulting in more contact with the wall surface. Therefore the rise of viscous friction acts to reduce the rising slug despite the buoyancy force rises with the inclination. In this regard, the interactions of these two forces contribute to break loose the vapor slug, and one can see that some smaller bubbles breaks free from the tail of the large vapor slug. Conversely, this phenomenon does not occur for vertical downhill flow. This is because the flow direction is opposed to the buoyancy force. The observed flow pattern is generally in line with the findings of heat transfer measurement. With the rise of mass flux ($G = 200\ \text{kg m}^{-2}\ \text{s}^{-1}$, $x = 0.4$, and $q = 25\ \text{kW m}^{-2}$), the slug flow is first seen in the horizontal configuration, followed by a longer slug and wispy annular flow pattern for the 45° uphill inclination accompanying with visible waviness amid slugs. But the conspicuous waviness disappears when the inclination angle is raised to vertical upward configuration, indicating the slowing down of the translational velocity by the surface contact. Similarly, the waviness disappears for −45° inclination as well as vertical downward. The existence of waviness suggests a higher heat transfer performance for horizontal and 45° upward inclination than other inclinations. In fact, the heat transfer coefficient between horizontal and 45° inclination is comparatively indiscriminate due to the presence of waviness. Note that the waviness is not so evident for $G = 100\ \text{kg m}^{-2}\ \text{s}^{-1}$, thereby the 45° shows a much higher heat transfer performance than those of horizontal one. On the other hand, the influence of inclination is decreasing when the mass flux is further increased to $300\ \text{kg m}^{-2}\ \text{s}^{-1}$ as shown in Table 1. This is somehow expected due to the rise of the Froude number, leading to a more symmetric flow pattern.

5. Conclusions

This study examines the effect of inclinations on the two-phase convective boiling heat transfer characteristics of the dielectric fluid HFE-7100 within a multiport microchannel heat sink having a hydraulic diameter of 825 μm. The inclinations spans from −90° (vertical downward) to 90° (vertical upward), and a flow visualization is also conducted in this study. In general, it is found that the heat transfer coefficient amid the vertical upward and horizontal is comparable for $G = 100\ \text{kg m}^{-2}\ \text{s}^{-1}$, and the heat transfer coefficient for 45° upward considerably exceeds other configurations. For $G = 200\ \text{kg m}^{-2}\ \text{s}^{-1}$, the heat transfer coefficient for horizontal is marginally lower than that of 45° upward inclination, yet other configurations are inferior to the 45° upward inclination. The results also indicate that downward arrangements always impair the heat transfer performance, and more than 50% deterioration of the heat transfer coefficient is encountered for $G = 100\ \text{kg m}^{-2}\ \text{s}^{-1}$ relative to that of 45° arrangement.

On the other hand, the enhancement of heat transfer coefficient under inclined arrangement is reduced when the mass flux is increased to $300\ \text{kg m}^{-2}\ \text{s}^{-1}$ due to a significant rise of the Froude number where the elongated bubble/slug are more symmetric in form. The flow visualization confirms that upward inclination increase the slug velocity. In addition, it is found that the influence of inclination on the flow pattern is more pronounced at lower mass flux or low vapor quality region.

Acknowledgments

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