Visualizing and disrupting liquid films for filmwise flow condensation in horizontal minichannels

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11 **Abstract**

12   This paper investigates the effects of hemispherical mounds on filmwise condensation heat
13 transfer in micro-channels. Also investigated were the impacts that spatial orientation of the three-sided
14 condensation surface (i.e., gravitational effects) on steam condensation, where the cooled surfaces were
15 either the lower surface (i.e., gravity pulls liquid towards the condensing surfaces) or upper surface
16 (i.e., gravity pulls liquid away from the condensing surfaces). Two test coupons were used with 1.9-
17 mm hydraulic diameters and either a plain copper surface or a copper surface modified with 2-mm
18 diameter hemispherical mounds. Heat transfer coefficients, film visualization, and pressure drop
19 measurements were recorded for both coupons in both orientations at mass fluxes of 50 kg/m^{2}s and
20 125 kg/m^{2}s. For all test conditions, the mounds were found to increase condensation heat transfer
21 coefficients by at minimum 13% and at maximum 79%. When the test section was inverted (i.e.,
22 condensing surface on the top of flowing steam), minimal differences were found in mound
23 performance, while the plain coupon reduces heat transfer coefficients by as much as 14%. Flow
24 visualization suggests that the mounds enhanced heat transfer due to the disruption of the film as well
25 as by reducing the thermal resistance of the film. Pressure drops followed parabolic behavior with
26 quality, being higher in the mound coupon than the plain coupon. No significant pressure drop
27 differences in the inverted orientation were observed.

28 1. **Introduction**

29   Closed-loop thermal management systems utilize their working fluid to remove heat from a
30 source, such as electronics or motors, and reject that heat into a heat sink, such as a radiator (Ho and
31 Leong, 2021). In space, it is particularly important to be able to move the thermal energy since cooling
32 of the entire system is only possible by thermal radiation (Mudawar, 2017). By using a two-phase fluid
33 in the closed-loop, the latent heat of the working fluid can be used advantageously by either evaporating
34 or condensing it, allowing for significant heat transfer without significant temperature gradients in the

35 fluid. For condensation processes, the condensed liquid (i.e., condensate) acts to insulate the cooling
36 surface from the working fluid (Sun and Wang, 2016; Niu et al., 2017; Alizadeh-Birjandi et al., 2019).
37 While condensation can be dropwise or filmwise, filmwise condensation is prevalent in internal flows
38 as it does not require any special coatings or conditions to occur (Ma et al., 2010; Chen and Derby,
39 2016; Alizadeh-Birjandi et al., 2019). Since the film in filmwise condensation prevents direct heat
40 transfer between the cooling surface and the vapor, heat transfer coefficients tend to be an order of
41 magnitude lower compared to dropwise condensation (Rose, 2002; Orejon et al., 2017; El Fil et al.,
42 2020). As such, when the film is thinner, for example, occurring when the quality is high, heat transfer
43 coefficients increase. Similarly, heat transfer coefficients increase with increasing mass flow rate where
44 convection becomes more significant relative to conductive heat transfer through the film.

45   In mini-channels, where the hydraulic diameter is between 200 µm and 3 mm and shear forces
46 become dominate, the film wets the entire surface, thins, and forms an annulus of condensate through
47 which the vapor flows (Soliman et al., 1968; Soliman, 1982; Kandlikar and Grande, 2003). Kim and
48 Mudawar (2013) developed a correlation for determining the particular flow regime (i.e. slug or wavy
49 annular) determined by the fluid properties, flow rate, and quality (Kim and Mudawar, 2013). While
50 the thinner film and flow regimes improve heat transfer rates, the decreased diameter increases the
51 pressure drop across the condenser (Kim and Mudawar, 2012). This increased pressure drop requires
52 additional work from the closed-loop compressor or pump, which can negate the positive impact on
53 heat transfer. Therefore, enhancements that increase the heat transfer coefficient without substantially
54 increasing the pressure drop in condensers are of particular interest (Mudawar, 2017).

55   One approach for enhancing condensation in micro-channels is to coat the surface in hydrophobic
56 substance such as Teflon, which transitions the filmwise condensation into dropwise condensation,
57 resulting in enhancements up to 10 times, yet coatings are not always durable (Miljkovic and Wang,
58 2013; Chen and Derby, 2016; Chen et al., 2017; Antao et al., 2020; Hoque et al., 2022). However,
59 durable hydrophobic coatings remain an active research area, thus condensation processes in industry
60 are filmwise condensation instead of dropwise condensation (Ahlers et al., 2019; Chang et al., 2020).
61 Another means of condensation heat transfer is to roughen hydrophilic surfaces to decrease the
62 wettability and increase the roughness Reynolds number, in order to increase the heat transfer
63 coefficient (Dipprey and Sabersky, 1963; Nicol and Medwell, 1966; Nguyen and Ahn, 2021).

