Heat transfer enhancement of a loop thermosyphon with a hydrophobic spot-coated surface

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Abstract
Heat transfer characteristic of a closed two-phase thermosyphon with enhanced boiling surface is studied and compared with that of a copper mirror surface. Two-phase cooling is widely used in application of thermal engineering and considerably more efficient than single-phase liquid cooling. The evaporator surfaces, coated with a pattern of hydrophobic circular spots (0.5 - 2 mm in diameter and 1.5 - 3 mm in pitch) on Cu substrates, achieve very high heat transfer coefficient and low incipience temperature overshoot with water as working fluid. Sub-atmospheric boiling on the hydrophobic spot-coated surface shows a much better heat transfer performance. Tests under heat loads 30 W to 260 W reveal the coated surfaces enhance nucleate boiling performance by increasing the bubbles nucleation-site density. The surface with hydrophobic spots with diameter 1 mm and pitch 1.5 mm achieves the maximal heat transfer enhancement with the minimum boiling thermal resistance as low as 0.03 K/W. A comparison of three evaporator surfaces with identical wettability patterns but with different surface topographies and coating thicknesses is carried out experimentally. The results show superior heat transfer rates and wear resistance on the surface coated with HNTs spots thanks to the large contact angle, great thickness, and durability of the coating layer.

Keywords: Thermosyphon, Wettability, Hydrophobic spot, Boiling heat transfer, Bubble behaviour

1. Introduction

Two-phase closed thermosyphon (TPCT) has been studied comprehensively in recent years due to its low thermal resistance, reliability, and flexibility (Maydanik et al., 2014). TPCT is considered one of the most promising cooling devices, which can be used in many applications such as desktop computers and servers (Webb, 2002), power chips (Li et al., 2011), spent fuel pool (Xiong et al., 2015), telecommunication equipment (Samba et al., 2013), smartphones (Han et al., 2013), and solar water heaters (Li et al., 2014). The two main parts of a TPCT are evaporator and condenser, which are connected by two flexible pipes, a riser for flowing vapour, and a downcomer for returning liquid. Heat is transferred by liquid vaporization from the evaporator to the condenser without any mechanical devices.

For a gravity-assisted thermosyphon, the basic mechanism of heat transfer is nucleate boiling in the evaporator and condensation in the condenser. Hence, to evaluate a thermosyphon, critical heat flux (CHF) and heat transfer coefficient (HTC) of boiling are recognized as the important performance benchmarks. Many efforts have been made to enhance CHF and HTC in boiling heat transfer research. Techniques to enhance CHF such as surface wettability engineering have been well established and studied. To characterize surface wettability, contact angle (CA) is an essential criterion: hydrophilicity (CA < 90°) and hydrophobicity (CA > 90°). Takata et al. (2003) made a superhydrophilic TiO₂-coated surface to enhance CHF by two times compared with an uncoated surface. The enhanced surface exhibited very high
wettability with the contact angle close to 0° after being exposed to UV light for several hours. Sarwar et al. (2007) performed a sub-cooled flow boiling experiment with microporous TiO$_2$ and Al$_2$O$_3$ surfaces to enhance CHF by 20% to 25% compared to a smooth one. They found that microporous Al$_2$O$_3$ surface had high wettability, which led to water moving in pores easily. Chang et al. (1997) also demonstrated CHF enhancement on DOA (Diamond particles, Omegabond 101 epoxy, and alcohol)-coated surfaces with different particle sizes. Significant CHF increase was achieved, which delayed the onset of film boiling effectively. Furthermore, nanoparticle deposition was found to be an advanced method to modify the surface wettability significantly and in turn to improve CHF. Forrest et al. (2010) developed a PAH/SiO$_2$ (poly, allylamine, and hydrochloride) nanoparticle coating, whose application on a nickel wire in a boiling experiment brought about surface wettability change that led to 100% enhancement of CHF. Nanoparticle coatings of Al$_2$O$_3$-water/ethanol was studied in pool boiling experiments by Kwark et al. (2010). They found a strong relationship between quasi-static contact angle and CHF. That is, with increasing wettability, CHF increased gradually.

Prior studies have demonstrated how to enhance the performance of HTC. The influence of surface wettability on HTC was made clear by Phan et al. (2009). From a comparison between weakly wetted surface and strongly wetted surface, they found that excellent HTC can be achieved both at very low contact angle close to 0° and at a contact angle close to 90°. They explained HTC increased for the superhydrophilic surface due to the existence of a larger and thinner liquid microlayer below the bubble across which a large amount of heat was transported. However, for the weakly wetted surface, the enhancement was due to high bubble emission frequency. HTC has been found to be significantly improved by Takata et al. (2006) using a super water-repellent (SWR) surface coated with patterns of nickel-PTFE particles. A very large contact angle over 150° led to bubble nucleation at extremely low superheats. HTC of nucleate boiling on checker/spot patterned surfaces was improved by seven times compared with that of a non-patterned surface, which confirmed the vital role of wettability on HTC.

