Bubble Dynamics and Nucleate Pool Boiling Heat Transfer on Microporous Copper Surfaces

Suraj Joottu Thiagarajan^{1,a}, Ronggui Yang^{1,b}, Charles King^{2,c}, Sreekant Narumanchi^{2,d}

¹ Department of Mechanical Engineering, University of Colorado, Boulder, CO, 80309, USA

² National Renewable Energy Laboratory, 15013 Denver West Parkway, Golden, CO, 80401, USA

^a Suraj.Thiagarajan@Colorado.Edu Phone: 720-505-9517

^b Ronggui.Yang@Colorado.Edu Phone: 303-735-1003 (Corresponding author)

^c Charles.King@NREL.gov Phone: 303-275-4419

^d Sreekant.Narumanchi@NREL.gov Phone: 303-275-4062 (Corresponding author)

Abstract

Nucleate pool boiling experiments were performed on microporous copper surfaces and plain surfaces using saturated HFE-7100 as the working fluid. Quantitative measurements of the bubble dynamics, such as the nucleation site density, bubble diameter at departure, and bubble departure frequency, were obtained using high-speed visualization. The microporous surfaces, with coating thicknesses in the range of 100 μm to 700 μm , porosity of 55% to 60%, and cavity sizes in the range of 0.5 μm to 5 μm , showed a significantly lower boiling incipience temperature, which enhanced the heat transfer coefficient by 50% to 270% and enhanced the critical heat fluxes by 33% to 60% when compared to the plain surface. At low heat flux levels, the surface with a thicker microporous coating showed better performance than the thinner one. However, the thinner microporous coating resulted in higher critical heat flux than the thicker surface. The site density, departure diameter, and departure frequency were compared against the

predictions using various correlations from the literature. Based on a heat flux partition model, using the measured values of the active site density and bubble departure diameter and frequency, and neglecting the single-phase heat transfer effects of bubble coalescence, the individual modes of heat transfer (evaporative, quenching, and convective) were computed. Reasonably good agreement between the partition model results and the experimental data was obtained. On the plain surfaces, the evaporative and quenching components were approximately equal. On the microporous surfaces, the evaporative component was found to be significantly higher.

Keywords

power electronics cooling; pool boiling; surface enhancements; microporous surface; HFE-7100

Nomenclature

A	area of target surface
C_P	specific heat of liquid at constant pressure
D	diameter of bubbles at departure
D_{crit}	critical diameter of nucleating cavity at given wall superheat
f	frequency of bubble departure from a site
h_{lv}	latent heat of vaporization
h	heat transfer coefficient
k	thermal conductivity of copper
k_l	thermal conductivity of liquid
l_{12}	distance between the two thermocouple locations
l_{1w}	distance between the first thermocouple and target surface
l_{2w}	distance between the second thermocouple and target surface
L_c	capillary length of the liquid
L_h	target surface size
N	bubble site density
ONB	onset of nucleate boiling
P_l	liquid pressure
P_{sat}	saturation pressure
P_v	vapor pressure
q''	heat flux based on the area of target surface
q_{in}	power input to heater
r	radius of curvature of a bubble nucleus
R_a	mean surface roughness
T_{sat}	saturation temperature in the chamber

T_w	surface temperature
$\Delta T_w = T_w - T_{sat}$	superheat temperature at the surface

Greek Symbols

μ_l	liquid viscosity
ρ_l	liquid density
ρ_v	vapor density
ϕ	contact angle
σ	liquid surface tension

1 Introduction

With the rapid miniaturization and high density integration of electronic components and the consequent increase in power density, conventional air cooling methods or single-phase liquid cooling technologies for thermal management of electronic components are becoming inadequate [1]. Electric-drive vehicles of the future will have high-power electrical systems for propulsion that dissipate large amounts of heat fluxes of the order of more than 100 W/cm² at temperatures that need to be below 175°C [2]. Liquid–vapor phase-change heat transfer processes, including pool boiling, gas-assisted evaporative cooling, jet impingement, and spray cooling, which exploit the latent heat of the cooling liquids to reduce the mass flow rates, promise to meet the increasingly stringent thermal management requirements of power electronics. Due to the relative simplicity of implementation, it is desirable to obtain high heat transfer coefficients (h) and critical heat flux (CHF) with immersion pool boiling, without the use of forced convective methods [1]. Surface modification techniques have been used over the past few decades to enhance heat transfer performance by providing larger surface area, higher density of nucleation sites, and potentially smaller superheat for the phase change heat transfer [3]. Honda and Wei [4] reviewed the effect of several forms of microstructured surfaces, including laser-drilled cavities, reentrant cavities, microfins, porous coatings, and sputtered surfaces. Specifically, microporous

surfaces fabricated by electrolytic deposition, flame spraying, sintering, or brazing of particles, have been quite extensively studied in the past for enhancing pool boiling heat transfer [5–8]. The heat transfer is likely enhanced due to a combination of the following factors: (i) an increase in the effective surface area that leads to enhanced interaction of the liquid with the surface, (ii) an increase in the nucleation site density and embryonic bubbles that grow at low superheat, (iii) the presence of capillaries that facilitate liquid return to dry spots, and (iv) an increase in the solid-liquid-vapor contact line length throughout the pores. Although boiling heat transfer enhancement with microporous coatings is well documented, the physical mechanisms responsible for the enhancements are not fully understood, which hinders the design and deployment of such structured surfaces. Bergles and Chyu [9] studied pool boiling on a commercially available porous coating (Linde High Flux) that consisted of micron-sized copper particles brazed onto a cylindrical copper heated surface, with a diameter of 2.5 cm and a length of 15 cm. The porous coating with a thickness of 0.38 mm was found to increase boiling heat transfer coefficients by as much as 800% and 250% for R-113 and water, respectively. They hypothesized the enhancement is due to that nucleation occurring within cavities (inside the porous layer) that are not flooded by the liquid. The resulting vapor then exits through preferred escape channels, and the rewetting liquid is then supplied through other interconnected channels within the porous matrix. Chang and You performed pool boiling experiments [6] with FC-72 on microporous organic coatings of various thicknesses, made with thermally conductive diamond particles in an organic binder (“DOA” coating: 2–70 μm synthetic diamond particles/OMEGABOND 101 epoxy binder/acetone carrier). Despite the low thermal conductivity (~ 0.95 W/mK), these coatings led to a significantly lower nucleate boiling incipience superheat (< 10 K) as compared to ~ 30 K for the plain surface, and resulted in a

significant increase in the heat transfer coefficient and CHF. Kim et al. [10] studied the mechanism of pool boiling with FC-72 on cylindrical platinum wire with a diameter of 390 μm using high-speed photography. The 17- μm -thick coating was made from dielectric DOM (8–12 μm synthetic diamond particles/OMEGABOND 101 epoxy binder/methyl-ethyl-ketone carrier). They found that the diameter at departure of bubbles on the microporous surface was lower than the plain surface. At heat fluxes below 12 W/cm^2 , latent heat contribution was enhanced on the microporous surface, due to activation of dormant nucleation sites. At high heat fluxes, convection heat transfer became larger than latent heat, possibly due to the decrease in the superheated liquid layer at the surface, which inhibits latent heat transfer. Thiagarajan et al. [11, 12] studied pool boiling and spray impingement boiling on a vertically oriented 10-mm x 10-mm copper surface coated with a 100- μm -thick microporous layer (similar to the one used in the current work). They found that the pool boiling heat transfer coefficient was enhanced by 500% on the microporous surface compared to the plain surface [10]. With spray impingement, the heat transfer coefficient was enhanced by 300% to 600% and the CHF was enhanced by up to 80% over the plain surface [11].

In general, it is found that heat transfer occurs through the following modes during saturated nucleate boiling on a plain surface, as illustrated in Fig. 1 (a): (i) evaporation, where vapor is formed by the evaporation of the thin microlayer of liquid under a growing bubble and/or of the superheated liquid on the bubble wall [13, 14]; (ii) quenching [15, 16] where, after the departure of a bubble, heat transfer to the cooler liquid that rushes in to fill the void occurs by transient conduction; (iii) microconvection [17–19], where the agitation of the liquid by the bubble motion results in increased heat convection; and (iv) natural convection, at locations on the target surface far away from the bubble activity. In a relatively simple model, the following bubble nucleation

quantities are usually considered in developing theoretical models to analyze heat transfer characteristics: average bubble departure diameter (D), average bubble departure frequency (f), and average active nucleation site density (N). Other researchers have utilized high-speed visualization in order to characterize bubble dynamics on plain surfaces and determine the contributions of the phase-change and single-phase convective heat ([20–22], for example).

On a surface with a porous coating, heat transfer can be enhanced due to the presence of a few additional modes within the passageways [3], as illustrated in Fig. 1 (b). Vapor is generated by nucleation and bubble growth within the porous layer, evaporation of liquid thin-film on the internal surfaces, and capillary evaporation at the menisci. Single-phase convective heat transfer takes place from the particles to the liquid due to bubble pumping and vapor motion. Little data exist in the literature that detail the characteristics of bubble dynamics on microporous surfaces, and there appears to be none that relates the bubble characteristics to the heat transfer on a microporous surface except [10].