64   Other approaches to enhancing filmwise condensation include physically modifying the
65 condensing surface with structures designed to increase condensing surface area, penetrate the film, or
66 disrupt the film by the addition of fins, twisted tape, or corrugation inside of the condensing
67 channel(Cavallini et al., 2003; Dalkilic and Wongwises, 2009; El Kadi et al., 2021). Ho and Leong
68 (2019) used conical pin fins in a circular tube and compared them to dome shaped fins evaluated by
69 Wang et. al. (2018). Using R134a in 8.7-mm channels, Ho and Leong (2019) determined that helically
70 arranged conical fins could enhance the heat transfer coefficient by up to 2.44 times more than pressure
71 drop was increased (Ho et al., 2019). This increase in heat transfer coefficient enhancement relative to
72 increased pressure was not found to be true of the dome fins used by Wang et al. (2018) also using
73 R134a in 8.7-mm channels (Wang et al., 2018; Ho et al., 2019). Aroonrat and Wongwises (2019) used
74 a 8.1-mm circular channel with R134a where dimples on the exterior of the tube were used to create
75 hollow pin fins with depths of 0.5 mm, 0.75 mm, and 1 mm with diameters of 1 mm, 1.5 mm, and 2
76 mm, respectively, in the channel. While they were able to increase the heat transfer coefficient by up
77 to 83%, the pressure drop was increased by up to 892% compared to a non-modified channel (Aroonrat
78 and Wongwises, 2019).

79   Although steam flow condensation has applications in air-cooled condensers and solar energy
80 production, there are limited flow condensation data for condensation of steam in mini-channels
81 (Dirker et al., 2019; El Kadi et al., 2021). Research conducted using fins and other structures to enhance
82 heat transfer properties of steam have largely focused on vertical gravity-driven condensation
83 processes. Winter and McCarthy (2020) used a series of open, parallel amphiphilic channels to promote

84   droplet nucleation in the bottom of the channel while the top of the channel is hydrophobic, cause the
85   droplets to de-wet and be removed from the surface to increase the condensation rate (Winter and
86   McCarthy, 2020; Winter et al., 2021). Modak et al (2019) performed a numerical analysis and
87   experimentally verified the impacts a monolayer of spherical particles on a vertical condensing plate
88   and found that, depending on the particle size, they will act as either a fin, a wick, or as surface
89   roughness with reductions in thermal resistance occurring for particle diameters between 1 μm to 50
90   μm and above 700 μm (Modak et al., 2019). Ho and Leong (2020) studied a vertical plate packed with
91   conical fins to increase the gravity-driven condensation heat transfer coefficient 146% over a plain
92   surface(Ho and Leong, 2020).
93   While these enhancement methods can be easily used within Earth's gravity, for microgravity
94   applications, such as satellites or spacecraft, the impacts that gravity has on the condensation process
95   need to be understood. Since performing experiments in micro-gravity settings is time and resource
96   intensive, changing the orientation of test sections so that any gravitational impacts will work with or
97   against the condensation process allows researchers to predict how the process would change in a
98   micro-gravity setting (Mudawar, 2017). As the diameter of a channel decreases, the relative strength
99   of surface tension and shear forces increase relative to gravitational forces (Wen et al., 2018). For low
100 mass velocity flows in particular though, gravitational forces can still distort the shape of the annulus,
101 preventing radial symmetry in horizontal circular channels (Mudawar, 2017). For channels that have
102 neither radial symmetry nor condensation on all surfaces, such as the channels investigated in this
103 paper, there is little information available regarding the effects of channel orientation on the
104 condensation properties of the channel.
105   The objectives of this study are to investigate the impacts of hemispherical structures in rectangular
106 mini-channels on liquid film thickness and its impacts on heat transfer during flow condensation.
107 Observations were made by measuring the heat transfer, pressure drop, and visualizing the condensate
108 film. Additionally, this study examines the effects of gravity on the horizontal two-phase flows by
109 changing the orientation of the test section such that condensation would either occur on the lower or
110 upper surface of the channel.