As we know, thermosyphon is a boiling heat transfer device operating at sub-atmospheric pressures for lowering the saturation temperature when water is selected as working fluid (Chu et al., 1999; Chehade et al., 2014; Tsoi et al., 2011). The boiling performance of water at low pressures decreases significantly compared with that at the atmospheric pressure. The deterioration of boiling heat transfer at low pressures has a direct impact on thermosyphon performance. It was shown that the maximum heat removal and the total heat resistance of heat pipes increase generally with increases of the system pressure (Marcus, 1972). Research on boiling of liquids at sub-atmospheric pressures has mainly focused on the effects of reduced pressures on the bubble nucleation process, critical heat flux, incipient superheat and surface temperature. One of the early studies on boiling at sub-atmospheric pressures of 13.3 - 101.3 kPa was carried out by Van Stralen (1956). A reduction in heat transfer during boiling at sub-atmospheric pressures was found. He observed that decreases in pressure delayed the onset of nucleate boiling, leading to rising bubble sizes while reducing the maximum heat flux attained. He also experimentally investigated the growth rate of vapour bubbles in water using a nickel-plated copper surface for a pressure range of 2 - 26.7 kPa. It was considered that the bubble departure time and departure radius increased substantially with decreasing operating pressure (1975). Niro et al. (1990) showed, by combining the Clausius-Clapeyron equation with the Laplace equation, the superheat necessary for bubble nucleation at low pressures was much higher than that at high pressures. The average departure diameter increases with decreasing pressure. McGillis et al. (1990) investigated the boiling of water in a thermosyphon configuration at sub-atmospheric pressures using a plain surface. They observed that lowering the pressure generated larger nucleation bubbles, which disturbed growth of active nucleation sites, resulting in larger wall superheats.

Hence, how to alleviate the negative effects of reduced pressures on the bubble nucleation process has become an issue of great importance. Using particle-coated enhanced surface is a promising way to improve heat transfer by lowering the wall superheat and increasing HTC and CHF at sub-atmospheric pressures. Liu et al. (2011) investigated the heat transfer performance of a heat pipe with nanoparticles experimentally, under pressures of 7.5 kPa, 12.4 kPa, and 20.0 kPa. The total thermal resistance of heat pipe with particle layer coating on the surface decreased significantly compared with an uncoated surface at all operating pressures. The heat transfer enhancement were attributed to the reduction of solid-liquid contact angle and increase of thermal conductivity. It was found that both the solid-liquid contact angle and the surface tension would decrease with increasing nanoparticle mass concentration. On account of the effect of surface wettability on HTC and CHF, Betz et al. (2010) were able to perform a pool boiling experiment with surfaces combing hydrophobic (Teflon) and hydrophilic (SiO$_2$ treated with Hydrofluoric acid) zones. HTC and CHF were enhanced by 100% and 65% respectively, compared with hydrophilic surface.

Inspired by enhanced heat transfer under the effect of surface wettability, this study aims to present improvement of
the overall performance of a two-phase closed thermosyphon by using a hydrophilic/hydrophobic (mixed-wettability) evaporator surface. The combination of hydrophobic coating and the hydrophilic surface leads to a consistently large number of active nucleation sites even at sub-atmospheric pressures. New experimental results of non-electroplating Ni-PTFE (polytetrafluoroethylene) coated surfaces are presented, and the influences of the pattern geometry and heat input are discussed. Average enhancement of about 200% of HTC was achieved for patterned surface compared with that of copper mirror surface. In addition, the experimental results of three evaporator surfaces with different hydrophobic coating materials (same spot diameter 1 mm and pitch 3 mm) are carried out experimentally. Boiling and total thermal resistances, HTC, evaporator surface temperature, and bubble behaviours are investigated, compared, and analysed.

Nomenclature

\[ A \] : evaporator surface area (m\(^2\))  
\[ c_p \] : heat capacity at constant pressure (J/kg K)  
\[ d \] : diameter (mm)  
\[ FR \] : filling ratio (%)  
\[ f_c \] : bubble frequency (s\(^{-1}\))  
\[ G \] : mass flow rate of condenser (kg/s)  
\[ h \] : heat transfer coefficient (kW/m\(^2\)K)  
\[ k \] : thermal conductivity (W/m K)  
\[ p \] : pitch (mm)  
\[ P \] : pressure (Pa)  
\[ \Delta P \] : pressure difference (Pa)  
\[ q \] : heat flux (kW/m\(^2\))  
\[ Q \] : heat transfer rate (W)  
\[ R \] : thermal resistance (K/W)  
\[ T \] : temperature (°C)  
\[ t_g \] : growth time (s)  
\[ t_w \] : waiting time (s)  
\[ \Delta T \] : temperature difference (K)  
\[ x \] : distance of two thermocouples (mm)