Assuming condensation of vapor does not occur anywhere from the point of generation of the vapor bubble until its departure from the exterior of the microporous layer, the total heat transfer from the target surface to the fluid by phase-change may be estimated using the rate of vapor departure from the surface. However, the single-phase heat transfer modes within the layer cannot be captured in a visualization study. The objectives of this study are therefore to investigate the saturated pool boiling heat transfer performance and bubble dynamics on the thermally conductive microporous copper coated surface with HFE-7100 and to quantify the contributions to heat transfer through different modes using high-speed visualized bubble dynamics information. In the present study, the average bubble departure diameter (D), the average bubble departure frequency (f), and the average active nucleation site density (N), are

calculated from high-speed movies of saturated pool boiling on plain copper surfaces of two different roughness levels and on three microporous copper surfaces at different heat fluxes (in the isolated bubble regime). These parameters are then used to obtain a quantitative measure of the contributions due to various modes of heat transfer on the surfaces using the partition model.

2 Experimental apparatus and procedure

2.1 Pool boiling setup

The experimental setup was designed to study pool boiling heat transfer in a horizontal configuration with plain or microporous copper surfaces. Because of the desirable properties of the fluid HFE-7100 (very low global warming potential, zero ozone depletion potential, nonflammability, high dielectric strength, and good thermophysical properties), it has been identified as a potential coolant for power electronics in electric-drive vehicles [23].

Fig. 2 shows the schematic diagram and photograph of the setup. The setup primarily consisted of an aluminum chamber to hold the test fluid (liquid-vapor) at saturation conditions, and the heater and temperature sensor assembly that contained the target surface (plain or with microporous coating) to be tested. The aluminum boiling chamber with a diameter of 12 cm and a width of 5 cm had glass windows in the front and the back to allow for the observation of boiling phenomena on the target surface. The chamber was filled with HFE-7100 liquid up to a height of about 6 cm above the target surface. Flexible silicone heaters were attached to the aluminum walls of the chamber to heat the liquid/vapor to maintain the liquid at the saturation temperature. The vapor would rise to the top region of the chamber by buoyancy, go through the reflux condenser located at the top of the chamber, and condense back to liquid pool. The pure liquid-vapor mixture in the chamber was maintained at $54 \pm 0.2^\circ\text{C}$, which is the saturation temperature corresponding to the atmospheric pressure of HFE-7100 in Golden, Colorado (80

1 kPa at an elevation of 1,730 m). Table 1 gives the thermophysical properties of HFE-7100 at
2 various temperatures [24].

3 The inset in Fig. 2 (b) shows a photograph of the heater and temperature sensor assembly.

4 The assembly consisted of a cuboidal oxygen-free copper block (dimensions 2 mm x 10 mm x 30
5 mm, thermal conductivity $k = 393$ W/m-K [25]) with a beryllium oxide ceramic, thick film
6 heater with a resistance of $20\ \Omega$ (Component General CCT-375) epoxied to it using a thin layer
7 of a relatively high thermal conductivity ($k = 1.6$ W/m-K) compound (EPO-TEK EE129-4 from
8 Epoxy Technology). Two K-type thermocouples with a junction diameter of 0.25 mm were
9 epoxied to 0.3-mm-diameter holes in the copper block, at distances of 1 mm and 4 mm from the
10 target surface using Arctic Silver thermal adhesive, such that the thermocouple junctions were
11 located at the center of the copper block at a position 1 mm from the vertical surface. This
12 copper-heater-thermocouple assembly was then placed in the cavity of a custom-fabricated
13 Teflon block ($k = 0.2$ W/m-K) with 3M Scotch-Weld DP420 epoxy ($k = 0.18$ W/m-K) filling up
14 the empty volume, so that only the 2-mm x 10-mm target surface of the copper block (along with
15 the surrounding epoxy) was exposed to the boiling chamber when assembled. The perimeter of
16 the copper surface was covered by the epoxy to prevent boiling at the edges. The heater and
17 temperature sensor assembly was placed under a circular slot in the aluminum chamber and
18 fastened tightly to the chamber to prevent any liquid leakage.

19 High-speed videos of the boiling dynamics, that is, bubble formation and departure from the
20 target surfaces, were obtained using a visualization setup that consisted of a high-speed camera
21 (Photron FASTCAM SA3 with frame rates of 1,500–3,800 fps) fitted with a long-distance
22 microscopic lens (Infinity K2S/C with CF-2 objective lens) and a cold light emitting diode
23 (LED) light source (SugarCUBE quad LED). The camera is placed at an angle of 30° relative to

the horizontal target surface. The target copper surface that was exposed to the liquid in the chamber was rectangular with a width of 2 mm and length of 10 mm. The width of the surface was kept small so that the nucleation sites on the entire surface were visible during boiling in order to accurately determine the bubble site density. The inputs from the thermocouples were fed into a data acquisition system and read using LabVIEW, while the power supply to the target heater was manually operated. The power levels were incremented by 0.1 to 0.4 W at heat flux levels below 10 W/cm^2 and by 0.5–1 W at higher heat flux levels. To verify that the pool boiling setup built for high-speed visualization on these target surfaces could be used to obtain reliable data, a few tests were also conducted using a 10 mm (width) x 10 mm (length) x 2 mm (depth) copper block in a setup similar to that used by El-Genk and Bostanci [26]. As discussed in Section 3.1.1, the pool boiling curves in this study were found to agree well with the results in [26] for the plain surface, thus validating the current setup.

2.2 Sample surfaces

Two types of surfaces were studied: the baseline case of plain copper surfaces with no coating and surfaces with microporous copper coatings. The plain surfaces were prepared by polishing the surfaces with #600 grit paper, and then with #2000 or #1500 grit paper, resulting in mean surface roughness, R_a , of $0.33 \text{ }\mu\text{m}$ and $0.78 \text{ }\mu\text{m}$, respectively, measured with a Fowler Pocket Surf roughness profilometer. The surface was deoxidized before testing by wiping with cotton swabs dipped in hydrochloric acid and cleaning with isopropanol. The scanning electron microscope (SEM) images of the plain surfaces are shown in Fig. 3 (a) and (b). The SEM images show that the cavities in the plain surfaces are of dimensions less than $1 \text{ }\mu\text{m}$ in diameter, and the surface with $R_a = 0.33 \text{ }\mu\text{m}$ is much smoother than the one with $R_a = 0.78 \text{ }\mu\text{m}$. The surfaces with the microporous copper coating were obtained from 3M Company. The procedure for fabricating

the microporous copper layer on the plain copper surface involved fusing of copper-rich microparticles of diameters between 5 μm and 20 μm on the copper surface at a temperature of $\sim 850^\circ\text{C}$ [27,28]. Three samples with coating thicknesses of 100 (± 8) μm , 360 (± 30) μm , and 700 (± 30) μm were prepared. The porosity of the microporous coating, calculated using weight and volume measurements, is found to be 55% to 60%. SEM images of the microporous coating, shown in Fig. 3 (c) and (d), show a microstructure with irregular, elongated pore structures, with cavity sizes in the range of 0.5 μm to 5 μm , which could potentially serve as nucleation sites. The presence of the microscale channels also results in a much larger surface area (compared to the nominal external surface area) that is available for thin film evaporation of the liquid, which could potentially augment the heat transfer. Furthermore, the network of interconnected channels that facilitate liquid replenishment could potentially delay dry-out on the surface to higher heat flux values as compared to the plain surface.

2.3 Experimental procedure and data reduction

Before each experiment, the HFE-7100 liquid in the pool boiling chamber was degassed for 60 to 80 minutes to expel any dissolved non-condensable gases. This was accomplished by boiling the liquid vigorously at 54°C using the guarded silicone heaters around the chamber, and allowing the vapor mixed with the non-condensable gases to flow through a condenser located above the chamber. The portion of the condensed vapor dripped back into the chamber, while the non-condensable gases escaped. After this procedure, the system was allowed to reach a steady state, with no power input to the target copper surface, so that saturation conditions existed within the chamber. Then the power to the target surface was incrementally increased and measurements were taken after the temperatures reached a steady state (that is, when the temperature variation was $< 0.05^\circ\text{C}$ over a period of 120 seconds). The thermocouple readings

were averaged over a period of 30 seconds. Assuming one-dimensional heat conduction in the copper block, the heat flux through the surface (q'') was determined by the temperature difference across these two thermocouples using Eq. (1),

$$q'' = k \frac{(T_1 - T_2)}{l_{12}}, \quad (1)$$

where k is the thermal conductivity of oxygen-free copper, T_1 and T_2 are the temperatures farther away and closer to the target surface, respectively, and l_{12} is the distance between the thermocouple locations. The calculated heat flux was then used to determine the wall temperature of the target surface T_w by assuming one-dimensional heat flow using Eq. (2),

$$T_w = T_1 - \frac{l_{1w} q''}{k}, \quad (2)$$

where l_{1w} is the distance between the thermocouple location and the surface. The heat transfer coefficient, h , was calculated using Eq. (3):

$$h = \frac{q''}{(T_w - T_{sat})}. \quad (3)$$

CHF was defined as the highest heat flux at which stable target surface temperature was maintained, before a sharp increase in temperature occurred ($>10^\circ\text{C}$ per second) with a slight increase in the power input.

2.4 Experimental uncertainties

The uncertainty of the measured heat flux was mainly from the accuracy of the position and calibration of the thermocouples. A National Institute of Standards and Technology (NIST)-traceable resistance temperature detector (RTD) was used along with a Hart Scientific micro calibration thermal bath model 7102 to calibrate all the thermocouples in the temperature range from 20°C to 105°C . The uncertainty in the temperature measurement was within $\pm 0.05^\circ\text{C}$. The uncertainty in the position of the centers of the thermocouples was estimated to be ± 0.05 mm for

1 a 0.25-mm thermocouple bead placed in a 0.30-mm-diameter hole. The uncertainties in voltage
2 and current were negligible ($<\pm 0.12\%$ and $<\pm 0.05\%$, respectively). The uncertainty in the heat
3 flux measurement was estimated using the Kline and McClintock method [29] to be $\pm 10\%$ at
4 heat fluxes below 5 W/cm^2 , $\pm 6\%$ at heat fluxes above 8 W/cm^2 , and in the intermediate range,
5 between $\pm 6\%$ and 10% .