111 **2.   Materials and methods**
112 **2.1. Experimental apparatus**

113   Experiments were conducted using an open-loop steam apparatus, shown in Figure 1, designed to
114 measure condensation heat transfer coefficients, pressure drops and allow for visualization of the
115 condensation process. Steam is regulated to 250 kPa for all test conditions and passes through a
116 separator to remove any water and a filter system to remove any contaminates which may be in the
117 steam supply. Due to uncertainty regarding the initial steam quality, the steam is superheated by 20°C
118 to 30°C. The superheated steam then passes through a pre-condenser to control the quality of the steam
119 entering the test section. The cooling water is supplied by a constant-temperature bath (Neslab RTE-
120 221), its mass flow rate measured via a Coriolis flow meter (CMFS015M, Micro Motion), and its
121 temperature is measured by two T-type thermocouples.
122   The steam temperature was measured before and after the pre-condenser using T-type
123 thermocouples (Omega; T-Q-SS-116-G-3) and as well as after passing through the test section. A
124 differential pressure transducer (Omega; PX409) was used to measure the pressure drop in the test
125 section. Five T-type thermocouples (Omega; TJC36-CPSS-062U-2) installed in the cooling block were
126 used to calculate the heat flux leaving the steam and entering cooling water supplied by a second
127 constant-temperature bath. A sixth T-type thermocouple in the test coupon was used to determine the
128 wall temperature in the channel. To test the effects of gravity on flow condensation, the same apparatus
129 was used where the test section was inverted such that the cooling surface was above the steam flow

130 as seen in Figure 2. In both orientations, visualization of the film was conducted using Leica Z16 APO
131 macroscope and a FASTEC IL3 high-speed camera. Temperature and pressure data were collected
132 using LabVIEW and a cDAQ-9174 with NI TB-9214 and NI 9207 modules.
133   Two test coupons, shown in Figure 3, were constructed out of oxygen-free copper and with a
134 channel whose hydraulic diameter was 1.9 mm. The first coupon had a channel with a flat rectangular
135 cross section and no additional structures added. The second test section had copper hemispheres added
136 via the following process. The physical and chemical contaminations on the copper coupon surface
137 were removed by using acetone, and then dried under ambient conditions. The 2-mm copper balls were
138 then placed in the hemi-spherical, pre-machined grooves in the channel of the coupon. A stainless-steel
139 mold with the same hemi-spherical groove pattern was placed over the copper balls with a light
140 pressure to secure the copper balls in place. Note that the stainless-steel mold was coated with carbon
141 powder to prevent from sintering the copper balls and the mold. The copper coupon with the copper
142 balls and the stainless-steel mold was then placed in a tube furnace (OTF-1200X) for sintering. The
143 temperature of the furnace during the sintering process was controlled with respect to time as shown
144 in Figure 4; the peak temperature used in this study was 1000°C. To avoid possible mechanical damage
145 due to sudden heating and cooling, the furnace heated and cooled at 3-5 °C/min with constant
146 temperature periods of 10-20 minutes between each temperature increase or decrease. At the end of
147 the sintering process, the stainless-steel mold was mechanically removed, leaving only the sintered
148 copper balls on the coupon. For the mound coupon, the hydraulic diameter was calculated by using the
149 length averaged cross-sectional area and perimeter.

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2.2. Data Reduction

153   Heat transfer coefficients, h, were calculated using Fourier's Law to determine the heat transfer
154 rate through the copper block, $\dot{Q}_{block}$, then determining h using the convection heat transfer equation,

$$\dot{Q}_{block} = -kA_{block}\frac{dT}{dy}\bigg|_{block} \qquad\qquad 1$$

$$h = \frac{\dot{Q}_{block}}{A_{surf}(T_{steam} - T_{wall})} \qquad\qquad 2$$

155 where the temperature gradient in the cooling block, dT/dy, was calculated using the least squares
156 method; A_{surf}, is the surface area where condensation occurs; A_{block} is the cross-sectional area of the
157 cooling block; and k is the thermal conductivity of the oxygen-free copper. For the mound coupon, the
158 planform area was used. Fluid temperature, T_{steam}, was determined using pressure of the two-phase
159 steam in the coupon, averaged using the pressure transducers before and after the test section. The wall
160 temperature, T_{wall}, was found using a thermocouple in the coupon 1.5 mm below the bottom of the
161 micro-channel.