Subscripts

\[ \text{amb} \] : ambient  
\[ \text{boil} \] : boiling  
\[ \text{cond} \] : condenser  
\[ \text{in} \] : input  
\[ \text{out} \] : outlet  
\[ \text{safe} \] : safety condition  
\[ \text{sat} \] : saturation  
\[ \text{v} \] : vapour  
\[ \text{w} \] : wall

2. Methodology
2.1 Experimental apparatus

The experimental setup is shown in Fig. 1. The setup used in the experiments consists of an evaporator and the condenser, and two connecting pipes of an internal diameter of 10mm. Both pipes are insulated in order to reduce heat loss. To activate this loop thermosyphon, the condenser is placed at 10mm higher than the evaporator, which helps the condensed liquid flow from the condenser to the evaporator continuously. The evaporator is a rectangular chamber (made of transparent polymer material) of 33 mm in height, 96 mm in length and 90 mm in width, which allows viewing of the liquid level inside the chamber and the bubble behaviours. The condenser chamber is 40 mm in height, 118 mm in length and 108 mm in width.
The coil heat exchanger, which works as a condenser, provides the condensing power. Two smooth-walled pipes (10 mm in diameter, 130 mm in length) are used to separate liquid and vapour pathways to exclude both thermal and viscous interactions between counter-currents of vapour and liquid. The press-fitted O-ring is used on the bottom of the evaporator chamber for good sealing. Heating is provided by three cartridge heaters embedded below the heating surface. The evaporator surface is shown in Fig. 2, 1.5 mm in thickness, 30 mm × 38 mm in width and length of the heating centre. A high-temperature-resistant thermally-conductive paste (16 W/m·K) is used between the top of heating block and the bottom of evaporator surface so as to reduce thermal contact resistance.

Leakage check is carried out to ensure that the setup maintains a consistent performance over a long period of time. Initially at 10 kPa, the pressure of the closed system is found to increase by only 0.2 kPa after a 24-hour period, which is considered an acceptable amount of leakage. After charging and degassing, the system valve is closed and air initially dissolved in the test liquid can be removed by vacuum degassing for 2 hours prior to the measurement. Each experiment run lasts 4 - 5 hours.

The operating principle of TPCT is as follows: the working fluid (water) is heated by the heater below the surface, and starts to boil on the evaporator surface. Then the generated vapour moves along the horizontal pipe driven by the pressure difference between the hot region and the cold region of the thermosyphon. In the condenser chamber, the vapour flowing from the evaporation section is condensed into the liquid, and the heat is dissipated into the circulating cooling water in the coil-tube. Finally, the liquid from the condenser returns to the evaporator by gravity forming a circulation system. The cycle then repeats itself.

![Diagram](image1)

**Fig. 1** Measurement scheme and thermocouples arrangement on the thermosyphon setup.

![Diagram](image2)

**Fig. 2** Non-electroplating mixed-wettability evaporator surface.
A number of process parameters are actively measured and monitored in the experiment, which are fed to a data acquisition system at a sampling rate of 3 Hz. During the experiment, the heating block assembly is insulated to assure one-dimensional heat flow to the boiling evaporator surface. All measurements have been conducted in a steady state, which is judged by monitoring the outputs of the thermocouples. The actual heat input $Q_{in}$ is extrapolated based on steady-state temperature readings (averaged 50 consecutive data points) of the three Ni-Cr thermocouples embedded in the heat transfer block. Various values of heat load ranging from 30 to 260 W are tested.

### 2.2 Data analysis

The temperature measurement of the whole system is described in Fig. 1. The heat flux is calculated from the thermocouple temperature difference in the heating block over the distance between:

$$ q = k \frac{T_1 - T_3}{x_1 - x_3} $$

Here $T_1$ and $T_3$ are the temperatures of the two thermocouples inserted in the heating block. And $x_1 = 4.28$ mm and $x_3 = 14.36$ mm are the distances to the upper end of the heating block. Additionally, $k$ is the thermal conductivity of the copper. The heat input $Q_{in}$ is the heat flux $q$ multiplied by the effective area of the heating surface:

$$ Q_{in} = qA $$

The thermal resistance is defined as the ratio of the temperature difference to the heat flow. The boiling thermal resistance is thus defined as the difference between $T_w$ and $T_{sat}$ to be divided by $Q_{in}$:

$$ R_{boil} = \frac{T_w - T_{sat}}{Q_{in}} $$

Here $T_w$ is the wall temperature at the centre of the evaporator surface measured by the thermocouple inserted in the hole inside. In consideration of the precision of the thermocouple and pressure gauge, $T_{sat}$ is the saturation temperature measured using the thermocouple immersed in the bulk liquid. The condensation thermal resistance is defined as the difference between $T_v$ (vapour temperature in the boiling chamber) and $T_{in}$ to be divided by the amount of heat removed at the condenser $Q_{cond}$:

$$ R_{cond} = \frac{T_v - T_{in}}{Q_{cond}} $$

Here $T_{in}$ is the temperature of the cooling water in the condenser. And $Q_{cond}$ is calculated by the temperature increase of the cooling water in the tube as follows:

$$ Q_{cond} = Gc_p (T_{out} - T_{in}) $$

Here $G$ is the cooling water mass flow rate, $c_p$ is water specific heat, $T_{out}$ is the temperature of the cooling water at the outlet. The total thermal resistance is the sum of the boiling thermal resistance $R_{boil}$ and the condensation thermal resistance $R_{cond}$:

$$ R_{total} = R_{boil} + R_{cond} $$

The flow rate of circulating water in the condenser part is set at 0.014 kg/s. The evaporation heat transfer coefficient, as one of the crucial performance parameters of TPCT, is defined by the following equation:

$$ h = \frac{Q_{in}}{A(T_w - T_{sat})} $$

Here $A$ is the evaporator surface area.

### 2.3 Uncertainty of the experimental data

The uncertainties of the measurements are analysed. The thermocouple uncertainty is 0.2 K. The uncertainty for the
distance measurement between \( x_1 \) and \( x_2 \) is 2\%. The thermal conductivity uncertainty is considered negligible. The extrapolation uncertainty of the wall superheat is 4\%. The uncertainty resulting from the evaporator surface area is 0.1\%.

The uncertainty of the heat flux measurement can be calculated by

\[
\frac{\Delta q}{q} = \sqrt{\left(\frac{\Delta k}{k}\right)^2 + \left(\frac{\Delta (\Delta T)}{\Delta T}\right)^2 + \left(\frac{\Delta x}{x}\right)^2}
\]

which gives 4.2\%.

For HTC and the thermal resistance, the measurement uncertainties,

\[
\frac{\Delta h}{h} = \sqrt{\left(\frac{\Delta q}{q}\right)^2 + \left(\frac{\Delta (T_w - T_{sat})}{T_w - T_{sat}}\right)^2}
\]

\[
\frac{\Delta R}{R} = \sqrt{\left(\frac{\Delta q}{q}\right)^2 + \left(\frac{\Delta (\Delta T)}{\Delta T}\right)^2 + \left(\frac{\Delta A}{A}\right)^2}
\]

are calculated to be 4.5\% and 5.8\%, respectively.

### 2.4 Surface preparation

The evaporator surfaces are coated with a pattern of hydrophobic circle spots (non-electroplating, 0.5 - 2 mm in diameter and 1.5 - 3 mm in pitch). As shown in Fig. 3, the fabrication procedure of non-electroplating mixed-wettability surface is shown as follows:

(a) Polishing the copper surface to mirror finish, to be followed by cleaning with acetone and alkaline.
(b) Patterning with photolithography and baking at 110 \(^\circ\)C for 5 min.
(c) Exposure to UV light for 2 min 40 seconds with the mask attached.
(d) Immersion in developing solution for 6 min.
(e) Base Ni-plating for 20 min at 85 \(^\circ\)C and main PTFE-plating for 40 min at 85 \(^\circ\)C.
(f) Removal of the photoresist-mask at 50 - 70 \(^\circ\)C for 20 min.

![Fig. 3 Process of Ni-PTFE coating by photolithography.](image-url)
The surfaces we used in the loop thermosyphon experiments are shown in Fig. 4. Table 1 summarizes the hydrophobic spot pattern parameters (namely, spot diameter and pitch). The objective is to study the effect of different spot diameters and pitches on the boiling performance, such as bubble departure diameter and frequency, active nucleation site density, liquid down-flow efficiency and bubble coalescence. Particularly, in the case of Type F surface, it is fully covered with hydrophobic material on the heating area. Type A surface is a copper mirror surface without any coating. Figure 5 shows a SEM micrograph of non-electroplating Ni-PTFE. Ni-PTFE layer is a compact film without any pores, adhering to the copper mirror substrate tightly. The images shows qualitatively that the coating is generally smooth and homogenous on the microscale. The contact angle of the coated surface is measured to be around 150°.

Table 1  Hydrophobic coating parameters of surfaces Type A, B, C, D, E, and F.

<table>
<thead>
<tr>
<th>Case</th>
<th>Spot diameter, (d) [mm]</th>
<th>Pitch, (p) [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type A</td>
<td>NA</td>
<td>NA</td>
</tr>
<tr>
<td>Type B</td>
<td>0.5</td>
<td>3</td>
</tr>
<tr>
<td>Type C</td>
<td>1</td>
<td>1.5</td>
</tr>
<tr>
<td>Type D</td>
<td>1</td>
<td>3</td>
</tr>
<tr>
<td>Type E</td>
<td>2</td>
<td>3</td>
</tr>
<tr>
<td>Type F</td>
<td>Fully covered</td>
<td>Fully covered</td>
</tr>
</tbody>
</table>