6 Finite element analysis of heat conduction was conducted on the test article to quantify
7 thermal losses and to ensure that the heat flux could be reliably calculated using the two
8 thermocouples, as shown in Fig. 4. A three-dimensional computer-aided design model of the test
9 article was created and imported into ANSYS Workbench for a steady-state heat transfer
10 simulation. The analysis was performed for multiple heat input levels, assuming a uniform heat
11 transfer coefficient on the target surface, and a set of heat transfer coefficients for the different
12 modes of heat loss. The heat loss modes included natural convection to the air surrounding the
13 Teflon block, convective heat transfer to the liquid from the epoxy, and Teflon surrounding
14 heated copper block.

15 Fig. 4 shows the results of the simulation for two cases of input heat of 1.74 W and 5.83
16 W , with experimentally calculated pool boiling heat transfer coefficients, as shown in Eq. (3), of
17 $4,200 \text{ W/m}^2\text{-K}$ and $13,200 \text{ W/m}^2\text{-K}$, respectively. The boundary conditions are shown in Fig. 4 (a).
18 In the model, the input values were the power input ($q_{in} = V \cdot I$, where V is the voltage and I is the
19 current through the cartridge heaters), the pool boiling heat transfer coefficient, and the loss heat
20 transfer coefficients. The set of coefficients for heat loss was obtained by iteratively trying
21 different values until the temperatures obtained in the ANSYS model (and heat flux from the
22 surface) matched the corresponding experimental values. The heat transfer through the different
23 surfaces (including the copper target surface) and the temperatures at all points were the outputs.

Fig. 4 (b) shows the temperature of the copper block along the direction of heat flow. Fig. 4 (c) and (d) show the temperature contours in the planes where the thermocouples are placed in the copper block (T_1 and T_2 , respectively). In both cases, the temperature in each plane is uniform within 0.05°C in both planes, and the average calculated temperature is within $\pm 0.05^\circ\text{C}$ of the measured values. This confirmed that the pool boiling heat flux and surface temperature could be calculated using the temperatures measured by the two thermocouples.

3 Results and discussion

3.1 Experimental boiling curves

3.1.1 Measurements on 10-mm x 10-mm surfaces

To verify that the pool boiling setup with visualization capability designed in this work could be used to obtain reliable data, a few tests were conducted using a 10-mm (width) x 10-mm (length) x 2-mm (depth) copper block in a setup similar to that used by El-Genk and Bostanci [26]. Saturated pool boiling heat transfer data of HFE-7100 by El-Genk and Bostanci [26] on a 10-mm x 10-mm plain copper surface that was sanded with a #1500 grit paper is shown in Fig. 5. In the comparison tests, only one thermocouple was placed at 1 mm below the surface to estimate the surface temperature. The heat flux through the surface was assumed to be equal to the $q_{in} (= V \cdot I)$. As the design of this heater assembly is similar to [26] and [30], where simulations showed that the heat loss was below 2%, the losses in the current work are also assumed to be small, and therefore neglected.

The surface was polished using #1500 and #2000 grit paper resulting in roughness of $0.78\ \mu\text{m}$ and $0.33\ \mu\text{m}$, respectively. As expected, Fig. 5 shows that the onset of nucleate boiling (ONB) occurs at a higher superheat on the smoother sample ($R_a = 0.33\ \mu\text{m}$) than that on the rougher sample ($R_a = 0.78\ \mu\text{m}$). A lower heat transfer coefficient is observed on the smoother

surface, as evidenced from the higher surface superheat temperature at the same heat flux. The variation of the CHF due to the level of surface roughness was found to be within experimental uncertainties [31]. A repeated measurement on the 0.78- μm surface after a period of 44 hours of idle waiting (with the surface in contact with the liquid maintained at 20°C after one cycle of measurement), showed that the ONB increased to 17°C as compared to 11°C in the first measurement on the surface. This indicates that the highly wetting nature of HFE-7100 results in flooding and deactivation of larger cavities, and thus the size of the nucleating vapor embryo changes over time despite the cavities' geometry remaining intact [31]. However, the changes in the heat transfer coefficient and the CHF are found to be negligible.

Based on the hydrodynamic instability model, the CHF on an infinitely large plain surface [32] is computed using the Lienhard-Dhir correlation, which gives a value of 16.7 W/cm² for HFE-7100. In our measurements, the 10-mm x 10-mm samples gave a slightly higher value of $\sim 18.5 \pm 1.1$ W/cm². Interestingly, the pool boiling curve corresponding to the surface polished with #2000 grit paper agreed closely with the data corresponding to the surface polished using #1500 grit paper in [26]. However, the CHF that we obtained was significantly lower than ~ 25 W/cm² as reported in [26]. Overall, we consider our measurement on the 10-mm x 10-mm surface to be in good agreement with the literature. In Fig. 5, the pool boiling curve on a 2-mm x 10-mm target surface is also shown, which is also found to be in the same range as the larger surface in the nucleate boiling regime, while the CHF is enhanced as expected for the smaller target surface, consistent with [33].

3.1.2 Measurements on 2-mm x 10-mm surfaces

Visualization of the bubble dynamics on the target surface requires unobstructed view of all the bubble sites to accurately measure the site density, diameter, and frequency of departure.

As the depth-of-focus of the lens is limited to about 2 mm, the width of the target surface was kept at 2 mm. However, if the target surface size is too small, boiling on the surface may become dominated by the surface tension effects rather than buoyancy, making the data unrepresentative of boiling on larger surfaces. By studying pool boiling data on target surfaces of different sizes under various acceleration-due-to-gravity levels, Raj and Kim [34] determined that when the size of the target surface $L_h > 2.1L_c$, where $L_c = [\sigma/(g(\rho_l - \rho_v))]^{1/2}$ is the capillary length, the boiling process is dominated by buoyancy rather than surface tension. At $T_{sat} = 54^\circ\text{C}$, the capillary length of HFE-7100 is 0.85 mm. The boiling characteristics on the 2-mm x 10-mm target surface is considered to be representative of larger boiling surfaces, because it satisfies the criterion $L_h = 2 \text{ mm} > 2.1L_c$.

Fig. 6 (a) shows the boiling curves obtained for the five surfaces with the surface area 2 mm x 10 mm exposed to the liquid: two plain surfaces with different roughness ($R_a = 0.33 \text{ }\mu\text{m}$ and $R_a = 0.78 \text{ }\mu\text{m}$) and three microporous copper surfaces with different thickness of coatings (with 100- μm , 360- μm , and 700- μm -thick layers). As expected, the ONB, h , and CHF all differ with each other depending on the surface condition of these samples. As the heater power level is raised from zero, and the surface temperature is increased from the saturation temperature (corresponding to $\Delta T_w = T_w - T_{sat} = 0$), nucleate boiling on the surface does not start until the “onset” happens. The ONB results in a sudden reduction in the surface temperature, a phenomenon called “incipience temperature drop,” which could result in damage to the device due to thermal shock [1]. This is particularly pronounced in highly wetting dielectric fluids such as HFE-7100, as they tend to flood even sub-micron cavities and deactivate them. Also, a large incipience superheat temperature can result in the temperature of the device exceeding the safe operating limit. Therefore, the incipience overshoot needs to be eliminated or minimized to

increase device life and reliability. In the Fig. 6 (a), the ONB for the two plain surfaces is as high as 17°C– 38°C. The difference in the ONB is due to the difference in the surface roughness (see Fig. 4) and the resultant variation of the size of the cavities that allow for initiation of bubble nucleation. In comparison, the ONB superheat on the microporous surfaces occurs at temperatures below 11°C, which would be much more favorable for use in electronics cooling. Repeated measurements showed that the ONB superheat varies by as much as $\pm 7^\circ\text{C}$ on the plain surfaces. The variation between the first and subsequent measurements on a given surface is likely due to the random manner in which cavities of various sizes are activated resulting in nucleation, and/or due to the flooding of some cavities in the reworked surface which results in no embryo for nucleation [35]. The ONB superheat is much more repeatable on the microporous surface, with only $\pm 3^\circ\text{C}$ variation.