2.3. Uncertainties

163   All measurement devices were calibrated. The T-type thermocouples were calibrated in a water
164 bath at 5°C increments from 5°C to 60°C, as well as in boiling water and in an ice bath, and compared
165 against a reference thermometer, resulting in a thermocouple uncertainty of ± 0.2 °C. For the
166 temperature gradient from the test section, the uncertainty was calculated using the following equation
167 (Kedzierski and Worthington III, 1993).

$$\omega_{\frac{dT}{dy}} = \sqrt{\omega_T^2 + \left(\frac{q'' D_{hole}}{6k}\right)} \sqrt{\frac{1}{\sum_{i=1}^{N}(y_i - \bar{y})^2}} \qquad 3$$

168   where D_{hole} is the diameter of the thermocouple hole, q'' is the heat flux, y is the position of each hole
169 along the direction of the thermal gradient, $\bar{y}$ is the average thermocouple position, and ω represents
170 the uncertainty. With the high thermal conductivity of the oxygen-free copper, and the relatively small
171 size of the thermocouple holes compared to the distance between them, the gradient uncertainty was
172 less than ± 10°C/m for all cases and was the largest contributor of uncertainty for the heat transfer
173 through the cooling block, and by extension, for the heat transfer coefficient. Pressure transducers were
174 calibrated using a deadweight tester, which allowed for a 0.25% full scale error which corresponded to
175 an uncertainty of ±1.7 kPa. Since the pressure and temperature of a two-phase fluid are not independent,
176 the pressures from the inlet and outlet of the test section were averaged and used to determine the
177 temperature of the steam, T_{steam}, in the test section with an uncertainty of ± 0.5°C. Subsequently, the
178 uncertainties for heat transfer coefficients were calculated,
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$$\omega_h = \qquad 4$$

$$\sqrt{\left(\frac{\partial h}{\partial \dot{Q}_{block}}\omega_{\dot{Q}_{block}}\right)^2 + \left(\frac{\partial h}{\partial A_{surf}}\omega_{A_{surf}}\right)^2 + \left(\frac{\partial h}{\partial T_{steam}}\omega_{T_{steam}}\right)^2 + \left(\frac{\partial h}{\partial T_{surf}}\omega_{T_{surf}}\right)^2}$$

184   where the uncertainties were dependent on uncertainties in the block heat transfer rate, $\dot{Q}_{block}$, surface
185 area, A_{surf}, two-phase saturation temperature of the condensing steam, T_{steam}, and surface
186 temperature, T_{surf}.

187 **3. Results and discussion**
188 **3.1. Single-phase validation**

189   To validate heat transfer measurements, steam was cooled to subcooled water at 250 kPa and 50
190 °C to 70°C for a single-phase validation using the plain coupon. This validation used two metrics, the
191 first was to compare the amount of energy which was exiting the water, $\dot{Q}_{water}$, to that which was
192 passing through the cooling block, $\dot{Q}_{block}$, from equation 1,

$$\dot{Q}_{water} = \dot{m}c_p(T_{in} - T_{out}) \qquad 5$$

193   where $\dot{m}$ is the mass flowrate of the fully condensed water, c_p is the specific heat at constant pressure,
194 and T_{in} and T_{out} are the temperature entering and exiting the test section respectively.

$$f(Re) = \left[\left(\frac{12}{\varepsilon^{1/2}(1+\varepsilon)\left(1 - 192\left(\frac{\varepsilon}{\pi^5}\right)tanh\left(\frac{\pi}{2\varepsilon}\right)\right)}\right)^2 + \left(\frac{3.44}{z^{+1/2}}\right)^2\right]^{1/2}$$

195   The second validation compared measured Nusselt numbers to the correlation made by Muzychka
196   and Yovanovich (2004) for non-circular ducts, which provides an upper and lower bound for the single-
197   phase Nusselt number (Muzychka and Yovanovich, 2004),

$$Nu = \left[\left(c_4\frac{f(Pr)}{z^*}\right)^m + \left(\left(c_2 c_3\left(\frac{f(Re)}{z^*}\right)\right)^{\frac{5}{3}} + \left(c_1\frac{f(Re)}{8\pi^{0.5}\varepsilon^\gamma}\right)^5\right)^{\frac{m}{5}}\right]^{\frac{1}{m}}$$

6

$$f(Pr) = \frac{0.564}{(1 + (1.909Pr^{1/6})^{9/2})^{2/9}}$$

8

198   where the values of c_1, c_2, c_3, c_4, and $f(Pr)$ are determined by the boundary conditions, ε is the aspect
199   ratio of the channel, and γ is the shape parameter which is 1/10 for the upper bound and -3/10 for the
200   lower bound. Additionally, $f(Pr)$, $f(Re)$, and Nu are the functions for the Prandtl number, Reynolds
201   number, and Nusselt number, respectively. The energy transfer from the subcooled water to the cooling
202   block was within 20% for all experimental points. Additionally, all Nusselt numbers were within or
203   within the error of the predicted value range, as shown in Figure 5. The energy balance and Nusselt
204   number analysis confirmed that the apparatus operated as predicted and that two-phase heat transfer
205   measurements are accurate.