Fig. 4 Copper mirror surface Type A and patterned surfaces of Type B, C, D, E, F.
3. Results and discussion

3.1 Effect of hydrophobic coating pattern

3.1.1 Heat transfer rates

We have studied experimentally the heat transfer characteristics of thermosyphons whose evaporator surface is coated with non-electroplating of Ni-PTFE particles. In this study, the filling ratio ($FR$, defined as the ratio of the water volume to the total volume of the system including the boiling chamber, condenser chamber, and connecting pipes) is set at 27% and condenser temperature (inlet water temperature in coil-tube) is 45 °C consistently, which are found to be the optimal values in our recent study (He et al., 2017). In Figs. 6 to 9 are shown the comparison of the thermal resistance of the evaporator ($R_{boil}$), the total thermal resistance ($R_{total}$), the surface temperature ($T_w$), and the heat transfer coefficient ($h$), respectively. The error bars in Fig. 6 to Fig. 9 represent the results obtained from the uncertainty analysis. Figure 6 and Figure 7 show the comparison of the experimental results of surfaces Type A to Type F, with the heat input ranging from 30 to 260 W. The boiling thermal resistance of the patterned surface is reduced by more than 60% on average (78%...
at maximum) compared with that of the mirror surface. The copper mirror surface (hydrophilic) appears to result in a very high superheat for the initiation of nucleation. Upon nucleation, the wall superheat drops to a lower value (overshoot) with the increasing heat flux. For the case of surface Type B (with spots diameter 0.5 mm), it is very difficult to start nucleate boiling on such small hydrophobic spots, especially at low heat fluxes. There are still inactive spots on the surface when the heat input reaches to 130 W. As a result, the boiling thermal resistance is much larger than that of the other spot-coated surfaces at relatively low heating powers. However, at higher heat fluxes up to 150 kW/m², due to the higher saturation pressure/temperature in the evaporator, nearly all the coated spots become active sites of bubble nucleation, which leads to lower boiling and total thermal resistances. Based on the results in Fig. 6 and Fig. 7, we note that surface Type C possesses the lowest boiling and total thermal resistances, due mainly to the large density of the active bubble nucleation sites. The surface temperature is lowered by more than 15 K on average between surfaces Type A and Type C. For hydrophobic spot-coated surface, a large amount of small bubbles depart from the surface, and hence much more heat is transferred to the condenser, which leads to the surface temperature being reduced significantly. As shown in Fig. 9, at a given heat flux (< 130 kW/m²), a comparison of surfaces Type B, D, and E (with different diameters but the same pitch) shows that the HTC values follow the order of Type D > Type E > Type B, which means surfaces coated with small-diameter spots (i.e., 0.5 mm) are not available for bubble nucleation at low heat flux. For $q > 130$

![Graph](image)

**Fig. 7** Comparison of the experimental results of surfaces Type A - Type F. The total thermal resistance $R_{total}$ vs. the heat input $Q_{in}$. The filling ratio is 27%, and the condenser temperature is 45 °C.

![Graph](image)

**Fig. 8** Comparison of the experimental results of surfaces Type A - Type F. The evaporator surface temperature $T_w$ vs. the heat input $Q_{in}$. The filling ratio is 27%, and the condenser temperature is 45 °C.
kW/m², the HTC order changes to Type D > Type B > Type E. Due to the larger diameter spots size, larger bubbles are generated from surface Type E leading to neighbouring bubbles merging into an even bigger one, which affects the frequency of bubble departure and heat transfer efficiency.

Taking into account the thermal resistances as a whole, we show the results of the boiling and condenser thermal resistance in Fig. 7. Fully covered surface Type F presents the worse heat transfer performance compared with spot-coated surfaces Type B to E, in terms of the thermal resistance, surface temperature, and HTC. In fact, very few bubble emissions are observed on the fully covered hydrophobic surface at low heat fluxes. However, at higher heat fluxes, bubbles start spreading over the surface, resulting in bubble coalescence and eventually film boiling (Phan et al., 2009). Similar heat transfer deterioration on a superhydrophobic surface was also observed by Takata et al. (2006). The lowest total thermal resistance has been obtained from both surfaces Type C and Type D, which decreases by 65% at maximum compared with the uncoated surface on account of the reduced boiling thermal resistances.

From the above analysis of the results, it is concluded that the boiling thermal resistances of the patterned surfaces Type B - Type F are much lower than that of the non-coated surface Type A. The enhancement is attributable to the outstanding bubble nucleation performance of the hydrophobic coatings. With Ni-PTFE spots being water repellent, the cavities on the spots act as seeds for nucleation. For a given heat flux, the hydrophobic spot-coated surfaces show the greatest improvement in boiling HTC, up to 200% compared with the non-coated one. Surface Type C presents the lowest boiling and total thermal resistances. That is because a large amount of coated hydrophobic spots (4 times larger than that of surfaces Type B, D, and E) provide more opportunities for bubble nucleation at a given heat flux. Low pressure boiling relies on a small number of large cavities, which is unsuitable for spot diameters as low as 0.5 mm. With the increase of heat input (usually leading to higher saturation pressures), it is possible to induce nucleate boiling even on smaller cavities, such as surface Type B. It is widely acknowledged in the literature that boiling heat transfer strongly depends on the nucleation site density, and bubble departure size and frequency. Large bubbles generated from a given area reduce the heat transfer efficiency due to the reduction of growth rate and emission frequency (Giraud et al., 2014).