The Laplace-Young equation describes the pressure difference across the surface of a vapor bubble, $P_v - P_l = 2\sigma/r$, where P_v is the vapor pressure, P_l is the liquid pressure, r is radius of curvature of the bubble and σ is the surface tension. Along with the Clausius-Clapeyron relation, the Young-Laplace equation is used to estimate the critical diameter at bubble nucleation from the ONB [35], resulting in the equation:

$$D_{crit,CC} = \frac{4\sigma T_{sat}}{\rho_v h_{lv} \Delta T_{w,ONB}}, \quad (4)$$

where σ is the liquid surface tension, T_{sat} is the saturation temperature, ρ_v is the vapor density, h_{lv} is the latent heat of vaporization, and $\Delta T_{w,ONB}$ is the wall superheat at the ONB. The critical diameter is an estimation of the diameter of the cavity where a vapor nucleus can grow into a bubble without condensing, at a given wall superheat. The prediction given by Eq. (4) is appropriate when the contact angle is large ($\phi \sim 90^\circ$), the incipience superheat is small, and the saturation curve is nearly linear over the range from the saturation temperature of the liquid to

the temperature of the superheated vapor within the bubble [36]. However, for a highly wetting liquid, the incipience superheat at ONB can be large, and thus the relation between T_{sat} and P_{sat} would be nonlinear, thereby invalidating the use of the Clausius-Clapeyron relation, and hence Eq. (4). An equivalent expression that takes into account the nonlinearity is given by the following equation [36]:

$$\Delta T_{w,ONB} = T_{w,ONB} - T_{sat}(\text{at } P_l) \cong T_{sat} \left(P_l + \frac{4\sigma(\text{at } T_{w,ONB})}{D_{crit}} \right) - T_{sat}(\text{at } P_l). \quad (5)$$

The relation between T_{sat} and P_{sat} for HFE-7100 [24] is given by:

$$\ln(P[\text{pascal}]) = \frac{-3641.9}{T[\text{kelvin}]} + 22.415. \quad (6)$$

The critical diameters of the nucleating cavity computed using the minimum and maximum values of $\Delta T_{w,ONB}$ from Eqs. (5) and (6) for the various surfaces are shown in Table 2. The data show that the microporous surface contains cavities $>1 \mu\text{m}$ that can get activated at low superheats, resulting in onset of nucleation much before the plain surface. Fig. 7 shows the critical diameters for the various surfaces plotted as a function of the surface superheat, overlaid on a plot of the size range of active nucleation sites calculated using Hsu's criterion [37] for a variety of contact angles. The fact that many of the measured data points lie outside the curves corresponding to $\phi \leq 7.5^\circ$ suggests that the relevant contact angle of liquid-vapor interface is higher than the static contact angle ($<1^\circ$).

Fig. 6 (a) also shows that the CHF on the microporous surface is enhanced by about 33%–60% as compared to that on the plain surfaces. Though the difference is small, the CHF decreases monotonically with an increase in the microporous layer thickness. This is likely due to the choking of the liquid pathways occurring at a relatively lower heat flux level in the thicker layer.

Fig. 6 (b) shows the heat transfer coefficient as a function for the heat flux. The h is enhanced by 50% to 270% on the microporous surfaces over the plain surfaces. The h on the 700- μm microporous sample is the highest of all in the isolated bubbles and fully developed nucleate boiling regimes. The h in this regime increases with the increasing thickness of the layer. This is probably due to the presence of a larger number of nucleating sites within the thicker layer, resulting in more intensive phase-change heat transfer. At wall superheat $>15^{\circ}\text{C}$ (corresponding to heat flux >20 to 25 W/cm^2) the slopes of the boiling curves, as shown in Fig. 6 (a), of the microporous surfaces start to decrease, resulting in the reduction in the h after reaching the peak value. At high heat fluxes, the increased rate of vapor generation and motion from within the layer to the outside leads to the decrease in the pathways available for liquid reflow, which leads to a decrease in the heat transfer coefficient.

3.2 Bubble dynamics

High-speed visualization offers the possibility of obtaining detailed and temporally resolved information on the dynamics of bubbles growth and departure from the target surface at a high magnification level (25x–30x). In the following sections, we will discuss the bubble dynamics on various surfaces as measured using high-speed video. The Photron FASTCAM Viewer program was used for the measurement of the various parameters. Visualization measurements were used to obtain the bubble characteristics up to a heat flux (and surface superheat) value that allowed for discerning individual bubbles. The crowding of the bubbles on the surface and their coalescence make measurements beyond this heat flux ($\sim 10\text{ W/cm}^2$) difficult and not reliable. Fig. 8 shows representative images of bubble dynamics on plain and microporous surfaces at various heat flux levels.

3.2.1 Bubble site density

Each video file was analyzed to determine the locations of the active nucleation sites on the test surface. The instantaneous number of active sites captured by a still frame was generally not as high as the total, cumulative number of sites that were active at least once over several departure periods [38]. The cumulative number of sites was determined by counting all the sites that were active over any part of an arbitrarily selected 100 frames that represented several bubble cycles. The area density of active nucleation sites, N , was then calculated by dividing the counted number of active sites by the area of the surface. The uncertainty in the measurement is estimated to be 10% for low heat fluxes and up to 20% for high heat fluxes where the field of view was frequently obstructed, by repeatedly counting the number of active sites on successive sets of 100 frames each. The number of active sites increased with increasing wall superheat and with increasing surface roughness for the plain surface. Incipience of nucleate boiling on the microporous surface occurred at a much lower temperature; and consequently, the bubble site density in the microporous surface was many times higher than that of the plain surface at any given superheat. Fig. 9 and Fig. 10 show the nucleation site density as a function of the wall superheat on the plain and microporous surfaces, respectively. Since nucleation in the microporous surface takes place within the layer, the visual image only gives us the number of sites of bubble release from the surface. Hence, site density is referred to as “bubble site density” to be more general. As expected, the plain surface with the higher roughness level produced more bubbles than the smoother plain surface. In regard to the microporous surfaces, the bubble site density was observed to increase strongly with the increasing temperature. However, no clear pattern in terms of the site density versus the layer thickness could be discerned.

The bubble site density on the plain surfaces were compared with models developed by Hibiki-Ishii (H-I) [39], Kocamustafaogullari-Ishii (K-I) [40], and Benjamin-Balakrishnan (B-B)

[41] models. The H-I correlation was developed by fitting to a wide array of experimental nucleation site density data for boiling on plain surfaces. They assumed that nucleating cavity number and size distributions would be statistically similar for most boiling systems, accounting for surface characteristics by including a static contact angle term. However, given that the static contact angle of HFE-7100 is $<1^\circ$, the predicted values of the site density (N_{HI}) were several orders of magnitude smaller than the experimentally observed values (N). Very good agreement was obtained if ϕ was assumed to be 41° to 48° for the plain surfaces, with only the contact angle being the varying parameter. It is reasonable to assume that during a nucleate boiling process, the dynamic contact angle (which tends to be much larger than the static contact angle and dependent on the velocity of the liquid-vapor interface) is the relevant parameter [42], instead of the static contact angle. This is reinforced by the observation that active nucleation cavity sizes lie outside of the size range given by Hsu's criterion with a contact angle of $<7.5^\circ$, as discussed in section 3.1.2. The H-I correlation is given by the following set of equations:

$$N_{HI} = N_n \left\{ 1 - \exp \left(-\frac{\phi^2}{8\mu^2} \right) \right\} \left[\exp \left\{ f(\rho^+) \frac{\lambda'}{R_c} \right\} - 1 \right]$$

$$R_c = \frac{2\sigma\{1 + (\rho_v/\rho_l)\}/P_l}{\exp\{h_{lv}(T_w - T_{sat})/(RT_w T_{sat})\} - 1} \quad (7)$$

$$f(\rho^+) = -0.01064 + 0.48246\rho^+ - 0.22712\rho^{+2} + 0.05468\rho^{+3}$$

$$\rho^+ = \log_{10} \left(\frac{\rho_l - \rho_v}{\rho_v} \right),$$

where $N_n = 4.72 \times 10^5$ sites/m², $\mu = 0.722$ rad, and $\lambda' = 2.5 \times 10^{-6}$ m. R is the gas constant based on the molecular weight. For HFE-7100, $R = 33.24$ J/(kg-K).

For the three microporous surfaces, the contact angle is kept constant at $\phi = 46^\circ$ and only N_n is varied to obtain good fits to the experimental data. The N_n in the H-I correlation is a coefficient that is related to the average cavity density on (smooth) surfaces. Therefore, it is reasonable to

assume that this could be the parameter that varies in the microporous surfaces depending on the thickness.

The K-I model assumes that the nucleation site density and the critical cavity radius can be normalized by the length scale of the bubble departure diameter. Using the departure diameter experimentally obtained (rather than use the Fritz correlation as recommended), reasonably good agreement was obtained for the plain surfaces (Fig. 9), while it severely underpredicted the N on the microporous surfaces. The B-B model only takes into account the average surface roughness and also was found to underpredict the observed N by more than one order of magnitude. Fig. 11 shows the experimentally measured N along with the predictions based on the H-I correlation for all the surfaces. The bubble site density is assumed to increase with the superheat temperature up to a value given by the condition of hexagonal close-packed bubble density on the surface, $N_{max} = 2/(\sqrt{3}D^2)$. After that superheat value, the bubble site density is assumed to be constant.

3.2.2 Bubble departure diameter

As a bubble grows in size, buoyancy eventually becomes larger than the force that pins the bubble to the surface. Using the videos, the bubble departure diameter was calculated by averaging three diameter measurements immediately after the departure over a period of about 1.5 to 3 milliseconds. Based on repeated measurement of the bubbles, it was determined that the diameter measurements obtained from the images have an uncertainty of $\pm 15\%$. The uncertainty is due to error in locating the bubble edges, the time-dependent non-spherical shape of the bubbles, and the fact that the bubbles emanating from the same site over multiple cycles also differ in size. The departure diameter also varies with the site. In all the experiments, the bubble diameters varied between 0.2 to 0.8 mm, with a majority around 0.45 to 0.6 mm. A representative distribution of the diameters on the plain and microporous surfaces is shown in Fig. 12 (a). The

average bubble departure diameters for the various surfaces are shown in Fig. 12 (b). The average diameter seems to be only weakly dependent on the superheat temperature on all the surfaces, and decreases slightly with increasing temperature. The average diameter at departure is about 0.5 mm on the plain surfaces. The bubble diameter at departure on the microporous surface is a little higher (~0.55 to 0.6 mm) than on the plain surfaces. This is probably due to the fact that the bubble growth in the case of the microporous surface happens due to vapor generation within the microporous layer, in addition to the growth that takes place due to evaporation after the bubble comes outside of the microporous layer. Since the cavity sizes are larger on the microporous layer relative to the plain surface, the surface tension force that pins the bubble to the surface will also be higher, thereby allowing the bubble to grow larger than that on the plain surface before departure.