206   **3.2. Flow condensation heat transfer coefficients**

207   Data collected from the plain coupon, in the standard orientation (i.e., condensation on lower
208   surface), served as a baseline to compare against for any enhancement. For a mass flux of 50 kg/m²s
209   and qualities from 0.20 to 0.87, the flow condensation heat transfer coefficients range from 22,200
210   W/m²K to 53,000 W/m²K. At the mass flux of 125 kg/m²s and qualities from 0.33 to 0.97, the flow
211   condensation heat transfer coefficients ranged from 40,800 W/m²K to 95,200 W/m²K. For both mass
212   fluxes, the heat transfer coefficients nearly linearly increased with quality; similarly, the heat transfer
213   coefficients are higher with the increased mass flux. To compare the other coupon and inverted test
214   section orientations, a third-degree polynomial was curve fit to these measured heat transfer
215   coefficients so that an enhancement factor could be calculated for any quality. The R^2 values for these
216   were 0.9908 and 0.9986 for the 50 kg/m²s and 125 kg/m²s data sets, respectively. Heat transfer
217   coefficients for all experiments are shown in Figure 6 and condensation heat transfer enhancements in
218   Figure 7. The two-phase heat transfer coefficients for the plain coupon in the standard configuration
219   (i.e., experiencing lower surface condensation) were compared to the values predicted by the Kim and
220   Mudawar (2013) correlation for condensing in mini/micro channels (Kim and Mudawar, 2013) in order
221   to compare to other experimental values obtained in similar channels. The Mean Absolute Percentage
222   Error (*MAPE*) for a mass flux of 50 kg/m²s was 38% and for a mass flux of 125 kg/m²s was 16%, as
223   shown in Figure 8,

$$MAPE = \frac{1}{n}\sum_{i=1}^{n} \left| \frac{h_{pred} - h_{exp}}{h_{exp}} \right|$$

224 where h_{pred} is the heat transfer coefficient predicted by the correlation, and h_{exp} is the observed heat
225 transfer coefficient from the experimental apparatus. Since the flow regimes (Section 3.3) influence
226 the predicted heat transfer coefficients, it should be noted that for the mass flux of 50 kg/m^{2}s, the flow
227 regimes predicted by the Kim and Mudawar (2013) correlation were all either in the transitional or slug
228 regimes, while for a mass flux of 125 kg/m^{2}s, the predicted flow regimes were either wavy-annular or
229 transitional.
230   For the mound coupon, condensation heat transfer coefficients were calculated using its planform
231 area, though the actual surface area of the mound coupon is 13.8% larger than that of the plain coupon.
232 With a mass flux of 50 kg/m^{2}K and qualities from 0.21 to 0.81, the heat transfer coefficients ranged
233 from 33,100 W/m^{2}K to 56,200 W/m^{2}K. This corresponds to a linearly decreasing enhancement of 47%
234 at a quality of 0.21, to a 13% enhancement at a quality of 0.81. At a mass flux of 125 kg/m^{2}s, the heat
235 transfer coefficients range from 44,300 W/m^{2}K to 132,000W/m^{2}K at qualities of 0.20 to 0.81,
236 respectively. The enhancement has a generally parabolic shape as a function of quality where the
237 greatest enhancement occurring at the low and high quality points. At a quality of 0.20, the heat transfer
238 coefficient is enhanced 48%, and at a quality of 0.81, it is enhanced by 79%. The lowest enhancement
239 occurs at a quality of 0.5 and corresponds to an enhancement of 20%. This parabolic enhancement
240 trend is best explained by the relative size of the mounds to the film as well as the disruption of film
241 flow which occurred due to them, as discussed in section 3.3. These heat transfer enhancements are
242 similar to those found in by Aroonrat and Wongwises (2019), using R134a, with hollow pin fins with
243 a maximum increase of 83% (Aroonrat and Wongwises, 2019), although this enhancement is smaller
244 than the 244% enhancement measured by Ho and Leong (2019) using R134a with conical fins lining
245 the interior of the channel (Ho et al., 2019). Ho and Leong (2020) investigated condensing steam on
246 vertical plates with an array of conical films, which resulted in an increase of 146%; however, this
247 differs from the present study as Ho and Leong (2020) studied gravity-driven film condensation (Ho
248 and Leong, 2020).
249   Flow condensation data for the inverted test sections, where condensation occurred on the upper
250 surface, had similar results to that of the standard test sections. For the plain inverted test section, at a
251 mass flux of 50 kg/m^{2}s, heat transfer coefficients ranged from 23,700 W/m^{2}K to 47,000 W/m^{2}K at
252 qualities from 0.20 to 0.87, which corresponded to an increase of 6.8% and a decrease of 10% in heat
253 transfer coefficients, respectively. At a mass flux of 125 kg/m^{2}s, the plain inverted test section had heat
254 transfer coefficients from 37,300 W/m^{2}K to 62,000 W/m^{2}K at qualities from 0.34 to 0.79. In this case
255 there was no heat transfer enhancement, with a decrease in performance of 9.5% at a quality of 0.34 to
256 a decrease of 14% at a quality of 0.79. The minimum decrease in heat transfer coefficient was 2.2%
257 and occurred at a quality of 0.51. For the inverted plain test section, all decreases in heat transfer
258 coefficient were less than 10%.
259   For the inverted mound test sections at a mass flux of 50 kg/m^{2}s, heat transfer coefficients ranged
260 from 37,300 W/m^{2}K to 74,800 W/m^{2}K at qualities of 0.23 and 0.90 respectively with corresponding
261 enhancements of 56% to 40% with a low of 37% occurring at a quality of 0.56. With a mass flux of
262 125 kg/m^{2}s, the heat transfer coefficients went from 42,300 W/m^{2}K to 110,000 W/m^{2}K at qualities
263 from 0.21 to 0.80. As with the standard orientation, the enhancement was parabolic with an
264 enhancement of 38% at quality of 0.21 and 51% at a quality of 0.80. The lowest enhancement was 27%
265 and occurred at a quality of 0.61.