### 3.1.2 Boiling behaviour

The contrasting bubble behaviours on surfaces Type A - Type F are shown in Fig. 10. The heat flux ranges from 130 kW/m² to 200 kW/m². In the case of the uncoated surface (Type A), the results show that at low heat flux levels only a couple of nucleation sites are active. The departure diameter of the bubble is large, leading to large waiting time before boiling occurs. It is clear that the high wettability causes the majority of cavities to be initially flooded, which, in turn, requires higher wall superheats for bubble nucleation. We can see clearly the surface temperature is 69.6 °C for surface
Type A when the heat flux is around 120 kW/m², whereas it is reduced to about 53 °C for surfaces Type C and Type D, owing to the excellent bubble behaviours. The apparent transition to intermittent boiling at high heat fluxes is responsible for large and undesirable temperature oscillations at the heated surface. Note that boiling with the patterned surfaces, there will always be a heat transfer enhancement owing to the added cavities on the evaporator. The substantial reduction in the wall superheat for the hydrophobic spot-coated surfaces (Type B - Type F) help explain the enhancement in the nucleate boiling performance. At a given heat flux about 130 kW/m², as shown in the image for surface Type B, there are more inactive spots on the surface compared with surfaces Type C and D. The excellent bubble behaviours for surfaces Type C and D correspond to the apparently more suitable hydrophobic spot diameter of 1 mm. However, the larger density of nucleation sites on surface Type C may lead to the facilitated coalescence of neighbouring bubbles to form even bigger bubbles at high heat fluxes, which reduces liquid supply to the surface. It is considered that suppressing bubble coalescence resulted in promoting liquid down-flow to the evaporating surface and penetration to nucleation spots. On the other hand, the bubbles generated on surfaces Type B and Type D seem to remain independently from each other even at high heat fluxes. The resulting strong mixing of liquid down-flow and vapour up-flow may explain the increasingly narrowing gap between the HTC curves with rising heat flux in Fig. 9.

![Type A](image1)

Type A: $q = 119.5 \text{ kW/m}^2$, $\Delta T_{sat} = 22.0 \text{ K}$, $T_w = 69.6 \text{ °C}$

Type B: $q = 202.5 \text{ kW/m}^2$, $\Delta T_{sat} = 16.8 \text{ K}$, $T_w = 66.5 \text{ °C}$

Type C: $q = 124.3 \text{ kW/m}^2$, $\Delta T_{sat} = 5.2 \text{ K}$, $T_w = 53.0 \text{ °C}$

Type D: $q = 197.7 \text{ kW/m}^2$, $\Delta T_{sat} = 7.7 \text{ K}$, $T_w = 58.1 \text{ °C}$

Type E: $q = 125.4 \text{ kW/m}^2$, $\Delta T_{sat} = 5.7 \text{ K}$, $T_w = 53.1 \text{ °C}$

Type F: $q = 197.9 \text{ kW/m}^2$, $\Delta T_{sat} = 7.2 \text{ K}$, $T_w = 57.7 \text{ °C}$

![Type D](image2)

Type D: $q = 124.4 \text{ kW/m}^2$, $\Delta T_{sat} = 7.1 \text{ K}$, $T_w = 55.5 \text{ °C}$

Type F: $q = 194.1 \text{ kW/m}^2$, $\Delta T_{sat} = 7.6 \text{ K}$, $T_w = 57.6 \text{ °C}$

Type F: $q = 124.4 \text{ kW/m}^2$, $\Delta T_{sat} = 9.4 \text{ K}$, $T_w = 59.8 \text{ °C}$

Type F: $q = 192.1 \text{ kW/m}^2$, $\Delta T_{sat} = 13.2 \text{ K}$, $T_w = 64.5 \text{ °C}$

![Type F](image3)

Fig. 10  Bubble behaviours on surfaces Type A - Type F at heat fluxes around 130 kW/m² and 200 kW/m². The operating pressures are 10 - 14 kPa, and the filling ratio is 27%.

The images for surface Type E in Fig. 10 show that much larger bubbles are generated on the coated-spots as the superheat increases. The period of the bubble growth ($t_g$) and the time between the bubble departure and the next bubble

appearance ($t_w$) are examined. It is obvious that a long time is required for a larger bubble form. The results show increases in both $t_g$ and $t_w$ for the hydrophilic copper surface compared with that of the hydrophobic spot-coated surfaces. Consequently, the bubble emission frequency, as determined by $f_e=1/(t_g+t_w)$, will decrease with increasing departed bubble diameter (Phan et al., 2009).