Several correlations have been proposed in the literature to predict the bubble departure diameter in pool boiling. The experimentally measured diameter are compared to correlations proposed by Cole [43], Kutateladze and Gogonin [44], Jensen and Memmel [45], and Phan et al. [46] in Fig. 12. Of these, the correlation by Phan et al., shown in Eq. (8), predicted the trend and values of the experimental data most closely (with the contact angle $\phi = 41^\circ$ to 48° , same as the value used in the H-I correlation for nucleate site density). This is an empirical correlation, developed with pool boiling of water on nanocoated surface resulting in the contact angle in the range $20^\circ < \phi < 110^\circ$:

$$D = 0.63 \frac{(2 + 3 \cos \phi - \cos^3 \phi)}{4} L_c, \quad (8)$$

where L_c is the capillary length and ϕ is the contact angle. At heat flux levels $q'' > 8$ to 10 W/cm^2 , lateral coalescence (between bubbles adjacent to each other on the surface) and vertical

coalescence (between subsequent bubbles at the same site) became significant enough to render the measurement impossible.

3.2.3 Bubble departure frequency

The number of bubble cycles per unit time at a given cavity location is defined as the bubble departure frequency. In the high-speed video files, bubbles growing at a location, within $\sim \pm 0.2$ mm, were all considered to be formed at the same nucleating cavity site. The bubble departure frequency was estimated by observing the number of bubbles that departed from a given nucleation site over a period of time (typically between 180 and 210 frames, corresponding to a time period of about 10 milliseconds). An average of the frequency of departure at a few sites (three to five) that are approximately of the same diameter at departure was taken to be the average departure frequency. The uncertainty in departure frequency is estimated to be $\pm 13\%$, based on the measurements from several sets of the bubble sites. The average value of the measured bubble departure frequencies are plotted with respect to the surface superheat temperature in Fig. 13, along with linear fits for each surface. On all the surfaces, the bubble departure frequency increases steeply with the superheat temperature. At the heat flux levels > 8 to 10 W/cm^2 , increased lateral bubble mergers made frequency measurements meaningless. In a bubble growth cycle, with t_g being the bubble growth period, and t_w the waiting period (between bubble departure and inception of a new bubble) at a given site, the frequency is essentially $f = 1/(t_g + t_w)$. In the experiments, at all heat flux levels, it was found that the bubble cycles were nearly continuous (that is, $t_w \ll t_g$). This is different from the $t_w > t_g$ observed during pool boiling of water [21]. This implies that transient conduction of heat from the surface to the liquid in the waiting period may not be a significant factor in our study, unlike the observation with pool boiling of water in [21], where the transient conduction was found to be the dominant mode

of heat transfer. Moghaddam and Kiger also observed in single-bubble pool boiling with FC-72 that the waiting period was often small [47].

For the microporous surface, once again the frequency of bubble departure does not seem to be clearly dependent on the thickness of the layer, though the thickness appears to positively affect the frequency. This again is probably due to the fact that the thicker layer allows higher rate of vapor generation.

Peebles and Garber [48] developed the following correlation for the relation between the frequency and bubble departure diameter, for the case where the bubble waiting time is negligible:

$$Df = 1.18 \left(\frac{t_g}{t_g + t_w} \right) \left[\frac{\sigma g (\rho_l - \rho_v)}{\rho_l^2} \right]^{0.25}, \quad (9)$$

where σ is the surface tension of the liquid, g is the acceleration due to gravity, ρ_l is the density of the liquid, and ρ_v is the density of the vapor. Because $t_w \ll t_g$, the term in the parentheses ~ 1 .

For the plain surface with the $D \sim 0.5$ mm, this gives $f \sim 214/\text{s}$, which is in the observed frequency range in the present study. However, because the diameter at departure is relatively constant with temperature, this equation implies that the frequency also should be constant, which is contrary to the measured data. The other departure frequency correlations tested [49] also are not applicable to the pool boiling of HFE-7100, in magnitude and trends [50].

3.3 Partition of heat flux

The heat flux partition model has been commonly used for mechanistically predicting the nucleate boiling heat transfer. In the model proposed by Bowring [51], the total heat transfer in a boiling process is partitioned into single-phase heat and phase-change heat. The former refers to the amount of heat that is transferred to the cold fluid to raise its temperature above the

saturation temperature, and the latter refers to the heat going into changing the phase of the liquid into vapor. The single-phase heat flux can be further subdivided into natural convection at the locations on the target surface that are not nucleating bubbles, and are far enough away to be unaffected by the churning motion of the liquid induced by the bubble motion.

The phase-change heat transfer is calculated from the total volume of vapor generated based on bubbles from all the nucleation sites. The volume of vapor generated is estimated from the vapor bubbles in the liquid just after they leave the surface. Originally developed for flow boiling, it has been applied to pool boiling also. Gerardi et al. [21] applied the partition model to pool boiling of deionized water and nanofluids and found that partition model was capable of predicting the correct order of magnitude of the heat flux, though the difference was typically large.

At the lower heat fluxes in the isolated-bubble regime, the following equations are used to obtain the following heat flux partitions (refer to Fig. 1 (a)).

1. The phase-change heat flux (q''_{evap}) is calculated from the total volume of vapor generated based on bubbles from all the nucleation sites:

$$q''_{evap} = \frac{\pi}{6A} \rho_v h_{lv} N D^3 f. \quad (10)$$

2. The single-phase portion of the heat transferred to the liquid is considered to be due to:
(1) natural convective heat transfer, $q_{NatConv}$, at locations undisturbed by the bubble activity, calculated using the McAdams correlation [52], and (2) the “quenching heat flux,” which refers to the amount of heat transferred to the cooler fluid that rushes to the surface after a bubble departs, which is the heat that is expended in the formation of a new thermal boundary layer after the departure of the bubble from the previous cycle [53]. The quenching heat flux (q''_{quench}) is a transient heat conduction process and is

modeled by the heat conducted to a semi-infinite layer of fluid during the waiting time between bubble cycles:

$$q''_{quench} = \frac{2\pi k_l \Delta T_w}{\sqrt{\pi \alpha_l}} N D^2 f \sqrt{t_w}, \quad (11)$$

where k_l is the thermal conductivity of the liquid, α_l is the thermal diffusivity of the liquid, and t_w is the waiting time period. From space- and time-resolved studies on pool boiling of a single bubble, Demiray and Kim [54] and Moghaddam and Kiger [55] observed that the transient heat conduction to the liquid happens not only during the “quenching process” during the waiting period, but also throughout the bubble growth and departure duration. Assuming gradual rewetting of the target surface by the liquid front after departure of a bubble, and for the limiting condition cases with very small waiting time, they derived the following equation for the transient conduction heat flux:

$$q''_{quench} = 2.15 \frac{k_l (0.65 \Delta T_w)}{\sqrt{\pi \alpha_l}} N D^2 \sqrt{f}. \quad (12)$$

As mentioned previously, from the visualization experiments, we found that the waiting time was much smaller than the bubble growth time. Thus Eq. (12) provides a physical picture that is closer to the observation than the quenching model, Eq. (11). The total heat flux was then computed as the sum of the three components—phase change heat flux, transient conduction heat flux, and natural convection: $q_{total\ proj} = q_{evap} + q_{quench} + q_{NC}$. The site density, departure diameter and frequency measurements made in the isolated-bubbles regime were extrapolated to higher superheat temperatures up to the CHF using the correlations discussed previously. In addition, the site density was assumed to reach a maximum (N_{max}) corresponding to the fully-packed condition and the frequency at $f_{max} = 280/\text{s}$ after which vertical bubble coalescence became large. Bubble

interactions and coalescence and their effect on the heat transfer are not accounted for in these models. Using these equations, the heat flux partitions for the plain and microporous coated surfaces were obtained.

3.3.1 Plain surfaces

Fig. 14 (a) and (b) show the heat flux partition calculated from the N , D , and f on the plain surface (with $R_a = 0.33 \mu\text{m}$ and $0.78 \mu\text{m}$, respectively) as a function of the superheat. The solid symbols indicate the partition fluxes calculated using the experimentally obtained bubble parameters. The open symbols denote the partition fluxes calculated using the projected values of N , D , and f from the correlations.

Based on the partition model, the proportions of heat that go into evaporative, quenching, and convection modes are plotted in the figures. The natural convection mode contributes little due to the small area of convection and the low natural convection heat transfer coefficient, and it decreases very quickly with increasing superheat. The evaporative and quenching (transient conduction) components contribute most to the total heat flux, and nearly equally (both contributing about 40% to 48% of the total calculated heat flux). In their experiments on pool boiling of water on plain surface, Gerardi et al. [21] observed that the quenching heat transfer was the dominant mode, and evaporative and convective modes were considerably smaller (<10%). This is probably because HFE-7100 is much more volatile than water, thus evaporates more readily resulting in a higher amount of heat going into the phase-change process.