266 **3.3. Flow visualization and condensate film disruption**

267   By using a high-speed camera, the flow condensation was observed at 500 frames per second. In
268   the plain coupon, condensate film flow was generally smooth for both mass fluxes observed. At the
269   lower mass flux and low qualities, the film had almost no disturbances apart from the occasional
270   "wave" across the entire micro-channel. As the quality or flow rate increased, the shear forces from
271   the vapor phase caused consisted small waves to form at the interface between the two phases. For the
272   mound test section, the mounds themselves served to disrupt the flow by preventing the film from
273   flowing in a straight line down the test section. Figure 9 shows the condensed film in both channels at
274   a quality of 0.6 for both mass flow rates. Film disruptions, particularly at the downstream side of the
275   mounds, were common and condensate was also observed to flow over the mounds as shown in Figure
276   10.
277   To show the significance of the mounds, a maximum film depth was calculated using the
278   Butterworth correlation to estimate the amount of condensed water available for the film (Butterworth,
279   1975). Figure 11 shows the maximum depth that the film could reach (i.e., estimated film thickness),

$$d_{film} = (1 - \alpha)d_{channel} \qquad\qquad 10$$

$$\alpha = \frac{1}{1 + 0.28X_{vv}^{0.71}} \qquad\qquad 11$$

$$X_{vv}^2 = \left(\frac{1 - x}{x}\right)\frac{\rho_g}{\rho_l}\frac{\mu_l}{\mu_g} \qquad\qquad 12$$

280   where d_{film} is the maximum possible film depth, $d_{channel}$ is the depth of the channel, α is the volumetric
281   fraction of vapor in the channel, X_{vv} is the Martinelli parameter for viscous-viscous flows, x is the
282   steam quality, ρ and μ are the density and viscosity of the liquid, l, and vapor, g, phases.
283   In the plain coupon, the thermal resistance is inversely related to the film depth; however,
284   relative to that film depth, the mounds' 1-mm radius proves a significant obstacle to the flow. For low
285   qualities where the film is thickest, $x < 0.4$, these disruptions in the film reduce the film thermal
286   resistance by forcing the flow to either pass over or around the mounds, increasing velocity, and
287   decreasing the relative thickness of film. For higher qualities, $x > 0.6$, the film is thin enough for the
288   mounds to have direct contact with the vapor, allowing them to act like fins, thereby increasing the
289   condensing surface area and increasing the planform heat transfer coefficient in addition to the film
290   disruptions. For all qualities both the film disruptions and fin effects occur; however, they are most
291   dominate for low and high quality respectively which is why for midrange qualities, $0.4 \leq x \leq 0.6$ the
292   lowest enhancement occurs. Based on the liquid film disruptions observed, it stands to reason that there
293   is some optimal grouping of mounds. If the mounds are too far apart, the opportunity for condensation
294   enhancement via film disruption and fin area is missed. However, if the fins are grouped too closely,
295   such as in a hexagonal close pack, the film may thicken due a smaller effective channel width and the
296   film could also stagnant at the bottom of the channel where the mounds meet, preventing convective
297   heat transfer from locally dominating over conductive heat transfer.
298   For both coupons, the inversion of the test section had little visual effect on the film, which
299   continued to flow in contact with the cooling surface despite that being the upper side of the flow
300   channel. Heat transfer performance was not significantly impacted by gravity and thus, two
301   nondimensional numbers were evaluated. The Eötvös number, Eo, which represents the ratio of
302   gravitational to surface tension forces, was calculated (Kim and Mudawar, 2012),