For the case of surface Type F, due to the whole surface being fully covered by hydrophobic coating, the resulting low heat transfer rates bring about high superheats. The generated bubbles cannot depart from the surface easily, but form a vapour blanket instead, which reduces the heat transfer efficiency of the surface, as shown in Fig. 10 ($q \approx 200$ kW/m$^2$).

Similar results were also noted by Gaertner (1967) and Hummel (1965). A comparison of the results for surfaces Type B, Type D, and Type E further reveals that the diameter and pitch of hydrophobic spots strongly correlate with the bubble diameters departing from the surface.

3.2 Effect of hydrophobic coating material

Next we investigate the effect of the hydrophobic coating on boiling heat transfer. We prepare three different patterned surfaces surfaces (new Type X, Type Y, and old Type D), whose photographs are in Fig. 11. The related coating parameters are shown in Table 2. Surface Type X surface is made from polished copper (CA $\approx 80^\circ$) and is then coated with HNTs (Halloysite Nanotubes, Al$_2$Si$_2$O$_5$(OH)$_2$·nH$_2$O) circular spots (CA $\approx 145^\circ$) (Li et al., 2015). The HNTs coating increases boiling heat transfer significantly, as is noted in a study (Ujereh et al. (2007)) where use of carbon nanotubes coating led to considerably reduced incipience boiling superheat and enhanced HTC. The surface roughness of the HNT coating is 50 - 100 nm and the coating thickness is about 200 $\mu$m.

Surface Type Y is made from polished copper (CA $\approx 80^\circ$) and is first coated with a TiO$_2$ layer using the sputtering method. After the TiO$_2$ surface is exposed to a UV light for more than 12 hours, its contact angle for water is close to $10^\circ$, which can provide a very high critical heat flux (Takata et al., 2003). This TiO$_2$ coating has a thickness of about 1.15 $\mu$m and its thermal resistance can be neglected in the moderate heat flux region. Then, a layer of FDPA (1H,1H,2H,2H-perfluorodecylphosphonic acid) is the pattern-coated on a TiO$_2$ surface, which results in a large contact angle (more than $150^\circ$). In the case of Type D (Ni-PTFE) surface, the contact angle for smooth nickel is typically between $60^\circ$ to $80^\circ$ (R. Wang et al., 2002), whereas it is $150^\circ$ for PTFE-coated area. The roughness of PTFE spot is about 1 $\mu$m, and the thickness is 30 $\mu$m. We use the same hydrophobic pattern design (diameter 1 mm and pitch 3 mm). All the experiments are performed under the same procedure and conditions. The filling ratio is 27%, and the condenser temperature is 45 $^\circ$C.

| Table 2 | Roughness and thickness of the hydrophobic spot-coated area. |

<table>
<thead>
<tr>
<th>Case</th>
<th>Coating material</th>
<th>Contact angle, CA [$^\circ$] (hydrophobic)</th>
<th>Roughness, $R_s$ [$\mu$m]</th>
<th>Thickness, $x$ [$\mu$m]</th>
<th>Substrate material</th>
<th>Contact angle, CA [$^\circ$] (hydrophilic)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type X</td>
<td>HNTs</td>
<td>145</td>
<td>0.1</td>
<td>200</td>
<td>copper</td>
<td>80</td>
</tr>
<tr>
<td>Type Y</td>
<td>FDPA</td>
<td>150</td>
<td>0.05</td>
<td>1</td>
<td>TiO$_2$</td>
<td>10</td>
</tr>
<tr>
<td>Type D</td>
<td>Ni-PTFE</td>
<td>150</td>
<td>1</td>
<td>30</td>
<td>copper</td>
<td>80</td>
</tr>
</tbody>
</table>
Figure 12 demonstrates qualitative differences between the boiling thermal resistance results of surfaces Type X, Type Y, and Type D for heating powers ranging from 30 W to 260 W. They all decrease with the increasing heater power due to the rising saturation pressure, which results in more active nucleation sites and departing bubbles. For surfaces Type X and Type D, the onset of nucleate boiling happens when the superheats are less than 3 K, whereas it is 6 K for surface Type Y. At the early stage of the boiling process (when the heat flux is 38 kW/m²), the boiling thermal resistance of surface Type D is only 0.07 K/W, whereas they are 0.1 K/W and 0.12 K/W for surfaces Type X and Type Y, respectively. Moreover, it should be mentioned that there is no remarkable difference between surfaces Type Y and Type D when the heat flux is larger than 100 kW/m². That is because higher system pressure leads to higher saturation temperature and all the spots are activated for bubble nucleation. With further increases in heat flux, surface Type Y behaves similarly to the other two surfaces.