The total calculated heat flux from the partition model matches much better with the experimental heat flux for the smoother plain surface, as shown in Fig. 14 (a), than the rougher surface, as shown in Fig. 14 (b). This is likely because the bubble site density on the smoother surface is much smaller than on the rougher surface, resulting in much less interference between

bubble sites. Thus the partition model, which neglects the heat transfer effects of the interference, is more accurate for the smooth surface. In the higher temperature range with projected values of the bubble site density, the total calculated heat flux falls significantly short of the experimental values. This is because the site density prediction from the H-I correlation for this surface does not grow as quickly with temperature as the experimental data (Fig. 11). The quenching and evaporative components follow a straight line as a function of temperature beyond 21°C (smoother surface) and 19°C (rougher surface), corresponding to the range where the site density and frequency are assumed to be constant.

3.3.2 Microporous surfaces

Fig. 15 shows the heat flux partition calculated from the N , D , and f on the three microporous surfaces as a function of the superheat. The solid symbols indicate the partition fluxes calculated using the experimentally obtained bubble parameters. The open symbols denote the partition fluxes calculated using the projected values of N , D , and f from the correlations.

In the low temperature range with the experimentally obtained values of the bubble parameters, the evaporative component is found to be the largest, followed by the quenching component. The evaporative component contributes (~65% to 73% of total calculated) significantly more than the quenching component (~20% to 25%). The natural convection component is close to zero. Compared to the plain surface, the contribution of the evaporative mode is much higher. This is due to the enhanced nucleation on the microporous surface versus the plain surface. In the high temperature range ($q''_{expt} > 25 \text{ W/cm}^2$), with the heat flux partitions calculated using the projected values of the bubble parameters, the slope of the total experimental heat flux is found to be about the same as that of the total calculated heat flux from

the partition model, with the offset being about 7 to 12 W/cm². This is possibly an indication that the bubble site density is indeed constant, just as assumed in the model.

In the microporous surface, the nucleation of bubbles takes place in cavities within the layer. The vapor bubble that is formed within the pores then moves through the pores before being released into the pool, detaching from the target surface. From the visualization method, we are able to observe the volume of bubble generated and therefore any heat transfer mode that results in the phase-change is indirectly accounted for. However, the bubble motion through the tunnels of the microporous layer will result in single-phase convective heat transfer to the liquid interface between the vapor and the solid. While the quenching component (q_{quench}) accounts for the single-phase convection during bubble growth outside the layer, the convective single-phase heat transfer within the microporous layer is not accounted for in the partition model. Taking this into consideration, it seems to suggest that a portion of the total heat flux to the pool from the heater with the microporous layer may actually be in the form of sensible heat transfer caused by the bubble motion through the pores, in addition to the enhancement due to the effects of bubble coalescence within and on the surface.

4 Conclusions

We experimentally studied the pool boiling heat transfer performance of copper surfaces coated with a thermally conductive microporous copper layer, in comparison with plain surfaces of two different roughness levels. Better heat transfer performances were observed with the microporous surface. Under all conditions tested, the microporous surface showed lower boiling incipience temperature (3°C–11°C) than the plain surface (17°C–38°C) due to the presence of cavities in the size range of 1 to 5 μm in large numbers on the microporous surface. The heat transfer coefficient increased by 50%–270% on the microporous surface as a result of the much

larger nucleate site density than on the plain surfaces. The CHF was enhanced by 33% to 60% for the microporous surface over the plain surface. High-speed visualization was utilized to obtain information on the bubble site density, departure diameter, and frequency of departure on the different surfaces. On all the surfaces, the bubble site density and frequency of departure increases sharply with temperature, while the diameter at departure stays relatively insensitive to temperature. Correlations available in the literature were used as given, or modified to fit the experimental data for the site density and diameter at departure, and extended them to the full temperature range until CHF. Using the information on the bubble site density, departure diameter, and frequency of departure, a simple heat flux partition model that neglects the effects of bubble coalescence was applied to obtain the contribution of various modes of heat transfer to the fluid from the various target surfaces. On the plain surfaces, both evaporative and quenching components contribute nearly equally to the total heat flux. On the microporous surfaces, the evaporative component contributes about 70% of the total heat, thus confirming that the microporous coating enhances nucleate boiling heat transfer as a direct result of its significantly higher active nucleation site density.

Acknowledgments

This work was performed under a subcontract from the National Renewable Energy Laboratory (NREL) to the University of Colorado – Boulder. The authors thank the support of Susan Rogers and Steven Boyd, Technology Development Managers of the Electric Drive Technologies Program in the U.S. Department of Energy Vehicle Technologies Office. The authors also thank Gilbert Moreno for help with the experimental apparatus, as well as Phil Tuma (3M), who supplied the target surfaces with the microporous coatings.

References

- [1] I. Mudawar, Assessment of high-heat-flux thermal management schemes, *IEEE Trans. Components Packag. Technol.* 24 (2) (2001) 122–141.
- [2] U.S. Department of Energy, Electrical and Electronics Technical Team Roadmap, Washington D.C., 2013.
- [3] J.R. Thome, Enhanced boiling heat transfer, Hemisphere Publishing Corporation, New York, 1990.
- [4] H. Honda, J. Wei, Enhanced boiling heat transfer from electronic components by use of surface microstructures, *Exp. Therm. Fluid Sci.* 28 (2004) 159–169.
- [5] J. Tehver, H. Sui, V. Temkina, Heat transfer and hysteresis phenomena in boiling on porous plasma-sprayed surface, *Exp. Therm. Fluid Sci.* 5 (6) (1992) 714–727.
- [6] J.Y. Chang, S.M. You, Boiling heat transfer phenomena from microporous and porous surfaces in saturated FC-72, *Int. J. Heat Mass Transf.* 40 (18) (1997) 4427–4447.
- [7] K.N. Rainey, S.M. You, Effects of heater size and orientation on pool boiling heat transfer from microporous coated surfaces, *Int. J. Heat Mass Transf.* 44 (14) (2001) 2589–2599.
- [8] M.S. El-Genk, Nucleate boiling enhancements on porous graphite and microporous and macro-finned copper surfaces, *Heat Transf. Eng.* 33 (3) (2012) 175–204.
- [9] A.E. Bergles, M.C. Chyu, Characteristics of nucleate pool boiling from porous metallic coatings, *J. Heat Transfer* 104 (2) (1982) 279–285.

- 1 [10] J.H. Kim, K.N. Rainey, S.M. You, J.Y. Pak, Mechanism of nucleate boiling heat transfer
2 enhancement from microporous surfaces in saturated FC-72, *J. Heat Transfer* 124 (3)
3 (2002) 500–506.
- 4 [11] S.J. Thiagarajan, S. Narumanchi, C. King, W. Wang, R. Yang, Enhancement of heat
5 transfer with pool and spray impingement boiling on microporous and nanowire surface
6 coatings, in: *Proc. 14th Int. Heat Transf. Conf.*, Washington, D.C., 2010: pp. 1–10.
- 7 [12] S.J. Thiagarajan, S. Narumanchi, R. Yang, Effect of flow rate and subcooling on spray
8 heat transfer on microporous copper surfaces, *Int. J. Heat Mass Transf.* 69 (2014) 493–
9 505.
- 10 [13] S.J.D. van Stralen, The mechanism of nucleate boiling in pure liquids and in binary
11 mixtures—part I, *Int. J. Heat Mass Transf.* 9 (10) (1966) 995–1006.
- 12 [14] B.B. Mikic, W.M. Rohsenow, P. Griffith, On bubble growth rates, *Int. J. Heat Mass*
13 *Transf.* 13 (4) (1970) 657–666.
- 14 [15] K.E. Forster, R. Greif, Heat transfer to a boiling liquid, *J. Heat Transfer* 81 (1959) 43–53.
- 15 [16] B.B. Mikic, W.M. Rohsenow, A new correlation of pool-boiling data including the effect
16 of heating surface characteristics, *J. Heat Transfer* 91 (2) (1969) 245–250.
- 17 [17] W.M. Rohsenow, A method of correlating heat transfer data for surface boiling of liquids,
18 *J. Heat Transfer* 74 (1952) 969–976.

- 1 [18] H.K. Forster, N. Zuber, Dynamics of vapor bubbles and boiling heat transfer, AIChE J. 1
2 (4) (1955) 531–535.
- 3 [19] C.L. Tien, A hydrodynamic model for nucleate pool boiling, Int. J. Heat Mass Transf. 5
4 (6) (1962) 533–540.
- 5 [20] A. Luke, D.-C. Cheng, High speed video recording of bubble formation with pool boiling,
6 Int. J. Therm. Sci. 45 (2006) 310–320.
- 7 [21] C. Gerardi, J. Buongiorno, L. Hu, T. McKrell, Study of bubble growth in water pool
8 boiling through synchronized, infrared thermometry and high-speed video, Int. J. Heat
9 Mass Transf. 53 (19-20) (2010) 4185–4192.
- 10 [22] J. McHale, S. Garimella, Nucleate boiling from smooth and rough surfaces–Part 2:
11 analysis of surface roughness effects on nucleate boiling, Exp. Therm. Fluid Sci. 44
12 (2013) 439–455.
- 13 [23] I. Mudawar, D. Bharathan, K. Kelly, S. Narumanchi, Two-phase spray cooling of hybrid
14 vehicle electronics, IEEE Trans. Components Packag. Technol. 32 (2009) 501–512.
- 15 [24] 3M product brochure, 3M NovecTM engineered fluid HFE-7100 for heat transfer, (2002).
- 16 [25] C.Y. Ho, R.W. Powell, P.E. Liley, Thermal Conductivity of the Elements, J. Phys. Chem.
17 Ref. Data 1 (2) (1972) 279.
- 18 [26] M.S. El-Genk, H. Bostanci, Saturation boiling of HFE-7100 from a copper surface,
19 simulating a microelectronic chip, Int. J. Heat Mass Transf. 46 (2003) 1841–1854.