$$Eo = gD_h^2 \frac{(\rho_l - \rho_g)}{\sigma} \qquad \text{13}$$

303   where g is the gravity constant, D_h is the hydraulic diameter, and σ is the surface tension.
304   The Eotvos number was 0.64 and 0.60 for the plain and mound coupons, respectively. Eötvös numbers
305   less than one support that gravitation forces are not dominant in these coupons. The modified Weber
306   number, We^*, which represents the ratio of inertial to surface tension forces, was employed to determine
307   the expected flow regime for each data point (Kim and Mudawar, 2012),

$$We^* = \frac{2.45 Re_g^{0.64}}{Su_g^{0.3}(1 + 1.09 X_{tt}^{0.039})^{0.4}} \qquad \text{14}$$

$$Re_g = \frac{G(1-x)D_h}{\mu_l} \qquad \text{15}$$

$$Su_g = \frac{\rho_g \sigma D_h}{\mu_g^2} \qquad \text{16}$$

$$X_{tt} = \left(\frac{\mu_l}{\mu_g}\right)^{0.1} \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_g}{\rho_l}\right)^{0.5} \qquad \text{17}$$

308   where Re_g is the Reynolds number of the gas phase, Su_g is the Suratman number of the gas phase, G is
309   the mass flux of steam, and X_{tt} is the Lockhart-Martinelli parameter. These results can be seen in Figure
310   12 and shows the flow regimes expected by either slug flow, transitional (between slug and annular),
311   or wavy-annular flow; few data are in the slug flow regime. This supports the observed behavior for
312   the film to adhere to the copper surface regardless of orientation as for these flow regimes the film wet
313   the entire channel surface.

3.4. Pressure drops

315   Pressure drops were parabolic for both coupons and orientations and were increased with quality
316   and mass flow rate. The pressure drops in the mound coupon were also higher; however, changing the
317   orientation of the test section had minimal impacts, as shown in Figure 13. For the plain coupon with
318   a mass flux of 50 kg/m^{2}s, the pressure drops ranged from 1.8 kPa to 8.8 kPa for qualities from 0.20 to
319   0.87. At a mass flux of 125 kg/m^{2}s, the pressure drops ranged from 15 kPa to 92 kPa for qualities from
320   0.33 to 0.97. When the test section was inverted, at the mass flux of 50 kg/m^{2}s pressure drops went
321   from 1.8 kPa to 9.9 kPa for qualities from 0.20 to 0.87 and from 16 kPa to 67 kPa for qualities from
322   0.34 to 0.79. The standard mound coupon at 50 kg/m^{2}s had pressure drops from 1.2 kPa to 9.2 kPa for
323   qualities from 0.21 to 0.81. At a mass flux of 125 kg/m^{2}s pressure drops ranged from 10 kPa to 110
324   kPa for qualities from 0.20 to 0.81. For the inverted mound coupon at 50 kg/m^{2}s pressure drops ranged
325   from 3.1 kPa to18 kPa for qualities from 0.23 to 0.90 and at a mass flux of 125 kg/m^{2}s the pressure
326   drops were from 11 kPa to 120 kPa for qualities from 0.21 to 0.80. The increase in pressure drop was
327   significantly lower than that observed by Aroonrate and Wongwise (2019), using R134a, where the
328   presence of hollow pin fins increased the pressure drop by as 251%, 578%, and 892% depending on
329   the pin fin size of 0.5 mm, 0.75 mm, and 1.0 mm, respectively, with a mass flux of 300 kg/m^{2}s
330   (Aroonrat and Wongwises, 2019). The much lower pressure drop observed in the mound coupon in

331   this study is likely due to the lower frequency and positioning of mounds not constricting the flow
332   through channel as much as pin fins, as well as the lower mass fluxes of 50 kg/m^{2}s and 125 kg/m^{2}s.