![Figure 12](image1)

Fig. 12 Comparison of the experimental results of surfaces Type X, Type Y, and Type D. The boiling thermal resistance $R_{\text{boil}}$ vs. the heat input $Q_{\text{in}}$. The filling ratio is 27%, and the condenser temperature is 45 °C.

![Figure 13](image2)

Fig. 13 Comparison of the experimental results of surfaces Type X, Type Y, and Type D. The heat transfer coefficient of the evaporator $h$ vs. the heat flux at the evaporator surface $q$. The filling ratio is 27%, and the condenser temperature is 45 °C.

It becomes obvious from Fig. 13 that HTC of these three surfaces rise monotonically with the increasing heat flux. The average HTC of surface Type Y is less than half that of surfaces Type X and Type D, which can be attributed to the worsening nucleate boiling performance and durability of the coated spot material. It is found that varying the coating
thickness yields significantly different HTC enhancement. As we know, initial bubble nucleation depends on absorbed gas in the cavities. Entrapped vapour lowers the required superheat for nucleation. The depth of active cavities increases as the roughness increases, which probably means a greater volume of vapour being trapped (Mpholo et al., 2010). Hydrophobic spots with a rough texture may contribute significantly to bubble nucleation because of the accumulation of entrapped vapour. In other words, the larger roughness and thickness of the coated-spot on the evaporator surface can lead to facilitated initial nucleation and lowered wall superheat at the onset of nucleate boiling (Forrest et al., 2010; Niro and Beretta, 1990; Parker and Genk, 2005).

The comparison of bubble behaviours on surfaces Type X, Type Y, and Type D at heat fluxes of 130 kW/m² and 200 kW/m² are presented in Fig. 14. The operating pressures are 10 - 14 kPa, and FR = 27%. The results for surfaces Type X and Type D show excellent bubble behaviour corresponding to the low superheats and surface temperatures. The large contact angle ensures that cavities on the surface will not be completely flooded, but covered in vapour and gas. On these cavities nucleation occur at very low wall superheat, and as a result, HTC is enhanced considerably. In the case of surface Type Y, however, there are a large amount of inactive spots even at the high heat flux of 200 kW/m², as shown in Fig. 14. Nucleate boiling performance is dependent on the hydrophobic coating property and durability. These results make it clear that micro-structured surfaces Type X and Type D with thicker coating are more suitable to bubble nucleation than the nano-structured surface Type Y. At the same heat input, the superheat is reduced by 2 K, and the surface temperature is dropped by 3 K. Surface Type D coated with Ni-PTFE and surface Type X coated with HNTs present the superior performance lies with their great coating roughness and thickness, and strong water-repellency. The stable spot-coated property and large roughness is conducive to enhancing heat transfer efficiency persistently. With the development of the experiments, the worsened coating for surface Type Y presents an obvious contrast.

The durability is always a problem for hydrophobic coating under boiling conditions. As Fig. 15 shows, there is no distinct difference between the first and second experimental runs with surface Type X. This confirms that the coating spots are not damaged. The results suggest inferior durability of surfaces Type Y and Type D. The greater coating thickness and thermostability of Type X (HNTs) appears to lead to excellent durability for the boiling heat transfer experiment. Taking into consideration the heat transfer rates and coating durability, we conclude that surface Type X (HNTs) exhibits the best applicability and stability for thermosyphon applications.
4. Conclusions

The evaporator surfaces coated with a pattern of hydrophobic circular spots (Ni-PTFE, 0.5 - 2 mm in diameter and 1.5 - 3 mm in pitch) by non-electroplating is proposed for enhanced thermosyphon design. For six different patterned surfaces, HTC of the patterned surface is enhanced by more than 200% on average compared with that of a copper mirror surface. The contrasting bubble behaviours are captured and examined in details in this study. The comparison among the three different hydrophobic coating materials are presented. The obtained results are as follows:

1. Based on the results of the boiling thermal resistance, total thermal resistance, and HTC, the hydrophobic spot-patterned surfaces are shown to be a promising way for heat transfer enhancement.

2. A large number density of spots coated on a given surface area provide more opportunities for bubble nucleation, especially at low heat fluxes.

3. Small-diameter spots (<1 mm) on the surface cannot induce bubble nucleation efficiently at low heat fluxes. For the case of homogeneously hydrophobic surface Type F, a vapour film is observed during boiling, which suppresses efficient heat transport from the evaporator surface to the liquid.

4. Hydrophobic spot-coated surfaces with micro-structures (Type X and Type D) are more suitable as boiling surface, which induces the onset of nucleate boiling at low superheats (about 2 K).

5. Compared with HNTs and FDPA coated surfaces, surface Type X (HNTs spot-coated) shows superior stability and durability, very low superheat at onset of nucleate boiling, and a very low thermal resistance.

References


Gaertner, R. F., Method and means for increasing the heat transfer coefficient between a wall and boiling liquid, U.S. Patent 3301, 314 (1967).