- [27] 3M product brochure, 3M microporous metallic boiling enhancement coating (BEC) L-20227, (2008).
- [28] P.E. Tuma, Evaporator/boiler design for thermosyphons utilizing segregated hydrofluoroether working fluids, in: 22nd Annu. IEEE Semicond. Therm. Meas. Manag. Symp., 2006: pp. 69–77.
- [29] S.J. Kline, F.A. McClintock, Describing uncertainties in single-sample experiments, Mech. Eng. 75 (1) (1953) 3–8.
- [30] G. Moreno, S. Narumanchi, C. King, Pool boiling heat transfer characteristics of HFO-1234yf on plain and microporous-enhanced surfaces, J. Heat Transfer 135 (11) (2013) 111014.
- [31] T.M. Anderson, I. Mudawar, Microelectronic cooling by enhanced pool boiling of a dielectric fluorocarbon liquid, J. Heat Transfer 111 (3) (1989) 752.
- [32] J.H. Lienhard, V.K. Dhir, D.M. Riherd, Peak pool boiling heat-flux measurements on finite horizontal flat plates, J. Heat Transfer 95 (4) (1973) 477.
- [33] J.H. Lienhard, V.K. Dhir, Hydrodynamic prediction of peak pool-boiling heat fluxes from finite bodies, J. Heat Transfer 95 (2) (1973) 152.
- [34] R. Raj, J. Kim, Heater size and gravity based pool boiling regime map: transition criteria between buoyancy and surface tension dominated boiling, J. Heat Transfer 132 (9) (2010) 091503.

- 1 [35] C.H. Wang, V.K. Dhir, Effect of surface wettability on active nucleation site density
2 during pool boiling of water on a vertical surface, *J. Heat Transfer* 115 (3) (1993) 659.
- 3 [36] S.M. You, T.W. Simon, A. Bar-Cohen, W. Tong, Experimental investigation of nucleate
4 boiling incipience with a highly-wetting dielectric fluid (R-113), *Int. J. Heat Mass Transf.*
5 33 (1990) 105–117.
- 6 [37] Y.Y. Hsu, On the size range of active nucleation cavities on a heating surface, *J. Heat*
7 *Transfer* 84 (3) (1962) 207–213.
- 8 [38] D. Gorenflo, E. Danger, A. Luke, S. Kotthoff, U. Chandra, C. Ranganayakulu, Bubble
9 formation with pool boiling on tubes with or without basic surface modifications for
10 enhancement, *Int. J. Heat Fluid Flow* 25 (2) (2004) 288–297.
- 11 [39] T. Hibiki, M. Ishii, Active nucleation site density in boiling systems, *Int. J. Heat Mass*
12 *Transf.* 46 (14) (2003) 2587–2601.
- 13 [40] G. Kocamustafaogullari, M. Ishii, Interfacial area and nucleation site density in boiling
14 systems, *Int. J. Heat Mass Transf.* 26 (9) (1983) 1377–1387.
- 15 [41] R.J. Benjamin, A.R. Balakrishnan, Nucleation site density in pool boiling of saturated
16 pure liquids: Effect of surface microroughness and surface and liquid physical properties,
17 *Exp. Therm. Fluid Sci.* 15 (1) (1997) 32–42.
- 18 [42] W. Tong, A. Bar-Cohen, T.W. Simon, S.M. You, Contact angle effects on boiling
19 incipience of highly-wetting liquids, *Int. J. Heat Mass Transf.* 33 (1) (1990) 91–103.

- [43] R. Cole, Bubble frequencies and departure volumes at subatmospheric pressures, *AIChE J.* 13 (4) (1967) 779–783.
- [44] S.S. Kutateladze, I.I. Gogonin, Growth rate and detachment diameter of a vapor bubble in free convection boiling of a saturated liquid, *High Temp.* 17 (1980) 667–671.
- [45] M.K. Jensen, G.J. Memmel, Evaluation of bubble departure diameter correlations, in: *Proc. 8th Int. Heat Transf. Conf.*, Begell House Inc., 1986: pp. 1907–1912.
- [46] H.T. Phan, N. Caney, P. Marty, S. Colasson, J. Gavillet, How does surface wettability influence nucleate boiling?, *Comptes Rendus Mécanique* 337 (5) (2009) 251–259.
- [47] S. Moghaddam, K. Kiger, Physical mechanisms of heat transfer during single bubble nucleate boiling of FC-72 under saturation conditions-I. Experimental investigation, *Int. J. Heat Mass Transf.* 52 (5-6) (2009) 1284–1294.
- [48] F.N. Peebles, H.J. Garber, Studies on the motion of gas bubbles in liquids, *Chem. Eng. Prog.* 49 (1953) 88–97.
- [49] V.P. Carey, *Liquid-vapor phase-change phenomena: an introduction to the thermophysics of vaporization and condensation processes in heat transfer equipment*, 2nd ed., Taylor and Francis, New York, 2007.
- [50] J.P. McHale, S. V. Garimella, Bubble nucleation characteristics in pool boiling of a wetting liquid on smooth and rough surfaces, *Int. J. Multiph. Flow* 36 (4) (2010) 249–260.

- 1 [51] R.W. Bowring, Physical model, based on bubble detachment, and calculation of steam
2 voidage in the subcooled region of a heated channel, in: HPR-10, Inst. Atomenergi,
3 Halden, Norw., 1962.
- 4 [52] W.H. McAdams, Heat transmission, 3rd ed., McGraw-Hill, New York, 1954.
- 5 [53] C.-Y. Han, P. Griffith, The mechanism of heat transfer in nucleate pool boiling—Part I,
6 Int. J. Heat Mass Transf. 8 (6) (1965) 887–904.
- 7 [54] F. Demiray, J. Kim, Microscale heat transfer measurements during pool boiling of FC-72:
8 effect of subcooling, Int. J. Heat Mass Transf. 47 (14-16) (2004) 3257–3268.
- 9 [55] S. Moghaddam, K. Kiger, Physical mechanisms of heat transfer during single bubble
10 nucleate boiling of FC-72 under saturation conditions-II: Theoretical analysis, Int. J. Heat
11 Mass Transf. 52 (5-6) (2009) 1295–1303.

12

Tables

Table 1. Thermophysical properties of saturated HFE-7100 at various temperatures.

T_{sat} (°C)	P_{sat} (kPa)	Liquid density ρ_l (kg/m ³)	Vapor density ρ_v (kg/m ³)	Liquid viscosity μ_l (Pa-s)	Surface tension σ (N/m)	Thermal conductivity κ_l (W/m-K)	Specific heat C_p (J/kg-K)	Latent heat of vaporization h_{lv} (kJ/kg)
70	133	1394	12.5	3.5×10^{-4}	8.5×10^{-3}	0.06	1273	109
61	100	1418	9.3	3.8×10^{-4}	9.3×10^{-3}	0.062	1254	112
54	80	1438	7.7	4.1×10^{-4}	10.2×10^{-3}	0.063	1241	114

Table 2. The range of superheat temperatures at onset of nucleate boiling on the different surfaces, and the corresponding critical nucleating embryo sizes.

Surface	Range of $\Delta T_{w,ONB}$ (°C)	Range of D_{crit} (μm)
Plain ($R_a = 0.33$ μm)	24.5 – 36.2	0.17 – 0.35
Plain ($R_a = 0.78$ μm)	14 – 24.3	0.37 – 0.85
Microporous (100 μm)	9 – 10.9	1.2 – 1.6
Microporous (360 μm)	3.4 – 10.1	1.4 – 7
Microporous (700 μm)	3.2 – 7	2.3 – 7.8

List of Figures

Fig. 1. Schematic diagram showing the different modes of heat and mass transfer on (a) plain surface and (b) inside and outside the microporous surface. In addition to the modes of heat transfer shown in the figure, bubble pumping action results in convective heat transfer inside the microporous layer..... 42

Fig. 2. (a) Schematic diagram and (b) photograph of the pool boiling facility. Inset in (b) shows the photograph of the test heater assembly with the 2 mm x 10 mm copper surface exposed on top. Photo credit: Suraj Thiagarajan (NREL). 43

Fig. 3. Scanning electron microscope images of (a) plain surface of $R_a = 0.33 \mu\text{m}$ and (b) plain surface of $R_a = 0.78 \mu\text{m}$; (c) and (d) microporous surface at two different magnification levels. Credit: (a, b) Suraj Thiagarajan; (c, d) Bobby To (NREL). 44

Fig. 4. Results of steady-state thermal simulation: two cases, corresponding to heat input (1) $q_{in} = 1.74 \text{ W}$ (with $h = 4,200 \text{ W/m}^2\text{-K}$) and (2) $q_{in} = 5.83 \text{ W}$ (with $h = 13,200 \text{ W/m}^2\text{-K}$) are shown. (a) Boundary conditions applied to the various portions of the heater assembly, (b) temperature contour in the copper block in the direction of heat transfer; (c) and (d) show the temperature contour plots in the plane perpendicular to the direction of heat conduction at thermocouple location: (T_1) 4 mm from the surface and (T_2) 1 mm from the surface, respectively. 45

Fig. 5. Pool boiling curves on plain surface with target surface dimension of 10 mm x 10 mm. A comparison is made with data from [26]. Inset shows the schematic of the cross-section of the test heater assembly. Data for the smooth plain surface of dimension 2 mm x 10 mm are shown here for comparison. The Lienhard-Dhir correlation for CHF on an infinitely large flat surface gives 16.7 W/cm^2 46