333   **4. Conclusions**

334   Filmwise condensation experiments were conducted on two 1.9-mm-hydraulic-diameter copper
335   coupons. One coupon was a plain rectangular channel while the other was modified with a staggered
336   line of 2-mm-diameter copper hemispheres. Heat transfer measurements, film visualization, and
337   pressure drop data were recorded for steam condensation for two mass fluxes, 50 kg/m^{2}s and 125
338   kg/m^{2}s, and qualities from 0.20 to 0.97. From this, the following conclusions may be drawn:

- The addition of copper hemispheres increased the heat transfer coefficient by as much as 79%, and which exceeded the area enhancement ratio of 14%.
- The presence of the mounds disrupted the condensate film, preventing the film from flowing as a smooth laminar sheet.
- The height of the mounds was significant compared to the maximum film depth for all tested cases and provided and lowered the effect thermal resistance of the condensate film.
- Changing the orientation of the test section had little effect in the mound coupon and no more than a 14% reduction in heat transfer coefficient in the plain coupon.
- Pressure drops were higher in the mound coupon but were not significantly affected by the change in orientation.
- Future work should include a focus on the placement and frequency of mounds in the channel to enhance filmwise flow condensation heat transfer.

351   **Conflict of Interests**
352   The authors declare that the research was conducted in the absence of any commercial or financial
353   relationships that could be construed as a potential conflict of interest.

355   **Author Contributions**
356   GAR, ME, GH, and MMD contributed to the design of the study. GAR and CEM performed the
357   experiments and performed data analysis. GAR wrote the manuscript draft. MMD and GH revised the
358   manuscript.

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466 **Figure captions**

467 **Figure 1) Open-loop experimental apparatus for steam condensation heat transfer and**
468 **simultaneous flow visualization.**

469 **Figure 2) Test section, including coupons and heat flux block, where (above) the test section is in**
470 **the standard orientation with the cooled surfaces were the lower surfaces (i.e., gravity pulls liquid**
471 **towards the condensing surfaces) or (below) the test section was inverted such that the cooled**
472 **surfaces were the upper surfaces (i.e., gravity pulls liquid away from the condensing surfaces).**

473 **Figure 3) Coupons with 1.9-mm hydraulic diameters; (left) plain coupon with no additional**
474 **structures and (right) coupon with 2-mm-diameter mound structures.**

475 **Figure 4) The sintering temperature as a function of time with the peak temperature shown.**

476 **Figure 5) Single phase validation, including (left) energy balance and (right) measured Nusselt**
477 **numbers compared to predictions by the single-phase Muzychka (2004) correlation.**

478 **Figure 6) Flow condensation heat transfer coefficients, for (left) a mass flux of 50 kg/m²s and**
479 **(right) a mass flux of 125 kg/m²s.**

480 **Figure 7) Heat transfer coefficient enhancement for the plain coupon in the inverted orientation**
481 **and the mound coupon in the standard and inverted orientations compared to the plain coupon**
482 **in the standard orientation (left) for a mass flux of 50 kg/m²s and (right) a mass flux of 125**
483 **kg/m²s.**

484 **Figure 8) Flow condensation heat transfer coefficients predicted by the Kim and Mudawar (2013)**
485 **model for the plain coupon in standard orientation.**

486 **Figure 9) Condensed film in standard plain and mound coupons, (A) plain coupon with a mass**
487 **flux of 50 kg/m²s, (B) plain coupon with a mass flux of 125 kg/m²s, (C) mound coupon with a**
488 **mass flux of 50 kg/m²s, (D) mound coupon with a mass flux of 125 kg/m²s; quality is 0.6 for all**
489 **cases.**

490 **Figure 10) Mound disruption of film at (Left) t = 0 ms, (Middle) t = 2.0 ms, and (Right), t = 4.0**
491 **ms at a mass flux of 50 kg/m²s and a quality of 0.2.**

492   **Figure 11) Predicted maximum liquid film thickness using the Butterworth (1975) void fraction**
493   **model.**

494   **Figure 12) Flow regimes predicted by Kim and Mudawar (2012) for the experimental conditions;**
495   **most data are in the transition or wavy-annular regimes.**

496   **Figure 13) Pressure drops for plain and mound coupons in both orientations (left) for a mass flux**
497   **of 50 kg/m^{2}s and (right) a mass flux of 125 kg/m^{2}s.**