1	Fig. 6. (a) Pool boiling curves and (b) heat transfer coefficients for the various surfaces.	
2	Arrows denote CHF. Horizontal bars indicate the range of the onset of nucleate boiling superheat	
3	in different test runs.	47
4	Fig. 7. Predicted size range of active cavity sites as a function of wall superheat based on	
5	Hsu's criterion, for the range of contact angle $1^\circ < \phi < 50^\circ$. The active sites are located in the	
6	concave portion of each curve. The range of critical diameters of the nucleating embryo	
7	calculated using Eq. (5) for each of the different surfaces from the superheat temperature at ONB	
8	is also shown.	48
9	Fig. 8. Images of bubble dynamics on a plain and a microporous surface at various heat flux	
10	levels. Credit: Suraj Thiagarajan (NREL).	49
11	Fig. 9. Experimental measurements of bubble site density compared with correlations for the	
12	two plain surfaces of roughness (a) $0.33\ \mu\text{m}$ and (b) $0.78\ \mu\text{m}$. The extended range corresponds to	
13	the superheat up to the CHF.	50
14	Fig. 10. Experimental measurements of bubble site density compared with correlations for	
15	the microporous surfaces of thickness (a) $700\ \mu\text{m}$, (b) $360\ \mu\text{m}$, and (c) $100\ \mu\text{m}$	51
16	Fig. 11. Bubble site density at departure on the various surfaces as measured using the high-	
17	speed video. The symbols are the measurements and the solid lines are the fits based on the H-I	
18	correlation as explained in section 3.2.1. N_{max} represents the maximum bubble site density when	
19	the surface is fully covered by bubbles at the respective departure diameter.	52
20	Fig. 12. (a) Representative distribution of departure diameters on plain and microporous	
21	surfaces. (b) Average bubble departure diameter for the various surfaces. Predictions based on	
22	several different models are also shown. The best agreement was found using the model from	
23	Phan et al. [46] with the same contact angle as used for the nucleate site density correlations. ..	53

1	Fig. 13. Average frequency of bubble departure for the various surfaces as a function of the	
2	superheat. The solid lines and equations in each case represent linear data fits.....	54
3	Fig. 14. The partition of heat flux model applied to the plain surfaces: (a) smooth ($R_a = 0.33$	
4	μm) and (b) rough ($R_a = 0.78 \mu\text{m}$). The solid line with * markers is the experimental data. The	
5	solid markers denote the data points of N , D , and f from experiments, and the open symbols	
6	denote the data from extrapolated N , D , and f	55
7	Fig. 15. The partition of heat flux model applied to the microporous surfaces: (a) $700 \mu\text{m}$, (b)	
8	$360 \mu\text{m}$, and (c) $100 \mu\text{m}$. The solid line with * markers is the experimental data. The solid	
9	markers denote the data points of N , D , and f from experiments, and the open symbols denote the	
10	data from extrapolated N , D , and f	57
11		

Figures

2

Fig. 1. Schematic diagram showing the different modes of heat and mass transfer on (a) plain surface and (b) inside and outside the microporous surface. In addition to the modes of heat transfer shown in the figure, bubble pumping action results in convective heat transfer inside the microporous layer.

Fig. 2. (a) Schematic diagram and (b) photograph of the pool boiling facility. Inset in (b) shows the photograph of the test heater assembly with the 2 mm x 10 mm copper surface exposed on top. Photo credit: Suraj Thiagarajan (NREL).

1

Fig. 3. Scanning electron microscope images of (a) plain surface of $R_a = 0.33 \mu\text{m}$ and (b) plain surface of $R_a = 0.78 \mu\text{m}$; (c) and (d) microporous surface at two different magnification levels. Credit: (a, b) Suraj Thiagarajan; (c, d) Bobby To (NREL).

Case 1:

Heat input $q_{in} = 1.74 \text{ W}$;

Heat flux on surface $q'' = 5.8 \text{ W/cm}^2$

Case 2:

Heat input $q_{in} = 5.83 \text{ W}$;

Heat flux on surface $q'' = 26 \text{ W/cm}^2$

Fig. 4. Results of steady-state thermal simulation: two cases, corresponding to heat input (1) $q_{in} = 1.74 \text{ W}$ (with $h = 4,200 \text{ W/m}^2\text{-K}$) and (2) $q_{in} = 5.83 \text{ W}$ (with $h = 13,200 \text{ W/m}^2\text{-K}$) are shown. (a) Boundary conditions applied to the various portions of the heater assembly, (b) temperature contour in the copper block in the direction of heat transfer; (c) and (d) show the temperature contour plots in the plane perpendicular to the direction of heat conduction at thermocouple location: (T_1) 4 mm from the surface and (T_2) 1 mm from the surface, respectively.

1

2

3

4

5

6

7

Fig. 5. Pool boiling curves on plain surface with target surface dimension of 10 mm x 10 mm. A comparison is made with data from [26]. Inset shows the schematic of the cross-section of the test heater assembly. Data for the smooth plain surface of dimension 2 mm x 10 mm are shown here for comparison. The Lienhard-Dhir correlation for CHF on an infinitely large flat surface gives 16.7 W/cm^2 .

Fig. 6. (a) Pool boiling curves and (b) heat transfer coefficients for the various surfaces.

Arrows denote CHF. Horizontal bars indicate the range of the onset of nucleate boiling superheat in different test runs.

1

2

3

4

5

6

7

8

Fig. 7. Predicted size range of active cavity sites as a function of wall superheat based on Hsu's criterion, for the range of contact angle $1^\circ < \phi < 50^\circ$. The active sites are located in the concave portion of each curve. The range of critical diameters of the nucleating embryo calculated using Eq. (5) for each of the different surfaces from the superheat temperature at ONB is also shown.

1 Fig. 8. Images of bubble dynamics on a plain and a microporous surface at various heat flux
2 levels. Credit: Suraj Thiagarajan (NREL).

(a)

(b)

Fig. 9. Experimental measurements of bubble site density compared with correlations for the two plain surfaces of roughness (a) $0.33 \mu\text{m}$ and (b) $0.78 \mu\text{m}$. The extended range corresponds to the superheat up to the CHF.

(a)

(b)

(c)

Fig. 10. Experimental measurements of bubble site density compared with correlations for the microporous surfaces of thickness (a) 700 μm , (b) 360 μm , and (c) 100 μm .

Fig. 11. Bubble site density at departure on the various surfaces as measured using the high-speed video. The symbols are the measurements and the solid lines are the fits based on the H-I correlation as explained in section 3.2.1. N_{max} represents the maximum bubble site density when the surface is fully covered by bubbles at the respective departure diameter.

(a)

(b)

Fig. 12. (a) Representative distribution of departure diameters on plain and microporous surfaces. (b) Average bubble departure diameter for the various surfaces. Predictions based on several different models are also shown. The best agreement was found using the model from Phan et al. [46] with the same contact angle as used for the nucleate site density correlations.

1

2

3 Fig. 13. Average frequency of bubble departure for the various surfaces as a function of the

4 superheat. The solid lines and equations in each case represent linear data fits.

5

(a) Plain smooth
($R_a = 0.33 \mu\text{m}$)

(b) Plain rough
($R_a = 0.78 \mu\text{m}$)

Fig. 14. The partition of heat flux model applied to the plain surfaces: (a) smooth ($R_a = 0.33 \mu\text{m}$) and (b) rough ($R_a = 0.78 \mu\text{m}$). The solid line with * markers is the experimental data. The solid markers denote the data points of N , D , and f from experiments, and the open symbols denote the data from extrapolated N , D , and f .

(a) 700 μm

(b) 360 μm

(c) 100 μm

Fig. 15. The partition of heat flux model applied to the microporous surfaces: (a) 700 μm , (b) 360 μm , and (c) 100 μm . The solid line with * markers is the experimental data. The solid markers denote the data points of N , D , and f from experiments, and the open symbols denote the data from extrapolated N , D , and f .

TITLE PAGE

1. TITLE OF THE PAPER

Bubble dynamics and nucleate pool boiling heat transfer on microporous copper surfaces

2. AUTHORS

- i. Name: Suraj Jootu Thiagarajan
Affiliation: Department of Mechanical Engineering, University of Colorado Boulder.
Address: Department of Mechanical Engineering
427 UCB University of Colorado
1111 Engineering Drive
Boulder, CO 80309-0427, U.S.A.
Email: Suraj.Thiagarajan@Colorado.edu
Phone: 1-720-505-9517.
- ii. Name: Ronggui Yang (*Corresponding author*)
Affiliation: Department of Mechanical Engineering, University of Colorado Boulder.
Address: Department of Mechanical Engineering
427 UCB University of Colorado
1111 Engineering Drive
Boulder, CO 80309-0427, U.S.A.
Email: Ronggui.Yang@Colorado.edu
Phone: 1-303-735-1003
Fax: 1-303-492-3498.
- iii. Name: Charles King
Affiliation: National Renewable Energy Laboratory, Golden, CO
Address: National Renewable Energy Laboratory,
15013 Denver West Parkway,
Golden, CO, 80401, U.S.A.
Email: Charles.King@NREL.gov
Phone: 1-303-275-4419
- iv. Name: Sreekant Narumanchi (*Corresponding author*)
Affiliation: National Renewable Energy Laboratory, Golden, CO
Address: National Renewable Energy Laboratory,
15013 Denver West Parkway,
Golden, CO, 80401, U.S.A.
Email: Sreekant.Narumanchi@NREL.gov
Phone: 1-303-275-4062
Fax: 1-303-275-4415.